Contents

Preface xxi
Author xxv
List of Symbols xxvii

1. Introduction to Electric Motors 1
1.1 History of Electric Motors 1
1.2 Motor Design Characteristics 2
1.2.1 Motor Torque 2
1.2.1.1 Static and Dynamic Torque 2
1.2.1.2 Motor Torque in Motor-Load System 4
1.2.1.3 Continuous Torque 6
1.2.1.4 Peak Torque 6
1.2.1.5 Stall Torque 7
1.2.1.6 Cogging Torque and Reduction Methods 8
1.2.1.7 Torque Ripple 13
1.2.2 Motor Speed 14
1.2.2.1 Continuous Speed 15
1.2.2.2 Peak Speed 15
1.2.2.3 Speed Ripple 15
1.2.3 Torque Density 15
1.2.4 Motor Power and Power Factor 16
1.2.5 Torque–Speed Characteristics 17
1.2.6 Mechanical Resonance and Resonant Frequency 20
1.2.7 Load-to-Motor Inertia Ratio 25
1.2.8 Duty Cycle 26
1.2.9 Motor Efficiency 29
1.2.9.1 Definition of Motor Efficiency 29
1.2.9.2 IEC Standards on Efficiency Classes of AC Electric Motors 30
1.2.10 Motor Insulation 31
1.2.11 Motor Operation Reliability 36
1.3 Classifications of Electric Motors 36
1.3.1 DC and AC Motors 38
1.3.2 Single-Phase and Three-Phase Motors 39
1.3.3 Induction and Permanent Magnet Motors 40
1.3.4 Synchronous and Asynchronous Motors 44
1.3.5 Servo and Stepper Motors 45
1.3.6 Gear Drive and Direct Drive Motors 49
1.3.7 Brush and Brushless Motors 53
1.3.8 Reluctance Motors 54
1.3.9 Radial-Flux and Axial-Flux Motors 55
1.3.10 Rotary and Linear Motors 56
1.3.11 Open and Enclosed Motors 57
1.3.12 Motor Classification according to Power Rating 57

Hesai Exhibit 1031, page 8

1.4 Motor Design and Operation Parameters 58
1.4.1 Back EMF Constant, K_e 58
1.4.2 Torque Constant, K_t 59
1.4.3 Velocity Constant, K_v 60
1.4.4 Motor Constant, K_m 60
1.4.5 Mechanical Time Constant, τ_m 61
1.4.6 Electrical Time Constant, τ_e 63
1.4.7 Thermal Time Constant, τ_{th} 64
1.4.8 Viscous Damping, K_{vd} 65
1.5 Sizing Equations 66
1.6 Motor Design Process and Considerations 67
1.6.1 Design Process 68
1.6.2 Design Integration 73
1.6.3 Mechatronics 74
1.7 Motor Failure Modes 74
1.8 IP Code 76
References 78

2. Rotor Design 83
2.1 Rotor in Induction Motor 83
2.1.1 Wound Rotor 83
2.1.2 Squirrel Cage Rotor 85
2.1.2.1 Factors Impacting Resistance of Squirrel Cage Rotor 86
2.1.2.2 Double-Cage Rotor 87
2.1.2.3 Casting Squirrel Cage Rotor 88
2.1.2.4 Skin Effect 91
2.1.3 Induction Motor Design Types and Their Performing Characteristics 91
2.2 Permanent Magnet Rotor 92
2.2.1 Discovery of Phenomenon of Magnetism 92
2.2.2 Permanent Magnet Characteristics 93
2.2.3 Permanent Magnet Materials 95
2.2.3.1 Ferrite Permanent Magnets 96
2.2.3.2 Rare-Earth Permanent Magnets 96
2.2.3.3 New Developments of Permanent Magnets 97
2.2.4 Magnetization 98
2.2.5 Factors Causing Demagnetization 99
2.2.6 Maximum Operation Temperature 102
2.2.7 Permanent Magnet Mounting and Retention Methods 102
2.2.8 Corrosion Protection of Permanent Magnets 107
2.3 Rotor Manufacturing Process 109
2.3.1 Lamination Materials 109
2.3.2 Lamination Cutting 111
2.3.3 Lamination Surface Insulation 112
2.3.4 Lamination Annealing 113
2.3.5 Lamination Stacking 113
2.3.6 Rotor Casting for Squirrel Cage Motor 113
2.3.7 Heat Treatment of Casted Rotor 114
2.3.8 Rotor Assembly 115

2.3.9 Rotor Machining and Runout Measurement 115
2.3.10 Rotor Balancing 115
2.3.10.1 Types of Unbalance 118
2.3.10.2 Rotor Balancing Machine 120
2.3.10.3 Balancing Operation 121
2.4 Interference Fit 122
2.4.1 Press Fit 124
2.4.2 Shrink Fit 128
2.4.3 Serration Fit 131
2.4.4 Fitting with Adjustable Ringfeder® Locking Devices 132
2.4.5 Fitting with Tolerance Rings 132
2.5 Stress Analysis of Rotor 135
2.6 Rotordynamic Analysis 137
2.6.1 Rotor Inertia 137
2.6.2 Motor Critical Speed and Resonance 139
2.7 Rotor Burst Containment Analysis 143
2.7.1 Rotor Burst Speed 143
2.7.2 Energy in Rotating Rotor 145
2.7.2.1 Kinetic Energy in Rotor 145
2.7.2.2 Elastic Potential Energy in Rotor 146
2.7.2.3 Ratio of Potential Energy to Kinetic Energy of Rotor 147
2.7.3 Rotor Burst Containment Design 147
References 150

3. Shaft Design 155
3.1 Shaft Materials 155
3.2 Shaft Loads 159
3.3 Shaft Design Methods 160
3.3.1 Macaulay's Method 160
3.3.2 Area–Moment Method 161
3.3.3 Castigliano's Method 162
3.3.4 Graphical Method 163
3.4 Engineering Calculations 163
3.4.1 Normal Stress for Shaft Subjected to Axial Force 164
3.4.2 Bending Stress for Shaft Subjected to Bending Moment 165
3.4.3 Torsional Shear Stress and Torsional Deflection 168
3.4.4 Lateral Deflection of Shaft 172
3.4.4.1 Lateral Deflection due to Bending Moment 173
3.4.4.2 Lateral Deflection due to Transverse Force 174
3.4.4.3 Lateral Deflection due to Shear Force 176
3.5 Shaft Design Issues 176
3.5.1 Shaft Design Considerations 176
3.5.2 Shaft Rigidity 178
3.5.3 Critical Shaft Speed 178
3.5.3.1 Shaft with Uniform Diameter 179
3.5.3.2 Stepped Shaft 179
3.5.4 Dimensional Tolerance 179
3.5.5 Dynamic Shaft Runout 180
3.5.6 Shaft Eccentricity 180

3.5.7 Heat Treatment and Shaft Hardness ... 180
3.5.8 Shaft Surface Finishing ... 182
3.5.9 Shaft Lead ... 184
3.5.10 Shaft Seal ... 186
3.5.10.1 O-Ring Seal ... 186
3.5.10.2 Universal Lip Seal ... 188
3.5.10.3 V-Shaped Spring Seal ... 189
3.5.10.4 Brush Seal ... 189
3.5.10.5 PTFE Seal ... 191
3.5.10.6 Noncontact Seal ... 191
3.5.11 Diametrical Fit Types ... 193
3.6 Stress Concentration ... 193
3.7 Torque Transmission through Mechanical Joints ... 194
3.7.1 Keyed Shafts ... 194
3.7.1.1 Selection of Key Material ... 194
3.7.1.2 Stress Analysis of Key and Keyseat ... 195
3.7.1.3 Key Fitting ... 198
3.7.1.4 Stress Concentration Factors of Keyed Shafts ... 199
3.7.2 Spline Shafts ... 200
3.7.2.1 Advantages of Spline Shafts ... 201
3.7.2.2 Spline Types ... 202
3.7.2.3 Stress Concentration Factors of Spline Shafts ... 203
3.7.3 Tapered Shafts ... 203
3.8 Fatigue Failure under Alternative Loading ... 205
3.9 Shaft Manufacturing Methods ... 206
3.9.1 Machined Shaft ... 206
3.9.2 Forged Shaft ... 207
3.9.3 Welded Hollow Shaft ... 207
3.9.4 Shaft Measurement ... 210
3.10 Shaft Misalignment between Motor and Driven Machine ... 210
3.10.1 Types of Misalignment ... 211
3.10.2 Correction of Shaft Misalignment ... 212
3.11 Shaft Coupling ... 213
3.11.1 Rigid and Semirigid Couplings ... 213
3.11.2 Flexible Couplings ... 216
References ... 216

4. Stator Design ... 219
4.1 Stator Lamination ... 219
4.1.1 State Lamination Material ... 219
4.1.2 Stator Lamination Patterns ... 219
4.1.2.1 One-Piece Lamination ... 219
4.1.2.2 T-Shaped Segmented Lamination ... 220
4.1.2.3 Connected Segmented Lamination ... 224
4.1.2.4 Two-Section Stator Lamination ... 224
4.1.2.5 Stator Lamination Integrated by Individual Teeth and a Yoke Section ... 225
4.1.2.6 Slotless Stator Core ... 226
4.1.2.7 Slinky Lamination Stator Core ... 227

4.2 Magnet Wire 230
4.2.1 Regular Magnet Wire 230
4.2.2 Self-Adhesive Magnet Wire 230
4.2.3 Litz Wire 231
4.3 Stator Insulation 232
4.3.1 Injection Molded Plastic Insulation 233
4.3.2 Slot Liner 233
4.3.3 Glass Fiber–Reinforced Mica Tape 234
4.3.4 Powder Coating on Stator Core 235
4.4 Manufacturing Process of a Stator Core 236
4.4.1 Stator Lamination Cutting 237
4.4.2 Lamination Fabrication Process 238
4.4.3 Lamination Annealing 239
4.4.4 Lamination Stacking 240
4.4.5 Stator Winding 242
4.4.5.1 Random Winding by Hand 243
4.4.5.2 Coil Formation: Distributed Winding 243
4.4.5.3 Coil Formation: Concentrated Winding 243
4.4.5.4 Coil Formation: Conductor Bar 243
4.5 Stator Encapsulation and Impregnation 245
4.5.1 Encapsulation 245
4.5.1.1 Encapsulation Materials 245
4.5.1.2 Encapsulation Process 248
4.5.2 Varnish Dipping 249
4.5.3 Trickle Impregnation 250
4.5.4 Vacuum Pressure Impregnation 250
4.6 Stator Design Considerations 253
4.6.1 Cogging Torque 253
4.6.2 Air Gap 253
4.6.3 Stator Cooling 254
4.6.4 Robust Design of Stator 254
4.7 Mechanical Stress of Stator 254
References 256

5. Motor Frame Design 259
5.1 Types of Motor Housing 259
5.1.1 Wrapped Housing 259
5.1.2 Casted Housing 260
5.1.2.1 Casting Materials 261
5.1.2.2 Casting Process 264
5.1.2.3 Pressure Casting 268
5.1.2.4 Heat Treatment 268
5.1.3 Machined Housing 269
5.1.4 Stamped Housing 269
5.1.5 Extruded Aluminum Motor Housing 269
5.1.6 Motor Housing with Composite Materials 270
5.1.7 Motor Housing Fabricated by 3D Printing Technology 270
5.1.8 Frameless Motor 271
5.2 Testing Methods of Casted Motor Housing 273

5.3 Endbell Manufacturing 276
5.3.1 Casted Endbell 277
5.3.2 Stamped Endbell 277
5.3.3 Iron Casting versus Aluminum Casting 278
5.3.4 Machined Endbell 279
5.3.5 Forged Endbell 279
5.4 Motor Assembly Methods 279
5.4.1 Tie Bar 279
5.4.2 Tapping at Housing End Surface 279
5.4.3 Forged Z-Shaped Fastener 280
5.4.4 Rotary Fasteners 282
5.4.4.1 Triangle-Base Rotating Fastener 283
5.4.4.2 Square-Base Rotating Fastener 285
5.4.4.3 Butterfly-Base Rotating Fastener 286
5.4.5 Other Types of Fasteners 286
5.4.5.1 Cylinder-Base Fastener Locked with Retaining Ring 287
5.4.5.2 Cylinder-Base Fastener with Self-Locking Aperture 287
5.4.5.3 Fastener Engaged with Housing from Housing Interior 289
5.5 Fastening System Design 290
5.5.1 Types of Thread Fasteners 290
5.5.2 Thread Formation 290
5.5.3 Fastener Preload 290
5.5.4 Fastener-Tightening Process 291
5.5.5 Tightening Torque 294
5.5.6 Thread Engagement and Load Distribution 297
5.6 Anticorrosion of Electric Motor and Components 298
5.6.1 Surface Treatment Methods 298
5.6.1.1 Electroplating 299
5.6.1.2 Electroless Plating 299
5.6.1.3 Physical Vapor Deposition 299
5.6.1.4 Inorganic Coating 300
5.6.1.5 Phosphate Coating 300
5.6.1.6 Electropolishing 300
5.6.2 Anticorrosion Treatment of Electric Motor 301
5.6.3 Hydrogen Embrittlement Issues 302
References 305

6. Motor Bearing 309
6.1 Bearing Classification 309
6.1.1 Journal Bearing 309
6.1.2 Rolling Bearing 313
6.1.2.1 Ball Bearing 313
6.1.2.2 Roller Bearing 319
6.1.3 Noncontact Bearing 320
6.1.4 Sensor Bearing 322
6.2 Bearing Design 323
6.2.1 Bearing Materials 323

6.2.2 Bearing Internal Clearances 324
6.2.3 Allowable Bearing Speed 328
6.2.4 Bearing Fitness 328
6.2.5 Prevention of Bearing Axial Movement 329
6.2.6 Bearing Load 331
6.2.6.1 Bearing Preload Arrangement 331
6.2.6.2 Radial and Axial Bearing Load 332
6.2.6.3 Load Distribution 332
6.3 Bearing Fatigue Life 335
6.3.1 Calculation of Bearing Fatigue Life 335
6.3.2 Bearing Failure Probability Distribution 336
6.3.3 Influence of Unbalance on Bearing Fatigue Life 338
6.3.4 Influence of Wear on Bearing Fatigue Life 341
6.3.5 Influence of Internal Radial Clearance on Bearing Fatigue Life 341
6.4 Bearing Failure Mode 343
6.4.1 Major Causes of Premature Bearing Failure 344
6.4.2 Lubricant Selection 344
6.4.3 Improper Bearing Lubrication 347
6.4.4 Lubricant Contamination 347
6.4.5 Grease Leakage 348
6.4.6 Bearing Sealing 349
6.4.7 Excessive Load 350
6.4.8 Internal Radial Interference Condition 351
6.4.9 Bearing Current 354
6.4.10 Impact of High Temperature on Bearing Failure 355
6.4.11 Bearing Failure Associated with Motor Vibration and Overloading 356
6.4.12 Improper Bearing Installation and Bearing Misalignment 357
6.4.13 Vertically Mounted Motor 358
6.5 Bearing Noise 359
6.6 Bearing Selection 360
6.6.1 Bearing Type Selection Based on Load 362
6.6.2 Bearing Type Selection Based on Speed 363
6.6.3 Selection of Bearing Size 363
6.7 Bearing Performance Improvement 364
References 365

7. Motor Power Losses 369
7.1 Power Losses in Windings due to Electric Resistance in Copper Wires 370
7.2 Eddy-Current and Magnetic Hysteresis Losses 373
7.2.1 Eddy-Current Loss 373
7.2.2 Magnetic Hysteresis Loss 374
7.2.3 Calculations of Eddy-Current and Magnetic Hysteresis Losses 374
7.2.4 Losses in Stator and Rotor Iron Cores 378
7.2.5 Losses in PMs 379
7.2.6 Power Losses in Other Core Components 381
7.3 Mechanical Friction Losses 381
7.3.1 Bearing Losses 381

7.3.2 Sealing Losses 385
7.3.3 Brush Losses 385
7.4 Windage Losses 386
7.4.1 Windage Loss due to Rotating Rotor 388
7.4.1.1 Taylor Vortex 388
7.4.1.2 Friction Factor 391
7.4.1.3 Prediction of Windage Loss due to Rotating Rotor 392
7.4.2 Windage Loss due to Entrance Effect of Axial Air-Gap Flow 393
7.4.3 Windage Loss due to Stator Surface Roughness 393
7.4.4 Fan Losses 395
7.4.5 Ventilating Path Losses 399
7.4.6 Methods for Reducing Windage Losses 399
7.5 Stray Load Losses 404
References 405

8. Motor Cooling 409
8.1 Introduction 409
8.1.1 Passive and Active Cooling Techniques 410
8.1.2 Heat Transfer Enhancement Techniques 411
8.2 Conductive Heat Transfer Techniques 412
8.2.1 Conductive Heat Flux and Energy Equations 412
8.2.2 Encapsulation and Impregnation of Electric Motor 414
8.2.3 Enhanced Heat Transfer Using High-Thermal-Conductivity Material 415
8.2.4 Using Self-Adhesive Magnet Wire for Fabricating Stator Winding 417
8.3 Natural Convection Cooling with Fins 417
8.3.1 Cooling Fin 418
8.3.2 Fin Optimization 418
8.3.3 Applications of Various Fins in Motor Cooling 421
8.3.4 Pin-Fin Heat Sink 425
8.3.5 Thermal Interface Materials 429
8.4 Forced Air Cooling Techniques 432
8.4.1 Thermophysical Properties of Air 432
8.4.2 Direct Forced Air Cooling Techniques 433
8.4.2.1 Forced Air Cooling at End-Winding Regions 434
8.4.2.2 Forced Air Flowing through Internal Cooling Channels across the Motor 434
8.4.2.3 Forced Air Flowing over Motor Outer Surfaces 438
8.4.2.4 Forced Air Flowing through Both Motor Outer and Inner Surfaces 439
8.4.2.5 Air Jet Impingement Cooling 440
8.4.3 Indirect Forced Air Cooling Techniques 441
8.4.3.1 Indirect Forced Air Cooling with Heat Exchangers 442
8.4.3.2 Indirect Forced Air Cooling via Indirect Evaporative Air Cooler 443
8.4.4 Fan and Blower 444
8.4.4.1 Fan Types 444
8.4.4.2 Forward-Curved, Backward-Curved, and Straight Blades of Centrifugal Fans 446

8.4.4.3 Fan Performance Curve and Operation Point 447
8.4.4.4 Fan Selection 448
8.5 Liquid Cooling Techniques 448
8.5.1 Thermophysical Properties of Coolants 449
8.5.2 Direct Liquid Cooling Techniques 450
8.5.2.1 Direct Liquid Cooling of Bundled Magnet Wires 451
8.5.2.2 Spray Cooling 452
8.5.2.3 Liquid Jet Impingement Cooling 456
8.5.3 Liquid Immersion Cooling 457
8.5.4 Indirect Liquid Cooling Techniques 458
8.5.4.1 Indirect Liquid Cooling via Cold Plates Attached to Motor Walls 458
8.5.4.2 Indirect Liquid Cooling via Helical Copper Pipes Casted in Motor Housing 458
8.5.4.3 Indirect Liquid Cooling with Cooling Channels on Casted Motor Housing 462
8.5.4.4 Indirect Liquid Cooling via Copper Pipe in Spacer 462
8.5.4.5 Indirect Liquid Cooling through Stator Winding Slots 463
8.5.4.6 Indirect Liquid Cooling through Microscale Channels 463
8.6 Phase-Change Cooling Techniques 465
8.6.1 Cooling with Heat Pipes 465
8.6.2 Cooling with Vapor Chambers 467
8.6.3 Evaporative Cooling 468
8.6.4 Mist Cooling 471
8.7 Radiative Heat Transfer 471
8.8 Other Advanced State-of-the-Art Cooling Methods 472
8.8.1 Microchannel Cooling Systems 472
8.8.2 Metal Foams 473
8.8.3 Heat Transfer Enhancement with Nanotechnology 474
8.8.3.1 Nanofluid 475
8.8.3.2 Carbon Nanotube 475
8.8.3.3 NanoSpreader™ Vapor Cooler 475
8.8.3.4 CarbAl™ Heat Transfer Material 476
8.8.3.5 Ionic Wind Generator 477
References 477

9. Motor Vibration and Acoustic Noise 481
9.1 Vibration and Noise in Electric Motor 481
9.2 Fundamentals of Vibration 483
9.2.1 Simple Harmonic Oscillating System 483
9.2.2 Damped Harmonic Oscillating System 485
9.2.3 Forced Vibration with Damping 487
9.2.4 Forced Vibration due to Mass Unbalance 492
9.2.5 Vibration Induced by Support Excitation 495
9.2.6 Directional Vibration 497
9.3 Electromagnetic Vibrations 498
9.3.1 Unbalanced Forces/Torques Caused by Electric Supply 498
9.3.2 Broken Rotor Bar and Cracked End Ring 499
9.3.3 Unbalanced Magnetic Pull due to Asymmetric Air Gap 500

9.3.3.1 Electromagnetic Force at Air Gap 500
9.3.3.2 Asymmetric Air Gap due to Nonconcentric Rotor and Stator 502
9.3.3.3 Asymmetric Air Gap due to Elliptic Stator 504
9.3.3.4 Asymmetric Air Gap due to Elliptic Rotor 505
9.3.3.5 Asymmetric Air Gap due to Rotor Misalignment 508
9.3.3.6 Asymmetric Air Gap Resulting from Shaft Deflection 508
9.3.4 Nonuniform Air Gap due to Stator Slots 508
9.3.5 Mutual Action Forces between Currents of Stator and Rotor 509
9.3.6 Vibration due to Unbalanced Voltage Operation 510
9.4 Mechanical Vibrations 510
9.4.1 Misaligned Shaft and Distorted Coupling 510
9.4.2 Defective Bearing 510
9.4.3 Self-Excited Vibration 511
9.4.4 Torsional Vibration 511
9.5 Vibration Measurements 512
9.6 Vibration Control 514
9.6.1 Damping Materials 514
9.6.2 Vibration Isolation 515
9.6.3 Tuned Mass Damper 518
9.6.4 Double Mounting Isolation System 521
9.6.5 Viscoelastic Bearing Support 522
9.6.6 Active Vibration Isolation and Damping 522
9.6.7 Measurements of Motor Vibration 525
9.7 Fundamentals of Acoustic Noise 526
9.7.1 Tonal Noise and Broadband Noise 527
9.7.2 Sound Pressure Level and Sound Power Level 527
9.7.3 Octave Frequency Bands 528
9.7.4 Three Sound Weighting Scales 530
9.7.5 Averaged Sound Pressure Level 531
9.7.6 Types of Noise 531
9.7.6.1 Structure-Borne Noise 532
9.7.6.2 Airborne Noise 532
9.7.6.3 Aerodynamic Noise 532
9.8 Noise Types and Measurements for Rotating Electric Machines 532
9.8.1 Noise Types in Rotating Electric Machines 533
9.8.1.1 Mechanical Noise 533
9.8.1.2 Electromagnetic Noise 534
9.8.1.3 Aerodynamic Noise 535
9.8.2 Acoustic Anechoic Chamber 535
9.8.3 Measurement of Motor Noise 536
9.8.4 Acoustic Noise Field Measurement 537
9.9 Motor Noise Abatement Techniques 537
9.9.1 Active Noise Reduction Techniques 538
9.9.2 Passive Noise Reduction Techniques 539
9.9.2.1 Blocking Noise Propagation Paths and Isolating Noise Sources 539
9.9.2.2 Using Noise-Absorbing Materials 540
9.9.2.3 Motor Suspension Mounting 543

9.9.2.4 Noise-Attenuating Structure 543
9.9.2.5 Smoothing Ventilation Paths 546
9.9.2.6 Selecting Low Noise Bearings 547
9.9.3 Innovative Noise Abatement Methods 548
References 548

10. Motor Testing 553
10.1 Motor Testing Standards 553
10.2 Testing Equipment and Measuring Instruments 554
10.2.1 Dynamometer 554
10.2.2 Thermocouples and Other Temperature Measuring Devices 559
10.2.3 Control System 560
10.2.4 Data Acquisition System 561
10.2.5 Torque Transducer 561
10.2.6 Power Supply 562
10.2.7 Motor Testbed 562
10.3 Testing Load Level 563
10.4 Testing Methods 564
10.4.1 Mechanical Differential Testing Method 564
10.4.2 Back-to-Back Testing Method 564
10.4.3 Indirect Loading Testing Method 564
10.4.4 Forward Short-Circuit Testing Method 567
10.4.5 Variable Inertia Testing Method 568
10.5 Off-Line Motor Testing 569
10.5.1 Winding Electrical Resistance Testing 570
10.5.2 Megohm Testing 570
10.5.3 Polarization Index Testing 571
10.5.4 High-Potential Testing 572
10.5.5 Surge Testing 572
10.5.6 Step-Voltage Testing 573
10.5.7 Determination of Rotor's Moment of Inertia 573
10.6 Online Motor Testing 576
10.6.1 Locked Rotor Testing 576
10.6.2 Motor Heat Run Testing 577
10.6.3 Motor Efficiency Testing 578
10.6.4 Impulse Testing 580
10.6.5 Cogging Torque Testing 580
10.6.6 Torque Ripple Measurement 583
References 585

11. Modeling, Simulation, and Analysis of Electric Motors 589
11.1 Computational Fluid Dynamics and Numerical Heat Transfer 589
11.1.1 Strategies in Modeling and Performing CFD Analysis 590
11.1.2 Rotating Flow Modeling 590
11.1.3 Porous Media Modeling 592
11.1.4 Numerical Simulation of Motor Cooling 595
11.1.4.1 Mathematical Formulations 596
11.1.4.2 Numerical Method 599
11.1.4.3 CFD Results 601

11.2 Thermal Simulation with Lumped-Circuit Modeling 607
11.3 Thermal Analysis Using Finite Element Method 610
11.4 Rotordynamic Analysis 614
11.4.1 Problem Description 614
11.4.2 Bearing Support's Stiffness and Damping 615
11.4.3 Rotordynamic Modeling 617
11.4.4 Results of Rotordynamic Analysis 618
11.5 Static and Dynamic Stress/Strain Analysis 621
11.5.1 Static Analysis 622
11.5.2 Dynamic Analysis 623
11.5.2.1 Centrifugal Force–Induced Stress on PMs 623
11.5.2.2 Structural Analysis Using Finite Element Method 625
11.5.3 Shock Load 626
11.6 Fatigue Analysis 627
11.7 Torsional Resonance Analysis 627
11.8 Motor Noise Prediction 629
11.9 Buckling Analysis 630
11.10 Thermally Induced Stress Analysis 632
11.11 Thermal Expansion and Contraction Analysis 632
References 635

12. Innovative and Advanced Motor Design 637
12.1 High-Temperature Superconducting Motor 637
12.2 Radial-Flux Multirotor or Multistator Motor 640
12.2.1 Radial-Flux Multirotor Motor 640
12.2.2 Radial-Flux Multistator Motor 642
12.2.3 Radial-Flux Brushless Double-Rotor Machine 643
12.2.4 Radial-Flux Double-Stator PM Machine 645
12.3 Axial-Flux Multirotor or Multistator Motor 646
12.3.1 Single-Sided and Double-Sided Axial-Flux Motors 646
12.3.2 Multistage Axial-Flux Motor 649
12.3.3 Yokeless and Segmented Armature Motors 650
12.4 Hybrid Motor 651
12.4.1 Hybrid Excitation Synchronous Machine 651
12.4.2 Hybrid Hysteresis PM Synchronous Motor 651
12.4.3 Hybrid Motor Integrating Radial-Flux Motor and Axial-Flux Motor 652
12.4.4 Hybrid-Field Flux-Controllable PM Motor 654
12.4.5 Hybrid Linear Motor 654
12.5 Conical Rotor Motor 655
12.6 Transverse Flux Motor 656
12.7 Reconfigurable Permanent Magnet Motor 660
12.8 Variable Reluctance Motor 661
12.9 Permanent Magnet Memory Motor 664
12.9.1 Variable Flux PM Memory Motor 664
12.9.2 Pole-Changing PM Memory Motor 664
12.9.3 Doubly Salient Memory Motor 665
12.10 Adjustable and Controllable Axial Rotor/Stator Alignment Motor 667
12.11 Piezoelectric Motor 669

12.12 Advanced Electric Machines for Renewable Energy 671
12.13 Micromotor and Nanomotor 672
12.13.1 Micromotor 672
12.13.2 Nanomotor 675
References 676

Index 679

Preface

Electric motors are extensively used in industrial, commercial, and military applications such as automobiles, elevators, electronic devices, robots, appliances, medical equipment, energy-conversion systems, machine tools, aircraft carriers, and satellites. Along with the rapid growth in energy demand all over the world in recent years, it has become a challenge for the motor industry to design and manufacture high-efficiency, high-reliability, low-cost, and quiet electric motors with superior performance.

Kollmorgen Corporation, a subsidiary of Danaher Corporation, is the global leader in the design, development, manufacture, and service of innovative and reliable products in the motion control industry. As the chief engineer of Kollmorgen, I have long been eagerly awaiting a modern book on the mechanical design of electric motors. However, most available books on electric motors in the market focus primarily on electromagnetic design rather than mechanical design. After a long unsuccessful search, one day I was suddenly struck with a thought: Why not write this book myself? This thought has inspired me to embark on the arduous task of compiling recent developments and advances in the mechanical design of electric motors into one book.

Despite significant advances in the last decades, the design, development, and implementation of modern electric motors still pose challenges for motor engineers and designers. This is because they involve a wide variety of disciplines, such as mechanics, electromagnetics, electronics, fluid and solid dynamics, heat transfer, material science, tribology, acoustics, control theory, manufacturing technology, and engineering economics, to name a few. In order to meet customer-specific requirements, today's motor manufacturers must effectively provide customized premium efficient products and services for accommodating multiple markets.

The ultimate goal of this book is to provide readers an in-depth knowledge base of designs, techniques, and developments of modern electric motors. The text is suitable for engineers, designers, and manufacturers in the motor industry, as well as maintenance personnel, graduate and undergraduate students, and academic researchers. This book has addressed in detail many aspects of motor design and performing characteristics, including motor classification, design of motor components, material selections, power losses, cooling, design integration, vibration, and acoustic noise. It also covers motor modeling, simulation, and engineering analysis. To reflect state-of-the-art electric motors, innovative and advanced motors developed in recent decades are reviewed in the book.

The book consists of a total of 12 chapters that cover electric motor fundamentals, practical design, engineering analysis, model setup, material and bearing selection, manufacturing processes, testing methods, and other design issues/practical considerations of electric motors. Chapter 1 presents the overview of the basic knowledge and fundamental principles of electric motors. Attempts have been made to introduce to readers motor operating characteristics, design parameters, manufacturing processes, and design methodologies.

Chapters 2 through 5 sequentially address design details of the most important components or subsystems of electric motors, including rotor, shaft, stator, and motor frame. In each of these chapters, design and manufacturing activities are addressed in detail to

Hesai Exhibit 1031, page 22

cover material selection, manufacturing process, assembly method, engineering analysis, etc. The potential applications of some newly developed manufacturing technologies on electric motors are also discussed.

Statistical data have revealed that the majority of electric motor failures are attributed to premature bearing failures. Thus, it is critical to understand bearing operating characteristics, determine bearing failure modes, identify bearing failure root causes, and evaluate the effects of some factors (e.g., temperature, lubrication condition, contamination, bearing current, excessive loading, misalignment, etc.) on bearing life. All these are addressed in Chapter 6.

When a motor operates under load conditions, a certain portion of the input power does not convert into the mechanical power to do useful work. Most of the wasted energy is converted into heat, which must be effectively dissipated into the environment. This results in a decrease in motor efficiency. As a matter of fact, motor efficiency varies widely with the technology used. In order to improve motor efficiency, it is highly desirable to minimize these power losses. The analysis and calculation of each type of power loss are discussed in Chapter 7.

Motor cooling plays a major role in ensuring optimum performance, operating reliability, and cost-effectiveness of electric motors. The adoption of high-efficiency cooling techniques can significantly increase in the power density and overall motor efficiency. A variety of cooling techniques, including air cooling, liquid cooling, mist cooling, and phase change cooling, are reviewed in Chapter 8. The selection of cooling techniques depends on many factors, such as cooling load, cooling effectiveness, thermophysical properties of coolant, complexity of cooling system, type of motor application, and others. State-of-the-art cooling techniques that have been successfully used in electronics cooling are also reviewed, and these have been suggested for application in motor cooling.

Chapter 9 describes vibration and noise issues of electric motors. Motor vibration can not only cause high noise and high fatigue stress of motor, but also deteriorate the performance of motor and, in the worst-case scenario, lead to premature motor failure. High acoustic noise is prohibited by many legislative regulations. For some special motor applications, acoustic sound pressure levels are restricted by law.

Computer technology has become a key component in motor development and design processes over recent decades. Computer-aided design (CAD), computer-aided manufacturing (CAM), and computer-aided engineering (CAE) techniques today are widespread in their application to motor design. However, no matter how perfect the engineering design and how precise the theoretical calculations, motor testing is a necessary step to validate the conceptual design, ensure normal and safe operation of the motor, and further optimize the motor design. Chapter 10 presents the motor testing methods and procedures for both component and system levels. The important testing equipment is also addressed.

A variety of engineering modeling, simulation, and analysis that are frequently encountered during motor design processes are addressed in Chapter 11. The successful integration of the engineering analysis into the product design cycle can significantly enhance the design quality, improve design standards, and accelerate the design process. In this chapter, some important engineering analyses are discussed in detail, including CFD analysis, rotordynamic analysis, and stress/strain analysis.

With the rapid strides made in computer technology, material science, manufacturing process, control technology, and other fields, the motor industry has developed immensely in recent years. In fact, sustainable growth of the motor industry requires innovation. In the age of globalization, motor manufacturers tend to aim at high-efficiency, high-precision,

durable, and energy-saving products. Chapter 12 reviews and discusses some innovative and advanced motors.

Electric motors and generators are commonly referred to as electric machines. While electric motors convert electrical energy to mechanical energy, electric generators convert mechanical energy into electrical energy. Due to the similarities between electric motors and generators in many aspects such as mechanical construction, electromagnetic operating principles, power losses, cooling methods, etc., some contents in this book can also be applied in generator design and serve as a reference for generator design engineers. Similarly, a few generator design practices covered in this book can also benefit motor design engineers.

"The details determine success or failure"—this is true not only for industrial production but also for many aspects of human society. As a consequence, engineers and designers should pay special attention to details in designing and manufacturing electric motors.

I am greatly indebted to the excellent reviewers of this book. Their valuable and pertinent comments have greatly enhanced the quality of the book. The experts who have kindly read different parts of this book include the following:

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From November 2010 to May 2013, I was sent to Shanghai, China, for building up an engineering team. I wish to acknowledge all the engineers in the team for offering me some useful information. While providing training to these engineers, I have also learned a lot from them.

I am especially grateful to Jonathan Plant, Jennifer Ahringer, and Jennifer Stair of CRC Press for their valuable advice and guidance. Vijay Bose as the project manager at SPi Global has done an excellent job in editing and proofreading the manuscript with splendid efficiency.

I deeply miss my parents. Their imperceptible influence in my growing years helped me set my own life goals. At the most difficult time of my life (during a special historical period), they encouraged me to look forward and keep my faith and hope for the future. My biggest regret is that they cannot see this published book.

Finally, I wish to express my sincere gratitude to my lovely wife, Zhangqing Zhuo, and daughter, Winnie Tong, for their love and patience. Without their continuous support and encouragement, this book could have only been published in my dreams.

I welcome and greatly appreciate feedback, corrections, and suggestions from the readers for improving the technical contents. The readers can directly contact me via e-mail at motor.mechanical.design@gmail.com.

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Hesai Exhibit 1031, page 25

Author

Dr. Wei Tong's career started as a mechanical technician in the early of 1970s. He currently serves as Chief Engineer at Kollmorgen Corporation, a subsidiary of Danaher Corporation, and is responsible for the development, design, analysis, and implementation of electric motors. He has made significant contributions in developing innovations in motor cooling, structure, and manufacturing.

Prior to joining Kollmorgen, Dr. Tong worked at GE Energy at Schenectady, New York, where he conducted designs of high-efficiency, high-powered generators and steam turbines. He was the project leader of a number of design and R&D projects, such as the advanced cooling system, hydrogen sealing system, noise attenuation system, dynamic thrust control system, steam turbine axial clearance design, and thermodynamic operation design. All these projects had been led to great technical and commercial success. Prior to GE, he worked at Babcock & Wilcox carrying out design and analysis for the superconducting magnetic energy storage system, solar-powered satellite, and medical isotope production reactor.

Dr. Tong received his PhD from the University of Minnesota in 1989 and MBA from State University of New York at Albany in 1999. He has served as an Associate Editor of the *ASME Journal of Heat Transfer* for two terms (2009–2012 and 2012–2015) and of the *International Journal of Rotating Machinery* (2008–).

Dr. Tong currently holds 28 US patents and 16 foreign patents. He has been a licensed Professional Engineer in Virginia since 1996. He is an internationally recognized expert on mechanical–electrical–thermal systems.

As a fellow (2006–) of the American Society of Mechanical Engineering (ASME), Dr. Tong has served on several ASME committees, including Heat Transfer under Extreme Conditions (K-18), Heat Transfer in Energy Systems (K-6), Aerospace Heat Transfer (K-12), and Long Range Directions and Issues in Heat Transfer (K-2). He was the chair of the K-18 Committee during 2006–2009.

In 2006, Dr. Tong was appointed as an adjunct professor in the Department of Mechanical Engineering, Virginia Polytechnic Institute and State University, where he taught an undergraduate course on mechanical engineering design.

Hesai Exhibit 1031, page 26

List of Symbols

A	area, m^2
a	acceleration, m/s^2
B	magnetic flux density, T (Tesla)
C_e	electric time constant, s
C_m	mechanical time constant, s
c	viscous damping coefficient, Ns/m
c	sound speed, m/s (Chapter 9)
c_p	specific heat, J/(kg-K) or J/(kg-°C)
D	outer diameter, m (Chapter 1)
d	diameter, m
d_e	effective diameter, m
E	modulus of elasticity or Young's modulus, Pa, MPa or GPa
E	energy, J (Chapters 2 and 9)
E_k	kinetic energy, J
E_p	potential energy, J
E_r	radiative energy, J
E_t	total energy, J
e	mass eccentricity, m
e	emissivity (Chapter 8)
e	specific internal energy, J/kg (Chapter 11)
F	force, N
F_{bt}	bolt axial tensile force, N
F_m	magnetomotive force, A (Ampere)
F_{mc}	fastener clamping force, N
F_u	unbalanced force resulting from unbalanced mass, N
f	frequency, Hz
f_c	centrifugal force, N
f_d	damping force, N
f_s	shape factor (Chapter 3)
f_R	resonant frequency, Hz
f_{tr}	torsional resonant frequency, Hz
G	shear modulus of elasticity, Pa, MPa or GPa
g	acceleration due to gravity, m/s^2
H	magnetic field strength, A/m (ampere per meter)
h	heat transfer coefficient, $W/(m^2\text{-}K)$
h	specific enthalpy, J/kg (Chapter 11)
I	electric current, A
I	moment of inertia, $kg\text{-}m^2$
I_a	second moment of area, m^4 (Chapter 3)
I_p	peak current, A
I_r	rated current, A
i	electric AC current, A
J	second polar moment of area, m^4
J	current density in conductor, A/m^2 (Chapter 2)

Hesai Exhibit 1031, page 28

Ja Jakob number
J_c coupling inertia, kg-m^2
J_{gb} gearbox inertia, kg-m^2
J_l load inertia, kg-m^2
J_m motor inertia, kg-m^2
J_o current density at conductor surface, A/m^2 (Chapter 2)
J_p polar moment of inertia, kg-m^2
J_{rl} reflected load inertia, kg-m^2
K tightening factor (Chapter 5)
K_ϕ ratio of electrical loading on rotor and stator
K_a loss coefficient for actual winding, $K_a = 2\Delta p_a / \rho u_a^2$ (Chapter 11)
KE kinetic energy, J
K_e back EMF constant, V/(rad/s) or V/rpm
K_i current waveform factor
K_m motor constant, N m/W$^{1/2}$ (Chapter 1)
K_p loss coefficient for porous media system, $K_p = 2\Delta p_p / \rho u_p^2$ (Chapter 11)
K_p electrical power waveform factor (Chapter 1)
K_t torque constant, N m/A (Chapter 1)
K_t normal stress concentration factor (Chapter 2)
K_{ts} shear stress concentration factor
K_v motor velocity constant, (rad/s)/V or rpm/V
K_{vd} viscous damping, N m/rpm or N m/(rad/s) (Chapter 1)
k stiffness, N/m
k shape parameter (Chapter 6)
k thermal conductivity, W/(m-K) (Chapters 8 and 11)
k spring coefficient, N-s/m (Chapter 9)
k radius of gyration, m (Chapter 11)
k_b bearing support stiffness, N/m
k_{eff} effective thermal conductivity, W/(m-K) (Chapters 11)
k_l linear stiffness, N/m
L inductance, H
L_e effective stack length, m
L_p sound pressure level, dB (Chapter 9)
L_w sound power level, dB (Chapter 9)
l length, m
l_g grid length, m
M bending moment, N m
M_a angular momentum, J · s or N m s
M_r rotor mass, kg
m mass, kg
m number of motor phases (Chapter 1)
m_c correction mass, kg
m_u unbalance mass, kg
$\dot{m}_a$ mass flowrate through actual winding, kg/s
$\dot{m}_p$ mass flowrate through porous media, kg/s
N_{phase} phase number
N_{slot} number of stator slots
n revolutions per minute, rpm
n_c factor of safety

n_s synchronous speed, rpm (Chapter 1)
n_{slot} slot number per pole
P_a apparent power, W
P_b bearing power loss, W
P_b brake power loss, W (Chapter 10)
PE potential energy, J
PF power factor
P_g gearbox power loss, W
P_{in} motor input power, W
P_{out} motor output power, W
P_p preload of fastener, N
P_r rated power of motor, W (Chapter 1)
P_r power loss due to electric resistance, W (Chapter 7)
P_{react} reactive power, W
P_{real} real power, W
P_s stator I^2R loss, W
P_{sc} stator iron core loss, W
P_s sealing power loss, W
P_w windage power loss, W
p pressure, Pa (Chapter 11)
p thread pitch, m (Chapter 5)
p_r number of pole pair per phase
Q_θ directivity factor (Chapter 9)
q'' heat flux, W/m^2
q''' volumetric heat generation, W/m^3
R electric resistance, Ohm
R supporting force of shaft, N (Chapter 3)
R_c room constant, m^2 (Chapter 9)
Re Reynolds number
R_m magnetic flux path, A(Ampere)/Wb(Weber)
R_{th} thermal resistance, °C/W or K/W
r radius, m
r_c radius from the center of correction mass to rotation axis, m
r_u radius from the center of unbalance mass to rotation axis, m
S material strength, Pa or MPa
S_h volumetric heat source, W/m^3
St Stribeck number (Chapter 6)
S_t torsional stiffness, N m/rad (Chapters 2 and 3)
S_{ut} ultimate tensile strength, Pa or kPa
S_y yield tensile strength, Pa or kPa
s slip
T torque, N m
T oscillating period, s (Chapter 9)
T temperature, °C or K (Chapter 8)
T_a ambient temperature, C or K
T_d damping torque, Nm
T_{dyno} dynamometer torque, N m
T_f final temperature, °C or K
T_{hs} hot spot allowance, °C or K

T_o mean motor torque, N m (Chapter 1)
T_p peak torque, N m
$T_p(t)$ periodic motor torque, as a function of time, N m (Chapter 1)
T_r continuous rated torque, N m
T_s stall torque, N m
t service time, s or h (Chapter 6)
t_f lubricant film thickness, m
U total strain energy, J (Chapter 3)
U unbalance, kg-m (Chapter 2)
u velocity, m/s
V voltage, V
v phase velocity of wave, m/s
W_{def} work done by friction of bolt head, J
W_f work done by friction of nut, J
W_t work done by fastener tightening torque, J
W_{th} work done by fastener thread friction, J

Greeks

α angular acceleration, rad/s^2
α half thread angle, radian or degree (Chapter 5)
α temperature coefficient of resistance, 1/°C or 1/K (Chapters 1, 7, and 10)
α thermal diffusivity (= $k/\rho c_p$), m^2/s (Chapter 8)
$\bar{\alpha}$ average sound absorption coefficient (Chapter 9)
β bearing contact angle, radian or degree (Chapter 6)
β coefficient of thermal expansion, 1/K or 1/°C (Chapter 8)
β frequency ratio, $\beta = \omega/\omega_n$ (Chapter 9)
β thread lead angle, radian or degree (Chapter 5)
Δp_a pressure drop through actual winding, Pa
Δp_p pressure drop through porous media, Pa
δ clearance, m (Chapter 2)
δ change in bolt length, m (Chapter 5)
δ interference, m (Chapter 6)
δ logarithmic decrement (Chapter 11)
ε strain, mm/mm or μm/mm (Chapter 11)
ε material elongation (%)
ε_h hoop strain
Φ magnetic flux, Wb (Weber)
φ torsional deflection angle, radian or degree (Chapter 3)
φ dissipation function, Pa-s^{-1} (Chapter 11)
γ shear strain
γ_g gear ratio
η efficiency (%)
η porosity in porous media, $\eta = A_a/A_p$ (Chapter 11)
φ phase angle, radian or degree (Chapters 2 and 9)
λ sound wavelength, m (Chapter 9)

λ	coefficient of bulk viscosity (or expansion viscosity), Pa-s or N s/m^2 (Chapter 11)
λ	scale parameter (Chapter 6)
μ	dynamic viscosity, Pa-s or N s/m^2 (Chapter 11)
μ	permeability, m^2 (Chapter 9)
μ_{eff}	effective viscosity, Pa-s or N s/m^2
μ_h	friction coefficient under bolt head
μ_n	friction coefficient under nut
μ_h	friction coefficient under bolt head
μ_{th}	fastener thread friction coefficient
μ_r	relative permeability
ρ	density, kg/m^3
ρ	electric resistivity, Ω·m (Chapters 7 and 10)
σ	stress, N/m^2 or Pa
σ_h	hoop stress, N/m^2
σ_{rad}	radiation efficiency (Chapter 9)
τ	shear stress, N/m^2
τ_e	electrical time constant, s
τ_m	mechanical time constant, s
τ_{th}	thermal time constant, s
ω	annular rotating speed, rad/s (Chapter 1)
ω	oscillating frequency, Hz (Chapter 9)
ω_{dyno}	dynamometer annular speed, rad/s
ω_n	natural frequency, Hz
ω_p	peak annular rotating speed, rad/s
ω_r	rated annular speed, rad/s
ξ	load-to-motor inertia ratio
ξ	porous media inertial resistance factor, $\xi = K_p/\Delta L$, m^{-1} (Chapter 11)
ζ	damping ratio, $\zeta = c/2(mk)^{1/2}$

Abbreviations

AC	alternative current
AF	axial flux
BDRM	brushless double-rotor machine
CAD	computer-aided design
CDF	cumulative distribution function
CFD	computational fluid dynamics
CFM	cubic feet per minute
CMM	coordinate measuring machine
CNC	computer numerical control
CS-PMSM	compound-structure permanent magnet synchronous machine
CTE	coefficient of thermal expansion
CVD	chemical vapor deposition
DC	direct current
DDR	direct drive rotary
DIR	direction of rotation

DSP	digital signal processor
DSPM	doubly salient permanent magnet
DVA	dynamic vibration absorber
EDM	electrical discharge machining
EMF	electromotive force
EPC	evaporative pattern casting
EVS	electric vehicle system
FCPM	flux-controllable permanent magnet
FC-TFM	flux concentrated transverse flux machine
FEA	finite element analysis
FFT	fast Fourier transform
FNM	flow network modeling
FS	factor of safety
FVM	finite volume method
HDD	hard disk drive
HESM	hybrid excitation synchronous machine
HNS	high nitrogen steel
HSM	hysteresis synchronous motor
HTS	high-temperature superconductor
ICE	internal combustion engine
IGBT	insulated gate bipolar transistor
IM	induction motor
IP	international protection rating
IPM	interior permanent magnet
LDV	laser Doppler vibrometer
LHM	linear hybrid motor
LIM	linear induction motor
LPM	lumped parameter model
LSM	linear synchronous motor
MF	magnification factor
MMF	magnetomotive force
MPM	mixing plane model
MRF	multiple reference frame
Nd-Fe-B	neodymium-iron-boron
NEMS	nano-electromechanical system
NSM	negative stiffness mechanism
OEM	original equipment manufacturer
PCB	printed circuit board
PCMM	pole-changing memory motor
PDF	probability density function
PF	power factor
PI	polarization index
PM	permanent magnet
PMHS	permanent magnet hysteresis synchronous
PMM	permanent magnet motor
PMSM	permanent magnet synchronous motor
PMW	pulse width modulation
PVD	physical vapor deposition
RF	radial flux

RMS	root mean square
RNG	renormalization group
RDT	rim-driven thruster
RTD	resistance temperature detector
rpm	revolutions per minute
rps	revolutions per second
Sm-Co	samarium-cobalt
SPL	sound pressure level
SPM	surface-mounted permanent magnet
SPM-TFM	surface permanent magnet transverse flux machine
SRF	single reference frame
SRM	switched reluctance motor
STL	sound transmission loss
TEFC	totally enclosed fan-cooled
TF	transverse flux
TFM	transverse flux machine
TFRM	transverse flux reluctance motor
TMD	tuned mass damper
UMP	unbalanced magnetic pull
VCA	voice coil actuator
VCM	variable-capacitance micromotor
VFMM	variable flux memory motor
VPI	vacuum pressure impregnation
VRM	variable reluctance motor
YASA	yokeless and segmented armature

1

Introduction to Electric Motors

Electric motors are devices that convert electric energy into magnetic energy and finally into mechanical energy. Electromagnetism is the basis of electric motor operation by generating magnetic forces necessary to produce either rotational or linear motion. For rotating electric motors, it is the interaction between the stator and rotor magnetic fields that creates motor torque to drive external loads.

Today, electric motors come in a wide variety of types, sizes, operating characteristics, and configurations to suit different applications. They are used almost everywhere in the world, including industrial drives, household appliances, medical devices, electronic products, robots, electric vehicles, machine tools, spacecrafts, and military equipment. As one of the fastest growing industrial sectors, electric motor manufacturing represents a major industry worldwide. Today, electric-motor-driven systems account for approximately 45% of total global electricity consumption. By 2030, energy consumption from electric motors is expected to rise to 13,360 terawatt hour (TWh) per year. End users now spend USD 565 billion per year on electricity used in electric-motor-driven systems; by 2030, that could rise to almost USD 900 billion [1.1]. In the United States, motor-driven equipment accounts for 64% of the electricity consumed in the manufacturing sector. That is approximately 290 billion kilowatt hours (kWh) of power per year [1.2]. There are more than 40 million electric motors used in manufacturing operation [1.3]. In addition, more than 95% of an electric motor's life-cycle cost is the energy cost. In China, it is estimated that about 60% of the annual power generation is consumed through motor-driven systems. All these clearly show how important it is to take a variety of measures to promote electric motor efficiency for the energy saving and carbon emission reduction.

1.1 History of Electric Motors

The discoveries of phenomena of static electricity can be traced back in ancient Greece about 2600 years ago. However, there was little real progress until the English scientist William Gilbert in 1600 described the electrification of many substances and coined the term *electricity*, the Greek word for amber.

Hans Oersted discovered electromagnetism in 1820 and additional works were made by a number of other scientists such as William Sturgeon, Joseph Henry, Ander Marie Ampere, Michael Faraday, and Thomas Davenport. In 1831, Michael Faraday discovered electromagnetic induction, the principle behind the electric motor and generator. This discovery was crucial in allowing electricity to be transformed from a curiosity into a powerful new technology.

Using a broad definition of *motion* as meaning any apparatus that converts electric energy into motion, it is widely accepted that Michael Faraday invented the first direct

Hesai Exhibit 1031, page 36

current (DC) electric motor in 1821. This motor was basically used to confirm his concept of electric motor and had no actual value in application. He succeeded in building the practical electric motor 10 years later. Following his groundbreaking work, many scientists had contributed to the developments of electric motors. William Sturgeon invented the first practical electric motor in the United Kingdom in 1832 [1.4]. The first US patent on electric motor was granted in 1837 to Thomas Davenport [1.5]. In 1887, Nikola Tesla introduced the world's first alternating current (AC) motor and gained a US patent in the next year [1.6]. Three-phase *cage-rotor* induction motors (IMs) were invented by Mikhail Dolivo-Dobrovolsky during 1889–1890 [1.7]. Even today, this type of motors is still in service for the vast majority of commercial applications.

1.2 Motor Design Characteristics

Electric motors are manufactured in a variety of types and configurations. Typically, an electric motor assembly is formed from a collection of parts, including a stator, a rotor, a shaft, a pair of end bells, bearings, and a motor housing supporting and enclosing the various components. In addition to these primary motor components, some motors may include electronic components that are used to modify operating characteristics for particular applications.

Motor design characteristics are the essential elements in the motor design and manufacturing processes. To select appropriate motors for specific applications, these design characteristics must be well understood.

1.2.1 Motor Torque

Torque is a measurement of the turning force acting on an object to cause that object rotating or twisting about an axis or pivot. Torque, like *work*, is measured as Newton meter (N-m) in the International System of Units (SI system) or pound-foot (lb_f-ft) in the English system. However, unlike *work* that only occurs during displacement, torque may exist even though no displacement or rotation occurs. A typical example is the static holding torque.

1.2.1.1 Static and Dynamic Torque

Torque can be divided into two major categories, either static or dynamic torque. From the standpoint of physics, the system is considered static if it has no angular or linear acceleration. Static torque refers to the amount of torque that an electric motor produces at zero speed (i.e., the motor is in a real static state) with the power output $P_{out} = 0$. By contrast, dynamic torque refers to the amount of torque that an electric motor produces at variable speeds of rotation with load applied (i.e., the motor is in a dynamic state) with $P_{out} > 0$. Simply, static torque is associated with forces that do not involve angular acceleration/deceleration, and dynamic torque is associated with dynamic forces that arise from acceleration/deceleration, following Newton's second law.

Some motor manufacturers may provide the information of continuous static torque rating for customers, indicating the motor is capable of supplying that static torque at zero speed of rotation continuously. However, this information may be not very useful in motor

Hesai Exhibit 1031, page 37

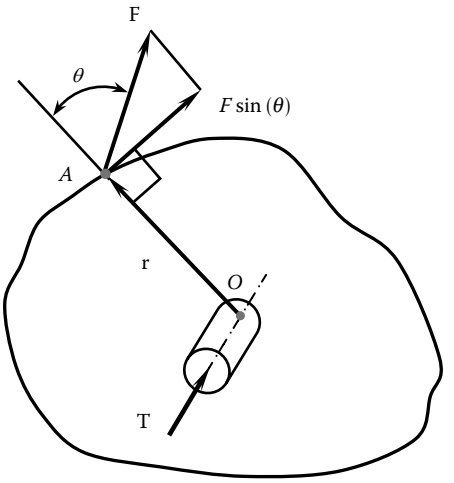


FIGURE 1.1
Torque vector ***T*** acting on a solid body to make it rotate.

selection because it does not define the continuous torque available from the motor at a specific speed (or in a range of speeds) to drive external load.

As a continuous torque rating is provided at a specific rated speed (or in a range of speeds), the torque is thus the dynamic torque, indicating the capability of the motor to provide up to a corresponding rated torque continuously. Furthermore, the maximum torque appearing on the motor nameplate refers to the highest dynamic torque for the motor at rated motor speed.

As shown in Figure 1.1, when a force vector **F** acts on a solid body at the point *A* to make the body rotate about its axis through the point *O*, the torque vector **T** around the point *O* is obtained by crossing product of the radial displacement vector **r** and the force vector **F**:

$$\mathbf{T} = \mathbf{r} \times \mathbf{F} \tag{1.1}$$

The direction of the torque vector **T** is determined by the right-hand rule, that is, it is perpendicular to both **r** and **F**. Correspondingly, the magnitude of the torque acting on the body is

$$T = rF\sin(\theta) \tag{1.2}$$

where the moment arm $r = \overline{OA}$, defined as the distance from the axis to the point where the force is applied, and θ is the measure of the smaller angle between the displacement vector **r** and the force vector **F**. It is worth to note that torque calculated from Equation 1.2 can be either static or dynamic torque depending on whether *F* is static or dynamic force.

For a rotating system with a fixed axis, the dynamic torque on the rotating system along axis of rotation is determined by the rate of change of the system angular momentum:

$$T = \frac{dM_a}{dt} \tag{1.3}$$

where M_a is the angular momentum of the rotating system, measured in N-m-s. M_a can be expressed as the product of the polar moment of inertia of the rotating system J_p and the rotating speed ω, that is,

$$M_a = J_p \omega \tag{1.4}$$

As a result, the torque on the rotating system can be expressed as

$$T = \frac{d(J_p \omega)}{dt} = J_p \frac{d\omega}{dt} = J_p \alpha \tag{1.5}$$

where α is the angular acceleration of the rotating system, measured in rad/s^2. This equation indicates that for electric motors, the less the motor inertia, the less torque the motor needs to produce to meet a desired acceleration rate. As a result, it is advantageous to minimize motor inertia to the greatest extent to maximize acceleration.

Servomotors are typically expected to accelerate loads from a stop to a given velocity and then decelerate the loads once again to a stop at precise position. Accordingly, to move or stop loads as fast as possible, the angular acceleration/deceleration α must be maintained high enough. As a result, the motor's polar moment of inertia J_p has to be kept at very low levels. For this reason, the motor inertia must be taken into account in the earlier stage of motor design.

1.2.1.2 Motor Torque in Motor-Load System

When a motor drives a load machine to perform work, it usually connects with some power transmission components such as coupling and gearbox. For demonstration purpose, a two-shaft gearbox is shown in Figure 1.2. The gear ratio γ_g is defined as the rotating speed of the gearbox input shaft that is directly coupled with the rotor shaft to the rotating speed of the gearbox output shaft, which is directly coupled with the load machine, that is,

$$\gamma_g = \frac{\omega_i}{\omega_o} = \frac{\omega_m}{\omega_l} \tag{1.6}$$

where

ω_i and ω_o are the rotating speeds of the input and output shaft of the gearbox
ω_m and ω_l are the rotating speeds of the motor and load, respectively

Unlike other components, because the input and output shaft of a gearbox have different rotating speeds, the inertia of each shaft must be calculated separately.

To determine the motor torque required for the system, a concept of reflected load inertia is introduced as the equivalent inertia of the load seen by the motor:

$$J_{rl} = J_c + J_{i,gb} + \frac{J_{o,gb} + J_l}{\gamma_g^2} \tag{1.7}$$

where J_c and J_l are the inertias of the coupling and load machine, respectively. This equation indicates that load inertia is reduced by the square of the gear ratio, which can be

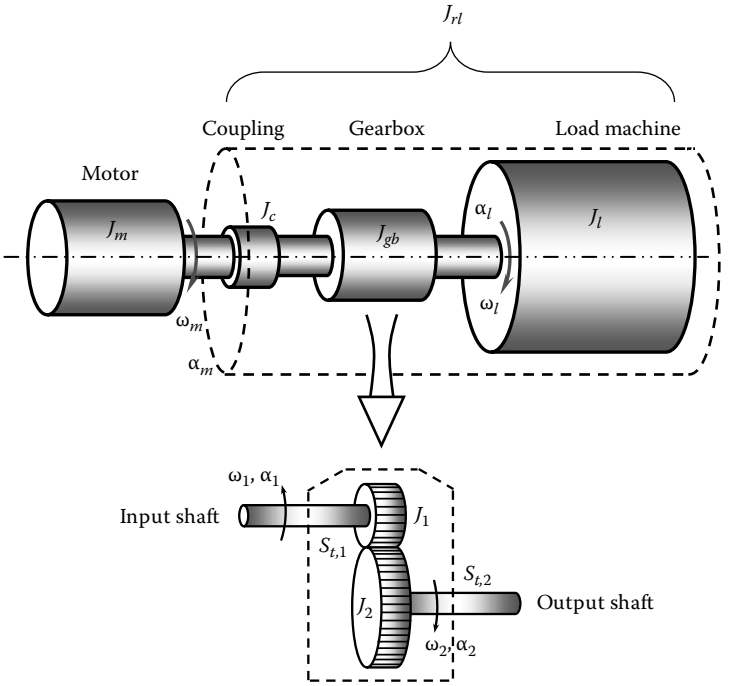


FIGURE 1.2
A motor is connected with a driven load machine via a coupling and a gearbox. The reflected load inertia is the equivalent inertia of the load seen by the motor.

derived from conservation of energy or power in a set of rotating components. In this resultant equation, it is assumed that the inertias of the coupling and gearbox are considered part of the load inertia.

The torque required at the motor thus becomes

$$T_m = \left(J_m + J_{rl}\right)\alpha_m = \left(J_m + J_c + J_{i,gb} + \frac{J_{o,gb} + J_l}{\gamma_g^2}\right)\alpha_m \tag{1.8}$$

The motor angular acceleration α_m has the relationship with the load angular acceleration α_l [1.8]:

$$\alpha_m = \alpha_l \gamma_g \tag{1.9}$$

Combining the aforementioned equations yields

$$\alpha_l = \frac{\alpha_m}{\gamma_g} = \frac{T_m}{\gamma_g}\left(J_m + J_c + J_{i,gb} + \frac{J_{o,gb} + J_l}{\gamma_g^2}\right)^{-1} \tag{1.10}$$

Taking the derivative of α_l with respect to γ_g, it follows that

$$\frac{d\alpha_l}{d\gamma_g} = T_m \frac{\left(J_l + J_{o,gb}\right) - \left(J_m + J_c + J_{i,gb}\right)\gamma_g^2}{\left[\left(J_l + J_{o,gb}\right) + \left(J_m + J_c + J_{i,gb}\right)\gamma_g^2\right]^2} \tag{1.11}$$

Let $d\alpha_l/d\gamma_g = 0$; thus,

$$\left(J_l + J_{o,gb}\right) = \left(J_m + J_c + J_{i,gb}\right)\gamma_g^2 \tag{1.12}$$

Thus, the optimum gear ratio is found at

$$\gamma_g = \sqrt{\frac{J_l + J_{o,gb}}{J_m + J_c + J_{i,gb}}} \tag{1.13}$$

The torque on a motor shaft can be measured using a number of strain gages to the shaft in a proper orientation. This allows directly measuring torsional shear strains that can be calibrated to output torque.

1.2.1.3 Continuous Torque

It is the interaction of the stator revolving field and the rotor induced field that produces the motor torque to drive load machines. Consequently, the motor torque is a function of both the field and armature currents and acts on both the rotor and stator simultaneously. The continuous rated torque T_r (in N-m) of a motor can be determined from its rated power P_r (in W) and rated rotor angular speed ω_r (in rad/s):

$$T_r = \frac{P_r}{\omega_r} \tag{1.14}$$

At normal operation, a motor provides a continuous torque, known as nominal torque, to drive an external load device smoothly. It can be seen from the previously mentioned equation that for a given motor power, the continuous torque is inversely proportional to the motor rotating speed. In practice, the continuous torque generated in a motor commonly has a cyclic variation as a result of the cyclic permeance variation that occurs as the rotating member moves with respect to the stationary member. The instantaneous torque of a motor can be expressed as (Figure 1.3)

$$T(t) = T_o + T_p(t) \tag{1.15}$$

where T_o and $T_p(t)$ are the mean component and the periodic component of the motor torque, respectively.

1.2.1.4 Peak Torque

The motor peak torque, which is always associated with the peak current, is the maximum torque a motor can produce for short periods of time without exceeding the motor

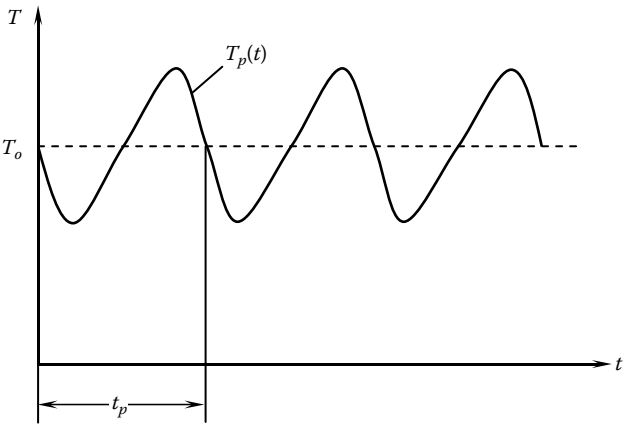


FIGURE 1.3
Mean component and periodic component of motor torque.

temperature limit or safe operating torque. A motor that operates under a peak torque condition is typically associated with a quick temperature rise. When the motor temperature exceeds the allowable value, it will cause degradation of insulation materials, irreversible demagnetization of permanent magnets (PMs) in PM motors (PMMs), and winding damage and, in turn, lead to the degradation of motor performance or even motor failure.

Peak torque contains two components [1.9]: (1) acceleration torque as inflicted by inertia forces with the maximum angular acceleration and (2) constant torque due to all other non-inertial forces such as gravity, friction, preloads, and other push–pull forces.

The continuous torque and peak torque curves are usually determined through motor testing and provided by motor manufacturers.

1.2.1.5 Stall Torque

The stall torque is the torque that a motor produces at zero rotating speed where $P_{\text{out}} = 0$. The stall torque, also known as locked rotor torque (LRT), can be measured with the rotor being locked. For a DC motor, the motor torque has a linear relationship with the motor rotating speed. The maximum torque occurs at zero rotating speed and zero torque occurs at maximum rotating speed. For an IM, the stall torque can be calculated as [1.10]

$$T_s = \frac{C_1 k_2 \left(P_{in,s} - P_s - P_{sc} \right)}{n_s} \tag{1.16}$$

where

C_1 is a reduction factor ($0.9 \le C_1 \le 1.0$) to account for nonfundamental losses
$k_2 = 9.549$ for torque in the unit of N-m
$P_{in,s}$ is the input power to stator (in W)
P_s is the stator I^2R loss (in W)
P_{sc} is the stator iron core loss (in W)
n_s is the synchronous speed in the unit of rpm

1.2.1.6 Cogging Torque and Reduction Methods

The torque generated from a PMM usually consists of two components: the effective driving torque, which is basically proportional to the supplied electric current, and the *no-current torque*, which is independent of the current, such as cogging torque.

Cogging torque originates from the interaction between rotor-mounted PMs and the stator teeth, which produces reluctance variations depending on rotor position [1.11]. As depicted in Figure 1.4, when the rotor rotates, the magnets attached to the rotor successively pass through the stator teeth and slot openings, resulting in the periodic variations of the magnetic field. During the process, the PM rotor tends to lock onto the position where the permeability reaches the largest. When the rotor deviates from this equilibrium position, tangential forces are produced between the magnets and stator teeth, either to return the rotor back to the old equilibrium position or to push the rotor to the next equilibrium position, leading to cogging torque.

Cogging torque is highly undesirable because it is the major cause of motor vibration and acoustic noise, particularly at light loads and low speeds. Even for high-speed applications, lower cogging torque always benefits smooth operation. For instance, in a servo system, the motor may come to a stop and then accelerate to another high speed after the stop. As the motor approaches zero speed, it is often to use settling time or settling error to measure how smoothly the motor approaches zero speed. This can definitely be impacted by cogging torque.

Because the rotor always tends to lock onto a position where it is aligned with the stator poles, it makes precise positioning of the rotor difficult. In addition, cogging torque is also an important source of torque ripples that have adverse effects in many demanding motion control applications.

Motor manufacturers often adopt the cogging torque ratio to quantitatively describe the level of cogging torque. This torque ratio is defined as the absolute cogging torque, which is characterized as peak-to-peak cogging torque, to the rated continuous torque. It is widely accepted that for regularly controlled servo systems, the maximum cogging torque ratio should keep under 5%. More ideally, for precisely controlled servo systems, the cogging torque ratio is 2% or less. Cogging torque below 1% really requires special design. For example, in some high-precision film roller systems that need super smooth operation at

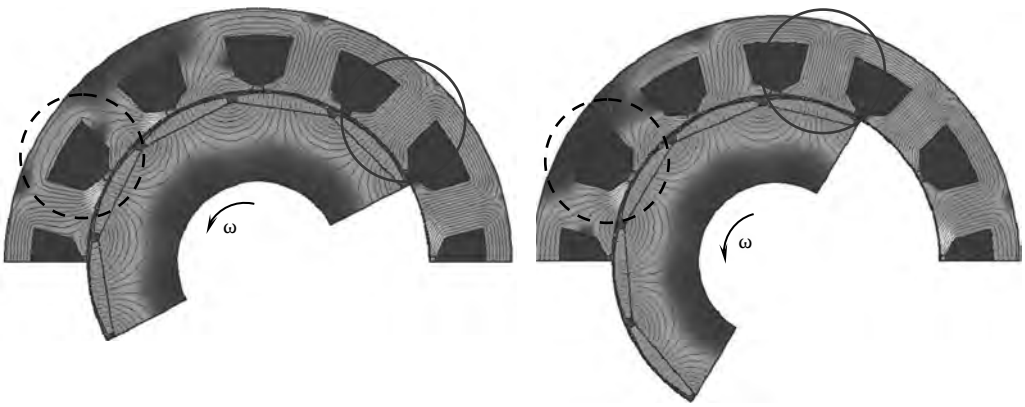


FIGURE 1.4
As a rotor rotates, magnets successively pass through stator teeth and slot, leading to different magnetic flux distributions, as shown earlier for magnet-teeth (solid circle) and magnet-slot (dashed circle) conditions.

very low speeds, an effective solution is to use DC torque motors for excellent operation. Three-phase brushless motors have been designed for very smooth telescope azimuth motion, that rotate at extremely slow earth rates of rotation of one revolution in 24 h. It is worth to note that each application may allow for different extremes of cogging torque.

Though cogging torque can be calculated accurately using numerical approaches such as finite element methods (FEMs), with simplified motor models, analytical methods may provide greater physical insight into the mechanism of cogging torque production. This is especially useful at the initial stage of the motor design. One approach to predict cogging torque is represented by [1.12]

$$T_{cog} = -\frac{\phi_g^2}{2}\frac{dR}{d\theta} \tag{1.17}$$

where

ϕ_g is the air gap flux
R is the air gap reluctance
θ is the position of the rotor

This equation indicates that in order to reduce cogging torque, either the air gap flux ϕ_g or the rate of change of the air gap reluctance $dR/d\theta$ must be minimized. In practice, cogging torque is reduced by forcing the air gap reluctance R as close as possible to constant with respect to rotor position.

Another method for calculating cogging torque in PMMs was proposed by Lu et al. [1.13]:

$$T_{cog} = -\frac{p_r B_r}{2}\frac{l_m}{\mu_o \mu_r}\frac{d\Phi}{d\theta} \tag{1.18}$$

where

p_r is the number of magnet pole pairs
B_r is the remanent magnetic flux density at $H = 0$
Φ is the magnetic flux calculated over a surface perpendicular to its direction of magnetization
l_m is the magnet length along the magnetization direction
μ_o and μ_r are the permeability of free space and the relative permeability of the magnet material, respectively

This equation shows that the magnitude of cogging torque is proportional to the number of magnet pole pairs, the remanent flux density value, the magnet length in the direction of magnetization, and the variation of the magnetic flux with respect to the rotor position. It has shown a good agreement of the predicted cogging torque with the cogging torque calculated using Maxwell stress method and measured results.

There are a number of techniques available for reducing cogging torque:

a. The most effective way in practice to reduce cogging torque is to skew either the stator teeth relative to the rotor centerline (Figure 1.5) or rotor magnets relative to the stator teeth, where Figure 1.6 is for step-skewing of segmented magnets

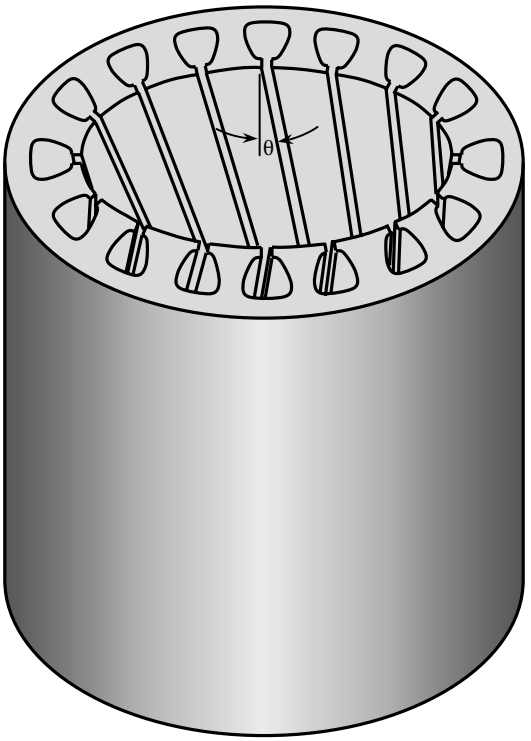


FIGURE 1.5
Skewed stator teeth with respect to its centerline for reducing cogging torque, where θ is the skew angle.

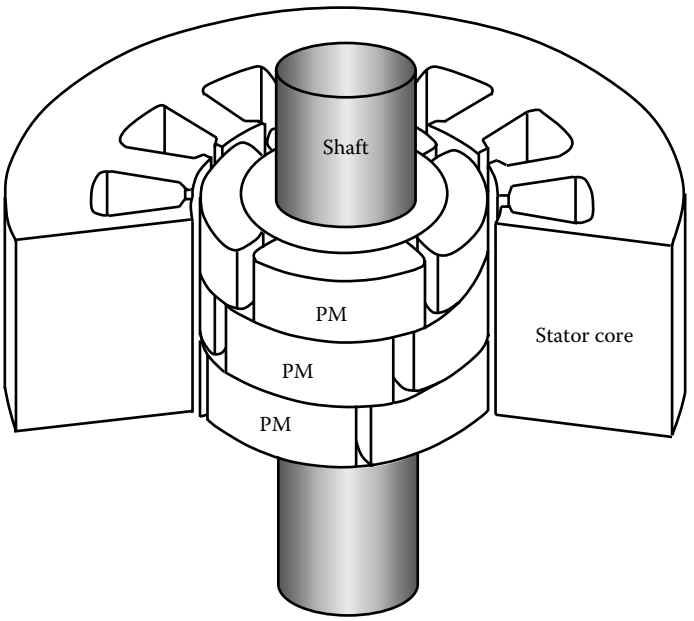


FIGURE 1.6
Step-skewed permanent magnets relative to stator teeth for reducing cogging torque.

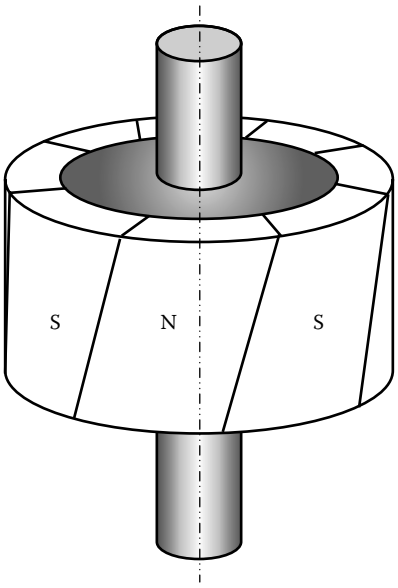


FIGURE 1.7
Skewed PMs on rotor core for reducing cogging torque.

and Figure 1.7 [1.14] is for skewing whole magnets. More detailed descriptions can be found in reference [1.15]. Skewing stator teeth needs special care in stacking steel laminations. Skewing whole magnets may have smoother rotor performance than step-skewing segmented magnets. However, it is to be noted that rotor skewing may also decrease the rotor saliency and thus reduces the back electromotive force (EMF) and motor effective torque. All forms of skewing will reduce the back EMF and motor effective torque as skewing affects total flux linkage to the coils.

b. Another technique is to use a fractional number of slots per pole. The use of this method not only reduces the amplitude of the cogging torque but also increases the fundamental order. This is because the stator slots are located at different relative circumferential positions with respect to the edges of the magnets [1.16–1.18]. To address the problem, the parameter q, which is defined as the slot number per pole n_{slot} divided by the phase number N_{phase}, is introduced as

$$q = \frac{N_{slot}}{2p_r N_{phase}} = \frac{n_{slot}}{N_{phase}} \tag{1.19}$$

where
p_r is the number of pole pair
N_{slot} is the number of the stator slots

The fractional slot winding arrangement with $q < 1$ is attractive for lower cogging torque. The investigations have shown that the cogging torque of the fractional slot motors can be less than 1% of the rated torque. In the case of multipole machines, the cogging torque of 0.05% could be estimated [1.19].

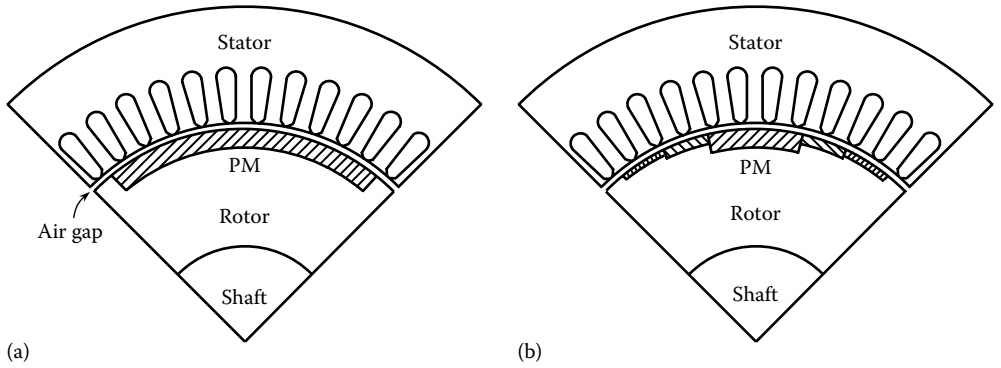


FIGURE 1.8
Rotors with surface-inset single PM (a) and stepped PMs (b).

c. The reduction in cogging torque can be also achieved by modulating drive current waveform [1.20].
d. It is a well-established fact that the magnet pole arc can have a large effect on the magnitude of the cogging torque [1.21,1.22]. The optimization of the magnet pole arc can reduce the harmonics of the air gap flux wave and the permeance wave [1.23].
e. Another technique is called magnet segmentation in which a pole magnet consists of several elementary magnet segments with the same polarities [1.24]. Furthermore, as shown in Figure 1.8, the segmented magnets may be selected with different thickness and lengths to obtain more sinusoidal flux density waves [1.25].
f. The shape of the magnetic pole has a strong impact on the uniformity of the stator–rotor air gap. A comparison of uniform and nonuniform air gap with different pole shapes of magnets is illustrated in Figure 1.9. It has been reported that the nonuniform pole shape of magnets can reduce cogging torque as high as 50% [1.25]. The normalized cogging torque to the peak torque between uniform air gap and nonuniform air gap is given in Figure 1.10. The simulation studies of the pole surface effect on cogging torque were performed by Lao et al. [1.26].
g. Other mechanical factors can affect cogging torque. Geometry factors such as rotor or magnet concentricity of the magnetic air gap can create eccentric magnet to stator teeth flux linkage. These variations in flux linkage can cause variations in cogging torque and torque ripple.
h. In order to achieve decreasing torque ripple, iron losses, and cogging torque for interior PM (IPM) synchronous machines, Soleimani et al. [1.27] proposed a novel structure of rotor. In their design, three layers of PM have been used and each layer has a fragmental trapezoid structure, as shown in Figure 1.11. With the optimized dimensions and shapes of the buried rotor magnets, the cogging torque ratio of 1.82% is achieved, comparing with torque ratio of 5% in conventional IPM machines.

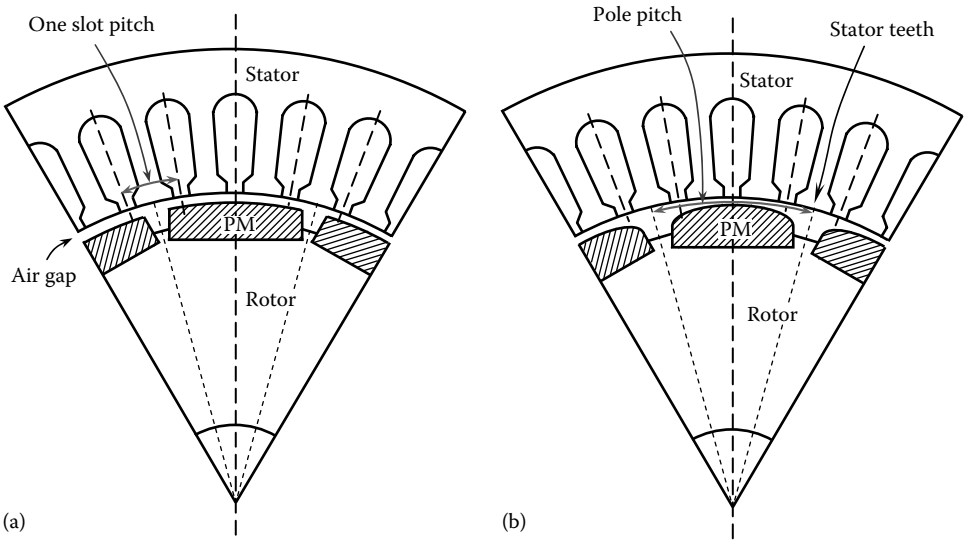


FIGURE 1.9
Surface-mounted PMs with different shapes for (a) uniform air gap and (b) nonuniform air gap.

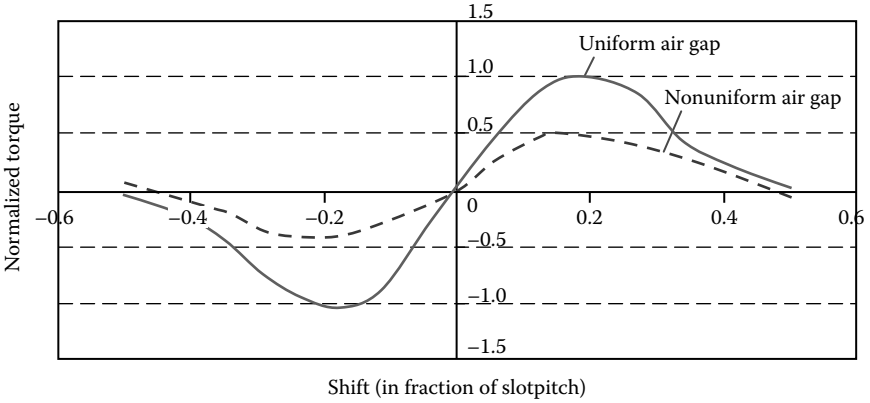


FIGURE 1.10
Comparison of cogging torque for uniform and nonuniform air gap (bread-loaf magnet shape).

1.2.1.7 Torque Ripple

PM synchronous motors usually generate torque ripple during their normal operation. The studies of Hsu et al. [1.28] have revealed that torque ripple can be classified into four types depending on the nature of their origin:

- Pulsating torque, which is inherently produced by the trapezoidal back EMF. The torque ripples caused by pulsating torque may be reduced by purposely produced fluctuating counter torques.
- Fluctuating torque, which is produced by altering the magnitudes of phase current in the same ratio.

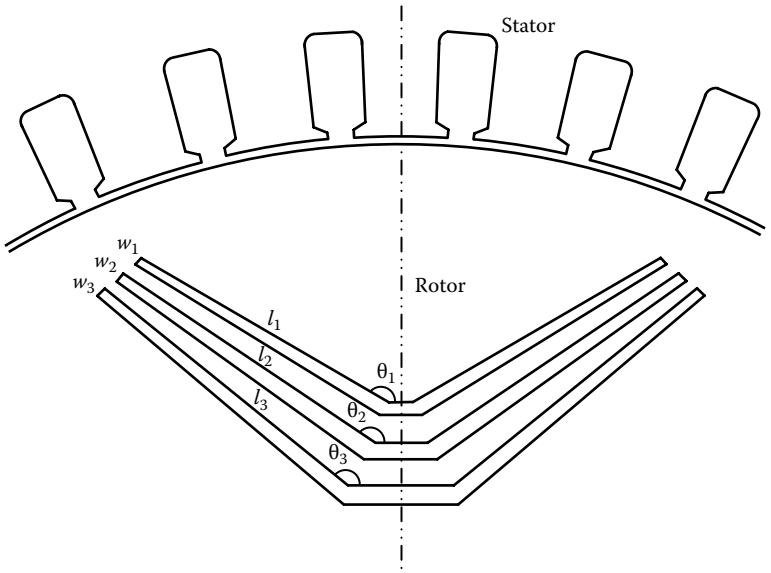


FIGURE 1.11
Novel structure of rotor for minimizing cogging torque for IPM synchronous motors.

- Reluctance cogging torque, which is produced by nonuniformly distributed air gap permeance associated with the teeth and PMs. Actually, this type of torque exists even when the motor is not energized.
- Inertia and mechanical system torques, which are generated by the dynamic motions of the mechanical components of motor. They are affected by the driven device.

Later, Holtz and Springob [1.29] have investigated the different sources of torque ripple in PM machines, including the distortion of the stator flux linkage distribution, variable magnetic reluctance at the stator slots, and secondary phenomena. In addition, the feeding power converter also contributes to torque ripple due to the time harmonics in the current waveform's time-varying delays between the commanded and the actual current.

1.2.2 Motor Speed

There are a few terms to describe rotational motion. Angular speed represents the change in angle per unit of time, typically measured in radians per second (rad/s) in the SI system and in degrees per second in the English system. Rotational speed is the measurement of revolutions per unit of time, expressed as either revolutions per second (Hz) or revolutions per minute (rpm). It is to be noted that speed is a scalar quantity and velocity is a vector. Thus, the difference between angular (or rotational) speed and angular (or rotational) velocity is that the latter contains the information of the rotational direction. Angular speed ω (in rad/s) and rotational speed n (in rpm) can be converted into each other:

$$\omega = \frac{\pi n}{30} \tag{1.20}$$

1.2.2.1 Continuous Speed

The continuous speed of a motor is an important design parameter, defining the nominal speed at which the motor continuously operates at the rated voltage and current to drive the full motor load. It depends on the number of the motor pole, the frequency of AC, and the amount of torque.

1.2.2.2 Peak Speed

The peak speed is the maximum speed a motor can reach during operation. It must be noted that at the peak rotating speed, all rotating components are subject to high centrifugal forces. Even a tiny unbalance in a high rotating speed system can cause severe motor vibration or serious damage of rotating components. Therefore, the motor peak speed needs to be carefully determined during motor design phase.

1.2.2.3 Speed Ripple

Speed or velocity ripple refers to the variations in steady-state speed in time. Velocity ripple values are usually defined as a percentage (%) of deviation from the ideal value. In practice, torque ripple has the tendency to cause speed ripple.

There are a large number of factors that affect motor velocity ripple and torque ripple [1.30], including

- Load-to-rotor inertia ratio
- Motor pole count—higher pole count leads to lower ripple
- Encoder resolution
- Commanded velocity
- System bandwidth (different between an analog and a digital drive)
- Commutation type (sinusoidal vs. trapezoidal commutation)
- System trajectory update rate (when using a digital drive)
- Natural (harmonic) frequencies of the system as a whole
- Mechanical resonance of motor components
- Mechanical friction in the system
- Physical alignment of mechanical components
- Inherent performance variances between two like components
- Load damping
- Consistency of the AC supply voltage to the drive (i.e., power supply integrity)
- Sampling rate/resolution of the tachometer (or velocity-measuring device)

1.2.3 Torque Density

Torque density is defined as the ratio of the nominal continuous torque T to the motor volume V, as the measure of the torque-carrying capability per unit volume. High torque density and high efficiency are two of the most desirable features for electrical motors. Torque density is a measure of the torque-carrying capability per unit volume of a motor, expressed in units of N/m^2 or lb_f/ft^2. Torque density is a system property since it depends

on the design of motor components and their interconnections. One of the main design goals in motor design is to improve torque density of motors.

1.2.4 Motor Power and Power Factor

The power output P_{out} of rotary motors is expressed as the product of the motor torque and the angular rotating speed, that is,

$$P_{out} = T\omega \tag{1.21}$$

Since the angular rotating speed ω (in rad/s) is related to the rotating speed n (in rpm) as $\omega = \pi n/30$, Equation 1.21 can be also expressed as

$$P_{out} = \frac{\pi n T}{30} \tag{1.22}$$

When an IM is connected to a power supply but still at rest, it appears just like a short-circuited transformer, drawing a very high current known as the locked rotor current (LRC). The torque corresponding the LRC is defined as the LRT. A motor that exhibits a high starting current will generally produce a low starting torque, and vice versa. Both the torque and current of the locked rotor are a function of the terminal voltage of the motor. Under a constant voltage, the torque and current vary with the rotor speed during the motor acceleration/deceleration process.

The most important two parameters for motor performance are motor efficiency and power factor. In the electric power industry, power factor PF is defined as the ratio of real power P_{real} to apparent power P_a. As shown in Figure 1.12, power factor PF (where $0 \le PF \le 1$) can be also expressed as the cosine of the impedance phase angle ϕ:

$$PF = \cos\phi = \frac{P_{real}}{P_a} \tag{1.23}$$

The power supply system provides both real and reactive power to operate the motor. Useful mechanical work is developed from real power P_{real} and is measured in watts (W)

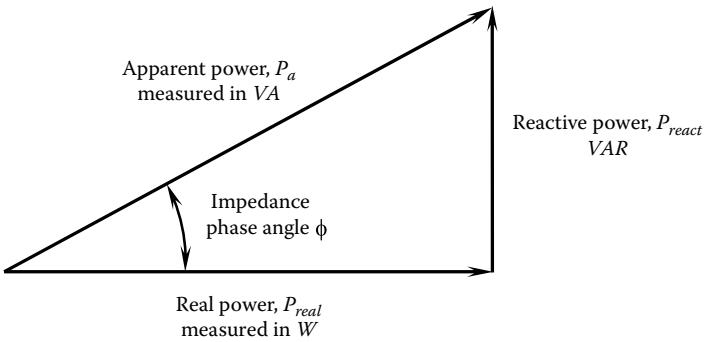


FIGURE 1.12
The power triangle: the relationship of real power, reactive power, and apparent power.

or kilowatts (kW). For AC motors, reactive power P_{react} is to develop magnetic fields. It is worth to note that reactive power does not provide any mechanical work. From the power triangle in Figure 1.12, apparent power P_a can be expressed as

$$P_a = \sqrt{P_{real}^2 + P_{react}^2} \tag{1.24}$$

Power factor depends upon motor load. When a motor is in normal operation, the electric current drawn by the motor varies with the external load. Under a no-load condition such as the motor start, the power factor is minimum, typically 0.1–0.25. As the shaft load increases, the load current through the stator windings increases significantly, and consequently, the power factor increases until reaching the maximum at the full-load point. Then, the power factor falls again as the motor approaches the full speed. In engineering practice, *PF* is determined at each load point using the following formula:

$$PF = \frac{P_{in}}{\sqrt{3}\,VI} \tag{1.25}$$

where
P_{in} is the motor input power
V and I are line-to-line voltage and current, respectively

The full-load power factor of an IM can vary from 0.5 for a small low-speed motor up to 0.9 for a large high-speed machine.

1.2.5 Torque–Speed Characteristics

In the power industry, a commonly used method of displaying motor performance characteristics graphically is to use motor torque–speed curves, as shown in Figure 1.13 for

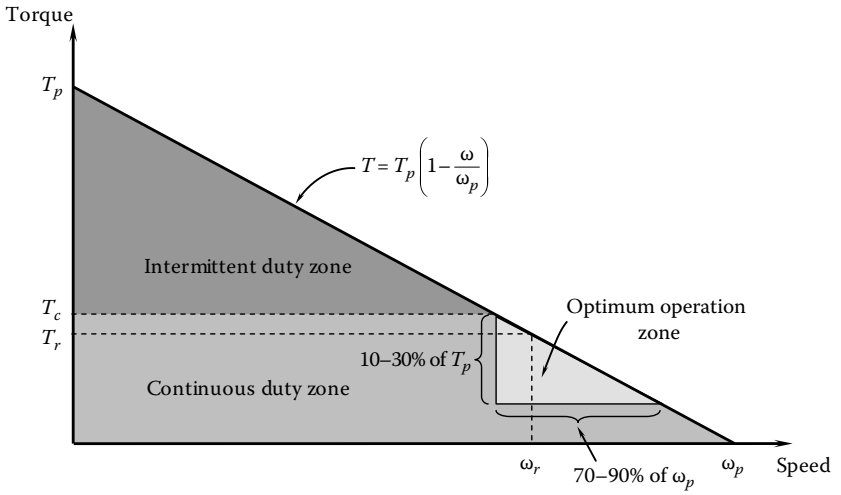


FIGURE 1.13
Torque–speed characteristics of a typical DC motor.

a typical DC motor. There are two pairs of parameters used to define the DC motor performance: one pair of torque, which includes peak torque T_p and rated torque T_r, and another pair of speed, which includes peak speed ω_p and rated speed ω_r. Since the peak torque occurs at $\omega = 0$, it also called the stall torque. The peak rotating speed is also called no load speed because the motor generates no torque at the point. As depicted in the figure, the linear relationship between the motor torque and the rotating speed can be expressed as

$$T = T_p\left(1 - \frac{\omega}{\omega_p}\right) \tag{1.26}$$

As shown in Equation 1.21, the motor output power is the product of torque and angular rotating speed. Thus, the power of a DC motor is given as

$$P_{out} = T_p\omega\left(1 - \frac{\omega}{\omega_p}\right) \tag{1.27}$$

This indicates that $P_{out} = 0$ at both $\omega = 0$ and $\omega = \omega_p$. Differentiating P_{out} with respect to ω and letting $dP_{out}/d\omega = 0$, the maximum power output is found to occur at $\omega = \omega_p/2$ where $T = T_p/2$. However, at this operating point, the motor efficiency is rather low, causing higher power losses and higher temperature rises. For optimal continuous performance, the operating speed may be set between 70% and 90% of the no load speed and the operating torque between 10% and 30% of the stalled torque [1.31]. The optimum operating zone is shown in Figure 1.13 as a small triangle.

With the two pairs of parameters (torque and speed), two motor operation zones can be identified: the intermittent duty zone and the continuous duty zone. In the intermittent duty zone, the motor operates intermittently with a higher torque. This is especially useful when a rotor requires frequent starts and frequent reversals in rotating direction. In such cases, extra torque is required to overcome the inertia of the load as well as the rotor itself. However, it is specially noted that in the intermittent duty zone, the motor can only produce torque and speed for a limited amount of time; otherwise, it will cause the motor overheating. As the combination of torque and speed produced by the motor falls in the continuous operation zone, the motor can be loaded until the rated torque remains constant for a speed range up to the rated speed. In this zone, the motor can run as long as needed without any chance of overheating. The slope of the torque–speed curve represents inherent motor damping. For DC motors, damping is a constant.

The torque–speed characteristics of a typical AC IM are displayed in Figure 1.14. When the motor is initially started from standstill at the starting current I_s, the starting torque (also called LRT) is produced by the motor to overcome the inertia of the motor drive system. The starting torque depends on the terminal voltage and the stator and rotor design. As the motor accelerates, the torque generated by the motor may drop slightly to the local minimum point known as the pull-up torque. In the case that the pull-up torque of the motor is less than that required by its application load, the motor will overheat and eventually stall. Then, a further increase in motor speed will lead to the increase in torque until it reaches the breakdown torque, which is the highest torque the motor can attain without stalling. Starting from this point, the continuous increase in speed causes the sharp

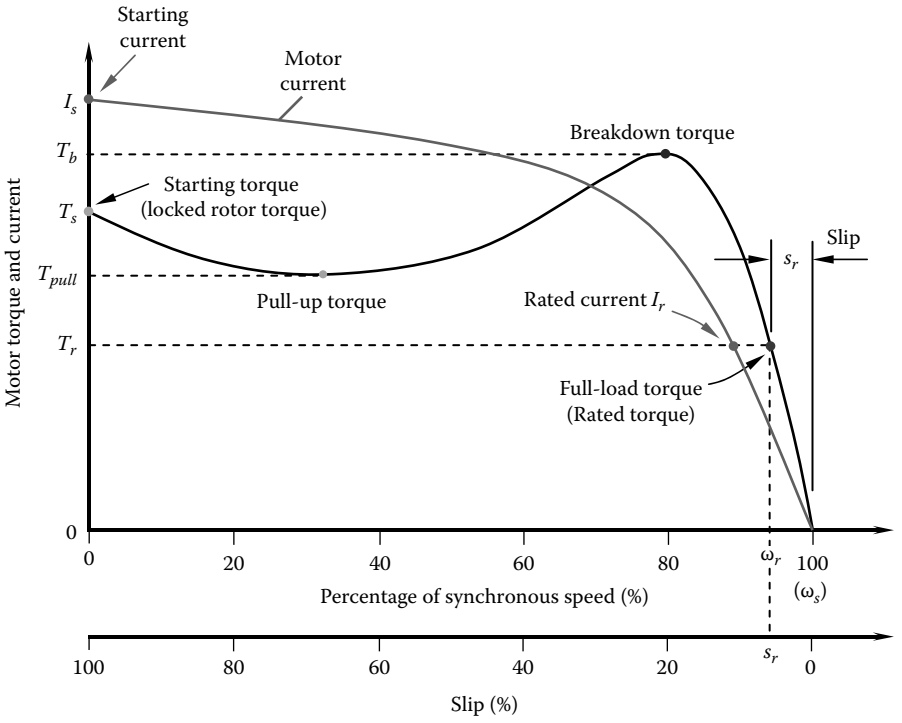


FIGURE 1.14
Torque–speed characteristics of a typical AC IM.

decrease in torque, as well as in motor current. When the motor reaches its full operation speed ω_r, it is loaded to its full-load torque T_r and the corresponding rated current I_r and slip s. At the synchronous speed, no torque can be developed as zero slip ($s=0$) implies no induced rotor current ($I=0$) and thus no torque ($T=0$). This situation only occurs for motors that run while not connected to a load. Therefore, in the strict sense, an IM can never reach the synchronous speed.

In order to clearly present the relationship between rotor torque and speed, an absolute value of torque (usually N-m) is used in Figure 1.14. More frequently, torque is expressed in terms of a percentage of full-load torque, together with speed in terms of a percentage of synchronous speed.

The torque–speed curve for a servomotor is given in Figure 1.15. The whole working zone is defined by the peak torque line $T=T_p$, the peak speed line $\omega=\omega_p$, and a diagonal line equation

$$T = T_p - \left(T_p - T_{mc}\right)\frac{\omega - \omega_k}{\omega_p - \omega_k} \tag{1.28}$$

where T_{ms} is the peak torque at the motor maximum speed ω_p and ω_k is the speed at the knee in the peak envelop.

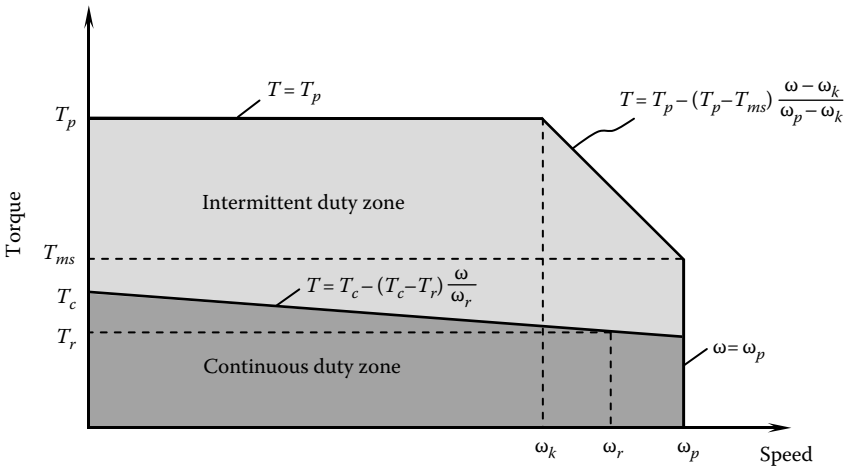


FIGURE 1.15
Torque–speed characteristics of a servomotor.

The two duty zones, that is, the intermittent duty zone and the continuous duty zone, are separated by the maximum continuous torque line, which is expressed as

$$T = T_c - \left(T_c - T_r\right)\frac{\omega}{\omega_r} \tag{1.29}$$

At normal operation, a motor runs at the continuous duty zone to provide a continuous torque to drive external loads.

Unlike DC motors, motor damping in AC motors always vary along with the motor speed.

1.2.6 Mechanical Resonance and Resonant Frequency

Mechanical resonance occurs when an external source amplifies the vibration level of a mass or structure at its natural frequency. For a rotating mass like a motor or a pump, this occurs at what is called the *critical speed*.

Every mechanical system can be resonated at a certain frequency, defined as resonant frequency. However, when two systems (e.g., a motor and a driven machine) are connected together by some power transmission components such as shaft coupling, gearboxes, and belts, it forms a new resonant frequency, based on all components in the system.

As illustrated in Figure 1.16, in a regular motor drive system, the total system inertia J_t is the sum of the motor inertia J_m and the reflected load inertia J_{rl}:

$$J_t = J_m + J_{rl} = J_m + J_c + J_{i,gb} + \frac{J_{o,gb} + J_l}{\gamma_g^2} \tag{1.30}$$

The unit of inertia is kg-m^2 in the SI system and lb$_m$-ft^2 in the English system. It is worth to note that 1 kg-m^2 = 1 N-m-s^2 and 1 lb$_m$-ft^2 = 1/32.185 lb$_f$-ft-s^2.

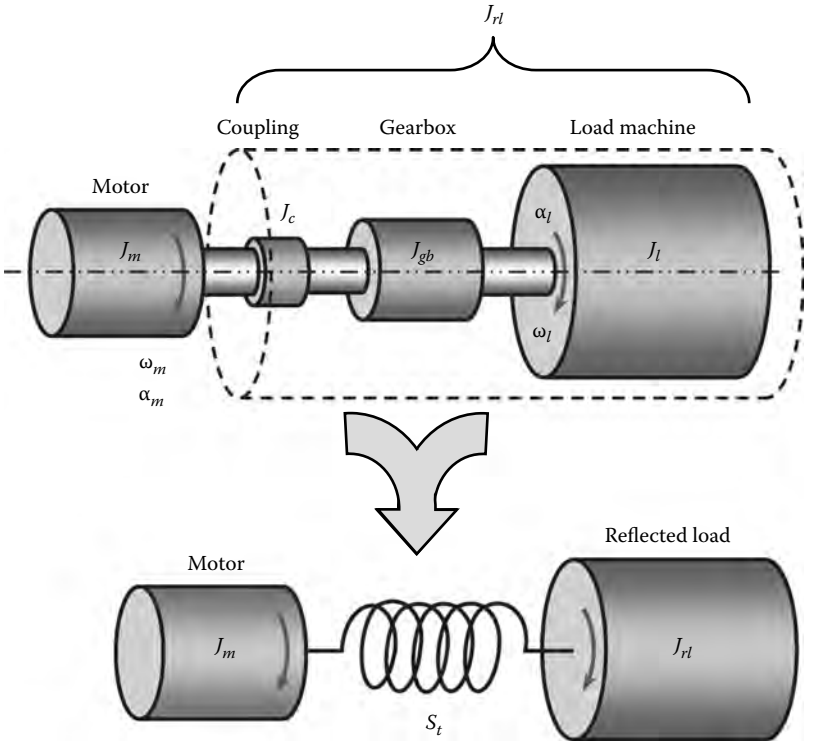


FIGURE 1.16
The equivalent motor-load model, consisting of two masses connected by a massless torsionally elastic spring.

Because the motor-load system is not rigid (i.e., the stiffness is not infinitely large), as the motor torque is applied on the system, each of these connecting components twists slightly like a torsional spring.

To calculate the resonant frequency, motor and load inertias must be distinguished from each other. As discussed previously, load inertia is the inertia of the load reflected to the motor shaft J_{rl}. It is the lumped inertia of the shaft coupling, gearbox, and driven load machine:

$$J_{rl} = J_c + J_{i,gb} + \frac{J_{o,gb} + J_l}{\gamma_g^2} \tag{1.31}$$

The total motor inertia J_m is the lumped inertia of the rotor (including the rotor core and shaft), bearing, and other rotating components:

$$J_m = J_{rotor} + J_{bearing} + \sum J_{other} \tag{1.32}$$

The effects of interference fit on the shaft stiffness can vary depending on the actual interference fit. For a very tightly fitted rotor, the rotor assembly can be viewed as one body with variable outer diameters (ODs) along its axis. Thus, the rotor core virtually increases

the shaft stiffness. On the contrary, if the rotor is loosely coupled with the driven machine, its compliance can contribute to resonance.

For a rotor with a hollow shaft, the torsional stiffness (or spring constant) at each segment is calculated as

$$S_{t,rotor,i} = \frac{\pi\left(d_{out,i}^4 - d_{in,i}^4\right)G}{32l_i} \tag{1.33}$$

where

$d_{out,i}$ and $d_{in,i}$ are the outer and inner diameter (ID) at ith segment, respectively
l_i is the length of the ith segment
G is the shear modulus of elasticity

It is noted that the stiffness of the motor shaft can be influenced by the various types of construction. In general, the machined or welded webs on the motor shaft can add significant stiffness (typically 10%–40% over the base shaft diameter stiffness), while keyed on laminations typically add minimal stiffness. A finite element analysis (FEA) has confirmed that with six welded spider arms on an IM shaft, the equivalent diameter is 7% larger than the base shaft diameter, which corresponded to a 33% increase in torsional stiffness [1.32].

Similarly, the torsional stiffness of the load system at each segment becomes

$$S_{t,oad,i} = \frac{\pi\left(d_{out,i}^4 - d_{in,i}^4\right)G}{32l_i} \tag{1.34}$$

For a solid shaft, $d_{in,i}=0$.

Thus, the total torsional stiffness (spring constant) for the rotor and load machine are given as, respectively,

$$S_{t,rotor} = \frac{1}{\sum\left(1/S_{t,rotor,i}\right)} \tag{1.35a}$$

$$S_{t,load} = \frac{1}{\sum\left(1/S_{t,load,i}\right)} \tag{1.35b}$$

For two geared shafts with a gear ratio γ_g (where $\gamma_g=\omega_i/\omega_o$), the torsional stiffness is given as [1.33]

$$\frac{1}{S_{t,gb}} = \frac{1}{S_{t,i}} + \frac{1}{S_{t,o}/\gamma_g^2} \tag{1.36}$$

It follows that

$$S_{t,gb} = \frac{S_{t,i}S_{t,o}}{S_{t,i}\gamma_g^2 + S_{t,o}} \tag{1.37}$$

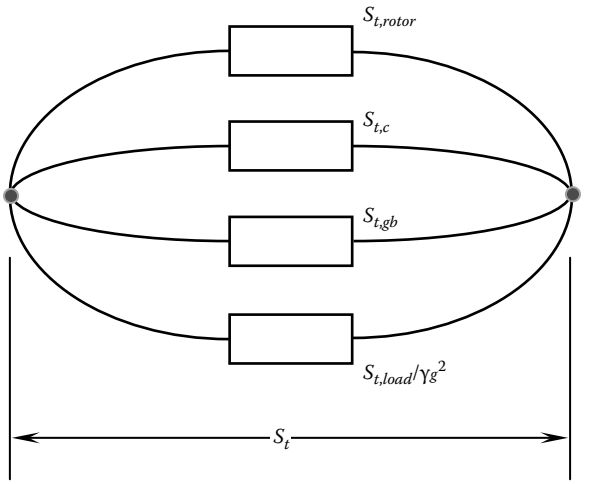


FIGURE 1.17
Taking advantage of the analogy between mechanical and electric systems to calculate the total torsional stiffness S_t.

Utilizing the analogy between the mechanical and electric systems, an electric circuit is developed for the total torsional stiffness S_t, as shown in Figure 1.17. Thus, S_t, which is measured in newton-meter per radians (N-m/rad), can be determined as

$$S_t = \frac{1}{\frac{1}{S_{t,rotor}} + \frac{1}{S_{t,c}} + \frac{1}{S_{t,gb}} + \frac{1}{S_{t,load}/\gamma_g^2}} \tag{1.38}$$

Resonant frequency (Hz) is thus calculated as

$$f_R = \frac{1}{2\pi}\sqrt{S_t\left(\frac{1}{J_m} + \frac{1}{J_{rl}}\right)} = \frac{1}{2\pi}\sqrt{\frac{S_t\left(J_m + J_{rl}\right)}{J_m J_{rl}}} \tag{1.39}$$

This shows that J_m and J_{rl} have the same effect on the resonant frequency. From this equation, it can be seen that there are four ways to increase the system resonant frequency:

- Make the motor shaft more rigid—this increases the stiffness and, in turn, increases the overall torsional stiffness S_t.
- Reduce motor inertia—this actually increases the ratio of load-to-motor inertia, but it also results in higher performance of servo system.
- Reduce reflected load inertia.
- Increase stiffness of components attached/mounted to shafts.

In a direct drive system, a direct drive rotary (DDR) motor is connected with a load without using a gearbox (Figure 1.18). The coupling inertia is considered as a part of the load

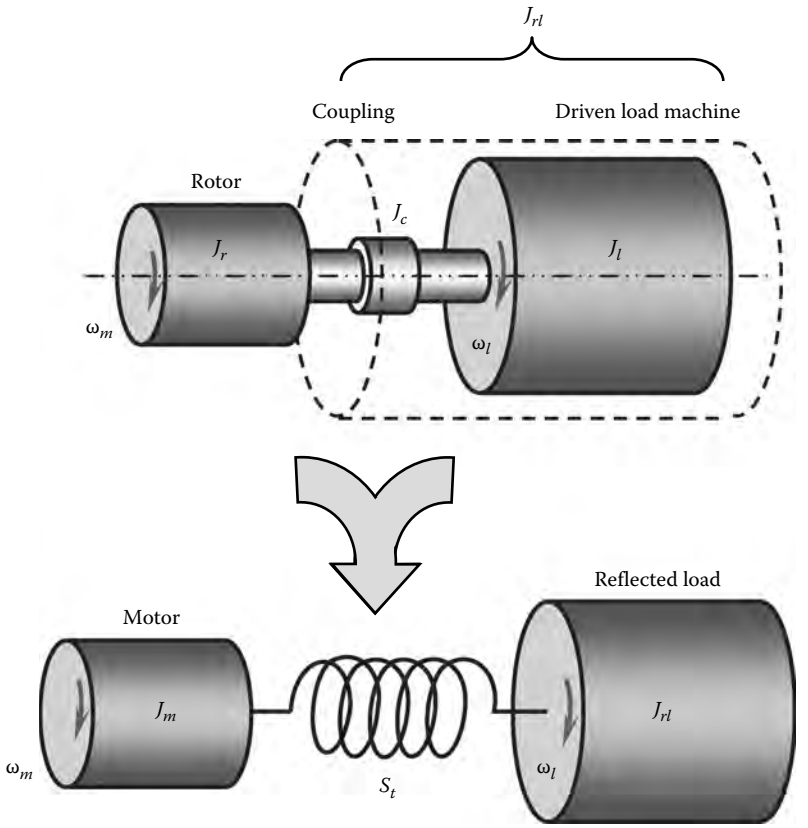


FIGURE 1.18
The equivalent motor-load model for direct drive system.

inertia. Therefore, the system model can be reduced to two lumped masses; the resonant frequency of the system is given by

$$f_R = \frac{1}{2\pi}\sqrt{\frac{S_t\left(J_m + J_{rl}\right)}{J_m J_{rl}}} = \frac{1}{2\pi}\sqrt{S_t\frac{J_m + \left(J_c + J_l\right)}{J_m\left(J_c + J_l\right)}} \tag{1.40}$$

The torsional stiffness is

$$S_t = \frac{1}{\dfrac{1}{S_{t,rotor}} + \dfrac{1}{S_{t,c}} + \dfrac{1}{S_{t,load}}} \tag{1.41}$$

For a three-mass system shown in Figure 1.19, the resonant frequencies are [1.33]

$$f_R = \frac{1}{2\pi}\sqrt{A \pm \left(A^2 - B\right)^{1/2}} \tag{1.42}$$

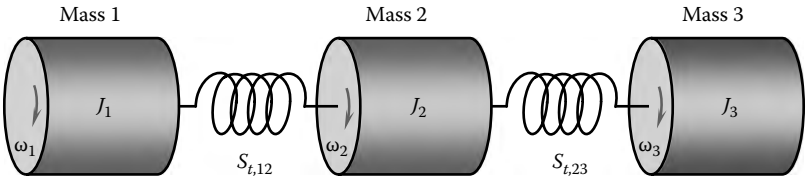


FIGURE 1.19
Three-mass system with massless torsionally elastic springs between them.

where

$$A = \frac{S_{t,12}(J_1 + J_2)}{2J_1J_2} + \frac{S_{t,23}(J_1 + J_2)}{2J_1J_2}$$

$$B = \frac{(J_1 + J_2 + J_3)S_{t,12}S_{t,23}}{J_1J_2J_3}$$

The notations $S_{t,12}$ and $S_{t,23}$ indicate that the torsional stiffness applies to the shaft between rotors 1 and 2 and rotors 2 and 3, respectively.

1.2.7 Load-to-Motor Inertia Ratio

Load-to-motor inertia ratio is defined as

$$\xi = \frac{J_l}{J_m} \tag{1.43}$$

The mismatch of the load-to-motor inertia is very important for the performance of electric motors. The importance of the inertia mismatch is relative to how responsive the system needs to be and how stiff the system is. In designing motion systems, the generally used rule of thumb is that the motor inertia should match the load inertia, that is, a ratio of 1:1 between load and motor inertia would be the ideal scenario. Regularly, the load-to-motor inertia ratio should stay within 10:1. Lower load-to-motor inertia ratios improve motor response, reduce mechanical resonance, and minimize power dissipation. An inertia mismatch of greater than 10:1 may cause motor speed oscillations (Figure 1.20), produce less than optimal response, waste power, and reduce system bandwidth.

In terms of efficiency, a 1:1 ratio between load and motor inertia provides the optimum power transfer. However, a 1:1 ratio is rarely useful in an actual application, because it requires an oversized and unnecessarily expensive motor. Furthermore, it wastes energy and may not perform to specifications [1.8]. In a typical servo system with a *stiff* coupling methodology, a load-to-motor inertial mismatch of 5:1 is generally accomplished.

However, in some cases, the inertia ratio for servomotors can be much higher than 5:1. As an example, direct drive elevator PM traction machines can have 40:1 inertia mismatches. More recently, Kollmorgen has developed a new generation of drive with digital biquadratic filters, enabling servomotors to be successfully applied to medical imaging gantry applications with high inertia mismatches up to 1000:1 [1.34]. To overcome the challenges of inertia mismatch, DDR motors, which directly couple to the load, have been developed in last two decades. DDR motors do not require a high degree of responsiveness and thus

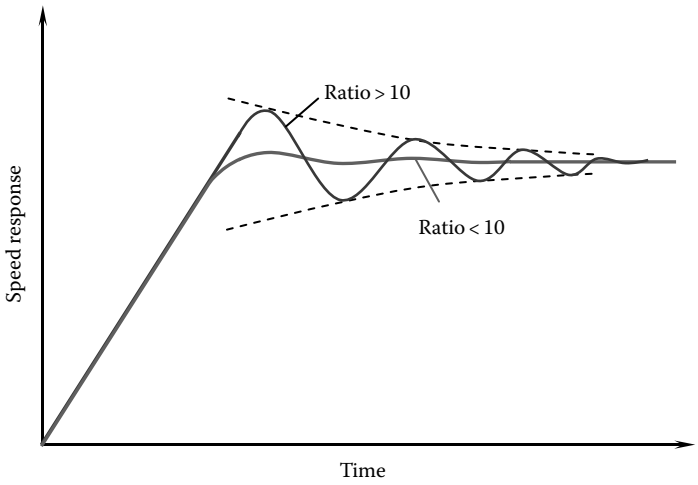


FIGURE 1.20
The effect of load-to-motor inertia ratio on motor speed response. For ratio >10, the speed response curve oscillates and asymptotically approaches to its final value in a long period of time.

significantly reduce inertia mismatch concerns. Directly coupled motors have been successfully tuned to ratios as high as 1600:1 [1.35].

Some servomotor applications require larger inertia rotors to achieve good system controllability. This is due to mass or inertia acting as mechanical filters to load disturbances. The easiest way is to attach an inertia disk or wheel to the shaft [1.36]. If possible, it is preferred to arrange the inertia disk inside the motor or have a larger diameter rotor in the same frame size. Otherwise, the inertia disk has to be arranged outside of the motor (Figure 1.21). As an example, by adding an inertia wheel or wheels to servomotors, Kollmorgen's engineers increase the motor inertia by a factor of 7. Thus, low-inertial motors that have a 14:1 inertia ratio can be changed to medium-inertia motors with a 2:1 inertia ratio. Correspondingly, the resonant frequency decreases from almost 4 times the antiresonant frequency to about 1.6 times the antiresonant frequency.

1.2.8 Duty Cycle

In the power industry, duty cycle is a measure of the fraction of time that a power device is in an active state, that is, $t_{on}/(t_{on}+t_{off})$. For electric motors, duty cycle is defined as the ratio the motor produces rated continuous power divided by the total elapsed time. In fact, duty cycle is a variation of load over a given period of time. The load variation may have a repetitive pattern or a fluctuating pattern.

Duty cycle is used to determine the acceptable level of running time so that the rated motor temperature is not exceeded. For a fixed repetitive load pattern, duty cycle is determined as the ratio of on-time to total cycle period. When operating cycle is such that electric motors operate at idle or a reduced load for more than 25% of the time, duty cycle becomes a factor in sizing electric motors. Also, energy required to start electric motors (i.e., accelerating the inertia of the electric motor as well as the driven load) is much higher than for steady-state operation, so frequent starting could overheat the electric motor.

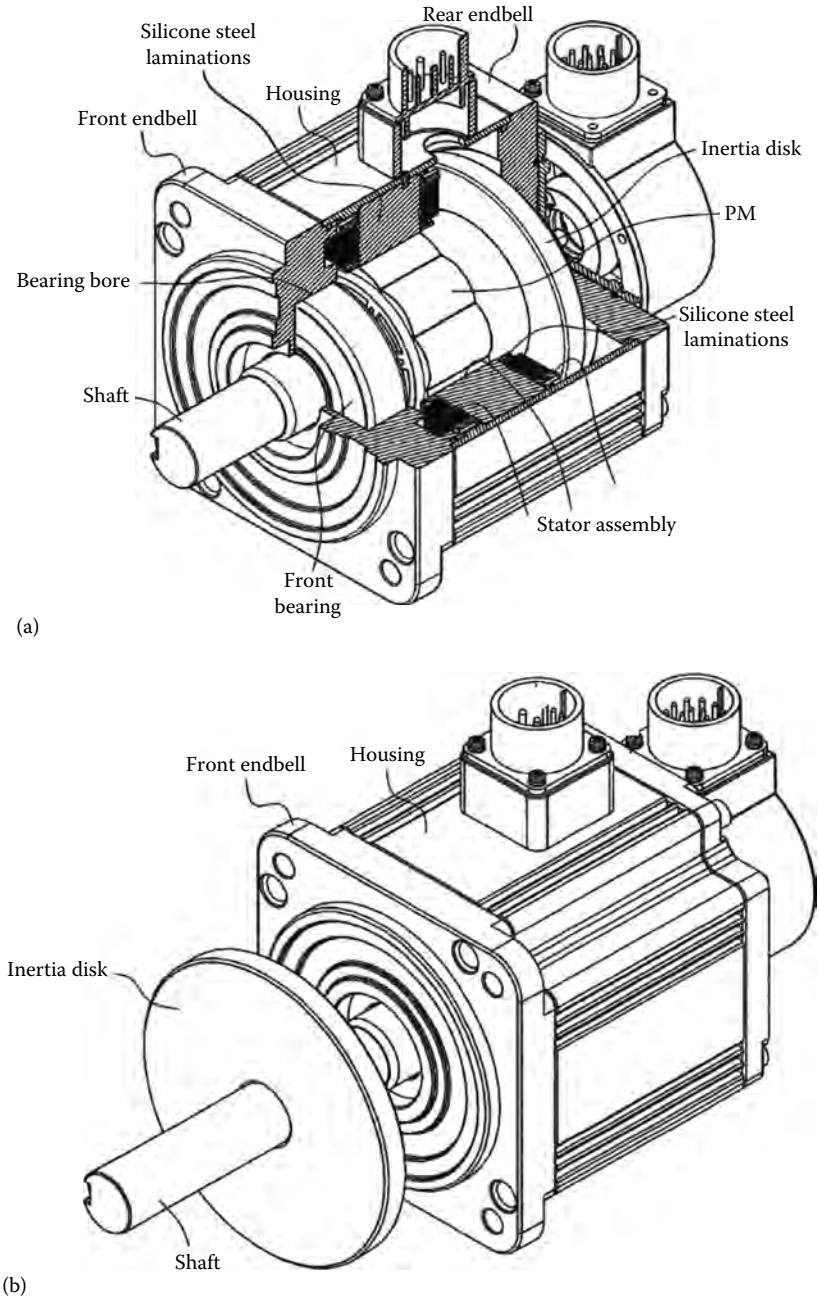


FIGURE 1.21
Addition of an inertia disk either inside (a) or outside (b) of a motor to increase the rotor inertia (U.S. Patent 7,911,095) [1.36]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 62

According to the applications of electric motors, a running load on an electric motor can be either steady or variable (e.g., follows a repetitive cycle of load variation or has pulsating torque shocks). For instance, electric motors in ventilating fans or blowers run continuously over an extensive period of time with almost constant loads. By contrast, electric motors in electric vehicle systems (EVSs) have wide variations in running loads. Elevator machines have typical duties of 120, 180, and 240 starts per hour ratings but may only run at those rates for a few hours a day during heavy traffic time. The temperature variations of electric motors under different operation conditions are presented in Figure 1.22.

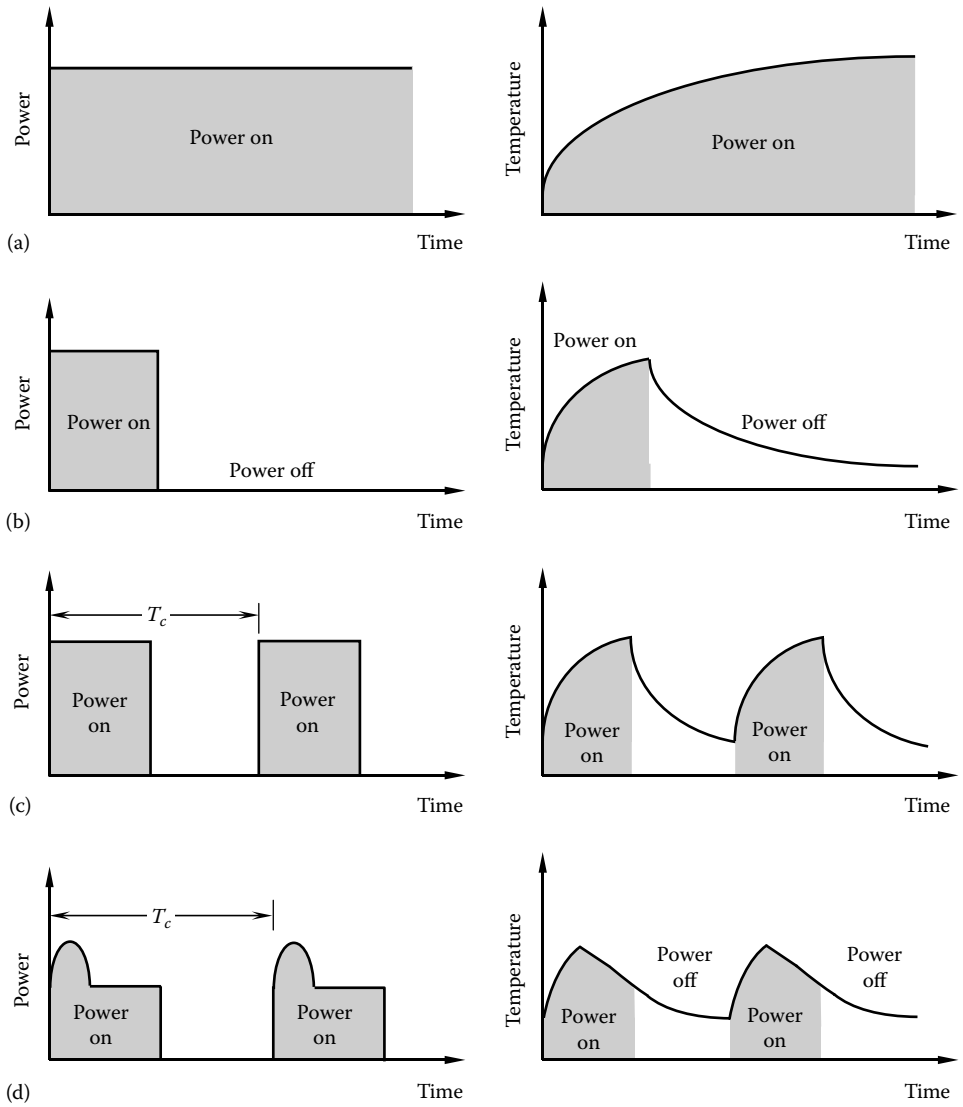


FIGURE 1.22
Temperature variations under different operation conditions: (a) S1—continuous operation; (b) S2—short time operation; (c) S3—intermittent periodic operation, where the start current has no impact on temperature rise; and (d) S4—intermittent periodic operation, where the start current has an impact on temperature rise.

TABLE 1.1

Operating Conditions of Electric Motors

Class	Motor Duty	Note
S1	Continuous duty	The motor operates at a constant load over an extensive period of time to reach temperature equilibrium.
S2	Short-time duty	The motor operates at a constant load over a period of time not sufficient to reach the thermal equilibrium. The rest periods are long enough for the motor to cool down.
S3	Intermittent periodic duty	The motor operates with repeated cycles consisting of a constant output power period followed by an off period. Temperature equilibrium is never reached. Starting current has little effect on temperature rise.
S4	Intermittent periodic duty with starting	Sequential, identical start, run, and rest cycles with constant load. Temperature equilibrium is not reached, but starting current affects temperature rise.
S5	Intermittent periodic duty with electric braking	Sequence of identical duty cycles—starting, operation, braking, and rest. Thermal equilibrium is not reached.
S6	Continuous operation with intermittent load	Sequential, identical cycles of running with constant load and running with no load. No rest periods.
S7	Continuous operation periodic duty with electric braking	Sequential identical cycles of starting, running at constant load, and electric braking. No rest periods.
S8	Continuous operation with periodic changes in load and speed	Series of identical repeating duty cycles, where within each cycle the motor operates at several different load levels and speed. There is no stopped time and thermal equilibrium is not reached.
S9	Duty with nonperiodic load and speed variations	Load and speed vary periodically within the permissible operating range. Frequent overloading may occur.
S10	Duty with discrete constant loads and speeds	Duty with discrete number of load/speed combinations, with these maintained long enough to reach thermal equilibrium.

The International Electrotechnical Commission (*IEC*) defines ten duty cycle designations to describe operation conditions of electric motors, denoted S1–S10, as shown in Table 1.1 [1.37].

1.2.9 Motor Efficiency

The fast rising energy demands all over the world and continuously increasing energy costs have created motivations for many developed and developing countries to focus on energy saving and consumption reduction. Improvements in energy efficiency are most often achieved by producing more efficient machines or adopting more advanced technologies. Because electric motors consume a significant amount of electric energy, motor manufacturers in recent years have committed to the development of more efficient electric motors to save energy.

1.2.9.1 Definition of Motor Efficiency

Motor efficiency is a measure of how effectively a motor converts electrical energy into mechanical energy. Given as a percentage, motor efficiency is defined as the ratio of the power output P_{out} to the power input P_{in}. Since the power output P_{out} represents the difference between the power input P_{in} and the variety of power losses $\sum P_{loss}$ and the

power input P_{in} represents the summation of the power output P_{out} and the variety of power losses $\sum P_{loss}$, motor efficiency can be expressed in several different forms:

$$\eta = \frac{P_{out}}{P_{in}} = \frac{P_{in} - \sum P_{loss}}{P_{in}} = \frac{P_{out}}{P_{out} + \sum P_{loss}} \tag{1.44}$$

This equation shows that in order to increase motor efficiency, it must minimize various power losses in motor, such as copper losses, iron losses, mechanical losses, and windage losses. Thus, the efficiency of electric motors can be boosted by the following [1.38]:

1. Reducing the copper losses in the motor windings. This can be done by increasing the cross-sectional area of the conductor or by improving the winding technique to reduce the winding length, especially at the end turns.
2. Using better materials for lowering eddy-current-related power losses.
3. Taking advantage of advanced nanotechnology to fabricate thinner laminations.
4. Improving the aerodynamics of motor ventilating system for reducing motor windage losses.
5. Applying more efficient cooling methods in motor cooling. (6) Improving manufacturing tolerances.

For electric motors, high efficiency and high torque density are two of the most desirable features. Today, motor efficiency is normally in the range of 80%–95%. Usually, larger motors with higher power output have higher efficiency than smaller motors. High-efficiency motors can provide significant benefits, including reductions in energy consumption and carbon emissions over their entire life cycles. Because operating costs comprise the majority of lifetime motor costs, even a 1% gain in efficiency can make a big difference to costs.

1.2.9.2 IEC Standards on Efficiency Classes of AC Electric Motors

In 2008, IEC published the standard IEC 60034-30: 2008 [1.39] on efficiency classes of AC electric motors. The scope of this standard covers almost all motors (e.g., standard, hazardous area, marine, and brake motors) but excluded motors made solely for converter operation and completely integrated into a machine:

- Single speed, three phase, and 50 and 60 Hz
- 2, 4, or 6 pole
- Rated output from 0.75 to 375 kW (1–500 hp)
- Duty type S1 (continuous duty) or S3 (intermittent periodic duty) with a rated cyclic duration factor of 80% or higher

This standard defines the requirements for the efficiency classes and aims to create a basis for international consistency. The international efficiency (IE) classes IE1, IE2, and IE3 defined in this standard are based on test methods specified in IEC 60034-2-1 [1.40]. It is noted that the methods with this standard determine efficiency values more accurately than the methods previously used.

Hesai Exhibit 1031, page 65

TABLE 1.2

Electric Motor Efficiency Level, Class, Testing Standard, and Regulation over Time

Efficiency Level	Efficiency Class Standard	Test Standard	Regulation over Time
Standard efficiency	IE1 IEC 60034-30	IEC 60034-2-1 Medium uncertainty	China Taiwan 2003 Switzerland 2010
High efficiency	IE2 IEC 60034-30	IEC 60034-2-1	Australia 2006 Brazil 2009 Canada 1997 China 2009 Europe 2011 Korea 2008 Switzerland 2011
Premier efficiency	IE3 IEC 60034-30	IEC 60034-2-1 Low uncertainty	Europe 2015–2017 United States 2011
Super premier efficiency	IE4 IEC 60034-31		

In order to promote a competitive motor market transformation, a new international standard IEC 60034-31 [1.41] was released in 2010 for the addition of the IE4 motor efficiency level. Since no sufficient market and technological information is available to allow IE4 standardization, this efficiency class is intended to be informative. It is expected that advanced technologies will be developed in the near future that can enable manufacturers to design motors for the IE4 class efficiency levels, while maintaining motor dimensions compatible with the existing motors having lower efficiency classes. The four electric motor efficiency classes, testing standards, and regulation over time are listed in Table 1.2.

A rated efficiency of a motor is a function of its rated output. The comparisons of the efficiencies for all four efficiency classes (IE1, IE2, IE3, and IE4) are presented in Figures 1.23 and 1.24 for 50 Hz and 60 Hz electric motors, respectively.

Furthermore, the comparison of four efficiency classes for 50 Hz and 4-pole motors is given in Figure 1.25.

1.2.10 Motor Insulation

The maximum operating temperature of a motor depends on the selection of motor insulation. As the maximum temperature is determined, the maximum motor load, typically specified as the amount of power the motor can deliver on a continuous basis, can be also determined. Thus, motor insulation needs to be defined at the motor's conceptual design stage.

According to the maximum allowable operating temperature that insulation systems can withstand, electric insulation can be categorized into several classes, characterizing the capability of the insulator to resist aging and failures due to overheating. During motor operation, an insulation material may gradually lose its insulating ability to perform the task. The lifetime of insulation materials depends on thermal, chemical, electrical, and mechanical factors. Among them, thermal and electrical stresses are most important.

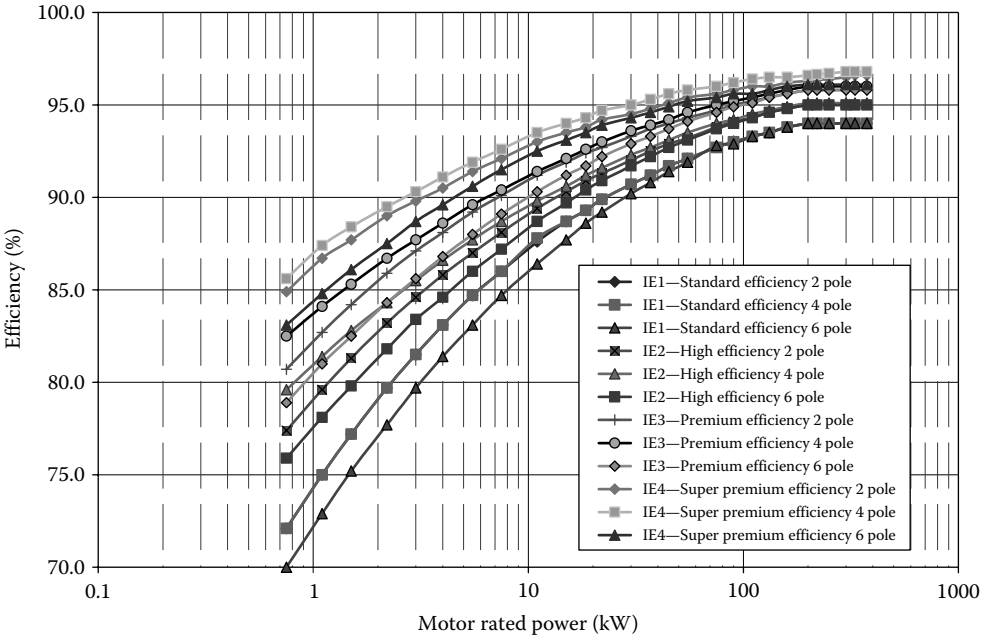


FIGURE 1.23
Comparison of four efficiency classes for electric AC motors with 50 Hz.

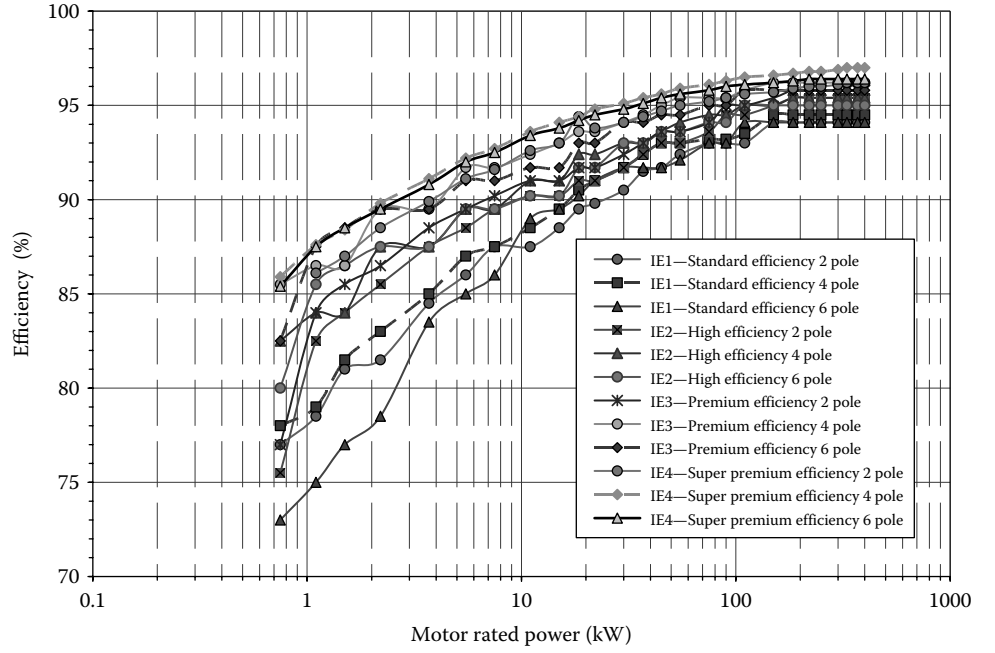


FIGURE 1.24
Comparison of four efficiency classes for electric AC motors with 60 Hz.

Hesai Exhibit 1031, page 67

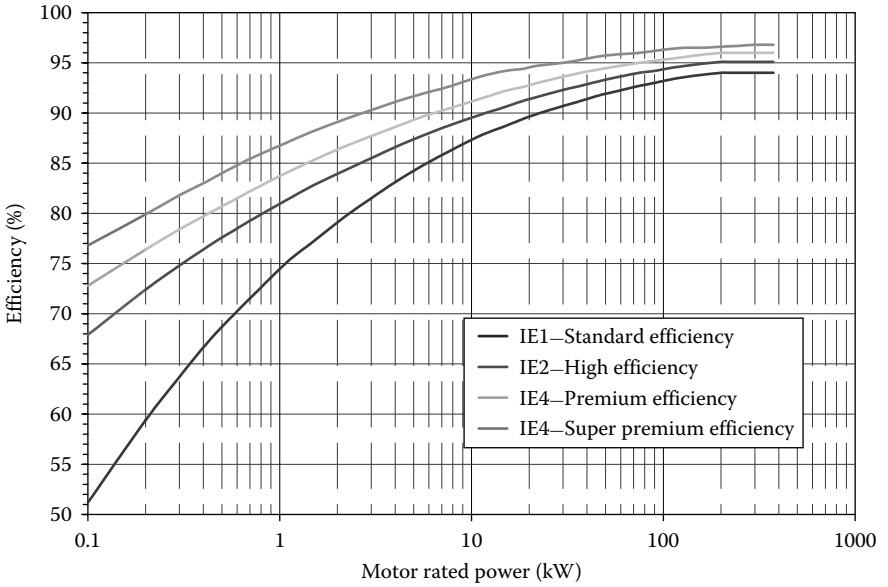


FIGURE 1.25
Comparison of four efficiency classes for 50 Hz, 4 pole electric motors.

In fact, dielectric strength of insulation is very sensitive to temperature aging. Average insulation lifetime decreases rapidly with the increase in motor internal temperatures. A generally accepted rule of thumb is that each 10°C rise above the rating temperature may reduce the motor lifetime by one-half.

There are generally five specialized insulation elements used in an electric motor, including turn-to-turn insulation between separate wires in each coil, phase-to-phase insulation between adjacent coils in different phase groups, phase-to-ground insulation between windings and the electrical ground, slot wedge to hold conductors firmly in the slot, and impregnation to bring all the other components together and fill in the air space.

Dielectric strength refers to the maximum electric field (in V/m in the SI system, and V/mil in English system, where 1 mil = 0.001 in.) that a material can withstand without breaking down and losing its insulating capabilities. The dielectric strength of a material depends on the specimen thickness, the electrode shape, the rate of the applied voltage increase, the shape of the voltage–time curve, and the medium surrounding the sample (e.g., air, gas, or liquid). It should be noted that in strong electric fields, Ohm's law does not hold for insulation materials that have extremely high electric resistance. The current density increases almost exponentially with the electric field, and at a certain value, the current jumps to very high magnitudes at which a specimen of the material is destroyed [1.42].

Dielectric strength values of some solid insulation materials at room temperature and normal atmospheric pressure are presented in Table 1.3.

Based on an average 20,000 h lifetime, the maximum allowable temperature and insulation materials for each insulation class are listed in Table 1.4. The motor classification is based on the temperature rating of the lowest-rated component in the motor.

Thermal aging is an irreversible, permanent reduction of material properties, defined as deterioration. Many of the insulation failures can be attributed to the fact that the

TABLE 1.3

Dielectric Strength of Some Solid Insulation Materials

Insulation Material	Dielectric Strength ($V/m \times 10^6$)	Reference
Ceramics		
• Porcelain	35–160	[1.43]
• Titanates of Mg, Ca, Sr, Ba, and Pb	20–120	[1.44]
Glasses		
• Fused silica, SiO_2	470–670	[1.43]
• Alkali–silicate glass	200	
Insulating films and tapes		
• Low-density polyethylene film	300	[1.45]
• Poly-*p*-xylylene film	410–590	[1.46]
• Aromatic polymer films		[1.47]
– Kapton H (DuPont)	389–430	
– Ultem	437–565	
– Hostaphan	338–447	
– Amorphous Stabar K2000 (ICI film)	404–422	
– Stabar S100 (ICI film)	353–452	
• Polyetherimide film (26 μm)	486	[1.48]
• Parylene N/D (25 μm) film	275	[1.49]
• Cellulose acetate film	157	[1.49]
• Cellulose triacetate film	157	[1.49]
• Polytetrafluoroethylene film	87–173	[1.49]
• Fluorinated ethylene-propylene copolymer film	157–197	[1.49]
• Ethylene-tetrafluoroethylene film	197	[1.49]
• Ethylene-chlorotrifluoroethylene copolymer film	197	[1.49]
• Polychlorotrifluoroethylene film	197	[1.49]
	118–153.5	[1.49]
Micas		
• Muscovite, ruby, natural	118	
• Phlogopite, amber, natural	118	[1.49]
• Fluorophlogopite, synthetic	118	
Potassium bromide, KBr, crystalline	80	[1.43]
Sodium chloride, NaCl, crystalline	150	[1.43]
Varnish		
• Vacuum-pressure-impregnated baking-type solventless varnish	79.9	
• Epoxy baking-type varnish	90.6	
– Solventless, rigid, low viscosity, one part	82.7	
– Solventless, semiflexible, one part	106.3	[1.49]
– Solventless, semirigid, chemical resistant	181.1	
– Solvable, for hermetic electric motors		
• Polyurethane coating—clear conformal, fast cre	78.7	
– Standard conditions	47.2	
– Immersion conditions		

TABLE 1.3 (continued)

Dielectric Strength of Some Solid Insulation Materials

Insulation Material	Dielectric Strength (V/m × 10^6)	Reference
Various insulations		
• Natural rubber	100–215	[1.43]
• Silicon rubber	26–36	[1.44]
• Aramid paper, calendered	28.7	[1.44]
• Aramid with mica	39.4	[1.44]

TABLE 1.4

Standard Insulation Classes

Insulation NEMA Class [1.50]	Maximum Temperature Rating (°C)	Insulation Materials
Class A	105	Cotton, silk, paper, synthetic fibers, vinyl acetate
Class E[a]	120	Polyurethane, epoxy resins, polyethylene terephthalate, phenolics, alkyds, leatheroid
Class B	130	Shellac, bitumen, silk, mica, polyesters
Class F	155	Epoxy, polyamides, silicone, mica, glass
Class H	180	Silicone elastomers, epoxy, polymides, silicone, mica, glass
Class N	200	Glass fibers, mica, asbestos, Teflon
Class R	220	Glass, silicone, mica, Nomex, Teflon

[a] IEC 60085 Thermal class [1.51].

operation temperature exceeds the temperature limits of insulation materials. In fact, temperature has a strong impact on the lifetime of insulations. As the temperature rises above the normal operating temperature, the lifetime of insulation can be quickly shortened (Figure 1.26). This phenomenon is called thermal aging. As discussed by Bonnett and Soukup [1.52], the increase in temperature may have resulted from various causes, including the following: (1) voltage variation or unbalance (per Bonnett and Soukup, 3.5% voltage unbalance per phase will lead to an increase of winding temperature of 25% in the phase with the highest current), (2) frequent motor starts and stops, (3) improper motor cooling, (4) severe environmental conditions such as high ambient temperature, and (5) motor operation under overloading conditions.

In many motor manufacturers, the reference ambient temperature T_a is assumed to be 40°C. The temperature rise of a motor is referred to as the difference between the measured temperature of the motor winding and the ambient temperature, that is, $\Delta T = T_w - T_a$. However, the standard method of measuring the winding temperature involves taking the ohmic resistance of the winding. This provides the average temperature of the whole winding, including the motor leads, end turns, and wires deep inside the stator slots. Therefore, to reflect the temperature difference within the winding, a so-called hot spot allowance must be added to adjust the allowable temperature rise. The hop spot allowance T_{hs} is usually assumed to be 5°C–15°C, depending on the insulation class. For a specific insulation class, the allowable temperature rise of a motor is determined for preventing motor overheating under all loading conditions (no load, full load, locked rotor, etc.):

$$\Delta T = T_{\max} - (T_a + T_{hs}) \tag{1.45}$$

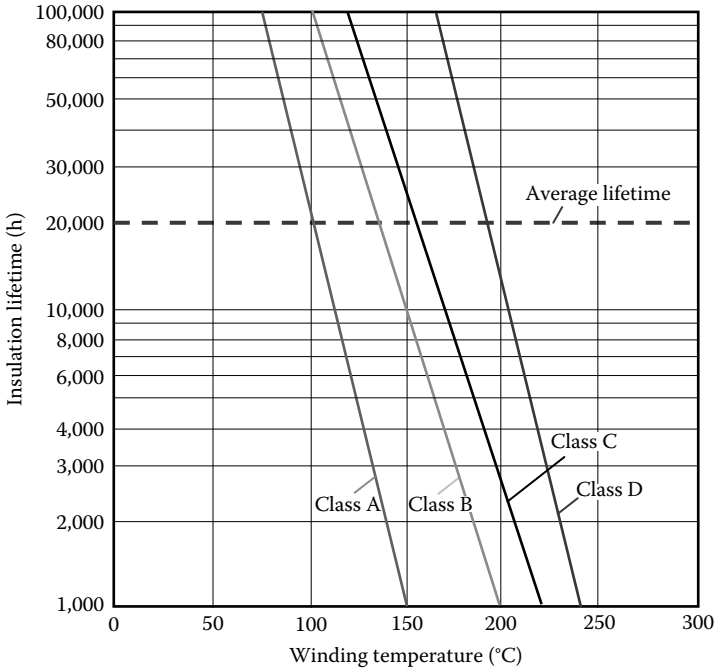


FIGURE 1.26
Thermal aging effect on lifetime of different insulation classes.

1.2.11 Motor Operation Reliability

Motor operation reliability and testing are two key components for design optimization and safe operation of motors. Motor reliability is the measure of the chance that the motor can operate normally over a period of time without failure. In the motor industry, one of the useful parameters is the mean time between failures, defined as the reciprocal of the failure rate (identified as $\ddot{e}$).

Two failure modes dominate the life of motors: bearing failures and winding failures. In all motor failures, bearing failures account the majority. Winding failures generally occur in the early stage of motor running and are attributed to shorts and grounds resulted from assembly quality rather than the long-term insulation degradation. By utilizing more than 2000 actual failure cases of fractional horsepower motors in the existed data failure bank, Wilson and Smith [1.53] developed a mathematical reliability model for each failure mode by means of Weibull cumulative distribution function and regression technique for use in predicting overall motor life and failure rates.

1.3 Classifications of Electric Motors

Electric motors can be classified in a variety of ways according to their operating characteristics, such as the source of electric power, type of rotor winding, type of motion, control pattern, the magnetic flux orientation, structure topology, power rating, and the cooling methods. A brief classification of rotary electric motors is given in Figure 1.27.

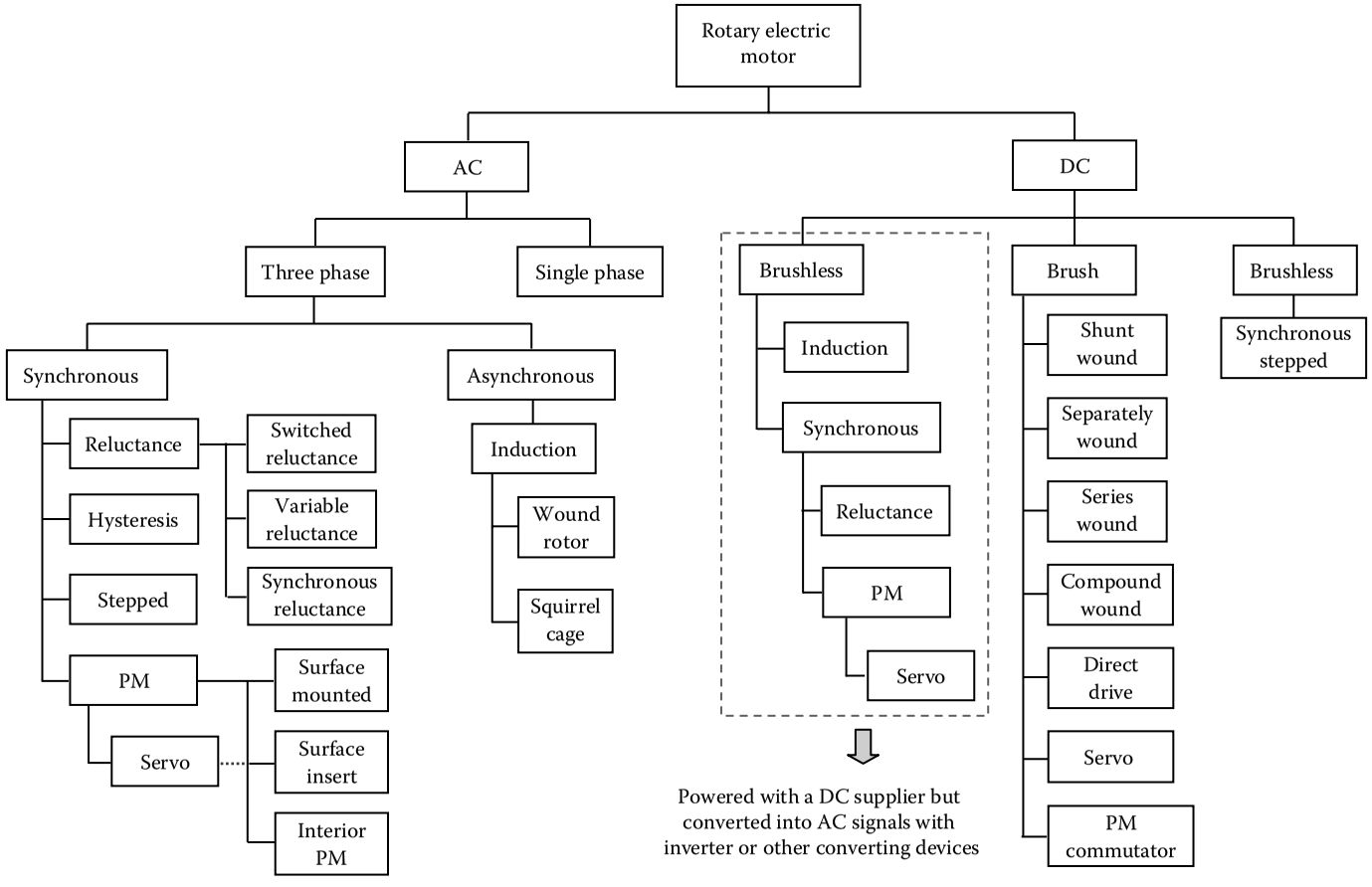


FIGURE 1.27
The classification of rotary electric motors.

Hesai Exhibit 1031, page 72

1.3.1 DC and AC Motors

DC motors are designed to run on DC power. This type of motors was invented much earlier than the type of AC motors. They are often used when high torque at low speed is required. The speed adjustments of DC motors can be as much as 20:1, and they can operate at 5%–7% of the motor's base speed (some can even operate at 0 rpm) [1.54]. There are a number of different types of DC motors:

- Shunt wound motor—In this motor, the rotor and stator windings are connected in parallel. Hence, the current in the rotor and stator windings are independent of one another. The characteristics of a shunt motor provide it very good speed regulation.
- Separately excited motor—In this motor, the rotor and stator winding are connected with different power suppliers. For a long time, it has been the most common configuration used in industrial applications for DC motors with electronic speed control. Due to their excellent controllability and simple operational performance, separately excited motors are extensively used in speed or position control systems.
- Series motor—As its name indicates, the rotor and stator windings in a series motor are connected in series. The torque is proportional to I^2 so it gives the highest torque per current ratio over all other DC wound field motors at economical costs. It is therefore used in starter motors of cars and some old generation elevator motors [1.55].
- Compound motor—As the combination of the series motor and the shunt motor, the motor has the torque characteristics of the series motor and the regulated speed characteristics of the shunt motor. A compound motor comprises an armature winding on the rotor and two field windings on the stator. The stator is connected to the rotor through a compound of shunt and series windings. Compound motors are often adopted to drive loads such as shears, presses, and reciprocating machines.
- Brushed DC PM field motors—Field windings are now replaced by PM materials and the winding has a series of commutator bars with carbon brushes arranged to provide mechanical commutation of the motor. Commutation is the switching of the current direction in the winding at the right timing to produce continuous torque vectors in one direction for continuous rotating speeds of the shaft.
- Universal motors with brushes and commutators have been designed to operate on either AC or DC power. For this type of motor, the stator's windings are connected in series with the rotor windings through a commutator. The advantages of universal motors are high starting torque, compact size, and ability to run at high operating speed. The negative aspects include the short lifetime caused by the commutator and brushes and high noise and vibration. Therefore, this type of motor is often used in applications where the motor only operates intermittently. Universal motors generally run at high speeds, making them suitable in home appliances (e.g., vacuum cleaners and food mixers) and home power tools (e.g., electric drills) where single-phase wall plug in power is abundant.
- Brushless DC motor—This special type of motors has been developed to overcome the limitations and drawbacks of brushed DC motors. Practically, brushless motors are powered by a DC electric source via an integrated inverter/switching power supply, which produces an AC electric signal to drive/commutate the motor. It usually requires sensors and electronic control system for controlling the

inverter output amplitude, waveform, and frequency. Though a number of brushless motors are available (e.g., induction, switched reluctance), typical brushless motors are made with PMs. With built-in PM, the motor does not need a separate excitation winding, resulting in reduced power losses and increased efficiency. The size of the PMDC motor is the smallest (high power density) among all other DC motors. Often, PMDC motors are served as servomotors.

Advantages of brushed DC motors include low initial cost, high reliability, and simple speed and torque control characteristics. Brushless DC motors have evolved from brush DC motors as power electronic devices and became available to provide electronic commutation in place of the mechanical commutation provided by brushes. Brushless motors have increased reliability, longer lifetime, higher efficiency, less maintenance, reduced noise, elimination of ionizing sparks from commutator, and overall reduction of electromagnetic interference.

Today, DC motors are still used in a wide range of applications, especially for those applications requiring precise speed control over a large range around the rated speed, such as steel mills, mines, and electric trains. However, DC motors usually require additional operational elements such as brushes and commutators that transfer electric power to motor armatures. However, commutators usually can cause power ripples and limit the rotor speed and brushes increase the frictional power lower and radiofrequency interference. Furthermore, as brushes wear and tear, carbon dust spreads throughout the motor, causing some operation and performance problems. As a consequence, the maintenance of the interface between the brush and commutator becomes critical for motor operation reliability. In addition, the use of brushed can enhance the motor acoustic noise.

AC motors are driven by AC power sources. By eliminating commutators and brushes, AC motors offer several advantages over DC motors, including increased operation reliability, longer lifetime, higher efficiency, less maintenance, reduced cost, shorter frame size, and simpler motor structure. AC motors are dominant in industrial motion control for cost/performance reasons. As the most common type, single-phase AC motors are mainly used for residential and commercial applications. Three-phase AC motors are especially suitable for high-power applications. In general, single-phase motors operate less efficiently than three-phase motors.

Brushless DC motors are similar to AC synchronous motors. The major difference is that the waveform used to drive AC motors is typically sinusoidal and could come directly from an AC source or could be using the pulse-width modulation (PWM) technique. Therefore, AC synchronous motors develop a sinusoidal back EMF, as compared to a rectangular or trapezoidal back EMF for brushless DC motors.

Motors used in hospital equipment or other patient-care facilities are required to comply with low noise level standards to endorse patient comfort and reduce anxiety. Brushless DC motors are ideal for noise-sensitive environments due to the lack of brushes, which emit audible noise during rotation [1.56].

The disadvantages of AC motors include difficulty of speed control, high control complexity, less torque density, inability for operating at low speeds, induced eddy current and hysteresis power losses in the stator and rotor cores, high cost, and poor positioning control.

1.3.2 Single-Phase and Three-Phase Motors

In most countries, household power is usually single phase due to the low cost of single-phase power distribution. A single-phase motor is run from an AC single-phase power

system. The most standard frequencies of single-phase power systems are either 50 or 60 Hz, although other frequencies may be also available. However, unlike three-phase motors, a single-phase motor is unable to produce the start torque itself; it must be started by some external means such as an auxiliary start winding or a start capacitor. Single-phase motors need some other forcing functions to set direction of rotation (DIR). The auxiliary start winding or start capacitor creates a simulated phase to set the DIR.

Three-phase power is a common form of electric power due to its inherent benefits in high-powered transmission and electric equipment operation. In a three-phase power system, ACs are carried by three circuit conductors with the same frequency and amplitude but different phases. As sinusoidal functions of time, the current at each conductor has shifted 120° in phase from each other. Correspondingly, in a three-phase motor, there are also three windings (separated equally in space by 120°) per pole on stator to produce a rotating magnetic field.

Because of higher efficiencies and favorable torque–current characteristics versus single-phase motors, the three-phase AC motors dominate in almost all industries and consume more than half of all the electricity used in industry. In contrast, three-phase motors are more efficient and compact than single-phase motors of comparable power rating. Three-phase motors have generally lower vibrations and last longer than single-phase motors under the same conditions. As a matter of fact, the effectiveness and low cost of three-phase motors are major reasons for three-phase power to be extensively used in industry.

1.3.3 Induction and Permanent Magnet Motors

Depending upon the method of generation of the magnetic field in the rotor, an AC motor can be classified either a PM AC motor, where the magnetic field of the rotor is directly produced by PMs, or an IM, where the magnetic field of the rotor is produced though the induction effect onto the rotor bars/winding.

In an IM, the AC power supply is connected to the stator winding to generate a rotating magnetic field. Because of this rotating field, the rotor is powered by means of electromagnetic induction. The change in magnetic flux through the rotor induces AC in rotor windings and in turn creates its own magnetic field. The induced current in the rotor gives rise to magnetic forces, which cause the rotor to rotate in the direction defined by the stator rotating magnetic field. In actual operation, the rotor speed always lags the magnetic field speed, which is defined as the synchronous speed, allowing the rotor windings or conducting bars to cut magnetic lines of force and produce useful torque.

According to the rotor structure, IMs can be further divided into two subcategories: (1) squirrel cage motors in which the rotor is made of conductive aluminum/copper bars that are parallel (or have a small skew angle) to the rotor centerline and short-circuited by the end rings (Figure 1.28) and (2) wound rotor motors where windings are made on the rotors (Figure 1.29).

IMs are perhaps the simplest and most rugged motors, which have been extensively used in a variety of applications, from household appliances to heavy industrial equipment, due to their simple structures and low costs. In modern squirrel cage IMs, the conducting bars are formed in the skewed slots distributed axially along the rotor surface by casting. These conducting bars are connected at the ends of the rotor slots as the end rings to form the closed electric circuit.

For IMs, the stator operates with the power supply frequency f ($f=60$ Hz in North America and $f=50$ Hz in most countries of the world) and the rotor winding contains

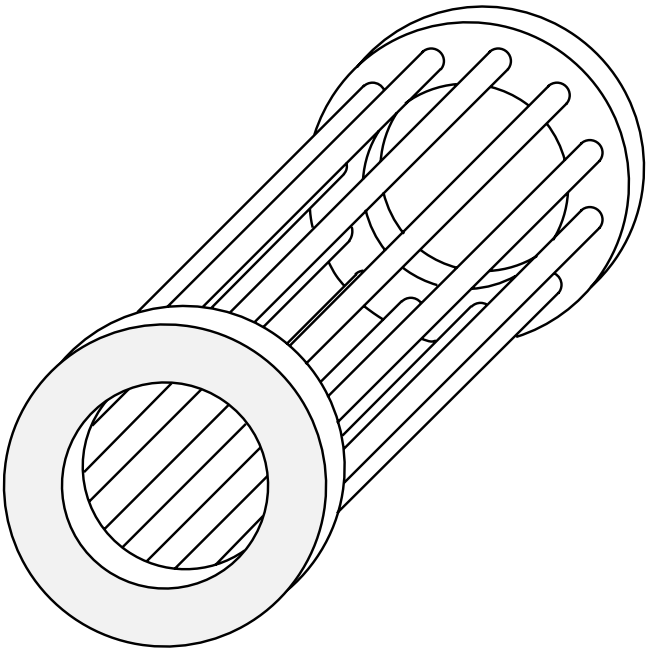


FIGURE 1.28
Conductive bars and end rings in a squirrel cage motor with closed slots.

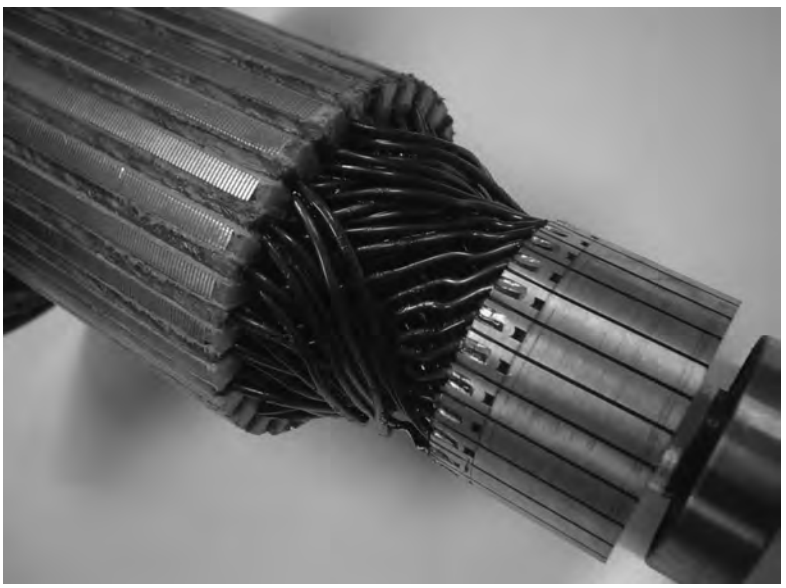


FIGURE 1.29
Rotor winding in a wound IM.

current and voltage with slip frequency f_{slip} or *sf*, where f_{slip} is defined as the frequency corresponding to the slip speed $n_{slip} = (n_s - n_r)$.

A low rotor resistance will result in the current being controlled by the inductive component of the circuit, yielding a high out-of-phase current and a low torque.

PMMs use permanent magnets to generate the required magnetic field and thus separated excitation windings are no longer required. A permanently excited synchronous motor has a sinusoidal back EMF and therefore operates with a sinusoidal voltage. PMMs are designed to provide a wide operation speed and load range with high motor efficiency, usually in excess of 90% from less than 50% speed driving a typical pump or compressor load to a peak range in excess of 97% for high-speed applications [1.57]. The structure of a PMM is shown in Figure 1.30.

PMMs have numerous advantages over IMs. First, the built-in PMs in a PMM eliminate rotor windings that are required in an IM. Hence, it greatly reduces the rotor inertia and power losses and, consequently, increases operational reliability and improves dynamic load response. Second, since the stator current in a PMM is only for torque production and thus magnetizing current through the stator is no longer necessary, a PMM operates at a higher power factor and thus a higher efficiency over an IM for the same power output. Third, the application of rare-earth PMs makes PMMs with high power density and high torque-to-inertia ratio, which lead to fast dynamic response capability. As a matter of fact, rare-earth PMMs have the highest power density of any motor type. This feature is especially desirable for applications in which the motor size is a main consideration. Today, PMMs have become increasingly popular in various industrial and commercial sectors. They are especially ideal in high-accuracy, high-performance motion control applications, for instance, computer numerical control (CNC) machine tools, robots, embedded motion, engraving machines, packaging and printing machines, semiconductor fabrication facilities, medical equipment, and satellite servo systems.

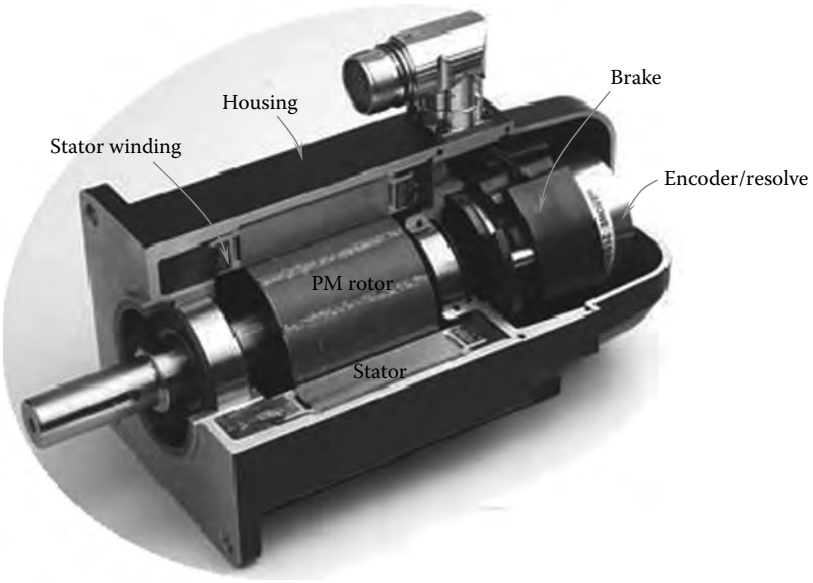


FIGURE 1.30
The typical structure of a PMM. (Courtesy of Kollmorgen Corporation, Radford, VA.)

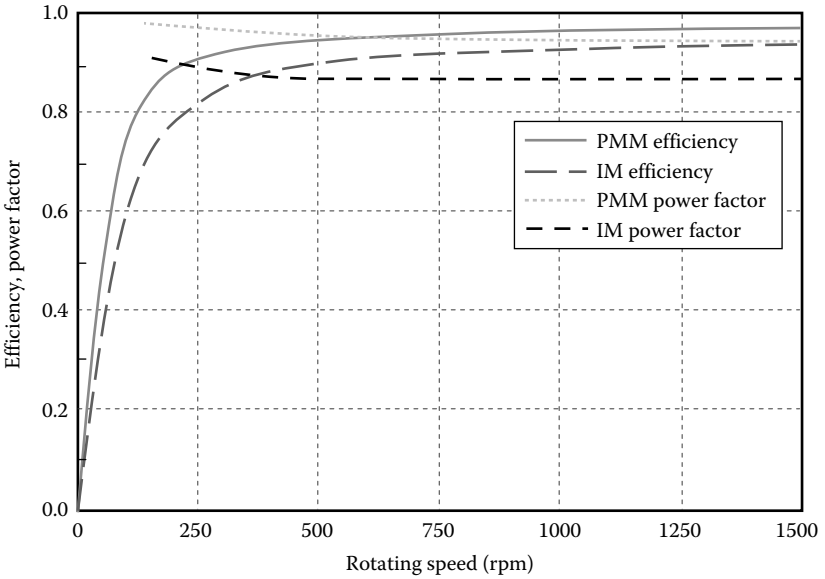


FIGURE 1.31
Comparison of calculated motor efficiency and power factor between a 55 kW, 1500 rpm IM and a PMM under an identical operation condition.

The comparison of motor efficiency and power factor between a 55 kW, 1500 rpm IM and a PMM under an identical operation condition and power rating is given in Figure 1.31 [1.58].

According to the mounting pattern of PMs, PMMs can be further divided as surface mounted, surface insert, core insert, and pole shoe. Surface-mounted PM (SPM) motors have dominated for decades. In recent years, some emerging markets and perhaps the sharp price rise of rare-earth magnets have boosted demand for IPM motors. With advantages such as high-speed performance, robust mechanical structure, low heat generation in rotor, and special designs that can support near-constant power over a broad range of speed, IPM motors provide an excellent solution for applications like traction motors, machine tools, or high-speed rotors.

In the surface-insert method, PMs are buried inside the rotor core. This method can offer some distinct advantages, for instance, the simple and robust rotor structure, high torque density, easy-to-achieve flux-weakening operation, and potentiality of fully sensorless operation [1.59].

Torque produced by an IPM motor is based on two different mechanisms [1.60]: one is the same as an SPM motor; PM torque is generated by the magnetic flux linkage between the PM rotor field and the electromagnetic field of the stator. Another is known as reluctance torque. PMs buried inside a rotor pole piece exhibit high reluctance directly along the magnetic axis due to the low permeability of the PMs and pole pieces between the magnetic poles or magnet barriers, creating inductance saliency and reluctance torque. Thus, IPM motor designs augment PM torque with reluctance torque. As a result, the magnets used in IPM motors can be thinner, achieving significant cost reductions.

1.3.4 Synchronous and Asynchronous Motors

A synchronous motor refers to a three-phase AC motor in which the rotor runs at the same speed of the rotating magnetic field of the stator. The synchronous speed n_s can be expressed in different ways, depending on the unit it adopted:

$$n_s = \frac{f}{p} \quad \text{in rps (Hz)} \tag{1.46a}$$

$$n_s = \frac{60f}{p} \quad \text{in rpm} \tag{1.46b}$$

where
f is the frequency of the AC supply current in Hz
p is the number of magnetic pole pairs per phase

The synchronous speeds of 50 and 60 Hz machines are presented in Figure 1.32 for various magnetic pole pairs.

If N denotes the number of magnetic poles per phase, then it gives that $N = 2p$ and yields

$$n_s = \frac{120f}{N} \quad \text{in rpm} \tag{1.47}$$

Therefore, synchronous speed can be altered by changing either the frequency applied to the motor or the number of magnetic poles. Some multispeed motors adopted external connections that enable to switch the stator poles, for example, from 4 to 6 poles. By using an adjustable frequency drive, motor speed can vary in a large speed range under a constant voltage.

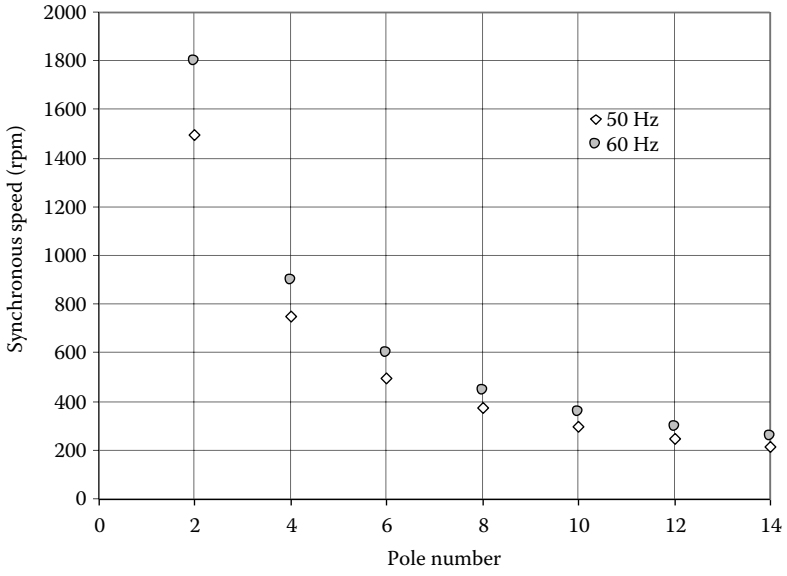


FIGURE 1.32
Synchronous speeds of 50 and 60 Hz machines under various pole numbers.

In the unit of radians per second (rad/s), the angular synchronous speed ω_s is given as

$$\omega_s = \frac{\pi n_s}{30} = \frac{2\pi f}{p} = \frac{4\pi f}{N} \tag{1.48}$$

In contrast, in an IM, the stator windings are wound around the rotor to produce a rotating magnetic field with a three-phase power supply. It is the varying magnetic field that induces currents in the rotor conductors/end rings (or rotor windings). Thus, the interaction between the magnetic fields of the stator and rotor causes a rotational motion of the rotor.

Because current in the rotor conductors/end rings is induced by the stator windings, the rotating speed of the rotor must lag the rotating speed of the stator's magnetic field (i.e., the synchronous speed). This speed difference is called slip, which is defined as

$$s = \frac{n_s - n_r}{n_s} = \frac{\omega_s - \omega_r}{\omega_s} \tag{1.49}$$

where

n_s and n_r are the synchronous speed and rotor speed in rpm

ω_s and ω_r are the angular synchronous speed and angular rotor speed in rad/s, respectively

Thus, the rotor rotating speed of an asynchronous motor becomes

$$n_r = n_s(1-s) \quad \text{in rpm} \tag{1.50a}$$

$$\omega_r = \omega_s(1-s) \quad \text{in rad/s} \tag{1.50b}$$

Slip increases with load. Usually, full-load slip ranges from less than 1%–3% for larger-power motors to 4%–6% for small-power motors.

Synchronous motors can be built with either salient pole rotor or nonsalient pole (cylindrical) rotors, depending on the customers' specifications and applications. Generally, salient pole motors are typically found in high-power, low-speed applications. Motors with cylindrical rotors are typically found in high-power, high-speed applications such as spindle motors in machine tools.

1.3.5 Servo and Stepper Motors

There are two main types of motion control systems: closed loop and open loop. A closed-loop system applies a feedback system to verify whether or not the desired output has been reached. For instance, an encoder is commonly attached with a servomotor to measure the velocity and position of the servomotor for providing the information to the motion controller. Obviously, servomotor systems require the use of the closed-loop system. As a contrast, an open-loop system does not need any feedback for verifying the output. In practice, most step motor systems use open-loop system. The difference between the two control schemes is whether or not the use of feedback system (Figure 1.33).

Hesai Exhibit 1031, page 80

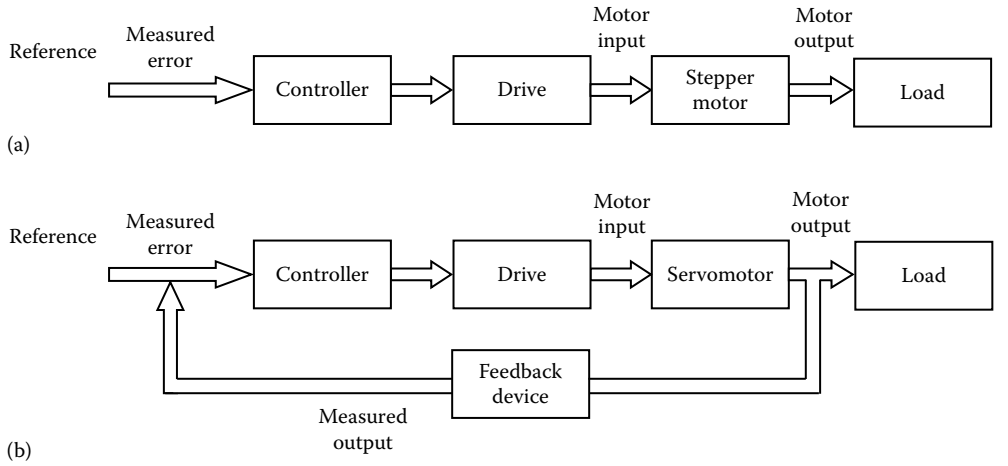


FIGURE 1.33
Comparison of two control schemes: (a) the open control loop and (b) the closed control loop.

An open motion control system usually consists of a motion controller, a feedback system, an electric motor, and a motor drive. The motion controller acts primarily as the brain of the motion control system. Its main function is to make sure the output of the system is as close as possible to the desired result. Based on the information from the feedback system (such as encoder, resolver, sensors), the motion controller sends the electronic signals to the motor drive to adjust motion path or trajectory. The feedback system is to obtain the motor motion information (e.g., displacement and speed), which are then converted into a set of digitized output signals and fed to the motion controller for making necessary adjustments. The motor drive takes the low-power electronic signal from the motion controller and converts it into high-power current/voltage to the motor, which executes the tasks from the drive as an actuator. All these components are integrated to form the complete motion control system to provide the desired movement for various applications.

The word *servo* originally comes from the Latin word *servus*, meaning slave or servant. NEMA [1.61] defined a servomotor as "an electric motor that employs feedback with the purpose of producing mechanical power to perform the desired motion of the servo mechanism." Simply speaking, any control element that employs feedback is a servo.

Today, servomotors are increasingly being used in a variety of applications due to their high power density, high efficiency, and excellent dynamic performance as compared with other motor drive technologies. Servomotors operate with closed-loop control systems. The ability of the servomotor to adjust to differences between the motion profile and feedback signals depends greatly upon the types of control systems and servomotors. There are two types of servomotors: one is classical DC servomotor and another is AC servomotor. Generally, AC servomotors can handle higher current surges compared to DC servomotors. It is to be noted that in some references, AC servomotors are referred to as brushless DC motors, causing confusion to some readers. An AC servomotor or the so-called brushless DC motor is essentially a three-phase AC synchronous motor. It has a position transducer inside the motor to transmit motor shaft position to the drive amplifier for the purpose of controlling current commutation in the three phases of the motor windings.

Hesai Exhibit 1031, page 81

Stepper motors are special motors used in motion control systems. This type of motor has high torque at low speeds, high reliability, low cost, and simple rugged construction that operates in almost any environment. Typical applications include printers, image scanners, CNC machines, and volumetric pumps. Unlike common AC and DC motors rotating continuously, a stepper motor moves in fixed angular increments, called the step angles. In a stepper motor, a full revolution is divided into a large number of discrete steps. In fact, a stepper motor is an actuator that converts electrical pulse signals into angular displacements. Once the stepper motor receives an electrical pulse signal, it moves a fixed step angle in a predefined direction. Thus, by controlling the pulse number, the angular displacement (i.e., the motor positioning) can be controlled precisely. By altering the pulse frequency, the accurate control of the motor rotating speed and acceleration can be achieved. It is important to note that in this control mechanism, no feedback systems are required. This allows the stepper motor to operate in an open-loop system, making the motion control easier and less expensive. In addition, stepper motors can operate at a set speed regardless of load as long as the applied load is less than the limited torque rating.

As an open-loop control system, the stepper motor requires only the input of the current state. The number of steps per revolution varies by model and manufacturer. Stepper motors were introduced in the early of 1960s as an economical replacement to closed-loop DC servo systems. Unlike conventional motors that rotate continuously, stepper motors rotate in fixed angular increments. One revolution of a stepper motor involves taking a number of steps, depending on the number of rotor teeth, motor construction, and type of drive scheme used in the motor control.

Stepper motor technology does have some disadvantages. The most critical drawback is the loss of synchronization and torque if a large load exceeds the motor's capacity. A high-inertia load can cause the rotor to slip, or not advance when the step pulse is given. Consequently, the user typically selects a stepper motor's capability with 2:1 factors of safety (FS) to torque to minimize or eliminate loss of synchronization. Stepper motors also tend to run hot because phase current is independent of the load. In some applications, if the motor needs to be overdriven by the load, it may be undesirable to feel the poles of the stepper motor as the rotor is being pulled by the load [1.62].

Today, stepper motors are available in many topologies and step sizes for various industrial and commercial applications due to their advantages of low cost, high reliability, no cumulative error, high torque at low speeds, simple and rugged construction, and excellent response to start-up, stopping, and reverse operation.

There are basically three types of stepper motors: PM, variable reluctance, and hybrid [1.63]. The PMM has relative low torque and low speed with large step angles. Its simple construction and low cost make it an ideal choice for nonindustrial applications. The variable reluctance motor (VRM) usually has three-phase stator windings. It can achieve large torque outputs. However, due to their large vibrations and high noise emissions, VRMs are gradually out of the main industrial market. The hybrid stepper motor combines the best characteristics of the variable reluctance and PMMs by constructing multistator poles and PMs inside the rotor. Therefore, the hybrid stepper motor is the most widely used stepper motor today. The comparison of three types of stepper motors is listed in Table 1.5.

A cutaway diagram of hybrid stepper motors is demonstrated in Figure 1.34. The stator coils are wound on stator poles. The rotor is a cylindrical PM, magnetized along the axis with radial soft iron teeth. Thus, one pole of the rotor may align with the stator in distinct positions.

The polarity of rotor laminations is shown in Figure 1.35. Unlike conventional motors, the magnet in a stepper motor is magnetized axially rather than radially. As shown in

TABLE 1.5

Comparison of Three Types of Stepper Motors

Stepper Motor Type	Phase	Step Angle	Torque	Vibration and Noise	Note
Permanent magnet	2 or 4	7.5° or 15°	Low	Low vibration at low frequencies, low noise	High efficiency, low current, low heat generation
Variable reluctance	3	1.5°	High	Large	Out of main industrial market
Hybrid	2 5	1.8° 0.72°	High	Low	With the most wide applications

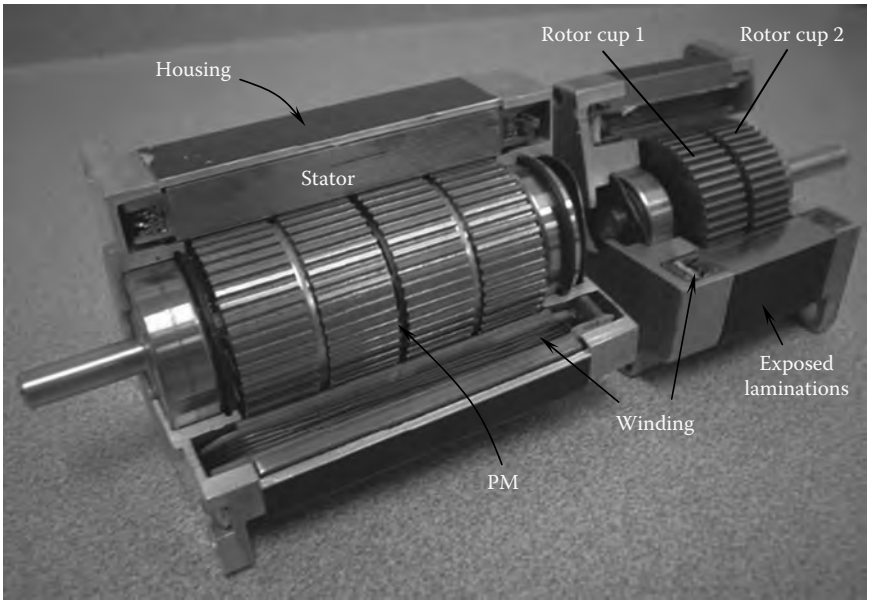


FIGURE 1.34
Structures of hybrid stepper motors, which combines features of the PM stepper and the variable reluctance stepper motors.

Figure 1.35, two axially spaced sections each formed with radially projecting and angularly spaced teeth. In addition, the teeth on the north end of the rotor are displaced by a half of a pole pitch from the teeth on the south end. Both of these two features make stepper motors unique from other motors.

For standard reluctance stepper motors (e.g., without PM), the rotors look similar to the aforementioned rotor except there is no PM and the magnetic field is not permanently present in the rotor lamination. The coils in the stator create a magnetic field in the stator that attracts or repels the soft iron rotor laminations into position. Then the coil current is reversed to reverse the coil magnetic field and keeps the rotation moving. A stepper motor drive is required to switch the currents.

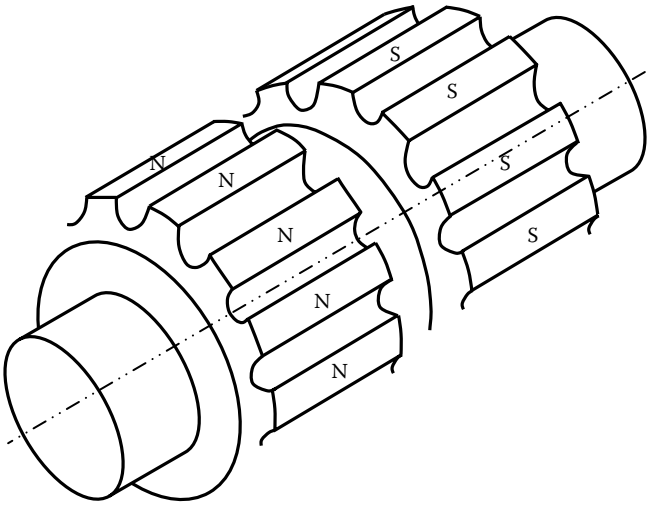


FIGURE 1.35
Magnetic polarity of rotor laminations in a hybrid step motor. The PMs are magnetized axially and placed in the center of lamination stacks. The rotor teeth on the north end of the magnet are magnetized north and the rotor teeth on the south end of the magnet are magnetized south. The north and south halves are displaced by a half of a pole pitch from the teeth on the south end.

1.3.6 Gear Drive and Direct Drive Motors

There are several motor driving patterns in motion control applications: (1) gear drive, (2) direct drive, (3) tangential drive (e.g., belt, chain), (4) ball/lead screw drive, and (5) worm drive.

Gear drive is a conventional mechanism in connecting electric motors to external mechanical loads. The shaft of a motor is attached to one end of the gearbox and, through the internal configuration of gears of the gearbox, provides a given output torque and speed determined by the gear ratio. In such a way, a motor is coupled with a transmission device (e.g., gearbox) to reduce the rotating speed and increase the output torque. The most important design parameter of gearboxes is the gear ratio γ_g, which is defined as the ratio of the input speed to the output speed. In a gear drive system, the ratio of the load torque T_l (the output torque) to the motor torque T_m (the input torque) is determined by the gear ratio γ_g and the gearbox efficiency η_{gb}:

$$\frac{T_l}{T_m} = \gamma_g \eta_{gb} \tag{1.51}$$

Gearboxes are available in many different types, sizes, gear ratios, efficiencies, and backlash characteristics. They can be generally categorized into several different types: parallel shaft gearbox (e.g., spur and helical gearbox), perpendicular shaft gearbox (e.g., bevel gearbox), planetary gearbox, and worm gearbox, just to name a few. For the type of parallel shaft gearbox, according to the number of shafts, gearboxes can be further classified as single-stage (two shafts), dual-stage (three shafts), and multistage (more than three shafts) gearboxes. Usually, this type of gearboxes takes more space than other gearbox types.

Hesai Exhibit 1031, page 84

Planetary gearboxes are the most commonly used gearboxes in the motion control market due to their high torsional rigidity, high torque, high efficiency, low inertia, and low backlash.

However, there are some disadvantages in using gearboxes. The main problem is backlash introduced by gearboxes. Backlash is the gap between the teeth of two adjacent gears. Thus, the rotational backlash of a gearbox is the accumulated backlash from all paired gears. It can be measured as the free rotational angle at the gearbox output shaft when the input shaft is locked, or vice versa. In precise motion control applications involving frequent load reversals (e.g., CNC machines, elevators, wind pitch control systems), backlash plays a crucial role to determine the repetitive positioning accuracy. For this type of applications, gearboxes must be made with low, strictly controlled backlash and high stiffness. For demonstration purpose, the theoretical backlash of gearbox and torsional stiffness are shown in Figure 1.36.

The use of separated motor and gearbox may greatly increase the system volume and lower the torque density and system reliability. To solve these problems, some motor manufacturers have designed motors with built-in gearboxes for achieving the solution of high torque capacity, smooth operation, minimal maintenance, and high-efficiency motors in compact sizes, as shown in Figure 1.37.

DDR motors are designed to directly drive an external loaded machine. They can be housed or frameless component sets of a PM field assembly and wound stator. The frameless part set can be built into a machine as *embedded motion* onto the user's shaft, housing, and bearing assembly and offers the most compact form factors. By eliminating the need for additional mechanical transmission devices, it allows improved motor reliability, efficiency, dynamic performance, torque density, motion accuracy, and disturbance response. This also leads to large mass savings in the mechanical structure and enables further cost reductions. There are other advantages such as better load inertia matching, ease of control, low noise emission, and streamlined machine design.

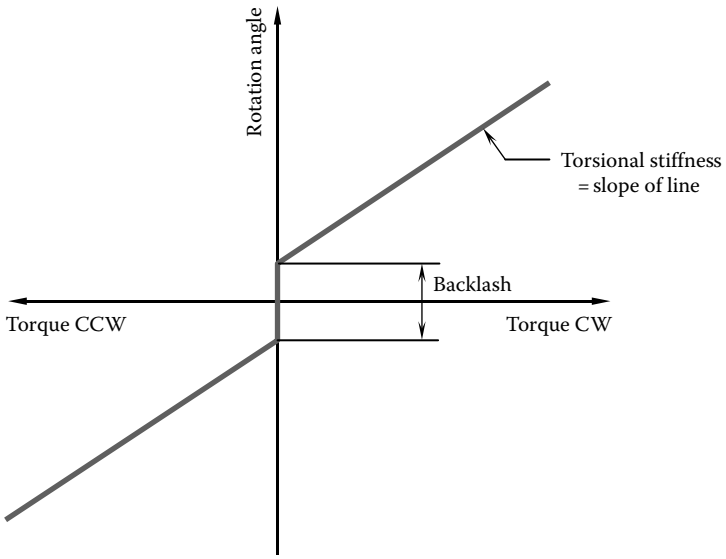


FIGURE 1.36
Theoretical backlash of a gearbox and stiffness curve.

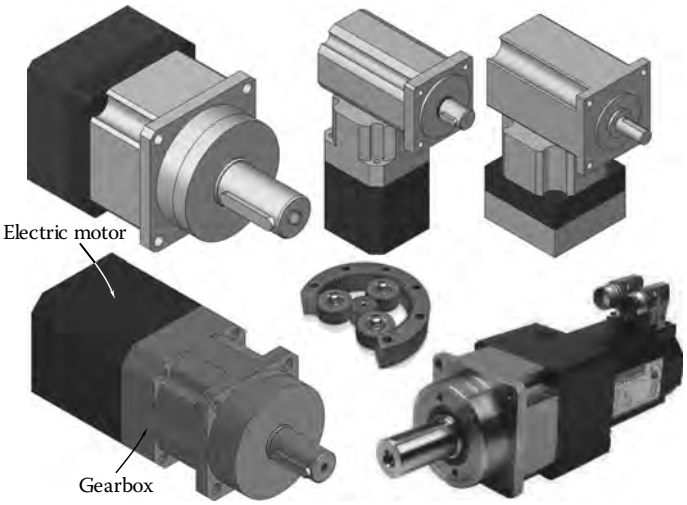


FIGURE 1.37
Integration of electric motor and gearbox as gearmotor.

Typical DDR motors are shown in Figure 1.38, which offer many high-performance features and zero maintenance in a precision servo solution. Among them, the cartridge DDR motor, developed by Kollmorgen in 2005, is the first in the motor industry to combine the space-saving and performance advantages of frameless DDR technology. The cartridge DDR motors are supported by customers' bearings, thus providing up to 50% more torque density than comparably sized conventional servomotors and the ability to remove the motor without dissembling the machine. By making direct drive benefits available to simple mechanisms, this type of motors has been successfully used in many applications such as packaging, printing, medical equipment, converting, and others.

For a DDR motor, torque is proportional to rotor diameter squared. Thus, in order to increase motor torque, DDR motors are typically designed with high diameter-to-length ratios, that is, large diameters and short axial lengths. In addition, a DDR motor can simultaneously have both a very large OD and ID. This implies that the DDR motor looks more like a thin ring. In fact, the large OD and ID help develop the large motor torque. This thin-ring shape may be particularly suitable for some applications requiring large diameters and short lengths such as computerized tomography (CT) scanners, magnetic resonance imaging (MRI) scanners, and wind turbines. As an example, a DDR motor for a telescope drive has a diameter of 2.5 m and a length of less than 50 mm. This motor can produce a continuous torque exceeding 10,000 N-m [1.64]. In large machine tool applications, DDR diameters of 1 m or so are commonly encountered.

Frameless DDR motors do not contain housings, bearings, or feedback devices, as shown in Figure 1.39. This implies that a frameless DDR motor must be integrated into the user's machine. To help integrate the motor, motor manufacturers often provide a reusable assembly tool called *bridge* to maintain the alignment of the rotor and stator during assembly.

The design of DDR motors is mainly influenced by its rotating speed, torque, and power. The achievable power of a motor is approximately proportional to the square of the air gap diameter and to the axial length of the motor at the air gap. This indicates that with a doubling of the air gap diameter, the motor power can increase four times. As a matter of fact, direct motors with low rotating speeds usually have a large diameter and a short length.

Hesai Exhibit 1031, page 86



FIGURE 1.38
Various DDR motors. (Courtesy of Kollmorgen Corporation, Radford, VA.)



FIGURE 1.39
Frameless DDR motors. (Courtesy of Kollmorgen Corporation, Radford, VA.)

Comparing with conventional servo systems, the advantages of direct drive systems include the following:

- It significantly increases system efficiency by eliminating inefficient gearboxes.
- Because the gearbox in a motor system is a primary noise source, the elimination of the gearbox greatly reduces the noise emission from the motor.
- Motor reliability increases due to the reduction in the number of rotating parts.
- The positioning inaccuracy from transmission compliance no longer existed.
- Lower inertia enables fast and accurate positioning.
- A direct drive servomotor is capable of precise revolution control with high resolutions.
- Without speed reduction devices, mechanical backlash, hysteresis, and elasticity in gear transmission systems are eliminated.
- There is no need for gearbox lubrication and it simplifies maintenance.

However, direct drive motors require more precise control systems. Since the direct drive motors need special design, their individual costs are much higher than other servomotors, but system costs utilizing them can be lower than the total installed cost.

1.3.7 Brush and Brushless Motors

Brush motors have a long history of applications since the first electric motor was invented in the late nineteenth century. For a classical DC motor, an electrical power source is connected to the rotor winding through a commutator and brushes. A commutator usually consists of a set of contact bars attached to the rotating shaft of a motor. As the stator windings are sequentially energized to generate a rotating magnetic field, the rotor aligns itself with the magnetic field of the stator. A number of stationary brushes come into contact with corresponding the contact bars of the commutator for reversing the flow of current in the rotor winding. In order to reduce the contact electric resistance at the contacting surface, brushes are pressed against the commutator and thus result in some friction between the brushes and commutator. In addition, at high rotating speeds, brushes become increasingly difficult to maintain reliably in contact with the commutator. Brushes may bounce off from the commutator surface to produce sparks and consequently break the circuit in a very short period of time. Continuous sparking can cause overheat, erode, or even melt the commutator. For this reason, brush motors are restricted to relative low speeds to avoid brushes excessively bouncing and sparking.

To overcome these drawbacks, brushless motors have been developed to replace brushed motors in some motion control applications. Brushless DC motors typically operate with PM rotors, offering distinct mechanical and electrical advantages over conventional systems. Attachment of PMs on rotor and elimination of rotor coil windings allow significant reductions in rotor inertia and increase in motor acceleration and efficiency. Winding heat can be dissipated directly from the stator into the environment and rotor heating is much lower than that in a brush motor. Generally, comparing to brush motors, brushless motors offer higher operation reliability, higher dynamic accuracy, higher efficiency, less product variation, and smaller size but require complex and expensive electronic control systems.

With the widespread use of brushless AC motors in industrial, commercial, and military applications, brushes are gradually exiting the electrical machine market.

1.3.8 Reluctance Motors

Reluctance motors are special motors that use salient pole rotors but without windings and PM on the rotors. There are several types of reluctance motors, with different construction and slightly different functions [1.65]:

- Switched reluctance motor (SRM)—By eliminating windings and PMs from the rotor, this type of motor has inherent advantages over some DC and AC motors, such as simple structure, structure robustness, low cost, and operation in high temperatures. SRMs have been designed mainly for the applications in high-power, high-efficiency, variable speed drives, enabling to deliver a wide range of torque. This type of motor requires the closed-loop position control.
- VRM—This type of motor is an evolution of the stepper motor but with less salient poles. The motors are generally designed for use in low-power, open-loop position and speed control systems where efficiency is not of prime importance. In fact, there are similarities between switched and VRMs such as the operating principles, mechanical structures, performing characteristics, and power losses. Both of the types are similar to the brushless PMMs except that the rotors are made from laminated *soft* magnetic materials.
- Synchronous reluctance motor—This type of motor is similar to synchronous AC motors. The rotor has salient poles but the stator has smooth, distributed poles, whereas both the switched and variable motors have salient poles for both the rotor and stator.

SRMs can be traced back to the invention of Robert Davidson in 1938 [1.66]. However, this type of motors did not find widespread use until the late 1970s due to the difficulty in controlling the machine. For an SRM, there are no windings or PMs on the rotor. When the stator windings are powered, the rotor's magnetic reluctance produces a force that attempts to align the rotor with the exited stator poles. The coils in the adjacent slots are powered successively so that the rotor can rotate continuously. This operation principle is based on the difference in magnetic reluctance for magnetic field lines between aligned and unaligned rotor position when a stator coil is excited; the rotor experiences a force that will pull the rotor to the aligned position [1.67].

Due to the elimination of magnets and rotor windings, SRMs offer several performance, efficiency, and cost advantages. In an SRM, torque is produced by the tendency of its moveable part to move to a position of least reluctance. The SRM has salient poles on both the rotor and the stator, but only the stator poles carry windings. When the stator windings are energized, they create a magnetic field that pulls the nearest pole on the rotor toward it. Consequently, the performance of SRMs is largely a function of the power electronics that control the sequencing of pole energizations. Besides, the SRMs have characteristically high power-to-weight ratios and are well suited for vehicle applications [1.68].

SRMs have been found to offer important advantages over conventional motors in many industrial and commercial applications. In recent years, SRMs have been receiving increasing attention because they can provide similar performance to brushless DC motors without using expensive rare-earth magnets. In addition, their robust construction and efficiency over a wide speed range make them well suited to a variety of applications including hybrid and electric vehicles. However, the main drawbacks are its high-torque ripple and consequently high audible noise. As shown in Figure 1.40, the stator of an SRM is similar to a brushless DC motor but the rotor has no magnets or windings attached.

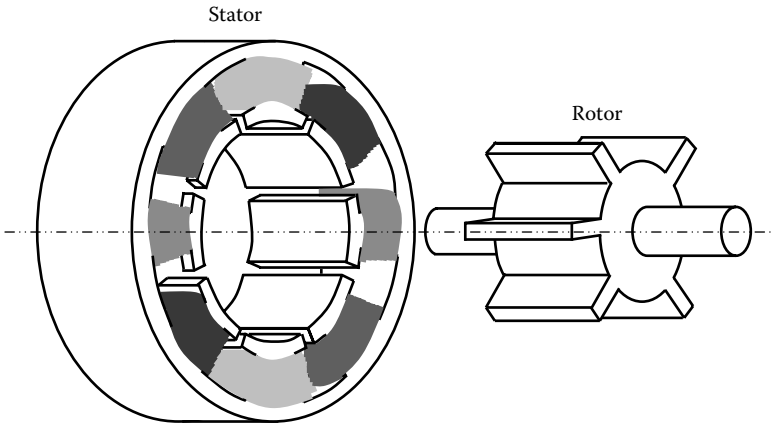


FIGURE 1.40
The salient rotor in an SRM.

1.3.9 Radial-Flux and Axial-Flux Motors

The configuration of the electromagnetic field in a motor determines the motor's geometry and structural topology. Electric motors in which the magnetic flux travels in the radial direction are classified as radial-flux (RF) motors. This type of motors is cylindrical in shape, and the rotor is typically located inside a stator but can also be placed outside the stator in some special applications, sometimes referred to as *inside out* motors or outer rotating rotor motors. As a contrast, motors in which the magnetic flux travels in the axial direction are classified as axial-flux (AF) motors. This type of motors usually has multiple disk- or pancake-shaped rotors and stators. A typical brushless AF motor is demonstrated in Figure 1.41, showing a stator–rotor–stator structural topology. AF motors offer low axial

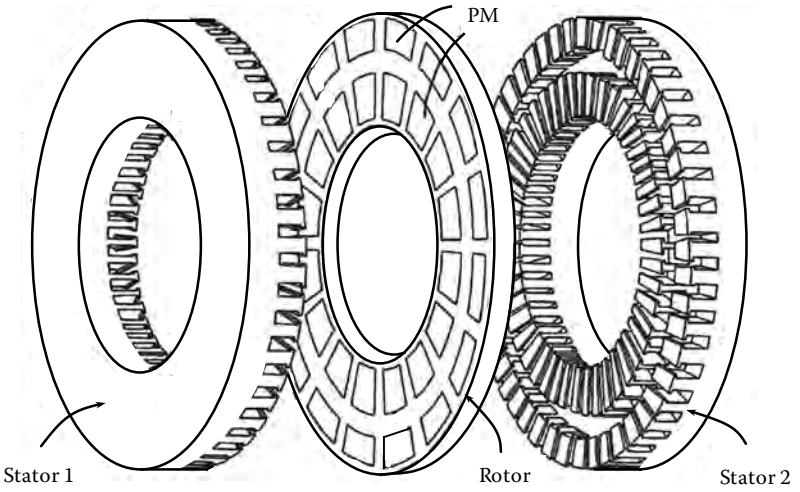


FIGURE 1.41
Brushless AF motor with the stator-rotor-stator structural topology, where PMs mounting on the rotor side surfaces and rotating with the rotor.

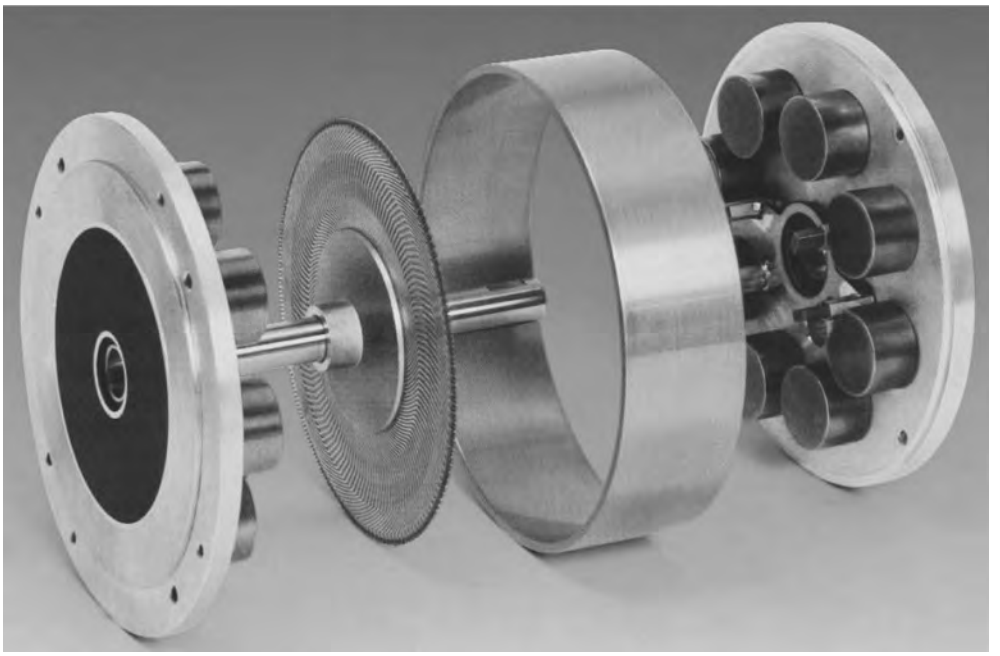


FIGURE 1.42
DC brush AF motor, where PMs are mounted on the stators to create an axial magnetic field. (Courtesy of Kollmorgen Corporation, Radford, VA.)

length and are common in automotive radiator cooling fan applications, table top medical or lab devices, and specialized elevator machines to name but a few. The structure of a DC brush AF motor is shown in Figure 1.42. As can be seen from the figure, the armature is constructed from several layers of copper conductors in a unique flat-disk configuration. Because it adopts ironless design, the thin, low-inertia armature design leads to exceptional torque-to-inertia ratios. The ironless armature enables the motors to deliver more torque over their entire speed range. In fact, the torque is almost constant from 0 to 3000 rpm, which is not attainable with conventional ironcore motor designs. In addition, this type of motors can accelerate from 0 to 3000 rpm in only 60^{o} of rotation [1.69].

1.3.10 Rotary and Linear Motors

According to the motion pattern of motor, an electric motor can be classified as either rotary or linear motor. As indicated in their names, the rotor in a rotary motor always rotates with respect to its rotating axis and the rotor in a linear motor moves linearly along the flat stator. Actually, a linear motor has the motion of a rotary motor if it was laid out in a flat state.

Linear motors are typically used in the applications that require linear or reciprocating motions (Figure 1.43). A famous example is the high-speed maglev train that was built in 2004 at Shanghai, China, and has been in normal commercial operation thereafter. Similar to rotating motors, linear motors can be grouped as induction or PM linear motors. When very fast linear acceleration or high linear forces are required, linear motors are often employed. A disadvantage is the mechanical structure to support them is more expensive than rotary motor structures.



FIGURE 1.43
A linear motor. (Courtesy of Kollmorgen Corporation, Radford, VA.)

1.3.11 Open and Enclosed Motors

An open motor refers to the motor that has ventilating openings that permit passage of external cooling air to get in the motor for flowing over and around the windings. With the openings on the motor housing, it makes possible for the external objects such as dusts and moisture to enter into the motor, causing motor damage even failure. Therefore, this type of motors cannot work under harsh environments.

An enclosed motor refers to the motor that is completely enclosed to prevent the free exchange of air between inside and outside of the motor housing. In such a configuration, heat is usually dissipated relying on conduction from the heat source (i.e., stator windings) to the motor housing and then from the housing to the environment by convection. This cooling mode is applicable only to small motors with relative low heat loads. For high-powered motors, a build-in cooling system such as a liquid cooling system or using heat pipes is necessary. This type of motors usually has high International Protection Rating or Ingress Protection Rating (IP) values and thus can be used in harsh and dangerous environments.

1.3.12 Motor Classification according to Power Rating

According to the motor nominal power, electric motors can be briefly divided into five categories:

- Micromotors are electric motors with rated output poser of 0.05 hp (<37 W) or less. Micromotors have been used across a wide range of commercial and industrial applications for light duty, especially in microelectronics, computer, and precision instrument industries.

- Small motors are larger than 0.05 hp but less than 1 hp (37–746 W). Their applications focus primarily on power hand tools, appliances, medical devices, small fans, optical devices, electrical cars, precise motion control systems, and other small machinery.
- Medium motors are considered to be in the range of 1–100 hp (746 W–74.6 kW). The majority of medium motors are used in various industrial applications such as industrial fans, pumps, motion-control systems, machine tools, and vehicles.
- Large motors occupy the power range of 100–1000 hp (74.6 kW–746 kW). They are normally designed for use in medium-duty applications as in elevators, large vehicles, industrial blowers/fans, printing machines, package machines, air compressors, and other industrial large machines.
- Extra large motors range from 1000 to 10,000 hp (746 kM–7.46 MW), which are usually used in heavy-duty applications, such as large rolling mills, large machine tools, high-speed trains, skyway elevators, ship propulsion, and mining machinery.
- Ultra large motors are considered to be larger than 10,000 hp (>7.46 MW). NASA used a motor that is rated 135,000 hp for a wind tunnel. In addition, the industry's largest players such as GE, Siemens, TECO-Westinghouse, ABB, and Toshiba have developed their own ultra large motors.

It is worth to note that the range for each motor category shown earlier has not been clearly defined by international, national, or professional standards yet.

1.4 Motor Design and Operation Parameters

In designing, selecting, and repairing electric motors, it often involves a number of constants that reveal the relationships of torque, current, speed, voltage, power loss, and other operating characteristics. Among these constants, the back EMF constant and torque constant are the two most important constants for evaluating motor performance.

1.4.1 Back EMF Constant, K_e

The induced voltage in motor conductors under a rotating magnetic field is defined as the back EMF V_e and is directly proportional to the angular velocity of the motor. Thus, the proportionality constant, referred to back EMF constant, is defined as the ratio of back EMF to motor rotational speed, in the unit of V/(rad/s) or V/rpm (in some applications, V/krpm):

$$K_e = \frac{V_e}{\omega} \tag{1.52}$$

Thus, K_e is a measure of how many volts per rpm the motor would produce if driven as a generator. It can be also used to determine how fast a motor will run with a given voltage applied to it.

1.4.2 Torque Constant, K_t

K_t is called torque constant (N-m/A) or torque sensitivity, which is defined as the torque T (in N-m) generated by a motor to the motor input current I (in A), that is,

$$K_t = \frac{T}{I} \tag{1.53}$$

For a sinusoidal commutated motor, the root-mean-square (*rms*) current i_{rms} should be used to replace I in the aforementioned equation, where i_{rms} is related to the peak AC current i_p as

$$i_{rms} = \frac{i_p}{\sqrt{2}} \tag{1.54}$$

Figure 1.44 presents the block diagram of conventional circuit for estimating motor torque constant. This conventional engineering method assumes that when the motor is accelerated, the motor torque constant is proportional to a position change as the output of the motor, regardless of the input current of the motor. However, motor torque constant actually depends on the input current during acceleration of the motor. Hence, the conventional method cannot provide the accurate result for motor torque constant. Based on a new block diagram shown in Figure 1.45, a new method was proposed to calculate motor torque constant by a multiplicity of measured current and speed values [1.70]:

$$K_t = \frac{\sum_1^n i(k-1)\left[v(k)-v(k-1)\right]}{\sum_1^n i(k-1)^2} \tag{1.55}$$

where

$i(k)$ and $v(k)$, respectively, indicate the current and the speed of the motor at a sampling time k

n indicates a natural number greater than 1

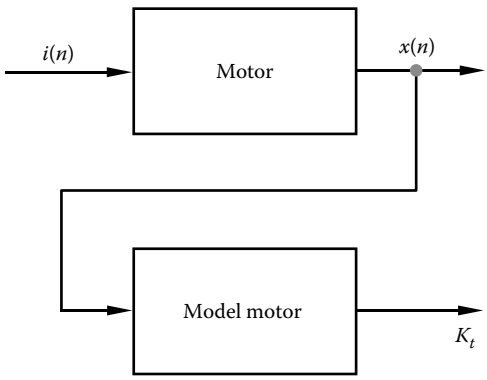


FIGURE 1.44
The block diagram of conventional circuit for estimating motor torque constant.

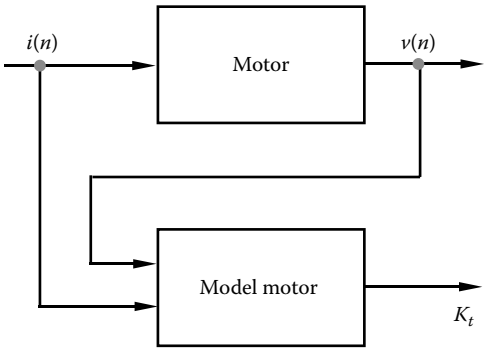


FIGURE 1.45
The block diagram of proposed new circuit for estimating motor torque constant.

1.4.3 Velocity Constant, K_v

K_v is the motor velocity constant, measured in rpm or rad/s per volt. In fact, K_v is the reciprocal of K_e. For brushless motors, K_v is the ratio of the motor's unloaded rpm or rad/s to the peak voltage. This parameter is used for selecting the winding in the motor.

It can be shown that if the torque constant K_t is in N-m/A and the back EMF constant K_e in V/(rad/s), then,

$$K_t = K_e = \frac{1}{K_v} \tag{1.56}$$

1.4.4 Motor Constant, K_m

K_m is the motor constant, defined as the torque constant K_t divided by the square root of the resistive power loss P_r:

$$K_m = \frac{K_t}{\sqrt{P_r}} \tag{1.57}$$

- For DC motors,

$$P_r = I^2 R_{DC} \tag{1.58a}$$

- For three-phase AC motors,

$$P_r = i_{rms}^2 R_{AC} = \frac{3 i_p^2 R_{AC}}{2} \tag{1.58b}$$

where R_{DC} and R_{AC} are the winding resistances for DC and AC motors, respectively.

Substituting (1.58a) into (1.57), it yields

$$K_m = \frac{K_t}{I\sqrt{R_{DC}}} = \frac{T}{I^2\sqrt{R_{DC}}} \tag{1.59}$$

Since $P_{in} - P_{out} = \sum P_{loss} > I^2 R_{DC}$, it follows that

$$\sqrt{P_{in} - P_{out}} > I\sqrt{R_{DC}} \tag{1.60}$$

Therefore, the minimum torque constant is obtained as

$$K_{m,\min} = \frac{T}{\sqrt{P_{in} - P_{out}}} \tag{1.61}$$

Similarly, substituting P_r for the corrected 3-phase AC motor and thus obtaining another K_m equation as

$$K_m = \frac{K_t}{1.225 i_p \sqrt{R_{AC}}} \tag{1.62}$$

In fact, K_m represents the ability of a motor to convert electrical power to mechanical power. A motor with a higher value of K_m can generate torque more efficiently. K_m is winding independent and is used as a rating comparison factor in selecting the motor size in various motor applications. In some references, K_m is also called motor size constant.

1.4.5 Mechanical Time Constant, τ_m

In design and analysis of servomotors, it often deals with two time constants: mechanical time constant τ_m and electrical time constant τ_e. A servomotor's dynamic motion response is controlled by these two time constants. Usually, τ_m and τ_e are listed in the servomotor specifications. However, it should be cautioned that these two time constants in the specifications are for the motor alone with no load inertia connected to the motor shaft. It is important to understand the impact of actual load conditions on the time constants [1.71].

The mechanical time constant τ_m (in unit of second s) is usually defined as the ratio of motor moment of inertia to the damping factor with a zero-impedance power source:

$$\tau_m = \frac{RJ_m}{K_t K_e} \tag{1.63}$$

where

R is the motor winding resistance
J_m is the motor moment of inertia

To take into account the impact of load on τ_m, the previously mentioned definition may be modified as

$$\tau_m = \frac{R_t J_t}{K_t K_e} \tag{1.64}$$

where

R_t is the total motor winding resistance of all phases plus the external circuit resistance and the total inertia
J_t is the sum of the motor inertia and the reflected inertia from the load to the motor shaft

According to NEMA [1.61], the mechanical time constant refers to the time taken by an unloaded motor to reach $(1 - 1/e)$ (where $1 - 1/e \approx 63.2\%$) of its maximum rated speed in a no-load condition. τ_m can be measured by applying a constant voltage to the motor, then measuring the velocity, and determining the time it takes to reach 63.2% of maximum rated speed.

One important factor to affect the mechanical time constant is temperature. During a normal operation process, the motor temperature rises from the ambient time (usually 40°C) to its normal operation temperature that is below the allowable maximum value. The electric resistance of the winding changes accordingly as

$$R(T) = R(T_o)[1 + \alpha(T - T_o)] \tag{1.65}$$

where α is temperature coefficient of resistance ($°C^{-1}$ or K^{-1}).

Similarly, for PMMs, the back EMF constant K_e and torque constant K_t are also affected by temperature. Both K_e and K_t have the same functional dependence on the motor's air gap magnetic flux density that is produced by the motor's magnets. All PMMs are subject to both reversible and irreversible demagnetization. It has been reported [1.72] that within the temperature range of −60°C to 200°C, all four types of PMs (aluminum–nickel–cobalt [alnico], SmCo, NdFeB, and ferrite/ceramic magnets) exhibit linear, reversible thermal reduction in field strength such that the amount of magnetic flux density produced by each magnet decreases linearly with increasing magnet temperature. Hence, the expression for the reversible decrease in both $K_e(T)$ and $K_t(T)$ with increasing magnet temperature is given as

$$K_e(T) = K_e(T_o)\left[1 - \alpha(T - T_o)\right] \tag{1.66a}$$

$$K_t(T) = K_t(T_o)\left[1 - \alpha(T - T_o)\right] \tag{1.66b}$$

where temperature coefficient of resistance α takes different values according to the different types of PM:

$$\alpha\ (\text{Alnico}) = 0.0001/°C$$
$$\alpha\ (\text{SmCo}) = 0.00035/°C$$
$$\alpha\ (\text{NdFeB}) = 0.001/°C$$
$$\alpha\ (\text{Ferrite}) = 0.002/°C$$

Thus, a 100°C rise in magnet temperature causes a reversible decrease in K_e and K_t as –1% for alnico, –3.5% for SmCo, –10% for NdFeB, and –20% for ferrite or ceramic magnets. This indicates that the motor operation temperature has the strongest impact on ferrite magnets.

By taking into account both load and temperature, the mechanical time constant becomes

$$\tau_m(T) = \frac{R_t(T)J_t}{K_t(T)K_e(T)} \tag{1.67}$$

Taking the ambient temperature of 40°C (104°F), the normalized mechanical time constant ratio, $\tau_m(T)/\tau_m(40°C)$, for each type of PMs is plotted in Figure 1.46 as a function of

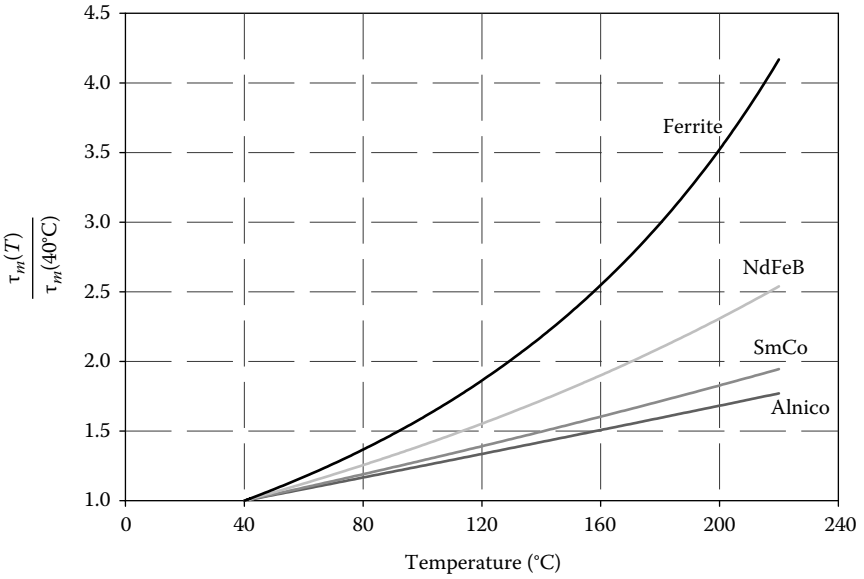


FIGURE 1.46
The temperature influence on mechanical time constant ratio for four types of PMs, with the ambient temperature of 40°C.

temperature. The increase in the mechanical time constant ratio is resulted from the combined effects of increasing electrical resistance and thermal demagnetization. It can be seen from the figure that the largest increase occurs in ferrite magnets with the mechanical time constant ratio increasing by a factor of 2.45 at 155°C. At the same temperature, it increases by a factor of 1.85, 1.57, and 1.49 for NdFeB, SmCo, and alnico magnets, respectively. These results have confirmed that the operation temperature has a strong impact on servomotor's dynamic motion response. Therefore, ignoring the temperature effect can lead to erroneous predictions of the servomotor's dynamic motion response to the applied voltage command. Rare-earth and ferrite/ceramic magnets have increased in magnetic flux at cold temperatures. Rare-earth magnets have improved resistance to demagnetization at cold temperatures, whereas ferrite/ceramic magnets have an increase in demagnetization risk at low temperatures.

1.4.6 Electrical Time Constant, τ_e

According to NEMA [1.61], the electrical time constant of a servomotor is the time required for the current to reach $(1 - 1/e)$ (i.e., 63.2%) of its final value after a zero source impedance, and the stepped input voltage is applied to the motor that maintained in the locked rotor or stalled condition (i.e., $\omega = 0$). A small electrical time constant indicates the high dynamic response of a servomotor.

Mathematically, the electrical time constant τ_e is defined as the ratio of armature inductance L to its winding electric resistance R:

$$\tau_e = \frac{L}{R} \tag{1.68}$$

Similarly, by taking into account the load and temperature effect, the equation becomes

$$\tau_e(T) = \frac{L}{R_t(T)} \tag{1.69}$$

where $R_t(T)$ is the total resistance, including the resistance of all phase windings and the external circuit resistance, at the temperature T. This equation indicates that with the increasing temperature, electrical time constant τ_e decreases due to the increase in $R_t(T)$.

1.4.7 Thermal Time Constant, τ_{th}

The thermal time constant τ_{th} is an indicator of the heat capacity of a motor, showing how fast or how slow the generated heat can be built up in the motor and can be effectively dissipated to the environment. In another words, it is a measure of how long it takes a motor to reach thermal equilibrium.

The thermal time constant τ_{th} can be expressed as

$$\tau_{th} = \frac{\rho c_p V_e}{h A_s} \tag{1.70}$$

where

ρ is the density
c_p is the specific heat
V_e is the effective motor volume (this is because a motor is not a solid)
h is the convective heat transfer coefficient
A_s is the motor outer surface area

This equation indicates that larger mass m ($m = \rho V_e$) and specific heat c_p lead to larger τ_{th}, that is, larger heat storage capability and slower changes in temperature. Higher heat transfer coefficient h and larger motor outer surface area A_s help dissipate the generated heat quickly from the motor to the ambient, leading to smaller τ_{th}, that is, faster changes in temperature. Obviously, a lower value of thermal time constant is highly desired.

The motor thermal time constant may be used to estimate the temperature rise of a motor. The motor temperature rise ΔT can be predicted as [1.73]

$$\Delta T = T - T_a = P_{loss} R_{th} + \left(T_a + T_i - P_{loss} R_{th}\right) e^{-t/\tau_{th}} \tag{1.71}$$

where

T_a is the ambient temperature
T_i is the motor initial temperature at the start of operation
P_{loss} is the motor total power losses that must be dissipated from the motor
R_{th} is the overall motor-to-ambient thermal resistance
t is the motor operation time

If ΔT is specified, the aforementioned equation can be used to determine how long of the motor could reach the specified value of temperature rise.

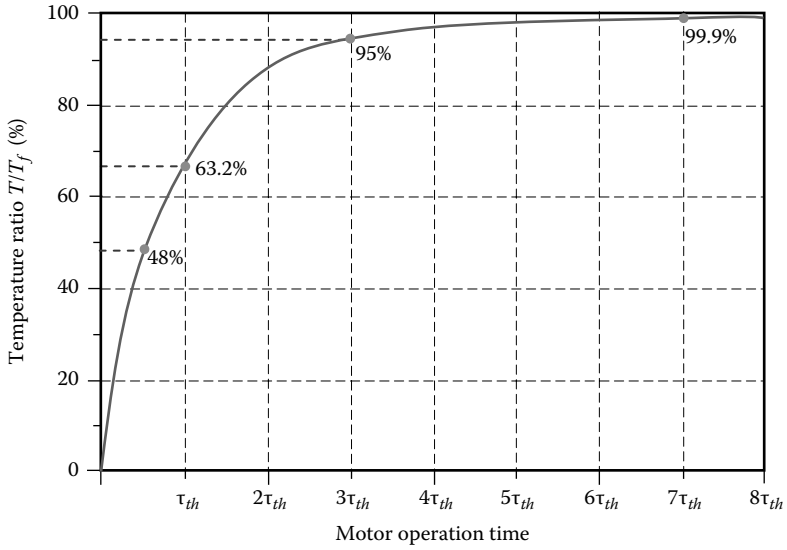


FIGURE 1.47
Variation of temperature ratio T/T_f as a function of motor operation time for a constant applied power.

Mathematically, as t approaches infinity (i.e., $e^{-t/\tau_{th}}$ approaches zero), the system attains thermal stability and the motor temperature reaches its final temperature T_f. Hence, the temperature rise ΔT becomes

$$\Delta T = T - T_a = P_{loss} R_{th} \tag{1.72}$$

The variation of the temperature ratio T/T_f with respect to motor operation time is plotted in Figure 1.47. At $t = 0$, the power is supplied to the motor and the motor temperature starts to rise exponentially until it reaches its final temperature at the thermal equilibrium. According to the engineering definition, thermal time constant is the time required to reach 63.2% of the final temperature.

It is worth to note that in practice, it is found that the thermal time constant of motor is not constant. It varies correspondingly to the change in heat load rates. For example, motors running in intermittent operation will have a shorter time constant than a motor running up to temperature in a continuous load operation. This is due to the definition of τ_{th} that assumes no internal heat generation and uniform heat distribution. Both are not purely true in motor operation.

1.4.8 Viscous Damping, K_{vd}

In general, damping can be divided into three types: viscous damping, coulomb or dry-friction damping, and hysteretic or structural damping. From the standpoint of physics, viscous damping is the dissipation of energy as occurred in liquid or air between moving parts. An example of viscous damping is ball bearing lubrication. It results in lower torque delivered at the output shaft to the torque developed at the rotor. Viscous damping in a single-degree-of-freedom torsional system is referred to as torsional viscous damping $K_{vd,t}$,

which is directly proportional to the damping torque T_d and inversely proportional to the angular velocity ω and is always opposite to the direction of motion, that is,

$$K_{vd,t} = \frac{T_d}{\omega} \tag{1.73}$$

The unit of viscous damping is N-m/rpm or N-m/(rad/s).

For reciprocating or linear motion systems, viscous damping $K_{vd,r}$ is defined as the ratio of the damping force F_d over the velocity $\dot{x}(t)$ with the unit of N-s/m:

$$K_{vd,r} = \frac{F_d}{\dot{x}(t)} \tag{1.74}$$

Another example of viscous damping is iron losses in a motor that are functions of frequency, circulating currents and iron mass.

1.5 Sizing Equations

The determination of appropriate motor size is an essential task in motor design. An oversized motor can provide itself a higher safety factor in stress and structural firmness but waste energy and can potentially create performance problems with the driven equipment, especially in turbomachinery such as fans or pumps. In some circumstances, an oversized motor may compromise the reliability of both the components and the entire system.

Empirical sizing equations are very useful for motor engineers to preliminarily determine the motor design at the early stage of motor design. Various sizing equations have been developed by different researchers and designers in the history of motor development. A set of sizing equations was proposed by Honsinger [1.74] for induction machines. This method focuses on the optimal electrical loading and the machine internal geometry for a given power level:

$$\frac{P}{n_s} = \xi_s D_{s,o}^3 L_e \tag{1.75}$$

$$\frac{P}{n_s} = \xi_r D_{r,o}^2 L_e \tag{1.76}$$

Based on these two sizing equations, the $D^{2.5}L_e$ sizing equations can be derived as

$$\frac{P_o}{n_s} = \xi_s' D_{s,o}^{2.5} L_e \tag{1.77}$$

$$\frac{P_o}{n_s} = \xi_r' D_{r,o}^{2.5} L_e \tag{1.78}$$

where

$D_{s,o}$ and $D_{r,o}$ are the OD of stator and rotor, respectively
P_o is the output power
n_s is the rotor rotating speed
L_e is the effective stack length and ξ_s, ξ_s' and ξ_r, ξ_r' are coefficients for stator and rotor, respectively

These coefficients contain the information regarding motor structure.

However, traditional sizing equations are based on the premise that the excitation of the electrical machine is provided by a sinusoidal voltage source to produce a sinusoidal EMF. In order to eliminate the deficiencies of traditional sizing equations, scientists at University of Wisconsin [1.75–1.77] have developed a general-purpose sizing equation that could take into account different waveforms of back EMF and machine characteristics. A particular effort has been made to express the sizing equation that characterizes the output power P_o as a function of overall volume of the machine. This sizing equation is easily adjustable for different motor topologies, such as RF, AF, and transverse-flux (TF) motors. The general-purpose sizing equation takes the form of

$$P_o = \frac{1}{1+K_\phi} \frac{m}{m_1} \frac{\pi}{2} K_e K_i K_p \eta B_g A \frac{f}{p} \lambda_o^2 d_{s,o}^2 L_e \tag{1.79}$$

where

$K_\phi = A_r / A_s$ is the ratio of electrical loading on rotor and stator
$A = A_r + A_s$ is the total electrical loading
m is the number of phases of the motor
m_1 is the number of phases of each stator (if there is more than one stator, each stator has the same m_1)
K_e is the EMF factor
K_i is the current waveform factor, defined as the peak phase current I_p to the *rms* value of the phase current I_{rms}
K_p is termed the electrical power waveform factor
η is the motor efficiency
f is the frequency
p is the motor pole pairs
λ is the ratio of the air gap surface diameter to the stator OD ($\lambda = d_g / d_{s,o}$)

The application of this general-purpose sizing equation can provide motor engineers with a useful tool in designing new high power density motors.

1.6 Motor Design Process and Considerations

The design of electric motor is a complex task, involving multidisciplines such as electromagnetics, mechanics, thermal science, material science, vibrations, acoustics,

rotordynamics, electronics, tribology, control theory, and mathematics. The design process of electric motors involves continuous iterations between electromagnetic, thermal, structural, rotordynamic, and systematic designs based on a variety of theoretical analyses, numerical simulations, and lab tests. To achieve an optimum design, all design parameters and criteria must be considered comprehensively.

There are two basic approaches in motor design: a subsystem/component approach and a system approach. Traditionally, a common engineering approach is to break down the system into subsystems or components (e.g., stator, rotor, feedback, cooling, coupling), design and optimize these subsystems, and then assemble them as a whole system. This is more likely the conventional bottom-up design strategy in engineering design. In the subsystem/component approach, the design of each subsystem/component is essentially independent of each other and all the work carry out in parallel. Thus, one of the benefits of this approach is its short design time with simplified problem. However, this approach ignores the interactions among different subsystems/components. This leads to a possible result that even if each subsystem/component performs well, the system as a whole may not perform well. This is because the sum of the functioning of the individual subsystem/component is quite often not equal to the functioning of the whole.

A system approach focuses on overall system performance and creates a technical solution that satisfies the functional requirements for the system. This is the result of the synthetic mode of thinking applied to physical problems. This approach takes account of the intrinsic connections and interactions among different subsystems/components. In this approach, the design engineers evaluate the entire system to determine how end-use requirements can be provided most effectively and efficiently. The system approach is actually based on a top-down design strategy, in which the requirements are always satisfied at every step of the design process.

In modern motor designs, engineers often involve some degree of compromise and trade-offs among various competing requirements and design features such as motor efficiency, operation reliability, torque density, IP rating, cooling, noise, and cost-effectiveness, to name a few. Decisions on trade-offs involve systematic comparisons of all benefits and costs for achieving a satisfactory overall design.

It is worth to note that the two design approaches are not always absolutely opposite. Under some circumstances, engineering designs may gainfully employ both methodologies [1.78].

1.6.1 Design Process

Good design of electric motor involves many engineering aspects upon which the success of the design work depends, including customer's specifications, motor operation condition, material selection, system integration, technical analysis, and manufacturing process, as demonstrated in Figure 1.48. In finding the best design solution, engineers must make trade-offs among many factors that determine the final design and cost of the motor. In addition, identifying and understanding the design constraints and limitations are critical to the success of the motor design.

The general design flow chart of electric motor is presented in Figure 1.49. The design process starts with the careful reviewing of customer's specifications and requirements, especially the requirements of motor operation and performance. Based on these information, the motor type and topology can be determined, such as PMM or IM, synchronous or asynchronous motor, and RF or AF motor. Then, the main motor design parameters are primarily set by using empirical equations. The overall sizing of the machine is of

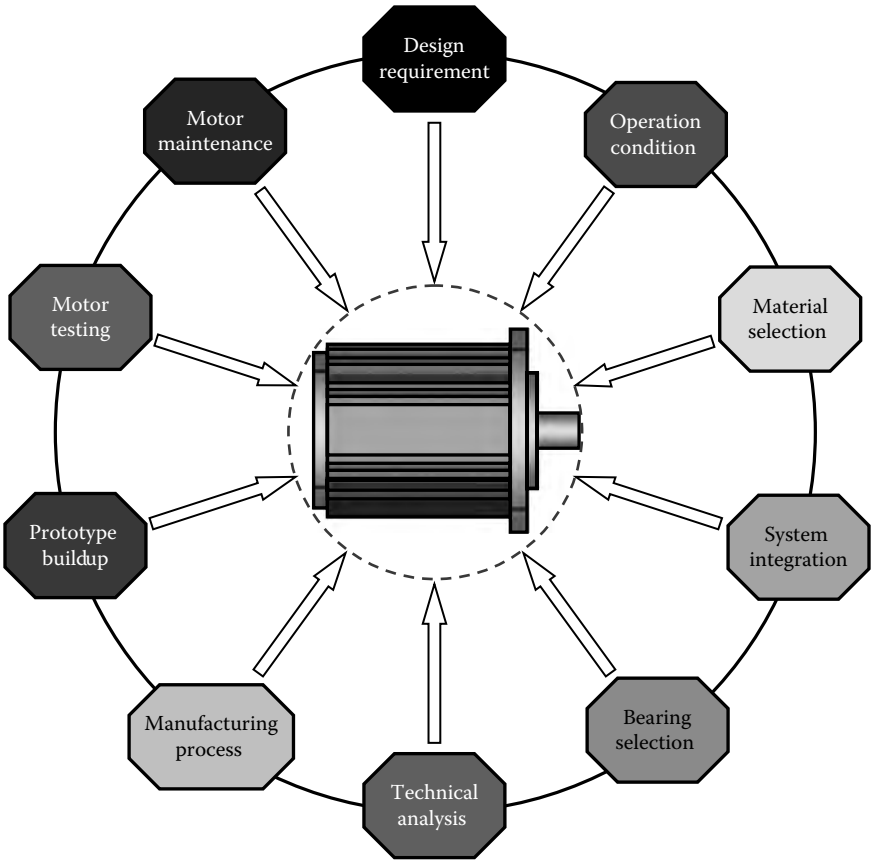


FIGURE 1.48
Motor design chart, indicating that in the motor design process many different design aspects must be taken into account.

great importance in terms of performance, structural integrity, cost, and weight. The major motor dimensions are determined via sizing equations available in the literature.

In the design of modern electric motors, none of technical design is isolated from the other designs. As a matter of fact, all designs in different fields have strongly influenced each other. Hence, these design processes are neither in series nor in parallel, rather, they form a crossover network. During this design stage, engineers/designers must communicate frequently to exchange their design concepts, innovative ideas, experience, and other useful information, especially the modifications that could impact on other designs. The design iterations between different design processes continue to carry out until the initial design is fully implemented.

Electromagnetic design is the core part of the motor design for determining motor key operating and performing characteristics. The electromagnetic design is highly dependent on the motor structure and rotordynamics analyses performed on the rotor. Recently, with the increasing demands for high-efficiency and high-reliability motors, one of the trends in novel motor development is to integrate the electromagnetic structure and mechanical transmission structure into a functional system for not only simplifying the motor structure but also increasing torque density, motor efficiency, and operation reliability.

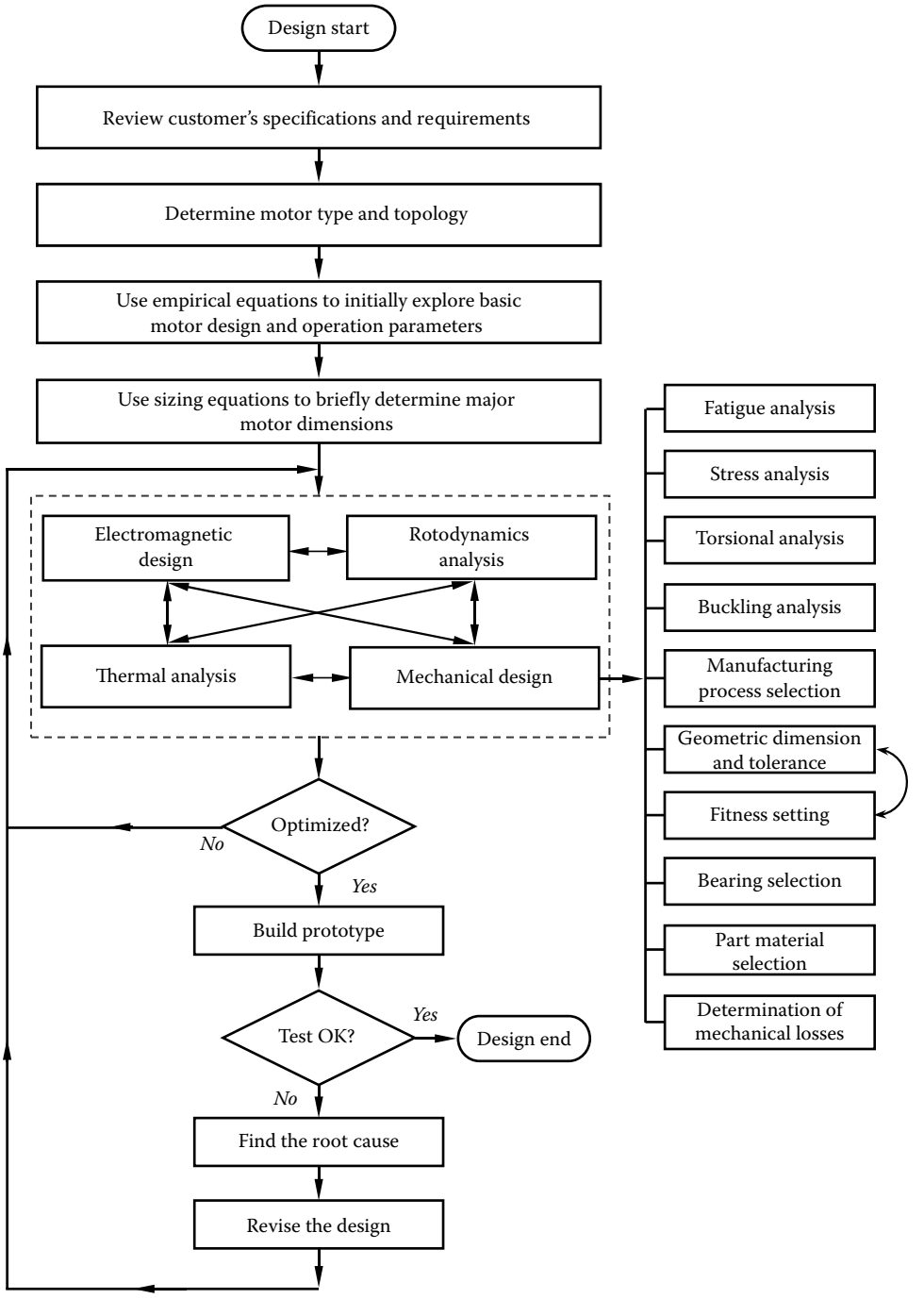


FIGURE 1.49
The flowchart of motor design.

Hesai Exhibit 1031, page 105

The calculated power losses from electromagnetic design are passed into thermal analysis. With the fast development of computational technologies in last several decades, it becomes more popular today for thermal analysis to be executed by means of advanced computational fluid dynamics (CFD) software package. In the current CFD market, many powerful commercial software packages are available, either for general-purpose or specific applications, to solve heat transfer and fluid dynamics problems. CFD methods are used to provide detailed information of motor cooling such as temperature distribution, velocity field, pressure drop, and thermal interaction between solid surfaces and fluid flows in electric motors and to confirm and refine the preliminary designs. However, performing a CFD analysis still remains a challenging task and requires highly skilled engineers due to the complexities involved, such as the complexity of highly nonlinear partial differential equations, CFD model meshing, large amount demanding of computational resources, determination of models and parameter of turbulent flows, convergence of simulation, and complexity of inputs and related tasks. Extensive design iterations exist between the thermal, mechanical, and other design processes for achieving the optimized motor design.

Mechanical design consists of a number of design components:

- Fatigue analysis—As a material is subject to cyclic loads, a fatigue crack nucleus may be initiated on a microscopically small scale, and then the crack grows to a macroscopic size, finally causing material failure suddenly at a stress level below the material ultimate tensile strength or even yield tensile strength. During normal operation, some motor components are subject to alternative loads. One example is the motor shaft that is subject to gravitational force from the rotor core and drag force from a load machine (e.g., using belt or gear driving). When the rotor rotates, the shaft experiences cyclic loads creating a compression–expansion cycle for every revolution.
- Mechanical stress/strain analysis—The performance of mechanical stress/strain analysis is to determine whether the motor structure or components can withstand various loads without failure. The analysis allows for design optimization to achieve the best performance of electric motors. Due to the increasing complexity of modern motors, the stress/strain analysis today is mostly carried out using 3D finite element models. Recently, 3D finite element models have been used to perform thermomechanical analyses for providing more complete and accurate simulation results.
- Torsional analysis—When a motor is in normal operation, it may be subject to torsional vibration due to various causes. In analytically determining the torsional response, it requires to calculate the torsional resonance frequencies of the motor. To do this requires the torsional stiffness and mass inertia of both the motor and the load system, as addressed previously.
- Buckling analysis—Bucking is a phenomenon that a structural member subjected to compressive stress suddenly fails due to the loss of its stability. The actual compressive stress of the structural member at the point of failure is much lower than the ultimate strength that the material can withstand. For instance, some motors have a number of ventilation ducts within the lamination core (either stator or rotor or both). These ventilation ducts are constructed by I-beams or Π-beams that function like spacers. It is highly desired to perform bucking analysis to avoid the instability failure of beams.
- Determination of manufacturing process for motor components—Each motor part can be fabricated by many different manufacturing processes. The selection

of manufacturing process is based on many factors, including cost-effectiveness, productivity, process cycle time, availability, motor part geometry, requirements for part strengths, vibration and dampness, surface finishing, material properties, porosity/void, and environment friendliness. As an example, for a large volume of motor endbells, the most suitable manufacturing method is die casting.

- Determinations of geometric dimension and tolerance (GD&T)—The purpose of GD&T is defined as describing the geometric requirements for part and assembly geometry. Proper applications of GD&T can ensure allowable part and assembly geometry defined on the drawings leads to right size, shape, form, location, orientation, and location, define the proper assembly of mating parts, and enable assembled motors to function as intended. Mechanical engineers are responsible to provide the information for drafters and manufacturing engineers for the fabrication of motor components and the layout of motor assembly. Based on the information, 3D solid models can be generated to explicitly describe nominal geometries and their allowable variations at both the component level and system level.
- Setting up an appropriate fitness between mechanically mating components—Improper fitness between mating parts may greatly reduce the motor lifetime or lead to motor failure. For example, a loose fitting between a shaft and a bearing can cause the relative movement between the two components, resulting in a fast wear of the shaft and thus high heat generation due to the friction. In addition, an improper fitness may cause motor vibration.
- Selection of motor bearings—It has been reported that more than a half of motor failures can be attributed to the failure of bearings. Among them, improper selection of bearings is one of the major causes, usually dominated by wrong grease or lubrication selection.
- Selection of materials of motor components—Material selection in the motor industry is an artful balance between material performance, motor reliability, manufacturing process, and total cost. Apparently, material selection can heavily influence the motor performance and lifetime. None of the materials is perfect to satisfy all requirements. It is the responsibilities of engineers to make trade-offs between various design factors, for example, material performance and cost, and strength and durability.
- Determination of mechanical, electric, and electromagnetic power losses—The mechanical losses in a motor usually consist of friction losses and windage losses. The electric losses (also referred to as copper losses) represent all the resistive losses in motor windings and cables and are usually the main loss component in electric motors. The electromagnetic losses (i.e., iron losses), including eddy current and hysteresis losses in motor cores and other components, are often obtained through electromagnetic analysis conducted by electrical engineers. All the information is requested by thermal engineers for performing CFD or numerical heat transfer analysis.

Design optimization is the top goal of motor design. When the preliminary design is complete, the engineers need to examine whether the predetermined optimum objectives of the design have been reached. For different applications, these optimum objectives can be different, namely, the maximum peak torque, the highest torque density ratio, the most efficient cooling, and the total weight. If the design does not satisfy certain requirements,

the design engineers need to review their designs to identify the design gap and do the design iteration again until all optimum objectives are gained.

The next step is to build a prototype motor for validating the motor design as a whole unit. This is the critical step for the success of products. No matter how perfect the results achieved from engineering analyses are, prototype motor testing is an essential and necessary step to validate the conceptual design and ensure normal motor operation. Only after a series of successful tests, the motor can start mass production.

1.6.2 Design Integration

An electric motor usually consists of dozens to even thousands of components. In order to increase motor operation reliability, reduce the manufacturing cost, and simplify the assembly process, it is desired to integrate some of the components together in one self-contained package, including (1) motor-transmission device (e.g., gearbox, belt, chain) integration, (2) mechanical–electromagnetic system integration, and (3) motor control (e.g., feedback, drive).

As one of the most important constituents in an automated electric driving system, an electronic drive is used to deliver a useable form of power to control the performance of a servomotor for an end user. As presented in Figure 1.50, the integrated motor drive assembly system includes a motor, a fan, and a drive unit. One remarkable advantage of such integrated assembly systems is their compactness in size and ease of installation into a small industrial or other application. Generally, the drive is disposed on the motor or arranged in an integral housing with the motor. Both the motor and drive are shared with the same cooling fan [1.79].

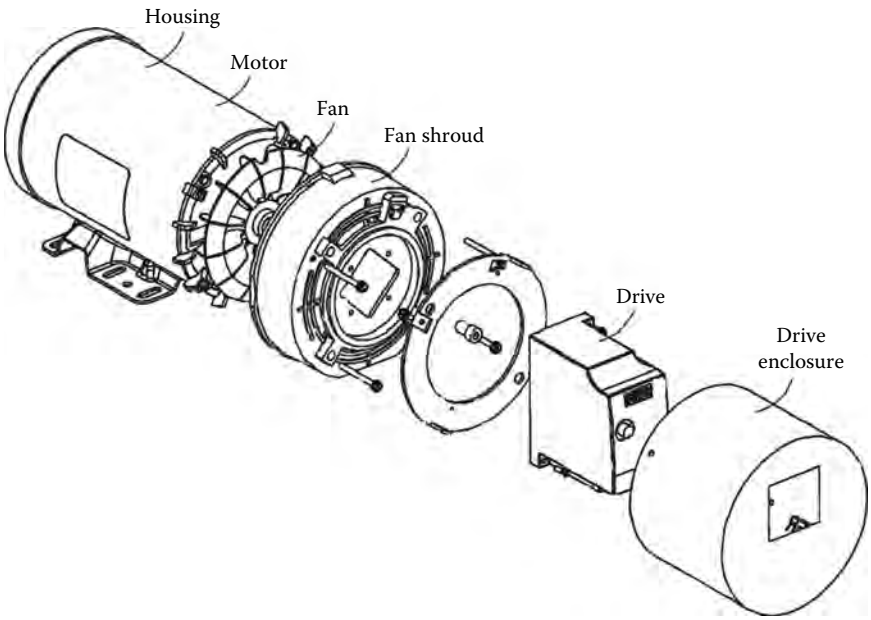


FIGURE 1.50
Integration of electric motor, drive, and fan as one unit (U.S. Patent 7,362,017) [1.79]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

A growing trend in motor control is the integrated system that combines motor, feedbacks, and controller in one self-contained package. By integrating the controller with the motor, the possibility of mismatch is eliminated. Based on the integrated modular motor drive concept, an integrated traction drive has been developed recently. The integration of motor and drive offers a number of attractive features such as reduced drive volume and the elimination of power transmission cables. Correspondingly, it reduces radiated electromagnetic interference and voltage transient due to power transmission over long cable distances [1.80]. Furthermore, the integration of motor and drive can offer fault-tolerant features not possible with conventional drives [1.81,1.82]. On the caution side, significant thermal management and analysis are required to protect the electronic drive and motor combination.

1.6.3 Mechatronics

Mechatronics is the confluence of classical engineering disciplines with mechanical, electrical, and electromagnetic engineering, sensor technology, drive and actuator technology, control theory, and computer science. In recent years, one of the design trends in the electric machine industry is to design intelligent mechatronics products. Mechatronics is not just a new design strategy but also a new way of doing business to gain competitive advantage in the global market. As a result, it has been extensively used to design improved products and processes [1.83].

In fact, the mechatronic design approach that involved multidisciplinary is expected to become a key technology to gain a competitive advantage in the era of modern manufacturing. The development of mechatronics will therefore be crucial to the continued competitiveness of national economies.

The magnetic hard disk drive (HDD) is believed to be one of the most successful examples of modern mechatronics systems. In a state-of-the-art HDD servo system, a high-speed servomotor, a read/write head, a data storage disk, and other components are designed as one unit. The precision position control of HDD enables the tolerance to be less than one micrometer while operating at high speed [1.84]. As the mechatronics technology advances, new application areas of motor mechatronics can be developed in promising directions, such as mobile robots and medical devices.

1.7 Motor Failure Modes

There are a number of different approaches to assess motor failure modes. Generally, there are three primary failure modes identified from motor root cause analyses. The most important failure mode is electrical-related failure, caused by the breakdown of motor winding insulations, overloading condition, winding short circuit, and other issues. In a typical case, the breakdown of insulation material is attributed to exceeding of the temperature limit of windings. An overloading condition can cause the winding current exceeding its limit to damage coils. The second failure mode is magnetic failure, which usually resulted from the thermal demagnetization of PM (for PMMs). The thermal demagnetization of PM may be attributed to high motor operation temperature, shock, vibration, or strong magnetic field generated by a stator. The third failure mode is mechanical failure of motor structure and components. In this failure mode, bearing failures are encountered frequently. Another common failure mode is loose/broken rotor bars. Usually, most

Hesai Exhibit 1031, page 109

mechanical failures result from slow processes such as mechanical wear and gradual degradation of material properties. During the process, the defective parts continue to display characteristic signs until they completely fail. This is different from most electronic component failures that happen suddenly and without warning.

Statistics have shown that despite the reliability and simplicity of construction of IMs, annual motor failure rate is conservatively estimated at 3%–5% per year, and in extreme cases, up to 12%, as in the pulp and paper industry [1.85]. The downtime in a factory due to motor failure may lead to a high cost, even exceeding the cost of motor replacement. In some instances, a motor failure may cause the entire production line to stop and thus interrupt the production process.

Based on the data collected from 1141 motors, those larger than 200 hp and less than 15 years in service, *IEEE* published a long report in three parts [1.86–1.88] to address motor failure modes according to the root causes of failure. In this report, Part I presented general results based on categories of motor types. Part 2 combined various categories and addressed some questions resulting from Part 1. Part 3 of the survey results was to address new questions and comments and to add more specific analyses of areas not yet explored previously.

A similar investigation of motor failure modes has been conducted by *Electric Power Research Institute* (*EPRI*) [1.89]. Though the approach of *EPRI* is different from *IEEE* (*EPRI* focused on failed motor components), their results are consistent. The integration of these two surveys was made by Venkataraman et al. [1.85], as presented in Table 1.6.

TABLE 1.6

Comparison of Failure Modes for Large-Size Motor

***IEEE* Survey**		***EPRI* Survey**		
Failure Contribution	**%**	**Failure Contribution**	**%**	**Average (%)**
Persistent overload	4.2	Stator ground insulation	23	Electrical-related failure
Normal deterioration	26.4	Turn insulation	4	
		Bracing	3	
		Core	1	
		Cage	5	
Electrical-related total	30.6	*Electrical-related total*	36	33.3
High vibration	15.5	Sleeve bearings	16	Mechanical-related failure
Poor lubrication	15.2	Antifriction bearings	8	
		Thrust bearings	5	
		Rotor shaft	2	
		Rotor core	1	
Mechanical-related total	30.7	*Mechanical-related total*	32	31.35
High ambient temperature	3.0	Bearing seals	6	Failures related to environmental maintenance and other reasons
Abnormal moisture	5.8	Oil leakage	3	
Abnormal voltage	1.5	Frame	1	
Abnormal frequency	0.6	Wedges	1	
Abrasive chemicals	4.2			
Poor ventilation cooling	3.9			
Other reasons	19.7	Other components	21	
Environmental reasons and other reasons total	38.7	*Maintenance-related and other parts total*	32	35.35

Data from *IEEE* and *EPRI* motor reliability surveys.

From the table, it can be seen that the most common reasons of the motor failures are bearing failures and failures related to the stator insulation. High vibration can lead to mechanical failures. Furthermore, many failures are related directly or indirectly to high operation temperature.

1.8 IP Code

Electric motors are designed to withstand various harsh environments such as flammable and explosive environments, high ambient pressure or vacuum, extreme temperatures, and high shock loads and vibrations. In some cases, they may operate in space or under deep water.

The IP code is an international standard for electric devices. It applies to the classification of degrees of protection provided by enclosures for electrical equipment against the insertion or intrusion of solid and liquid objects, external influences, or conditions such as dust, moisture, water, icing, corrosive solvents, and mechanical impacts. The IP code consists of two digits and optional letters. The indications of digits and optional letters are shown in Tables 1.7 and 1.8, respectively [1.90].

TABLE 1.7

Indications of IP Digits

First Digit: Protection against Ingress of Particles			Second Digit: Protection against Ingress of Water	
IP	**Protection of Person**	**Protection of Equipment**	**IP**	**Protection of Equipment**
0	No protection	No protection	0	No protection
1	Protected against contact with large areas of the human body (back of hand)	Protected against objects over 50 mm in diameter	1	Protected against vertically falling drops of water, e.g., condensation
2	Protected against contact with fingers	Protected against solid objects over 12 mm in diameter	2	Protected against direct sprays of water up to 15° from vertical direction
3	Protected against tools and wires over 2.5 mm in diameter	Protected against solid objects over 2.5 mm in diameter	3	Protected against sprays to 60° from vertical direction
4	Protected against tools and wires over 1 mm in diameter	Protected against solid objects over 1 mm in diameter	4	Protected against water sprayed from all directions (limited ingress permitted)
5	Protected against tools and wires over 2.5 mm in diameter	Protected against dust (limited ingress, no harmful deposit)	5	Protected against low-pressure jets of water from all directions (limited ingress permitted)
6	Protected against tools and wires over 2.5 mm in diameter	Totally protected against dust	6	Protected against strong jets of water
			7	Protected against the effects of immersion between 15 cm and 1 m
			8	Protected against long periods of immersion under pressure
			9K	Protected against ingress of high-temperature (steam) and/or high-pressure water

Hesai Exhibit 1031, page 111

TABLE 1.8

Indications of Optional Letters in IP

First Letter (Optional)		Second Letter (Optional)	
Letter	**Protected against Access to Hazardous Parts with**	**Letter**	**Indication**
A	Back of hand	H	High-voltage device
B	Finger	M	Device moving during water test
C	Tool	S	Device standing still during water test
D	Wire	W	Weather condition

TABLE 1.9

Typical IP Codes Used for Different Motor Applications

IP Classification	Application
IP00	Open motor
IP12	Open drip-proof motor [1.91]
	Drip-proof motor [1.91]
IP13	Splash-proof motor [1.91]
IP21	Elevator motor
IP23, IP24	Weather protected motor [1.91]
IP42	Commercial refrigeration application
IP44	Totally enclosed fan-cooled motor [1.91]
	Totally enclosed pipe-ventilated motor [1.92]
IP54	Regular for commercial applications (fans, blowers)
	Totally enclosed force-ventilated motor [1.91]
	Totally enclosed air-to-air–cooled motor [1.92]
	Totally enclosed air-over motor [1.91]
	Totally enclosed air-to-water–cooled motor [1.91]
	Totally enclosed water-cooled motor [1.92]
IP55	Waterproof motor [1.92]
IP65	Motor used in hybrid vehicles
	Elevator motor
	High-speed spindle servomotor
	Some direct drive motor
IP67	Motor used under severe operation conditions (e.g., pitch and yaw control motors in wind turbines, driving motors in chemical reactors and in machine tools)
	Motor used outdoor under harsh environments
	Motor used in aerospace and defense applications (military vehicle, submarine, satellite, etc.)
	Motor used in food processing, beverage, and pharmaceutical applications
IP69K	Motor operating under ultra-harsh or aggressive environments (e.g., autoclaving, high-pressure spray, and frequent washdown with caustic chemicals)
	Motor used in deep water such as in propelling systems
	Liquid immersion pumps

Hesai Exhibit 1031, page 112

The IP code has a strong impact on the motor design. With the IP defined, a number of design factors, such as motor construction, insulation, sealing method, coating, cooling technique, and component material, must be specifically determined to satisfy the requirements and regulations of the IP code.

As a reference, common IP codes used in electric motors in practice are summarized in Table 1.9.

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Hesai Exhibit 1031, page 117

2

Rotor Design

An electric motor primarily consists of a rotor and a stator, which are separated by an air gap and integrated together via bearings and a motor frame. The rotor is the rotating component to drive an external loading machine, and the stator is the stationary component to produce a rotating magnetic field.

Rotor design involves a wide range of knowledge, skills, and experience across various disciplines including material science, rotor dynamics, electromagnetics, structural mechanics, aerodynamics, manufacturing technology, tribology, thermal engineering, and fluid dynamics. Furthermore, during the rotor design, a large number of design parameters must be carefully considered and determined.

As discussed previously, an AC motor can be primarily classified as either an induction motor (IM) or a permanent magnet motor (PMM), depending on how the rotor is constructed. Because the structure and the working principle of these two types of motors are fundamentally different, IM and PMM are addressed separately in the following sections.

2.1 Rotor in Induction Motor

A typical rotor in an IM comprises a cylindrical core that is made up of silicon steel laminates and pressed onto the cylindrical motor shaft. There are two types of IMs: wound rotor IMs and squirrel cage rotor IMs. The primary difference between these two types of IM motor is the design of the motor rotor. While the design of stators for all IMs is almost the same, the rotor construction is quite different for different types of IMs.

2.1.1 Wound Rotor

In a wound rotor, a set of insulated rotor windings, which are similar to the stator windings, are utilized to accept external impedances. A set of rotor windings are correspondingly connected to a set of insulated slip rings mounted on the rotor shaft. The carbon-composite brushes riding on the slip rings connect to a set of external rheostats (Figure 2.1). This rotor construction design allows the external rheostats to be placed in series with the rotor windings. These rheostats, varying from almost short-circuit condition to an open-circuit condition with infinite external resistance, can remarkably improve motor starting characteristics, of which low inrush current is being the most significant.

Wound rotor IMs have some distinct advantages over other types of IMs. A special feature of the wound rotor motor is its capability to control the motor velocity at different load levels by controlling the rotor currents. By varying the rheostat resistance, it is possible to change the slip and thus to change the rotor rotating velocity. This feature has been used to improve the motor performance. For instance, a wound rotor has ability to gradually bring up to speed for high-inertia and large load equipment.

Hesai Exhibit 1031, page 118

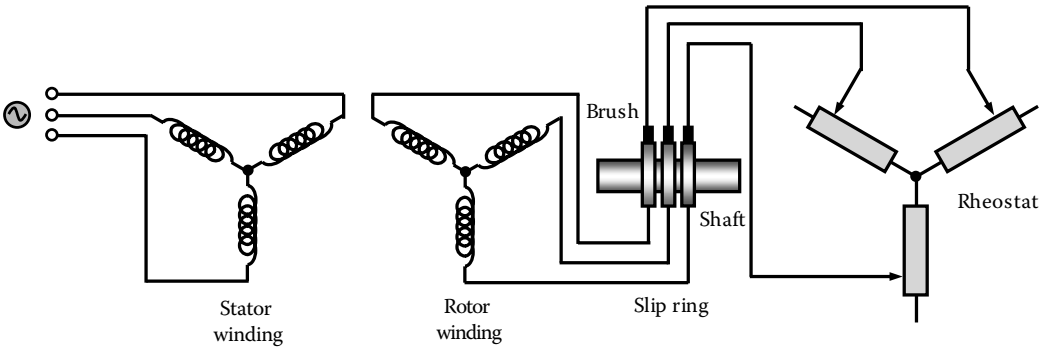


FIGURE 2.1
Wound rotor IM using slip rings and brushes to contact rotor winding and external resistances for improving motor start characteristics

Long motor life is ensured with the use of external resistor banks during motor start-up. For these reasons, the wound rotor IMs are conventionally used for high-power industrial applications.

Using this feature can optimize the torque–speed characteristics for different applications [2.1]. The impact of the rotor resistance on the torque–speed characteristics is shown in Figure 2.2. It can be seen that increasing rotor resistance can result in the shift of the breakdown torque toward zero speed. This can eventually alter motor performance characteristics: (1) The increase in the rotor resistance causes the significant

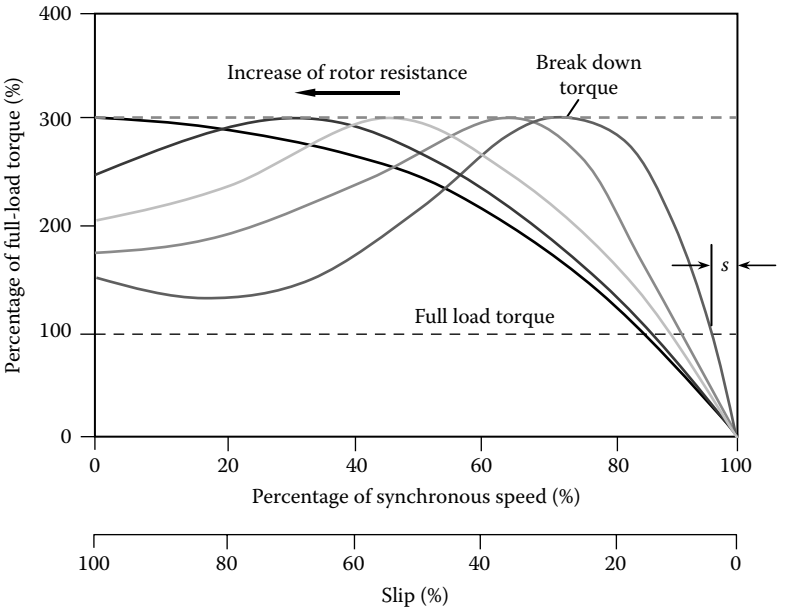


FIGURE 2.2
Variations of torque–speed characteristics of wound IMs as a function of rotor resistance. Increasing rotor resistance results in the shift of the breakdown torque toward zero speed, causing the increase of slip and starting torque and the decrease of full-load speed.

increase in the starting torque and the decrease in the corresponding starting current. (2) A higher rotor resistance leads to the reduction of the rotating speed of the rotor, that is, the increase of slip. (3) For a given full-load torque, the motor full-load speed decreases as the motor rotor resistance increases. However, the reduction of the full-load speed in such a way is highly undesirable since it lowers the motor efficiency and power output as the increase in the resistive losses (i.e., I^2R losses). (4) Under high rotor resistance conditions, the pull-up torque may be eliminated from the torque–speed curves. In fact, a high rotor resistance causes the motor torque to decrease more rapidly than those with lower rotor resistances. (5) The higher rotor resistance is always related to the larger variation in the rotation speed as the torque varies. This indicates that wound rotor IMs having high rotor resistances are more susceptible to speed variations as the load torque changes [2.2].

The rotor windings are made up of copper coils inserted into rotor slots. Essentially, the rotor windings exhibit inductance and resistance, and these characteristics can effectively depend on the frequency of current flowing in the rotor. Today, the wound rotor IMs are less used in industrial applications due to frequent maintenances for slip rings and brushes, high rotor inertia, and requirement of bulky resistor banks.

2.1.2 Squirrel Cage Rotor

Most IMs used today are of squirrel cage type. A squirrel cage rotor consists of a rotor core made by stacked laminations and a shaft fitted in the rotor core. A number of equally spaced open or closed slots are punched at the outer circumference of the laminations (Figure 2.3). In fact, the selection of open or closed rotor slot design involves the trade-off because each design has its own advantages and disadvantages, depending on different applications. Open rotor slots allow a substantial reduction of the rotor leakage inductance with the possibility to increase the constant power range speed regulation. As contrast, closed slots tend to produce lower surface loss and have higher rotor leakage inductance for a better filtering in presence of inverter supply [2.3,2.4].

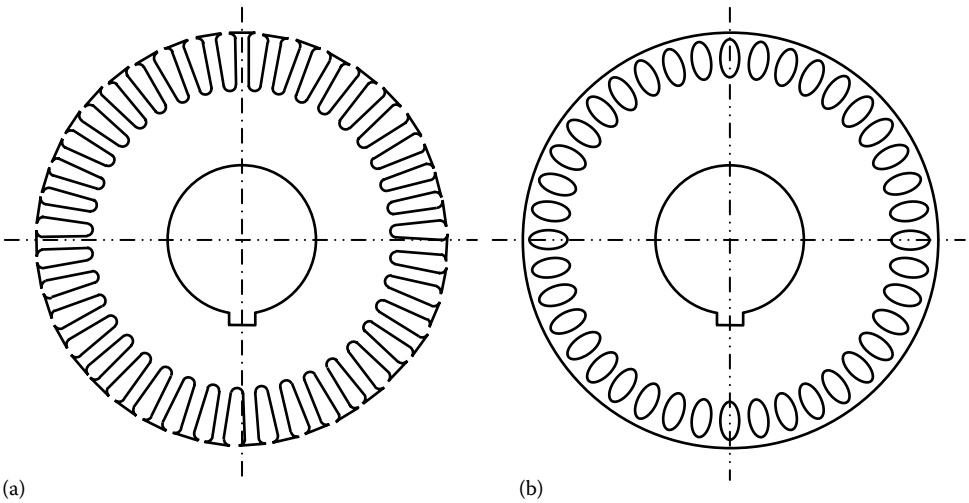
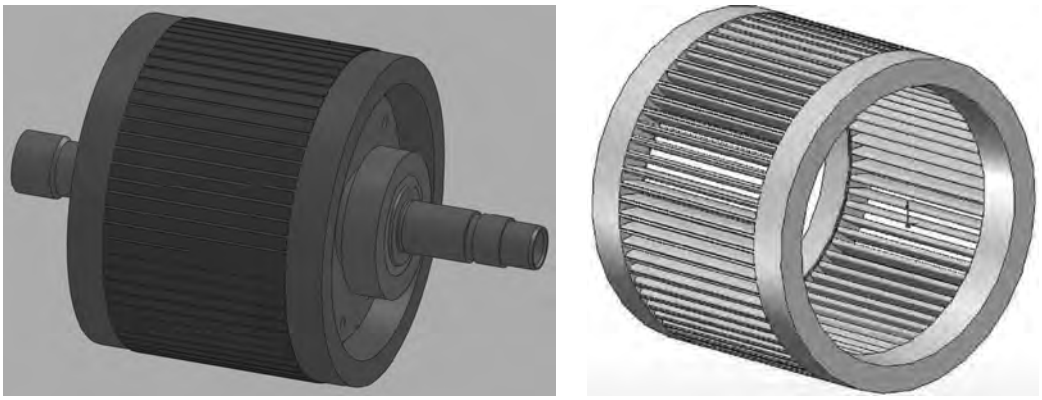


FIGURE 2.3
Rotor slots of a squirrel cage motor: (a) open slots and (b) closed slots.



(a) (b)

FIGURE 2.4
Casted copper rotor assemble: (a) the rotor is assembled by integrating two hollow shafts with the casted copper rotor core. (b) Conducting bars and end rings form a closed loop for inducted current as the steel rotor laminations are removed from the rotor core.

A transient 2D FEA has shown that the closed slot rotor geometry gives less spatial rotor harmonics and less stator current harmonics, leading to a significantly lower level of stator vibrations for frequencies above 1 kHz of induction machines [2.5]. In addition, closed slot rotor helps improve the die-casting process.

Rotor conducting bars are made by casting a conducting material (usually aluminum or copper) directly into rotor slots. These casted conducting bars are usually slightly skewed to the axis of the rotor for reducing cogging torque and connected together with an end ring at each of the rotor end to form a closed electric circuit. Thus, the stator windings set up a rotating magnetic field around the rotor to induce electric current in the conductive bars and end rings. In turn, the induced currents in the conducting bars/end rings react with the magnetic field of the stator to produce forces acting at a tangent orthogonal to the rotor, resulting in continuous torque on the motor shaft to drive external loads. As shown in Figure 2.4, by removing the steel laminations, the conducting bars and end rings form a squirrel cage. Thicker end rings can not only reduce current densities but also increase the heat dissipation capability of the rotor.

2.1.2.1 Factors Impacting Resistance of Squirrel Cage Rotor

There are a variety of factors impacting characteristics and performance of squirrel cage rotors, including the conducting bar material, number of conducting bars, cross-sectional area, bar configuration and shape, skew angle, and bar position relative to the rotor surface.

A conducting bar with a large cross-sectional area exhibits a low resistance. Moreover, the electric conductivity of copper is nearly 60% higher than that of aluminum. Thus, it is commonly accepted that copper rotor bars are superior over aluminum bars because of lower power losses, better performance, and higher motor efficiency. It has been reported that the incorporation of casting copper for conducting bars and end rings in place of aluminum would result in attractive improvements in motor energy efficiency through reductions in motor losses ranging from 15% to 20% [2.6]. With an FEA software, Daut et al. [2.7] have analyzed and investigated the performance of the

three-phase 0.5 hp IM using either copper or aluminum rotor bars. The FEA results fall into four groups: torque versus speed, torque versus slip, power loss versus speed, and power loss versus slip. Comparisons between copper and aluminum rotor bars have been performed in these four groups and shown that the copper rotor bars excel in all investigated performance issues.

The position of conducting bars in a squirrel cage IM is an important design parameter to affect the motor performance. The impedance of a conducting bar is comprised of both the resistance and inductance. According to the position of the conducting bar relative to the rotor surface, the conducting bar exhibits different magnetic leakage flux and inductance. Positioning the conducting bars near the rotor surface (i.e., near the stator) will produce a relatively small leakage flux and a small inductance. In contrast, positioning the conducting bar deeper radially into the rotor slots (i.e., away from the stator) will result in more leakage and a higher inductance and consequently a higher impedance. Due to the skin effect, at high frequencies, the AC impedance of the outer portion of the conducting bar is lower than that of the inner portion.

2.1.2.2 Double-Cage Rotor

The invention of double-cage rotor was motivated to develop a type of motor that is characterized by a variable rotor resistance during motor operation. This type of motor offers preferably high rotor resistance at the motor start-up, high locked torque, low starting current, and high pull-up torque, while maintaining higher efficiencies when compared to the same power ratings with single-cage designs. For the specific rotor designs and operating conditions, the comparison of torque–speed curves of double-cage and single-cage rotors is presented in Figure 2.5.

As presented in Figure 2.6, a double-cage consists of a large cage buried deeply in the rotor and a small cage located near the rotor surface. The conducting bars in the large cage are loosely linked with the stator and have low resistance and high leakage inductance. In contrary, the conductors in the small cage are closely linked with the stator and have high resistance and low leakage inductance. At the motor start-up stage, only the conductor

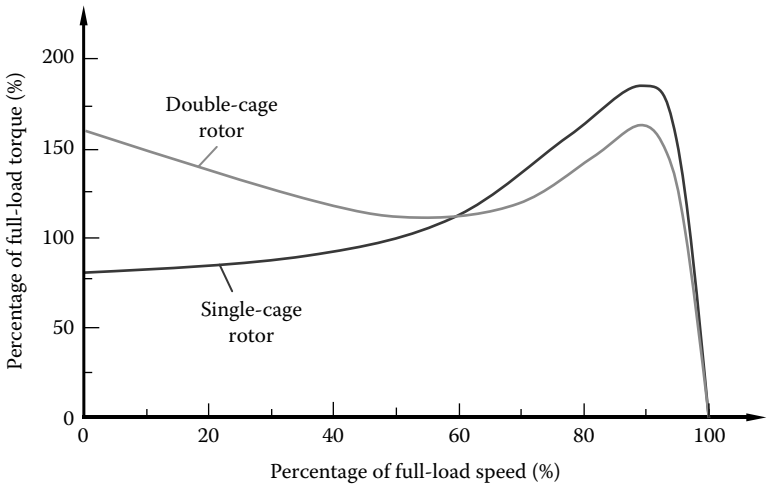


FIGURE 2.5
Comparison of torque–speed curves between double-cage and single-cage rotors.

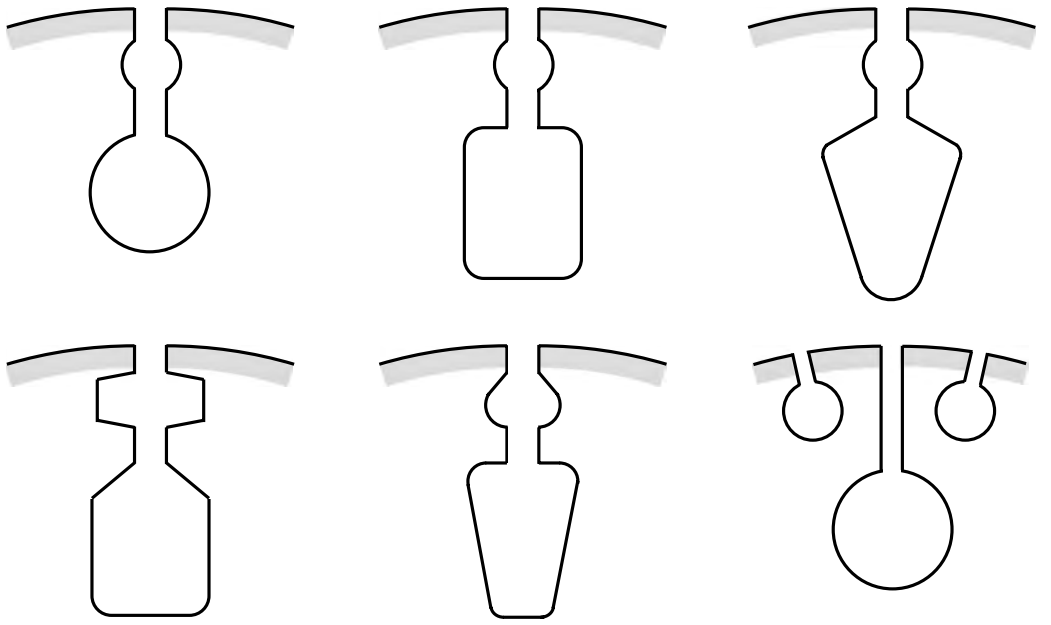


FIGURE 2.6
Various double-cage rotor designs for improving motor start-up and normal operation.

bars in the small cages that have high resistance are effective, providing high starting torque. In the motor normal operation stage, the conducting bars in both large and small cages become effective. Because the two resistances are connected in parallel, the rotor resistance is

$$R_r = \frac{R_i R_o}{R_i + R_o} \tag{2.1}$$

where R_i and R_o are the resistances of the conductors in the inner and outer cages, respectively. In a case that $R_o \gg R_i$, $R_r \approx R_i$. Therefore, under normal operation conditions with low rotor frequencies, the rotor current mainly goes through the conducting bars in the inner cage with lower I^2R losses. For this reason, some motor manufacturers use a high-resistance material such as brass for the outer cage and a low-resistance material such as copper for the inner cage.

Some large-size motors with double-cage rotors may employ separated end rings to connect the inner and outer cage, respectively. One of the reasons for this is to reduce thermal stresses induced in the end rings due to the different coefficients of thermal expansion of the inner and outer cages.

2.1.2.3 Casting Squirrel Cage Rotor

Several casting processes can be used for making squirrel cage rotors, including centrifugal casting and high- and low-pressure die casting. Low-pressure die casting is usually used for large rotors. In a casting process, the stack of laminations is placed in a die-casting

TABLE 2.1

Comparison of Properties of Copper and Aluminum

Property	Copper UNS C10100[a] [2.8]	Aluminum 383 [2.9]
Density (kg/m^3)	8890–8940	2700
Hardness, Vickers	75–90	85
Yield tensile strength (MPa)	69–365	152
Ultimate tensile strength (MPa)	221–455	310
Elongation at break (%)	55	3
Modulus of elasticity (GPa)	115	71
Poisson ratio	0.31	0.33
Shear modulus (GPa)	44	27
Machinability (%)	20	70
Melting temperature (°C)	1083	660
Thermal conductivity (W/m·K)	391	96.2
CTE (μm/μm/K)	1.692×10^{-5} (20°C–100°C)	2.11×10^{-5} at 20°C
	1.728×10^{-5} (20°C–200°C)	2.25×10^{-5} at 250°C
	1.692×10^{-5} (20°C–300°C)	

[a] Due to casting, C10100 is assumed the annealed properties.

die or a permanent mold containing a space at the top and bottom for the simultaneous casting of end rings. The molten material is poured or forced into the mold.

Two conducting materials are commonly used in casting squirrel cage rotors: (1) aluminum, which is the popular casting material of choice by most motor manufacturers due to its low melting temperature (660°C), low cost, and low density, which means that an aluminum rotor is subject to less starting inertia, less stress from centrifugal forces, and less vibration, and (2) copper, which is the promising material to significantly reduce rotor power losses as the result of high electric conductivity. The melting temperature of copper is 1083°C. The comparison of properties of copper and aluminum is presented in Table 2.1.

It is well known that incorporation of copper for the rotor bars and end rings in place of aluminum would result in attractive improvements in motor energy efficiency. Manoharan et al. [2.10] have reported that for the same output power, by merely replacing an aluminum rotor with a copper rotor and keeping all other parameters unchanged, the motor efficiency is increased by nearly 2.6%. It is observed that this increment in efficiency in the copper rotor is attributed to the higher electrical conductivity of the copper rotor. For instance, under a temperature of 20°C, the electric resistance of copper is 44% lower than that of aluminum. Hence, for the same current, the substitution of aluminum by copper can result in 44% reduction in the electric resistance loss.

Figure 2.7 presents an example of applying copper rotor in vehicles. It is benefited with low resistance losses and low starting torque and increased torque density. This leads to small motor size with the same power output or more power output for the same motor size. Furthermore, the much simpler design of a copper rotor compared to an aluminum rotor can significantly lower the manufacturing costs [2.11].

It was reported [2.12] that the advantage of using die-casting copper motors is not only the reduction in power losses but also the increase in power density. The die-cast copper rotor motors would be 18%–20% lighter and 14%–18% less expensive than the aluminum rotor motor at the same efficiency when a frame size reduction is possible.

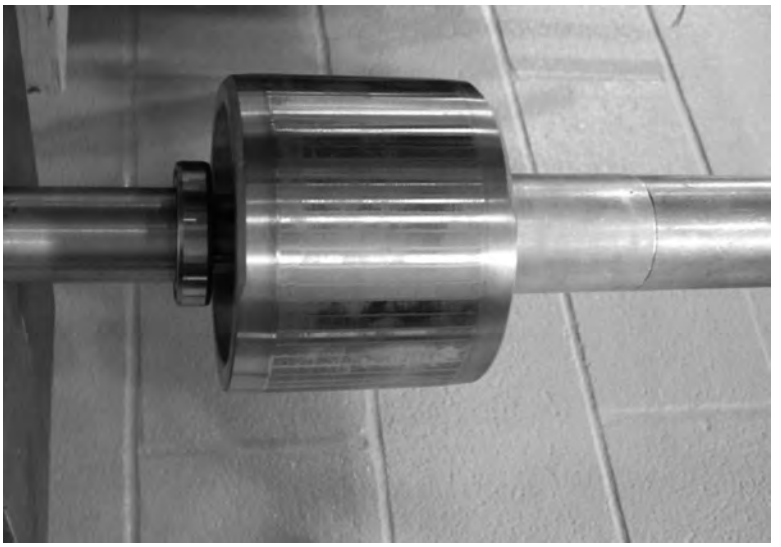


FIGURE 2.7
Rotor with casted copper conducting bars and end rings.

However, two difficulties must be addressed before cast copper rotors are fully commercialized: (1) Casting copper into a rotor is still a challenge today due to high casting temperatures (>1000°C) that are detrimental to the rotor laminations. (2) Copper has a higher coefficient of thermal expansion (CTE) than aluminum. As a motor passes through temperature cycles during its lifetime, the difference in CTE between copper conductors and steel laminations may develop fatigue stresses at the interfaces between the rotor conducting bars and stator core slots. The fatigue stresses may initiate cracks on the conducting bars and eventually lead to fatigue failure of the conductor bars. In some cases, it may result in the separation of the conducting bars/end rings from the rotor core. For the same reason, sometimes breaks may occur at the joints of the conducting bars to the end rings.

The advantages of squirrel cage IMs include their low cost, ruggedness, and maintenance free operation. However, since the rotor conducting bars and end rings are permanently short-circuited, it is unable to add any external resistance in series with the rotor circuit for starting purpose, leading to high starting current and low starting torque. In fact, this is the major disadvantage of the squirrel cage motors. Because squirrel cage IMs have good constant speed characteristics, they are preferred for driving machinery with constant operating speeds such as fans, blowers, water pumps, grinders, lathes, and printing and drilling machines.

As a rotor is subjected to a very high rotating speed, a retention mechanism may be required to keep the end rings connected to the rotor against high centrifugal forces. A number of solutions are effective in reducing the stress concentration at certain areas and are also effective in keeping the rotor together at the high speed. These include the following: (1) Each end ring can be secured using a band of a high-strength material. (2) Rivets can be cast into the rotor to secure the end rings to the body. (3) Specially designed laminations can be used for securing the end rings in rotors used today with the rotor body. (4) By breaking the end ring into sections, limiting the thermal growth of each section, thus reducing the thermal growth of the entire the ring. The specially designed geometry of the end ring in rotor used today helps contain the end ring at high speed by acting like a band as used in rotors today [2.13].

2.1.2.4 Skin Effect

In power systems, skin effect refers to the tendency of AC current to concentrate near the outer surface of a conductor and to decay exponentially toward the center. This indicates that the induced current in conducting bars is unevenly distributed over the cross-sectional area of the conducting bars. The current density increases from the center to the outer surface of the bar. Because of the skin effect, as well as the proximity effect between the conductor bars, the nonuniform current distribution causes more resistive loss. As a result, the AC resistance of a conducting bar is much higher than its DC resistance. Generally, the AC resistance increases with the increase of the frequency of rotor currents. The distribution of current density in a round conducting wire is given as

$$J = J_o e^{-d/\delta} \tag{2.2}$$

where

J_o is the current density at the conductor surface
e is the base of the natural logarithm ($e = 2.718$)
d is the distance measured from the surface toward the center of the conductor
δ is the skin depth, which is defined as the distance at which the current decreases to e^{-1} ($e^{-1} = 0.3679$) of its original value

The skin depth δ is inversely proportional to the square root of the frequency (in Hz).

2.1.3 Induction Motor Design Types and Their Performing Characteristics

The National Electrical Manufacturers Association (NEMA) standard has specified four major design classifications of AC IMs: designs *A*, *B*, *C*, and *D*. These design classifications apply particularly to the rotor design and hence affect the performance characteristics of the motors. Figure 2.8 displays a series of torque–speed curves for these design classifications with identical power rating and rotating speed [2.14]. It illustrates that the design characteristics such as starting torque, speed, and breakdown torque can be combined in different ways:

- Design *A* motors are used for applications that may undergo frequent but short-time overloading and thus require high breakdown torque, such as injection molding machines. This type of motor has normal starting torque (typically 150%–170% of the rated torque) and relatively high starting current.
- Design *B* motors are general-purpose motors, designed for applications that require normal starting and running torques, such as machine tools, fans, and centrifugal pumps. This type of motor usually has a comparable starting torque with Design *A* but offers low starting current. The locked rotor torque is good enough to start many loads encountered in the industrial applications. The slip is about 5%.
- Design *C* motors are selected for applications that require high starting torque, such as inclined conveyors and gyratory crushers.
- Design *D* motors, called *high slip* motors, are sometimes used to power hoists and cycling loads, such as oil well pump jacks and low-speed punch presses.

Hesai Exhibit 1031, page 126

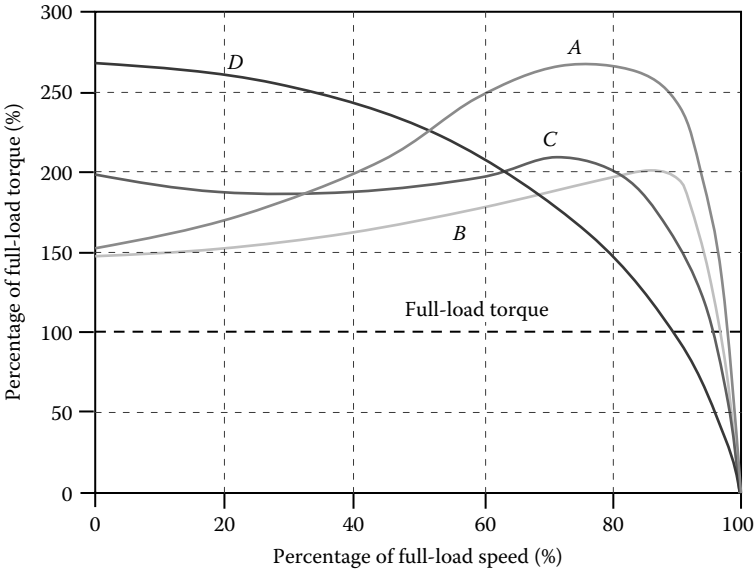


FIGURE 2.8
Torque–speed characteristics for design *A*, *B*, *C*, and *D* motors defined by NEMA.

Note that the starting torque ranges broadly among the design classifications, from about 150% of full-load torque for Design *A* motors to about 260% of rated torque for Design *D* motors. In addition, the speed at full-load for Design *A* motor is several percent less than the Design *D* motor.

2.2 Permanent Magnet Rotor

A PMM adopts PMs attached to the rotor in place of rotor windings or conductors/end rings to generate magnetic fields. The first motor invented by Faraday in 1821 was practically a primitive PM DC machine [2.15]. The first attempt to introduce rare-earth PM motors into the market of electric machines was in the 1980s using samarium–cobalt (Sm–Co) magnets that were invented in 1960s. Since then, more manufacturers have successfully launched rare-earth PM motors into the market. Neodymium–iron–boron (Nd–Fe–B) magnets were added into the rare-earth PM family in the mid-1980s. Like Sm–Co, Nd–Fe–B has demonstrated many design advantages and superior performance over common PMs. Because Nd–Fe–B is more abundant than Sm–Co, it is extensively used in energy-conversion devices today. In the recent years, modern PM motors have presented a momentum of rapid development. However, PM motor applications have been generally limited to relatively low-power motors, usually less than 500 kW, in most cases, less than 100 kW.

2.2.1 Discovery of Phenomenon of Magnetism

Ancient Chinese discovered the phenomenon of magnetism. Utilizing the magnetism of natural lodestone, the first compass was invented around the fourth century BC and was



(a)

(b)

FIGURE 2.9
Invented compass in ancient China: (a) Appeared on a Chinese stamp of 1953. (b) A replica.

called *south pointer* or *Si Nan* (司南). In the book of *Guiguzi* (鬼谷子), which was written in the early third century BC, it stated that "the lodestone makes iron come or it attracts it." The author Wang Yu (a thinker and strategist at approximately 300 BC) noted that in addition to its main purpose of serving as designators of direction, the Chinese primarily used it to order and harmonize their environments and lives (it refers to as *Feng Sui*—风水). The compass could be carried with jade hunters to prevent them from getting lost during their journeys.

The earliest Chinese records show the spoon-shaped compass *south pointer* could be dated back to sometime during the Han Dynasty (from the second century BC to the second century AD). As shown in Figure 2.9, a spoon-shaped lodestone is placed on a cast bronze plate that had the 8 trigrams (i.e., *Ba Gua*—八卦), as well as the 24 directions (based on the constellations) and the 28 lunar mansions (based on the constellations dividing the equator). Because the shape of the spoon bottom is spherical, the spoon can rotate freely on the bronze plate. Toggling the spoon to rotate, when it stops, its handle will point to the south.

2.2.2 Permanent Magnet Characteristics

The use of PMs in electric motors is highly desirable since they have great potential for reducing motor weight and increasing efficiency. In a PMM, the magnetic field of rotor

is produced by PMs. Because there is no need of rotor windings or conducting bars, the motor structure is greatly simplified. However, due to the limited magnetic strength of PMs, IMs are still dominant in large-size electric motors.

PMs are key parts in PM motors to generate magnetic fields along with stator windings. The advantages of PM motors include the following:

- Having higher efficiency than IMs
- Simplifying the rotor structure
- Requiring less maintenance
- Lowering the rotor inertia under the same power rating
- Eliminating the need for field exciter and slip rings
- Reducing cooling requirements due to less rotor losses
- Offering more precise speed control
- Lowering bearing currents
- Achieving higher power/torque density

With the fast developments of advanced magnetic materials and continuous improvements in operation performance, PM motors have been extensively used in various military and industrial applications. However, in comparison with conventional IMs, PM motors are generally more expensive due to high costs of magnets, especially for the rare-earth magnets.

The most important relationship of a magnetic material is the nonlinear behavior between the magnetic flux density **B** and the magnetic field strength **H**, where **B** and **H** are measured in gauss and oersteds and are analogous to current and voltage in an electric circuit, respectively. A typical **B**–**H** hysteresis loop of a magnetic material is shown in Figure 2.10. This hysteresis loop is useful for evaluating the response of a magnetic material to the applied magnetic field. It is readily noted the **B**–**H** hysteresis loop is skew-symmetric with respect to the *H*-axis. An unmagnetized PM is composed of a large number of small magnetic domains that randomly oriented and cancel one another, resulting in a net magnetic moment of zero on a large scale. This indicates that for the material in the unmagnetized state, both **B** and **H** are at the origin in Figure 2.10. During a magnetizing process, these randomly oriented domains are forced to be aligned with respect to the applied external magnetic field, and the magnet is thus magnetized following the initial magnetization curve (as market as curve #1). As it reaches the point of magnetic saturation, an additional increase in **H** will produce little increase in **B**. When driving magnetic field strength **H** to zero (curve #2), the magnet still retains a considerable degree of magnetization because of hysteresis of the magnet. This phenomenon is referred to as residual magnetism and the corresponding magnetic flux density is denoted as $\mathbf{B}_r$. To move the magnetic flux density **B** to zero, **H** must be reversed. This is referred to as coercivity (curve #3—also refers to demagnetization curve). The force required to remove the residual magnetism from the magnet is called the coercivity force, expressed as $-\mathbf{H}_c$ in Figure 2.10.

As the magnetic field strength continues to decrease in the negative direction, the magnet is magnetized again but in the opposite direction (curve #4). Similarly, as it reaches the point of magnetic saturation (but in opposite direction), any changes of −**H** in the negative direction will result in little variation of −**B**. From this point, increasing **H** back in the positive direction will lead the increase of **B** (curve #5). As **H** becomes zero again, the material

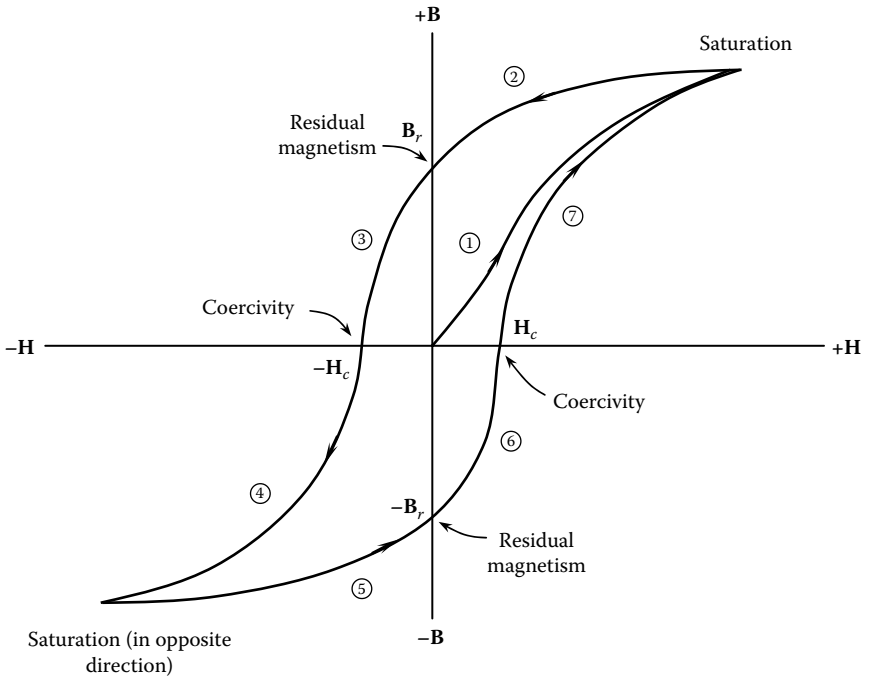


FIGURE 2.10
The magnetization hysteresis loop of a PM.

still remains negative flux density $-\mathbf{B}_r$. Further increasing **H** will bring **B** to zero (curve #6) and reach the point of magnetic saturation (curve #7). It is noted that this curve did not pass through the origin because the positive magnetic field strength is required to remove the residual magnetism.

A magnetized PM may lose partly or completely its magnetism under certain circumstances. The demagnetization of the PMs in a motor can cause performance degradation or complete failure of the motor.

The electromagnetic properties of PMs are strongly influenced by temperature [2.16]. An increase in temperature can increase kinetic energy of the molecules in a magnetic material, causing them to wiggle and jiggle around and the domains to lose alignment. Studies have shown that temperature variations normally produce changes in the hysteresis loop width, induction amplitude, maximum differential permeability, and other magnet properties [2.17], impacting the performance of PM motors. An FEA was conducted to investigate the influence of PM materials and temperature on the performance of interior PM motors [2.18]. The results show that along with the increase in temperature, remanent flux density $\mathbf{B}_r$ decreases, resulting in the decrease of the average torque and cogging torque but the increase in torque ripple.

2.2.3 Permanent Magnet Materials

PMs have been used in electric machines for more than a century. However, the modern history of PMs started in the 1930s with the introduction of alnico. As the strongest type of magnets at that time, alnico alloys dominated the market of electric machines due to their

TABLE 2.2

Magnetic Characteristics for Various Magnetic Materials

Magnetic Material	Grade	B_r (Gauss)	H_c (Oe)	H_{ci} (Oe)	BH_{max}[a] (MGOe)	T_{max} (°C)	T_c (°C)	Reference
Ferrite magnet	8	3,900	3,200	3,250	3.5	300	450	[2.19]
Nd–Fe–B	39H	12,800	12,300	21,000	40	150	310	[2.19]
Sm–Co	26	10,500	9,200	10,000	26	300	750	[2.19]
Alnico	5	12,500	640	640	5.5	540	860	[2.19]
Ceramic	8	3,900	3,200	3,250	3.5	300	460	[2.19]

[a] BH_{max} represents energy product, given in MGOe. Nd–Fe–B magnets have the highest energy product of any magnetic materials available below 200°C.

excellent remanence, high thermal stability, and high chemical stability until the development of rare-earth magnets in the 1970s.

In the last several decades, dramatic improvements have been made in magnetic properties and availability of PMs. Applications of PMs in electric machines are increasingly growing. There are several main types of PM materials used in motor manufacturing, including ferrite PMs, alnico PMs, and rare-earth PMs. The comparison of magnetic characteristics for various magnetic materials is presented in Table 2.2.

2.2.3.1 Ferrite Permanent Magnets

Ferrite PMs are made from some oxides (SrO, BaO, or Fe_2O_3). Due to low costs of raw materials and simple manufacturing technique, the magnets are considerably cheap. The corrosion resistance of the oxides is considered excellent, and no surface treatments are required. Ferrite PMs can work under severe environmental conditions such as high temperature, dust, salt fog, and moisture. Therefore, they are extensively used in automobile motors and power tools.

Alnico magnets are alloys made of aluminum, nickel, cobalt, copper, iron, and other materials by either sintering or casting. The remarkable advantage of alnico magnets is that they have the widest range of temperature stability and the lowest temperature coefficient of any commercial magnet materials (0.02% per degree centigrade). Their maximum working temperatures can be up to 450°C–550°C. Due to good corrosion resistance, there are no surface treatments required. This type of magnet is usually used in temperature-sensitive devices such as instruments and sensors.

2.2.3.2 Rare-Earth Permanent Magnets

Rare-earth PMs are made from alloys of rare-earth elements such as neodymium (Nd) and samarium (Sm). In comparing with regular ferromagnets, rare-earth magnets have higher residual magnetic flux density and much higher coercivity but lower Curie temperature. Today, rare-earth magnets are widely used in various PM motors for producing high-efficiency, high-torque-density, and high-performance motors and reducing motor vibration and noise. Because Sm–Co magnets can work at higher temperatures (up to 250°C–350°C) than Nd–Fe–B magnets (usually 80°C–220°C), they are more suitable for manufacturing various high-performance PM motors. In addition, due to the strong anticorrosion ability, electroplating of Sm–Co magnets is generally unnecessary. However, because of the low reserve of samarium on the earth, Sm–Co magnets are very expensive.

Among rare-earth magnets, Nd–Fe–B magnets offer the largest energy product of any magnetic materials available below 200°C (392°F). Compared to Sm–Co and alnico magnets, Nd–Fe–B magnets are capable of picking up more than 600 times its own weight. Nd–Fe–B magnets have not only the excellent qualities of high remanent magnetization, high coercive force, and high magnetic energy product but also the better mechanical properties for processing of complex shapes. According to the current trend, the Nd–Fe–B magnetic material may become the dominant magnetic material in the future in electric machines, energy, transportation, medical devices, computers, home appliances, and other fields. However, because the material contains large amounts of neodymium and iron, the Nd–Fe–B magnets are easily corroded and thus must be coated with nickel (Ni), zinc (Zn), gold (Au), chromium (Cr) epoxy resin, etc. One of the disadvantages of Nd–Fe–B magnets is its low performance temperature. The magnetic loss at high temperature is relative large.

2.2.3.3 New Developments of Permanent Magnets

In the last two decades, new materials have played a major role in the technological innovations needed to propel the rapid development of high-performance, high-powered electric machines that are ever lighter and smaller.

The rapid developments of nanotechnology in recent years shed a new light on research and development of nanoalloy magnets. Nanoalloy magnets are especially interesting to the motor industry because it can further reduce the power density of electric motors. Previous studies showed that arrays of nanomagnets displayed various magnetization patterns that are closely related to the nature of the magnetic interactions between neighboring nanomagnets [2.20,2.21].

It is known that the strength of a magnet is measured in terms of coercivity and saturation magnetization values. These values increase with a decrease in the grain size and an increase in the surface to volume ratio. Therefore, the magnets made of nanocrystalline yttrium–samarium–cobalt grains may possess very unusual magnetic properties due to their extremely large surface area [2.22].

Japanese engineers and scientists have committed to developing new types of magnets for replacing rare-earth magnets. The government-affiliated New Energy and Industrial Technology Development Organization and Hokkaido University announced in 2010 that it had developed motors for hybrid vehicles that used cheap and readily available ferrite magnets [2.23].

More recently, a team of researchers mainly from Tohoku University and Toda Kogyo Corporation has succeeded in generating $Fe_{16}N_2$ powder with a purity of 91% and a reproducibility for the first time in the world. Thus, a new powerful magnet can be made from the material using the synthesizing technology developed by Toda Kogyo without any rare-earth element. The new magnet is aimed to be commercialized in 2023 [2.24]. Toshiba recently announced that the company has developed a high-iron-concentration samarium–cobalt (Sm–Co) magnet that is free of dysprosium, a rare-earth mineral. The magnet exceeds the heat-resistant neodymium magnet in magnetic force by 1% at an operating temperature of 100°C and by 5% at 150°C [2.25].

At the end of 2012, a company based in the United States [2.26] announced that it has made a breakthrough in the development of non-rare-earth magnet electric motor. The key design is the patent-pending motor geometry, which in part defines the shape and magnetization direction of the PMs. This new technique provides an alternative way in PM motor designs. According to this company, the electromagnetic design and analysis task was completed. The next phase is the mechanical design and the concept unit will be built in 2013.

Scientists at University of Minnesota have developed a new type of magnets using the iron nitride ($Fe_{16}N_2$) magnet and achieved 130 megagauss–oersteds (MGOe), which is more than twice the maximum reported magnet energy product for a rare-earth neodymium magnet [2.27].

2.2.4 Magnetization

The full performance of PMs can only be guaranteed if it is magnetized properly to saturation. The magnetization of PMs can be implemented either prior to or posterior to rotor assembly. Premagnetized magnets can easily achieve 100% of saturation. This approach may be especially suitable for some cases. For example, since a large PM motor is hard to place in a magnetizer, the premagnetization of magnets is the only solution. However, there are some apparent disadvantages for premagnetized magnets. First, the magnetized magnets are difficult to handle during rotor assemblies because of its attraction to steel parts or attraction/repulsion with other magnets. Second, premagnetized magnets may pick up magnetically permeable dirt in their assembly and transportation processes. Third, magnets are usually brittle and easily damaged in shipment if they are not packaged properly. Therefore, it is preferred to magnetize assembled systems at the assembly locations. However, in such a way, the magnetized saturation level may be slightly lower (about 99%) due to the fact that the magnets are too close to one another [2.28].

Usually, postmagnetization works efficiently for surface-mounted PM and interior PM that are located near the rotor surface. In the motor industry, because of the high production efficiency and ease of assembly, a majority of small PM motors are postmagnetized by means of magnetizers, as depicted in Figure 2.11.



FIGURE 2.11
Postmagnetizing of a PM motor by means of a magnetizer.

Hesai Exhibit 1031, page 133

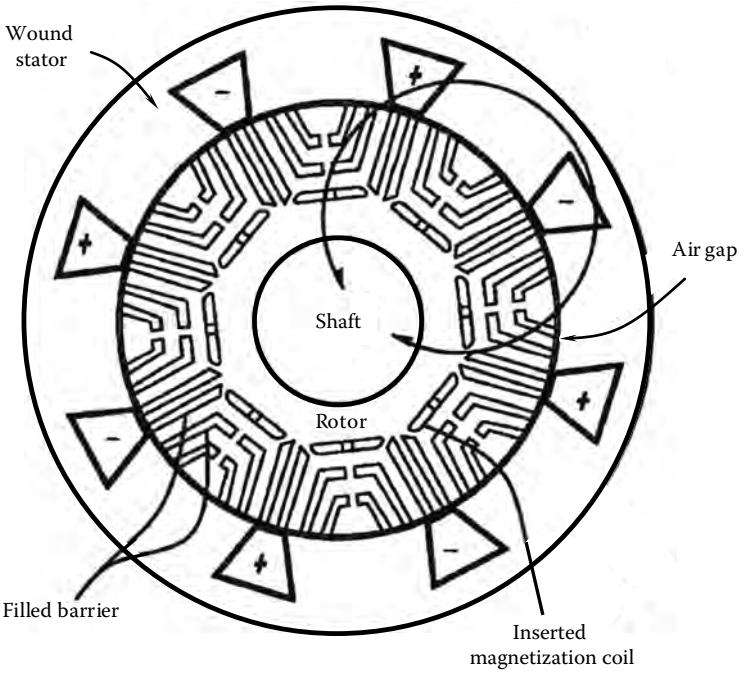


FIGURE 2.12
Cross-section of a multilayer interior magnet motor with a magnetizing auxiliary winding for magnetizing interior magnets (U.S. Patent 6,674,205) [2.29]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

For multilayer buried magnet rotors, postmagnetization would result in a large amount of magnetic material buried deep within the rotor to be only partially magnetized or even not magnetized at all. To solve this problem, an innovative method is presented in Figure 2.12. It removes magnetic material from the regions of the rotor that cannot be effectively or strongly magnetized during the postmagnetization process and inserts magnetizing coils. Thus, the inserted magnetizing coils enhance the magnetizing field produced by the stator or other magnetizing fixture for improving the rotor magnetization. Then, the magnetizing coils are removed from the rotor [2.29].

2.2.5 Factors Causing Demagnetization

During assembly process, magnetic properties of PMs may change due to the improper installation processes and changes in environmental conditions. As a result, the variation in magnetic properties can directly affect the reliability and performance of electric motors. A primary cause of demagnetization of PMs is high temperature. When a PM is heated, the internal energy increases to make the atoms in the PM vibrate actively. The vibration of the atoms may shift the alignments of some atoms from an ordered pattern to a nonaligned disordered pattern. As a result, the PMs may be partially demagnetized. In a worse case, excessive temperatures may lead the PMs to completely lose their magnetic properties, causing electromechanical failure, power loss, or erratic power fluctuation. This is especially true for rare-earth PMs.

The temperature effect on magnet demagnetization usually falls into three categories:

1. Reversible losses: At the early stage, the rise in temperature will gradually decrease the magnetism of a magnetic material. For example, the reduction in magnetism of Nd–Fe–B magnets is 0.13% per degree Celsius. At this stage, the losses in magnetism can be recovered automatically as the magnetic material returns to its original temperature.
2. Irreversible but recoverable losses: Continuously increasing the temperature will result in the irreversible drop in the magnetism due to the reduction in the coercive force. The losses in magnetism can be only recovered by remagnetizing. This is also applicable for the partial demagnetization due to the interference of external magnetic fields.
3. Irreversible and unrecoverable losses: Each magnetic material has a specific temperature, the Curie temperature T_c, at which all magnetic properties are completely lost. Because the crystal microstructure of magnetic materials changes at the Curie temperatures, remagnetization cannot bring its magnetism back again. When the temperature is much higher than the Curie temperature, the lattice of the magnetic material will be melted and recrystallized, resulting in the permanent loss of the characteristics as the permanent magnetic material.

The impact of temperature on the demagnetization characteristics of PMs has been extensively investigated by many researchers and engineers. This is normally studied by looking at the **B**–**H** hysteresis loop at the second quadrant. A demagnetization curve represents the relationship of the magnetic flux density generated by a magnet and the demagnetization force imposed on the magnet. As examples, the demagnetization curves for a typical ferrite PM under various temperatures are shown in Figure 2.13 [2.30]. It can

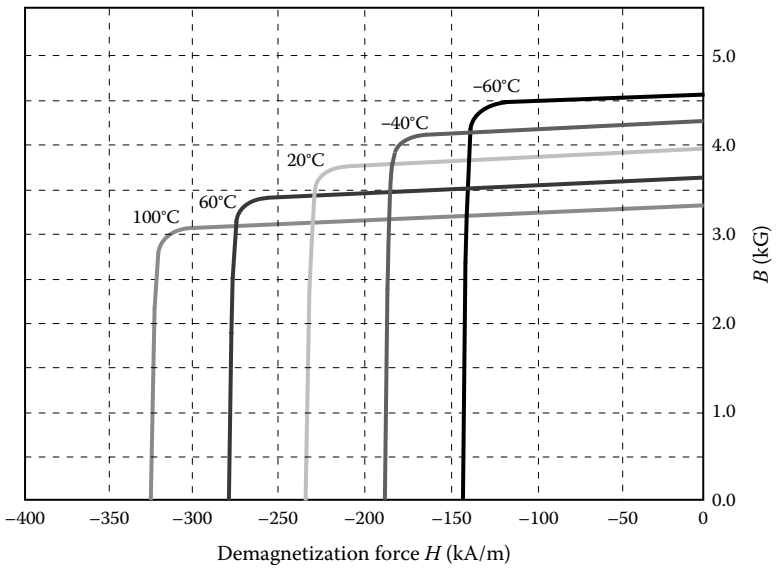


FIGURE 2.13
Impact of temperature on demagnetization characteristics of a typical ferrite magnet.

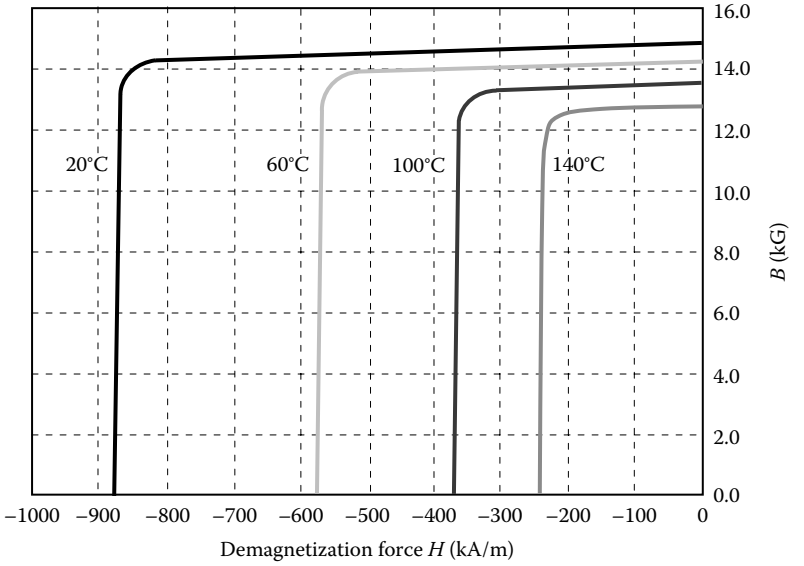


FIGURE 2.14
Impact of temperature on demagnetization characteristics of a typical Nd–Fe–B magnet.

be seen from this figure that the shapes of all demagnetization curves are similar. These curves decrease slowly initially with the increase in **–H** and then drop off abruptly when the *knees* are passed. At the knee points, the irreversible demagnetization starts to take place. The knee occurs at approximately –130 kA/m at the temperature of –60°C but occurs –320 kA/m as the temperature is raised to 100°C. Increasing operation temperature from –60°C to +100°C, the demagnetization curves in sequence reduce in both their magnetic field strength **H** (i.e., move toward **–H**) and magnetic flux density **B** (i.e., move toward **–B**).

For a typical Nd–Fe–B PM, the demagnetization curves are shown in Figure 2.14 [2.31]. Unlike in ferrite magnets, the temperature impact on the magnetic characteristics of Nd–Fe–B magnets is somewhat different. When the operation temperature increases from 20°C to 140°C, the demagnetization curves increase in their magnetic field strength **H** (i.e., move toward **+H**), while they decrease in their magnetic flux density **B** (i.e., move toward **–B**). This indicates that for a PM motor subject to a large range of operating temperature, its maximum torque capacity and efficiency of the motor can vary over a wide range.

Furthermore, it has been observed that the magnetic properties change with temperature due to the change in the magnetic domain structure. The effect of the operational temperature on Nd–Fe–B magnets has been investigated on a 2.5 kW, 24-pole surface PM prototype motor by Beniakar et al. [2.32]. Their results show that the rise of the operational temperature from 20°C to 140°C causes a significant decrease of the air gap flux density, thus deteriorating the machine's overall performance.

In addition to temperature, other factors can also cause demagnetization of magnetic materials. A strong external magnetic field can distort the magnetic field of a PM, causing some atoms to orient their electron spins to conform to the external magnetic field. This will turn some magnetic domains in the PM from an aligned orientation to a random orientation. In fact, it is possible to demagnetize a PM by applying an external magnetic field in a direction opposite to the direction of magnetization of the PM.

Mechanical disturbances also tend to randomize the magnetic domains and thus to demagnetize PMs. Thus, it is important to avoid strong mechanical shocks such as hammering or jarring on PMs. It is usually preferable to ship PMs in an unmagnetized state to prevent damages to themselves and things close to them. In order to safely and reliably ship magnetized PMs, special care must be taken to ensure that the PMs are packaged well with damping materials.

Radiation-induced demagnetization of PMs has been studied for various insertion devices [2.33,2.34]. It has been reported that Sm–Co magnets exhibit significant demagnetization when irradiated with a proton beam of 10^9–10^{10} rad. Nd–Fe–B test samples were shown to lose all of their magnetization at a dose of 7×10^7 rad and 50% at a dose of 4×10^6 rad [2.35]. When a PM motor is subjected to electron beam or γ-ray irradiation, this type of losses must be carefully considered.

Studies have demonstrated that PMs are affected by time, known as magnetic creep. PMs undergo changes immediately after magnetization. At this stage, the PMs are less stable, showing significant variations in magnetic properties by fluctuations in thermal and magnetic energy. After a period of time, the PMs become more stable as unstable magnetic domains decrease.

The time effect on modern PMs is minimal. It has been reported that over 100,000 h, the losses are in the range of essentially zero for samarium–cobalt materials to less than 3% for alnico five materials at low permeance coefficients [2.36]. Due to the extremely high coercivities, rare-earth magnets experience a much less time effect than other types of magnets. Furthermore, most losses of rare-earth magnets are reversible or recoverable.

2.2.6 Maximum Operation Temperature

Maximum operation temperature is defined as the temperature below which magnets can operate normally with a small demagnetization. For PM motors, the maximum operation temperature provides an important reference temperature for the design of motor cooling. It is important to maintain the maximum operating T_{max} well below the Curie temperature T_c for preventing losses in magnetism, as well as the structural or mechanical damage of magnets. In fact, for a particular magnetic material, T_{max} is determined by not only the magnetic material but also the material processing process.

2.2.7 Permanent Magnet Mounting and Retention Methods

In practice, PMs may be either mounted on the rotor outer surface or inserted into retention slots passing axially through the rotor core. A motor with surface-mounted PM is simple in construction and easy to manufacture. However, the magnets in this design are exposed to all the time and space harmonics of the stator winding magnetomotive force (MMF).

For surface-mounted PM motors, one of the most important issues is to hold the magnets in place firmly to prevent them from flying off during motor operation [2.37]. Some surface mounting methods are presented in Figure 2.15, where PMs are positioned either directly on the rotor surface or on the sunken portions made on the rotor surface.

A retention device must be capable of withstanding both high centrifugal forces due to high rotating speed of the rotor and the axial forces that may arise during manufacturing, assembly, or installation. In addition, the retention device must have low permeability to avoid short-circuiting the magnetic flux, low electric conductivity to minimize space and time harmonic losses, and sufficient strength to provide for magnet retention and must be thin enough to not increase the air gap.

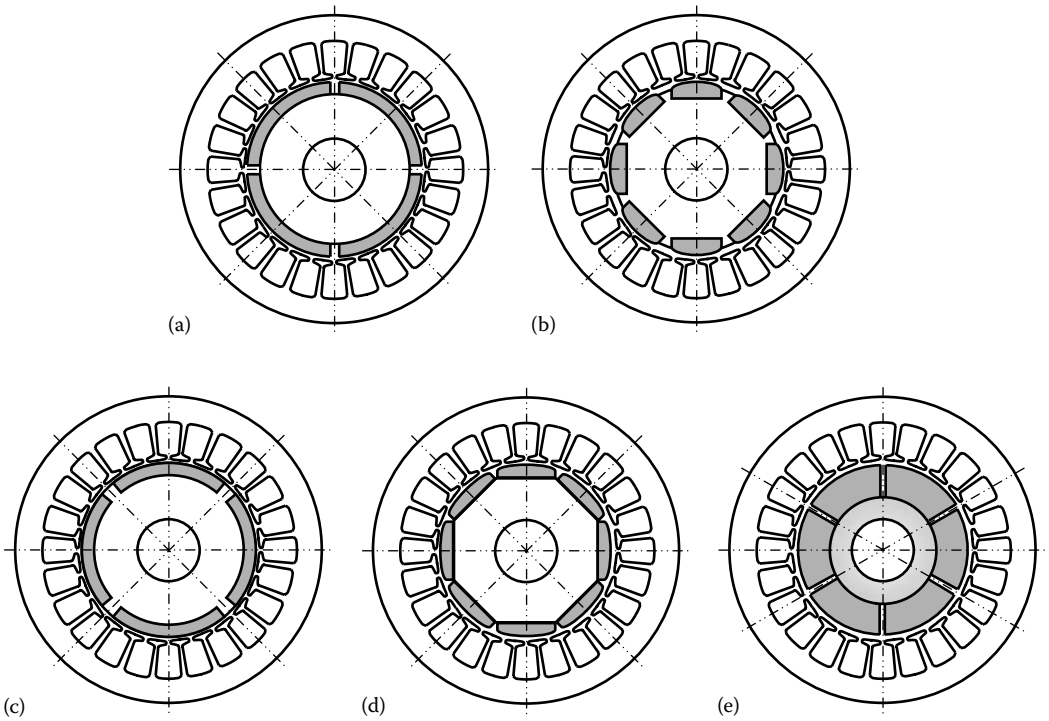


FIGURE 2.15
Various mounting methods on rotor surface: (a) tile-shaped magnets mounted on the rotor surface, (b) bread loaf magnets partially positioned in the sunken portions of the rotor, (c) tile-shaped magnets fully positioned in the sunken portions of the rotor, (d) bread loaf magnets mounted on the surface of an octagon rotor, and (e) thick magnets mounted on the thin rotor core, generating strong magnetic field.

PMs can be mounted on rotor surfaces by either mechanical retaining methods (e.g., using clamps, wedges, and wrapped fibers), chemical retaining methods (e.g., using adhesives), physical retaining methods (e.g., using soldering), or combinations of these methods. A common method is to glue arc magnets on the rotor core surface with adhesives and secure the magnets with reinforcing fine fiber (e.g., Kevlar™ aromatic polyamide fiber) or fiberglass bands/strings wrapped around the magnets (Figures 2.16 and 2.17). This technique offers a simple and effective solution for the magnet assembly but decreases the effective air gap and consequently increases the magnetic leakage flux. In order to minimize eddy-current losses in the magnets, it is desirable to use multiple small magnets. However, the installation of small magnets increases the working load, leading to low efficiency.

Adhesives provide strong bonds between magnets and the rotor core. Thus, the magnets become load-bearing members, thus subject to centrifugal load that may potentially exceed their strength limitations. Additionally, the bonded magnets become permanently attached to the rotor, making it difficult or otherwise infeasible to replace them, wholly or individually. Adhesive is also susceptible to failure due to extreme temperatures, aging, and chemical exposure [2.38].

An innovative retention method [2.39] combines the two magnet mounting methods, that is, surface-mounted and slot insertion methods, as shown in Figure 2.18. This method utilizes a number of dovetail slots that are made circumferentially on the rotor core surface

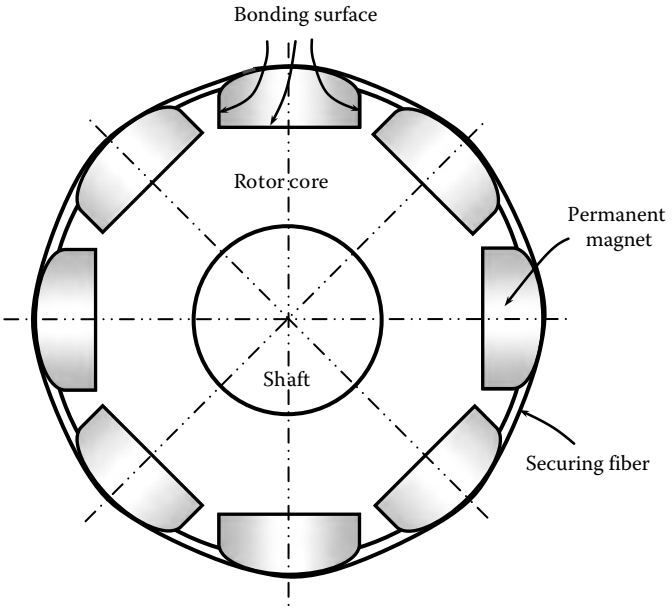


FIGURE 2.16
Gluing PMs on rotor surface and securing with wound special fiber string outside the magnets.

to hold wedge-shaped magnets robustly. Correspondingly, the magnet shape must match with that of the dovetail slots. Because magnets are usually fragile and unable to withstand large forces, the interference fit is inapplicable in assembling the magnets into the dovetail slots. Instead, the magnets are fixed in the dovetail slots by an adhesive.

Compared to mechanical retention methods, soldering takes less space that can be critical for high-power-density motors. It has a lesser outgassing rate than gluing. However, the primary disadvantage of soldering in comparison to mechanical attachment and gluing is the necessity to raise the temperature of the PM to above the solder melting point during the soldering process, which can easily demagnetize the magnets. Therefore, special measures in magnet cooling must be carefully considered and taken.

Another alternative method for the magnet retention is to use high-strength metallic sleeves over the magnets [2.40], as shown in Figure 2.19. The metallic sleeve is advantageously resistant to high temperature and easy to process with a high productivity. The thin-wall metallic sleeve has a capability of restraining the magnets against centrifugal forces. By extending the sleeve beyond the ends of the rotor core and bending the sleeve toward the magnets, the metallic sleeve thus prevents the magnets from shifting in the axial direction (Figure 2.20). However, because of electrically conductive properties of the material, the metallic sleeve produces an eddy-current loss, which in turn reduces the efficiency of the motor. To minimize the influence of the metallic sleeve on motor performance, the sleeve is preferably made of the materials with low conductivity such as titanium or chromium alloys. However, these materials are expensive and hence impractical. In practice, stainless steel and corrosion-resistant Inconel™ alloys that contain primarily nickel, chromium, and iron are commonly used for metallic sleeves. For small motors, the thickness of the sleeve preferably ranges from about 0.1 mm (0.004 in.) to about 0.6 mm (0.024 in.). For large motors, the thickness can go up to 3 mm (0.12 in.). With such a thin wall, the sleeve can be mounted over magnets by various operations such as heat shrinking or pressing



FIGURE 2.17
PM rotors are commonly wound with reinforcing fine fibers to provide large mechanical strength for retaining PMs on rotors.

fitting [2.41]. However, the use of metallic sleeves increases the thermal resistance and thus lowers the rotor cooling efficiency.

Magnets can be also retained by a molded plastic cylindrical sleeve. As illustrated in Figure 2.21, the magnets are encapsulated entirely by the plastic sleeve. This method provides a perfect solution for the magnet corrosion problem and increases the magnet retention strength against the centrifugal forces [2.42].

The schematic diagram for PMs embedded in the interior of a lamination rotor core is shown in Figure 2.22. There are a number of advantages with this technique over the surface-mounted technique: (1) Inserted PM can generate higher torque because it utilizes both the PM and reluctance torques (e.g., the design *f* in Figure 2.22) generated by the magnetic saliency. (2) Because magnets are placed inside slots that pass axially through the rotor core, magnet retention is thus enhanced and yields to simplification in the manufacturing process. This is especially true for high-speed motors. (3) With this technique, rectangular magnets, which are less expensive than arc magnets, can be used. (4) The centrifugal forces acting on the magnets will be transferred to the rotor core. (5) When a magnet is broken due to corrosion, heating, or shock load, its fragments will not directly fly off into the air gap to damage the motor. However, for the same power and same machine size, the surface-mounted magnet machine requires less magnet material than the corresponding embedded magnet machine.

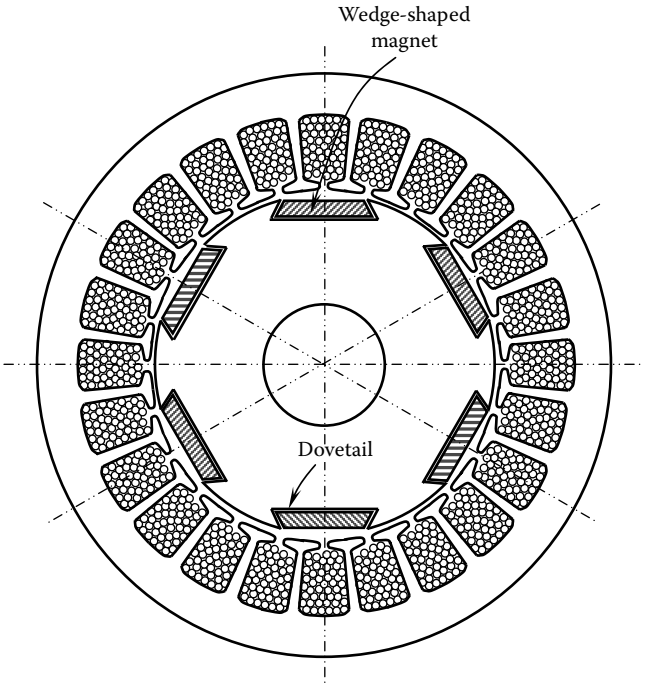


FIGURE 2.18
An innovative method for magnet retention by inserting wedge-shaped magnets into the dovetails on the rotor surface.

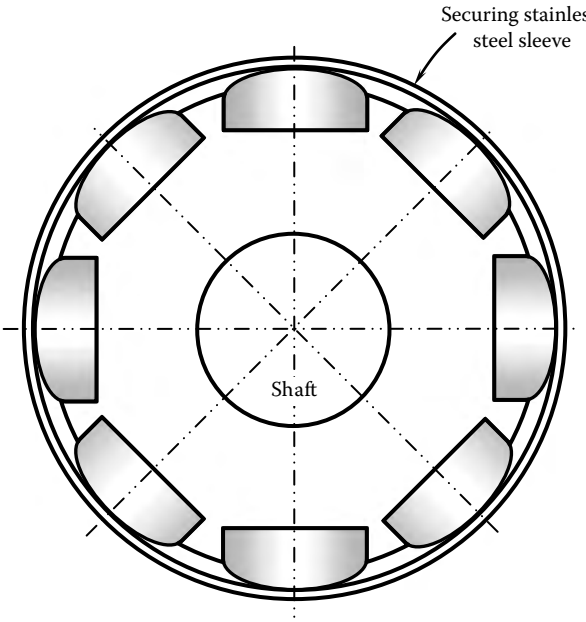


FIGURE 2.19
Using securing stainless steel sleeve for holding magnets on the rotor core.

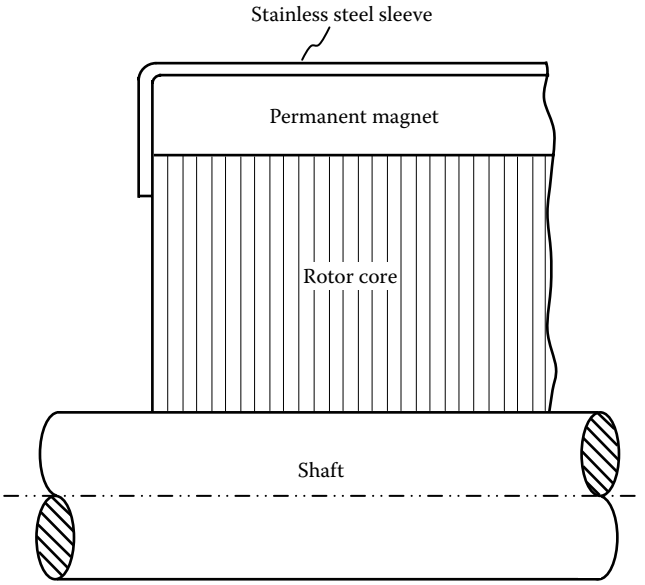


FIGURE 2.20
Bending of stainless steel sleeve at the end of rotor core for preventing magnets from shifting in the axial direction.

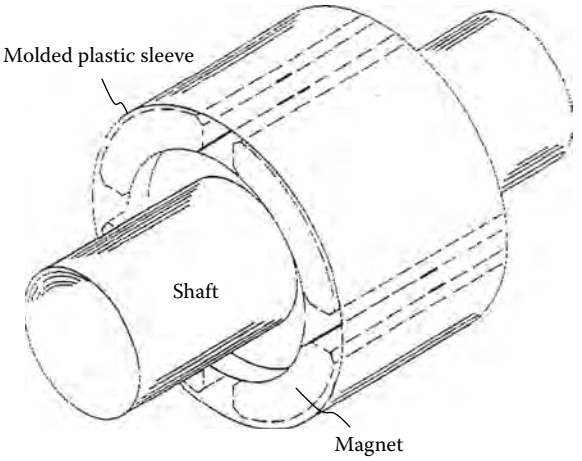


FIGURE 2.21
Retention of magnets with a molded plastic sleeve (U.S. Patent 4,973,872) [2.42]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

For a motor with embedded magnets, the stator synchronous inductance in the d-axis is lower than the synchronous inductance in the q-axis.

2.2.8 Corrosion Protection of Permanent Magnets

Electric motors may work under severe environmental conditions. Therefore, the protection of PMs from corrosion could be critical for motor normal operation and lifetime in

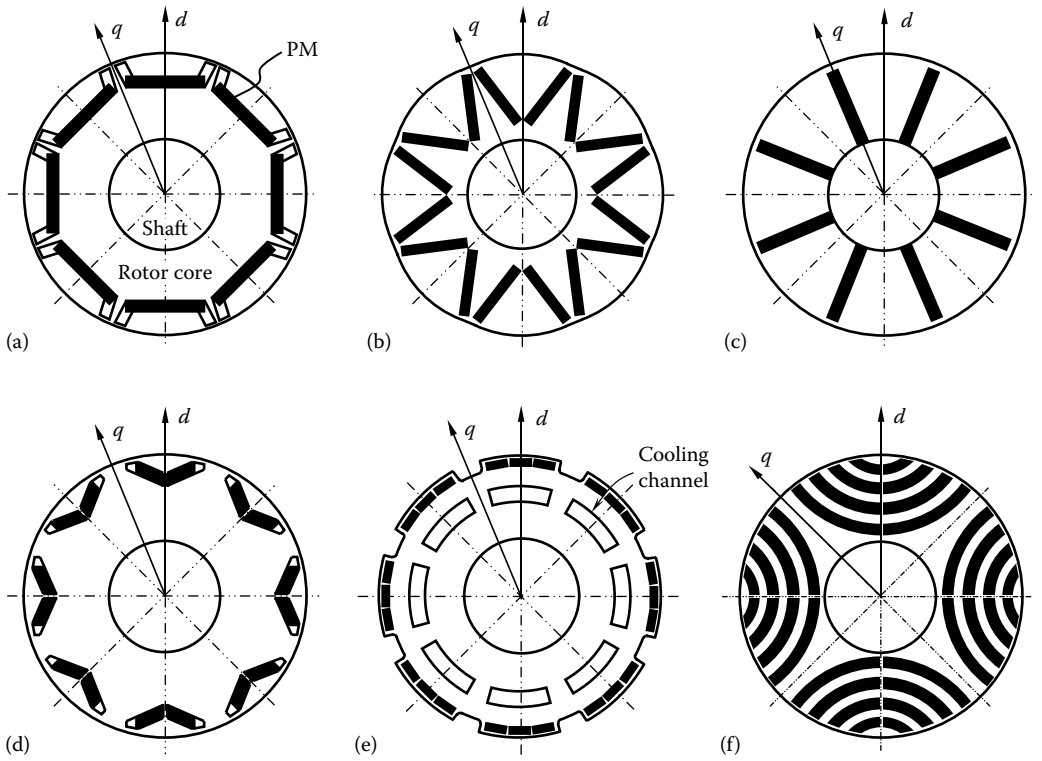


FIGURE 2.22
PMs placed in the interior of the rotor core, allowing for high-speed operation than surface-mounted magnet design: (a) buried tangential magnets near the rotor surface; (b) buried splitted V-magnets with 1/cosine-shaped air gap outline; (c) buried radial magnets so that their magnetization is azimuthal; (d) embedded splitted magnets in *V*-shaped slots near the rotor surface; (e) embedded splitted magnets in slots near the rotor outer surface (EP 1420501), similar to surface mount method but more secure magnet retention; (f) buried multilayer splitted arc magnets in saliency slots, exhibiting excellent performing characteristics of a motor.

some applications. For instance, rare-earth magnets exposed to acid/alkaline solution, salt, deleterious gas, or some chemical substances can result in corrosion of them. Even under a high-humidity condition, some magnets are easy to be eroded. It is noted that though the corrosion itself does not actually cause the part failure, the corrosion is the catalyst that starts a long chain of events over a long time of motor operation that eventually leads to the motor failure.

The detrimental effects of magnet corrosion include the following: (1) Excessive corrosion on magnet surfaces can cause premature wear of magnets. This in turn will cause magnet degradation and even failure of motor. (2) Debris and powders detached from corrosive magnets may block the air gap between the stator and rotor. (3) Microscale cracks may develop on magnet surfaces as a result of corrosion. This will eventually lower the strength of magnets.

The corrosion process of a magnet begins with the diffusion of oxygen, water vapor, or hydrogen along grain boundaries inside the magnet. The most common method to protect magnets against corrosion is to use coatings on magnet surfaces. Coatings can isolate magnets from all harmful chemical substances and thus enhance the corrosion resistance. A number of materials can be used for coating, such as polyurethane, phtal-uretane and

epoxy lacquer, polymer epoxide, polyester, polyester–epoxy, zinc, nickel, copper, and chromium. Experimental results [2.43] show that polymer coatings demonstrate the best resistance. The lacquer coatings are resistant in the water environment. Metal coatings have a lower corrosion resistance compared to the polymer and lacquer in corrosive environments.

Another method is to improve the intrinsic properties of the magnetic material by controlling the chemistry of the grain boundary in the magnet. For example, the addition of certain transition elements (e.g., Co, Ga, Nb, Mo, and V) can reduce the corrosion rate of rare-earth magnets [2.44].

2.3 Rotor Manufacturing Process

The manufacturing process for both induction and PM rotors starts from the material section. This is a critical step because it determines rotor core losses, rotor operating performance, and motor efficiency. Usually, motor manufacturers directly purchase thin lamination sheets from steel suppliers. The lamination sheets are punched by stamping machines or cut by laser devices to make desired lamination patterns. The stamped laminations are usually required to pass a heat treatment process to stabilize their electromagnetic, thermal, and mechanical properties and to release the internal stresses for dimension stabilization. Finally, these laminations are stacked together to form rotor cores.

The rotor assembly is achieved when a shaft is interference fitted into a rotor core. The design of the fitting method is very important for reliable motor operation and securely transmitting its torque to an external loaded machine. An improper fitting may lead to the separation of the shaft and the rotor core during motor operation. The final step in the rotor manufacturing is to carefully perform the rotor balance under a dynamic condition.

2.3.1 Lamination Materials

It has been commonly accepted that high motor torque can be achieved with the magnetic core material having high saturation magnetic flux density, low coercivity, and low-core-loss characteristics. The selection of rotor core material is based on the electromagnetic, thermal, and mechanical properties of materials, as well as the material cost. It has long been recognized that no material is perfect in all aspects and optimum for all applications. During design processes, trade-offs are often made between material properties (e.g., permeability, saturation flux density, density, and tensile strength), core losses, cost, fabrication processes, formation of lamination insulations, and other factors.

Of all the soft magnetic core materials, the most widely used materials are known as electrical steels, which are divided into several general classes. Among them, a major class is silicon steel, in which silicon is the principal alloying element. Alloying the steel with silicon can considerably increase the volume resistivity of the steel and thereby reduce the eddy-current loss. In addition, silicon can affect the grain structure of the steel and thus gives somewhat improved core loss by the reduction of the hysteresis loss in nonoriented electrical steels [2.45]. Hence, the electric and magnetic characteristics of silicon steel make it well suited for making rotor and stator lamination cores.

Silicon steel consists of body-centered cubic crystals. During a rolling process, these crystals are stretched and flattened. If they are left in that state, the magnetic properties

are maximized in the rolling direction (i.e., along the length of the strip). Hence, this type of steel is defined as oriented steel. Taking advantage of this characteristic, the maximum performance can be achieved by applying the material in the same direction that magnetic flux is expected. When an annealing heat treat (usually 850°C–1100°C) is performed, it may eliminate the grain structure in the material needed for achieving approximately isotropic properties within the material. This material is defined as nonoriented steel.

Silicon content in electrical steels is usually between 1.5% and 3.5%. Oriented silicon steel contains more silicon than nonoriented steel, approximately 3.0%–3.5%. However, it must be noted that very high silicon content (e.g., >4.0%) may result in lowering of induction permeability saturation density. To minimize hysteresis losses, the impurities such as oxides, nitrides, and sulfides in silicon steel must be tightly controlled.

Carbon content in all silicon steels is usually considerably low, approximately 0.003%. This helps minimize the hysteresis loss in the silicon steels, make them easier in rolling and other fabrication processes, and increase the lifetime of tooling.

Silicon steel is available in an array of grades and thicknesses, suited for applications in different types of electric motors. According to American Iron and Steel Institute (AISI), the silicon steel is graded by core loss, represented by a series of M numbers. The lower the M number, the lower the core loss and thus the higher the cost.

As a reference, some silicon steels are listed in Table 2.3. Among all grades, M19 is probably the most common grade for motion control products, as it offers nearly the lowest core loss in this class of material.

An alternative material commonly used for rotating electric equipment is the nickel alloy due to its high permeability and low core losses. These characteristics make it ideal for motors. However, its cost is significantly higher than silicon steel. Nickel alloys require a very careful annealing cycle. During the annealing process, surface insulation films are formed on the surfaces of the nickel alloy laminations.

In some applications that require high flux density without saturation, cobalt alloys may be used to make motor laminations. This type of alloys is also used in weight-sensitive applications such as space shuttles and satellites. The commonly used cobalt alloys contain 48%–50% cobalt and 2% vanadium, making them high tensile strengths. Hence, for some high-speed, large-power motors, silicon steel is no longer applicable due to its low mechanical strengths, and cobalt alloys become an excellent choice for the lamination material. Like nickel alloys, cobalt alloys require careful annealing after stamping and making oxide coatings on lamination surfaces is a separate process.

Other elements such as manganese and aluminum can also help reduce core losses due to the different mechanisms from silicon. The addition of these elements into steel will alter the metallurgical grain structure to contribute to lowering of the core loss.

Cold rolled lamination steel is the most cost-effective and most common material for core laminations of low-cost motors. Similar to carbon steel, this material has relative high core losses. It usually requires annealing after stamping to develop optimum properties and to add oxide coatings on the lamination surfaces.

The rotor core is assembled onto the rotor shaft with an interference fit. The radial interference value is optimized to provide adequate radial contact pressure to transmit torque under full-load and full-rotating speed.

One of the most critical issues for a high-speed motor is the extreme high centrifugal force acting on the rotor core, resulting in a high level of spin stress. In addition, interference fit preload stress also contributes to the loading of the rotor core. Thus, it becomes critical to carefully select lamination materials based on not only their magnetic properties but also mechanical properties, especially for high-speed motors.

TABLE 2.3

Silicon Steel Grades

Silicon Steel	AISI Grade	Thickness, mm (in.)	Note
Nonoriented silicon steel	M15	0.36 (0.014) 0.47 (0.0185)	The magnetic properties are practically the same in any direction of magnetization in the plane of the material.
	M19	0.36 (0.014) 0.47 (0.0185) 0.64 (0.025)	
	M22	0.36 (0.014) 0.47 (0.0185) 0.64 (0.025)	
	M27	0.36 (0.014) 0.47 (0.0185) 0.64 (0.025)	
	M36	0.36 (0.014) 0.47 (0.0185) 0.64 (0.025)	
	M43	0.36 (0.014) 0.47 (0.0185) 0.64 (0.025)	
	M45	0.47 (0.0185) 0.64 (0.025)	
	M47	0.47 (0.0185) 0.64 (0.025)	
Orientated silicon steel	M2	0.18 (0.007)	The magnetic properties are strongly oriented with respect to the direction of rolling.
	M3	0.23 (0.009)	
	M4	0.27 (0.011)	
	M6	0.36 (0.014)	
High permeability orientated	—	0.23 (0.009) 0.27 (0.011)	Low core losses with very thin laminations.

Source: Data from AK Steel, Selection of electrical steel for magnetic cores, http://www.aksteel.com/pdf/markets_products/electrical/Mag_Cores_Data_Bulletin.pdf.

2.3.2 Lamination Cutting

There are primarily two ways used by lamination manufacturers in lamination cutting: laser beam cutting and stamping machine cutting, depending on applications. For large-volume production manufacturing, stamping machine cutting is more appropriate for its high efficiency and low cost. For achieving high productivity, most lamination manufacturers adopted high-speed stamping machines in production lines. By supplying silicon steel sheets automatically, stamping machines operate continuously with press capabilities of over 250 strokes per minute and up to 1250 tons of force. To fully utilize the material, some motor manufacturers stamp slotted laminations of the stator and rotor from the same sheet of core steel simultaneously, with the outer doughnut-shaped punching as the stator lamination and the inner as the rotor lamination.

One problem often encountered in stamping cutting is that burrs are left at the lamination edges after a stamping process. Without passing through special treatments, these burrs may form electric paths for interlaminar eddy current under assembly pressures. Consequently, control of stamping operation must be carefully designed to keep the stamping burrs as small as possible. If necessary, all stamped edges are debarred when effective control is difficult or impossible.

Laser cutting is typically applied to large-size motor laminations in which stamping machines may not be suitable. This technique is also suitable for fast fabrication of motor prototypes for its flexibility, dimensional accuracy, and short lead time. By concentrating a very high temperature into a tiny spot during a short time, laser cutting provides superior cutting quality with small burrs, low deformation, low residual stress, and tight control of dimensions. During the cutting process, an oxide film is naturally generated on the burrs, thereby reducing the conductivity of burr contact and reducing interlaminar losses. The drawbacks of laser cutting are its low productivity and high cost.

2.3.3 Lamination Surface Insulation

When a laminated core is subjected to an alternating electric field, eddy current is induced in each lamination and between the laminations (Figure 2.23). The resistance to the eddy current within a lamination depends on the lamination dimensions (thickness, length, and width) and material properties. In most circumstances, the eddy-current losses vary approximately as the square of the thickness of flat-rolled magnetic materials, that is, $P_e \propto t^2$, where t is the lamination thickness. This indicates that reducing the lamination thickness by half will lead to the reduction in the eddy-current losses by one-fourth. With today's advanced nanotechnology, magnetic core laminations may be possibly made extremely thin so that eddy-current losses within the laminations become negligibly.

Referring to Figure 2.23, in addition to the eddy current within each lamination, eddy current may occur between laminations, called interlaminar eddy current. In order to minimize eddy-current losses between laminations, silicon steel laminations must be insulated from one another to maximize the lamination surface resistance by coating a thin layer on two sides of each lamination with either an organic, inorganic, or magnetite material.

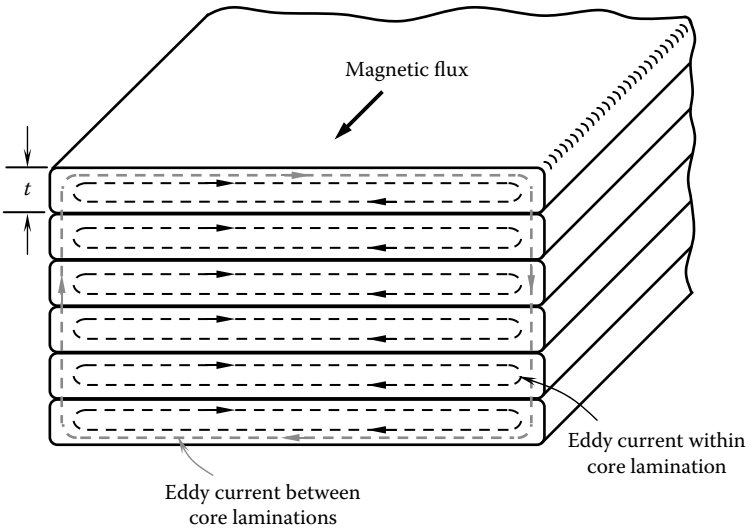


FIGURE 2.23
Eddy current within each core lamination and between core laminations.

The simplest way for making lamination insulation is to oxide laminations through annealing. This will create an oxide film on all lamination surfaces, as well as burrs. Thus, it greatly reduces the conductivity of laminations and minimizes eddy-current losses.

Lamination recoating restores interlaminate resistance to bring magnet core laminations back to optimal operating specifications. Recoating on stamped or laser cut laminations eliminates the possibility of interlaminate shorts.

2.3.4 Lamination Annealing

Lamination annealing is used to restore the electromagnetic, thermal, and mechanical properties of newly stamped or laser cut laminations. During a stamping process, plastic deformation, stress, and strain are produced at the cut zone, causing nonfavorable effects on magnetic properties of laminations. Though laser cutting produces little plastic deformation at cut edges, it induces a thermal shock wave and high temperatures to the cut laminations, resulting in thermal stresses and/or permanent damage to magnetic properties, such as reduced remanence and permeability. It has been reported that annealing processes can increase permeabilities and reduce power losses for both stamping and laser cutting [2.46]. Usually, the annealing temperature for the recrystallization ranges at 830°C–890°C. A higher annealing temperature up to 1100°C is also applicable for some lamination materials.

2.3.5 Lamination Stacking

A stack of laminations is assembled for a rotor core. Laminations are often stacked on a mandrel and then compressed under high pressure for obtaining the rigidity of the rotor core in the axial direction. Traditionally, laminations have been bonded using either adhesives or pins. Obviously, these stacking methods require additional operations and thus increase manufacturing costs and production lead time. In last several decades, a number of manufacturing processes have been developed to simplify the lamination stacking operation without using adhesives or pins. A simple method is to weld the rotor core when it is compressed. Another process involves interlocking the laminations at their outer tips with a die-punching machine so that corresponding laminations can interlock one another during assembly. Interlocking dimples on internal regions is another method. For large motors, the finished rotor cores are kept tightly in the axial direction by through bolts with a plate at each end of the stacked rotor core. A coated thin layer of insulation on each surface of each lamination reduces eddy current to minimize energy loss and boost performance.

An alternative method for fabricating rotor core is referred to as the slinky method. As demonstrated in Figure 2.24 [2.47], a rotor core is made by coiling up a straight continuous strip of sheet metal with respect to its centerline. To help bend the strip into a generally spiral and arcuate configuration, a V-shaped notch is formed between adjacent teeth at the ID of the core. The predetermined holes are used to receive suitable fastening means such as rivets to hold the rotor core firmly. All the slots, notches, and holes are punched together into a continuous longitudinally extending metal band of relative ductile material such as silicon steel.

2.3.6 Rotor Casting for Squirrel Cage Motor

A squirrel cage rotor has conducting bars and end rings made by casting either aluminum or copper into the rotor core slots and at the end surfaces of the rotor. The conducting bars

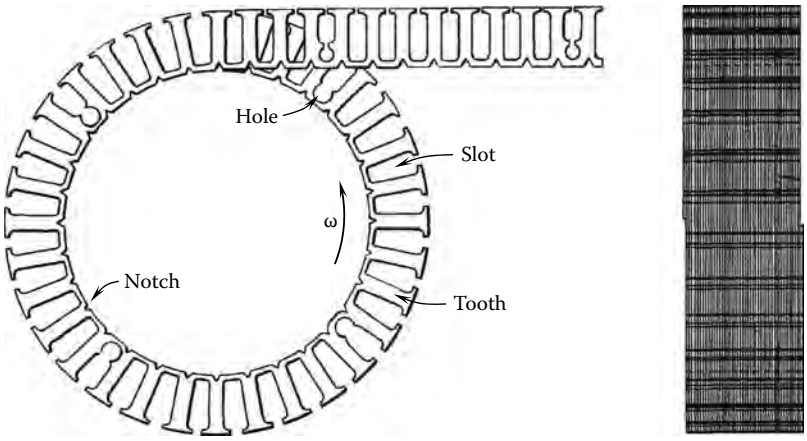


FIGURE 2.24
Applying slinky method for fabricating motor core for using as either rotor or stator (U.S. Patent 3,188,505) [2.47]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

are parallel to the rotor axis with a small skew angle, and the end rings serve to connect electrically all of the rotor bars, forming closed loop paths for electric current.

Cast rotors can be made by either permanent casting or die-casting process. In a permanent casting process, a rotor core is placed in a permanent mold with a desired space at the core top and bottom for the casting of end rings. Then, the mold is clamped together and a molten aluminum alloy is poured or forced into the mold. In a die-casting process, a rotor core can be cast either vertically or horizontally. One of the advantages using a die-casting process is its high pressure acting on the molten aluminum to force it to fill the rotor slots in a very short period of time for minimizing porosities in cast rotors. To achieve high casting quality, the key casting parameters including the aluminum temperature, injection speed, casting pressure, and casting time must be controlled strictly. In addition, to achieve high casting efficiency and rotor strength, all conducting bars, end rings, and fans are cast as one integral unit, resulting in a robust and rigid assembly to withstand high loads.

In a rotor casting process, a shaft may or may not be inserted into a rotor core, depending on the succeeding finishing steps required and the particular manufacturing process being used. With the shaft insertion, high casting temperatures may result in the deformation of the shaft and the reduction of the hardness on the shaft surface.

2.3.7 Heat Treatment of Casted Rotor

Heat treatment of casted rotors is important for optimizing the motor performance. By heating casted rotors at 300°C–450°C (570°F–840°F) for about 1–2 h, it helps break metallurgical bonds between steel laminations and the conductor bars, as the result of a large differential in thermal expansion of the two metallic materials. This greatly increases the electric resistance between the conducting bars and laminations. A theoretical analysis has shown that even a partial improvement in the insulation between the conducting bars and laminations can immediately result in noticeably reduced eddy-current losses. In addition, heating of casted rotor creates a thin protective insulation of naturally formed oxide film at the cutting edges of the laminations to further reduce interlaminar eddy-current losses.

2.3.8 Rotor Assembly

The most economic and reliable rotor assembly method is to use interference fits to firmly integrate shafts and rotor cores together, such as press fit and shrink fit. Generally, press fit is used for relative small rotors without exceeding the capacity of axial hydraulic pressing forces. However, the axial force required to assemble large rotors is considerably high in some instances and thus may bend or damage shafts or cause overstressed rotor cores. As a result, heat shrink processes are often used for larger rotors. The rotor assembling process is shown in Figure 2.25. In this case, shrink fit is made by heating rotor cores with induction heating (Figure 2.25a) while maintaining shafts at room temperature. When a rotor core reaches a preset temperature, its ID expands to become a little larger than the OD of the shaft, allowing the shaft to be easily inserted into the bore of the rotor core with little or no axial force (Figure 2.25b). The preset heating temperature is determined based on the rotor-shaft interference and ID of the rotor core. The rotor core is then shrunk as cold compressed air is applied on the rotor assembly. Once the shaft seizes to the core, shrinkage causes the core to pull away from the shaft shoulder. The end of the rotor core is hit with a rubber hammer to ensure the shaft shoulder is banked on the core face (Figure 2.25c). Finally, the rotor assembly is covered with a cooling hood for complete cooling of the rotor assembly.

2.3.9 Rotor Machining and Runout Measurement

A rotor assembly process involving heat shrinking or high mechanical pressuring may cause a change in shaft runout. Therefore, after rotor assembly is complete, it is preferred to check the runout of shafts to ensure the changes of runout are in the range of acceptance (Figure 2.26).

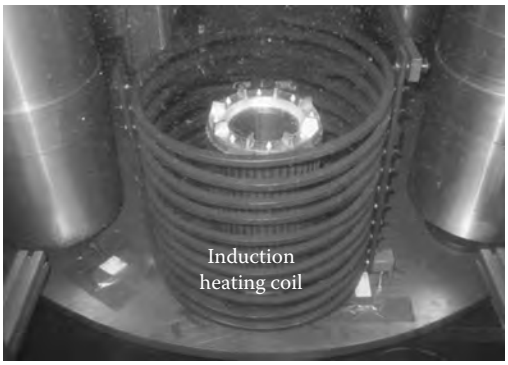
Kollmorgen engineers have investigated the effect of heat shrinking on shaft runout [2.48]. Thirty shafts were measured before and after the heat-shrinking assembly process. The runout of the shafts was inspected in seven locations with a 0.00254 mm (0.0001 in.) dial indicator using the bearing journals as the datum. The mean change in runout for shafts was 0.00414 mm (0.000163 in.) with a standard deviation of 0.00676 mm (0.000266 in.). The average runout from the rotor OD (measured at two locations) to the bearing journals was 0.01707 mm (0.000672 in.) with a standard deviation of 0.01034 mm (0.000407 in.). Data collected from the runout testing have shown that the heat shrink process for rotor assembly has minimal to no effect on shaft runout.

Then, the rotor is placed on a lathe, where the rotor outer surface is machined to a smooth finish for ensuring a uniform air gap between the rotor and stator and for minimizing the runout of the rotor outer surface to the axis of rotation (Figure 2.27). However, machining the rotor outer surface may smear the casted conducting bars and steel laminations, resulting in the increase in the core losses, creating some hot spots and the decrease in motor efficiency. To reduce the smearing effect on motor performance, some motor manufacturers grind the rotor outside surfaces.

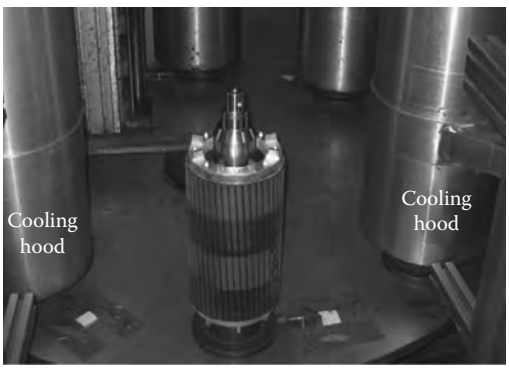
After the rotor is machined, it requires inspecting the runout on both the rotor and shaft for ensuring the centricity of the rotor to the rotating axis and uniform air gap, as demonstrated in Figure 2.28. The measured shaft runout data should be compared with those measured prior to the rotor assembly.

2.3.10 Rotor Balancing

Motor manufacturing involves a variety of mechanical processing techniques, such as machining, milling, forging, stamping, welding, grinding, drilling, and casting. When



(a)



(b)



(c)

FIGURE 2.25
Rotor assembly process with the shrink fit technique: (a) heating the rotor core, (b) inserting the shaft, and (c) quenching the assembly and hammering the shaft to reaching its desired position.

a rotor has been fabricated and assembled, unbalance of the rotor is likely experienced due to a number of factors, including improper manufacturing tolerances, material defects (e.g., porosity, voids, and blowholes), unsymmetrical structures (e.g., keys and key ways), poor assembly, and improper fabrications (e.g., misshapen casting and eccentric machining). Rotor unbalance can result in vibration and variable stress in rotors and their related

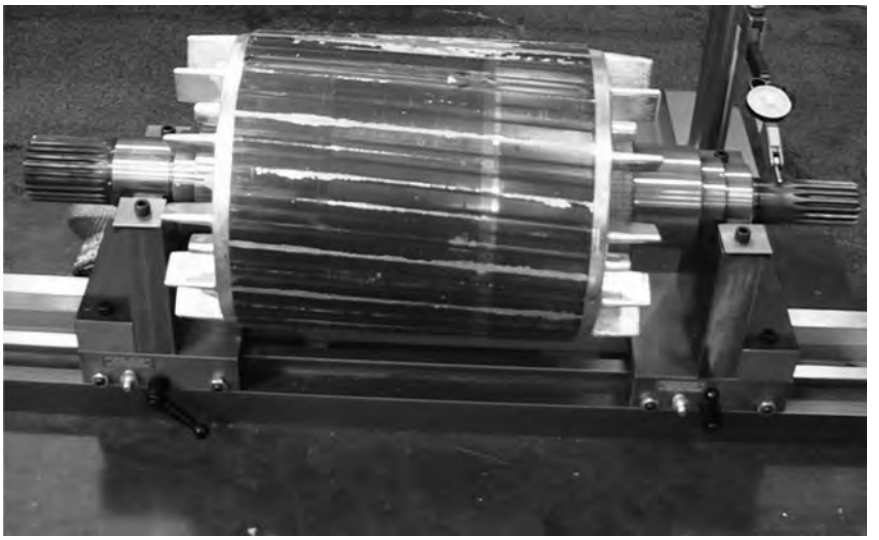


FIGURE 2.26
Measurement of shaft runout after rotor assembly but before rotor machining.

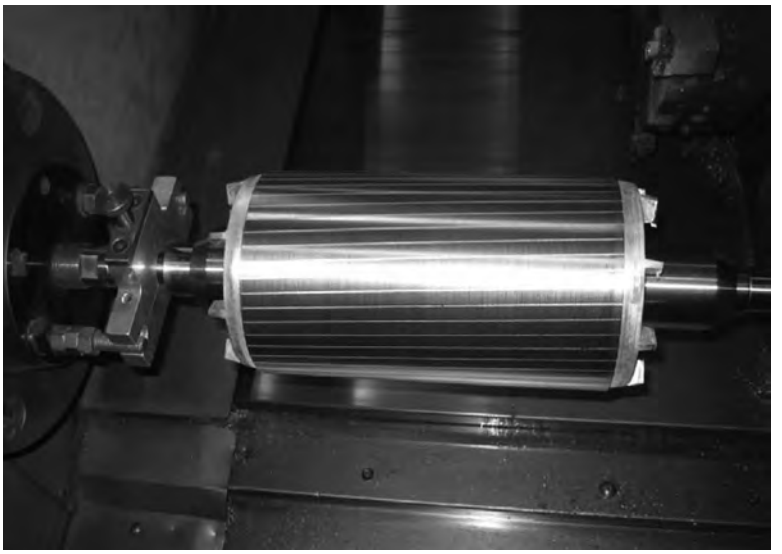


FIGURE 2.27
Machining the outer surface of a casted rotor with a CNC.

supporting structures. Rotor balancing is extremely important to motors that have very high rotating speeds for ensuring safe and normal operation.

Rotor balancing involves the entire rotor structure, including a shaft, a rotor core, rotor conducting bars/end rings, fans, and other auxiliary components.

There are a number of international standards available for rotor balance. One commonly used standard in motor industry is ISO Standard 1940/1 [2.49], which provides a method for applying unbalance tolerances based upon the static and coupled

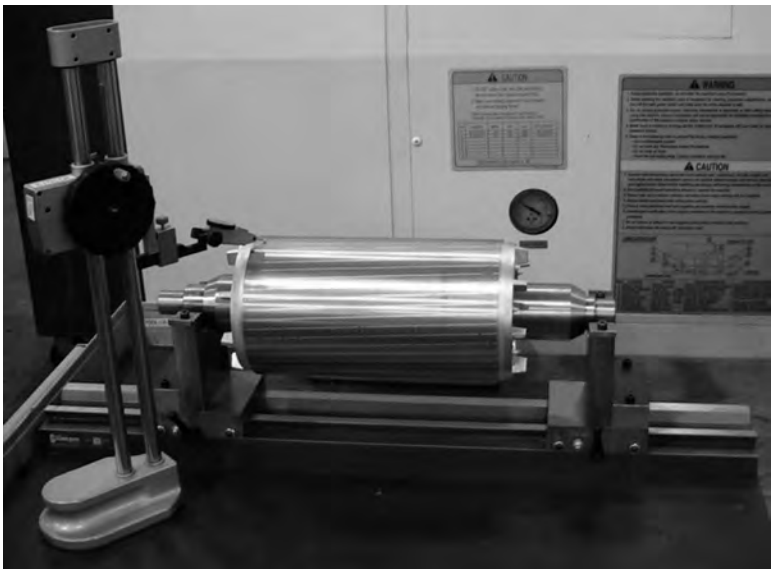


FIGURE 2.28
Runout measurement made on the outer surface of a cast rotor after it was machined.

components. This standard defines common levels of acceptable unbalance for various types of machines and applications.

2.3.10.1 Types of Unbalance

There are three types of unbalance: (1) static unbalance is when the principal inertia axis of a rotor is offset from and parallel to the axis of rotation and is corrected only in one axial plane (Figure 2.29a); (2) coupled unbalance is when the principal inertia axis intersects the axis of rotation at the center of gravity (Figure 2.29b); and (3) dynamic unbalance is the vectorial summation of static unbalance and coupled unbalance (Figure 2.29c). Thus, it is equivalent to two unbalance vectors in two specified planes that completely represent the total unbalance of the rotor. Because dynamic unbalance is a multiplane unbalance, the correction of dynamic unbalancing requires at least two correction masses. In all cases, the center of gravity C_g is located on the principal inertia axis, rather than on the axis of rotation.

It has been widely accepted that dynamic unbalance is one of the most common sources of motor vibration and noise. As reported previously, dynamic unbalance is the main source in about 40% of the excessive vibration situations [2.50]. In severe situations, this type of unbalance can cause failures of rotor and bearings.

As the simplest form of unbalance, static rotor unbalance U_s can be determined as

$$U_s = M_r e = m_u r_u \tag{2.3}$$

where

M_r and m_u are the rotor and unbalance masses, respectively
e is the mass eccentricity measured as the distance between the principal inertia axis and the axis of rotation
r_u is the distance between the center of the unbalance mass and the axis of rotation

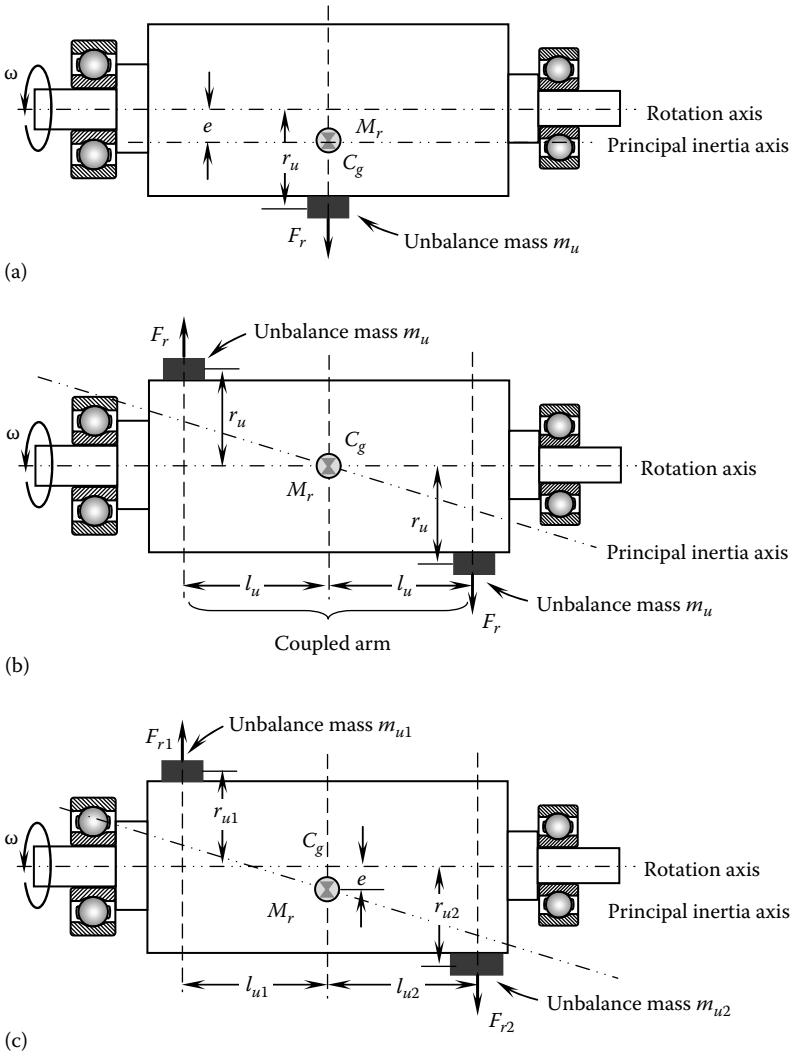


FIGURE 2.29
Three types of unbalance: (a) static unbalance, (b) coupled unbalance, and (c) dynamic unbalance. In all cases, the center of gravity is located on the principal inertial axis.

Thus, the mass eccentricity e can be obtained:

$$e = \frac{m_u r_u}{M_r} \tag{2.4}$$

Referring to Figure 2.29a, the magnitude of unbalance force or centrifugal force is expressed as

$$F = (m_u r_u + M_r e)\omega^2 \tag{2.5}$$

This indicates that centrifugal force is exponentially sensitive to the rotor rotating speed.

Coupled unbalance is expressed as

$$U_c = m_u r_u (2l_u) \tag{2.6}$$

where $(2l_u)$ is defined as the coupled arm, which is the distance between the two unbalance masses. Thus, coupled unbalance is described as a mass times a length squared.

2.3.10.2 Rotor Balancing Machine

To prevent possible damages to motor due to vibration caused by rotor unbalance, balancing operation is a necessary corrective action before assembling rotors to motors. Many motor manufacturers use balancing machines to detect the amount and location of unbalanced masses on rotors. Some precise balancing machines can sensitively and accurately identify any mass axis 0.001 mm off the axis of rotation.

A simplest type of balancing machine is used for static balancing only. This type of machine is suitable for balancing disk-shaped rotors. For a majority of rotors, the type of balancing machine required must be capable of identifying dynamic unbalance in two axial planes. This type of machine is suitable for balancing rotors with long span length and small shaft diameter (i.e., $L/D \gg 1$).

The principle of rotor balancing is very similar to that of car wheel balancing. With an advanced balancing machine, masses have to be either removed from or added to a rotor to minimize the uneven mass distribution so that rotor vibration can be minimized. It must be noted that the added mass must be firmly installed to the rotor and thus will not fall off when the rotor rotates at a high speed. For this reason, the material removal method is ideal.

An advanced balancing machine is shown in Figure 2.30. This machine can implement the whole balancing process with full automation, from measuring the location and



FIGURE 2.30
Rotor balancing machine.

amount of unbalance, displaying unbalancing results in polar graphs, drilling on the testing rotor, and repeating the process until unbalance is lower than an allowable level.

2.3.10.3 Balancing Operation

Not all unbalanced rotors need to be corrected. In common practices, a permissible unbalance is set by motor manufacturers as the balancing acceptance limit. The permissible unbalance can be determined based on experiments from similar machines, permissible bearing forces, or standards. Balancing corrections would be taken only when the existing balance is larger than the permissible unbalance.

The principle of rotor balancing is that all forces and moments acting on the rotor must be balanced under both static and dynamic conditions, so that the resultant force and moment on the rotor must be zero, that is,

$$\sum_{i=1}^{n} F_i = 0 \tag{2.7a}$$

and

$$\sum_{i=1}^{n} M_i = 0 \tag{2.7b}$$

Static unbalance can be detected without spinning the rotor and can be most easily corrected. As shown in Figure 2.31, a static unbalance can be corrected in two ways. One method is to add directly a correct mass m_c at the opposite location of the unbalance mass m_u in the plane that contains the center of gravity (Figure 2.31a). As long as the product of the correct mass m_c and the radius from the center of the correct mass to the axis of rotation r_c is equal to the product of the unbalance mass m_u and the radius from the center of the unbalance mass to the axis of rotation r_u ($m_c r_c = m_u r_u$), the static balance is achieved. In a case that $r_c = r_u$, it leads to $m_c = m_u$.

An alternative method is to use two pieces of correct mass on the rotor in planes equidistant from the plane that contains the center of gravity at the radius r_c (Figure 2.31b). The static balance is achieved when $2m_c r_r = m_u r_u$. If $r_c = r_u$, then $m_c = m_u/2$.

An advantage of using these methods for correcting static unbalance is their simplicity and flexibility. Rotor balancing can be done without using a complex balance machine. However, these methods are only suitable for the disk-shaped rotors that are dominated by static unbalance.

Unlike static unbalance that is measured under a nonrotating condition, coupled unbalance can only be measured by spinning the motor rotor. There are two methods to correct coupled unbalance. The first method is to use two equal masses, in which each is to be added on the rotor at an angle of 180° apart from the unbalance mass, as shown in Figure 2.32a. In the second method, coupled unbalance is corrected in any two planes. However, each correcting mass and the distance to the plane that contains the center of gravity must be carefully determined, as shown in Figure 2.32b.

Because dynamic unbalance is the vectorial summation of static unbalance and coupled unbalance, the correction of dynamic unbalance is thus the combination of the corrections of static unbalance and coupled unbalance.

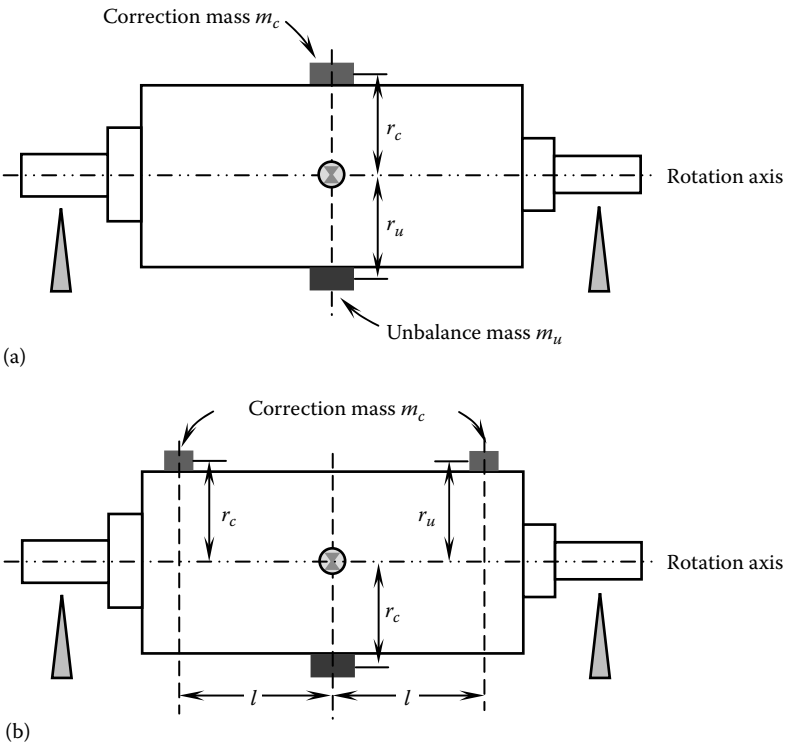


FIGURE 2.31
Two alterative balancing operations for static unbalance: (a) adding one piece of correction mass in the plane containing the center of gravity and (b) adding two pieces of correction mass in two planes with equal distance from the plane containing the center of gravity and equal radius from the rotation axis.

A real balancing operation is shown in Figure 2.33. An assembled rotor is placed in a rotor balancing machine. The bearing journal at each end of shaft is put on a pair of bearing rollers. The rotor is driven by a driving belt that is placed over the rotor surface near the rotor center.

2.4 Interference Fit

A rotor primarily consists of a rotor core and a shaft. In order to integrate these two parts together as the rotor to transmit the motor torque, the fit between the rotor core and shaft must be carefully designed. For example, too loose a fit could result in a corroded or scored rotor core and shaft, while too tight a fit could result in unnecessarily large mounting and dismounting forces, even damages of the mating surfaces or deformation/cracks of the fitting parts.

The term *interference* refers to the fact that one part slightly interferes with the space that the other is taking up. Interference fit is often used to joint two mating parts together either semipermanently or permanently. Interference fit can be generally achieved by shaping

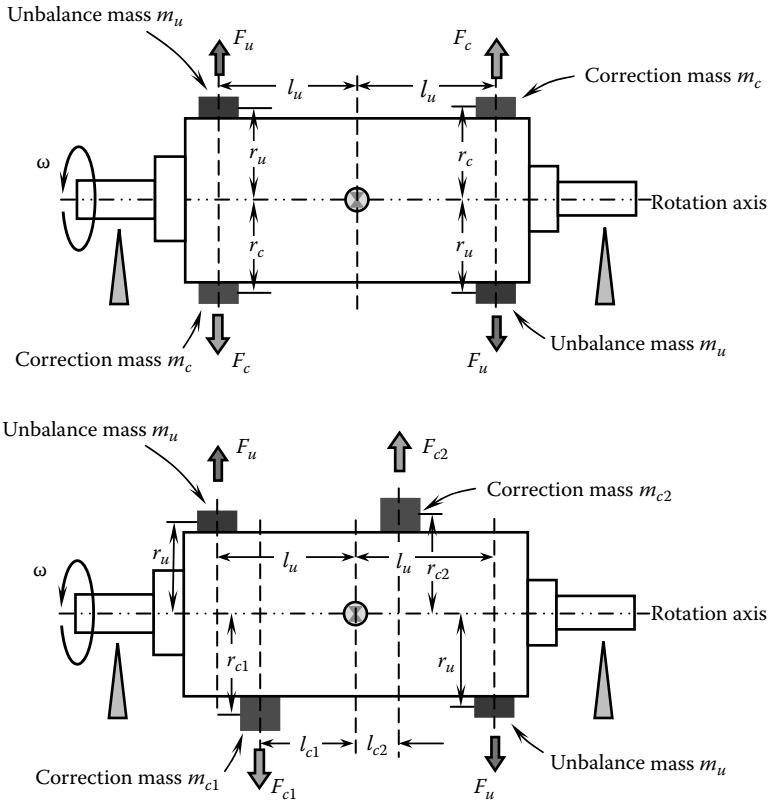


FIGURE 2.32
Balancing operation for coupled unbalance.

the two mating parts so that one or the other (or both) slightly deviates in size from the nominal dimension. In the motor industry, interference fit is extensively used to join rotor cores and shafts, as well as stator cores and housings, for its high joining strength and concentricity between the mating parts. Interference fit can be divided into several types as discussed in the following.

CTE is the material property of primary influence on dimensional stability as it represents the material response to changes in temperature. This material property can be exploited in an interference fit such as shrink fit. It is to be noted that in the interference fit design, it is critical to carefully consider the temperature effect on the fitting strength and component stresses in the whole temperature range including both motor operation and storage. In some improper designs, interference fits work fine near room temperature. However, as the temperature becomes extremely high or extremely low, due to the large differential in thermal expansion between the assembly parts, the two assembled parts may separate from each other or pressed against each other, causing one of them to be broken.

Consequently, to avoid failure of the interference-fitted assembly and minimize thermal stresses arising from the difference in CTE during temperature variations, it is highly desired to select materials of the mating parts that have close CTE values.

Hesai Exhibit 1031, page 158

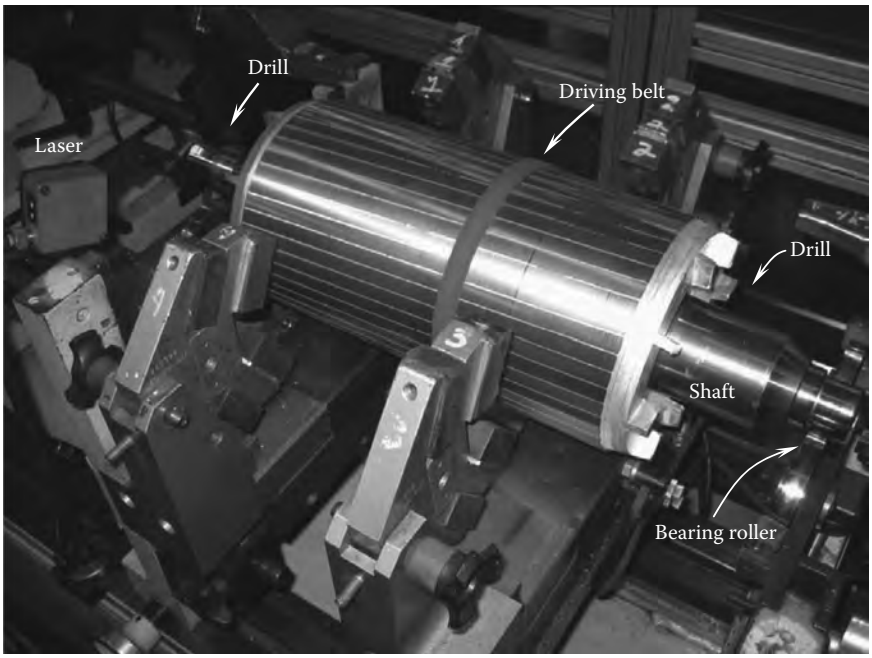


FIGURE 2.33
Automatically controlled rotor balancing process.

2.4.1 Press Fit

Press fit is generally chosen for its high operation reliability in rotor assemblies. Press fit can be obtained by selecting the proper interference between a shaft and a rotor core, that is, the OD of the shaft is slightly larger than the ID of the rotor core. Thus, at room temperature, the shaft must be pressed into the rotor core under an axial pressing force. During this insertion process, the shaft reduces its original OD due to the compressive force, and the rotor core increases its original ID due to the expansion force, producing a radial contact stress at the interface. This radial stress is referred to in the literature as contact pressure or interference pressure. In general, the contact pressure changes in response to the interference, ambient temperature, shaft dimensions, and material properties of both the shaft and rotor core.

As shown in Figure 2.34, the original inner and outer radii of a hollow shaft and the rotor core are a and b and c and d, respectively. For the case that $b > c$, the interference between the shaft and the rotor core is defined as

$$\delta = b - c \tag{2.8}$$

when the shaft is pressed into the rotor core, the shaft radius becomes b' and

$$b' = b - \delta_s \tag{2.9}$$

where δ_s is the decrease of the shaft radius. As the same matter, the internal radius of the rotor core becomes c'

$$c' = c + \delta_{rc} \tag{2.10}$$

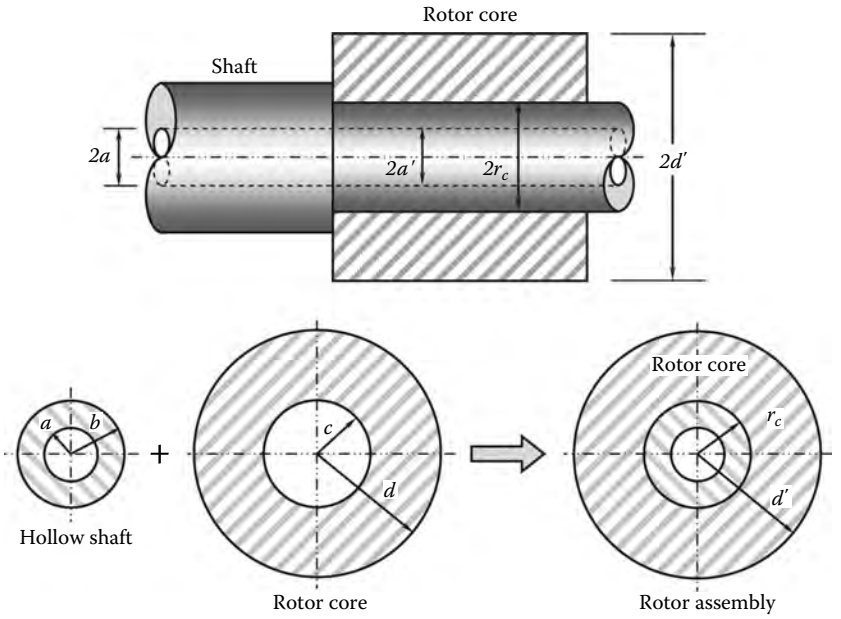


FIGURE 2.34
Dimensions of the hollow shaft and rotor core before and after assembly.

where δ_{rc} is the increase in the internal radius of the rotor core. Because that $b' = c' = r_c$, it follows that

$$\delta = b - c = \delta_s + \delta_{rc} \tag{2.11}$$

A contact pressure p is thus created at the contacting surface at $r = r_c$ (Figure 2.35). It can be derived that the radial stresses $\sigma_r = -p$ in each member at the contacting surfaces. The tangential stresses of the rotor core bore and the shaft surface are [2.51]

$$\sigma_{t,rc} = p\frac{d^2 + r_c^2}{d^2 - r_c^2} \tag{2.12}$$

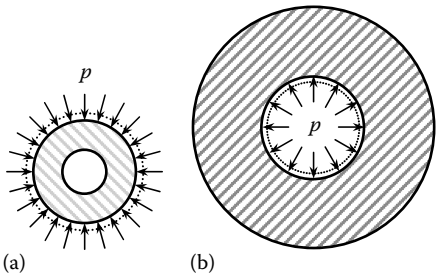


FIGURE 2.35
Contact pressure acting on (a) the hollow shaft and (b) the rotor core.

$$\sigma_{t,s} = -p\frac{r_c^2 + a^2}{r_c^2 - a^2} \tag{2.13}$$

respectively.

The tangential strain of the rotor core is determined as the ratio of the change in circumference to the original circumference, that is,

$$\varepsilon_{rc} = \frac{2\pi(r_c - c)}{2\pi c} = \frac{\delta_{rc}}{c} \tag{2.14}$$

This gives that

$$\delta_{rc} = c\varepsilon_{rc} \tag{2.15}$$

Since

$$\varepsilon_{rc} = \frac{\sigma_{t,rc}}{E_{rc}} - \nu_{rc}\frac{\sigma_r}{E_{rc}} \tag{2.16}$$

where

ν is Poisson's ratio

E is Young's modulus (modulus of elasticity)

Combining Equations 2.14 and 2.16, it yields

$$\delta_{rc} = \frac{c}{E_{rc}}\left(\sigma_{t,rc} - \nu_{rc}\sigma_r\right) \tag{2.17}$$

Noting that $\sigma_r = -p$ and substituting Equation 2.12 into 2.17 yields

$$\delta_{rc} = \frac{cp}{E_{rc}}\left(\frac{d^2 + r_c^2}{d^2 - r_c^2} + \nu_{rc}\right) \tag{2.18}$$

Similarly to Equations 2.14 and 2.16 for the tangential strain of the rotor core, the governing equations for the tangential strain of the shaft are as follows:

$$\varepsilon_s = \frac{2\pi(r_c - b)}{2\pi b} = -\frac{\delta_s}{b} \tag{2.19}$$

$$\varepsilon_s = \frac{\sigma_{t,s}}{E_s} - \nu_s\frac{\sigma_r}{E_s} \tag{2.20}$$

Therefore, from the previous two equations, the reduction in the shaft radius δ_s can be expressed as

$$\delta_s = \frac{bp}{E_s}\left(\frac{r_c^2 + a^2}{r_c^2 - a^2} - \nu_s\right) \tag{2.21}$$

Thus, Equation 2.11 can be rewritten as

$$\delta = \delta_s + \delta_{rc} = \frac{cp}{E_{rc}}\left(\frac{d^2 + r_c^2}{d^2 - r_c^2} + \nu_{rc}\right) + \frac{bp}{E_s}\left(\frac{r_c^2 + a^2}{r_c^2 - a^2} - \nu_s\right) \tag{2.22}$$

It must note that the interference δ is associated with radius, rather than diameter. This equation can be solved for the contact pressure p when the interference δ is known:

$$p = \frac{\delta}{\frac{c}{E_{rc}}\left(\frac{d^2 + r_c^2}{d^2 - r_c^2} + \nu_{rc}\right) + \frac{b}{E_s}\left(\frac{r_c^2 + a^2}{r_c^2 - a^2} - \nu_s\right)} \tag{2.23}$$

For a solid shaft (i.e., $a = 0$), the previous equation is simplified as

$$p = \frac{\delta}{\frac{c}{E_{rc}}\left(\frac{d^2 + r_c^2}{d^2 - r_c^2} + \nu_{rc}\right) + \frac{b}{E_s}\left(1 - \nu_s\right)} \tag{2.24}$$

The contact area between the shaft and the rotor core is

$$A_c = 2\pi r_c L \tag{2.25}$$

where L is the contact length between the rotor core and the shaft. Thus, for a given interference δ, the maximum torque that can be carried by the shaft is

$$T_{\max} = \mu_s F_c r_c = \mu_s p A_c r_c = 2\pi r_c^2 \mu_s p L \tag{2.26}$$

where μ_s is the static coefficient of friction between the shaft and the rotor core.

The axial force required for the press in assembly is (see Figure 2.36)

$$F_a = \mu_k p A_c = 2\pi r_c \mu_k p L \tag{2.27}$$

where μ_k is the kinetic coefficient of friction between the shaft and the rotor core.

It is noted that $b = r_c + \delta_s$ and $c = r_c - \delta_{rc}$. Because $(b + c) \gg (\delta_{rc} - \delta_s)$, r_c can be expressed as

$$r_c = \frac{b + c}{2} + \frac{\delta_{rc} - \delta_s}{2} \approx \frac{b + c}{2} \tag{2.28}$$

When a shaft is pressed into a rotor core, the force driving the shaft should be applied uniformly to the end of the shaft to avoid galling, peening, or damaging the rotor core. The mating surfaces of both the shaft and rotor core should be thoroughly cleaned and free of imperfections.

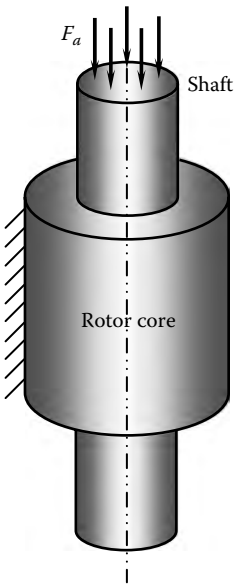


FIGURE 2.36
Axial pressing force applied in press fit rotor assembly.

2.4.2 Shrink Fit

Shrink fit is a semipermanent assembly system that can transmit large torques through the creation of high contact pressure at the interface of its mating components. Being a tight joining method, shrink fit has been extensively used in various industries due to its high joining strength and reliability. There are a number of benefits to using shrink fit over press fit: (1) shrink fit can minimize mechanical stresses associated with the pressing operation and thus reduce the deformation of the rotor core; (2) it allows a rotor core to shrink onto a shaft symmetrically; (3) the requirement of surface finishing for shrink fit is relatively lower than that for press fit; and (4) the process of shrink fit is completely controllable.

Most materials are subjected to thermal expansion as the temperature goes up and thermal contraction as the temperature goes down. Shrink fit techniques utilize such the material properties in machine assembly processes.

For a solid material, the coefficient of linear thermal expansion is typically a function of temperature, for measuring how much the material expands for a change in temperature. It is defined as the linear dimension change with respect to the change in temperature per unit linear dimension (i.e., length) under a constant pressure p_o:

$$\alpha_L = \frac{1}{L}\left(\frac{\partial L}{\partial T}\right)_{p_o} \tag{2.29}$$

If the desired expansion in length ΔL is provided, the temperature rise ΔT can be determined as

$$\Delta T = \frac{1}{\alpha_L}\frac{\Delta L}{L} \tag{2.30}$$

TABLE 2.4

Coefficient of Linear Thermal Expansion of Some Common Materials

Material	Coefficient of Linear Thermal Expansion × 10^{-6} (at 20°C) (mm/mm-°C)
Aluminum	23
Aluminum nitride	5.3
Brass	19
Carbon steel	10.8
Copper	17
Gold	14
Iron	11.8
Nickel	13
Stainless steel	17.3
Steel	11–13

The coefficients of linear thermal expansion for some common materials are listed in Table 2.4.

Similarly, the area and volume thermal expansion coefficients are defined as follows:

$$\alpha_A = \frac{1}{A}\left(\frac{\partial A}{\partial T}\right)_{p_o} \tag{2.31}$$

$$\alpha_V = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_{p_o} \tag{2.32}$$

In shrink fit applications, the ID of a rotor core is slightly smaller than the OD of a shaft. In an assemble process, to achieve a shrink fit joint between the two components, either the rotor core or the shaft (or both) must be treated. This can be done by either heating the core to increase the internal diameter or cooling the shaft to reduce the external diameter. Thus, according to the fitting conditions, shrink fit techniques can be categorized into three groups: (1) heating technique, (2) cooling technique, and (3) mixing technique, which is the combination of (1) and (2).

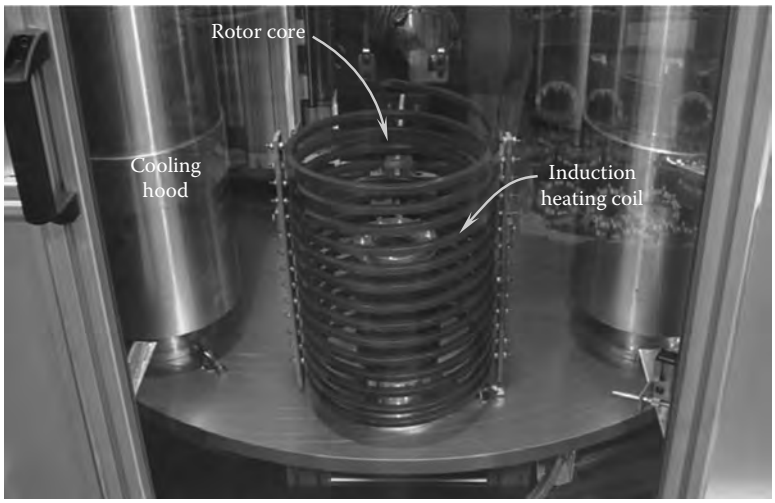
Most motor manufacturers utilize heating techniques in rotor shrink fits, that is, heating rotor cores to a certain temperature and maintaining shafts at room temperature. With the desired expansion and shrinkage, the rise in temperature can be determined. For example, if the shaft OD is 31.75 mm and the rotor core ID is 31.70 mm. This gives the diametrical interference fit of 0.05 mm. Thus, by adding the minimum desired slip fit clearance of 0.07 mm (≈ 0.003 in.), the total differential expansion is 0.12 mm. The temperature rise ΔT required on the rotor core to give 0.12 mm expansion in the diameter is calculated as

$$\Delta T = \frac{0.12/31.70}{10.8 \times 10^{-6}} = 350.5°\text{C}$$

Therefore, the total temperature would be 350.5°C plus the ambient temperature.



(a)



(b)

FIGURE 2.37
Inducting heating process for rotor core prior to shaft insertion: (a) loading and inducting heating coil on the rotor core and (b) heating the rotor core.

Induction heating is a noncontact process of providing fast, consistent, and controllable heating for bonding, hardening, or softening metals or other electrically conductive materials. The process relies on induced electric currents within metallic components for heating themselves. By using this technique, shrink fit can be achieved by heating rotor cores to preset temperatures while maintaining shafts at room temperature. As demonstrated in Figure 2.37, a rotor core is placed in an induction heating equipment. As the induction heating coil is placed around the rotor core, it requires only a few seconds to achieve the desired temperature of the rotor core. The power supply to the heating coil is controlled and monitored with the front panel LCD and sealed touchpad.

As a rotor core is heated up, its internal diameter increases, allowing a shaft to insert into position. Then, as the rotor core is cool down, it shrinks back to its original size and holds the shaft tightly. By contrast, in a cooling technique, shrink fit is achieved by chilling shafts with a suitable median, such as liquid nitrogen or dry ice. An alternative way is to place shafts into a freezer. During the time, the rotor core remains at the room temperature. In such a way, the shaft shrinks to allow it to pass through the rotor core bore. When the shaft restores to the room temperature, it expands to its original size to gain a tight fit with the rotor core. One of the advantages of the cooling technique is that it causes little distortions on effected parts.

In some applications, it is preferred to cool down shafts rather than to heat up rotor cores. This is because high temperature may cause the damage of the rotor winding insulation (as in IMs) or irreversible demagnetization of magnets (as in PMMs). In addition, since a shaft has a relatively low weight compared with that of a rotor core, it is easier to deal with the shaft than the rotor core. During the cooling shrink fit process, the shaft is usually cooled via exposure to a cryogen, typically solid carbon dioxide (–78.5°C at normal atmospheric pressure), liquid carbon dioxide (–56.6°C at 518 kPa), or liquid nitrogen (–195.8°C at normal atmospheric pressure) in order to reduce its size through the contraction.

In comparison with heating, the cooling process cannot achieve as much change in diameter. Thus, for some special fitting applications, both the heating and cooling technique are used to provide the desired fitting strength.

The mechanical design of shrink fit set is based on either the classical Lamé elastic solution of a thick-walled cylinder or the elastoplastic solution that is based on the yield criterion of von Mises. Various analytical, experimental, and numerical studies have been performed by many investigators. Horger and Nelson [2.52,2.53] discussed the detail design of shrink fit assemblies based on linear elasticity solutions. An analysis of the shrink fit in the context of nonlinear elasticity was presented by Antman and Shvartsman [2.54]. Using the von Mises yield criterion and the Hencky deformation theory, Lundberg [2.55] presented the first elastoplastic solution for the shrink fit problem. Late, based on the work of Lundberg, Gao and Atluri [2.56] proposed an analytical solution for the axisymmetric shrink fit problem with a thin strain-hardening hub and an elastic solid shaft.

The difference between shrink fit and press fit is in method of assembly: press fit applies an axial force during the assembling process at room temperature, and shrink fit takes advantage of thermal expansion or shrinkage of the materials. In some cases, shrink fit is the only way to join parts that have low mechanical strengths.

2.4.3 Serration Fit

Serration fit is referred to a special press fit technique under partial interference conditions. In serration fit, a number of serrations are made symmetrically on the outer circumference of a shaft at the room temperature to form local deformations on the shaft (Figure 2.38). Due to the hardening effect of cold forging, the local strength near the serrations can increase 15%–30% of the material strength. When the shaft is pressed into a rotor core, the interference occurs at the vicinity of each serrated groove on the shaft. During the fitting process, the serrations are deformed to generate a contact pressure at the small areas near deformed serrated grooves. Unlike in press fit and shrink fit, the requirements for surface finish and dimension tolerance of mating components can be significantly reduced because of the limited contact area in serration fit.

Hesai Exhibit 1031, page 166

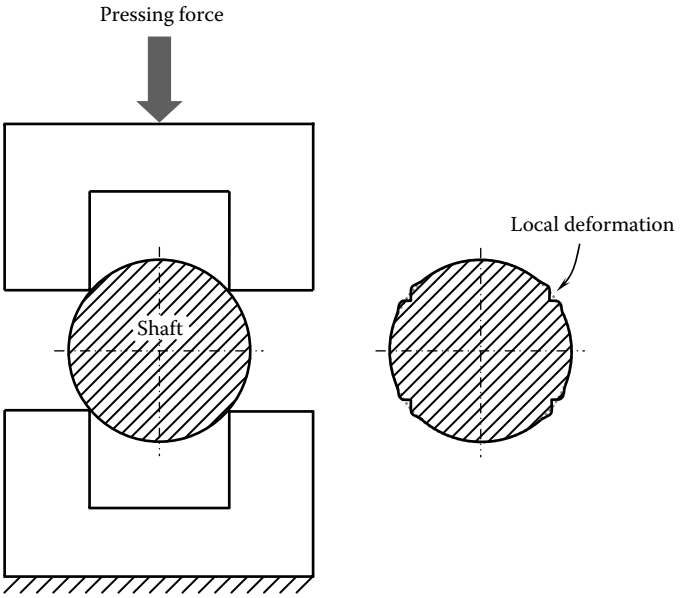


FIGURE 2.38
Fabrication of serrations on a shaft.

In the motor industry, the serration fit technique is adopted in rotor assemblies for relative low torque applications. Figure 2.39 demonstrates the V-shaped serrated grooves disposed in the axial direction on the outer surfaces of the shafts.

2.4.4 Fitting with Adjustable Ringfeder® Locking Devices

Ringfeder locking devices generate easily adjustable and releasable mechanical fits. As shown in Figure 2.40, these locking devices feature either a single- or double-tapped thrust rings with self-releasing tapers. Thus, they can bridge relatively large fit clearances between shafts and rotor cores. Because of their advantages over other interference fitting methods, they have been successfully used in various industries for more than a half century.

2.4.5 Fitting with Tolerance Rings

Tolerance ring is a precision-engineered device usually made from a thin strip of spring steel or stainless steel. The application of tolerance rings can provide robust fitting joints in rotor assemblies. Wave pitches pressed on tolerance rings are used to provide the radial contact pressure between mating components. When the tolerance ring is assembled between mating parts, the wave pitches are compressed and elastically deflected, resulting a large contact pressure between the mating parts for tightly holding them together. In motor manufacturing, a tolerance ring presented in Figure 2.41 can be used for assemblies of shafts and rotor cores and bearings. The wave pitch of the tolerance ring is carefully designed so that the desired spring rate can be achieved.

Tolerance rings work on the two physics principles of spring and friction. The corrugations on a tolerance ring act like stiff radial spring. Like regular springs, the



FIGURE 2.39
Serrations on motor shafts.

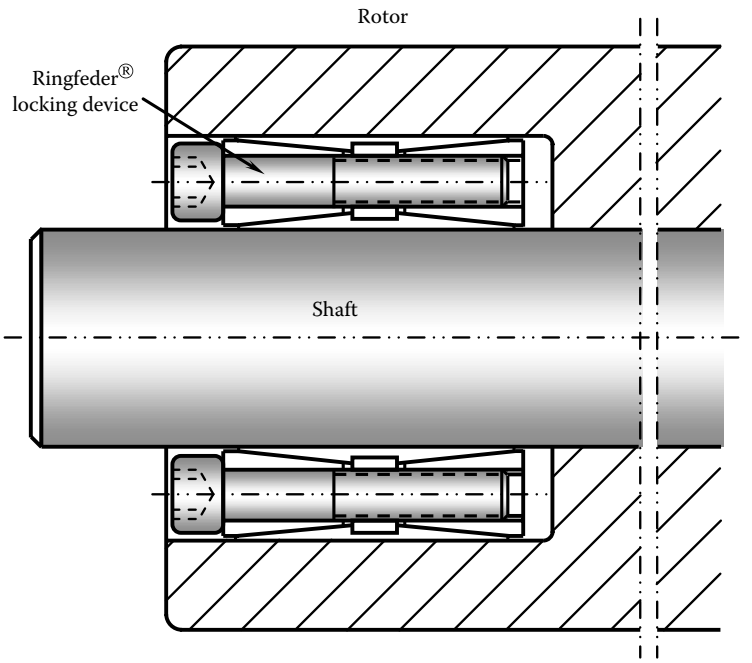


FIGURE 2.40
Keyless connection between shaft and rotor/hub using Ringfeder® locking device.

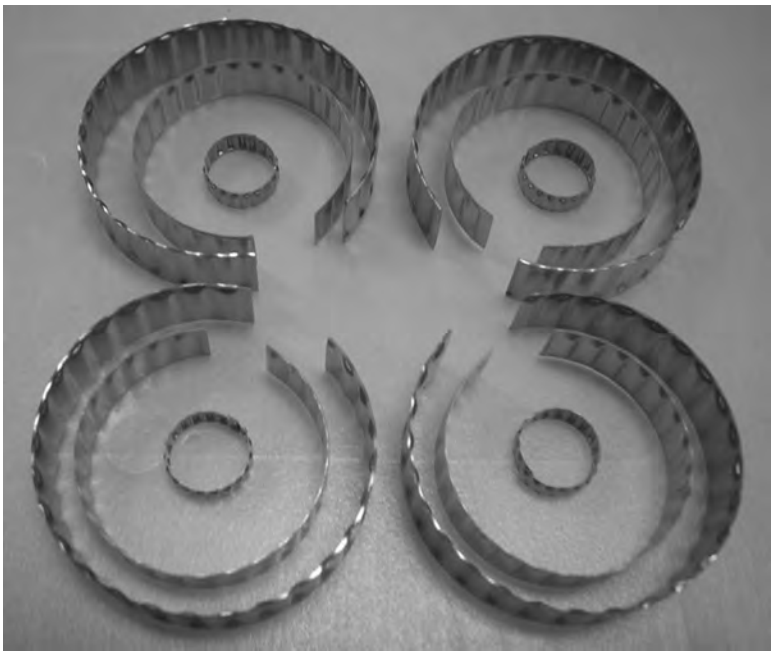


FIGURE 2.41
Using tolerance rings between mating parts such as shaft and rotor core.

relationship between the spring force F_s applied on the tolerance ring and the deflection of the ring is given as

$$F_s = -kx \tag{2.33}$$

This minus sign means that the spring force F_s is always in the opposite direction of the displacement x. In fact, the compressive force acting on the tolerance ring always flattens the corrugations.

The friction force of tolerance springs is determined as

$$F_f = \mu N \tag{2.34}$$

where

μ is the coefficient of static friction
N is the normal force on the contacting surfaces

This force is important for holding the mating parts together firmly without sliding on each other. While motor torque capacities are related to the amount of interference and coefficient of static friction, motor radial load capacities relay on yield limit of the material, the preload, and the cumulative compression of the corrugations caused by interference fit.

Tolerance rings provide many design advantages over other conventional fitting methods. They are used with great success in compensating for different thermal expansion of mating parts. As shown in Figure 2.42, without using tolerance rings, the pressing force between mating parts reduces sharply as the temperature goes higher due to the

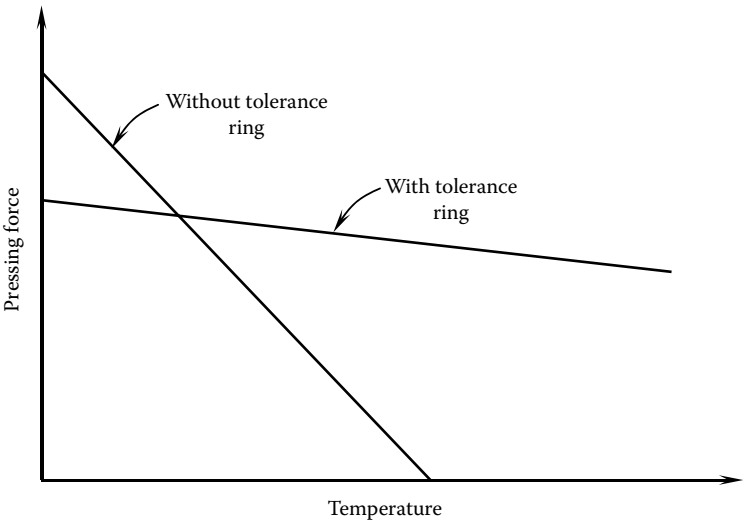


FIGURE 2.42
Comparison pressing forces at different temperatures with and without tolerance rings.

thermal expansion of the mating parts. In contrast, with the tolerance rings, the pressing force only reduces slightly with the temperature increase. For instance, for a 50.8 mm (2 in.) diameter ball bearing on an aluminum housing, the installation pressing force is 2224 N (500 lb_f) at room temperature. The bearing becomes loose at a temperature of 77°C (170°F) as the result of the higher thermal expansion of the aluminum housing than that of the steel bearing. With the tolerance ring, an initial installation pressing force is only (320 lb_f), and retention still remains high at 1023 N (230 lb_f) even at the temperature of 132°C (270°F).

Because tolerance rings allow for a broad range of tolerance, this permits the use of loosen tolerances for the mating parts (e.g., shaft and rotor core). As a result, the requirements for the surface finish on the mating parts are obviously reduced. Another advantage is that the use of tolerance rings can reduce the vibration of the system and lower the noise emission from the system. Finally, the use of tolerance ring can significantly simplify the assembly and disassembly processes.

However, the transmitted torque through the tolerance ring fitting is not as high as those through press fitting and shrink fitting, indicating that the application of tolerance rings is limited to small-rating motors.

2.5 Stress Analysis of Rotor

One of the most important parameters in the rotor design is the length-to-diameter aspect ratio (i.e., *L/D* ratio). A rotor with a low aspect ratio has high rotor stiffness and thus has a high critical speed. The determination of *L/D* ratio is based on factors such as motor torque, spin stresses, rotor dynamic characteristics, and heat loads. Small *L/D* ratios mean high tangential speed and high centrifugal forces at the rotor surface. However, a rotor with a large diameter and a small *L/D* ratio may have large windage losses, especially for

high-speed motors. Normal *L/D* ratio ranges 0.5–1.0 for wound rotor motors and 1–3 for PM motors [2.57]. The tangential speed on the rotor surface u_t is calculated as

$$u_t = r\omega = \frac{d\omega}{2} \tag{2.35}$$

To restrict the centrifugal force of a rotor, the upper limit of the rotor tangential speed is set in the range of 100–250 m/s, depending on motor applications.

During rotor rotation, centrifugal forces are produced in the rotor and in turn generate stresses in the circumferential and radial directions. In general, hoop tensile stresses are dominant and play a decisive role in the selection of rotor material. The stress resulted from centrifugal loading is governed by the radial equilibrium equation in the cylindrical coordination system [2.58]:

$$\frac{d\sigma_r}{dr} + \frac{1}{r}\left(\sigma_r - \sigma_\theta\right) + \rho\omega^2 r = 0 \tag{2.36}$$

where

σ_r and σ_θ are the radial and circumferential stresses, respectively
ρ is the density
ω is the rotational angular speed

Thus, a linear relationship between stress vector $\boldsymbol{\sigma}$ and strain vector $\boldsymbol{\varepsilon}$ can be written as

$$\boldsymbol{\sigma} = \mathbf{k}(\boldsymbol{\varepsilon} - \boldsymbol{\alpha}\Delta T) \tag{2.37}$$

where

$\mathbf{k}$ is the stiffness matrix
$\boldsymbol{\alpha}$ is the vector of thermal expansion coefficient
ΔT is the temperature difference

The previous equation can be expressed in the matrix form as

$$\begin{Bmatrix} \sigma_\theta \\ \sigma_z \\ \sigma_r \end{Bmatrix} = \begin{bmatrix} k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{31} & k_{32} & k_{33} \end{bmatrix} \begin{Bmatrix} \varepsilon_\theta \\ \varepsilon_z \\ \varepsilon_r \end{Bmatrix} - \begin{bmatrix} \alpha_\theta \\ \alpha_z \\ \alpha_r \end{bmatrix} \Delta T \tag{2.38}$$

By ignoring the quadratic terms of the deformation, the strains in circumferential and radial directions can be lineally related to the radial displacement u_r; the strain in the axial direction (z) is assumed to linearly vary along the radial direction:

$$\begin{cases} \varepsilon_\theta = \dfrac{u_r}{r} \\ \varepsilon_r = \dfrac{\partial u_r}{\partial r} \\ \varepsilon_z = \varepsilon_o + \varepsilon_1 r \end{cases} \tag{2.39}$$

Substituting Equations 2.38 and 2.39 into 2.36, the governing equation for radial displacement u_r is obtained. A closed-form solution is derived in detail by Ha et al. [2.59].

2.6 Rotordynamic Analysis

Rotordynamics is a specialized branch of applied mechanics concerned with the behavior and stability of rotating machinery. It plays an important role in improving the safety and reliability of electric motors. To design a robust motor, it is necessary to perform both steady-state and transient lateral and torsional calculations.

With the fast development of advanced computing techniques in recent decades, there are many software packages that are capable of solving the rotordynamic system of equations. Each software package has its specific capabilities and applications. Today, because of the complexity of the modern rotating machinery, rotordynamic software packages have been extensively used to analyze the behavior of structures ranging from gas and steam turbines to auto engines.

Rotordynamic results strongly rely on many factors such as motor structure, bearing type, rotor inertia, rotating speed, mass distribution in rotating system, motor component materials, and motor mounting pattern. As illustrated in Figure 2.43, a motor is fixed on the frame of a machine via the drive-side endbell of the motor. In order to perform rotordynamic analysis of the rotor, the stiffness and damping of the bearings, endbells, and the motor housing must be determined prior to the analysis. The rotordynamic model is shown in Figure 2.44, where S_{br1} and S_{br2} and c_{br1} and c_{br2} are the stiffness and damping of the bearings 1 and 2; S_{ed1} and S_{ed2} are the stiffness of the endbells 1 and 2, respectively; and S_h is the housing stiffness. The bearing stiffness and damping are functions of the rotor rotating speed ω. The support properties of the endbells and housing may be also speed dependent.

2.6.1 Rotor Inertia

In high-speed or large-scale motors, the rotor inertias can be considerably large. The total inertia of the motor rotating system can be obtained by summing the component

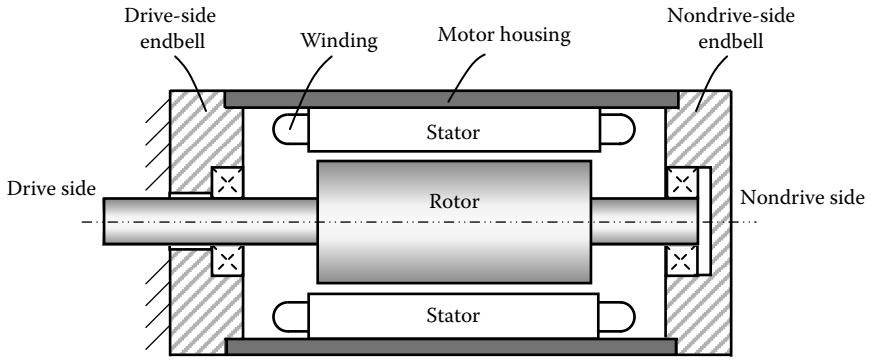


FIGURE 2.43
A motor is mounted to the driven machine via drive-side endbell.

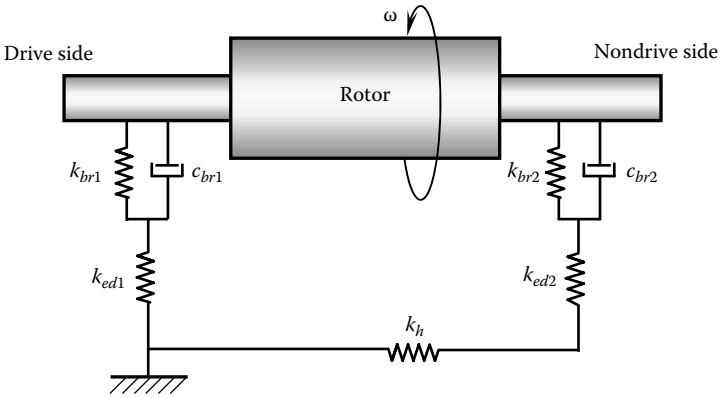


FIGURE 2.44
Rotordynamics model for a motor mounted at one end.

inertias together, such as the rotor core inertia, shaft inertia, and bearing rotating component inertia:

$$J_r = J_{core} + J_{shaft} + J_{bearing} \tag{2.40}$$

The formulas for determining the inertia and spring constant of rotating bodies with various shapes are given in Figure 2.5. The bearing inertia consists of two components: the inertias of the inner raceway and rolling elements

$$J_{bearing} = J_{ir} + NJ_{rolling} \tag{2.41}$$

where N is the number of rolling elements. The inertia of the inner ring is

$$J_{ir} = \frac{\pi l \rho (d_{ir,o}^4 - d_{ir,i}^4)}{32} \tag{2.42}$$

where $d_{ir,o}$ and $d_{ir,i}$ are the outer and inner diameters of the bearing inner ring, respectively (Figure 2.45). For a rolling bearing, the inertia of each rolling element with respect to the bearing centerline is

$$J_{roll} = m\left[K\left(\frac{d_{roll}}{2}\right)^2 + l^2\right] \tag{2.43}$$

where

m and d_{roll} are the mass and diameter of the rolling element, respectively
l is the distance from the center of the rolling element to the bearing centerline
K is the geometric constant of the rolling element, where $K = 2/5$ for sphere bolls and $K = 1/2$ for rollers

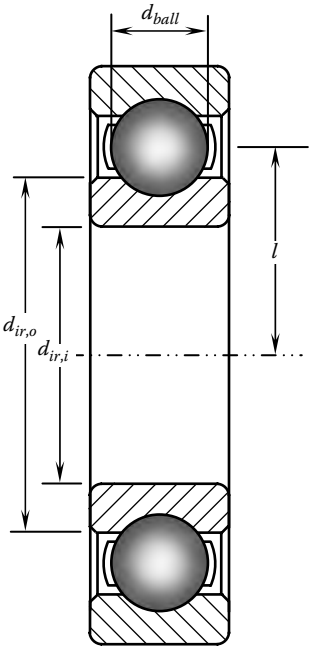


FIGURE 2.45
Calculation of inertia of rolling bearing with respect to the bearing centerline.

2.6.2 Motor Critical Speed and Resonance

A rotor assembly consists of a rotor core and a shaft, fitted together tightly. The rotor critical speed refers to the speed at which a system resonance is excited. In such a case, the centrifugal force associated with even a small mass unbalance can cause a vibration of high amplitude. In worse cases, it may lead to a disintegration of a motor within a few seconds.

For some small lightweight motors with low rotating speeds, the operation speed is far below the critical speed. Therefore, the verification of critical speed for this type of motor may be unnecessary. Some motors are designed to operate at a rotational speed above the critical speed. It can work well if the motors accelerate quickly through the critical speed before the vibration buildup to an excessive amplitude.

Every rotating machine has its own natural frequency at which all attached components and the machine structure self-vibrate, known as resonance. Theoretically, natural frequency is directly proportional to the stiffness and inversely proportional to the mass of the machine. The accurate calculation of critical speed and natural vibration frequency is quite complex if all factors are taken into account. In practice, there are two methods used to predict critical speed and natural frequency: Rayleigh-Ritz and Dunkerley methods. Both Rayleigh-Ritz and Dunkerley methods are an approximation to the first natural frequency of vibration. Generally, the Rayleigh-Ritz equation overestimates and the Dunkerley equation underestimates the natural frequency. Good practice suggests that the maximum operation speed should not exceed 75% of the critical speed [2.60].

Both resonance and critical speeds are frequencies that are governed by natural frequencies, damping, and vibration forces. A resonance is a condition in a structure in which the

frequency of the vibrating force is equal to the natural frequency of the system. For the rotation-excited vibration, the resonance is defined as the critical speed.

During operation, a motor is subject to a variety of forces, including mechanical and electromagnetic forces. Though the calculation of all motor forces is considerably complex, some factors can be identified as follows:

- When a rotor is spinning, any unbalance on the rotor causes a large centrifugal force deflection.
- Because of finite number of stator and rotor slots (or magnets in a PM motor), it generates the unbalanced magnetic force.
- Force due to the torque is transmitted to the load.

Regularly, torsional critical speed is less of a problem in motors but can be significant for large *L/D* ratio motors.

Considering a rotor assembly in a motor as presented in Figure 2.46a, the rotor is positioned at the center of the system, supported by two roller bearings at the each end of the rotor shaft. This physical model can be converted into a rotordynamic model, as shown in Figure 2.46b.

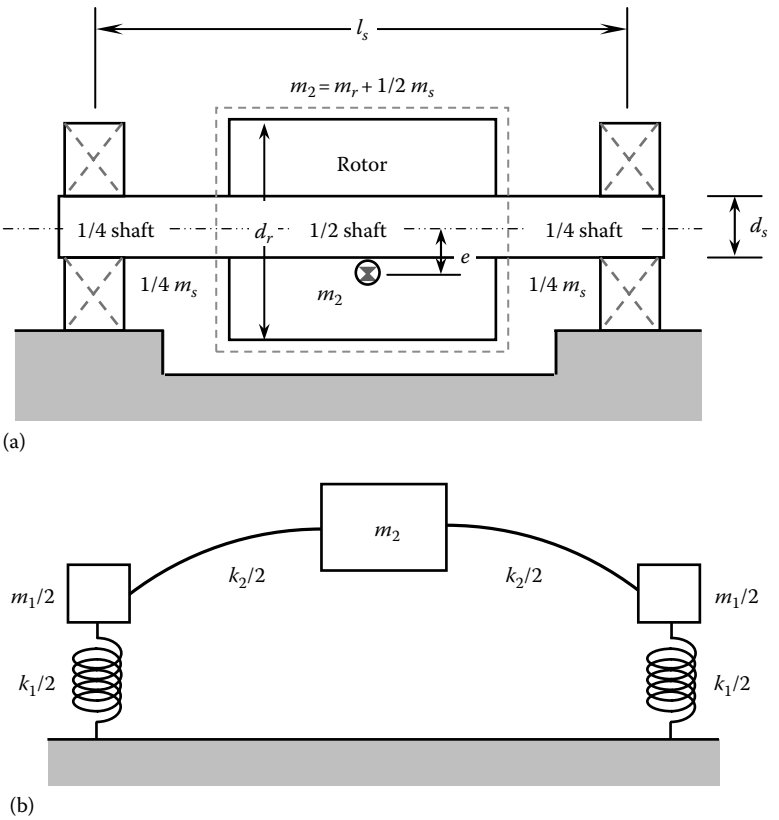


FIGURE 2.46
Rotor models: (a) physical model and (b) rotordynamic analytical model.

The system can be divided into three sections: the middle section, which contains the rotor core and one-half of the shaft, and two end sections, each containing one-quarter of the shaft. In order to perform the rotordynamic analysis, this system can be further simplified to a two-degree-of-freedom model with two masses, m_1 and m_2 [2.61]:

$$m_1 = \frac{1}{2} m_s \tag{2.44}$$

$$m_2 = m_r + \frac{1}{2} m_s \tag{2.45}$$

where m_s and m_r are the shaft and rotor mass, respectively. The exciting force F_u is resulted from the eccentric mass m_2 in the rotor system and is given by

$$F_u = m_2 e\omega^2 \sin(\omega t - \varphi) \tag{2.46}$$

The above system can be simplified as a two-degree-of-freedom rotordynamic model, as shown in Figure 2.47. There are two equations of motion for this system, one for each degree of freedom. These two equations are generally in the form of coupled differential equations:

$$m_1 \frac{d^2 x_1(t)}{dt^2} + (k_1 + k_2)x_1 - k_2 x_2 = 0 \tag{2.47}$$

$$m_2 \frac{d^2 x_2(t)}{dt^2} + k_2 x_2 - k_2 x_1 = (m_2 e\omega^2)\sin(\omega t - \varphi) \tag{2.48}$$

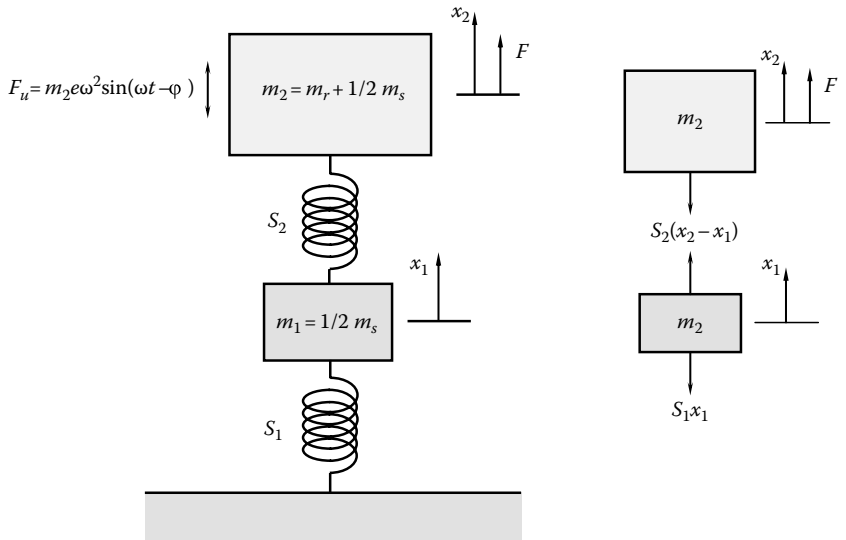


FIGURE 2.47
Reduced two-degree-of-freedom rotordynamic model for rotor assembly, where the exciting force is produced due to the rotor unbalance.

Hesai Exhibit 1031, page 176

The solutions of the motion equations may take the form

$$x_1(t) = X_1 \sin(\omega t - \varphi) \tag{2.49}$$

$$x_2(t) = X_2 \sin(\omega t - \varphi) \tag{2.50}$$

where
X_1 and X_2 are constants denoting the maximum amplitudes of $x_1(t)$ and $x_2(t)$
φ is the phase angle

Thus, the motion equations become

$$[-m_1\omega^2 + (k_1 + k_2)]X_1 - k_2X_2 = 0 \tag{2.51}$$

$$-k_2X_1 + (-m_2\omega^2 + k_2)X_2 = m_2e\omega^2 \tag{2.52}$$

These two equations can be expressed in a matrix form

$$\begin{bmatrix} (k_1 + k_2) - m_1\omega^2 & -k_2 \\ -k_2 & k_2 - m_2\omega^2 \end{bmatrix} \begin{Bmatrix} X_1 \\ X_2 \end{Bmatrix} = \begin{bmatrix} 0 \\ m_2e\omega^2 \end{bmatrix} \tag{2.53}$$

This can be solved to obtain X_1 and X_2:

$$X_1 = \frac{k_2m_2e\omega^2}{AB - k_2^2} \tag{2.54}$$

$$X_2 = \frac{m_2e\omega^2}{B - \frac{k_2^2}{A}} \tag{2.55}$$

where
$A = (k_1 + k_2) - m_1\omega^2$
$B = k_2 - m_2\omega^2$

By setting the determinant of the dynamic stiffness matrix in Equation 2.53 to be equal to zero, the fourth-order frequency equation can be obtained as

$$\omega^4 + \left(-\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2}\right)\omega^2 + \frac{k_2(k_1 + k_2)}{m_1m_2} = 0 \tag{2.56}$$

This equation can be solved using the quadratic formula,

$$\omega^2 = \frac{-B \pm \sqrt{B^2 - 4C}}{2} \tag{2.57}$$

where

$$B = -\frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2}$$

$$C = \frac{k_2\left(k_1 + k_2\right)}{m_1 m_2}$$

This leads to the two natural frequencies

$$\omega_1 = \left[\frac{-B + \sqrt{B^2 - 4C}}{2}\right]^{1/2} \tag{2.58a}$$

$$\omega_2 = \left[\frac{-B - \sqrt{B^2 - 4C}}{2}\right]^{1/2} \tag{2.58b}$$

2.7 Rotor Burst Containment Analysis

For high-speed motors, a key technical issue is the operation safety of electric motor. This is especially true for some applications such as elevators, roller coasters, and cableways. According to rotordynamics, rotor burst refers to the phenomena that a rotor breaks suddenly at high rotating speeds. When this failure occurs, the motor stator and housing must function as a containment to retain all debris to avoid severe personal injuries or disasters. For this reason, rotor burst containment is a critical design requirement regarding motor safety.

2.7.1 Rotor Burst Speed

The rotor burst speed can be predicted using different approaches. A logical approach is that rotor burst occurs when the hoop stress of a rotor is equal to the ultimate tensile strength of the rotor material. Consider a rotor core assembled with a shaft to form a rotor, with an inner and outer radii r_i and r_o, respectively, and a length l. The rotor is contained inside a stator and a housing (Figure 2.48). A differential element in the rotor has the radial thickness dr and the circumferential length $rd\theta$. The centrifugal force acting on the rotor element due to the rotor rotating is given as

$$dF = (r\omega^2)dm \tag{2.59}$$

where the mass of the element dm is found as the product of the material density ρ and the element volume dV:

$$dm = \rho dV = \rho l(rd\theta)dr \tag{2.60}$$

At an equilibrium condition, all forces acting on the element must be balanced, that is,

$$\sum_i F_i = 0 \tag{2.61}$$

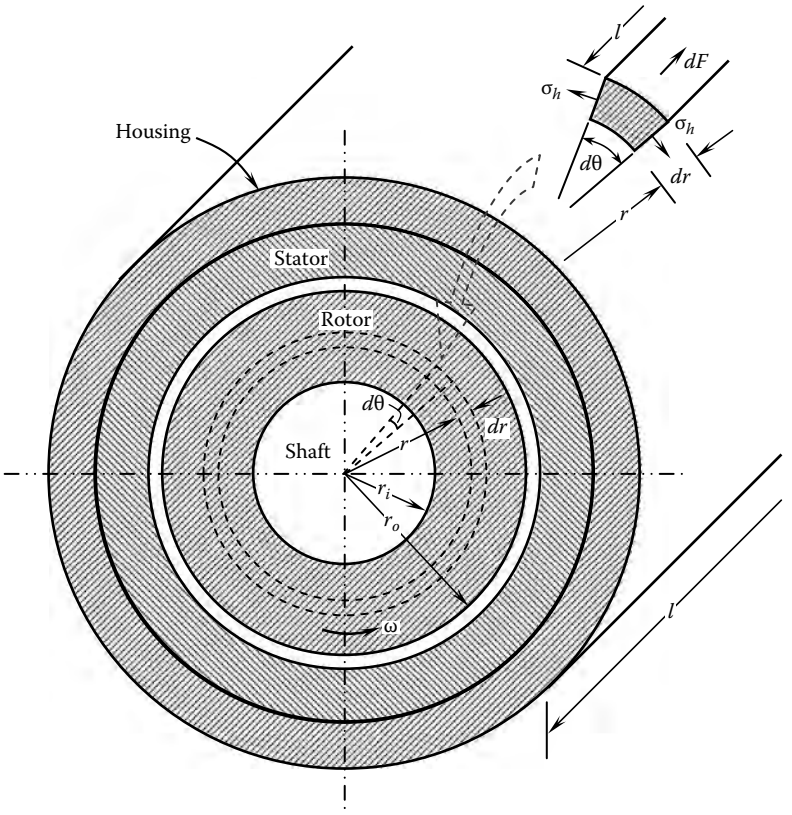


FIGURE 2.48
Rotor is contained inside a stator and a housing that functions as the containment of the rotor.

It follows that

$$dF = 2l(dr)\sigma_h \sin\left(\frac{d\theta}{2}\right) \cong l(dr)\sigma_h\theta \tag{2.62}$$

where σ_h is the hoop stress on the element and $\sin(d\theta/2) \cong d\theta/2$ for small angle of $d\theta$. Thus, the hoop stress σ_h at any radius r becomes

$$\sigma_h = \rho(r\omega)^2 \tag{2.63}$$

This equation indicates that the hoop stress of the rotor is independent of the rotor length l. The maximum hoop stress occurs at the rotor outer surface where $r = r_o$:

$$\sigma_{h,\max} = \rho(r_o\omega)^2 \tag{2.64}$$

The rotor burst speed ω_b is thus determined as $\sigma_{h,\max}$ is equal to or larger than the ultimate tensional strength of the rotor material S_{ut}:

$$\omega_b = \frac{1}{r_o}\sqrt{\frac{S_{ut}}{\rho}} \tag{2.65}$$

This indicates that the rotor burst speed ω_b is both design-dependent (the rotor OD r_o) and material-dependent (square root of the material's ultimate strength-to-density ratio).

2.7.2 Energy in Rotating Rotor

For a rotating rotor, the dominate energy stored in the rotor is the kinetic energy about the center of the rotor mass. In addition, the elastic potential energy is also stored in the rotating rotor because of circumferential tensile stress and elongation during operation.

2.7.2.1 Kinetic Energy in Rotor

The kinetic energy of a rotating rotor is proportional to the product of the polar moment of inertia of the rotor J_p and the square of rotor rotating speed ω, that is,

$$KE = \frac{1}{2} J_p \omega^2 = \frac{1}{2} J_p \left(\frac{\sigma_h}{\rho r^2} \right) \tag{2.66}$$

From this equation, it can be deduced that the kinetic energy of the rotor increases quadratically with the angular velocity ω. Furthermore, Equation 2.66 indicates that the rotor diameter also has a greater influence on kinetic energy. In the previous equation, J_p can be expressed as

$$J_p = \int r^2 dm = \int r^2 \rho dV \tag{2.67}$$

Referring to Table 2.5, the differential volume of the rotor dV is

$$dV = 2\pi r l dr \tag{2.68}$$

Hence, for a hollow cylinder, the polar moment of inertia becomes

$$J_p = 2\pi l \rho \int_{r_i}^{r_o} r^3 dr = \frac{\pi l \rho}{2} \left(r_o^4 - r_i^4 \right) \tag{2.69}$$

The kinetic energy of a rotor at any rotating speed of ω is given as

$$KE = \frac{1}{2} J_p \omega^2 = \frac{\pi l \sigma_h}{4 r_o^2} \left(r_o^4 - r_i^4 \right) \tag{2.70}$$

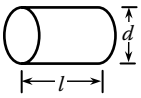
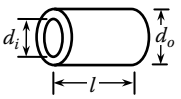
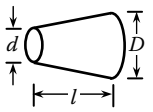
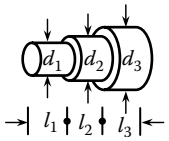
At the burst rotating speed $\omega = \omega_b$, the hoop stress is equal to the ultimate tensile strength of the rotor material. The kinetic energy of the rotor is

$$KE_b = \frac{1}{2} J_p \omega_b^2 = \frac{\pi l S_{ut}}{4 r_o^2} \left(r_o^4 - r_i^4 \right) \tag{2.71}$$

Hesai Exhibit 1031, page 180

TABLE 2.5

Polar Moment of Inertia J_p and Torsional Stiffness S_t for Various Types of Shafts

Shaft Type	Shaft Shape	Polar Moment of Inertia J_p (kg-m^2)	Torsional Stiffness S_t (Nm/rad)
Solid circular shaft		$J_p = \frac{\pi l \rho d^4}{32}$	$S_t = \frac{\pi G d^4}{32l}$
Hollow circular shaft		$J_p = \frac{\pi l \rho \left(d_o^4 - d_i^4\right)}{32}$	$S_t = \frac{\pi G \left(d_o^4 - d_i^4\right)}{32l}$
Tapped circular shaft		$J_p = \frac{3\pi}{32} \frac{l \rho d^4}{(n + n^2 + n^3)}$ $n = \frac{d}{D}$	$S_t = \frac{3\pi}{32l} \frac{G d^4}{(n + n^2 + n^3)}$ $n = \frac{d}{D}$
Stepped shaft		$J_p = \sum_i J_{p,i} = \sum_i \frac{\pi l_i \rho d_i^4}{32}$	$S_t = \frac{1}{\sum_i \frac{1}{S_i}}$

2.7.2.2 Elastic Potential Energy in Rotor

The elastic potential energy in a rotating rotor is given as [2.62]

$$PE = \frac{1}{2} \int_V \sigma_h \varepsilon_h dV \tag{2.72}$$

where ε_h is the hoop strain (circumferential strain) in the rotor and

$$\varepsilon_h = \frac{\sigma_h}{E} = \frac{\rho r^2 \omega^2}{E} \tag{2.73}$$

where E is the Young's modulus. Thus, the elastic potential energy is expressed as

$$PE = \frac{\pi l \rho^2 \omega^4}{6E} \left(r_o^6 - r_i^6\right) \tag{2.74}$$

At the rotor burst speed, the elastic potential energy reaches its maximum value

$$PE_b = \frac{\pi l \rho^2 \omega_b^4}{6E} \left(r_o^6 - r_i^6\right) = \frac{\pi l S_{ut}^2}{6E r_o^4} \left(r_o^6 - r_i^6\right) \tag{2.75}$$

This indicates that the elastic potential energy at the burst speed is dependent on the rotor geometry and material properties but independent of the rotor mass.

2.7.2.3 Ratio of Potential Energy to Kinetic Energy of Rotor

The ratio of the potential energy to the kinetic energy becomes

$$\frac{PE}{KE} = \frac{2}{3}\frac{\rho r_o^2 \omega^2}{E}\left[\frac{1-(r_i/r_o)^6}{1-(r_i/r_o)^4}\right] \tag{2.76}$$

When the rotor fails at the rotor burst speed, this ratio becomes

$$\frac{PE_b}{KE_b} = \frac{2}{3}\frac{S_{ut}}{E}\left[\frac{1-(r_i/r_o)^6}{1-(r_i/r_o)^4}\right] \tag{2.77}$$

As pointed out by Genta [2.63], the kinetic energy is at least one or two orders of magnitude greater that the energy needed to deform the rotor until failure occurs.

2.7.3 Rotor Burst Containment Design

At the rotor burst speed, a rotor splits into fragments that attempt to follow tangential trajectories to strike the internal wall of the stator. When the fragments impact with the stator, it is the kinetic energy of the fragments that get converted into kinetic and strain energy of the stator. Also, part of the energy gets dissipated in the form of heat energy due to the fragments rubbed with the stator inner surface. The released fragments can be contained within the stator if the kinetic energy of the fragments is less than the sum of the shear and strain energy of the stator material. In any cases, stators have to withstand very high-energy impact resulting from broken rotors.

However, the accurate prediction of the containment response is considerably complex due to the following factors: (1) the prediction of the shape and size of rotor debris would be very difficult, if not impossible; (2) there are uncertainties associated with the kinetics of the rotor debris; and (3) the determination of the impact parameters such as the impact area, points, and angle of each fragment relative to the containment wall is extremely difficult. Therefore, in practical containment design, rotor burst tests and FEMs are often used to determine the key design parameters of the containment structure.

Pichot et al. [2.64] have developed the loading models to analyze the containment response following rotor burst. With these models, the radial, axial, and torsional loadings on the containment wall are determined using fundamental energy and momentum principles. In order to overcome the difficulties due to the uncertainties associated with the kinetic energy of the debris, two separate models were developed. The first model is called the debris deflection model, in which the debris is assumed to impact the inside wall of the containment and to be immediately deflected axially without any accumulation on the wall. The second model is called the debris accumulation model, which assumes that all of the fragments pack into a debris bed against the wall without any axial deflection. In fact, these two models represent two extreme cases and thus provide the bounds for the real case. In analyzing the containment response, the debris deflection model tends to produce an impulsive loading that maximizes the magnitude of the axial loading. On the other hand, the debris accumulation model tends to produce longer duration loading and maximizes the torque on the containment.

Hagg and Sankey [2.65] have derived an analytical solution to predict the thickness of the containment shell in the event of turbine disk burst by comparing strain and shear energy with the total energy of disk fragmented missile before impact. Their solution is based on the assumption that the kinetic energy lost by the disk fragments during impact is converted into kinetic energy in the containment shell and energy loss to plastic strain and shear failure in the shell. The minimum perforation energy occurs with the normal force imposed by the translational motion of an impacting fragment.

In the studies of Hagg and Sankey [2.65], the containment process consists of two stage events. The first stage accounts for localized perforation failure where the fragments perforate the containment shell upon initial impact. The equations assume that perforation failure occurs when the energy transfer during impact as measured by the energy loss of the burst fragment exceeds the maximum compression and shear strain energy that the contact zone of the containment shell is capable of absorbing. The maximum compression and shear strain energies in the shell are based on the compressive flow stress, the failure strain, and the shear strengths of the shell material. Assuming that the impact is entirely inelastic and there are no losses to friction or heat, the energy lost during the impact process E_{loss} equals the kinematic energy of the fragment before the impact KE_f minus the residual energy E_{res} [2.66]:

$$E_{loss} = KE_f - E_{res} = \frac{1}{2} m_f v_f^2 \left(1 - \frac{m_f}{m_f + m_c} \right) \tag{2.78}$$

where the kinematic energy of the fragment KE_f is based on the normal velocity of the fragment to the wall of the containment shell v_f and m_f and m_c are the mass of the fragment and the effective mass of the containment shell that responds to the initial contact and momentum transfer, respectively.

The energy required for straining the fragments E_{strain} consists of compressive strain energy in compression E_{comp} and shear strain energy in shearing E_{shear} [2.67]:

$$E_{strain} = E_{comp} + E_{shear} = At\varepsilon\sigma_d + C\tau_d pt^2 \tag{2.79}$$

where

A is the striking area
t is the thickness of the containment shell
ε is the per unit plastic strain
σ_d is the dynamic stress of the containment shell in compression
τ_d is the dynamic shear stress of the containment shell in shearing
p is the perimeter of the sheared area
C is the experimental constant with a value in the range of 0.3–0.5

Hence, the necessary and sufficient condition for straining released fragments is

$$E_{loss} < E_{strain} \tag{2.80}$$

That is,

$$\frac{1}{2} m_f v_f^2 \left(1 - \frac{m_f}{m_f + m_c} \right) < At\varepsilon\sigma_d + C\tau_d pt^2 \tag{2.81}$$

Therefore, the minimum thickness of the containment shell is determined as

$$t_{\min} = \frac{1}{2C}\left(\frac{A}{p}\right)\left(\frac{\varepsilon\sigma_d}{\tau_d}\right)\left[-1+\sqrt{1+\frac{2C\tau_d p}{\left(A\varepsilon\sigma_d\right)^2}\left(m_f v_f^2\right)\left(1-\frac{m_f}{m_f+m_c}\right)}\right] \tag{2.82}$$

This calculated value of $t_{\min}$ is to be compared to the total thickness of the stator and housing. In a case that $t_{\min}$ is larger than the total thickness, either the stator or housing must be redesigned.

If localized perforation failure does not occur, the process moves to the second stage. In this stage, the residual energy is dissipated in the form of tensile strain throughout an extended volume of the shell material. As discussed previously, the residual energy is

$$E_{res} = \frac{1}{2}m_f v_f^2\left(\frac{m_f}{m_f+m_c}\right) \tag{2.83}$$

Failure occurs when the remaining energy exceeds the allowable strain energy in this extended volume.

Alternatively, several turbine manufacturers have presented their models to estimate the minimum thickness of the containment shell. Each of the models may involve different operation conditions and applications. These empirical formulas are listed here for the purpose of reference:

General Electric [2.68]

$$t_{\min} = C_1\sqrt{KE_f} \tag{2.84}$$

Pratt and Whitney [2.69]

$$t_{\min} = C_2\sqrt{\frac{B(KE_f)}{\tau_d p}} \tag{2.85}$$

Snecma [2.70]

$$t_{\min} = \sin(\alpha)\sqrt{\frac{KE_f}{\tau_d p}} \tag{2.86}$$

where

C_1 and C_2 are the empirical constants
KE_f is the translational kinetic energy of the fragment
B is the blade buckling factor
α is the fragment impact angle

It is worth to note that no matter how advanced computational techniques have been developed for predicting the rotor burst, development of a reliable and effective containment system for electric motor eventually requires extensive experimental testing to verify the analytical predictions. This usually requires a large amount of investment to build up a specific testing lab.

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Hesai Exhibit 1031, page 187

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3

Shaft Design

A shaft in an electric motor is used to transmit torque and power from the motor to an external loading machine. Because motor shafts are subjected to various combined effects of tension, compression, bending, and torsion during operation, they are typically designed for maximum stiffness and rigidity and minimum deflection to maintain shaft stress/strain below allowable limits under various loading and operating conditions. The achievement of such design objectives relies on the selection of shaft material, determination of suitable shaft dimensions (especially shaft diameter and span between the bearings) and structures, mitigation/elimination of stress concentrations, and other design activities. Obviously, each of these can affect the long-term reliability of motor.

As a rotating component, the motor shaft is subject to a completely reversed bending load that results in an alternating bending stress in the shaft. Furthermore, some motors require to frequently change the direction of rotation or experience frequent starts and stops during operation, causing cyclic torsional stress in motor shafts. Therefore, the prevention of fatigue failure is an important consideration in the shaft design.

A typical shaft of an electric motor is illustrated in Figure 3.1. Typically, the motor shaft is supported by two bearings at its ends and a rotor core mounted on the shaft between the two bearings. Motor shafts usually carry keyed or splined power-transmitting components (e.g., sheaves, pulleys, couplers, and sprockets) on the overhanging end to transmit torque to an externally driven machine. The shaft span is generally designed as short as possible to increase the shaft stiffness and reduce the shaft stress and lateral deflection. In addition, minimizing the shaft overhang is highly desired for lowering shaft bending stress.

The importance of engineering analysis and design of shafts has been widely recognized by engineers and designers. However, most machine design textbooks have mainly focused on shafts with uniform diameters. Such shafts are easy to produce but rare in practice. To help design motor shafts, this chapter emphasizes on the design for stepped motor shafts.

3.1 Shaft Materials

The selection of proper shaft material is critical for ensuring motor normal and safe operation. Depending on different applications, motor shafts can be made of low- to medium-carbon steel, cast iron, stainless steel, aluminum alloys, brass, and bronze. The criteria of the shaft material selection include material mechanical and thermal properties, rigidity, hardness, wear resistance, machinability, noise absorption, manufacturing process, and cost.

By far the most widely used shaft material is carbon steel. The mechanical and thermal properties of carbon steel primarily depend on the amount of carbon it contains. Increasing

Hesai Exhibit 1031, page 190

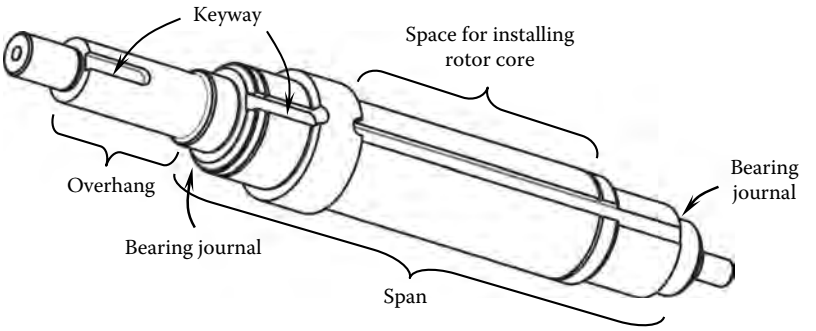


FIGURE 3.1
Structure of a motor shaft.

carbon contents can result in the increase of hardness, yield tensile strength, and ultimate tensile strength. However, higher carbon contents also increased material brittleness and reduced weldability, machinability, and elongation.

According to the content of carbon in steel, carbon steel can be categorized into approximately three types: low-carbon, medium-carbon, and high-carbon steel. Low-carbon steel contains carbon up to 0.25%. Medium-carbon steel has carbon content ranging from 0.25% to 0.70%. High-carbon steel contains carbon in the range of 0.70%–1.50%. Most motor shafts are made of steel containing 0.2%–0.5% carbon. With a single-digit elongation, high-carbon steel is too brittle to be used as shaft material.

To improve the mechanical properties of shafts, such as hardness, yield and ultimate tensile strength, fatigue strength, and other material properties, motor shafts are often heat-treated. Heat treatment processes include spheroidizing, annealing, normalizing, carburizing, quenching, martempering, and austempering. Each process has its own purpose and is applied to certain types of materials.

A spheroidizing process is especially used for high-carbon steel with carbon content more than 0.6%. Spheroidite forms when carbon steel is heated to approximately 700°C for more than 30 h. The purpose of this process is to soften high-carbon steels and allow more formability. An annealing process is often used to relieve residual stresses in the cold-worked steel with the carbon content of 0.3% or above. Fully annealed steel has no residual stresses and becomes ductile. Normalizing helps steel with a fine and more uniform pearlitic structure. Normalized steel still maintains high strength and durability. For low-carbon steel, carburizing can increase the hardness and improve other mechanical properties. During the process, carbon molecules penetrate into the material surface. The affected area and depth of carbon content are dependent on the heating time and temperature. Generally, higher temperatures and longer heating time lead to greater carbon diffusion into steel. The carbon content of quenchable steel must be 0.3% or above. A quenching process can increase yield and ultimate tensile strength, fatigue strength, and hardness but decrease material elongation. Usually, quenched steel is about three or four times harder than normalized steel. However, residual stresses can be introduced during the quenching process in the bulk of the shaft that may cause cracks on the shaft surface. In all heat treatment processes, the heating and cooling rate and temperature holding time are critical parameters to impact the metallographic microstructure of steel.

Either cold-rolled or hot-rolled carbon steel can be used for making motor shafts. Hot-rolled steel is produced at temperatures above the recrystallization temperature of steel.

As a contrast, cold-rolled steel is produced under the recrystallization temperature of steel (more regularly, at the room temperature). The benefits of cold-rolled steel over hot-rolled steel are as follows: (1) The strength of cold-rolled steel can increase by 15%–30% due to the strain-hardening mechanism. (2) Cold-rolled steel has smooth surface finish compared to hot-rolled steel. (3) Cold-rolled steel products have tight dimensional tolerances. However, the cold rolling process can generate residual stresses remaining in the rolled steel. Furthermore, the cost of cold rolling process is usually higher than that of hot rolling.

For some applications that require high material strengths or operate under severe environmental conditions (e.g., low or high temperatures, high humidity, acid and alkali corrosion), motor shafts may be also made of special steel alloys.

Stainless steel can be used as a shaft material when motors work under corrosive environments or for some special applications such as food industry and medical equipment. With a minimum chromium (Cr) content of 10.5%, stainless steel is continuously protected by a passive layer of chromium oxide that forms naturally on the surface through the combination of chromium and moisture in the air. This enhances corrosion resistance of stainless steel. The addition of nickel (Ni) can also raise the corrosion resistance of stainless steel. However, machining stainless steel is more difficult than carbon steel because of heat buildup during the machining process and difficulties in chip breaking. Stainless steel alloy 304 is one of the most widely used stainless steel that contains 18% of chromium, 8% of nickel, and lower carbon (0.08%) to minimize carbide precipitation. This type of stainless steel possesses an excellent combination of strength, corrosion resistance, and fabricability [3.1]. However, because the coefficient of thermal expansion (CTE) of stainless steel is approximately 25%–50% higher than that of carbon steel, high thermal stresses on stainless steel shafts may be developed as the result of temperature rise in motors.

Cast iron is a large family of ferrous alloys in which four types can be categorized: gray cast iron, ductile iron, white iron, and metallurgic iron. Cast iron has been widely used to make crankshafts in internal combustion engines (ICEs). Crankshafts convert the reciprocating motion into rotation by crankpins. One of the main reasons to use cast iron is that it has excellent vibration damping properties to reduce the pulsation influence of the four-stroke cycle. In addition, cast iron exhibits low notch sensitivity, low modulus of elasticity, high thermal conductivity, moderate resistance of thermal shock, and outstanding castability. It is interesting to note that unlike most ferrous materials, shear strength of gray cast iron is much higher than ultimate tensile strength, indicating that gray cast iron can withstand higher shear forces than tensile forces. However, until now, it is still rare to use cast iron as the shaft material in electric motors.

Aluminum is a versatile and corrosion-resistant material. Advanced high-tensile aluminum alloys such as scandium- and titanium-enhanced aluminum alloys have been used to replace steel today in making shafts, vehicle body, and other important components in the automotive industry. Their advantages are ease of fabrication, high strength-to-density ratio, and resistance to the corrosive atmospheres. High-performance race cars utilize extensively lightweight aluminum parts as engineers strive to improve speeds and reduce corrosion under high-temperature and high-stress conditions. For example, high-grade 7075-T6 aluminum alloy may be acceptable as a shaft material in some applications due to its high strength, low weight, and good fatigue strength properties [3.2]. In the motor industry, the use of aluminum shafts is limited to small-size and low-torque motors.

The properties of some typical shaft materials are listed in Table 3.1. The machinability rating is an indicator of how easy or how difficult a material can be mechanically processed based on the reference material AISI 1212 steel as 100% machinability.

TABLE 3.1

Properties of Some Shaft Materials

Material	Yield Tensional Strength (MPa)	Ultimate Tensional Strength (MPa)	Shear Strength (MPa)	Fatigue Strength (MPa)	Elongation (%)	Hardness Brinell (HB)	Machinability Rating (%)	Vibration Damping
AISI 1008 cold rolled	285	340	196	170	20	95	55	OK
AISI 1020 cold rolled	350	420	242	193[a]	15	121	65	OK
AISI 1045 cold rolled, annealed	505	585	338	268[a]	12	170	65	OK
304 SS	241	586	334	241[b]	55	149	45	A little better than steel
Ductile iron 65-45-12	310	448	336	179	12	131–220	160	Good
Gray cast iron 40	—	293	393	128[b]	<1	235	70	Best
A7075-T6	503	572	331	159[c]	11	150	120	Poor

[a] Based on 10^7 cycles.
[b] Based on 10^8 cycles.
[c] Based on 5×10^8 cycles.

Hesai Exhibit 1031, page 193

3.2 Shaft Loads

Motor shafts are generally subject to combined, variable loads during operation. The motor loads can be categorized into various types:

a. Because a shaft transmits torque from a motor to an externally driven machine, the shaft is subject to the torsional load. Hence, the shaft must have adequate torsional strength to withstand the load.
b. Transverse loads are most significant in various motor applications. As a shaft is subject to transverse loads, it undergoes bending or flexural deformations and possibly shearing at the joints of the shaft and the rotor core. There are several types of transverse forces acting on the shaft: (1) The gravitational force acting on the mass of the rotor assembly (the rotor core, shaft, and other components) is perpendicular to the shaft axis for a horizontally installed motor and (2) the shaft usually carries power-transmitting components such as sheaves, gears, pulleys, and chain sprockets, exerting forces at the end of the shaft in the transverse direction. (3) For a nonuniform air gap between a stator and a rotor, the unbalanced magnetic pull may develop and act on the rotor and shaft. As the shaft rotates, all these transverse forces cause reverse bending moments on the shaft. Thus, it is required to perform fatigue analysis to avoid fatigue failures.
c. A motor shaft is usually subject to low axial loading compared to its transverse loads. However, for some applications such as high-speed PCB drilling machines that experience high acceleration and deceleration in a very short time period in the motor axial direction, the axial load on the shaft can become considerably high. Furthermore, buckling analysis may be required for assessing rotor stability.
d. The coupling between a motor and a driven machine can be achieved by using flanges, splines, belts, chains, and other flexible or rigid coupling devices. For using belts or chairs, a bending moment is applied on the overhanging portion of the shaft.
e. One of the major concerns during motor design is the motor's resonant behavior. As a motor starts up, the motor shaft is subject to a shearing force. Mechanical resonance occurs as an external source amplifies the vibration level of a motor at its natural frequency. The increase in amplitude of vibration level leads to high shear stress on motor shaft and other components.
f. The preload acts on a shaft due to interference fitting of rotor core, bearing, and other mechanical components.
g. A motor may encounter shock loads at normal operation. Shock loads can be created when motors experience interrupted motions in rapid sequence, such as repeated sudden start and stop, high acceleration and deceleration, forward–reverse, and indexing motions. In addition, with high rotating speeds, electric motors act like flywheels. Typically, they represent approximately 80%–90% of the total kinetic energy in the drive systems. This flywheel effect can produce inertia shock loads far above the rated torque when the driven equipment is abruptly stopped [3.3].

If shock loads cannot be completely avoided, the motor components must be designed by taking into account an additional factor of safety (shock factors).

Earthquakes can always cause shock loads on motors. For the safe operation of the motor, this scenario must be carefully considered at the motor design stage. This is extremely important for some types of motors, such as elevator motors.

3.3 Shaft Design Methods

There are several methods available in design of motor shafts; each of them has its specific scope of application and advantages/disadvantages when compared with others.

3.3.1 Macaulay's Method

This method is relatively simple to deal with the stiffness, radius of curvature, deflection, and bending moments in a beam. The method enables discontinuous bending moment functions to be represented by a continuous function. It allows the contributions, from individual loads to the bending moment at any cross section to be expressed as a single function, which takes zero value at those sections where particular loads don't contribute to the bending moment [3.4].

As shown in Figure 3.2, when a beam is subject to a uniform load along its length, the bending moment M (in unit of N-m) is given by

$$M = \frac{EI_a}{R} \tag{3.1}$$

where

E is Young's modulus of elasticity (in unit of Pa)
I_a is the second moment of area (in unit of m^4)
R is the radius of curvature (in unit of m)

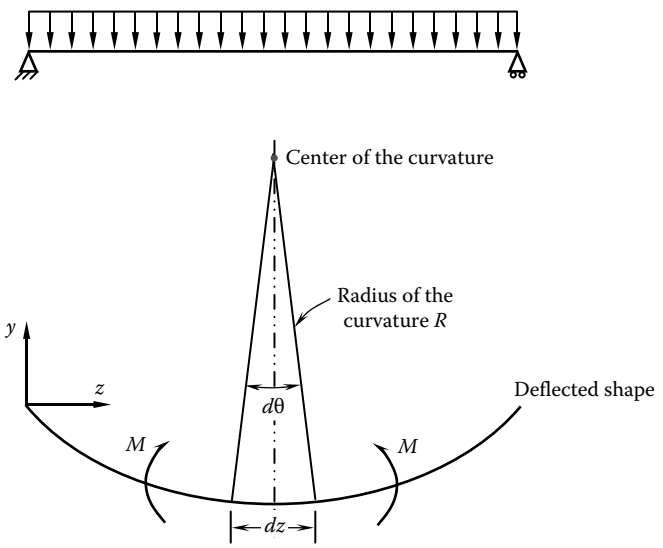


FIGURE 3.2
Beam deflection under uniformly distributed load.

In some textbooks, EI_a is defined as the beam flexural rigidity.

The curvature $1/R$ equals the reciprocal of the radius and is expressed as the second derivative with respect to the beam deflection y, that is,

$$\frac{1}{R} = \frac{d^2y}{dz^2} \tag{3.2}$$

It gives that

$$EI_a \frac{d^2y}{dz^2} = M \tag{3.3}$$

Hence, the differential equation of an elastic curve may be given as

$$\frac{d^2y}{dz^2} = \frac{M}{EI_a} \tag{3.4}$$

Macaulay's method enables discontinuous bending moment functions to be represented by a continuous function, thus avoiding the need to deal with the beam section by section between discontinuities in the bending moment function. This is very desirable since it avoids the need to evaluate, and therefore eliminate, a large number of constants of integration.

3.3.2 Area–Moment Method

The area–moment method is commonly used to determine the deflection and slope of a shaft based on two theorems: the first is used to find rotations and the second is used to find displacements. It is effective for calculating the deflections of shafts with various cross sections.

In Figure 3.3, consider a section of an elastic curve between points A and B. Both the slope and M/EI_a change along the elastic curve from A to B. Thus, integrating both sides of Macaulay's equation with respect to dz yields

$$\int_A^B \frac{M}{EI_a} dz = \int_A^B \frac{d^2y}{dz^2} dz = \left[\frac{dy}{dz}\right]_A^B = \theta_B - \theta_A \tag{3.5}$$

This indicates that the difference in slope between the points A and B on a beam is equal to the area of the M/EI_a diagram between these two points.

Multiplying both sides of Macaulay's equation by zdz and integrating from A to B, it gives that

$$\int_A^B \frac{M}{EI_a} z dz = \int_A^B \frac{d^2y}{dz^2} z dz = \int_A^B \left[\frac{d}{dz}\left(z\frac{dy}{dz}\right) - \frac{dy}{dz}\right] dz = \left[z\frac{dy}{dz}\right]_A^B - \left[y\right]_A^B$$

$$= z_B\left(\frac{dy}{dz}\right)_B - z_A\left(\frac{dy}{dz}\right)_A - y_B + y_A \tag{3.6}$$

Hesai Exhibit 1031, page 196

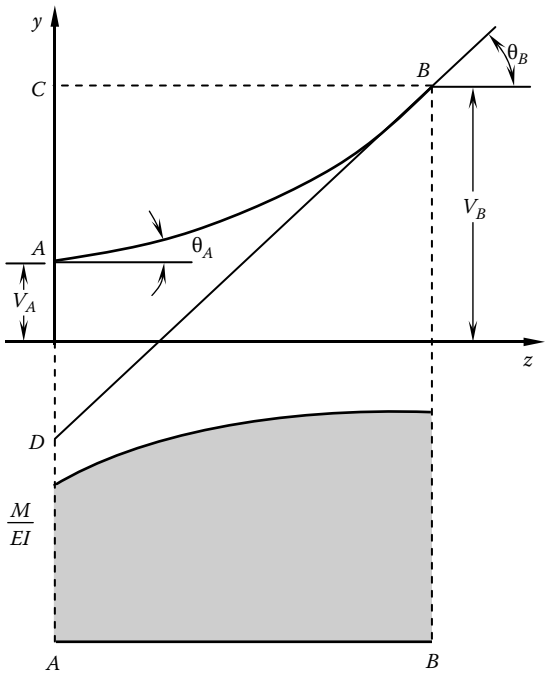


FIGURE 3.3
The changes in the slope and M/EI along the elastic curve between points A and B.

If the origin is shifted below the point A (i.e., $z_A = 0$), the second term in the previous equation vanishes, giving that

$$\int_A^B \frac{M}{EI_a} z dz = z_B \theta_B - y_B + y_A \tag{3.7}$$

where $z_B\theta_B$ is represented by the distance CD in Figure 3.3. This indicates that the moment about a point A of the M/EI_a diagram between points A and B provides the deflection of point A relative to the tangent at point B.

3.3.3 Castigliano's Method

Castigliano's method is widely used to determine displacements of linear-elastic systems based on strain energy of the systems. When such a system is subject to external loads, the deflection δ_i at any loading point in a direction of any external load Q_i can be determined by taking the partial derivative of the total strain energy U stored in the system with respect to the applied load Q_i in that direction, that is,

$$\delta_i = \frac{\partial U}{\partial Q_i} \tag{3.8}$$

If the displacement is in the same direction of the force, the displacement is positive.

The general energy and deflection equations corresponding to different load types are listed in Table 3.2, where F, M, and T are the force, bending moment, and torque acting on

TABLE 3.2

General Deflection Equations Using Castigliano's Method

Load Type	General Energy Equation	General Deflection Equation
Axial force	$U = \int_0^l \frac{F^2}{2EA}\,dx$	$\delta = \int_0^l \frac{F}{EA}\left(\frac{\partial F}{\partial Q}\right)dx$
Bending moment	$U = \int_0^l \frac{M^2}{2EI_a}\,dx$	$\delta = \int_0^l \frac{M}{EI_a}\left(\frac{\partial M}{\partial Q}\right)dx$
Torsional moment	$U = \int_0^l \frac{T^2}{2GJ}\,dx$	$\delta = \int_0^l \frac{T}{GJ}\left(\frac{\partial T}{\partial Q}\right)dx$
Transverse shear	$U = \int_0^l \frac{f_s V^2}{2GA}\,dx$	$\delta = \int_0^l \frac{f_s V}{GA}\left(\frac{\partial V}{\partial Q}\right)dx$

the shaft, respectively, E is the modulus of elasticity of the shaft material, A is the cross-sectional area of the shaft, I_a is the second moment of area, G is the shear modulus of elasticity, J is the second polar moment of area (sometimes improperly referred to as the *polar moment of inertia*), and f_s is the form factor for shear:

$$f_s(x) = \frac{A(x)}{I_a^2(x)} \int_A \frac{Q^2(x,y)}{t^2(y)}\,dA \tag{3.9}$$

The form factor is a dimensionless quantity that depends only on the shape of the cross section of shaft. In most cases, f_s is constant. For a circular shaft, $f_s = 10/9$.

3.3.4 Graphical Method

The graphical method has been primarily used in earlier time when computational techniques have not been fully developed [3.5]. With this method, design engineers graphically analyze shaft stresses and deflections. Today, with the fast development of sophisticated computational techniques, design engineers more and more concentrate on the analytical approach. However, the graphical approach to the shaft design and analysis has not lost its utility, especially in some applications where the graphical technique can provide the most efficient and quickest solution and physical insight to visualize the response of a shaft to applied loads.

3.4 Engineering Calculations

Engineering calculation is a key part in motor shaft design, which is conducted to ensure proper operation of shafts, including stress and strain, shaft deflection, critical speed, and stiffness.

For most loading types (forces, bending moments, etc.), the stress developed in a shaft is linearly related to the loading. As long as the stress in the material remains within the linear-elastic region, the stress is also linearly related to the deflection.

3.4.1 Normal Stress for Shaft Subjected to Axial Force

When a shaft of uniform diameter is subjected to an axial force F (Figure 3.4), the normal stress σ of the shaft is obtained as

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d^2} \tag{3.10}$$

where A is the cross-sectional area of the shaft.

More general, when the cross-sectional area of a shaft varies along the z direction, that is, $A = A(z)$, the normal stress becomes a function of z:

$$\sigma(z) = \frac{F}{A(z)} = \frac{4F}{\pi\left[d(z)\right]^2} \tag{3.11}$$

A common example is a tapered shaft that can be characterized by a small (or large) diameter d_1 (or d_2) and a cone angle α, as shown in Figure 3.5. Thus,

$$d(z) = d_1 + 2z\tan\alpha \tag{3.12}$$

Substituting Equation 3.12 into 3.11, it yields

$$\sigma(z) = \frac{4F}{\pi[d_1 + 2z\tan\alpha]^2} \tag{3.13}$$

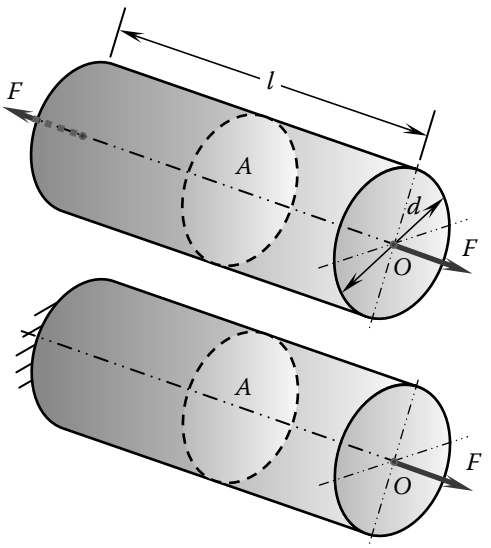


FIGURE 3.4
A single-section shaft subjected to axial force.

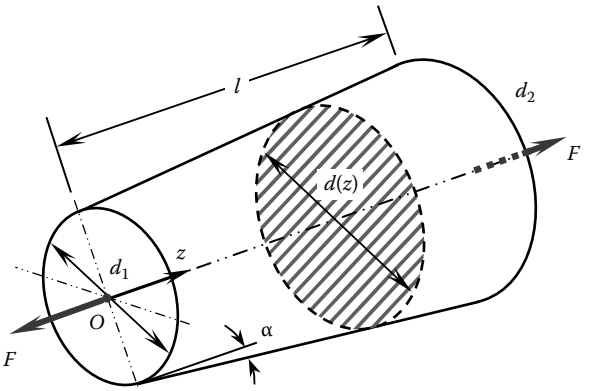


FIGURE 3.5
A tapered shaft with a cone angle α is subject to an axial force *F*.

For a stepped shaft that has an individual diameter at each segment, the normal stress at each segment *i* is given as (see Figure 3.6)

$$\sigma_i = \frac{F}{A_i} = \frac{4F}{\pi d_i^2} \tag{3.14}$$

This indicates that the maximum normal stress occurs at the segment having the smallest diameter.

3.4.2 Bending Stress for Shaft Subjected to Bending Moment

When a shaft is subjected to pure bending moment, there is bending stress generated in the shaft. As shown in Figure 3.7, compressive stresses are developed for $y > 0$ and tensional stresses for $y < 0$. At $y = 0$, there is neither compressive nor tensile stress. Thus, $y = 0$ is defined as the neutral axis. The classic formula for determining the bending stress is

$$\sigma(y) = \frac{My}{I_a} \tag{3.15}$$

- For a solid circular shaft,

$$I_a = \frac{\pi d^4}{64} \tag{3.16a}$$

- For a hollow circular shaft,

$$I_a = \frac{\pi\left(d_o^4 - d_i^4\right)}{64} \tag{3.16b}$$

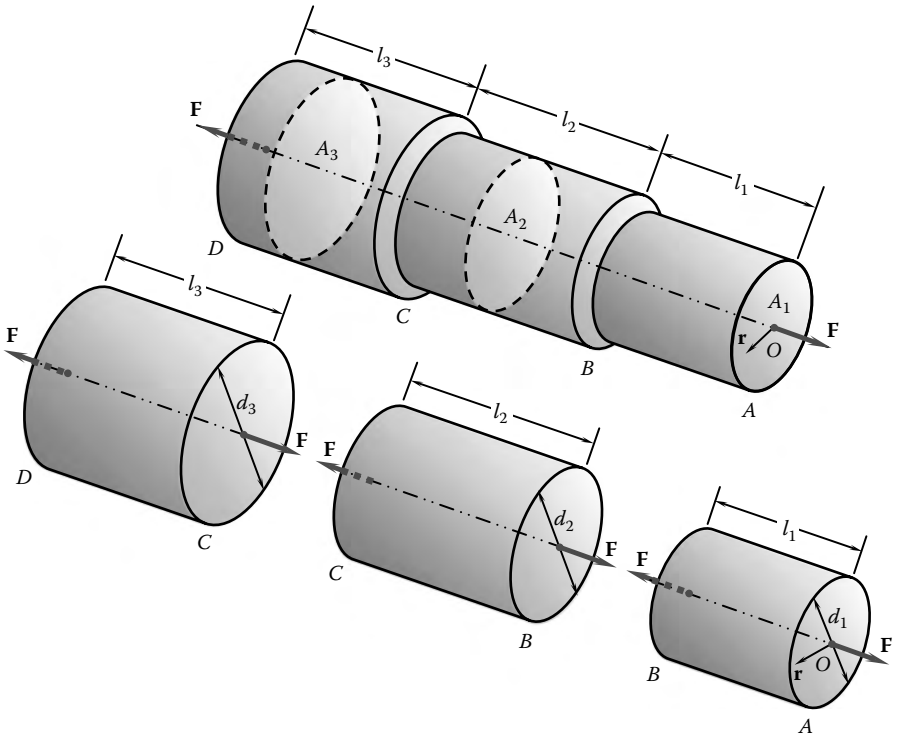


FIGURE 3.6
A stepped shaft subjected to an axial force.

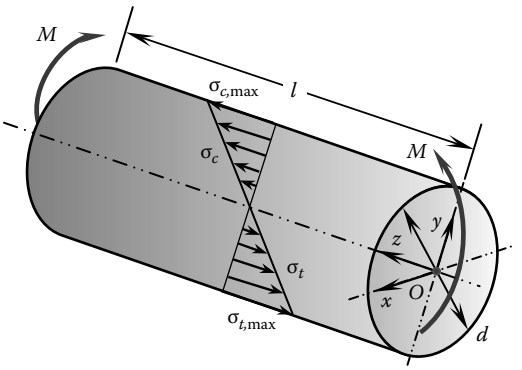


FIGURE 3.7
A single-section shaft subjected to pure bending moment.

The maximum compressive and tensional stresses occur at $y = d/2$ and $-y = d/2$, respectively:

- For a solid circular shaft,

$$\sigma_{c,\max} = \sigma_{t,\max} = \frac{32M}{\pi d^3} \tag{3.17a}$$

Hesai Exhibit 1031, page 201

- For a hollow circular shaft,

$$\sigma_{c,\max} = \sigma_{t,\max} = \frac{32Md_o}{\pi\left(d_o^4 - d_i^4\right)} \tag{3.17b}$$

For a stepped shaft subjected to multiple bending moments (Figure 3.8), the bending stress at each segment is given as

$$\sigma_i\left(y\right) = \frac{My}{I_{a,i}} \tag{3.18}$$

- For a solid circular shaft,

$$I_{a,i} = \frac{\pi d_i^4}{64} \tag{3.19a}$$

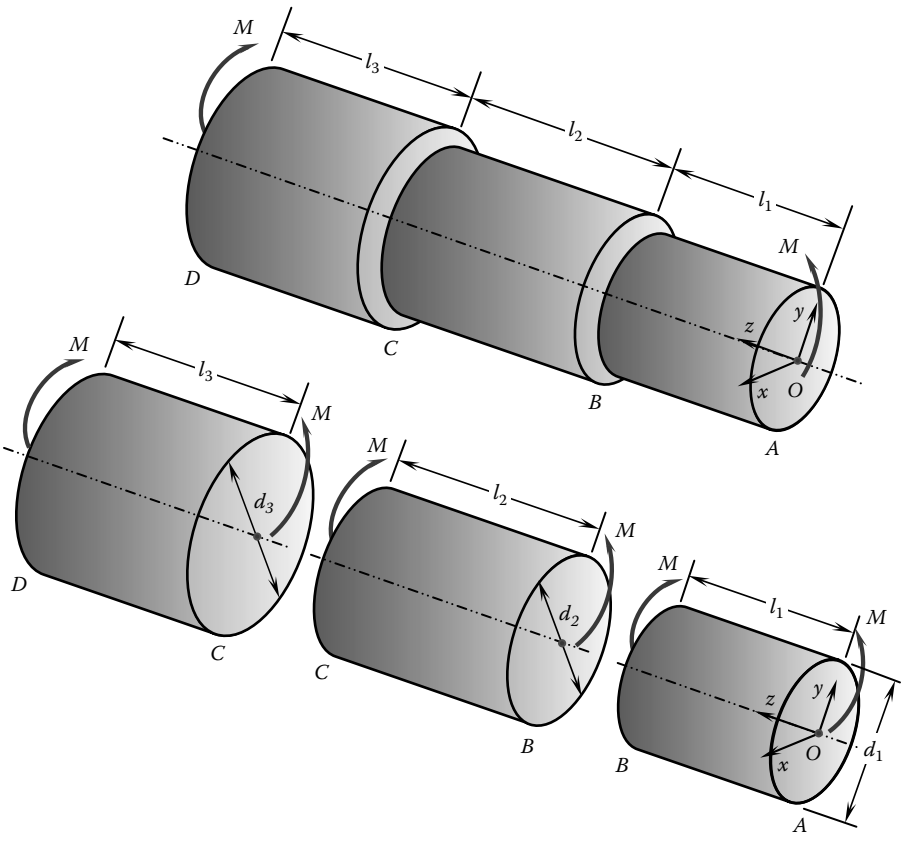


FIGURE 3.8
A stepped shaft subjected to pure bending moment.

- For a hollow circular shaft,

$$I_{a,i} = \frac{\pi\left(d_{o,i}^4 - d_{i,i}^4\right)}{64} \tag{3.19b}$$

3.4.3 Torsional Shear Stress and Torsional Deflection

Figure 3.9 shows that a pair of twist forces F is applied on a single-section shaft with a uniform diameter. This is equivalent to the case that the shaft is fixed at one end and a torque T is applied at the other end. In such a case, the radial line at the shaft end surface is rotated through an angle ϕ, defined as the shaft torsional deflection (or angular deflection in some textbooks). The length of the arc produced in the case is $R\phi$. Correspondingly, the straight line parallel to the shaft axis is rotated through an angle γ, defined as the shaft shear strain.

The rate of twist $d\phi/dz$ can be derived as

$$\frac{d\phi}{dz} = \frac{T}{GJ} \tag{3.20}$$

where

G is the shear modulus of elasticity (in unit of Pa)

J is the second polar moment of area (in unit of m^4)

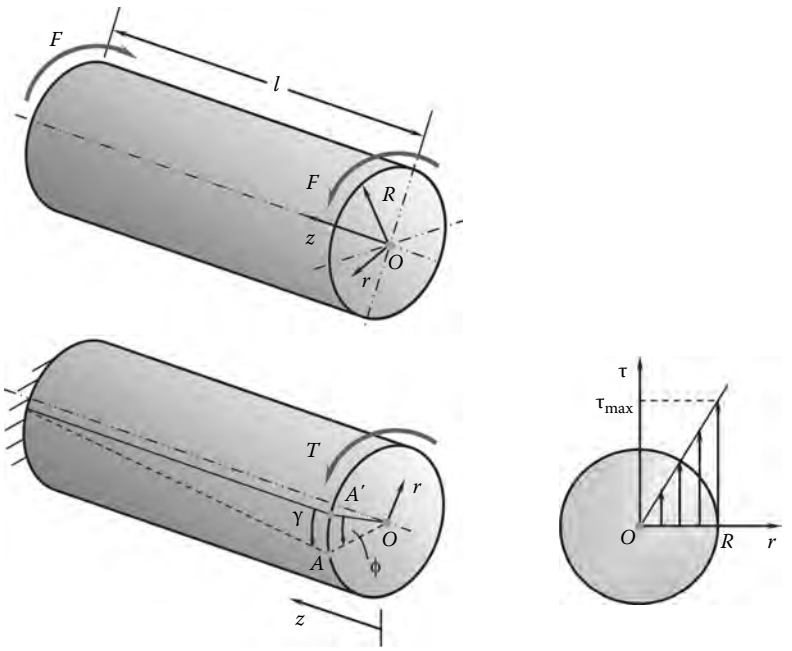


FIGURE 3.9
Torsional shear stress and torsional deflection of a single-section shaft subjected to torsional bending moment $M = F \times R$ or torque T.

It can be seen from this equation that the higher the value of *GJ*, the lower the rate of twist $d\phi/dz$ for a given torque *T*. This means that the rigidity of the shaft increases with the increase of *GJ*. The quantity *GJ* is thus defined as torsional rigidity.

For a solid circular shaft,

$$J = \frac{\pi d^4}{32} \tag{3.21}$$

For a hollow shaft with an OD and ID, d_o and d_i, respectively, the second polar moment of area *J* is

$$J = \frac{\pi\left(d_o^4 - d_i^4\right)}{32} \tag{3.22}$$

Integrating Equation 3.20 for obtaining torsional deflection of the shaft ϕ (in radians) yields

$$\phi = \frac{Tl}{GJ} \tag{3.23}$$

This equation shows that torsional deflection in a shaft is inversely proportional to the diameter of the fourth power. Taking a solid shaft as an example, a 10% increase in shaft diameter can result in a 31.7% reduction in torsional deflection ($1/1.1^4 - 1 = -31.7\%$). However, for practical and economical reasons, shaft size is often minimized to just satisfy the basic requirements.

Torsional stiffness of a shaft is a measure of the resistance offered by the elastic shaft to torsional deformation. This design parameter is very important in positional systems. In practice, shaft torsional stiffness is specified as a torque per unit torsional deflection, that is,

$$S_t = \frac{T}{\phi} = \frac{GJ}{l} \tag{3.24}$$

Torsional stiffness may be expressed in several different units, but the most common and easiest to work with is N-m/rad.

Another design parameter closely related to torsional stiffness is torsional flexibility, which is defined as the reciprocal of the torsional stiffness.

The torsional shear stress τ at a certain radius *r* of shaft is expressed as

$$\tau = \frac{Tr}{J} \tag{3.25}$$

This reveals that the torsional shear stress is independent of the choice of shaft material.

It can be seen from Figure 3.9 that the torsional shear stress is a linear function of shaft radius *r*; it vanishes at the shaft center (where $r = 0$) and reaches the maximum at the outer surface of the shaft (where $r = R = d/2$):

$$\tau_{\max} = \frac{TR}{J} = \frac{Td}{2J} \tag{3.26}$$

- For solid shaft,

$$\tau_{\max} = \frac{16T}{\pi d^3} \tag{3.27a}$$

- For hollow shaft,

$$\tau_{\max} = \frac{16Td_o}{\pi\left(d_o^4 - d_i^4\right)} \tag{3.27b}$$

This indicates that the shaft core has little contribution to the shaft torsional strength, and thus, the shaft can be designed as a hollow shaft minimal influence on its strength while significantly reducing weight.

Comparing Equations 3.25 and 3.26 gives

$$\tau = \frac{r}{R}\tau_{\max} \tag{3.28}$$

The shaft shear strain γ, measured as the twist angle, can be expressed as the product of the radial coordinate r and the rate of twist $d\phi/dz$:

$$\gamma = r\frac{d\phi}{dz} \tag{3.29}$$

Since $d\phi/dz$ is a constant, the shaft shear strain γ is a linear function of the shaft radius r and reaches its maximum value $\gamma_{\max}$ at the outer surface (where $r = R$):

$$\gamma = \frac{r}{R}\gamma_{\max} \tag{3.30}$$

From Hook's law, the shear stress τ can be expressed as

$$\tau = G\gamma = Gr\frac{d\phi}{dz} \tag{3.31}$$

Substituting Equation 3.20 into 3.31, it becomes

$$\tau = \frac{Tr}{J} \tag{3.32}$$

This is identical to Equation 3.25.

For a stepped shaft having a different diameter and length in each segment, when the shaft is subjected to a single torque (Figure 3.10), the torsional deflection at each shaft segment becomes

$$\phi_i = \frac{Tl_i}{GJ_i} \tag{3.33}$$

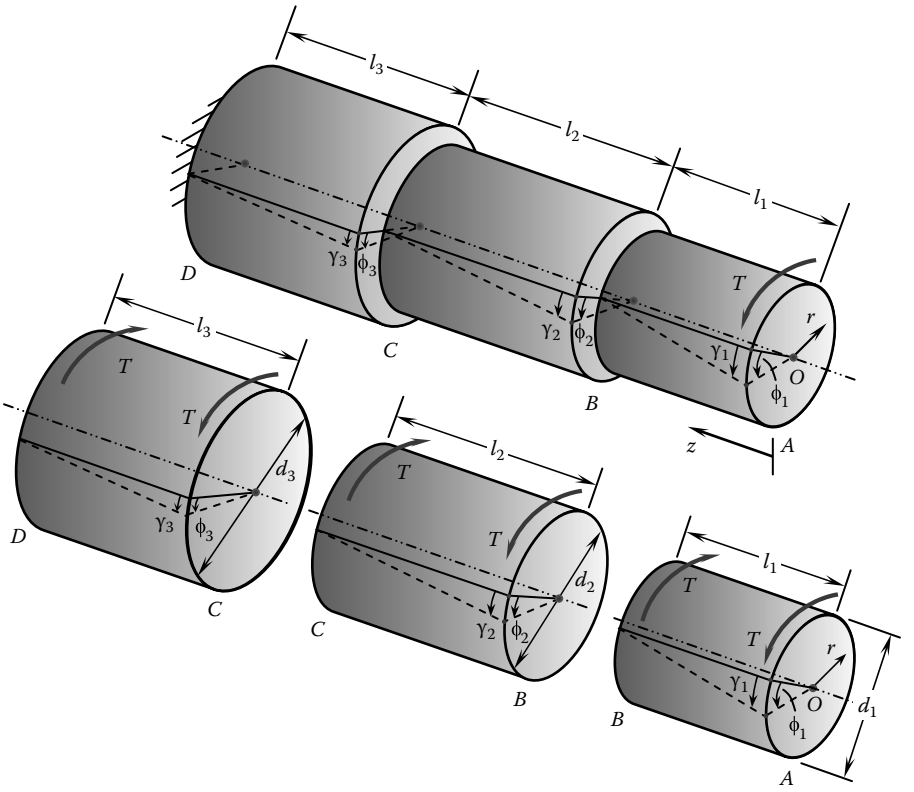


FIGURE 3.10
Torsional deflection of a stepped shaft subjected to a single torque and made from of single material.

It is noted that all torsional deflections ϕ_i have the same sign. The total torsional deflection is

$$\phi = \sum_i \phi_i = \frac{T}{G} \sum_i \frac{l_i}{J_i} \tag{3.34}$$

Similarly, the torsional shear stress on each shaft segment is

$$\tau_i = \frac{T r_i}{J_i} \tag{3.35}$$

For a stepped shaft with individual diameter d_i, length l_i, torque T_i, and material shear modulus G_i at each segment (Figure 3.11), the total torsional deflection becomes [3.6]

$$\phi = \sum_i \phi_i = \sum_i \frac{T_i l_i}{G_i J_i} \tag{3.36}$$

For this case, the sign of each torsional deflection ϕ_i depends on the torque applied on the segment.

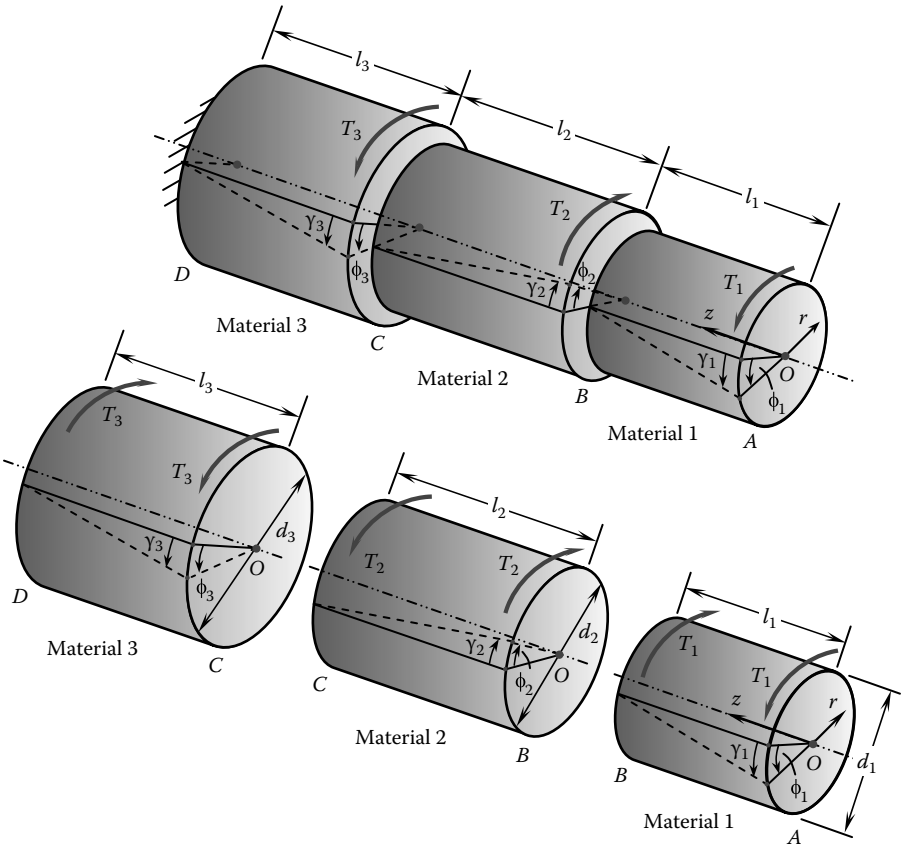


FIGURE 3.11
Torsional deflection of a stepped shaft having an individual diameter, length, torque, and material at each segment.

Correspondingly, shaft torsional stiffness at each segment is expressed as

$$S_{t,i} = \frac{T_i}{\phi_i} = \frac{G_i J_i}{l_i} \tag{3.37}$$

The total shaft torsional stiffness is given as

$$S_t = \left(\sum_i \frac{1}{S_{t,i}} \right)^{-1} \tag{3.38}$$

3.4.4 Lateral Deflection of Shaft

One of the critical design parameters of motor shaft is the maximum allowable deflection. If a motor shaft experiences a large lateral deflection, the rotating rotor may possibly contact the stator bore, causing a severe damage of the motor. Thus, the shafts must be rigid enough to avoid excessive deflection.

3.4.4.1 Lateral Deflection due to Bending Moment

When a shaft is subjected to a bending moment, the longitudinal centroidal axis of the shaft becomes a curve defined as elastic curve, as shown in Figure 3.12 with an exaggerated shape. When the bending moment is a function of the position along the shaft, that is, $M = M(z)$, the lateral deflection y of the shaft that has a uniform diameter can be derived from the following equation [3.7]:

$$\frac{d^2y}{dz^2} = \frac{M(z)}{EI_a} \tag{3.39}$$

Integrating the previous equation with respect to z, the slope of the shaft (in radians) is obtained as

$$\theta = \frac{dy}{dz} = \frac{1}{EI_a}\int M(z)dz \tag{3.40}$$

Continuously integrating the previous equation with respect to z, the lateral deflection of the shaft is found as

$$y = \int \theta dz = \frac{1}{EI_a}\iint M(z)dzdz \tag{3.41}$$

In an electric motor, the shaft diameter is often nonuniform. For a stepped shaft having an individual diameter at each segment, $I_{a,i} = I_{a,i}(z)$, the lateral deflection y and the slope of the shaft θ can be determined, respectively:

$$y_t = \sum_i y_i = \sum_i \int \frac{d\theta}{dz} = \sum_i \frac{1}{E}\iint \frac{M_i(z)}{I_{a,i}(z)}\,dzdz \tag{3.42}$$

$$\theta_t = \sum_i \theta_i = \sum_i \frac{dy_i}{dz} = \sum_i \frac{1}{E}\int \frac{M_i(z)}{I_{a,i}(z)}\,dz \tag{3.43}$$

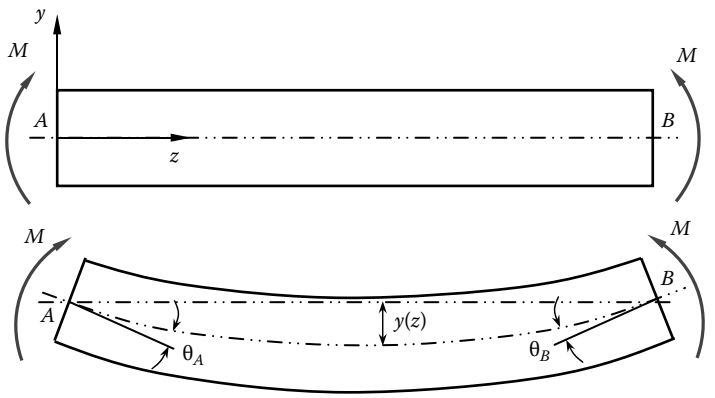


FIGURE 3.12
Deflection of a single-section shaft subjected to bending moment.

The previous equations can be used in a stepwise fashion to solve the deflection problem for stepped shafts.

For the simple case as shown in Figure 3.12, the deflection is given as

$$y(z) = \frac{Mz}{6EI_a l}\left(z^2 + 3a^2 - 6al + 2l^2\right) \tag{3.44}$$

$$\theta_A = \frac{M}{6EI_a l}\left(3a^2 - 6al + 2l^2\right) \tag{3.45}$$

3.4.4.2 Lateral Deflection due to Transverse Force

As shown in Figure 3.13, a shaft is simply supported at the two ends and loaded by a transverse force *F*. At the equilibrium condition, all forces and bending moments acting on the shaft must be balanced:

$$\sum F = 0 \Rightarrow R_A + R_B - F = 0 \tag{3.46}$$

$$\sum M = 0 \Rightarrow R_B l - Fa = 0 \tag{3.47}$$

This gives

$$\begin{cases} R_A = \dfrac{F}{l}(l-a) \\ R_B = \dfrac{Fa}{l} \end{cases} \tag{3.48}$$

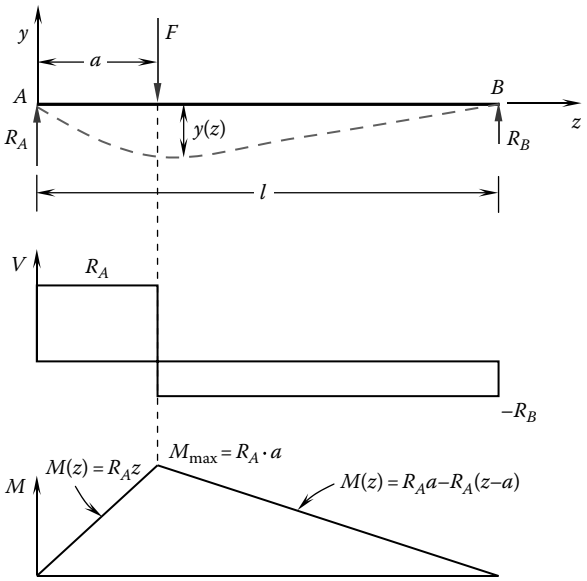


FIGURE 3.13
Deflection of a shaft subjected to a transverse force.

and

$$\begin{cases} M(z) = R_A z = \dfrac{Fz}{l}(l-a) & 0 \le z \le a \\ M(z) = R_A a - R_B(z-a) = \dfrac{Fa}{l}(l-z) & a \le z \le l \end{cases} \tag{3.49}$$

The deflection of the shaft is

$$\begin{cases} y(z) = \dfrac{F(l-a)z}{6EI_a l}\left[z^2 + (l-a)^2 - l^2\right] & 0 \le z \le a \\ y(z) = \dfrac{Fa(l-z)}{6EI_a l}\left(z^2 + a^2 - 2lz\right) & a \le z \le l \end{cases} \tag{3.50}$$

$$\begin{cases} \theta_A = \left(\dfrac{dy}{dz}\right)_A = -\dfrac{Fa}{6EI_a l}\left(2l^2 - 3al + a^2\right) \\ \theta_B = \left(\dfrac{dy}{dz}\right)_B = -\dfrac{Fa}{6EI_a l}(l^2 - a^2) \end{cases} \tag{3.51}$$

For this case, the maximum deflection occurs at $z = a$:

$$y_{\max} = -\frac{Fa^2(l-a)^2}{3EI_a l} \tag{3.52}$$

In Figure 3.14, a shaft with uniform diameter is subjected to multitransverse forces. In such a case, each deflection $y_i(z)$, which resulted from each of the transverse force F_i, is obtained as

$$\begin{cases} y_i(z) = \dfrac{F_i(l-a_i)z}{6EI_a l}\left[z^2 + (l-a_i)^2 - l^2\right] & 0 \le z \le a_i \\ y_i(z) = \dfrac{Fa_i(l-z)}{6EI_a l}\left(z^2 + a_i^2 - 2lz\right) & a_i \le z \le l \end{cases} \tag{3.53}$$

$$y_t(z) = \sum y_i(z) \tag{3.54}$$

Therefore, the total deflection is obtained by superimposing each of the deflection $y_i(z)$:

$$\begin{cases} \theta_A = -\sum \dfrac{F_i a_i}{6EI_a l}\left[2l^2 - 3a_i l + a_i^2\right] \\ \theta_B = -\sum \dfrac{F_i a_i}{6EI_a l}\left(l^2 - a_i^2\right) \end{cases} \tag{3.55}$$

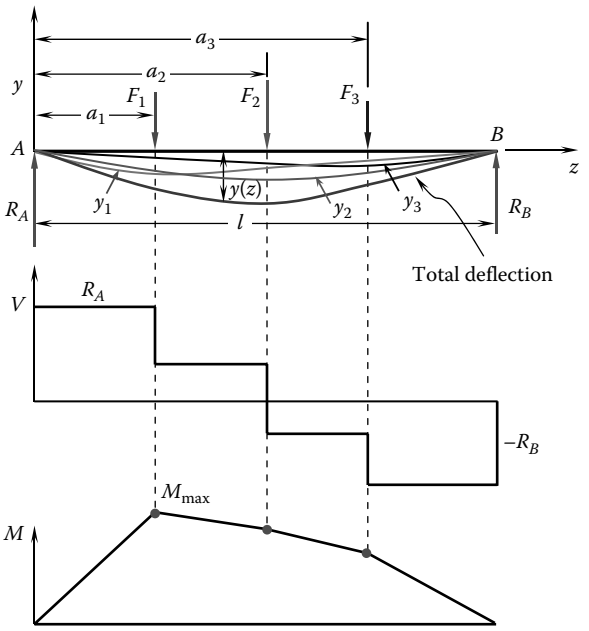


FIGURE 3.14
Deflection of a shaft subjected to multitransverse forces. The total deflection $y(z)$ is the superposition of each deflection $y_i(z)$, corresponding to each transverse force.

3.4.4.3 Lateral Deflection due to Shear Force

A shear condition is considered when a shaft is subject to a pair of equal, opposite, and parallel forces that are so close so that the bending moment is negligible. Typically, deflections due to direct shear is very small (<5%) compared to deflections due to bending moments. However, for short and heavily loaded shafts, this deflection may become significant.

As shown in Figure 3.15, two equal and opposite forces act on a shaft apart by Δz. Because Δz is small enough, the bending effect on the shaft can be ignored. The deflection due to each force is easily obtained. Thus, the total shaft deflection $y(F_1+F_2)$ can be determined using the superposition method.

3.5 Shaft Design Issues

3.5.1 Shaft Design Considerations

The design of highly reliable shafts involves many aspects of mechanical design; some of them are listed in the following:

- The determination of shaft diameter is greatly affected by many design factors such as shaft loads, transmitted torque, support span, bearings, mating methods, stress concentrations, and motor cooling. Many of these factors have been discussed previously and integrated into the derivation of the equations for calculating the shaft diameter.

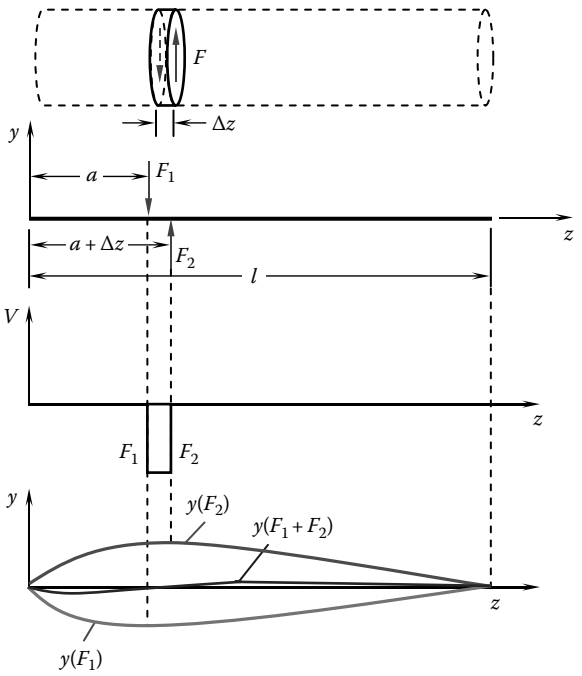


FIGURE 3.15
Deflection of a shaft subjected to shear load.

- In order to reduce stress concentration, the radius of the fillet between two adjacent steps must be larger than $0.2d$. Sharp corners are absolutely not allowed.
- To lower bending stress and shaft deflection, the shaft length must be kept as short as possible. So does the shaft overhang length.
- Factor of safety (FS) in shaft design can vary in a large range, mostly depending on applications. For some specific applications such as elevate motors, FSs are often mandated by applicable international, national, or industrial standards, policies, or law.
- The shaft shoulder in contact with the face of a bearing must keep perpendicular to the shaft centerline. To avoid interference, the shaft fillet radius must be smaller than the bearing radius. The shoulder height is an important parameter to provide enough support to bearing loads. This is especially true for tapered roller bearings subject to high axial loads. Sometimes the shoulder height is roughly assumed to be 50%–80% of the thickness of the inner race of the bearing that directly contacts with the shaft. For more accurate design, shaft shoulders are determined using bearing catalogs for the specific bearing type and different load conditions. When the shoulder at a bearing becomes critical, it is necessary to check the stress and stress concentration on the shaft shoulder.
- Although there are many methods of fixing bearings in position on shafts, the most common way is to use retaining (snap) rings, which fit into grooves of shafts and take axial loads. However, the grooves can cause stress concentrations in the shafts.

Hesai Exhibit 1031, page 212

3.5.2 Shaft Rigidity

Shaft design involves two types of shaft rigidity:

- Torsional rigidity: Torsional rigidity is a measure of a shaft's ability to resist twisting loads. It is expressed as the product of the shear modulus of elasticity G (in unit of Pa) and the second polar moment of area J (in unit of m^4). Thus, the torsion angle (in rad) of the shaft that is subject to a torque T is determined:

$$\theta = \frac{Tl}{GJ} \tag{3.56}$$

  where
  T is the torque or torsional moment
  l is the shaft length subjected to twist moment

  This equation indicates that the torsion angle θ decreases with the increased shaft torsional rigidity GJ. In the unit of degree, the torsion angle can be expressed as

$$\theta = \frac{180}{\pi}\frac{Tl}{GJ} \tag{3.57}$$

  The permissible angle of twist changes case to case. For instance, for machine tool applications, it is 0.25° per meter length of shaft.
- Lateral rigidity: Lateral rigidity is a measure of a shaft's ability to resist forces or bending moments, defined as the product of modulus of elasticity E and second moment of area I. Generally, the higher the lateral rigidity EI, the lower the shaft lateral deflection y. In certain applications, the shaft is designed on the basis of the lateral rigidity rather than on strength. Usually, the maximum permissible deflection for the transmission shaft is taken as $0.001l$–$0.003l$, where l is the span between two bearings. For electric motors, the gap depth must be taken into account in the determination of the maximum shaft deflection. Generally, the shaft lateral deflection depends on shaft dimensions, load condition, material properties, bearing span, and supporting patterns.

3.5.3 Critical Shaft Speed

A rotating shaft deflects during rotation due to its own weight, external loads, imperfect geometry, and uneven mass distribution. This deflection can significantly impact the motor performance. It has been found that there are a number of specific rotational speeds, at which the vibration amplitude increases dramatically. As the rotational speed passes beyond these speeds, the vibration amplitude decreases significantly. These speeds are defined as *critical speeds*.

The critical speed of a shaft generally depends on the magnitudes and locations of mass/load carried by the shaft, shaft support span, stiffness of the shaft, moment of inertia of the shaft, properties of shaft material (e.g., density, modulus of elasticity), shaft varying diameter, type of end supports, bearing type, and system configuration. Typical shaft design requires the critical speeds to be much higher than the anticipated operating speeds of motors, usually 20%–25% of the operating speed.

As discussed in Chapter 2, two methods are commonly used to calculate shaft critical speed. Rayleigh–Ritz method is an approximate method of finding displacements that is based on the theorem of minimum potential energy. However, the Rayleigh–Ritz method overestimates the critical speed by a few percent [3.8]. Another method is called the Dunkerley method, which underestimates the critical speed.

3.5.3.1 Shaft with Uniform Diameter

For a shaft of uniform diameter, the calculation of critical speed is relatively straightforward.

- For a shaft itself, the critical speed is

$$n_c = \left(\frac{\pi}{l}\right)^2 \sqrt{\frac{gEI_a}{A\gamma}} \tag{3.58}$$

- For the shaft with one element (e.g., rotor core) attached to it, the critical speed is

$$n_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{st}}} \tag{3.59}$$

where
n_c is the critical speed in rpm
g is the acceleration of gravity
δ_{st} is the static deflection

3.5.3.2 Stepped Shaft

For a stepped shaft that has multiple elements, the critical speed can be calculated by the Rayleigh method after calculating the static deflection at each and every change of cross section [3.9]:

$$n_c = \sqrt{\frac{g\sum_i^m F_i y_i}{\sum_i^m F_i y_i^2}} \tag{3.60}$$

3.5.4 Dimensional Tolerance

The key tolerances of a shaft can strongly influence motor normal operation and reliability. Motor design engineers should pay special attention to the shaft dimensional tolerances at these locations:

- Bearing journals. It usually requires the press fits for assembling shafts with bearings.
- Rotor core. To ensure a rotor to withstand various loads, the interference fit (e.g., press or shrink fit) adopted in the rotor–shaft assembly.
- Coupling attached to the shaft. Improperly installed couplings may initiate additional forces acting on shafts.

- Shaft seal. Because a seal may contact with the shaft, it generates a frictional force on the shaft. To help prevent scratches and nicks, the shaft must maintain certain levels of hardness (usually HRc > 45, more desirable hardness is at the range of 60–65 HRc), smoothness, and roundness.
- Keyseats, retaining ring grooves. These may also use the press fit method.

3.5.5 Dynamic Shaft Runout

Dynamic shaft runout refers to the phenomenon that during operation, a shaft does not rotate around the true center. A number of factors may cause shaft runoff such as shaft deflection, vibration, bearing misalignment, shaft imbalance, machining tolerance, and other inaccuracies. Shaft runouts can deteriorate motor operation, produce additional stresses inside the shaft, and enlarge sealing gap, which leads to a certain level of leakage.

Dynamic shaft runout data can be captured using a dial indicator when a shaft rotates slowly. Runout measurements are usually taken at a number of positions along the shaft. Alternatively, measurements can be performed more efficiently by using a coordinate measuring machine (CMM).

3.5.6 Shaft Eccentricity

Shaft eccentricity can be resulted from the shaft deformation due to unbalance forces on the shaft, the shaft-to-bore misalignment due to machining and assembly inaccuracies, as well as the bearing radial clearance, and dynamic runout from the shaft itself. Among them, the shaft-to-bore misalignment is involved by both the shaft and the bore and almost always exists to a certain extent in practice. Apparently, shaft eccentricity must be minimized to prevent vibration, noise, leakage, and possible motor failure, especially for high-speed motors.

Eccentricity is a measure of shaft bow that is caused by any or a combination of mechanical, thermal, and gravity bow. This bow is measured while a motor runs at slow speed for detecting the peak-to-peak motion of the shaft. The measured data are critical for identifying the primary root cause of shaft eccentricity. There are a number of ways to reduce shaft eccentricity, including the following: (1) Both the shaft and its mating bore must be machined within the tolerance. (2) The shaft runout is controlled within the tolerance. (3) To reduce the gravity-induced shaft deformation, shafts that have been completely machined should be placed vertically. (4) All fits need to be carefully inspected, including bearing to bearing bore, shaft to bearing, and shaft to rotor core. (5) Rotor unbalance must be minimized. In addition, unbalance inspection includes all mechanical components mounted on the shaft, such as couplings and pulleys. (6) Bearings need to be inspected to ensure their performance. (7) Since one of the root causes of shaft eccentricity is overloading, sometimes reducing load can mitigate the extent of shaft eccentricity. (8) The selection of appropriate couplings is an important part impacting motor's overall performance. The adoption of flexible couplings helps reduce shaft misalignment and eccentricity.

Because shaft eccentricity is so critical for motor performance, it is highly desired to proactively solve it in the early stage of design process.

3.5.7 Heat Treatment and Shaft Hardness

The purpose of heat treatment on motor shafts is to improve shaft mechanical properties as a result of the change in material microstructures. Medium- and high-carbon steel

is heat-treated to increase strengths, hardness, wear resistance, and fatigue life. For low-carbon steel, the improvement of material properties can be achieved by means of carburization technology.

Hardness is a measure of a material's ability to resist mechanical abrasion and indentation. In fact, there is a positive correlation between the material hardness and the yield tensile and ultimate tensile strengths. Shaft surface hardness has a direct influence on its load capacity, performance, lifetime, and sealing.

It has been reported that a shaft of 40 HRc hardness has only 20% of the rated life of a system with a shaft at 60 HRc. To get the most rated life, heat treating shafts to a surface hardness of 60–65 HRc is essential [3.10]. For instance, by changing a shaft hardness of 50–55 HRc to 60–65 HRc, the shaft life can increase eight times.

The ability of the material to absorb a significant deformation before fracture is referred as ductility. An indicator of the material ductility is its elongation under a tensile load. Toughness is the material's ability to sustain impact and shock loading. For a motor shaft, it is desired to maintain high hardness at the shaft surface and good ductility and toughness at its core.

There are many types of heat treatment processes available to motor shafts. The combination of inductive heating and water quenching is the most effective heat treatment process of carbon steel shafts. As illustrated in Figure 3.16, during an inducting heating process while a single inductive coil moves along the shaft axis to heat a certain length of the shaft, the motor shaft rotates slowly around its axis to ensure uniform heating all over the shaft surface and through the shaft cross sections. The surface temperature of the shaft during induction heating changes sharply from room temperature to above 1000°C. The optimal heating time for required shaft properties is found out to be a few seconds, followed by a quenching period with oil–water emulsion.

The thickness of the surface hardened layer of steel parts through the induction hardening process is determined by many factors such as frequency of the AC field, heating and quenching time, current, moving speed of the heating coil, shaft material, and shaft shape, to name a few. Microscopic photos are commonly used to demonstrate the variation of the microstructure underneath the shaft surface. It has shown that at the shaft surface is completely martensitic. At a certain depth, the microstructure is composed of martensite

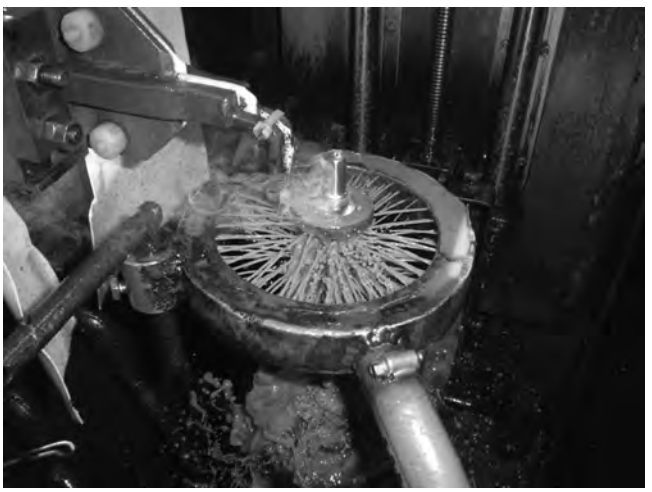


FIGURE 3.16
Inductive heating and water quenching of a motor shaft.

Hesai Exhibit 1031, page 216

and ferrite. The ferrite in the microstructure indicates that the transformation to austenite was not completed during induction heating period because the time for transformation was not long enough. Beneath the hardening depth, the microstructure is composed of pearlite and ferrite [3.11].

One of the most important factors in a shaft heat treatment is to minimize the distortions in shaft shape and size. The larger the mass of the metal heated, the greater the metal's thermal expansion and, thus, distortion.

3.5.8 Shaft Surface Finishing

Shaft surface finishing is important on the shaft fatigue life and seal performance, as well as the reduction of windage losses. Shaft surface finishing contains two features: surface texture and shaft lead. Surface texture is characterized by three parameters: average roughness R_a, average peak-to-valley height R_z, and average peak-to-mean height R_{pm}, as shown in Figure 3.17.

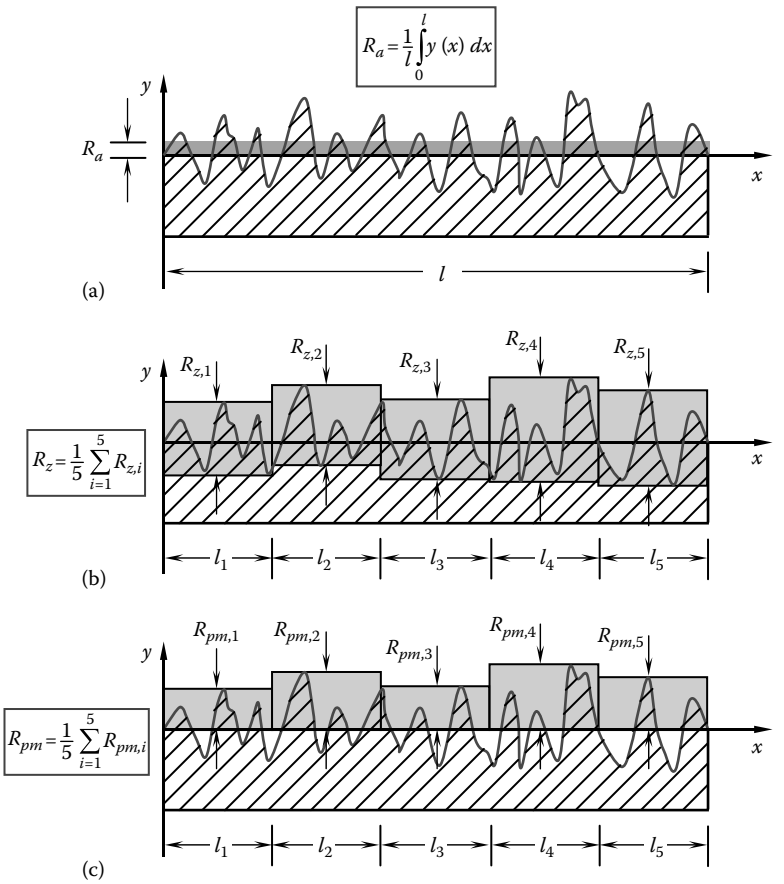


FIGURE 3.17
Three surface texture characteristic parameters: (a) average roughness R_a, (b) average peak-to-valley height R_z, and (c) average peak-to-mean R_{pm}.

R_a is the average value of the deviations from the mean line over the evaluation length l and can be determined as

$$R_a = \frac{1}{l}\int_0^l |y(x)|\,dx \tag{3.61}$$

In general, R_a less than 0.5 μm and a shaft lead angle less than 0.05° can achieve the optimum seal performance.

Steel manufacturing process has a strong impact on surface finish. For example, cold-rolled stainless steel has a roughness of $R_a = 0.2–0.5$ μm and therefore usually does not need to be polished to meet surface roughness requirements. The surface treatment of stainless steel and the resultant surface topography are listed in Table 3.3 [3.12].

Similarly, the rms roughness R_q is defined as follows:

$$R_q = \sqrt{\frac{1}{l}\int_0^l y^2(x)\,dx} \tag{3.62}$$

TABLE 3.3

Surface Treatments of Stainless Steel and Resultant Surface Topography

Surface Treatment	Approximate R_a Values (μm)	Typical Features of the Technique
Hot rolling	>4	Unbroken surface
Cold rolling	0.2–0.5	Smooth unbroken surface
Glass bead blasting	<1.2	Surface rupturing
Ceramic blasting	<1.2	Surface rupturing
Micropeening	<1	Deformed (peened) surface irregularities
Descaling	0.6–1.3	Crevices depending on initial surface
Pickling	0.5–1.0	High peaks, deep valleys
Electropolishing		Roundoff peaks without necessarily improving R_a
Mechanical polishing with aluminum oxide or silicon carbide		
Abrasive grit number		Surface topography highly dependent on process parameters, such as belt speed and pressure
500	0.1–0.25	
320	0.15–0.4	
240	0.2–0.5	
180	0.6	
120	1.1	
60	3.5	

R_z is the average peak-to-valley height and can be obtained by averaging $R_{z,i}$ values from a number of consecutive lengths along the contact surface, where

$$R_z = \frac{1}{N}\sum_{i=1}^{N} R_{z,i} \tag{3.63}$$

where N is the number of sections.

Besides R_a and R_z, another important parameter is the average peak to mean, defined as R_{pm}:

$$R_{pm} = \frac{1}{N}\sum_{i=1}^{N} R_{pm,i} \tag{3.64}$$

By cutting surface roughness with the depth of C ($C = R_z/2$), the area can be distinguished as the contact areas and noncontact areas. The surface contact area ratio R_c is defined as the summation of all surface contact areas to the total surface area (Figure 3.18), that is,

$$R_c = \frac{1}{l}\sum_{i=1}^{N} x_i \tag{3.65}$$

3.5.9 Shaft Lead

Fluid leakage in electric motors may be encountered when defective shafts that were not properly manufactured are used. A number of factors can affect the shaft sealing performance such as shaft surface finish, dimensional accuracy, and operating conditions. During a shaft finishing process, spiral grooves may be generated on the shaft surface. As shown in Figure 3.19, while the shaft rotates counterclockwise at an angular velocity **ω**, a finishing tool (a lathe cutter or a grinding wheel) moves axially along the shaft toward the lathe chuck at a feeding speed **u**, forming the shaft lead angle α

$$\alpha = \tan^{-1}\left(\frac{u}{r\omega}\right) \tag{3.66}$$

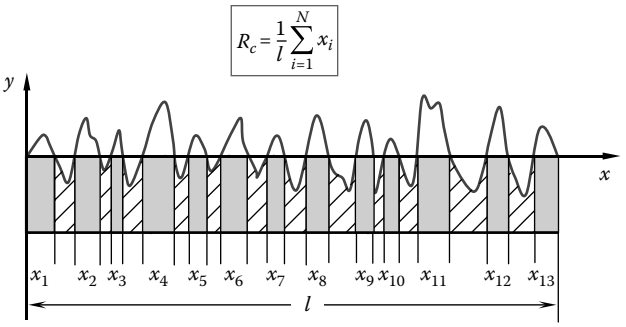


FIGURE 3.18
Surface contact area A_c, which is defined as the ratio of the summation of each contact area to the total area.

Hesai Exhibit 1031, page 219

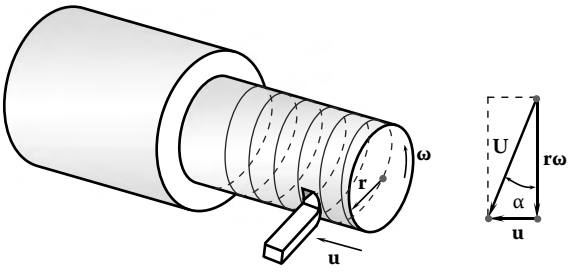


FIGURE 3.19
Formation of spiral grooves on shaft surface during shaft finishing machining.

and the magnitude of the resultant speed vector **U** is

$$U = \sqrt{u^2 + (r\omega)^2} \tag{3.67}$$

Figure 3.19 indicates that the lead angle α is proportional to the feeding speed u but inversely proportional to the angular velocity ω and the shaft radius r. Regardless of shaft finishing conditions, the shaft lead angle α is generally recommended to be controlled less than 0.05°. With a large feed speed or a slow shaft rotating speed, or both, spiral grooves are formed on the shaft surface. This phenomenon, known as shaft lead, can result in shaft seal failure during motor normal operation.

A quick and simple test can be adopted by using the suspended weight method for establishing the presence of shaft lead. As illustrated in Figure 3.20, a shaft is mounted on a lathe chuck and a small weight (e.g., 1 oz) is suspended on a testing shaft. As the lathe rotates slowly either counterclockwise or clockwise, observe whether the weight moves along the shaft and which direction it goes. The ideal case is that the weight remains in the same location with the rotating shaft, indicating there is no leakage problem.

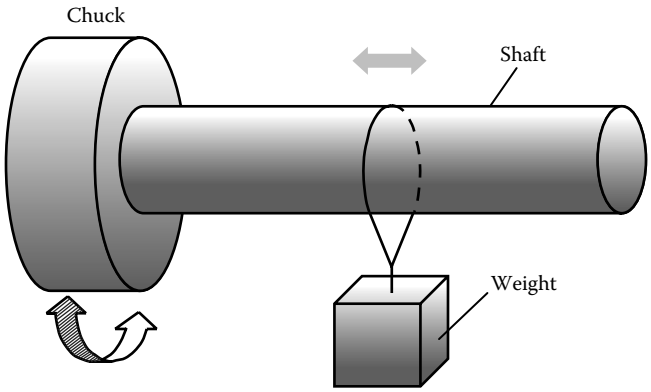


FIGURE 3.20
Motor shaft string testing.

3.5.10 Shaft Seal

Sealing is one of the main concerns in the design of rotating machinery such as motors, engines, gas turbines, generators, pumps, compressors, and other variety of rotational machines. A sealing system in a rotating machine is used to control (1) fluid (liquid or gas) leakage, (2) dynamic performance, (3) tolerance to boundaries, (4) coolant or lubricant flow, and (5) windage losses. All these functions can directly affect the performance of rotating machines through parasitic losses, life, or limit cycles [3.13]. In a worse case, a failure of the sealing system may lead to the whole system failure or disaster (e.g., exploration in a hydrogen-cooled machine).

In some motor applications, shaft seals can be critical for motor performance. For instance, in the food, pharmaceutical, and semiconductor industries, any leakages from motors that are associated with the production processes are not allowed. If using individual sealing elements, shaft seals may involve both static seals, where both the mating surfaces are stationary, and dynamic seals, where there is relative motion between the mating surfaces. For a motor using an individual sealing element, the static seal occurs between the stationary sealing element and the motor endbell and the dynamic seal occurs between the shaft and the sealing element. In most applications, seal elements are directly integrated into the motor components (e.g., motor endbell). For such a circumstance, there exists only dynamic seal between the seal element and rotating shaft. Because of the dimensional variations in dynamic seal surfaces, the design of a dynamic seal system is much more difficult than a static seal system.

Seal performance and lifetime depend upon many parameters, including shaft hardness and surface finish, shaft spiral lead, runout, rotating speed, operation temperature, pressure, seal material, and lubrication conditions. To achieve longer seal life, shaft hardness is expected to be higher than HRc 45. Most effective sealing is obtained with optimum shaft surface finishes. However, highly polished shafts may lead to high seal temperatures.

Many contacting and noncontacting shaft seals have been developed and used to rotating machinery, reflecting the need for specific solutions for individual applications. As always, each type of seal has its pros and cons. Some common seal types will be addressed in the following sections.

3.5.10.1 O-Ring Seal

O-ring seals are indispensable to a wide variety of applications, from vehicles, machine tools, pumps, and air conditioners to washing machines and vacuum cleaners. Advantages of O-rings include the following [3.14]:

- Having sealing ability over a wide range of pressure, temperature, and tolerance
- Normally requiring very little room and are light in weight
- Reusable in many cases
- Ease of service and installation
- Duration of life depending upon the normal aging period of its material
- Providing cost-effective sealing solution

The dimensions of O-ring are defined by the OD (or ID) and its cross section diameter d. O-rings are typically made of elastomers, including acrylonitrile–butadiene, carboxylated

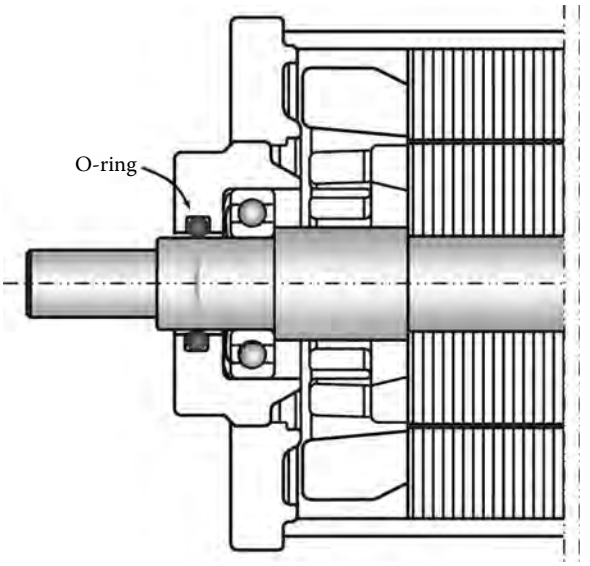


FIGURE 3.21
O-ring seal between a shaft and a motor endbell. Note that groove size prevents rotation of O-ring.

nitrile, ethylene–propylene rubber, fluorocarbon, and silicone rubber (high-temperature applications up to and above 300°C), to name a few.

As demonstrated in Figure 3.21, an O-ring is radially squeezed in an O-ring cavity groove that is machined on the endbell, providing dynamic sealing ability for the bearing. To ensure effective sealing and prevent O-ring rotation, the O-ring's OD must be slightly larger than the groove's diameter. Thus, the O-ring is slightly squeezed in the groove. Furthermore, it is important to keep the shaft diameter no greater than the O-ring's ID. When the O-ring operating temperature increases, it tends to contract. This contraction can result in a tendency for the O-ring to seize the rotating shaft and cause more friction.

There are several parameters for quantitatively describing the squeezing degree of O-ring in rotary O-ring sealing applications:

1. Relative interference, which is defined as the ratio between the differential diameter of the O-ring OD and the groove diameter, and the O-ring OD, that is, $(OD_{O\text{-ring}} - d_{groove})/OD_{O\text{-ring}}$. The maximum relative interference is usually 2%–5%, depending on different applications.
2. Compression squeeze, which is defined as the difference between the uncompressed O-ring cross-section diameter and its radial height after it compressed. Although the recommended minimum value of compression squeeze by many O-ring manufacturers is about 0.005 in. (0.127 mm), reduced compression squeeze, as small as 0.002 in. (0.051 mm), may be used in some rotary sealing systems to minimize friction.
3. Compression ratio, which is defined as the ratio between the compression squeeze and the uncompressed O-ring cross-section diameter. For rod- or piston-type seal, the recommended compression ratio ranges between 5% and 30%, with the target value of 20%.

Hesai Exhibit 1031, page 222

In a continuously rotary application such as an electric motor, the shaft continuously rotates in the ID of the O-ring. Heat due to friction is generated at the same location of the contact and builds up inside the O-ring. Because elastomers are poor thermal conductors, high temperatures can cause O-ring swelling and finally lead to seal failure. This is especially true as the shaft surface speed exceeds 180 ft/min (0.91 m/s) or an O-ring is excessively squeezed.

However, O-ring seals are not recommended for rotary applications under the following conditions [3.15]:

- Pressure exceeding 5.52 MPa (800 psi)
- Temperature lower than −40°C (−40°F) or higher than 107°C (225°F)
- Surface speeds exceeding 3 m/s (9.8 ft/s)

Because O-ring seals are directly in contact with rotating shafts, O-ring wear and sealing life are some of the main concerns in O-ring seal design. Furthermore, rotary seals do not efficiently dissipate heat themselves compared to reciprocating seals. Thus, special care must be taken on the seal operating temperature to avoid seal failures during the service life.

3.5.10.2 Universal Lip Seal

A universal lip seal is used to seal a rotating shaft against the stationary motor housing. As an example presented in Figure 3.22, the seal system has a dual-lip structure: an auxiliary lip seal rides in the face of the motor endbell to establish the first barrier for solid particles such as dusts, and a primary lip seal is closer to the bearing to set the second barrier for fluid leakage. The dimensions of the dual lip are set based on material properties, sealing size, rotating speed, and lip geometry. Optimization of these ensures larger sealing capability, more reliable operation, and longer seal life. The use of a lubricant at the seal tip can promote seal durability and increase the seal life. Contaminations are always harmful for seal performance and should be avoided. With the well-designed seal and proper selections of seal material and lubricant, seals can often last over 10,000 h under clean environmental conditions [3.16]. This type of seal is primarily designed for low-pressure and low-speed applications.

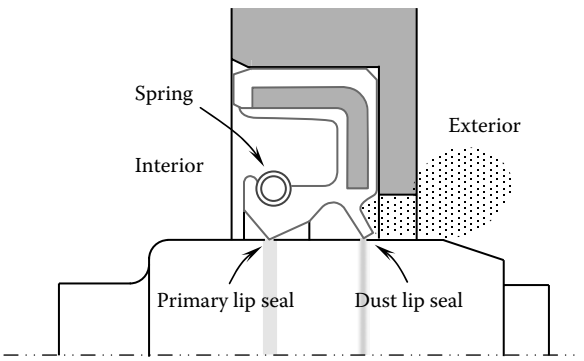


FIGURE 3.22
Universal lip seal with dual lip: a primary lip for sealing fluid and an auxiliary lip for sealing dusts.

3.5.10.3 V-Shaped Spring Seal

This type of seal differs from the previous one in some respects. First, the seal is stretched and mounted onto a rotating shaft rather than on a stationary motor endbell. Typically, the main function of the seal is to act as a face seal to prevent the ingress of dirt, dust, oil, and water. Secondly, the sealing face is the flat end surface of an endbell rather than the cylindrical shaft surface (Figure 3.23). Because the shaft radial runout is generally larger than its axial displacement, this sealing configuration can help improve the sealing performance. In fact, sealing on a flat face is easier than on a curved face. Third, the sealing pressure can be changed by adjusting the relative position of the seal to the sealing face. Finally, a high external pressure always pushes the seal lip closer to the sealing face, leading to the enhanced sealing performance but increased friction on the sealing face.

3.5.10.4 Brush Seal

The initial concept of brush seal was proposed by General Electric in 1955. Since then, the brush seal technique has been greatly developed and found more applications in a variety of industries. In the last three decades, brush seals have attracted more industrial attention in replacing conventional labyrinth seals for their improved resistance to leakage. Today, the brush seal technology represents a promising advance in constructing seal systems with high sealing effectiveness and possible small size. The brush seal uses a biased pattern of wire Kevlar® fibers in contact with the surface of a shaft. This type of seal is remarkably effective in reducing leakage.

Brush seals typically comprise a plurality of elongated Kevlar fibers or wire bristles in contact with a rotating surface. The Kevlar fibers provide a tight rub-tolerant seal that experiences only slight degradation over time. The Kevlar fibers of the seal are compliant in use, and this minimizes damage due to transient impact between the sealing components. A typical brush seal is formed by sandwiching Kevlar fibers between a pair of supporting metal plates that are welded together on their top surfaces (Figure 3.24).

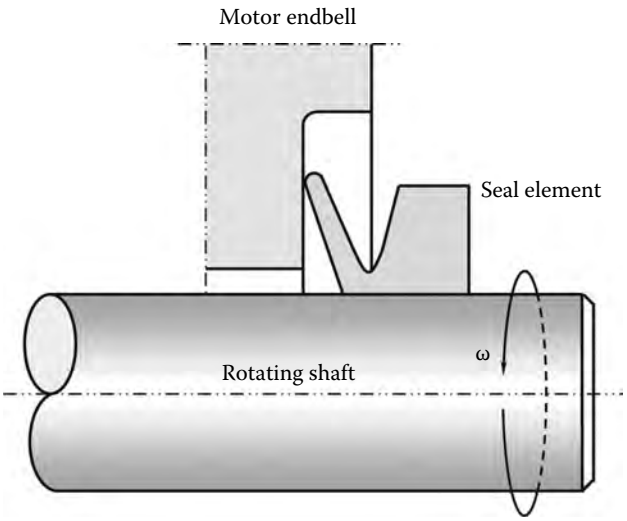


FIGURE 3.23
A V-shaped seal attached to rotating shaft for sealing the shaft at the end surface of motor endbell.

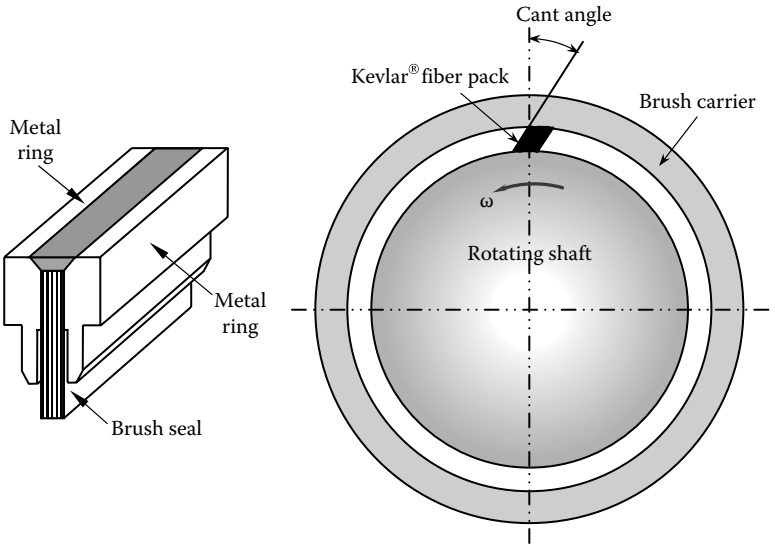


FIGURE 3.24
Brush seal employed Kevlar® fiber pack for sealing rotating shaft.

The Kevlar fibers or bristles of the seal are compliant in use, and this minimizes damage due to transient impact between the sealing components. Brush seal construction consists of three main components: Kevlar pack and back and front steel rings. The brush fibers lie at a cant angle affording the capability of accommodating radial excursions of the rotating shaft. For bidirectional rotating motors, the cant angle is set at zero.

Figure 3.25 presents an example that uses the combination of a flexible silicone rubber brush seal and a labyrinth seal to seal the shaft of rotating machinery for minimizing leakage flows. The brush seal comprises a plurality of elongated wire bristles in contact

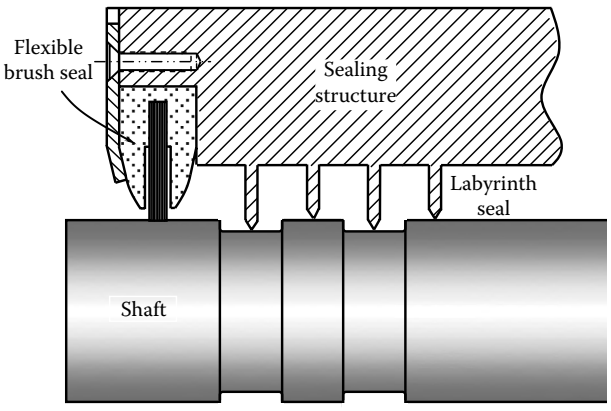


FIGURE 3.25
A flexible silicone rubber brush seal combined with a labyrinth seal to seal the shaft of rotating machinery. The brush seal body is made of heat-resistant magnetic silicone rubber material for high temperature applications (U.S. Patent 6,390,476) [3.17]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 225

with the rotating shaft. The bristles provide a rub-tolerant seal that experiences only slight degradation over time. In order to use the brush seals under high-temperature conditions, they are fabricated with the flexible heat-resistant magnetic silicone rubber material. Therefore, this type of brush seal can fit a wide range of sealing dimensions and complex sealing geometry. By changing the brush bending pattern, the brush seal can be used for either inner sealing where the bristles point radially inward, outer sealing where the bristles point radially outward, or axial sealing where the bristles point parallel to the rotor–shaft axis. Additionally, by forming the brush seal body from the magnetic silicone rubber, the seal may be readily adhered to the metallic sealing components. In fact, this type of brush seal is particularly suitable for applications with large sealing dimensions [3.17].

3.5.10.5 PTFE Seal

PTFE, or more formally polytetrafluoroethylene, was discovered by accident in 1938 by a DuPont scientist. This material has some unique and remarkable properties, including the following [3.18]:

- PTFE has thermal stability across a wide temperature range. It can be used in cryogenic temperatures to seal liquid nitrogen (–196°C), liquid hydrogen (–253°C), and liquid helium (–269°C). It also can keep normal service at temperature up to 260°C for virgin PTFE.
- PTFE has the lowest friction coefficient of any known solid. The material has self-lubricating capability, making it as the excellent seal material in dynamic seal applications.
- Comparing with other main polymeric materials (e.g., UHMW-PE, PEEK, PCTFE, and PI), PTFE has the highest elongation, up to 300%.
- PTFE has very strong chemical resistance and chemical stability. It is inert to virtually all industrial chemicals and solvents even at elevated temperatures and pressures.
- The exceptional insulating properties of PTFE make the material ideally suited for using in electric machines.

High-pressure, high-speed shaft seals are often encountered in equipment or rotating machinery relevant with chemical processing under severe operating conditions. PTFE seals are especially suitable for these applications due to their superior material properties. Some specially designed PTFE seals can withstand pressures up to 55 MPa (7252 psi) and surface speeds up to 15 m/s (49.2 ft/s) [3.18]. A high-pressure, high-speed PTFE shaft seal is presented in Figure 3.26.

3.5.10.6 Noncontact Seal

As one of noncontact seal types, labyrinth seals are the most commonly used sealing configuration in rotating machinery to control the internal leakage of fluid. A typical labyrinth seal is generally characterized by a series of grooves formed along the adjacent surfaces of a rotating shaft. The circumferential grooves make a tortuous axial path that restricts fluid flow across it, as shown in Figure 3.27a [3.19].

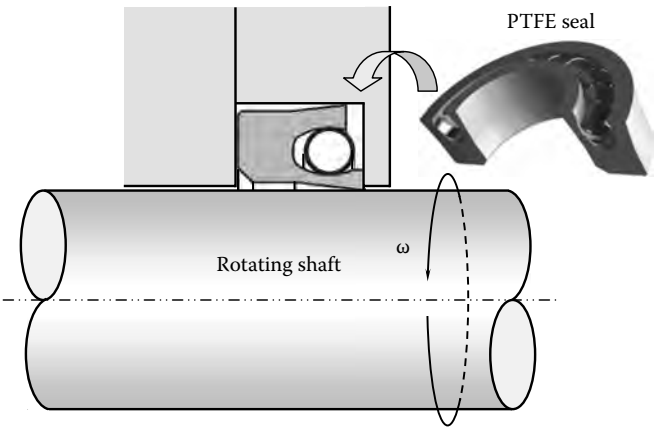


FIGURE 3.26
High-pressure, high-speed dynamic PTFE seal.

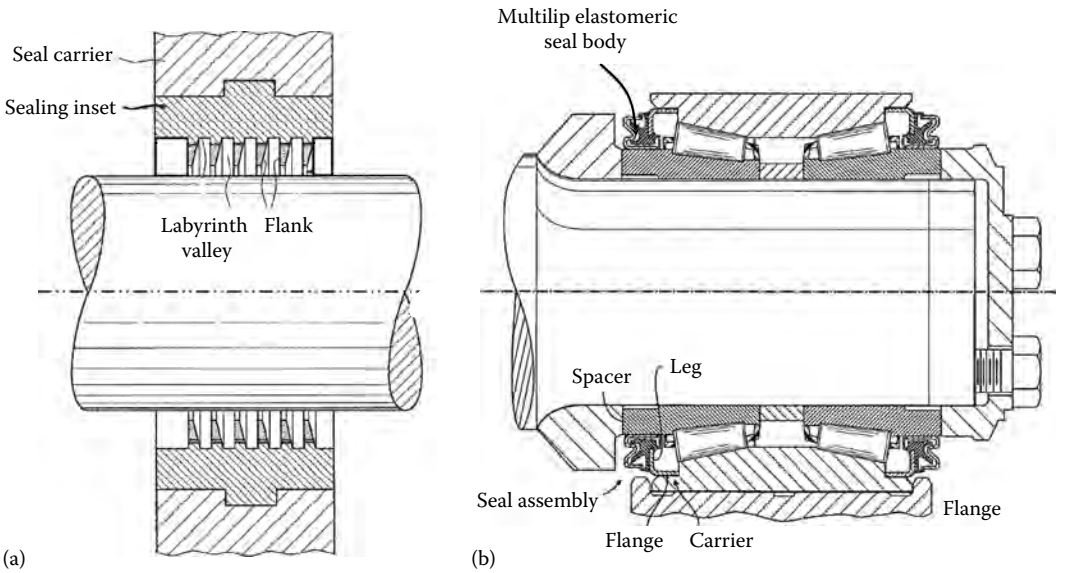


FIGURE 3.27
Noncontact labyrinth seals used in rotating machinery: (a) tooth-groove labyrinth seal (U.S. Patent Application 2012/0228830) [3.19] and (b) multilip labyrinth seal (U.S. Patent 8,342,535) [3.20]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

In motor operation, the labyrinth seal on a rotating shaft provides noncontact sealing action by controlling the passage of fluid. With the labyrinth design, the leakage fluid has to pass through a long and tortuous path, producing high friction forces against the movement of the fluid. In addition, at high rotating speeds, the leakage fluid is subjected to a high angular momentum and hence is flung radially outward. In such a way, the leakage fluid along the tortuous path is disturbed and pushed away from the passage. Consequently, the combination of these two effects makes this type of seal have good sealing capability.

Another type of labyrinth seal is presented in Figure 3.27b, where an elastomeric seal body has multiple radially outward lips in different directions, creating even more complicated flow path for improving sealing performance [3.20].

This type of labyrinth seals is known for its low manufacturing cost, small fiction (only fluid friction involved), no wear, tight radial clearance between the seal and shaft, and acceptable leakage control effectiveness. One important issue in using labyrinth seals is that they are prone to developing cross-coupled forces, which can induce rotor dynamic instability. The effects of pressure differential, rotor speed, entry flow conditions, and seal geometry on the rotordynamic stability were addressed by Benckert and Wachter [3.21] in detail.

3.5.11 Diametrical Fit Types

There are two diametrical fit types: hole-based fitting and shaft-based fitting. In the hole-based fitting, the dimension of the hole is taken as the basis and the clearance or interference is applied to the shaft dimension. In contrast, in the shaft-based fitting, the dimension of the shaft is taken as the basis. Generally, the hole-based fitting is considered better than the shaft-based fitting. This is because holes are often machined by standard tools (drills or reamers) with fixed dimensions and shafts are machined by CNC lathes to any given dimensions. In addition, some standard mechanical parts such as bearings, pulleys, and gears often have standard internal diameters with tight tolerances. However, the shaft-based fitting may be also used in some special applications.

3.6 Stress Concentration

It has long been recognized that any structural and discontinuity in machinery components can alter the stress or strain distributions in the vicinity of the discontinuity and significantly raise the local stress or stain levels. The structural discontinuities can be geometric discontinuity such as grooves, holes, notches, shoulders, keyseats, threads, splines, and sudden changes in geometry and material discontinuities such as voids, cracks, and porosities. As the sources of stress/strain intensification, these discontinuities affect only small regions of actual machinery components or structural members but have little impact on the overall stress/strain pattern. This is extremely dangerous because without any apparent signs, the machinery components or structural members may suddenly fail due to the cracks developed at the stress concentration locations.

In order to quantitatively describe the phenomenon of stress concentration, a dimensionless factor, known as stress concentration factor K, is introduced as the ratio of the actual maximum stress with the stress riser to the nominal stress without the stress riser, that is,

$$K = \frac{\text{Actual maximum stress}}{\text{Nominal stress}} \tag{3.68}$$

More specifically, the normal and shear stress concentration factors, K_t and K_{ts}, are defined respectively as

$$K_t = \frac{\sigma_{\max}}{\sigma_o} \quad \text{for normal stress (underbending, tension, or compression)} \tag{3.69a}$$

$$K_{ts} = \frac{\tau_{max}}{\tau_o} \quad \text{for shear stress (undertorsion)} \tag{3.69b}$$

where the stresses σ_{max} and τ_{max} are the real maximum normal and shear stresses and the nominal stresses σ_o and τ_o are the normal and shear stresses without stress raisers, respectively.

Stress concentration factors for different discontinuity types, loadings, and geometries can be found from some typical mechanical engineering design textbooks [3.6,3.22,3.23].

3.7 Torque Transmission through Mechanical Joints

For a shaft to transmit torque or power to a driven machine, some mechanical components are needed to be mounted on the shaft, such as coupling, pulley, gear, hub, sheaves, sprocket, and/or cam, to name a few. To ensure that these mechanical components are firmly connected with the shaft, it must satisfy the following design requirements: (1) no relative motion between the shaft and the attached components, (2) easy and rapid assembly/disassembly, (3) easy to manufacture, (4) minimized stress concentration, and (5) induced less vibration and acoustic noise.

Motor torque is normally transmitted from the motor shaft to the mounted components using keys, splines, pins, or other torque-transmitting elements.

3.7.1 Keyed Shafts

Most motor shafts use keys for securing mating elements rotating together to transmit motor torque to external equipment. The advantages of using keys include cost-effectiveness, ease of installation, and high reliability. In addition, keys used in the joint systems may function as mechanical *fuses* to prevent further damage to joints or other parts of the machine.

3.7.1.1 Selection of Key Material

In the design of keyed connections, it is important to select proper key material. An improper key material may cause damage to the key itself, shaft, hub, or even motor failures in severe cases. In transmitting power/torque, there are two basic failure modes: shear across the shaft–hub interface and compression due to the bearing forces between the key (lower portion) and shaft and key (upper portion) and hub. Since shafts are subjected to combined loads, the key material must have high strengths, high hardness, proper elongation, and tolerance to shock loads.

The required key strengths are closely associated with the key length. As the key length is 1.6 times the shaft diameter, the key stress is equal to the shaft stress. As the key length is comparable to the shaft diameter, the key stress becomes much higher than the shaft stress. However, because there is a high contact pressure between the shaft and the hub bore due to the interference fit to transmit a considerably large portion of power/torque, it is generally accepted that the key material can be chosen with the same strengths and hardness as the shaft. In practice, carbon steel is the most common material for keys.

3.7.1.2 Stress Analysis of Key and Keyseat

Numerous types of keys are available, including flat keys, square keys, round keys, gib-head keys, feather keys, and Woodruff keys. Among these, flat keys are widely used in the motor industry. To accept a key, longitudinal grooves are made into a shaft and its mating element (e.g., sheave, hub), known as keyways or keyseats. According to the milling pattern, a shaft keyseat can be classified as either sled runner keyseat or profile keyseat. The sled runner keyseats are milled with a disk milling cutter that has the same width of the keyseat. The profile keyseat is produced by a circular endmill that has a diameter equal to the keyseat width (Figure 3.28). Though the type of profile keyseat is widely used in industries, the type of sled runner keyseat is desirable for its lower stress concentration factor. Figure 3.29 presents a PM rotor with a profile keyseat.

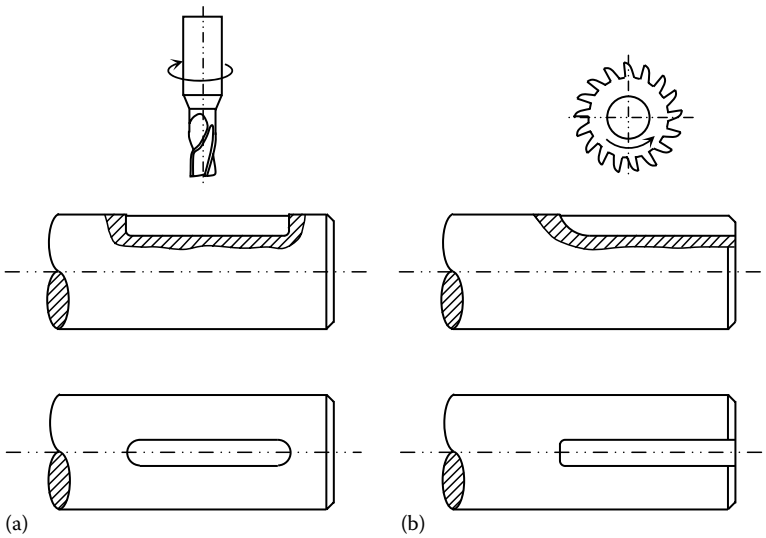


FIGURE 3.28
Keyseat types: (a) profile keyseat and (b) sled runner keyseat.

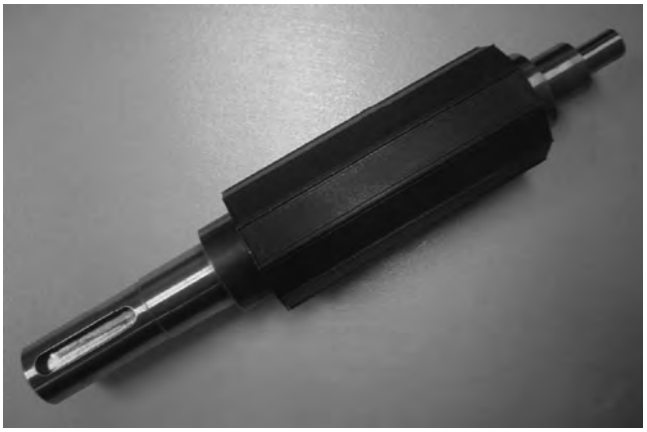


FIGURE 3.29
A keyseat is machined at the drive side of a PM rotor.

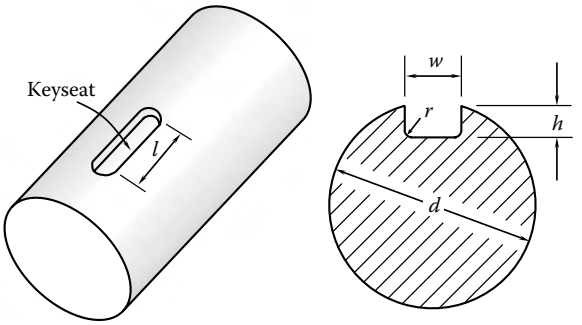


FIGURE 3.30
Dimensions of the profile keyseat made on the motor shaft. Small fillets are made at the keyseat corners to decrease stress concentration.

A keyed shaft is primarily characterized by five variables: shaft diameter d, keyseat width w, keyseat depth h, effective keyseat length l, and the fillet radius at the keyseat bottom edge r (Figure 3.30). Usually, the fillet radius is about 2% of the shaft diameter. This gives the stress concentration factor of the keyseat approximately 3. If a keyseat has sharp corners at the bottom, the stress concentration factor can be significantly high. As reported by Peterson, typical stress concentration factors are approximately 1.1–2.0 for stepped shafts and 2.5–4.0 for keyseats [3.24]. It is noted that the rounded key ends avoid high local stresses.

As illustrated in Figure 3.31, a key is inserted in the keyseats of the shaft and sheave/hub. To avoid interference with the keyseats at the key edges, the radii of the key must be larger than those of the keyseats. The key must be precisely fitted to the keyseat in the shaft and sheave/hub with a small radial clearance on the top. Thus, motor torque is transferred from the shaft to the sheave/hub through two contact surfaces at the sides of the key. While the contact force of the shaft F exerts on the lower section of the key, the opposite reaction force F' from the sheave/hub exerts on the upper section of the key (where F is slightly larger than F'). These resulting forces cause shear and compressive stresses. Thus, for this locking system, there are two major failure mechanisms: shear failure and crushing failure, which are applicable to all components (shaft, key, and sheave/hub).

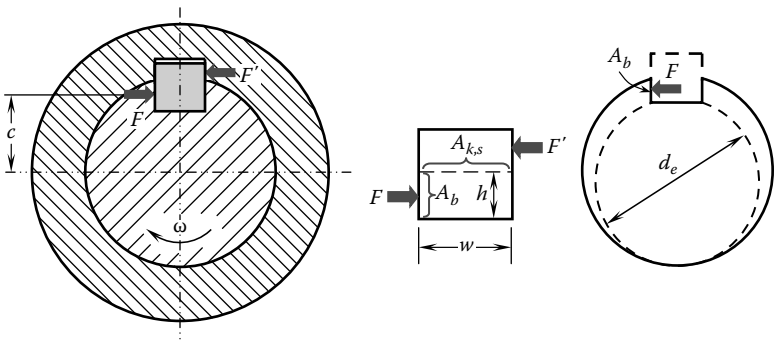


FIGURE 3.31
Shear failure and crushing failure applied on a flat key and a keyed shaft.

In the shear failure mechanism, a key withstands opposite forces on its upper and lower sections, causing a shear stress on the key shear plane $A_{k,s}$,

$$A_{k,s} = lw \tag{3.70}$$

The maximum shear stress τ_{max} is reached when the force F approaches its maximum value F_{max}:

$$\tau_{max} = \frac{F_{max}}{A_{k,s}} \tag{3.71}$$

The factor of safety for the shear failure is defined as

$$n_s = \frac{S_{sy}}{\tau_{max}} = \frac{0.577\, S_y}{\tau_{max}} \tag{3.72}$$

where S_{sy} and S_y are shear yield strength and tensile yield strength (according to the distortion-energy theory, $S_{sy} = 0.577 S_y$), respectively. Thus, the maximum torque that can be transmitted by the key is

$$T_{k,max} = F_{max} c \approx F_{max}\left[\frac{1}{2}(d-h)\right] = \frac{0.289\, lw(d-h) S_y}{n_s} \tag{3.73}$$

The stress analysis of a keyed shaft is different from the analysis of the key. As a good practice to estimate the torque that can be transmitted by the keyed shaft with a reduced shaft diameter, the concept of an effective diameter d_e is introduced into the analysis:

$$d_e \cong \sqrt{(d-h)^2 + w^2} \tag{3.74}$$

Correspondingly, shear stress resulting from the applied torque T is

$$\tau = \frac{Tr}{J} \tag{3.75}$$

where r is the shaft radius and

$$J = \frac{\pi d_e^4}{32} \tag{3.76}$$

The maximum shear stress is reached at $r = d_e/2$:

$$\tau_{max} = \frac{T_{max}(d_e/2)}{\pi d_e^4/32} = \frac{16 T_{max}}{\pi d_e^3} \tag{3.77}$$

Combining the previous equation, it follows that

$$T_{max} = \frac{0.1133 d_e^3 S_y}{n_s} \tag{3.78}$$

In the crushing failure mechanism, the compressive force F acts on the bearing area A_b of the shaft and the key, respectively. The maximum bearing stress is given as

$$\sigma_{\max} = \frac{F_{\max}}{A_b} = \frac{F_{\max}}{hl} \tag{3.79}$$

The factor of safety for the crushing failure is defined as

$$n_c = \frac{S_{cy}}{\sigma_{\max}} \tag{3.80}$$

where S_{cy} is the compressive yield strength. Hence, the corresponding maximum torque becomes

$$T_{\max} = F_{\max} c \approx F_{\max}\left[\frac{1}{2}(d-h)\right] = \frac{hl(d-h)S_{cy}}{2n_c} \tag{3.81}$$

For steady loads, $n_s = n_c = 2$ is frequently adopted. As for shock loads, the factor of safety can be chosen between 2.5 and 4.5 [3.25].

3.7.1.3 Key Fitting

Usually, higher motor torques require tight fits (e.g., interference fits) between keys and keyseats. When a key is loosely fitted into the mating keyseat as a clearance fit. Thus, as the key is subject to the transverse forces from both the shaft and hub. Thus, the key rolls in the keyseat to pit the edges and surfaces of the keyseat (Figure 3.32). This condition can cause high loadings between the key and keyseats, leading to the failure of either the key or shaft, or both. It becomes even worse if the motor frequently changes its rotating direction. On the other hand, too tight fitted keys and keyways may also result in mechanical failures. To ensure appropriate fit, the width and height dimensions of

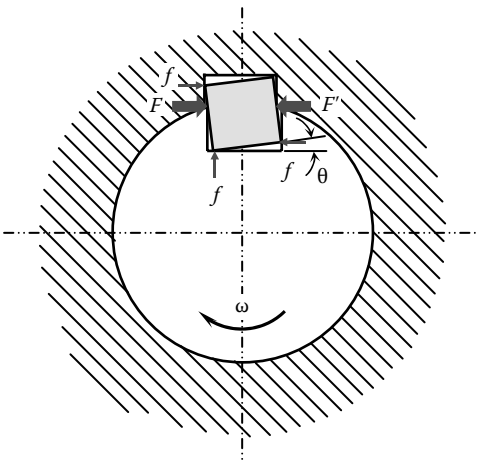


FIGURE 3.32
A key inserted loosely in a keyseat, causing high edge loadings between the key and keyseat.

keys and keyways must be held to recommended tolerances, as defined in industrial or national standards.

Because the stress analysis of key fitting is very complicated, it is usually performed using FEM tools.

3.7.1.4 Stress Concentration Factors of Keyed Shafts

A machined keyseat in a shaft does not only reduce the shaft strength and stiffness but also cause stress concentration in the shaft. The impacts of keyseats on the torsional stiffness of shafts were addressed first by Filon in 1900 [3.26]. Subsequent to that, a large number of theoretical, experimental, and numerical investigations have been conducted for determining stress concentration factors in keyed shafts [3.7,3.27–3.35, as some examples]. Pilkey [3.22] proposed a correlation of stress concentration factor for a specific case of $w/d = 1/4$ and $h/d = 1/8$:

$$K_t = 1.9753 + 0.1434\left(\frac{0.1}{r/d}\right) - 0.0021\left(\frac{0.1}{r/d}\right)^2 \quad r/d \in [0.005 : 0.07] \tag{3.82}$$

This correlation is found overestimating the K_t values. More recently, Pedersen [3.35] reported a better correlation for the same case:

$$K_t = 1.8755 + 0.1397\left(\frac{0.1}{r/d}\right) - 0.0018\left(\frac{0.1}{r/d}\right)^2 \quad r/d \in [0.003 : 0.07] \tag{3.83}$$

The comparison of the two previous correlations is given in Figure 3.33. It can be seen for $r/d < 0.005$, the predicted K_t values from two correlations are almost identical. For large r/d ratios, the correlation of Pilkey is about 4%–5% higher than that of Pedersen.

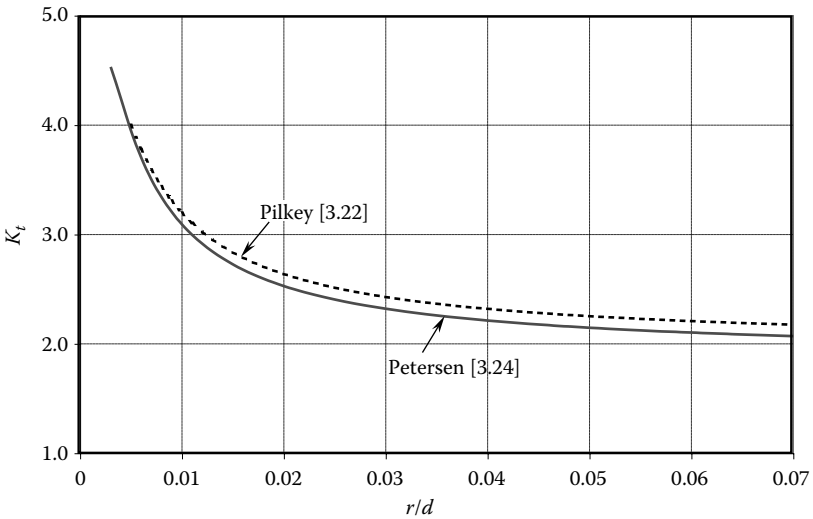


FIGURE 3.33
Comparison between two correlations of stress correlation of keyed shaft for the case of $w/d = 1/4$ and $h/d = 1/8$.

More general, stress concentration factors of keyed shaft depend on three ratios: fillet radius to shaft diameter ratio r/d, keyseat width to shaft diameter ratio w/d, and keyseat height to shaft diameter ratio h/d. As discussed by Pedersen [3.35], the width ratio and height ratio can be linked together and fall naturally into two groups depending on the shaft diameter:

$$\frac{w}{d} = 1.2662\frac{h}{d} + 0.0866 \quad d \in [6:38]\,\text{mm} \tag{3.84a}$$

$$\frac{w}{d} = 1.6683\frac{h}{d} + 0.1055 \quad d \in [38:500]\,\text{mm} \tag{3.84b}$$

Corresponding to these two diameter ranges, stress concentration factors can be correlated as

$$K_t = \left(1.4786\frac{h}{d} + 0.6326\right)\left(\frac{r}{d}\right)^{\left[0.868(h/d)^2 - 0.4392(h/d) - 0.2369\right]} \quad d \in [6:38]\,\text{mm} \tag{3.85a}$$

$$K_t = \left(1.0428\frac{h}{d} + 0.5355\right)\left(\frac{r}{d}\right)^{\left[2.8074(h/d)^2 - 0.8091(h/d) - 0.2476\right]} \quad d \in [38:500]\,\text{mm} \tag{3.85b}$$

respectively.

Many researchers and investigators have contributed to determine shear stress concentration factor K_{ts} for keyed shaft. Leven [3.28] did the early work for the stress concentration arising from torsion in a shaft containing a keyseat and obtained K_{ts} values at the straight part of a keyseat as a function of r/d. Using the electroplating method, Okudo et al. [3.32] experimentally examined stresses in keyseats under the most practical conditions. To obtain the limit value for a shaft twisted only through a key, the fitting is devised so as to minimize the surface friction at the fitted portion. From the Peterson's charts [3.22], for a ratio of $r/d = 0.02$, $K_{ts} = 2.60$ without the key in place, and $K_{ts} = 3.0$ with the key in place.

Thus, the allowable maximum torques transmitted by a keyed shaft in the shear failure and crushing failure become

$$T_{sh,\max} = \frac{0.289 d S_y}{n_s K_{ts}} A_{sh,s} \tag{3.86}$$

$$T_{\max} = \frac{hl(d-h)S_{cy}}{2n_c K_t} \tag{3.87}$$

3.7.2 Spline Shafts

Spline shafts are widely used in various industries such as automotive, aerospace, heavy machinery, machine tool, textile machinery, and mining. Splines perform the same function as keys in transmitting torque from a motor to a driven machine. In fact, a spline can be viewed as a series of axial keys machined into the shaft (Figure 3.34), with corresponding

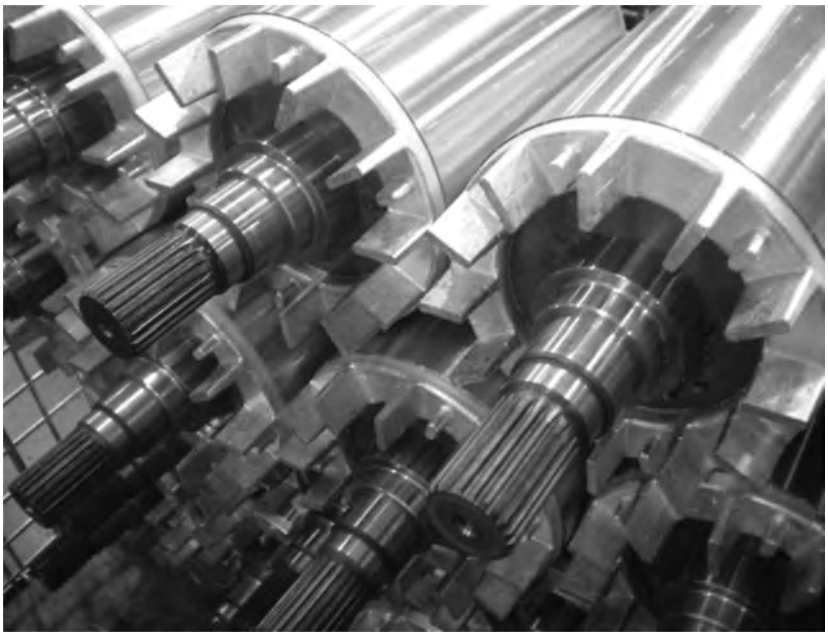


FIGURE 3.34
Spline shafts used in induction motors.

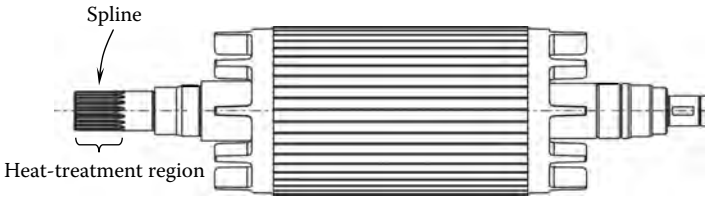


FIGURE 3.35
The spline is heat treated for increasing surface hardness, wear resistance and fatigue life.

grooves machined into the bore of the mating element. As demonstrated in Figure 3.35, the spline is machined at the drive side of the shaft. To increase the surface hardness, wear resistance, and fatigue life, shaft splines must go through heat treatment processes. It should be noted that to prevent shaft deformation, heat treatment is applied to the limited region that contains only spline. However, improper heat treatment of steel and alloys may lead to temper embrittlement, tempered martensite embrittlement, or hydrogen embrittlement, depending on different heat treatment processes [3.36]. Consequently, all of these embrittlement types could cause spline shaft failure, as shown in Figure 3.36.

3.7.2.1 Advantages of Spline Shafts

There are a number of advantages of spline connections over key connections: (1) Because the spline is machined directly on the shaft, no relative motion can occur between the spline teeth and the shaft. (2) With multiple spline teeth, the torsion and bending loads are uniformly distributed to the teeth. (3) Each spline tooth transfers a much lower torsion

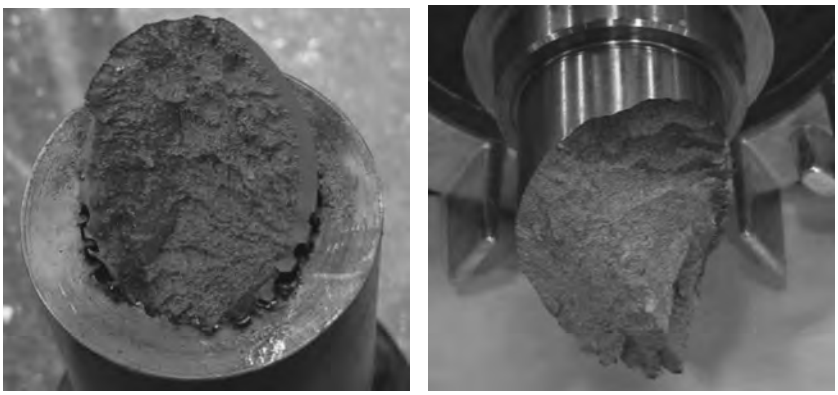


FIGURE 3.36
Failure of spline shafts due to improper heat treatment.

load compared with a keyed shaft. Thus, spline shafts can transmit large torque or power. (4) A spline shaft has an ability to accommodate large axial movements between the shaft and the mating part while simultaneously transmitting torque. (5) An involute spline shaft has an autocentering dynamics that allows a semi- or full-floating coupling. (6) The mating part can be indexed to various positions with respect to the shaft spline. (7) A helical spline shaft can drive the axial and rotary motion at the same time. (8) Using the hob to cut spline shafts can offer notable advantages both in terms of manufacturing efficiency and spline profile accuracy over the manufacturing method for keyed shafts. (9) Splines are usually heat-treated to increase the surface hardness and wear resistance. By contrast, it is very rare to do heat treatment to shaft keyseats. (10) Uniformly distributed spline teeth have little impact on shaft unbalance. For a keyed shaft, the keyseat can bring an unbalance problem. (11) While a shaft with an external key is subject to a high stress concentration at the mouth of keyseat, a spline shaft has much lower stress concentration due to the distribution of loads to all teeth.

3.7.2.2 Spline Types

Although a number of spline types exist, the most commonly used splines across a wide variety of industries are involute splines and straight-sided splines (sometimes also referred to as parallel key splines), as shown in Figure 3.37. Involute splines typically have pressure angles of 30, 37.5, or 45. This type of splines tends to be self-centering that equalizes bearing and stresses among all teeth. This characteristic feature also helps achieve better concentricity between the shafts of the motor and its mating machine. In contrast with involute splines, straight-sided splines have ridges with a rectangular profile. This type of splines has been standardized by SAE to have 4, 6, 10, or 16 splines, which is equally distributed around the circumference of a shaft. Today, straight-sided splines still remain popular in many applications, especially in the machine tool and automotive industries.

Involute splines are favored over straight-sided splines because of their self-centering capability, higher torque-transmitting capacity, and greater strength, as well as some manufacturing advantages (e.g., for any given pitch, the spline cutting tools can cut any number of teeth).

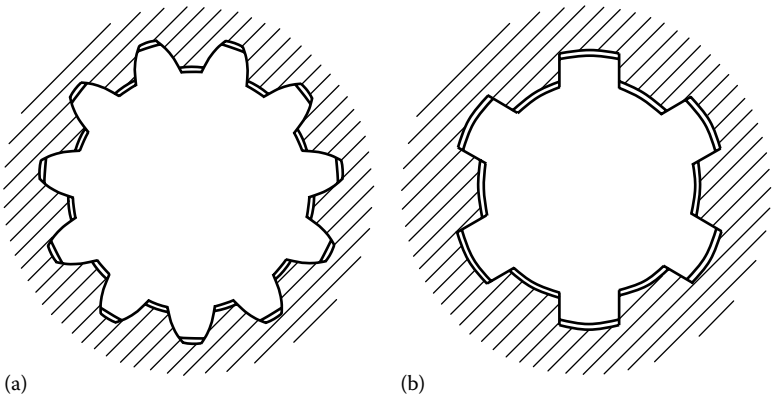


FIGURE 3.37
Spline types: (a) involute spline and (b) straight-sided spline.

3.7.2.3 Stress Concentration Factors of Spline Shafts

When spline motor shafts transmit torque or power to a driven machine, they are subject to torsion loads. A mathematical analysis was conducted by Okubo [3.37] for a shaft with a number of longitudinal semicircular grooves. It was found that as r/d approaches zero (where r is the fillet radius at the tooth root and d is the shaft diameter), the stress concentration factor $K_{ts}=2$. By using the 3D photoelastic technique, Yoshitake [3.38] performed an experimental investigation on eight-tooth splines. In his work, the torsion stress concentration factors were measured by the wedge method with varied tooth fillet radius. The test results indicate that the involute splines are more advantageous than parallel side splines from the standpoint of torsion stress concentration factor. It was reported that the torsion stress concentration factor is 2.8 for an involute spline and is 3.1 ($r=1.5$ mm), 3.4 ($r=1.0$ mm), and 4.4 ($r=0.5$ mm) for parallel splines.

3.7.3 Tapered Shafts

Since keys and splines usually produce stress concentrations on motor shafts, tapered shafts may provide a promising solution with keyless designs. As shown in Figure 3.38, a taper is made at the drive end of a motor shaft with a small cone angle α and inserted in the bore of the traction sheave/hub. A number of fasteners (screws or bolts) are engaged with the shaft to ensure the close contact of the tapered shaft with the sheave/hub. As the fasteners are tightened, the tapered shaft is further pushed into the bore of the sheave/hub, creating a contact pressure at the contact surface.

By comparing with conventional shafts, tapered shafts are more expensive and may require special tooling. In addition, this method may not be suitable for very high torque conditions.

Assume the tapered shaft is perfectly contacted with the sheave/hub, the contact diameter d is a function of the contact distance x, as

$$d(x) = d_1 + \frac{d_2 - d_1}{L} x \tag{3.88}$$

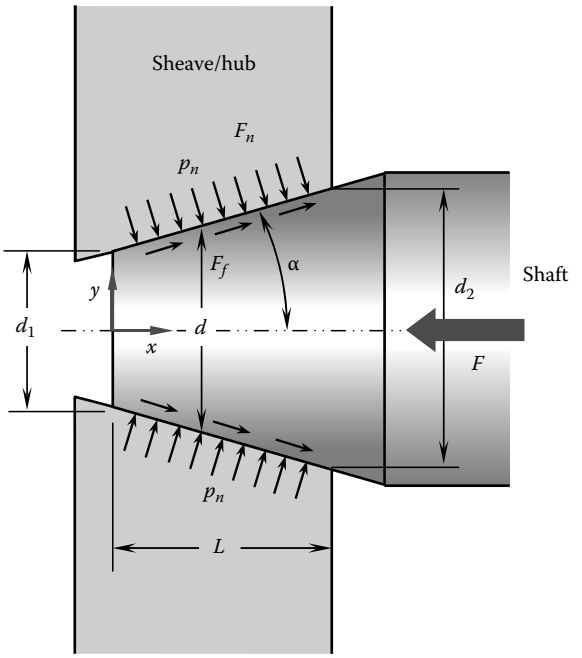


FIGURE 3.38
A tapered motor shaft against a sheave/hub.

The area of the contact surface between two components is thus

$$A = \pi \int_0^L d(x)\,dx = \pi \int_0^L \left(d_1 + \frac{d_2 - d_1}{L} \right) x dx = \frac{\pi L}{2}\left(d_1 + d_2\right) \tag{3.89}$$

With the normal contact pressure p_n applying on the interface of the tapered shaft and the sheave/hub, the force acting on the contact surface can be decomposed as the normal contact force F_n, which is perpendicular to the contact surface, and the friction force F_f, which is parallel to the contact surface. Since the two components are perfectly contacted each other, the contact pressure p_n is uniform on the interface, following that

$$F_n = \pi p_n \int_0^L d(x)\,dx = \pi p_n \int_0^L \left(d_1 + \frac{d_2 - d_1}{L} x \right) dx = \frac{\pi L p_n}{2}(d_1 + d_2) \tag{3.90a}$$

$$F_f = \mu F_n = \frac{\pi \mu L p_n}{2}\left(d_1 + d_2\right) \tag{3.90b}$$

where μ is the friction coefficient between two contact components.

At the equilibrium condition, all forces are balanced with each other, $\sum F = 0$. The axial pressing force F can be expressed as

$$F = p_n A\left(\sin\alpha + \mu\cos\alpha\right) \tag{3.91}$$

The torque transmitted by the shaft can be determined as

$$T = \int \frac{d}{2} \mu p_n \, dA = \frac{\mu}{2} \int_0^L \frac{\pi d^2 p_n}{\cos\alpha} dx = \frac{\pi \mu p_n L}{6\cos\alpha}\left(d_1^2 + d_1 d_2 + d_2^2\right) \tag{3.92}$$

In a similar manner, a governing equation for calculating the brake torque of a cone clutch was derived in Shigley's textbook [3.39]. By assuming uniform contact pressure, the brake torque can be expressed as

$$T = \frac{F_n \mu}{3\sin\alpha} \frac{d_2^3 - d_1^3}{d_2^2 - d_1^2} \tag{3.93}$$

where

$$F_n = \frac{\pi p_n}{4}\left(d_2^2 - d_1^2\right) \tag{3.94}$$

Combining Equations 3.93 and 3.94 yields

$$T = \frac{\pi \mu p_n}{12\sin\alpha}\left(d_2^3 - d_1^3\right) \tag{3.95}$$

Note that in Equation 3.92, the length of the tapered shaft L can be expressed as

$$L = \frac{d_2 - d_1}{2\tan\alpha} \tag{3.96}$$

Substituting Equation 3.96 into 3.92, Equation 3.92 is identical to Equation 3.95.

However, there is a difference in force (or pressure) equation between two systems. By replacing L in Equation 3.90a with 3.96, it gives

$$F_n = \frac{\pi p_n}{4\tan\alpha}\left(d_2^2 - d_1^2\right) \tag{3.97}$$

Comparing Equations 3.97 and 3.94, it can be clearly seen that a factor of $1/\tan\alpha$ is missing in Equation 3.94 (Shirgley's equation). This indicates that Equation 3.94 is a special case of Equation 3.97 with $\tan\alpha = 1$ (i.e., $\alpha = 45°$). This has been confirmed by Keesee [3.40] by using an FEA analysis to calculate the contact pressure and contact force on the interface. The numerical results have shown good agreement with the data obtained from Equation 3.97.

3.8 Fatigue Failure under Alternative Loading

Rotating shafts are frequently subject to partial or complete reversing loads, resulting fatigue failure.

The fatigue stress concentration factor is defined as

$$K_f = \frac{\text{Maximum stress with discontinuity}}{\text{Nominal stress without discontinuity}} \tag{3.98}$$

The relationship between K_f and K_t is found as

$$K_f = 1 + q(K_t - 1) \tag{3.99}$$

where q is the notch sensitivity of a material and $0 \le q \le 1$. Thus, it gives that $1 \le K_f \le K_t$. If notch sensitivity data are not available, it is conservative to use K_t in fatigue calculations.

3.9 Shaft Manufacturing Methods

There are a variety of shaft manufacturing methods in manufacturing industries. None of the manufacturing methods is perfect. Each has its own set of advantages and disadvantages. The determination of the optimal manufacturing process depends on many factors, such as production efficiency, dimension accuracy, surface finishing, influence on part's strength, processing cycle, shaft material, shaft size, equipment availability, and cost.

3.9.1 Machined Shaft

A majority of motor shafts are manufactured by machining for high productivity, low cost, and easy tolerance control.

A machined shaft usually passes through a series of machining processes. As the shaft is machined from a raw material, in order to increase the productivity, it is typical to use a common lath first to quickly machine it to the dimensions that are close to its final dimensions. Then, use a precise machine such as a CNC to machine the shaft to the final dimensions. For some important sections, such as bearing journals, the shaft may go through a heat treatment and finally grinding it to the final dimensions.

During these processes, the shaft has to be machined on different machines. Therefore, holding the shaft's geometric tolerances such as roundness, cylindricity, runout, concentricity, and straightness is critical in obtaining satisfactory shaft balance and lowing motor vibration. As a matter of fact, this depends entirely on the success of the initial centering phase of the machining operation. The key is providing a good reference point for machining between centers.

In motor shaft manufacturing, endworking of shafts is often incorporated as a primary operation of a series of machining processes. Endworking can eliminate some secondary machining operations by combining them with initial end finishing, thereby increasing productivity and helping to decrease the part process scrap rate for subsequent operations [3.41].

Major benefits of the endworking process include the improved concentricity, roundness, and squareness of the machined surfaces. This greatly reduces the shaft runout. Practically, the more accurate shaft centers allow for more accurate dimension and position controlling, in favor of shortening the production cycle time and costs.

Endworking is conventionally used for centering and facing a shaft that is loaded on a lathe between centers in high-volume production of shafts. Presently, the endworking process is often performed by a CNC double-end machine to work on both shaft ends at the same time for achieving cost savings and high productivity.

3.9.2 Forged Shaft

Forging is a manufacturing process that has been extensively used in industries for producing high-strength and high-quality machine parts. In a forging process, under high pressure, a metal workpiece undergoes large plastic deformation, resulting in an appreciable change in shape or cross section. Unlike in a conventional machining process that the material grain is cut off, forging increases the strengths and toughness of metal by aligning the grain along the line of potential stress. Moreover, because of the enormous pressure involved during forging process, a forged part is denser in the bulk due to the reduction or elimination of porosities and voids. Thus, the forged shafts have higher strength-to-weight ratio and higher corrosion resistance. However, because of the high cost of forging tooling and required multiple steps, forging is generally more expensive than most manufacturing processes. Motor manufacturers often use medium-carbon steel to make forged motor shafts.

Several forging processes are available, including cold forging and hot forging. Cold forging is performed at room temperature and requires a very high forging pressure to overcome the deformation resistance of the material. This indicates that cold forging is limited to relative small shafts with simple geometries. Cold forging can achieve product features such as smooth surfaces, tight tolerances, concentric diameters, and beneficial grain flow. It offers significant cost-saving advantages as the result of the material saving. In contrast, hot forging is performed above the material recrystallization temperature and thus generates oxide scales on the forged shaft surface.

A method for forging a stepped shaft through a combination of forging operations has been reported recently [3.42]. The method includes providing a billet of a predetermined mass; heating the billet; cross-wedge rolling the billet to form an intermediate workpiece having a first cylindrical portion and a second cylindrical portion that are axially spaced apart by a neck that has a smaller diameter than the two portions; and performing at least one upset forging operation on the end of the intermediate workpiece to enlarge the first cylindrical portion such that in at least one location, its diameter is larger than a diameter of the billet and a diameter of any other shaft portion (Figure 3.39).

3.9.3 Welded Hollow Shaft

Motor weight can be very sensitive in some applications such as satellites and aircrafts. Motor designers always take tremendous efforts to reduce the machine weight and increase its torque density. One of the effective methods is to use hollow shafts to lower the shaft weights without affecting their functions and performances. In fact, a hollow motor shaft can not only reduce its weight and inertia but also provide a passage for a cooling fluid (air or liquid) passing through the shaft interior to cool the rotor.

Hollow shafts are often used in direct drive motors and vehicle motors that have large shaft diameters. Because of the larger diameter, a hollow shaft has a higher stiffness/mass ratio, a higher natural frequency, and lower deflection, compared with a solid shaft having the same mass.

Hesai Exhibit 1031, page 242

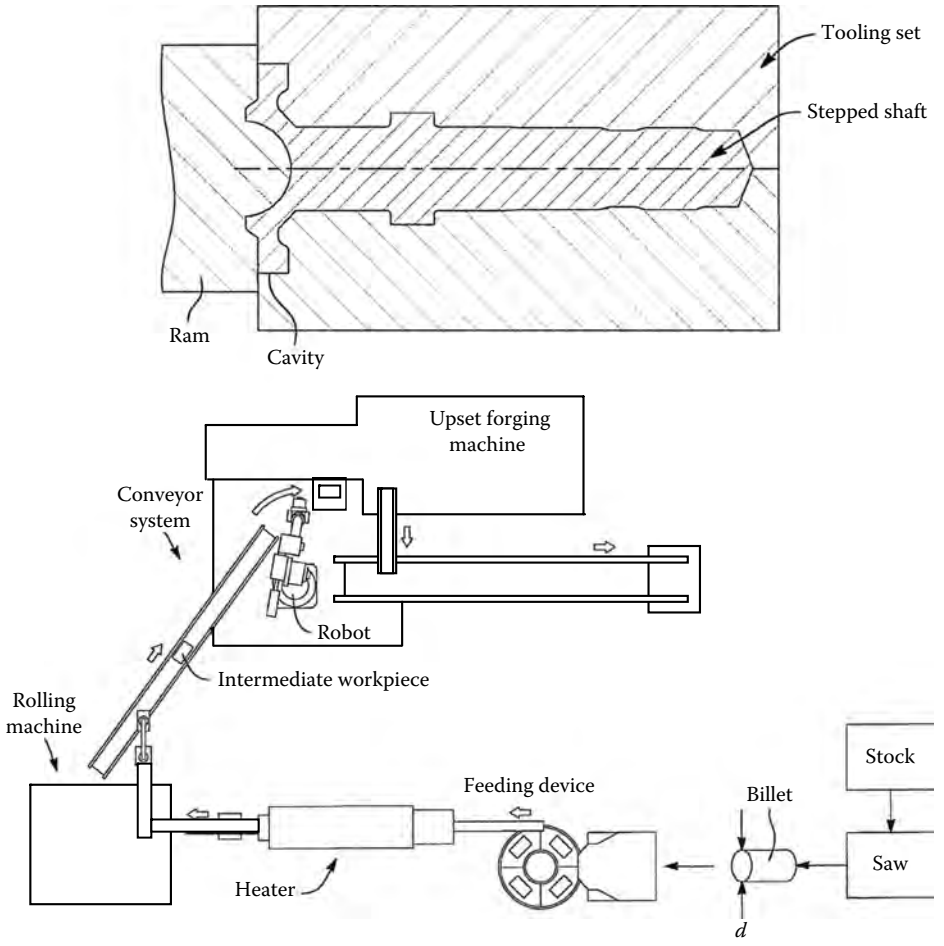


FIGURE 3.39
Fabrication system for producing forged stepped shafts (U.S. Patent 7,866,198) [3.42]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

The fabrication of hollow shafts may involve several different manufacturing processes such as welding, machining, and casting. Among them, welding is the most cost-efficient way for making hollow shafts. Therefore, it requires that the shaft material has a good weldability.

The weldability of steel strongly depends on the chemical composition of steel, especially carbon. Higher quantities of carbon and other alloying elements result in a lower weldability. To quantitatively evaluate the influence of the contributions of the various alloying elements on the difficulties encountered in alloy welding, a concept of carbon equivalent (CE) is introduced in an effort to measure weldability with a single number. CE is an empirical value expressed in weight percent. Several empirical formulas are available in literature. One popular expression, which is recommended by the International Institute for Welding and most often used in ASME applications [3.43,3.44], is given in the following (all elements in mass percentage):

$$CE = C + \frac{Cr + Mo + V}{5} + \frac{Mn + Si}{6} + \frac{Ni + Cu}{15} \tag{3.100}$$

Another popular formula for predicting CE is known as Dearden and O'Neill formula, which is similar to (3.100) but without the Si term. Dearden and O'Neill [3.45] first proposed the CE to assess the weldability of alloy steels. Later, their formula was revised slightly by the International Institute of Welding (IIW).

A CE value of 0.30 or less is considered to be good for welding and no special precautions are required. For $0.30 < CE < 0.4$, modest preheat is necessary. Steel with a CE value in excess of about 0.5 cannot be easily welded because of their increased tendency to develop hydrogen-induced cracking in the heat-affected zone. Hence, welding of steel with a high CE value requires a special care for the preheat process. Figure 3.40 is an example of hollow shaft used in hybrid vehicles.

An alternative design of hollow shaft is presented in Figure 3.41. It can be seen that the hollow shaft is formed by integrating two shafts together with the rotor laminated core through the interference fitting. In order to ensure the firmness of the rotor assembly, welding is used between shaft 1 and the rotor core.

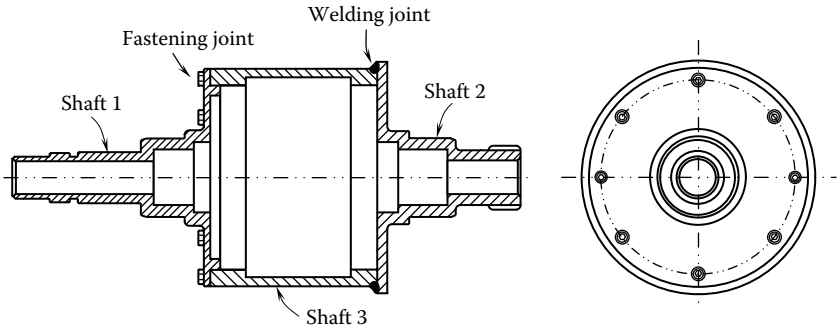


FIGURE 3.40
Three-piece hollow shaft design using fastening and welding joint methods.

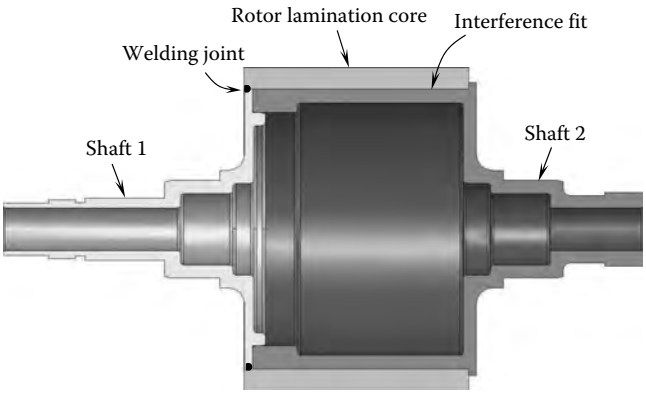


FIGURE 3.41
Alternative hollow shaft design: two shafts are integrated together with the rotor lamination core.

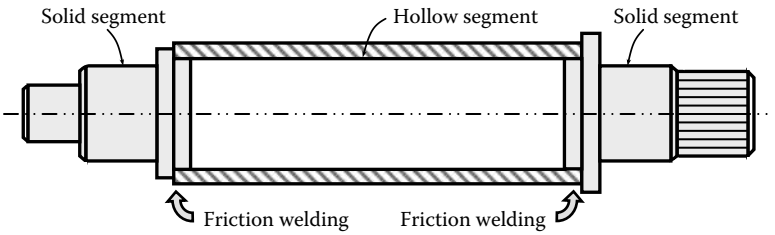


FIGURE 3.42
Hollow shaft fabricated by friction welding technology.

A hollow shaft can be also fabricated by utilizing the friction welding technology. Friction welding is a melt-free process that generates heat through mechanical friction between a rotating and a stationary component. Because there is no melt involved, this processing method is not a truly welding process in the traditional sense but more like a forging process. One of the advantages of this technology is to allow different materials to be joined together, such as aluminum and steel and stainless steel and carbon steel. A hollow shaft generally consists of three segments: two solid segments and one pipe segment. Two solid segments are friction welded to both ends of the center pipe segment (Figure 3.42). The formed shaft is then machined to its final dimensions. With this manufacturing technique, the hollow shafts can achieve up to 50% or more weight reduction compared to the similar solid shafts.

3.9.4 Shaft Measurement

Confirming that manufactured shafts meet all requirements requires validating shaft hardness, geometric dimensions and tolerances, and other items. Figure 3.43 shows a coordinate measurement machine (CMM) that is used to measure the geometric dimensions and other characteristics of shafts. The CMM is composed of three axes, that is, *X*, *Y*, and *Z*, forming a regular 3D coordinate system. In a measurement, the shaft is put on a pair of *V*-shaped stands on the table, which is defined as the *X*–*Y* plane. Measurements are executed using a probe attached to the movable third axis, that is, *Z*-axis. As the probe touches the shaft surface, the CMM takes readings in six degrees of freedom. In such a way, a measuring point is generated with its three coordinates (*X*, *Y*, and *Z*). All collected points are analyzed via regression algorithms for the shaft constructive features. Usually, dimensional measurement capabilities of a CMM include position, perpendicularity, parallelism, angularity, profile of a surface, profile of a line, straightness, flatness, circularity, cylindricity, symmetry, concentricity, and total/circular runout.

3.10 Shaft Misalignment between Motor and Driven Machine

Shaft alignment has a strong impact on motor and driven machine operation and performance. Shaft misalignment, occurring as the centerlines of the motor and the driven machine shafts are not in line with each other, can result in excessive vibration, noise, and

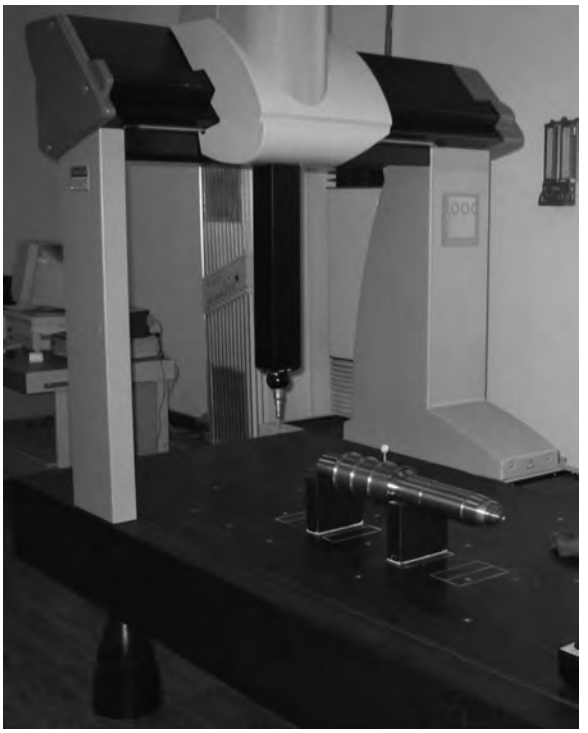


FIGURE 3.43
A coordinate measurement machine for measuring shaft geometric dimensions.

additional loads in motor and driven machine components, such as coupling, bearings, seals, shafts, and other machine components. It can lead to premature wear or even catastrophic failure of these components.

3.10.1 Types of Misalignment

Because the motor and driven machine are in 3D space, the misalignments between their shafts can exist in any direction. In actual applications, it is most convenient to break up this 3D space into the horizontal and vertical planes. As demonstrated in Figure 3.44, there are basically two types of motor misalignment:

- Angular misalignment can be further divided into the horizontal and vertical angular misalignments. Both of the misalignments occur when a motor is set at an angle to the driven machine in the horizontal or vertical direction, or both. It can be seen that from Figure 3.44a, as the centerlines of the shafts of the motor and driven machine are extended, the two centerlines cross each other with an angle. Angular misalignment can cause severe damage to motors and driven machines.
- Axial misalignment occurs when a motor has an axial distance with its mating machine (Figure 3.44b). This type of misalignment, sometimes referred to as *shaft end float*, often results from the dimensional variations due to the thermal expansion/contraction of the shafts. In practice, axial misalignment is easily measured and corrected compared to angular misalignment.

Hesai Exhibit 1031, page 246

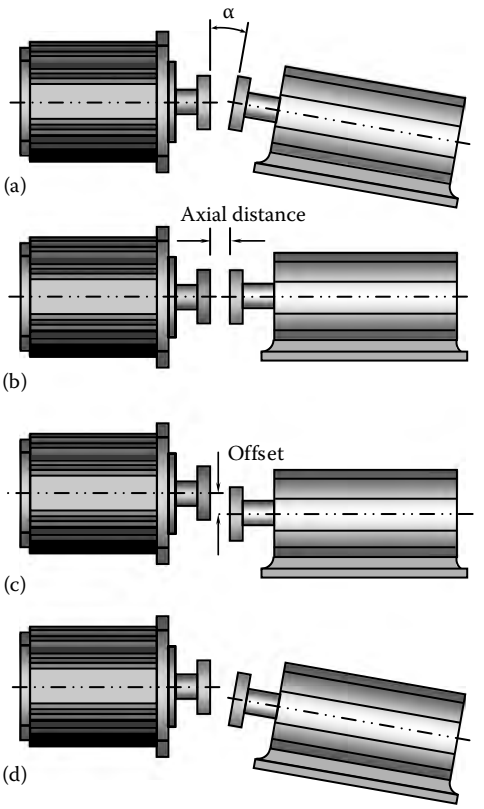


FIGURE 3.44
Misalignment of the motor shaft with the shaft of driven machine: (a) angular misalignment, (b) axial misalignment, (c) parallel misalignment, and (d) combination of angular, axial, and parallel misalignments. Only vertical misalignments are presented in this figure. Horizontal misalignments are in the similar manner.

- Similar to angular misalignment, parallel misalignment can be further divided into horizontal and vertical parallel misalignments. These two misalignments occur when the two shaft centerlines are parallel do not line up either horizontally or vertically or both (Figure 3.44c). Parallel misalignment may be resulted from manufacturing processes such as improper tolerances and shaft runouts, as well as poor installation practices.
- In practice, angular, axial, and parallel misalignments may occur simultaneously, as shown in Figure 3.44d.

3.10.2 Correction of Shaft Misalignment

Misalignments have long been recognized as one of the leading causes of electric machine damage and have been responsible for huge economic losses. In order to achieve best results in correcting shaft misalignment, it must take into account the machine sizes, shaft geometries, coupling type and condition, rotating speed, and operating temperature of both the motor and driven machine.

During motor normal operation, the alignment condition may change as the results of variations in a number of operating characteristics, such as operating temperature, transmitted torque, load, coupling, bearing play, and mounting conditions of both the motor and driven machine. Therefore, it is highly desired to measure motor-driving machine alignments under actual operating conditions (sometimes referred to as dynamic alignments).

There are good practices for measuring and correcting shaft misalignments [3.46]. The first step is to check alignments using appropriate tools. Modern laser alignment systems have been developed since the 1980s. Today, laser alignment becomes an essential component of a viable maintenance strategy for rotating machines. One of the notable features of laser alignment systems is their capability of performing real-time measurements to monitor the changes in machine's alignments.

As the misalignments have been detected, the next step is to follow suitable procedures to correct them. Soft foot is one of the most prevalent conditions found in shaft misalignments. Some laser alignment tools have a soft foot operation capability that guides engineers through the correction procedure. In practice, a set of precision shims is often used to bring the machines back into both vertical and horizontal alignment [3.47].

3.11 Shaft Coupling

The term coupling refers to a device that is used to connect two shafts together at their ends. The primary function of a motor coupling is to transmit torque or power from a motor shaft to the shaft of the driven machine. According to the connection pattern and relative motion, motor couplings can be briefly classified as rigid and flexible couplings.

3.11.1 Rigid and Semirigid Couplings

Rigid couplings are designed to draw two shafts together tightly so that no relative motion can occur between them. Rigid couplings are only applicable when the alignment of the two shafts can be maintained very accurately during motor operation lifetime. This type of coupling does not compensate for any misalignment between a motor and a driven equipment. If there is any significant angular, radial, or axial misalignment between them, additional loads (either bending moment, or tensional or axial stress) will be applied on the shafts of the motor and the driven equipment, as well as the coupling itself. The effects of these loads include premature bearing, seal or coupling failures, shaft cracking and breaking, and excessive radial and axial vibrations.

One of the most commonly used rigid couplings is a flange coupling, as shown in Figure 3.45. The motor shaft and the driven machine shaft are connected tightly through two flanges using bolt fasteners. Thus, in addition to tensile forces, the bolts are also subject to shear forces. This type of coupling uses a sleeve to align the two shafts.

A semirigid coupling is shown in Figure 3.46. This coupling is similar to the previous one, but there is no common sleeve shared between two shafts. As shown in the figure, the motor shaft and driven machine shaft are firmly assembled together with their hubs/flanges respectively. Each flange is bolted with a ring-shaped annular element which is securely fixed on the outer surface of the split tube. To avoid distortions of the split tubes under loads, the split tubes are supported internally with two reinforcing half-rings.

Hesai Exhibit 1031, page 248

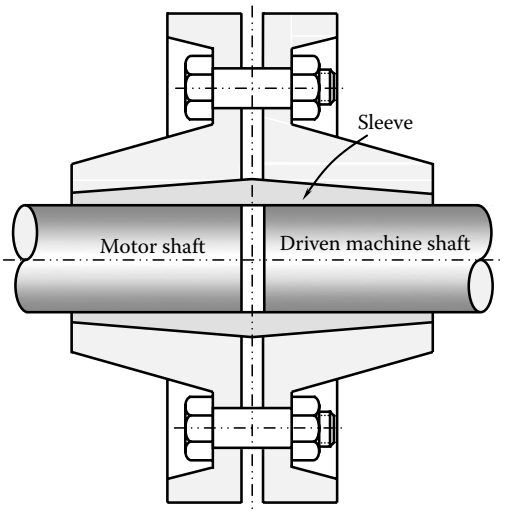


FIGURE 3.45
Flange rigid coupling. The sleeve is for aligning two shafts.

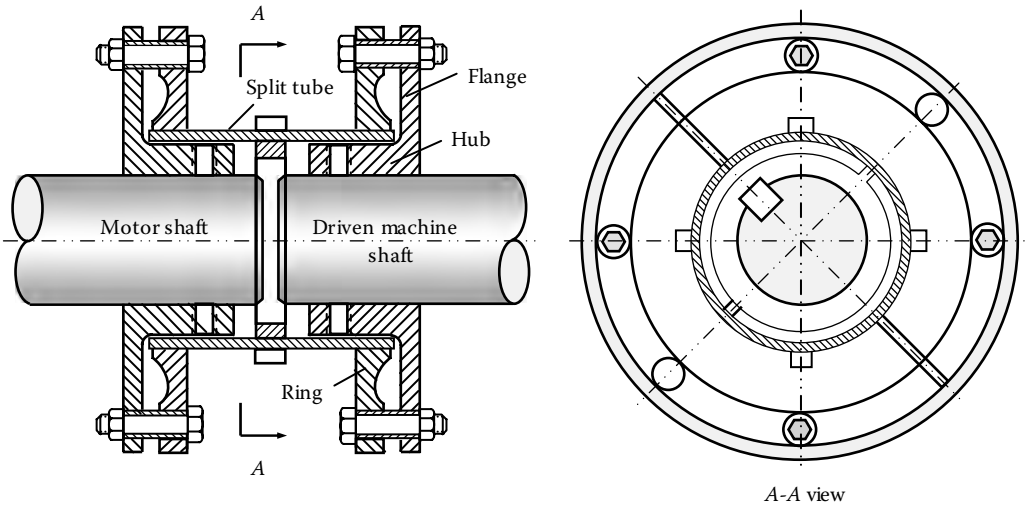


FIGURE 3.46
Semirigid coupling used in torque/power transmission. The motor shaft and driven machine shaft are indirectly connected together through two ring-shaped annular elements and split tubes for gaining a certain degree of flexibility.

This type of coupling can provide a certain degree of freedom to accommodate a small misalignment between two shafts.

In order to further increase the coupling flexibility, the two ring-shaped annular elements may be made of flexible elastomeric materials such as polyurethane elastomer, as suggested by Zilberman et al. [3.48]. However, one of the unfortunate consequences of increasing the coupling flexibility is a corresponding reduction in torque transmission

Hesai Exhibit 1031, page 249

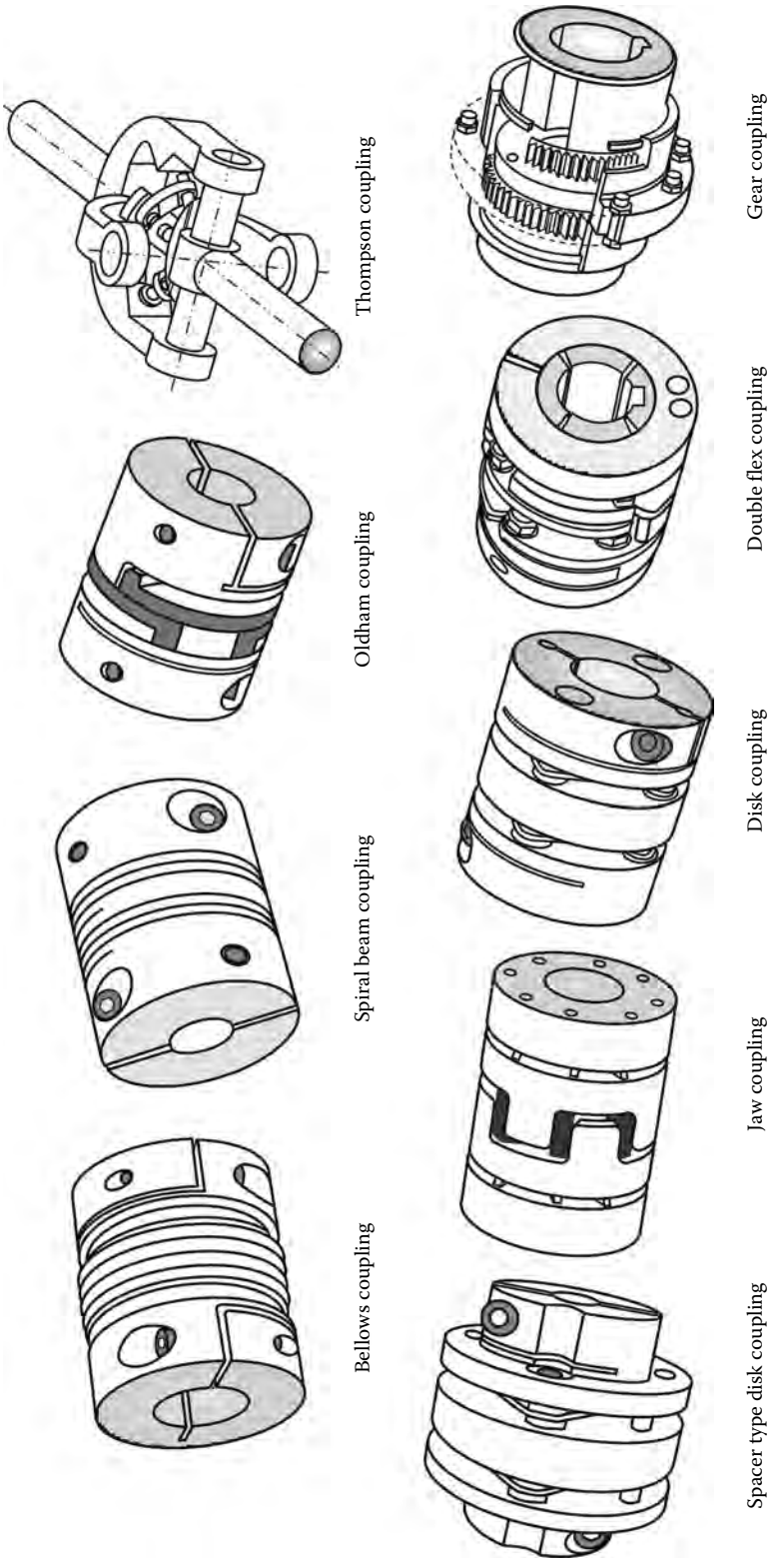


FIGURE 3.47
A variety of flexible couplings used in motor applications.

reliability. This indicates that such couplings may be restricted to use in relatively low-torque transmission applications.

3.11.2 Flexible Couplings

Flexible couplings are designed to transmit motor torque smoothly while accommodating minor misalignment in the axial, radial, or angular directions. The flexibility is such that when misalignments do occur, the coupling components move with little resistance. It is noted that with today's technology, flexible couplings are designed to tolerate more misalignments.

A large variety of flexible coupling types are available, as displayed in Figure 3.47. Each of them has its special features and is suitable for particular applications. Motor engineers need to trade-off their strengths and limitations to correctly select the coupling for the particular application.

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4

Stator Design

The stator is usually the outer body (or the inner body in some special cases) of an electric motor to house the driven windings on a laminated steel core for creating a rotating magnetic field. The stator core is made up of a stack of prepunched laminations assembled into a motor housing that is made of aluminum or cast iron or no separate housing designs. The inner surface of the stator is made up of a number of deep slots or grooves, distributed either uniformly or nonuniformly in the circumferential direction. It is into these slots that the stator windings are positioned. The arrangement of the windings within the stator determines the number of poles that the motor has.

4.1 Stator Lamination

The considerations in lamination design include the selection of the lamination material, optimization of the lamination stamping/cutting process, and determination of the lamination profile and pattern, dimension tolerances, core losses, grain orientation, unique shape requirements, and, of course, lamination manufacturing costs.

4.1.1 State Lamination Material

In practice, the base lamination material used for stators is fundamentally the same for rotors when it is desirable to punch the stator and rotor laminations simultaneously with the same die. However, some differences may exist between these two applications: (1) Because the main function of a stator is to generate a rotating magnetic field, the lamination material of the stator has higher requirements for electromagnetic properties. (2) Unlike a rotor that is subject to large centrifugal forces as it rotates, the forces acting on the stator are much lower, leading to lower requirements for mechanical strength properties. Based on these reasons, segmented cores are extensively used in stators for all types of motors but are restricted from use in high-speed rotors due to high centrifugal forces.

4.1.2 Stator Lamination Patterns

One of the factors that affect the efficiency of electric motors is the ability of steel laminations used in the stator to carry magnetic flux. There are several lamination patterns available for either enhancing the magnetic flux or simplifying the manufacturing processes, or both, depending on different applications.

4.1.2.1 One-Piece Lamination

Stators and rotors of electric machines are usually manufactured using a stack of one-piece laminations, made by punching the desired pattern from large insulated sheets of steel.

Hesai Exhibit 1031, page 254

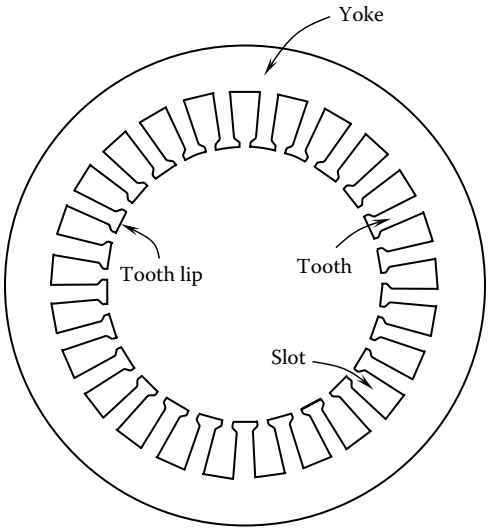


FIGURE 4.1
One-piece lamination is made by an undivided piece of steel sheet in full revolution. Stator slots are for receiving stator windings.

One-piece lamination is fabricated from an undivided piece of steel sheet and continuous in the 360° circumference, as shown in Figure 4.1. Obviously, the one-piece lamination method offers the advantage of fabrication and assembly simplicity. Multiple pieces are always used over the stator circumference and will be discussed in the following sections.

As illustrated in the figure, a one-piece lamination contains a number of teeth facing inward and connecting with the yoke. Slots are formed between adjacent teeth for receiving stator windings. The tooth tips, as well as the insulation sleeves or wedges, help keep the stator windings inside the slots. Tooth tips also have many electromagnetic functions.

4.1.2.2 T-Shaped Segmented Lamination

Segmented stator laminations are frequently encountered in large-size motors (Figure 4.2) wherein stamping one-piece laminations is no longer practical, either due to the limitation of the capability of stamping machines or due to the incredible high costs. For such cases, segmented laminations are the best choice for the stator core fabrication.

The recent trend toward using segmented laminations in small motors is a means to not only increase the slot-filling rate and facilitate automated fabrication of electric motors

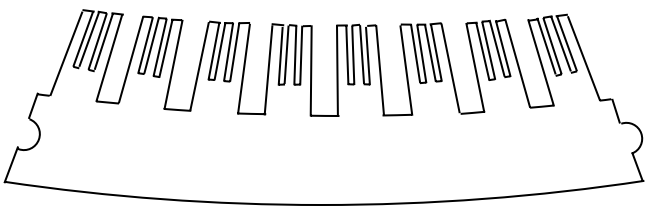


FIGURE 4.2
A segmented stator lamination containing a number of teeth for a large-size motor. The wide radial slots are used for receiving stator windings and the narrow radial slots for reducing eddy-current losses.

but also increase the continuous torque (well designed segmented lamination stators may increase motor torque by up to 50% over conventional motors). The segmented lamination technology is probably best suited to electric machines where the size of the effective air gap is insensitive to small variations, as in PM machines. It has been developed for high power density, cost-effective modern motors with better heat transfer within small overall packages, offering some benefits over traditional one-piece lamination stator cores [4.1,4.2]:

- The segmented design lowers the requirements for stamping machines and thus reduces both the machine and tooling costs. However, handling and assembly costs of the larger number of pieces to assemble increase or require automation tooling.
- The adoption of the segmented lamination technology can make a full use of the lamination material for significantly reducing the material consumption and material cost.
- Because each lamination section can be machine wound it allows placing more copper wires in stator slots. Generally, segmented lamination stators allow for 20%–30% higher slot-filling rate. This makes it able to use a larger diameter copper wire to fabricate the stator windings, thereby reducing the I^2R power loss.
- The segmented type stator may significantly increase in continuous motor torque over conventional integral type stator.
- The segmented lamination technology and single-tooth windings enable a motor to have significantly shortened end turns. This feature allows the motor to be much shorter, comparing with the long end turns found in conventionally random-wound servomotors.
- In manufacturing segmented stator coils, there are more choices in coil insertion methods. Many innovative ideas have been awarded patents over the past decade.
- Eddy-current loss is lowered due to the restriction of eddy-current paths. Thus, it could increase the motor efficiency. However, there are increases in electromagnetic ampere-turn drops due to the increase in mechanical gaps that can also reduce the total performance gain.
- T-shaped or I-shaped segmented laminations allow forming stator cores with different diameters to greatly save the tooling costs on very large motors. Again, one has to factor in the increase part handling and possible automation costs if employed on small motors.
- By taking advantages of the permeability of a grain-oriented material, the motor performance is correspondingly enhanced.

However, the adoption of the segmented core structure in electric machines comes with a penalty of increased core losses at the segment joints due to several factors, such as the degraded material conditions, the increased amount of punched edges with induced residual stress, and the effect of compressive stress applied on segmented stator cores. Furthermore, a potential cause of increased core losses may result from the increase in eddy current losses at the edge-to-edge butt joints [4.1]. Punching electrical steel drastically alters its magnetic properties near the cutting edges, resulting in the increase in core losses in two different ways [4.3]:

1. In a direct way, the cutting process creates residual stresses in the laminations at the vicinities of the cut edges, which can significantly increase the hysteresis losses and hence the total core losses at these locations.

2. In an indirect way, cutting can alter the magnetization profile inside the lamination. Due to the permeability drop, the polarization obtained with a given excitation is significantly reduced at the cut edge. Hence, in order to have the same flux across the sample (i.e., the same average polarization), a higher polarization in the bulk of the lamination is required. This higher polarization induces higher total losses.

Moses et al. [4.4] found that for a T-joint transformer core, the inner edges of the yoke and limb laminations are the highest power loss areas, up to 30% higher than the mean core loss. The third circulating harmonics causes additional losses calculated to be 20% of the nominal core loss [4.5].

Segmented laminations are made by punching one or several lamination teeth as an individually segmented piece. To save the tooling cost, all segmented laminations are usually designed to have an identical shape. As illustrated in Figure 4.3, T-shaped segmented laminations are jointed together by interlocking convex and concave portions to form a complete piece of a lamination. As all segmented lamination pieces are put into positions, the geometric pattern and magnetic circuit are approximately the same as for one-piece laminations. To enhance the motor performance, the grain orientation is arranged in the radial direction on all T-shaped segmented laminations.

A grain-oriented material has a superior performance in respect of less saturation and lower losses in the direction of the grain orientation. For this material, both core loss and permeability vary markedly, depending on the direction of the magnetic flux relative to the material rolling direction. To further take the advantage of the permeability and core loss of the grain-oriented material, it is to split the T-shaped segmented lamination into two pieces, as shown in Figure 4.4. Thus, by arranging the grain orientation in the radial direction on the tooth piece and in the circumstantial direction on the yoke piece, the stator performance is maximized.

A large number of patents are available on the segmented lamination design. Similar to the design in Figure 4.3, an innovative design that was awarded as a US patent [4.6] is presented in Figure 4.5. The lamination pieces are secured by means of the curved arc mating surfaces. A patented design is illustrated in Figure 4.6 [4.7], where both the teeth and yoke are made by segmented pieces. In Figure 4.7, the stator core consists of segmented large and small components, arranged alternately. Each stator slot is formed between a large and a small component, forming nonuniform slots in the circumference of the stator. It has been reported [4.8] that the periodicity of the cogging torque waveform for the motor with nonuniformly distributed slots is half that for the motor with uniformly distributed slots. However, the amplitude of the cogging torque is increased significantly due to the fact

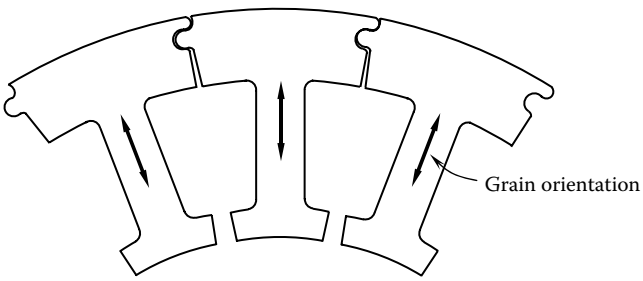


FIGURE 4.3
Segmented stator laminations, jointed with interlocking convex and concave portions.

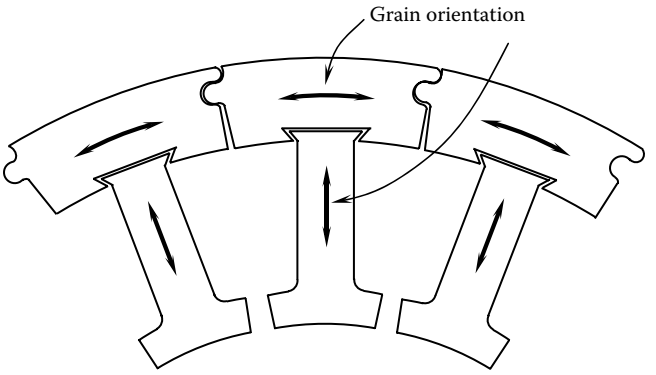


FIGURE 4.4
Grain-oriented segmented laminations.

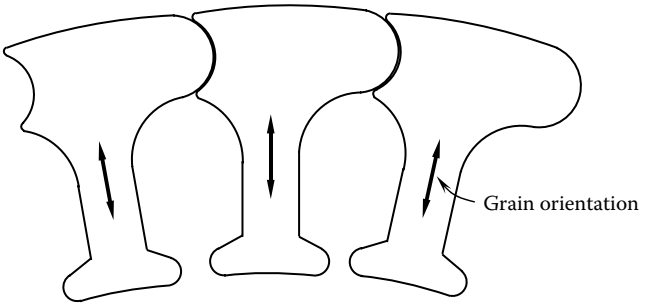


FIGURE 4.5
Patented segmented stator laminations with curved arc mating surfaces (U.S. Patent 5,212,419) [4.6]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

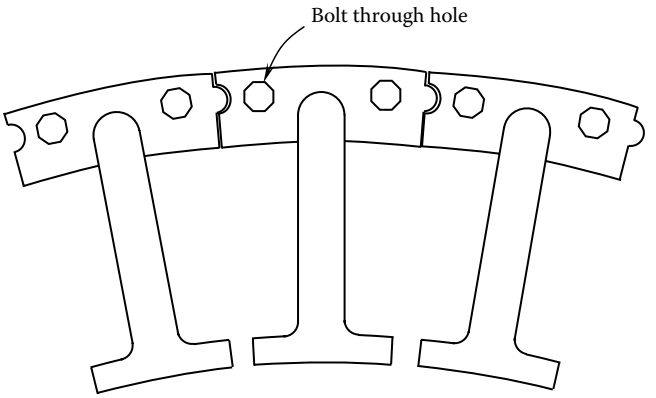


FIGURE 4.6
Patented segmented stator laminations in which both the tooth and yoke are segmented (U.S. Patent 7,816,830) [4.7]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 258

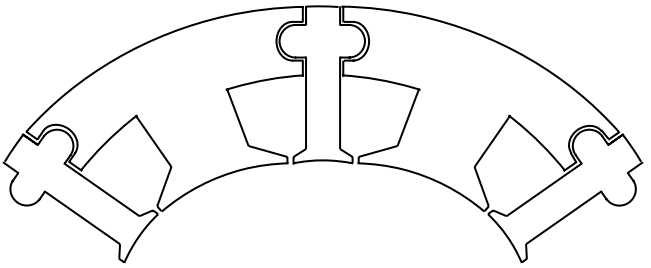


FIGURE 4.7
Alternative segmented stator laminations, showing each stator slot is formed between a large and a small segmented piece.

that the tooth tips of the wider teeth are approximately equal to the pole pitch in order to maximize the flux linkage per coil.

4.1.2.3 Connected Segmented Lamination

In order to increase the productivity of segmented laminations, the manufacturing process shown in Figure 4.3 can be improved. In this new process, instead of stamping individual segmented laminations, a long string of lamination pieces may be continuously stamped. These lamination pieces are connected together at the lamination outer tips (Figure 4.8). Then, these pieces are easy to roll together to form a complete lamination.

An alternative design of punched segmented laminations that connected together at the outer tips of the laminations is shown in Figure 4.9. The laminations are fabricated by stamping steel sheets in a progressive die. To reduce the bending force, circular edges and large semicircles are arranged at the tips of each lamination [4.9].

4.1.2.4 Two-Section Stator Lamination

In this design, a stator lamination is integrated by two lamination pieces: a section of teeth and a section of yoke (Figure 4.10). The stator windings are wound on the section of teeth. When all windings get in their positions, the section of yoke covers at the outer surface of the section of teeth, for housing the stator windings [4.10]. The section of teeth is positioned in the concave slots that are made at the yoke inner surface.

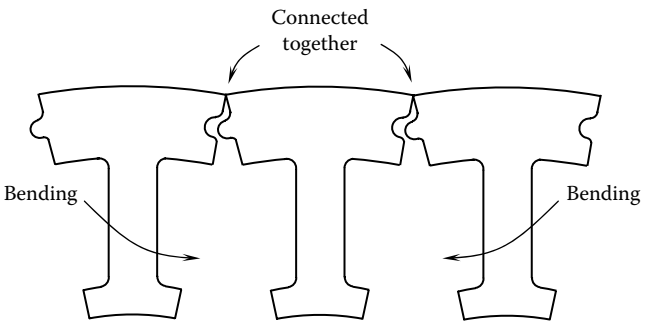


FIGURE 4.8
Punched segmented laminations are connected together at the outer tips of the laminations.

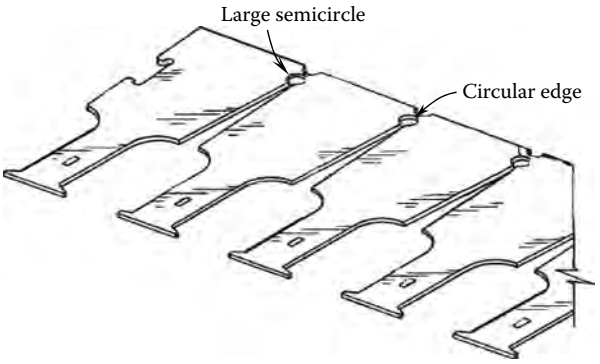


FIGURE 4.9
An alternative design of punched segmented laminations that connected together at the outer tips of the laminations (U.S. Patent 7,062,841) [4.9]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

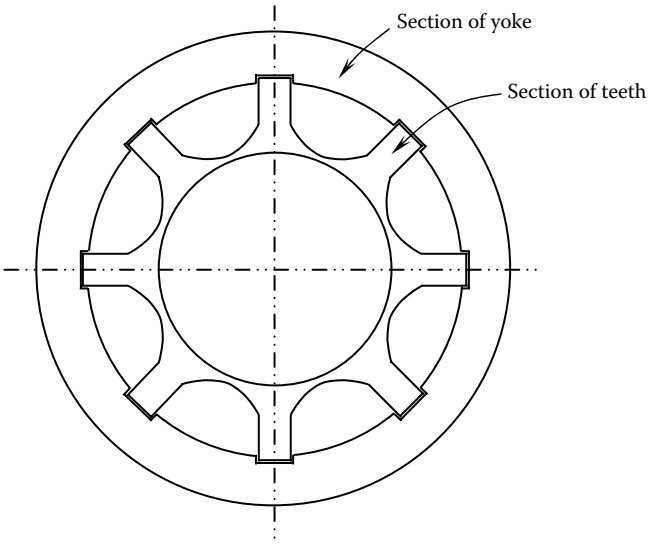


FIGURE 4.10
Integrated stator core with a section of teeth and a section of yoke; the section of teeth is positioned in the concave slots at the yoke inner surface.

For this design, the use of nonoriented materials, in which the magnetic properties are primarily the same in all directions, is more appropriate.

4.1.2.5 Stator Lamination Integrated by Individual Teeth and a Yoke Section

An alternative design of segmented laminated core is presented in Figure 4.11. It can be seen that all teeth are no longer connected together. Each tooth is individually positioned in the concave slot made at the inner surface of the yoke. Thus, this design can use the grain-oriented material to make the teeth, arranging the grain orientation in the radial direction.

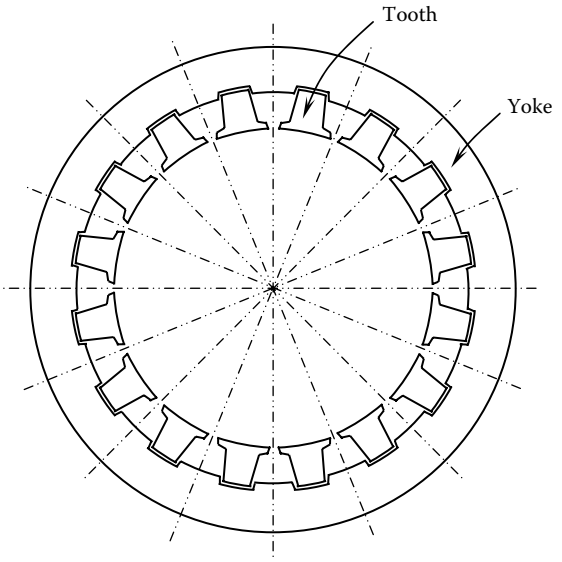


FIGURE 4.11
Integrated stator core with individual teeth and yoke; the teeth are positioned in the concave slots on the inner surface of the yoke.

The segmented lamination design in Figure 4.12 is similar to that in Figure 4.11 but with the opposite orientation of the yoke and the teeth. As a novel PM motor, the teeth are put outer forward and the yoke is located in the inner of the stator core [4.11]. In the assembly process, the sheets for a tooth are glued together to form a component, which are then glued onto the stator yoke (Figure 4.13). Rectangular-shaped concentrated fractional-pitch windings are inserted in the stator slots.

4.1.2.6 Slotless Stator Core

In brushless PM machines, cogging torque is generated from the interaction between the PMs on the rotor and the stator slots. Cogging torque can result in torque/speed ripples, rough running, vibration, and acoustic noise. Motor engineers always struggle with reducing cogging torque. As a result, slotless stator designs have emerged as a solution to zero cogging in conventional PM motors. The slotless stators totally eliminate the cogging torque, simplify the lamination process, and smooth the motor performance. In this type of motor, stator wires are wound into a cylindrical shape and assembled inside the stator core in a radial inward compressed condition [4.12]. Since the stator winding effectively functions as a spring pressure at the internal wall of the stator core, it is mechanically secured to the inner periphery of the stator core and a special fixation mechanism is not required. Then, the stator assembly is immersed in high-temperature epoxy resins and finally cured in an oven. Thus, the stator winding maintains its orientation with respect to the stator laminations and housing assembly (Figure 4.14).

However, the slotless winding requires increasing the air gap and in turn reduces PM excitation field. To maintain the same air gap flux density, the height of the PM must be increased. Therefore, slotless PM motors use more PM materials than slotted motors [4.13]. The cross-sectional view of the slotless stator is shown in Figure 4.15. Like all technologies, there are pros and cons to the differences.

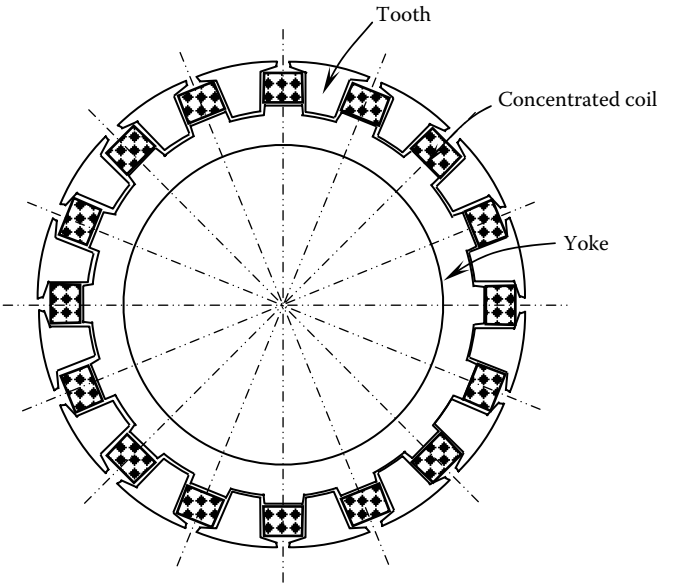


FIGURE 4.12
Integrated stator core with teeth and an internal yoke section; the teeth are positioned in the concave slots on the outer surface of the yoke, and the concentrated coils are inserted into the stator slots.

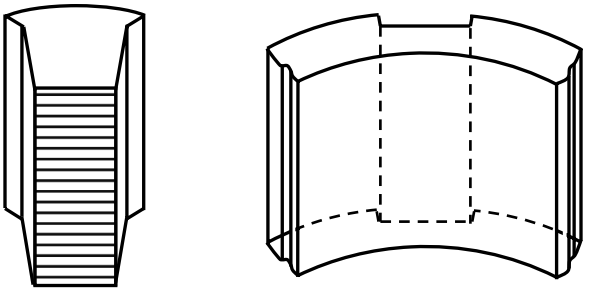


FIGURE 4.13
The sheets of teeth are held together as a subsection and then integrated with the yoke laminations.

4.1.2.7 Slinky Lamination Stator Core

Using new methods for fabricating stator cores is highly desired for both the cost reduction and manufacturing efficiency. One such design is known as the slinky method [4.14,4.15]. With this method, a stator core is built up from a continuous slotted strip of silicon steel rather than cross-sectional laminations in a conventional manufacturing process (Figure 4.16). The strip is wound edgewise in a helical configuration by a coiling machine that consists of three flanged rolls. By making the stacked strip to the desired thickness, the rolled core is compressed longitudinally and welded at the outer circumferential surface of the core, avoiding noise caused by vibration. The advantages of this manufacturing method include the following: (1) Using the slinky design provides the induced magnetic field with a preferable grain orientation in the rotor rotation, which would subsequently boost the overall efficiency of the motor. (2) It reduces the iron losses, noise generation, and

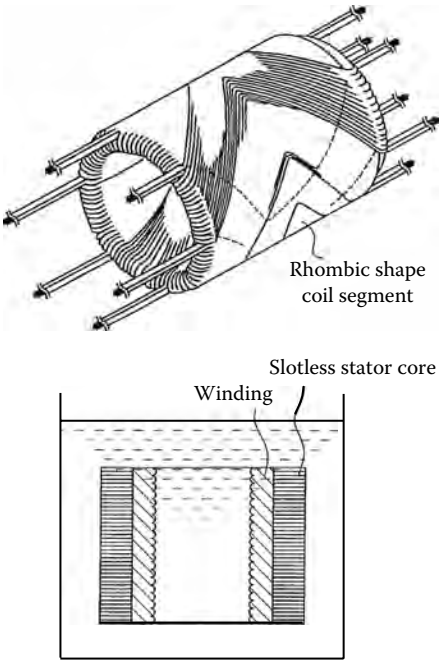


FIGURE 4.14
Fabrication of slotless stator (U.S. Patent 6,525,437) [4.12]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

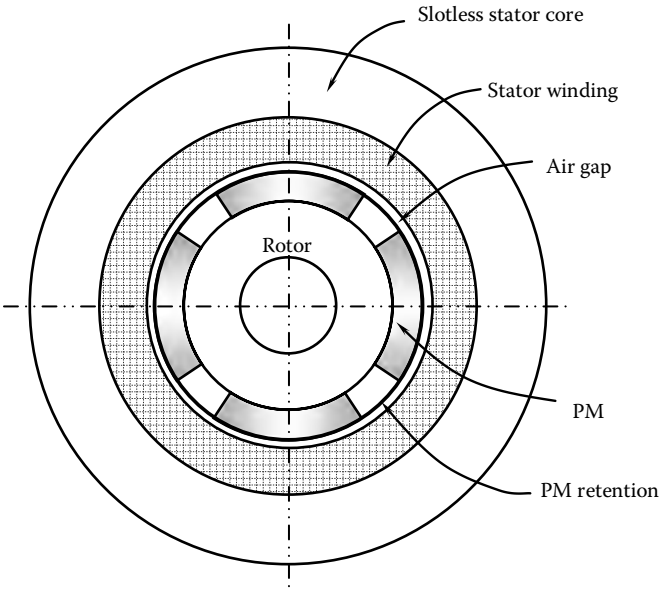


FIGURE 4.15
Structure of a four-pole slotless PM motor.

Hesai Exhibit 1031, page 263

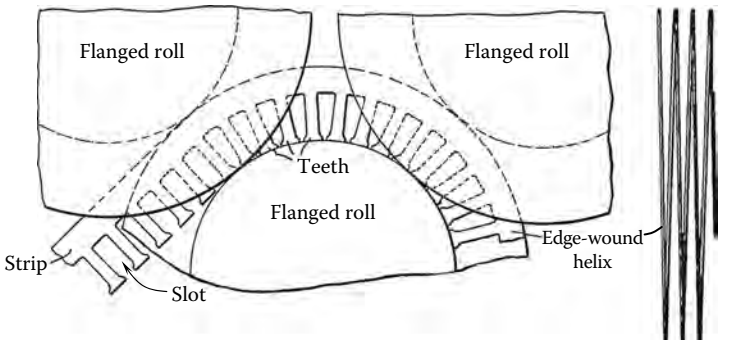


FIGURE 4.16
Fabrication of helical stator core with three flanged rolls (U.S. Patent 1,920,354) [4.11]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

power draw as compared to the conventional process. (3) This method can lead to great savings in material and winding labor.

This technique can be used in a slotless stator design. With the additional elimination of the teeth on the steel back iron, cogging of the motor is eliminated. It also simplifies the manufacturing process, as shown in Figure 4.17 [4.16].

In order to gain full advantages of core loss reductions from a laminated core, an adequate surface insulation must be maintained between the laminations. Stator laminations

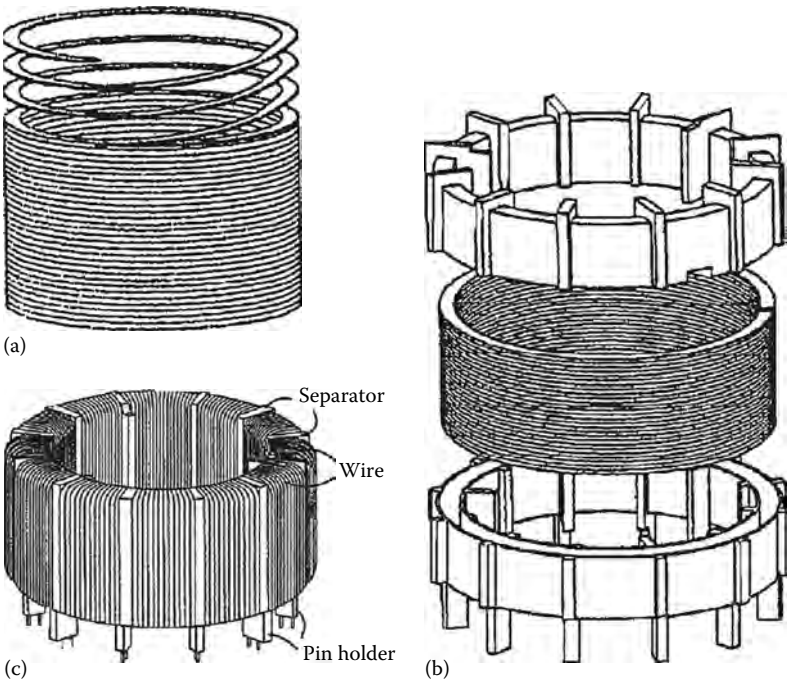


FIGURE 4.17
Using slinky method in designing slotless stator: (a) fabricating a helical stator core, (b) adding a molded plastic annular insulator at each end of the stator core, and (c) making the stator winding over the insulators (U.S. Patent Application 2005/0073210) [4.15]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

are usually coated with inorganic nonconductive materials as the insulation. The selection of coating materials is based on material properties especially electrical resistance, operation temperature, cost, and processing time. Furthermore, the application of a suitable coating can promote a better stamping performance such as low burr and long tool life.

4.2 Magnet Wire

The most common materials used for stator windings are copper and aluminum for their excellent electrical properties especially high electrical conductivity. By comparing with aluminum, copper has better electrical properties but higher melting temperature. At the temperature of 20°C, the conductivity of copper is 70% higher than that of aluminum. Though silver has the highest electrical conductivity (its conductivity is 5.7% higher than that of copper at 20°C) and highest thermal conductivity of any metal, its high cost and tarnishability have prevented it from being used for motor windings. Magnet wires are widely used in small and medium electric motors for fabricating stator windings. For heavy-duty large-size electric motors, electric conducting cables are more commonly used to make stator windings.

Magnet wire is coated with insulation materials (such as enamels) to allow winding wires to contact each other without forming electricity short circuits between them. The selection of the magnet wire is based on the motor operation conditions (working frequency, current, voltage, temperature, etc.), wire material properties, wire diameter for allowing maximum current density, thermal class, coating material, thickness of coating layer, and others. Magnet wire received its name undoubtedly from its use to make electromagnets.

4.2.1 Regular Magnet Wire

Regular magnet wire consists of a base metal, commonly copper or aluminum, and coated one- or multilayer of insulation materials, such as enamel, fibrous polyester, fiberglass yarn, and polyamide. For multilayer coatings, it is often necessary to use different compositions to achieve the most optimal insulation result. Several cross-sectional shapes of magnetic wires are available in stator windings, including round, square, and rectangular (Figure 4.18). The use of square wires can significantly increase the slot-filling rate. This is especially useful where space constraints are concerned.

4.2.2 Self-Adhesive Magnet Wire

As shown in Figure 4.19, a self-adhesive magnet wire has a thermoplastic adhesive film superimposed over standard film insulation. When activated by heat or solvent, the bond coating cements the winding turn-to-turn to create a self-supporting coil. This type of wires opens up new avenues for some special applications, especially where regular magnetic wires are not suitable. For instance, bobbin-free deflection yoke coils in television sets can be conveniently made by self-adhesive wires. With a coated thermoplastic cement on the outer surface of the wire, a strong turn-to-turn bond throughout a winding can be achieved, and thus the need for varnish impregnation can be eliminated.

The adhesive can be activated by different ways. An easy way is to allow the wire passing through a solvent while winding. As the solvent is evaporated, all turns are bonded

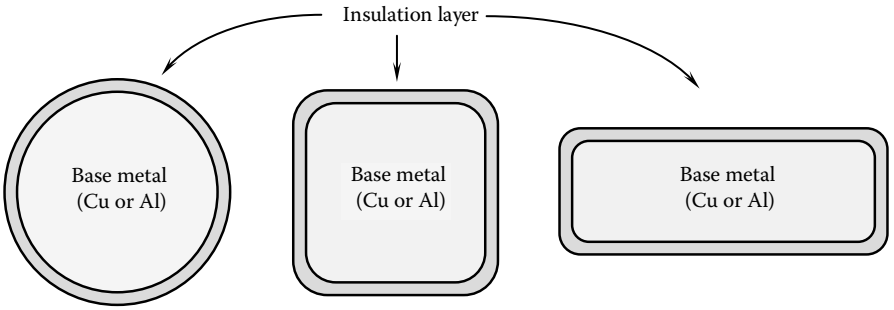


FIGURE 4.18
Modern magnet wires with different shapes.

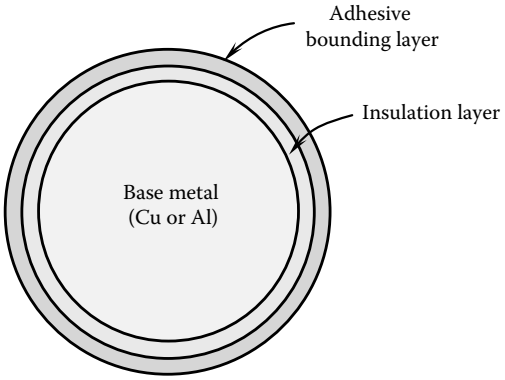


FIGURE 4.19
Self-adhesive magnet wire; an adhesive bounding layer is added on magnet wire.

together as the whole coil. Another effective way is called resistance heating, which is accomplished by passing electric current through the coil. Due to the resistance of the coil itself, heat is generated in the coil, and a desired temperature can be reached by controlling the heating time. When the desired temperature of the coil is reached, the coil is compressed so that the wires contact closely and thus bounded together. In fact, high resoftening temperature of cement allows the self-adhesive wires to compare with varnish-impregnated heavy-grade magnet wires.

As an example, the dimensions of a self-adhesive wire are presented in Table 4.1. It can be seen from the table that the thickness of both the insulation layer and the cement layer is proportional to the bare wire diameter. The thickness of the cement layer is about half of that of the insulation layer. This type of wires can withstand 180°C, which is well suited for electric motors.

The optimum bonding of the wires is reached when the coil temperature is raised to the range of 220°C–240°C, measured by the change in the coil electrical resistance.

4.2.3 Litz Wire

Litz wire is basically used for high-frequency applications. It contains many thin wire strands that are individually insulated and twisted together. Litz wire utilizes the full

Hesai Exhibit 1031, page 266

TABLE 4.1

Dimensions of Self-Adhesive Magnet Wires (Bondeze®M)

AWG Size	Nominal Bare Wire Diameter, mm (in.)	Minimum Insulation, mm (in.)	Minimum Cement, mm (in.)	Maximum OD, mm (in.)
15	1.4503 (0.0571)	0.0406 (0.0016)	0.0203 (0.0008)	1.5392 (0.0606)
16	1.2903 (0.0508)	0.0381 (0.0015)	0.0203 (0.0008)	1.3741 (0.0541)
17	1.1506 (0.0453)	0.0381 (0.0015)	0.0203 (0.0008)	1.2344 (0.0486)
18	1.0236 (0.0403)	0.0356 (0.0014)	0.0203 (0.0008)	1.1506 (0.0453)
19	0.9119 (0.0359)	0.0356 (0.0014)	0.0203 (0.0008)	0.9931 (0.0391)
20	0.8128 (0.0320)	0.0330 (0.0013)	0.0178 (0.0007)	0.8865 (0.0349)
21	0.7239 (0.0285)	0.0330 (0.0013)	0.0178 (0.0007)	0.7976 (0.0314)
22	0.6426 (0.0253)	0.0305 (0.0012)	0.0178 (0.0007)	0.7137 (0.0281)
23	0.5740 (0.0226)	0.0279 (0.0011)	0.0152 (0.0006)	0.6401 (0.0252)
24	0.5105 (0.0201)	0.0279 (0.0011)	0.0152 (0.0006)	0.5740 (0.0226)
25	0.4547 (0.0179)	0.0254 (0.0010)	0.0152 (0.0006)	0.5156 (0.0203)
26	0.4039 (0.0159)	0.0254 (0.0010)	0.0127 (0.0005)	0.4623 (0.0182)
27	0.3607 (0.0142)	0.0229 (0.0009)	0.0127 (0.0005)	0.4140 (0.0163)
28	0.3200 (0.0126)	0.0203 (0.0008)	0.0127 (0.0005)	0.3708 (0.0146)
29	0.2870 (0.0113)	0.0203 (0.0008)	0.0102 (0.0004)	0.3353 (0.0132)
30	0.2540 (0.0100)	0.0178 (0.0007)	0.0102 (0.0004)	0.2997 (0.0118)

cross-sectional area of the wire to carry current. Because each individual strand is thinner than the skin depth, the use of Litz wire can greatly minimize the skin effect and proximity effect losses in high-frequency windings.

A size of a Litz wire is often expressed in abbreviated format of *N*/*XX*, where *N* is the number of strands and *XX* is the AWG size of each strand. The twist per inch or meter is also an option to specify.

4.3 Stator Insulation

As magnet wires are wound into stator slots to form stator windings, electrical insulation is required wherever there is a difference of electric potential between two electric conductors to prevent the conductors from short circuiting to one another, including (1) one turn of a coil to adjacent turns, (2) coils to adjacent coils, (3) stator windings to the ground (i.e., the stator core), and (d) one phase winding to adjacent phase windings.

The insulation of stators can be done by various methods, for example, using aramid or mylar layered paper, fluidized-bed powder coating, or overmolding with thermoplastic materials. For low- or moderate-voltage stators, the magnet wire insulating varnish can usually satisfy the requirements of turn-to-turn insulation. However, as the stator voltage increases, an additional insulation becomes necessary. For instance, wrapping an insulating tape around the conducting wire can effectively enhance the dielectric strength. Similarly, for low-voltage stators, coil-to-coil insulation is provided by the wire insulation to avoid the risk of short circuiting in various series or parallel connected coils. The practical method for winding-to-ground insulation is usually to employ slot liners and slot wedges at the bottom and at the top of stator slots,



FIGURE 4.20
The damage of stator winding due to the failure of electric insulation.

respectively, to separate the stator windings to the adjacent lamination surfaces for preventing any part of the stator windings from shorting to the stator core. These are available in many shapes, sizes, and materials. Phase-to-phase insulation can be achieved by completely insulating the winding of one phase from those of adjacent phases. Failure of electric insulation can lead to the damage of the stator winding (Figure 4.20), causing permanent failure of the electric motor.

4.3.1 Injection Molded Plastic Insulation

Injection molded plastic insulation provides highly reliable insulating properties to laminated stator stacks. The use of molded plastic insulation prior to winding assures consistent and durable insulation of wound winding to the stator cores. Figure 4.21 is an example of the molded plastic insulation, which consists of a main body and an end cap. Using the molded plastic insulation can offer proven engineering advantages in the stator assembly, including consistent insulation thickness, reliable and long lasting insulation, and reduced labor time. The use of such insulation in small motors is presented in Figure 4.22.

A specially designed molded plastic insulator is applied at the motor end, providing the insulation for lead wires, as shown in Figure 4.23.

4.3.2 Slot Liner

As shown in Figure 4.24, heat-resistant and mechanically stable insulation papers or thermoplastic materials are inserted in stator slots for preventing coils from shorting to the stator core. The typical thickness of an insulation material may have a range of 0.1–0.65 mm, depending on liner materials and stator operation conditions. Although the material is inexpensive, it takes a longer labor time for fabricating stator windings.

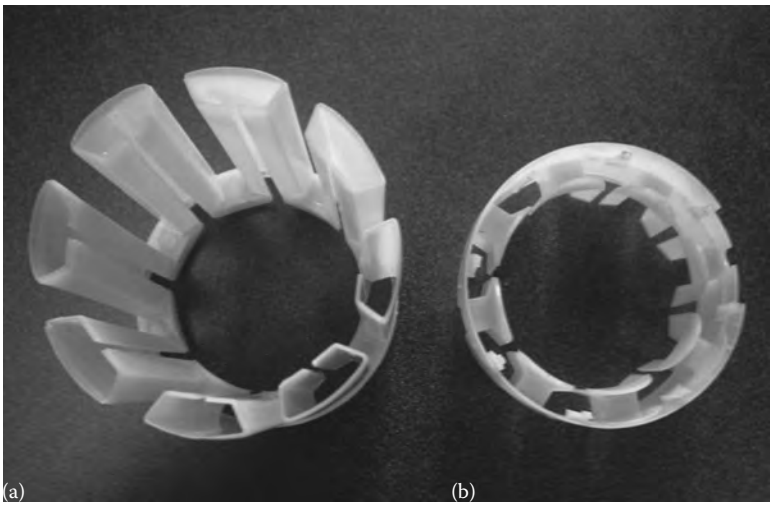


FIGURE 4.21
Molded plastic insulator: (a) main body; (b) end cap.



FIGURE 4.22
A variety of molded plastic insulators applied to stator windings.

4.3.3 Glass Fiber–Reinforced Mica Tape

For large-size *KV* motors, stator windings are often made of conducting bars that are continuously wrapped with mica tape. The taped winding bars are placed in a vacuum impregnation tank and flooded with a low-viscosity epoxy resin. After being completely impregnated, the insulation is cured at high temperatures in large chamber ovens.

Hesai Exhibit 1031, page 269



FIGURE 4.23
Specially designed molded plastic insulation part used at the motor ends for insulating the lead wires.

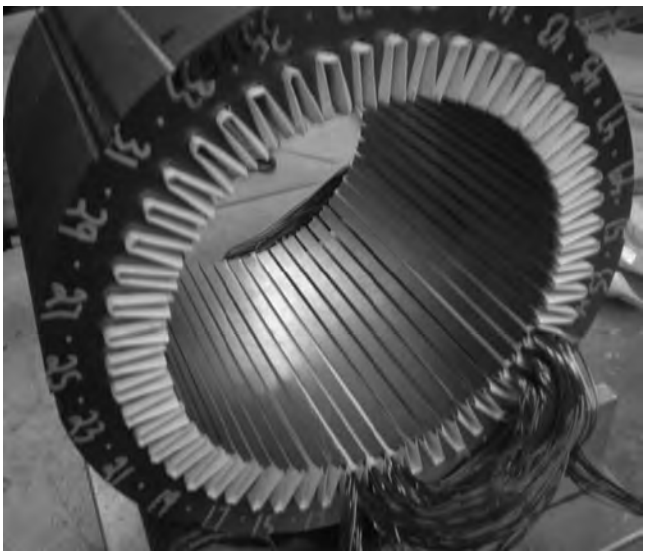


FIGURE 4.24
Insulation papers inserting in the stator slots for receiving winding coils.

4.3.4 Powder Coating on Stator Core

Powder coating technology (fluidized-bed coating) has been successfully applied in motor manufacturing since the early 1950s and is recognized as a superior and powerful method of applying a protective finish on electrical components. As an example, motor stators



FIGURE 4.25
A stator core having a coated insulation layer on the slot surfaces and partially end surfaces with powder coating technique.

are often shaped cylindrically with inwardly facing slots configured to receive stator windings. It is required to insulate the copper windings from the stator metal surfaces. This can be achieved by applying power coating techniques to provide uniform insulating coating layers on the stator slot surfaces, as well as partially on the stator end surfaces (Figure 4.25). There are a number of insulation materials suitable for powder coating such as epoxy, Glyptal®, and Loctite®. Epoxy powder coating has been effectively used as a high dielectric insulator on copper/aluminum conductors by many motor manufacturers for its high durability and superior dielectric strength. Numerous ways of distributing powder throughout motor parts are available. One approach that offers significant advantages over a wide range of other approaches is to use a specially designed apparatus that is equipped with a power fluidized bed to get uniform coating layers [4.17]. The thickness of the coating layer typically has a range of 0.1–0.5 mm.

4.4 Manufacturing Process of a Stator Core

The manufacturing process of a stator core includes lamination stamping, annealing, stacking, winding, and impregnating/encapsulating. The selection of stator core materials is similar to that of rotor cores. Since there are no centrifugal forces acting on stator cores, the requirement for material strengths can be lower than that of rotor cores.

In order to reduce eddy current and hysteresis losses, a stator core is manufactured by stacking a large number of thin laminations made of silicon steel. Without the lamination process, a solid steel core would become an electrical heater. The addition of silicon

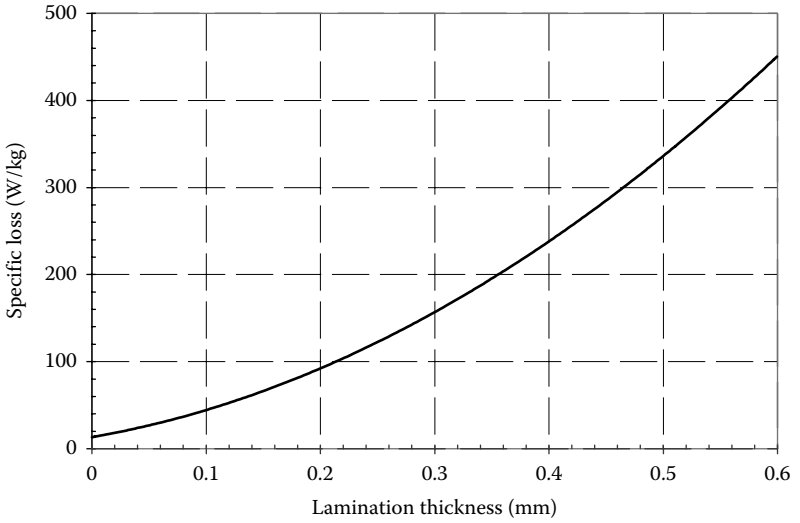


FIGURE 4.26
Specific iron loss in a PM motor versus lamination thickness at $B = 2T$ and $f = 1200$ Hz, showing that the reduction in lamination thickness can significantly reduce stator power loss.

to steel can significantly increase the permeability and volume resistivity to eddy current and thus enhance motor efficiency. It was shown that the iron loss of stator core can be reduced considerably by employing 6.5% silicon steel laminations rather than 3% silicon steel [4.18].

Silicon steel is available in an array of grades and thicknesses so that the material can be tailored for various applications. Generally, the thickness of motor laminations is in the range of 0.36 mm (0.014 in. for 29 gage) to 0.64 mm (0.025 in. for 24 gage). Thinner laminations are also available at 0.051 mm (0.002 in.), 0.102 mm (0.004 in.), and 0.178 mm (0.007 in.), considered for use in high-performance and high-frequency applications. It has been reported that at high flux densities and high frequencies, the specific iron loss is proportional to the square of the lamination thickness [4.19]. For the purpose of demonstration, the specific iron loss in a PM motor versus the lamination thickness for high saturated cobalt iron alloy at $B = 2T$ and $f = 1200$ Hz is presented in Figure 4.26. This indicates for a lower iron loss; the lamination thickness should be less than 0.1 mm. However, the determination of the lamination thickness is a trade-off between core loss and fabrication cost. Thinner laminations are always more expensive and have lower stacking efficiency.

Two major distinctions of laminated materials are the fully processed electrical steel and annealed electrical steel. The annealed electrical steels are most common in North America and have higher-performing electrical properties than the fully processed steels. A fully processed steel is usually a silicon electrical steel with an insulator coating and is not annealed after punching to save time and cost of the anneal process. When a lamination supplier is set up with high output flow through ovens, the anneal process can be down very economically.

4.4.1 Stator Lamination Cutting

Stamping is the most popular method used for lamination fabrication by most lamination manufacturers. The advantages of stamping include high productivity, low cost, and

simplicity of stamping process. The accuracy of the lamination cutting is typically controlled within ±0.0254 mm (±0.001 in.). However, burrs and warps near the lamination edges due to stamping can lead to difficulties in stacking and tolerance control. As discussed by Brown [4.20] for a high-speed stamping machine, these inaccuracies sometimes can be 50% or more of the design gap between laminations. In addition, burrs provide current paths between laminations and thus increase eddy-current losses.

Laser cutting is often used in special cases, such as prototype buildup, stators with extra large dimensions, or very complex geometries and shapes. Since the design information of lamination is computerized and loaded in a laser cutting machine, there is no need for tooling as in the stamping process.

In laser cutting, a high-energized laser beam is focused in a very tiny spot so that the local temperature rises extremely high to melt lamination sheets. Hence, this process generates much smaller residual stresses, lower distortion, better surface finishing, and high precise dimension control than the stamping process. However, laser cutting has lower productivity, higher power consumption, and higher costs. During a cutting process, some deterioration to the magnetic characteristics may occur, but it is too small to require annealing. This fabrication process is suitable for small quantities of laminations. The required tolerances for laser cutting are typically set at ±0.0254 mm (±0.001 in.).

Electrical discharge machining (EDM) is a manufacturing cutting process. An unwanted material is removed from the workpiece by a series of rapidly recurring current discharges between two electrodes—one is called the tool electrode and another is called the workpiece electrode. In a normal operation, the two electrodes are separated by a dielectric liquid. Similar to laser cutting, EDM can do a good job in cutting laminations but suffers its low productivity, excessive tool wear, and high power consumption.

Chemical etching is a very low-cost procedure that requires only a few chemicals (such as sodium persulfate and hydrochloric acid) and very little equipment. In practice, this method has been extensively used in manufacturing printed circuit boards (PCBs). The chemical etching process can generate burr-free and stress-free laminations that do not require annealing. The process starts to create a photographic image (*mask*) with the desired stator pattern. The mask is then illuminated with UV light onto lamination sheets. The lamination sheets need to be immediately placed into a developer solution such that the desired stator design is chemically etched onto the laminations.

4.4.2 Lamination Fabrication Process

An effective way to produce stator and rotor laminations together was presented by Isayama [4.21]. This method uses a sequence of stamping operation on strip materials to generate first the rotor sheet piece and then the stator sheet piece. As demonstrated in Figure 4.27, the stamping operation starts to punch pilot holes (Stage A) and then a shaft hole (Stage B), rotor ventilating holes (Stage C), and rotor teeth to complete the rotor lamination fabrication (Stage D). The following operation is for the stator lamination. The slots are made in Stage E. At Stage F, an annular punch is used to generate a concentric thin portion, located inside the ID end. In Stage G, round calking portions are formed. Following, stator teeth are punched (Stage H). In the last stage (Stage I), the stator lamination is blanked away from the metal sheet.

This method has numerous advantages over conventional methods, including (1) reducing the residual stress in stator and rotor laminations, (2) reducing the lamination deformation, (3) lowering production cost because all operations complete on a single strip material with the use of a single press machine and a single die machine, (4) enabling

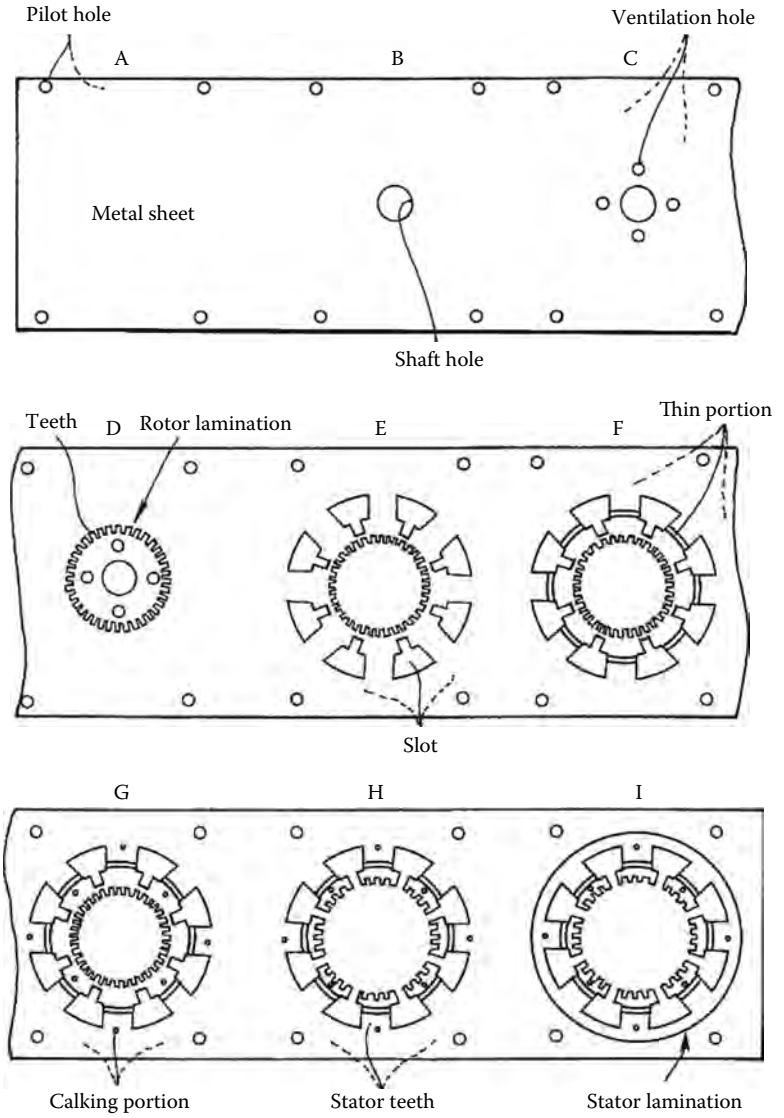


FIGURE 4.27
Rotor and stator laminations are made by a sequence of stamping operations (U.S. Patent 5,539,974) [4.21]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

to deal with complex shapes of stator and rotor, and (5) obtaining laminations that are excellent in such high shape precision as squareness and parallelism and highly suitable for use in accurate electric motors.

4.4.3 Lamination Annealing

During a lamination cutting process, residual stresses are introduced in processed laminations, leading to the degradation of material magnetic properties near the edges of the laminations and possible large cogging torque in motor operation. The smaller the stator

size, the larger the cut-affected zone relative to the whole lamination area. Therefore, annealing is a necessary step for stress relief and for optimum properties of laminations, especially for stamped small laminations. The annealing cycle requires to maintain processed laminations for several hours at the temperature range of 730°C–790°C under nonoxidizing and noncarburizing conditions (such as a vacuum or nitrogen-filled chamber) and then to naturally cool them to room temperature. Fully processed steels skip this step and perform very satisfactorily, but the highest performance is achieved with the anneal cycle.

4.4.4 Lamination Stacking

With the development of industrial automation, automatic lamination stacking becomes more popular today for modern motor manufacturers. This increases production efficiency and reduces the requirement for the storage capacity, especially for a rotary stacking.

A stator stack is formed by stacking laminations into a pack. There are several methods to assemble stack laminations into stator cores [4.22]:

- Welding—This method can robustly join laminations together to withstand large forces without lamination separation (Figure 4.28). However, welding across laminations may increase local stator core losses to impact motor performance, particularly for small motors. Designing geometry for welding can eliminate or minimize the shorting concerns. If improperly designed or welded, an increase in cogging torque can occur. Therefore, in order to avoid this undesired result, it must keep welds out of critical flux return path locations.
- Bonding with adhesive materials—This method is to deposit a thin film of adhesive on lamination surfaces to be bonded and assemble the laminations into the stator core. The core and winding are reheated to cure the adhesive material. Thus, it has bounded together the laminations of the core to provide adequate



FIGURE 4.28
Stator laminations are integrated by welding.

Hesai Exhibit 1031, page 275

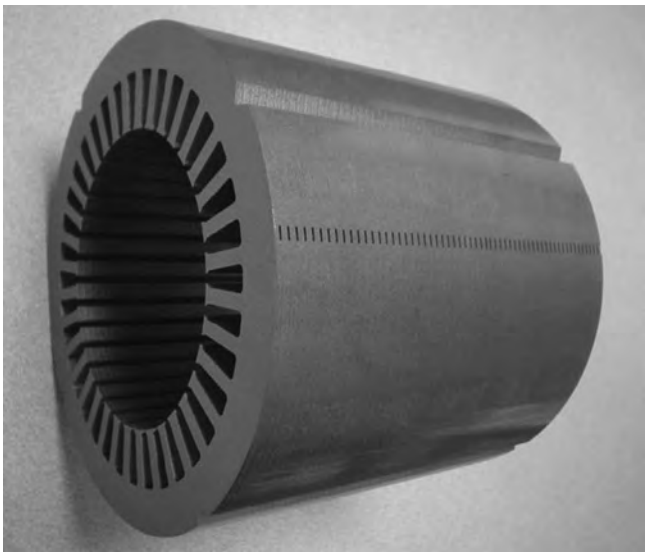


FIGURE 4.29
Stator core is fabricated using spot bonding technique with epoxy adhesive.

core strength (Figure 4.29). By comparing with other methods such as welding and riveting, this method provides the relatively low mechanical strength of the stack.

- Riveting—This is an economical and reliable method for securing stator cores. It requires punching small holes or slots on laminations. However, the heads of the rivets may obstruct the stator end windings.
- Fastened by pins—This method is similar to riveting but without the problem of the obstruction of the rivet heads to the stator end windings.
- Lamination interlocking—This is a convenient way of assembling a stack. As can be seen in Figure 4.30, 10 dimples are placed at the back iron on each stator lamination. When the laminations are stacked together, they nest inside each other. The main disadvantage of this process is that it requires a complex die that is very expensive.
- Self-cleating—This method rolls V-shaped strips and then flattens them down into dovetail slots at the OD of the stator laminations.
- Using slot liners—This method is commonly used for stator cores less than 150 mm. Cuffed slot liners hold the stator core together, as described in [4.23].
- Using thin sleeves—As demonstrated in Figure 4.31, this method for the stator core retention is to use thin metallic sleeves outside of the core. Due to high strength of the sleeve, the stator core can withstand large forces with only tiny deformation. The installation of the sleeve usually relies on the shrink fit technique.
- Bolting—This method is similar to riveting for providing reliable lamination integration. A plurality of bolts is extended through boltholes in the lamination stack and engaged with nuts. Upon tightening of the bolds, the laminations are forced together to establish a generally axial compression on the lamination stack. However, bold heads have even larger heads than rivets to obstruct the stator end windings.

Hesai Exhibit 1031, page 276

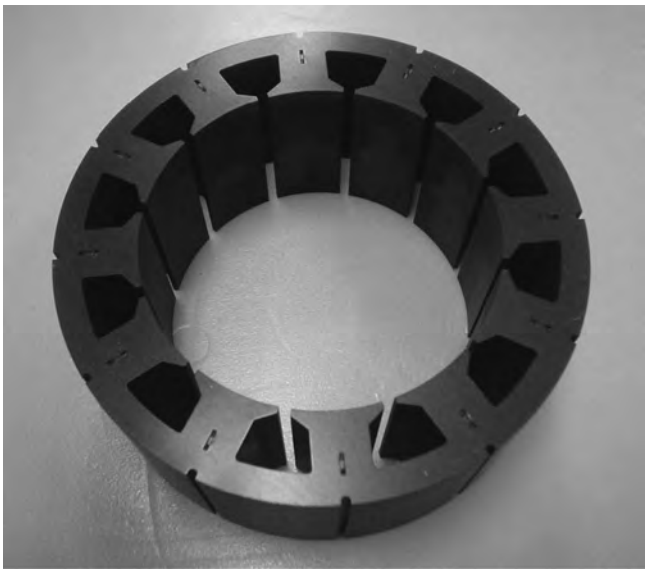


FIGURE 4.30
Stator core is made from dimpled laminations.

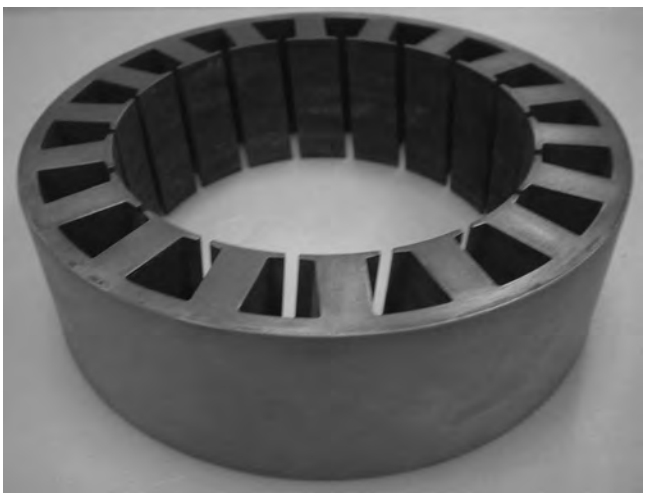


FIGURE 4.31
Stator core is secured with a thin sleeve at the outside of the core.

4.4.5 Stator Winding

One of the important parameters in stator winding is the slot fill ratio, which is defined as the percentage of the space occupied by magnet wires to the total available space of the slot. In order to lower the wire resistive loss and increase the power density, it is highly preferable to have the maximum copper fill, that is, maximum slot fill ratio.

A winding end turn refers to the amount of the winding extending beyond each end of the stator's magnetic core structure. Though the end turns are necessary to complete the

electrical path within the winding, they contribute little to the motor torque output. Motor torque is only generated by the winding that lies within the stator's magnetic core structure. Consequently, it is highly desired to minimize the length of the winding end turns. This can not only save the wiring material and lower the material cost but also reduce the copper loss and increase the motor efficiency. The shorter the winding stack length, the greater the impact of the end turn length on motor efficiency.

4.4.5.1 Random Winding by Hand

The advantages of random winding by hand include the flexibility and suitability to some manufacturing processes, such as prototypes, low-volume production, and/or motor rewinding. Slot fill can also be higher than single slot designs wound with typical needle winding machines. However, this winding method may have more winding variabilities and require highly skilled operators. Hand wiring can have different configurations and is best to clarify if (1) coils are hand wound and hand inserted, (2) coils are wound around stator teeth by hand, and (3) coils are machine wound and hand inserted.

4.4.5.2 Coil Formation: Distributed Winding

This type of winding is made by arranging winding turns in several full-pitch or fractional-pitch coil. Then, this coil is housed in the slots spread around the air gap periphery. Because a distributed overlapping winding generally results in a more sinusoidal MMF distribution and EMF waveform, it is used extensively in PM brushless AC machines [4.24]. High slot fills can be achieved depending upon the motor size. Generally, the larger the motor, the higher the slot fill.

4.4.5.3 Coil Formation: Concentrated Winding

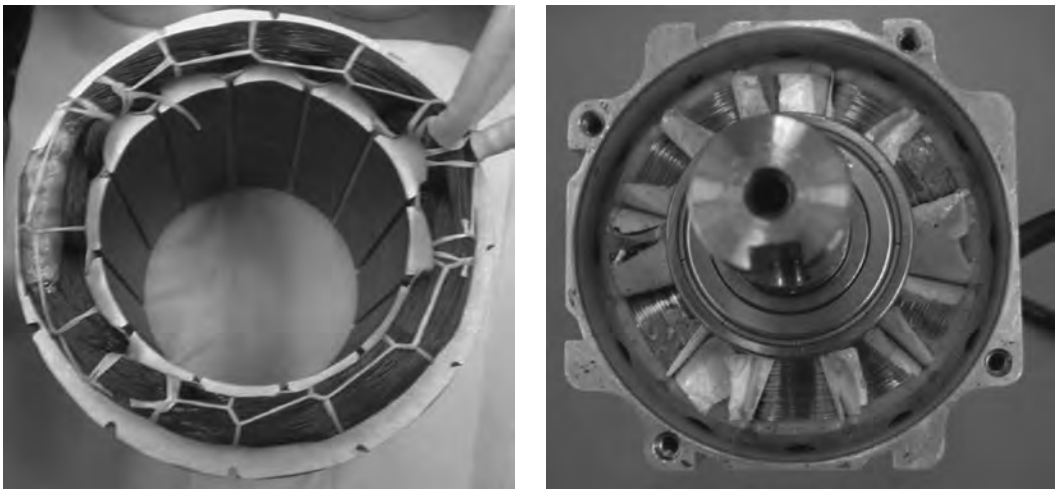
The interest in motors with concentrated windings is growing due to their short end windings and simple winding structures that are highly desired for high-volume automated manufacturing. For this type of windings, all the winding turns are wound together in series to form one multiturn coil. The concentrated winding motors have potentially more compact designs compared to conventional motors with distributed windings.

It has been reported that based on FEM calculations, the performance of an electric machine with concentrated winding is superior, because the minimization of the copper volume greatly reduces power losses and manufacturing costs. Induced EMF in concentrated windings is greater than distributed windings. In addition, harmonic or noise in distributed windings is lesser than concentrated windings. Furthermore, distributed windings have less armature reaction. The distributed and concentrated windings are given in Figure 4.32.

4.4.5.4 Coil Formation: Conductor Bar

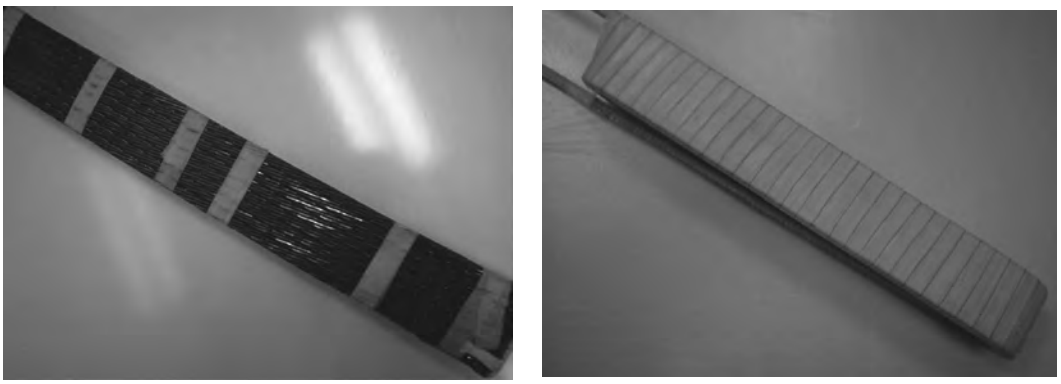
In building large-size motors and generators, it is often necessary to bundle several individual conductors that are insulated against one another to form a so-called conductor bar (Figure 4.33). The insulation between the conductor bar and stator core is defined as the main insulation. To fabricate stator windings, several conductor bars are connected to each other at the machine ends, as shown in Figure 4.34.

Conductor bars are prefabricated with either round or rectangular shape copper wires, coated with mica tape and impregnated with epoxy resin. They are then heated up and



(a) (b)

FIGURE 4.32
Coil formation: (a) distributed winding; (b) concentrated winding.



(a) (b)

FIGURE 4.33
Bundled conducting bars used in electric motors: (a) rectangular magnetic wires are twisted to reduce eddy-current loss and (b) stator winding is wrapped by an insulating strip.

cured resulting in high winding mechanical strength. Generally, a conductor bar with a rectangular cross section is preferred for its high filling rate in stator slots. In producing conductor bars, the individual conductor can be either twisted around each other or maintained parallel each other. During a manufacturing process, sometimes, conductor bars are required to be compressed to make them into a desired shape and compact size. It has been found that an improper compressing method may potentially damage the conductor's insulation, causing stator winding failure. The connection efforts are more labor intensive with individual coils.

Before and after the winding impregnation, stators must undergo a series of tests such as hipot (dielectric voltage withstand test), surge (impulse test), resistance, and contact resistance to verify the stator condition.

Hesai Exhibit 1031, page 279

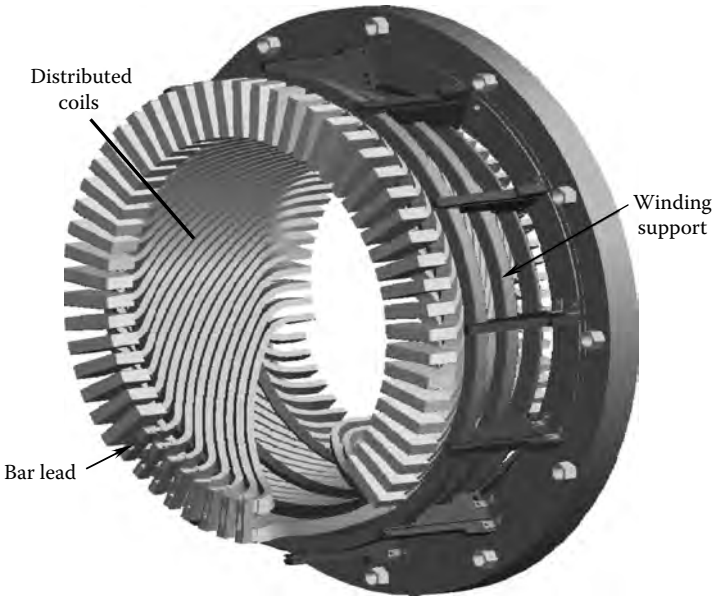


FIGURE 4.34
Conducting bars are commonly used in large electric machines. It shows the joints of conducting bars at the end turn region.

4.5 Stator Encapsulation and Impregnation

Encapsulating and impregnating stators can strengthen stator winding electrical insulation, provide reliable protection to chemicals and harsh environments, enhance thermal dissipation, promote stator structure integrity, and stabilize motor operation.

4.5.1 Encapsulation

Encapsulation has been widely used in electric machines. According to different applications, electric motors can be either partially or entirely encapsulated. Partial encapsulation is basically applied to the stator end windings for integrating them with other stator components against vibration and for enhancing heat transfer. Entire encapsulation is applied to the whole stator assembly for achieving better protection of the stator from moisture, dirt, debris, and erosions caused by chemicals such as acids. This treatment can provide very robust, durable, and long-life stators.

4.5.1.1 Encapsulation Materials

A wide variety of encapsulation materials are available for electric machines, as given in Table 4.2. These materials usually exhibit high dielectric strengths and relatively high thermal conductivity and are developed for different applications.

Thermoset plastics (also known as thermosets), such as epoxies, phenolics, and thermoset polyesters, have long been applied to electric machines as encapsulation materials. They can form irreversible chemical bonds during the curing process and have good

TABLE 4.2

Encapsulation Materials

Encapsulation Material	Thermal Conductivity (W/m-K)	Dielectric Strength (V/mil)	CTE (10^{-6}/K) (%)	Note	Reference
E88 epoxy	1.049 (at 23°C)	780	3.49	A two-component flame retardant resin system used for the impregnation and encapsulation of high-voltage electrical components.	[4.25]
C89 hardener	1.069 (at 50°C) 1.083 (at 75°C) 1.074 (at 100°C) 1.043 (at 125°C) 1.004 (at 140°C)				
Araldite CW 229-3 Aradur HW 229-1	0.75	508	27	It is a prefilled resin system that provides high crack and thermal shock resistance.	[4.26]
Altherm XB 2710 Aradur XB 2711	1.5	508	24	It is a good choice for stator encapsulation for its high thermal conductivity.	
Araldite XB 2252 Aradur XB 2253	0.7	508	60	The system is mineral filled with excellent flowability and impregnation capability.	
Araldite CW 1312 Aradur HY 1300	1.1	381	100	It has higher thermal conductivity and adequate impregnation capability. High filler loading results in higher viscosity.	
Arathane CW 5631 Arathane HY 5610	0.6	508	70	The curing reaction is fast and exothermal at room temperature. It is easy to process and has good impregnation capability.	
Aratherm CW 2731	3.0		20	One-component product. Developed for end turn encapsulation of motors and generators.	

Hesai Exhibit 1031, page 281

Catalyst 11	1.28	380	97.9[a]	It is a two-component, thermally conductive epoxy encapsulant. It features a low CTE and excellent electrical insulative properties.	[4.27]
EP 234	3.77	415		It is room temperature cured and designed for medium-voltage devices.	
EP 1121	0.14	440		Its low viscosity allows for good wicking and penetration into components and circuitry and also gives good air release.	[4.28]
EP 1282	0.14	440		EP 1282 is a two-part unfilled electronic grade epoxy encapsulant. It cures at room temperature to a tough, semirigid polymer.	
EP 1285	1–1.27	365–400		Designed for applications requiring a high degree of thermal conductivity and a low CTE combined with a moderate free-flowing viscosity.	
Thermoset® SC-320 silicone	3.2			A two-component silicone elastomer that provides excellent thermal conductivity, high thermal shock resistance, and improved coefficient of expansion.	[4.29]

[a] Samples tested were cured for 24 h at 25°C.

Hesai Exhibit 1031, page 282

mechanical and thermal properties. In recent years, with the recognitions by IEC-85 and UL-1446, thermoplastics as encapsulation materials are used for some encapsulated motors, solenoids, and transformers. Thermoplastics offer some performance benefits such as high strength and shrink resistance.

Though these two types of encapsulation materials sound similar, they have very different properties and applications. One of the remarkable differences is that the volatile organic solvents used in thermoset plastic encapsulation processes are eliminated in thermoplastic processes. As a result, the harmful solvent emissions to the environment are eliminated. Another important characteristic feature is that the curing process of thermoplastics is completely reversible as no chemical bonding takes place. This indicates that thermoplastics can be remelted at high temperatures. This special characteristic feature allows thermoplastics to be remolded and recycled but restricts their use for relatively low-temperature applications.

4.5.1.2 Encapsulation Process

In practical operation, prior to the encapsulation process, the stator is preheated in an oven to drive off any moisture or volatile components.

Thermally conductive thermosets are ideally suited for the full stator encapsulation due to their outstanding electrical and thermal properties and mold flow characteristics [4.30]. Among them, epoxy resins are most popular in the motor industry. They are generally hard and tough, exhibit low shrinkage on cure, and offer superior chemical and environmental resistance. The key features of encapsulating materials include high heat conductivity, high dielectric strength, and low shrinkage, thermal endurance, and chemical resistance. It is worth to note that thermal conductivity in an encapsulation system is determined by all its components, such as epoxy resin, hardener, and filler. Generally, thermal conductivity increases as more fillers are added. However, a high filler volume may significantly increase the viscosity of the mix, making it much more difficult to fill tight spaces. One way to get high filler volume in an encapsulation system is to use fillers with a spherical shape; angular fillers require much resin between them to maintain the necessary liquidity of the encapsulating materials.

Epoxy resins usually contain diluents or reducers to help their fluidity during the process. The hardener used with the epoxy resin plays a very important role for the final properties and has an impact on the speed of cure. Many epoxy resin systems use solid fillers as the constituent for cost reduction and increased hardness and stiffness of the cured product. Moreover, by carefully choosing the solid filler materials, it may lead to the increase in thermal conductivity in the cured encapsulation. It was reported that the types of filler that can help increase thermal conductivity include ceramic material, glass fiber, Kevlar® fiber, carbon fiber, silicon nitride fiber, and other materials [4.31]. The filler materials can be in the form of particles, granular powder, whiskers, fibers, or any other suitable forms. The thermally conductive polymer composition used to form the encapsulating polymer layer for the stator assembly comprises a base polymer and a thermally conductive filler material and has a thermal conductivity of at least 0.6 W/(m-K) and up to 5 W/(m-K) or higher.

Different encapsulation processes are available such as vacuum casting, gravity casting, and pressure gelation to allow highly automated production lines with short cycle time. Prior to casting, a sealing core is placed in the middle of the stator for preventing an encapsulant contamination of stator laminations.

A stator encapsulation process starts with the heating of an encapsulant to a certain temperature for the purposes of degassing and increasing the fluidity of the encapsulant. The higher the fluidity of the encapsulant, the shorter the filling time. With a gravity casting technique, liquid encapsulant is induced at the bottom of the stator to avoid air bubbles trapped in the encapsulation. Then, the stator is put in the oven for curing. The curing temperature and time can be optimized to minimize volume shrinkage and maximize mechanical strengths.

A standard casting process is typically controlled and executed by an encapsulating equipment, which consists of two independent systems for dealing with the encapsulant (e.g., epoxy resin) and the hardener individually before they are mixed. The epoxy resin is stored separately with the hardener in its packaging drum and goes to the vacuum premixers for degassing and the metering mixer for further mixing. This is the same process for the hardener. After this step, both materials get in the static mixer to get them to be mixed together. The well-mixed epoxy resin is pumped to fill the mold.

The cutaway view of an encapsulated stator shows that the epoxy completely penetrated into the stator coils (Figure 4.35).

The entire stator assemblies are immersed into a liquid epoxy resin in a vacuum tank. The vacuum helps to not only completely fill voids in the windings but also exhaust moisture and other vapors from the windings.

4.5.2 Varnish Dipping

Varnishing dipping is an effective way of securing stator windings. In this method, a stator is immersed into an open varnish tank. After a certain time, the stator is removed from



FIGURE 4.35
A cutaway view of an encapsulated stator shows that epoxy penetrates into stator coils and forms void-free encapsulation.

Hesai Exhibit 1031, page 284

the varnish tank to allow excess varnish dipping. Then, place the stator in an oven to dry off solvent.

However, during varnish dipping and baking processes, solvent is continuously evaporated and released into the environment air, causing air pollution problems.

4.5.3 Trickle Impregnation

Trickle impregnation is extensively used in many motor manufacturers. In practice, a trickle varnish machine is usually used to impregnate the stator winding with a varnish that rigidly secures the wires. Prior to the impregnation process, stators need to be preheated to the desired temperature. A thin stream of impregnating resin is poured/dripped onto the winding. In such a way, the impregnating resin fills the voids of the winding and gets polymerized in a short time. This method offers several advantages, including short processing time, high retention, no need of postoven curing, and consistent quality of impregnation.

The varnish improves heat transfer within the winding and between its surrounding magnetic core structure. This improves motor cooling and in turn increases the motor's continuous torque and power density. In addition, trickle impregnation helps the stator winding to withstand vibrations, avoiding motor failure due to the damage of wire insulation.

The utilization of a thermally conductive epoxy instead of a regular varnish to impregnate the stator winding can enhance motor cooling remarkably, for example, using a potting epoxy that is a recognized component in a UL-1446 insulation system that has a Class H (180°C) temperature rating. Actual measurement shows that potting the stator winding using this thermally conductive epoxy lowers its winding-to-ambient thermal resistance, R_{th} (in °C/W), by 50% compared to impregnating the winding with a typical varnish [4.32].

4.5.4 Vacuum Pressure Impregnation

The vacuum pressure impregnation (VPI) technology has been extensively used in a wide range of applications in the electric machine manufacturing industry from insulating coil windings to sealing porous metal castings. The characteristic of this technology is to use a VPI tank (Figure 4.36) that is vacuumed first and then pressurized to achieve the best insulation effect on stators. This method can provide the highest industrial standards for electric machines. By driving out voids from the electric winding through the VPI process, the thermal conductivity of the winding is remarkably enhanced so that the hot spots are eventually eliminated. This also reduces the risk of partial discharge in the winding. In fact, the VPI can make the high stator mechanical integration to reduce vibration of the motor. Since the resins contain no solvents and only a small amount of resin vaporizes during curing, the VPI process has a negligible environmental impact. Presently, this state-of-the-art method is primarily applied on heavy-duty motors.

The primary VPI process involves (Figure 4.37) the following:

- Place a stator in a VPI tank.
- Pull the vacuum of the container for a period of time to remove air and moisture from the stator winding.



FIGURE 4.36
VPI tank used for large electric motors.

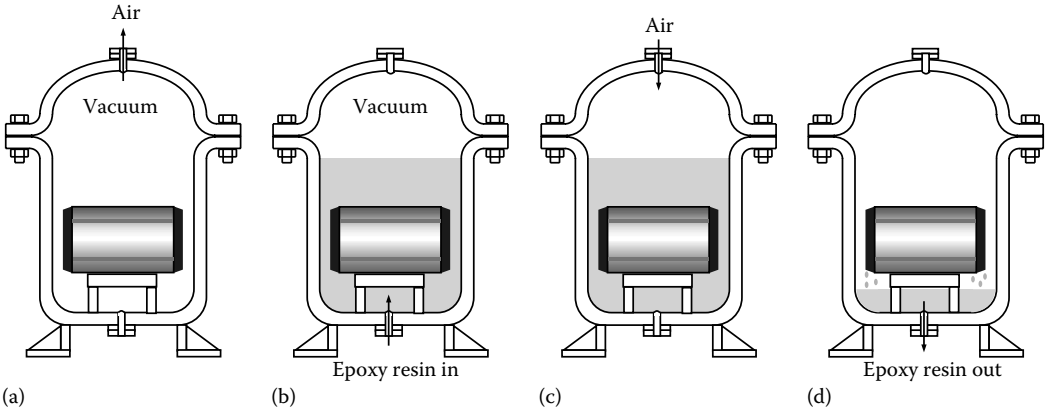


FIGURE 4.37
VPI process: (a) evacuating the container into deep vacuum, (b) introducing epoxy resin into the container until the stator is completely submerged, (c) releasing the vacuum of the container and applying pressure over the resin-covered stator to force resin completely into the stator, (d) draining epoxy resin out of the container, and (e) placing the stator into an oven and baking it for curing the epoxy resin (not shown).

Hesai Exhibit 1031, page 286



FIGURE 4.38
The temperature controlled oven for curing and baking epoxy resins of impregnated motors.

- Introduce a preheated solventless epoxy resin into the vacuum container until it fully covers the stator and hold the vacuum of the container for a period of time.
- Release the vacuum of the container and pressurize the container to a desired level to achieve maximum impregnation for deep penetration of epoxy resin into voids.
- Draw the epoxy resin from the container.
- Place the impregnated stator into a temperature controlled oven (Figure 4.38) and bake it to cure the epoxy resin and fully develop the properties of the insulation system for a period of time at a desired temperature. In today's motor workshops, the baking process is fully controlled by a built-in computer in the oven to followup the preset temperature chart. To prevent *thermal shock* to motor insulation systems, the oven temperature is brought up and down slowly. Ovens are typically equipped with digital panel meters to monitor baking temperatures.

This VPI process can achieve complete penetration of resin throughout turns, coils, slots, and insulation and thus is primarily applied on heavy-duty applications. The VPI technology is superior to solvent varnishes in many aspects. As discussed previously, solvent varnishes lose 50%–70% of volume during baking, leaving voids and air pockets in

the windings. In contrast, VPI provides a 100% solid mass structure that increases the mechanical strength, minimizes the hot spots, and reaches the highest level of environmental protection. A motor winding that undergoes a VPI process is virtually impervious to oil, moisture, and chemical contaminants [4.33].

4.6 Stator Design Considerations

4.6.1 Cogging Torque

Cogging torque is one of inherent characteristics of PM motors, resulted from the interaction of the PM MMF harmonics and the air gap permeance harmonics due to slotting. Since cogging torque can cause speed ripples, induce motor vibration, and deteriorate motor performance, it is one of the major design goals for motor engineers to reduce cogging torque.

Electromagnetic design primarily determines the level of cogging torque. As addressed in Chapter 1, there are many methods for reducing the amount of cogging torque, for instance, skewed stator stack, optimized slot and pole combination [4.34], uneven distribution of stator slots, stator tooth notching, and pole shifting.

In addition to EM design, a cogging condition may be inadvertently set up from improper stator fabrication processes, including the following: (1) The lack of a proper annealing after lamination stamping will directly lead to high residual stresses in the stamped laminations, as well as the degradation of magnetic properties, and thus results in cogging torque. (2) The lack of deburring will increase eddy-current losses. (3) Machining defects (e.g., concentricity and roundness) on the stator ID may result in unequal air gap and thus greatly induce a large amount of cogging torque. (4) Exceeding roundness or concentricity tolerance of the stator ID can significantly change cogging torque. A decrease in the air gap or air gap variations can considerably increase cogging torque. (5) Exceeding tolerance of slot opening is very sensitive to change cogging torque. (6) Welding a stack of stator laminations may increase the core loss at the welded area, cause the deformation of the stator core, and alter magnetic flux distribution. All these factors could change cogging torque. A more important issue is the welding location. Welding in the back iron between the teeth could increase cogging torque.

4.6.2 Air Gap

The radial distance between the rotor and stator in a motor is defined as the air gap. Normally, a smaller air gap provides a more efficient and powerful motor. Hence, it is highly desired to maintain the air gap dimension as small as possible and within a small variation in operation. The control of the air gap dimension involves the design of several components such as the stator, rotor, motor housing, and endbells. An important factor that affects the air gap dimension is the accuracy of the coincidence of the stator and rotor axes. Thus, to provide a motor with a small air gap dimension within only a small tolerance, preciseness in manufacturing of these parts is required.

In attempting to obtain concentricity, it has been proposed [4.35] to mechanically fit the metal rotor bearing support members to the stator stack in a matter that form a cavity. A hardenable plastic material is ejected to fill the cavity and bond the parts together. The assembly is then machined to the desired dimensions.

Hesai Exhibit 1031, page 288

4.6.3 Stator Cooling

An important objective in motor design is to control the motor temperature below its allowable value. Increased motor temperature often reduces motor efficiency and affects bearing life. The Arrhenius equation predicts that the failure rate of an electric device is exponentially related to its operation temperature.

In designing a motor cooling system, thermal engineers always focus on stator cooling for several reasons: (1) The stator winding is usually the main heat source in a motor. Test data show that in most applications heat generated in a motor is primarily attributed to the stator. (2) Cooling in the stator end-winding region is particularly difficult and still remains a challenge due to various factors [4.36]. (3) As a stationary component, the stator is much easier to be cooled compared with the rotor. In fact, the stator often serves as a heat sink for the motor. (4) For some electric motors, the pumping effect, which is resulted from the rotor rotation, is strong enough to generate turbulent circulating flows for making the motor self-cooling [4.37].

4.6.4 Robust Design of Stator

The root causes of motor failure are often related to mechanical deterioration such as vibration, static and cyclic loads (e.g., centrifugal and differential expansion forces), insulation fracture, and bearing lubricant contamination and leakage, to name a few. Robust design techniques can optimize the stator design and ensure adequate motor performance over a wide range of operating conditions.

Because the rotor is supported on bearings located at the endbells of the machine, the stator design is significantly impacted by the dynamic behavior of the rotor. As a matter of fact, the loads carried by the rotor are essentially transmitted to the stator and finally to the motor base. In this load configuration, the structural vibratory loads caused by the rotor, and loading caused by stator vibration that drives rotor behavior, are intertwined. This requires a full understanding of the structural interactions for all motor components, as well as the interaction between the motor and its base.

Motor vibration is greatly influenced by its base. A weak motor base usually results in high vibration. Thus, it is essential to design a motor base strong enough to withstand all motor loads for preventing resonance and vibration.

4.7 Mechanical Stress of Stator

Electric motors are exposed to many kinds of disturbances and stress. Some disturbances are due to imposed external conditions such as mechanical or electric unbalanced system; harmonics; supply interruptions; variations in voltage, current, and frequency; unstable cooling system; and large variations in ambient temperature and humidity. Stress factors include frequent successive start-ups, abnormal use of the motor, stall, and overload situations. These stress and disturbances deteriorate the winding insulation mechanically and increase thermal aging rate and thus may eventually lead to an insulation failure [4.38].

Stresses and strains in stator cores can not only cause mechanical problems but also change the magnetic properties of electrical steels (e.g., silicon steel). A stator is subjected

to a variety of forces during operation. When the forces are relatively small, only elastic deformations are generated on the stator core. Removal of the forces will permit the stator core to return to essentially a stress-free condition. However, under large load conditions, the stator core has to withstand high stresses that produce plastic deformations of the stator core. These plastic deformations can distort the metal crystals or atomic structure and, in turn, affect the magnetization characteristics of the material. As a result, a set of the maximum stress on magnetic core materials must consider both the mechanical and electromagnetic requirements.

There are three main sources of mechanical stress for stators. The first common mechanical stress is caused by power frequency current. When a motor starts, the current through the stator winding reaches its maximum value that gives rise to a magnetic force oscillating at twice the power frequency [4.39]:

$$F = \frac{ki_{rms}^2}{w} \tag{4.1}$$

where
i_{rms} is the rms current
w is the stator slot width
$k = 0.96$

If the current in a stator winding is

$$i_{rms} = A\sin\omega t \tag{4.2}$$

Equation 4.1 can be rewritten as

$$F = \frac{kA^2\left(1-\cos 2\omega t\right)}{2w} \tag{4.3}$$

Noting that $\omega = 2\pi f$, this equation describes the oscillation force in the radial direction at twice the power frequency. If the stator winding resonates at this frequency, the insulation of the winding will be damaged. More likely, the damage occurs at the stator end-winding regions. In fact, resonances cause serious vibration levels of end windings and develop high stress in these windings. As discussed previously, stator encapsulation can provide a good solution to avoid it.

In addition, there is also a force at twice the power frequency in the circumferential direction caused by the rotor's magnetic field interacting with the current in the stator winding. However, this circumferential force is only about 10% of the radial force.

The second mechanical stress is caused from the transient effect. For synchronous machines, switching on of motors or out-of-phase synchronization can give rise to a large transient power frequency current that may be five times (or even more) greater than the normal operating current in the stator. This results in that the magnetically induced mechanical force is 25 or more times stronger than the normal service. This alternative force tends to bend the stator winding, especially in the end-winding regions. If the force is too large, the stator winding may crack.

The third source of mechanical stress resulted from rotors. In a motor, the rotor rotates at a high speed with a coupled externally driven machine. While the rotor is subject to a larger centrifugal force, it transmits its radial and axial loads to the stator via bearings.

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Hesai Exhibit 1031, page 292

5

Motor Frame Design

Electric motors are manufactured in a variety of types and configurations. Typically, an electric motor assembly includes a hollow, substantially cylindrical-shaped housing with a stator and a rotor disposed within the housing interior and an endbell at each end of the housing for providing protections to the motor components against the insertion or intrusion of solid/liquid objects under various environmental conditions. In addition to rotors and stators, some motors may have electronic components that are used to improve operating characteristics for particular applications. These electronic components also need to be well protected.

For radial air gap motors, the stator coils are wound axially through the stator slots, and the end turns of the stator winding are positioned adjacent to the stator end surfaces. The endbells and the motor housing form an enclosure to protect the stator windings and end turns from inadvertent contact and grounding while providing a mounting surface for rotor bearings and shaft bushings. The endbells also prevent debris from entering the hollow interior of the stator and interfering with operation of the motor.

5.1 Types of Motor Housing

Motor housings are designed to provide the support for the stator core and protect motor components from foreign objects such as water, dust, sand, and moisture (see Figure 5.1). Motor housing can be made by a variety of manufacturing processes such as wrapping, casting, extruding, machining, and stamping. For rare-earth magnet motors, either magnetic or nonmagnetic materials can be used to fabricate motor housings. However, for Alnico magnet motors, the housings must be made from only nonmagnetic materials such as aluminum, brass, and stainless steel to preserve specified performance characteristics of the motors. The requirements for the motor housing design include high structural strength for supporting the stator, low vibration transmission, light weight, corrosion resistance, and low cost.

5.1.1 Wrapped Housing

A wrapped motor housing is made from a flat sheet metal that passes through a 3-roll bending machine to form a cylindrical-shaped form. Figure 5.2 illustrates the manufacturing process of wrapped housing. The first step is to prepare a metal sheet with prepunched apertures for installing motor fasteners and notches for receiving electric connection cables (Figure 5.2a). At each edge of the sheet metal, a small notch is left. When the two edges come together nicely, these notches form a narrow clearance for receiving welding operation. Subsequently, the sheet metal is rolled by a bending machine with three adjustable rolls to control the radius of the rolled sheet metal (Figure 5.2b). When the stator core is

Hesai Exhibit 1031, page 294

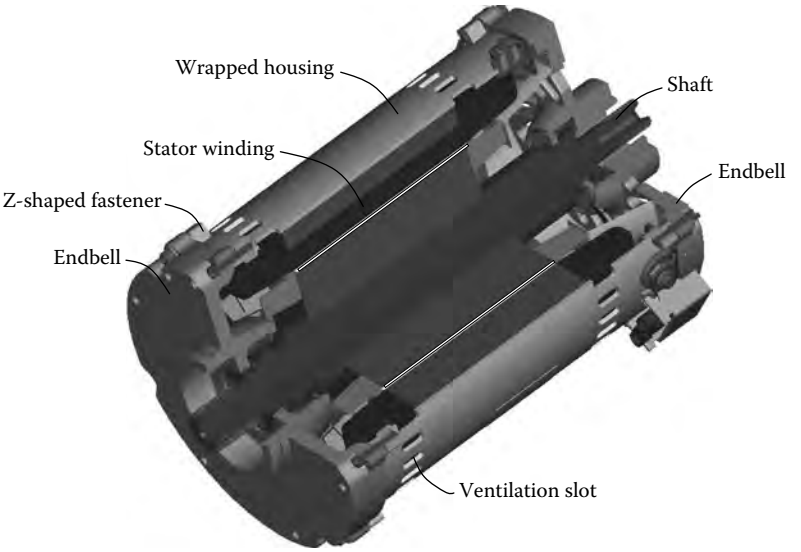


FIGURE 5.1
Cutaway view of electric motor with wrapped housing.

inserted, the wrapped housing is pressed tightly with a special tool for welding along the welding seam (Figure 5.2c). Lastly, the two ends of the housing are machined for receiving the endbells that block off the axial ends.

During the welding process of the wrapped housing, a large amount of heat is produced and conducted into the motor housing and stator core. Excessive heating can cause the deformation and distortion of the housing and stator core, resulting in permanent damage to these parts. High heat input can also adversely affect the mechanical properties of the welded parts. Therefore, it is critical to control the temperature on the welded parts and enable efficient heat dissipation from them to the environment. For example, using a higher welding speed, a lower welding current, and/or an active/passive cooling technique (e.g., cooling via cold plate) can essentially solve most welding overheating problems. While the control of heat input is highly desired in the welding process, it must be noted that the rapid cooling, on the other hand, may lead to the change of metallographic microstructure in welded materials, resulting in the formation of large grains and the reduction in resistance to cracking.

5.1.2 Casted Housing

Casting refers to a manufacturing process that involves a phase change of casting materials between liquid and solid. Metal casting processes can be traced back to approximately 3600 BC as ancient Egyptians utilized casting to make bronzes. In China, the discovery of "Three-Star Piles" civilization in the last century has revealed that about 4000 years ago, ancient Chinese casted out exquisite bronzes. The cast iron technique was invented in China in the seventh century BC. The earliest cast iron pieces in the world were discovered in Shanxi, China, by archaeologists in the 1980s.

A casting process begins by heating up of casting material to reach the material melting temperature or above, at which the casting material is in the liquid state. The molten casting material is maintained at the temperature for a period of time for degassing, then poured

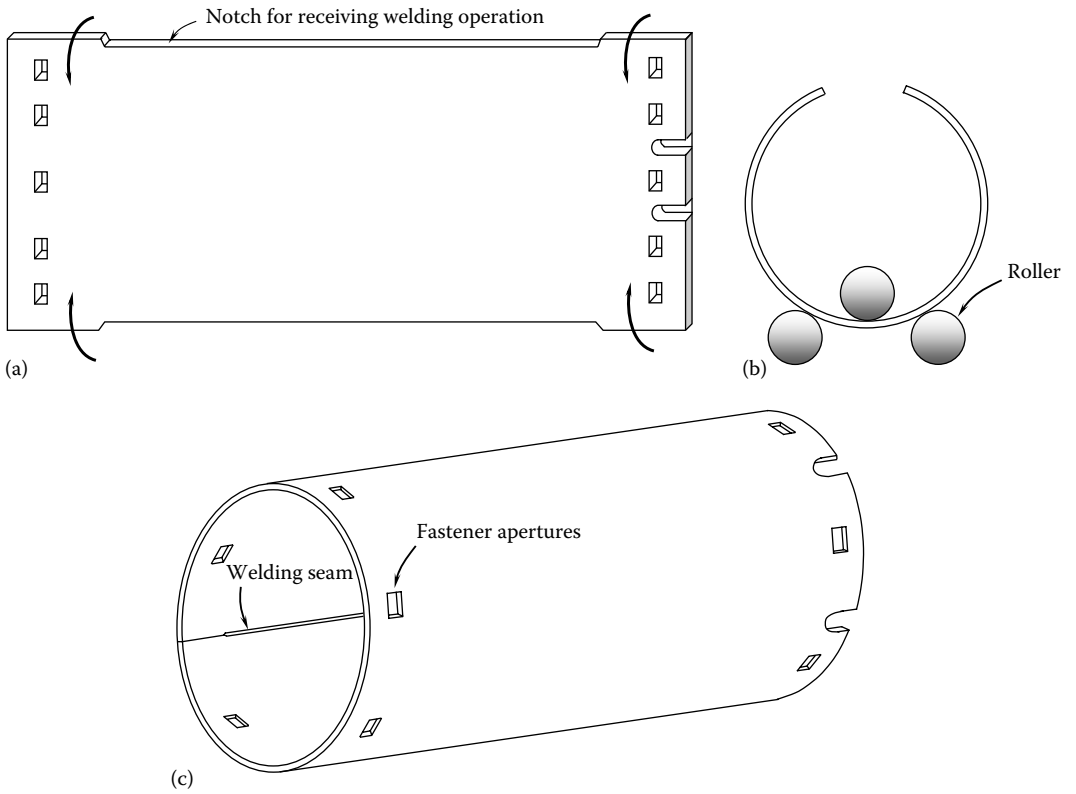


FIGURE 5.2
Manufacturing procedure of wrapped motor housing: (a) punching apertures for installing fasteners and notches for receiving motor cables on a sheet metal using a high-speed punching machine and cutting the sheet metal, (b) rolling the sheet metal to form an open hollow tube with a 3-roll bending machine, and (c) welding at the notched edges to form the motor housing.

into a mold with a desired shape, and allowed to solidify. Therefore, casting technologies are especially suitable for producing motor housings, endbells, and other components with a wide variety of shapes and geometries. In fact, with the ability to effectively produce high-quality and low-cost mechanical components with a wide range of sizes and materials, various casting processes are extensively used in almost all industries, especially in automobile, aerospace, and power industries. Figure 5.3 is a typical servomotor with a casted housing. In this design, the housing is integrated with the front endbell as a single-casted part.

5.1.2.1 Casting Materials

Common casting materials include aluminum, iron, copper, magnesium, zinc, and steel alloys. Among them, aluminum is the most popular material extensively used by motor manufacturers. Benefited with a low melting temperature, high liquidity, light weight, high thermal conductivity, and other good casting characteristics, aluminum alloys have been commonly used for manufacturing motor housings, endbells, and other components. Furthermore, reusability is one of the principal advantages for aluminum. With the improvement of aluminum alloys and casting processes and good balance between

Hesai Exhibit 1031, page 296



FIGURE 5.3
Casted servomotor housing, showing that the front endbell is integrated with the motor housing.

mechanical properties and lightness, it is expected that more and more aluminum-casting parts will be demanded by the motor industry.

Today, cast iron housings are commonly used in heavy-duty and large-size electric motors, as cast aluminum is unable to offer the required strength and hardness. Cast irons are ternary alloys of Fe–C–Si. The microstructure and mechanical properties of cast irons vary considerably depending on the form of the carbon element presented in the irons. There are four main types of cast irons with different carbon forms: gray, malleable, ductile, and white cast iron. Carbon is mainly present as graphite in gray, malleable, and ductile cast iron, while exists as cementite (Fe_3C) in white cast iron. Gray cast iron contains 2.5%–4% carbon and 1%–3% silicon [5.1]. Silicon is essentially needed as a graphite stabilizing agent that makes cementite (Fe_3C) more difficult to form in the cast irons. Graphite is a relatively soft material, and its presence in gray iron gives it some unique and useful properties:

- Excellent vibration damping capacity, being 25–100 times better than that of a 1080 steel [5.2]. Damping is the ability of a material to absorb vibration energy by converting it into other forms of energy such as heat. The quelling of vibration is very important for structures and devices with moving parts. Damping capacity of gray iron strongly depends on its graphite structure and size. For instance, the damping capacity for a gray iron with course flake graphite can be 5–10 times higher than that for a gray iron with fine flake graphite. The strength and hardness of the gray iron is provided by the metallic matrix in which the graphite occurs. Generally, damping capacity decreases with increasing strength since a smaller amount of graphite present in the higher strength irons decreases the energy absorbed. The primary damping mechanism is localized microplastic behavior. Under cyclic loading conditions, this microplastic behavior produces a hysteresis loop whose area is proportional to the energy absorbed during each cycle [5.3]. This property is especially desirable as it reduces the acoustic noise and vibration emitted by motors.
- Superior wear resistance as the metallic matrix microstructure being primarily pearlite. Among the casting materials in Table 5.1, gray cast iron has the highest hardness.
- High compressive strength. Gray iron's compressive strength is typically three to four times higher than its tensile strength [5.4]. Thus, the parts made by gray cast iron are mainly designed to withstand compressive forces rather than tensile forces.

TABLE 5.1

Properties of Some Common Casting Materials for Electric Motors

Material Property	Aluminum 383	Aluminum 356-T6	Aluminum 6061-T6	Ductile Iron 65-45-12	Gray Cast Iron
Melting point (°C)	516–543	557–613	582–652	1148	1180
Density (kg/m^3)	2700	2670	2700	7096	7150
Yield tensile strength (MPa)	152	>165	276	310	—
Ultimate tensile strength (MPa)	310	>234	310	448	276
Elongation (%)	3.5	3.5	17	12	1
Poisson ratio	0.33	0.33	0.33	0.29	0.29
Hardness (Brinell)	75	70–105	95	170–207	183–234
Modulus of elasticity (GPa)	71.0	72.4	68.9	163	90–113
Shear modulus (GPa)	27	27.2	26.0	65	36–45
Fatigue strength (MPa)	145	90	96.5	210	110
Shear strength (MPa)	187	143	207	400	334
Machinability (%)	70	50	50	0	0
Thermal conductivity (W/m-K)	96.2	151	167	32.3	47
Coefficient of thermal expansion @ 20°C (mm/m-K)	21.1	21.4	23.6	14.6	10.8

- Internal lubricating qualities and machinability. Graphite has little strength or hardness. The presence of the graphite provides several valuable characteristics to gray iron, including the ability to produce sound castings, dimensional stability, borderline lubrication, and excellent machinability. In fact, cast iron is the easiest ferrous material for dry machining.
- Improved thermal properties. Gray iron has a better thermal conductivity and a lower coefficient of thermal expansion compared with ductile iron.

Due to these superior properties, gray cast iron has been a staple in automotive manufacturing for many years. As a desired material in vibration damping for achieving low-noise performance, there is a steady increase in the use of cast iron in certain types of motors, such as those in medical equipment.

As a member of the cast iron family, ductile iron contains 3%–4% carbon and 1.8%–2.8% silicon. This type of cast iron has a high strength. Unlike gray cast iron in which graphite is presented as flakes, ductile iron contains tiny graphite nodules, which make cracking more difficult. Therefore, ductile cast iron is much stronger and has much higher ductility than gray cast iron. In addition to high strength, the toughness and shock resistance also increase for ductile cast iron [5.5]. Furthermore, the damping capacity of ductile iron is considerably greater than most other ferrite materials (except for gray iron). These superior properties of ductile iron make it ideally suited for various industrial applications. In the motor industry, ductile iron is often used to make motor housings and endbells.

For ductile iron with high silicon contents, a silicon-rich surface layer is formed to resist oxidation, corrosion, and heat. Stabilization of the ferrite phase reduces the influence of high temperature in two ways: (1) silicon raises the critical temperature at which ferrite transforms to austenite; (2) the strong ferritizing tendency of silicon stabilizes the matrix against the formation of carbides and pearlite, thus reducing the growth associated with the decomposition of these phases at high temperature [5.6].

Hesai Exhibit 1031, page 298

Steel casting is a specialized form of casting, adapted for parts that must withstand heavy loads and shocks. There are generally two types of steel castings to meet different needs of various applications: carbon steel casting and alloy steel casting. Carbon steel casting is the most popular type, which uses low-, medium-, and high-carbon steel in casting to obtain different ranges of strength. For alloy steel castings, special alloy elements (e.g., manganese and chromium) are added to casting steels to enhance the desirable material properties. Among alloy steel castings, stainless steel casting is designed to resist corrosion and thus widely used in the chemical, shipbuilding, and renewable energy industries. However, steel casting is seldom used for making motor components unless casting iron cannot deliver enough strength or motors have strict anticorrosion requirements.

Compacted graphite iron has been developed since the mid-1970s. Because the graphite in this type of iron takes a form between that in gray iron and ductile iron, it has the benefit of combining the high thermal conductivity of gray iron with the high strength and high modulus of elasticity of ductile iron [5.7]. Since it has high thermal-shock resistance, this type of iron is especially suitable in applications involving thermal shocks.

In the selection of materials for casting housings, a number of mechanical and thermal properties must be taken into account, including mechanical strength, elongation, hardness, modulus of elasticity, dimensional stability, shrinkage, machinability, thermal conductivity, among other indices. The properties of the most common casting materials are given in Table 5.1.

5.1.2.2 Casting Process

Casting processes can be divided into two broad groups, gravity-fed and pressure-fed processes, where the gravity fed involves pouring molten metal into mold cavities under gravity and the pressure fed involves injecting molten metal under pressure. The choice of casting process is greatly influenced by product geometric complexity, size, material, dimensional accuracy, surface finish, tooling and other costs, and production volume.

As one of the oldest known manufacturing techniques, casting still remains a highly efficient and cost-effective manufacturing process today. Modern advances in casting technology have led to a wide array of specialized casting methods. In manufacturing industries, four main casting methods are commonly used to make a variety of cast products—die casting, sand casting, permanent-mold casting, and evaporative pattern casting (EPC)—each having its own unique fabrication advantages and limitations.

Die casting is a manufacturing casting process that can produce geometrically complex metal parts through the use of reusable molds (i.e., dies). There are traditional high pressure and also low pressure die casting processes. The casting materials typically are nonferrous alloys such as aluminum, zinc, magnesium, and copper. In a die-casting process, molten metal is injected under high pressure over a very short period of time into a hardened steel or cast iron die (Figure 5.4). Then, the ejection die is removed, and the die-casting part is ejected by means of ejection pins. The total cycle time is very short, typically between a few seconds to less than a minute. High casting pressures help reduce porosities in casting parts, increase casting uniformity, and improve the surface finish and dimensional accuracy. Because die casting can provide good surface finishing and dimensional accuracy, only light machining is required to enforce the accuracy of a few critical dimensions. In some cases, post machining can be completely eliminated. A key feature in die design is the positioning of the runners and gates. Well-designed gates permit rapid flow into the die cavity, minimizing turbulence and long flow paths for the molten metal.

Hesai Exhibit 1031, page 299

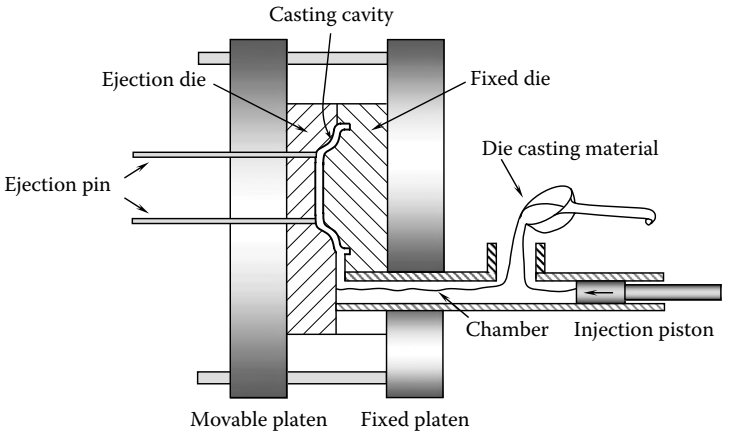


FIGURE 5.4
Illustration of die-casting process.

Die-casted housing is most popular in the motor industry due to its inherent advantages of light weight, low cost, high thermal conductivity, and high productivity. In the design of die-casting housing, the most important thing is to determine the departing line as it has a direct impact on the success of the casting process. Secondly, it is better to make the wall thickness as uniform as possible to reduce casting deflects caused by nonuniform cooling rates and residual stresses in casting housings. Reinforced ribs are often employed in motor housings to reduce the housing weight and casting material. All corners of casting parts should be radiused generously to avoid stress concentrations and cracks. The set of the draft angle allows casting parts to be easily removed from dies. The minimum draft angle per side in the parting direction depends on the casting materials: usually 0.25° for zinc alloys, 0.5° for aluminum alloys, and 0.75° for brass. In practice, a draft angle of 1°–2° on all walls in the parting direction is suitable for most applications. The arrangement of ejection pins is critical for allowing casting parts to be effectively ejected from dies.

Sand casting applies reusable sand to make sand molds. The advantages of sand casting include low cost of mold, large-size casting part, and wide variety of metals and alloys. However, sand-casted parts also suffer rough surface and poor dimensional accuracy. In this technique, the preparation and bonding of the sand mold are the critical steps. A typical structure of a sand-casting mold is shown in Figure 5.5. For casting parts with a simple geometry, the mold may be consisted of two components, that is, the upper mold and the lower mold. Depending on the geometry requirement, cores may be placed in the mold. The paths of molten metal are usually arranged on the department surface. Gas vents are left on the upper mold to allow air to escape from the mold during the casting process. In sand casting, molten metal is poured into the sand mold with the aid of the gravitational force.

Permanent-mold casting refers to the casting process that employs reusable metal molds to produce casting products repeatedly. This casting process usually requires the preheat of mold. Therefore, when the molten metal is poured into the mold under the gravitational force, the expansion of the mold is smaller than that in sand casting without preheating. The molds are usually made of high-strength steel or iron alloys. To achieve the desired lifetime of the mold, this casting process is mainly used for lower-melting-temperature

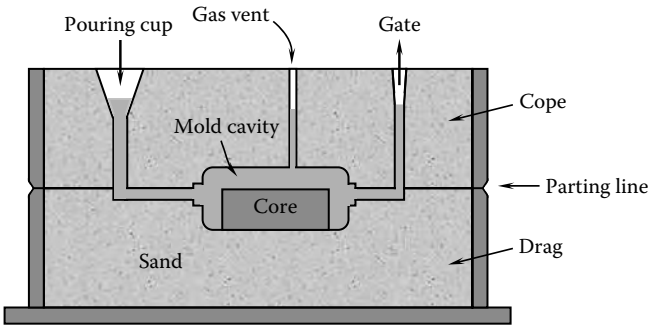


FIGURE 5.5
Typical sand-casting mold and casting process.

materials such as aluminum alloys. When the process is used to cast steel or iron alloys that have much higher melting temperatures, the mold life is extremely short [5.8].

For the ease of manufacturing and casting, the mold usually consists of two or more components. To reduce the temperature effect on the mold dimensions, a thin layer of heat-resistant material (such as clay or sodium sulfate) is coated on the inner surfaces of the mold.

Permanent-mold casting produces castings with excellent structural characteristics. There are some inherent advantages that the permanent-mold-casting process has over other casting processes: (1) because of the reusable molds, the unit tooling cost is lower than other casting processes under high-volume production conditions; (2) some mechanical properties of casted parts via permanent-mold casting are more favorable; (3) the dimensional accuracy is increased; (4) the surface finish is improved; and (5) the directional solidification rate can be controlled by varying the mold wall thickness or selectively heating or cooling certain positions of the mold. The fast cooling rates created by using a metal mold result in a finer grain structure than sand casting [5.8]. However, because of the high tooling cost, this casting process is obviously uneconomical for low-volume productions. The permanent-mold-casting process is especially suitable for making motor housings with thick walls. Figure 5.6 illustrates the permanent mold for elevator motor housings.

The EPC technique has been developed for providing a cost-effective and low-pollution solution in the foundry industry. EPC takes advantage of certain materials that are easy to evaporate under high temperatures (e.g., foam plastic) to fabricate casting patterns. The pattern is coated with refractory materials and buried into dry sand.

There are two main cast processes in EPC. The lost-foam-casting process is demonstrated in Figure 5.7 [5.9]. In this casting process, a pattern is positioned in a suitable container. Dry sand is placed in a container to surround and cover the pattern. Sand is vibrated to ensure that the pattern is completely surrounded without leaving undesirable voids. As the molten alloy is introduced into the mold through a sprue, the pattern material evaporates immediately upon contacting the molten alloy. As a consequence, the molten alloy replaces the space of the pattern and forms a seamless casting piece, which has exactly the same shape as the pattern. Therefore, there is no need for removing the pattern from the mold during the whole casting process.

Full-mold casting combines sand casting and lost-foam casting. It employs an expandable polystyrene (or foamed polystyrene) pattern, which is supported by sand in a single-piece sand mold. The molten metal is then poured through the sprue into the mold. With the

Hesai Exhibit 1031, page 301

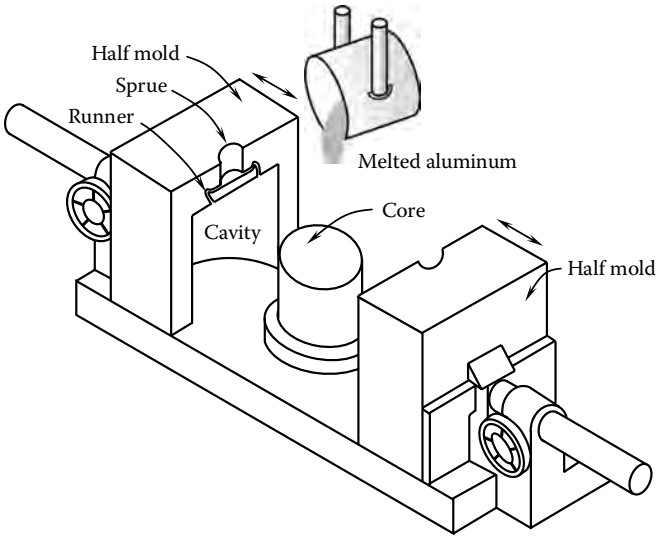


FIGURE 5.6
Permanent-mold casting for motor housing.

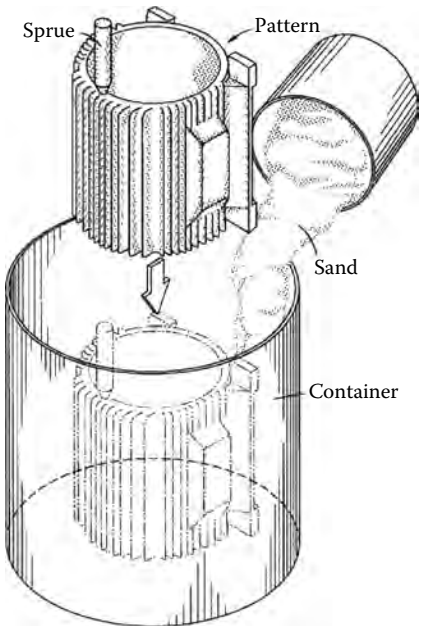


FIGURE 5.7
Making motor housing with lost-foam-casting process (U.S. Patent 6,109,333) [5.9]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 302

progressive evaporation of the pattern material, the molten metal fills the available space. This casting process can make complex-shaped parts (e.g., automobile engines) without using cores and drafts.

5.1.2.3 Pressure Casting

In order to reduce porosities in casting parts, pressure casting is widely adopted by many motor manufacturers, especially for large-size, high-power-density motors. In this casting technique, the molten metal is forced into a metal mold under pressure. The higher the applied casting pressure, the shorter the casting time and the less the porosities in casting parts.

Figure 5.8 presents a low-pressure-casting process. While the molten metal is poured into the crucible, a pressurized gap forms. The pressure acts on the surface of the molten metal to force the molten metal into the mold through the refractory tube. The air pressure is maintained until the casting part has hardened within the mold. Once the casting part has solidified, the mold is opened and the casting part is removed. In this casting process, since the molten metal is drawn from the lower part of the crucible, it has less exposure to the environment and thus contains less trapped gas and other undesired contaminants, compared with the molten metal at the upper part of the crucible. Therefore, casting quality is remarkably improved.

5.1.2.4 Heat Treatment

During a casting process, the solidification rate of the casting part is far from uniform. This is especially true for permanent-mold casting in which the liquid aluminum contacted with the metal mold surfaces solidifies much faster than that in the bulk of the part. Consequently, large residual stresses are created inside the casted part with nonuniform mechanical properties.

There are several heat treatment methods, each having a unique combination of temperature and time depending on the casting process, casting material, and cast requirements. Standard T6 heat treatment can enhance material properties such as improved ductility and fracture toughness through spheroidization of the eutectic silicon particles

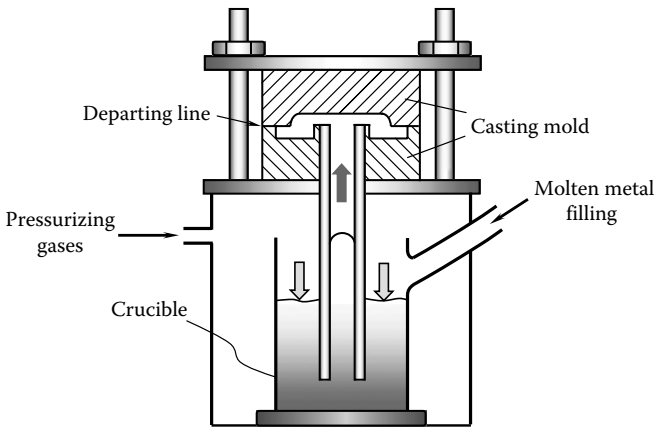


FIGURE 5.8
Pressure-casting process.

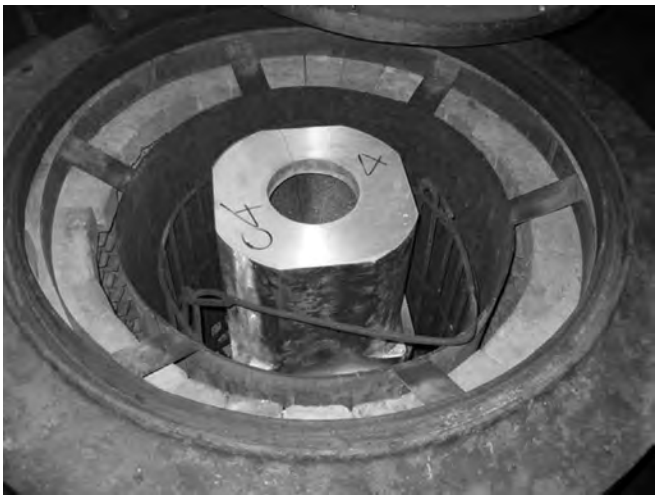


FIGURE 5.9
Furnace for heat treatment of casted housing.

in the microstructure and increase yield strength through the formation of a large number of fine precipitates [5.10].

As part of T6 heat treatment, artificial aging at a relatively low-temperature range of 120°C–210°C can effectively reduce residual stresses. During this stage, the precipitation of dissolved elements occurs. These precipitates are responsible for the strengthening of the material [5.11]. It is interesting to note that a casting process normally takes less than 10 min, while a typical T6 heat treatment cycle may take more than 10 h. Figure 5.9 shows a casted aluminum housing of an elevator motor placed in the furnace, waiting for the heat treatment.

5.1.3 Machined Housing

For fast fabrication of motor prototypes, motor housings are often machined to reduce the lead time and avoid high tooling costs in casting processes. In such a way, motor housings can be machined to exact dimensions from raw materials by means of CNC machining centers. However, this method is not suitable for housings with very large sizes and/or complicated structures.

5.1.4 Stamped Housing

In stamping processes, metal sheets are deformed to desired shapes and dimensions under high stamping forces at room temperature. Because of its high productivity and acceptable accuracy of shape and size, stamping process is extensively used in various industries. In motor manufacturing, stamped housings and endbells are typically used for small-size, large-volume, and ordinary motors such as vacuum cleaner motors (Figure 5.10).

5.1.5 Extruded Aluminum Motor Housing

In various industries, aluminum extrusions provide engineering solutions for a diverse range of machinery and equipment needs due to their complex shapes, light weight,

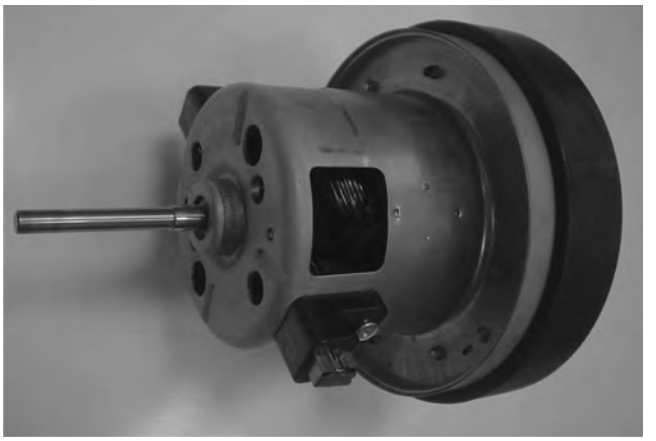


FIGURE 5.10
Stamped motor housing of a vacuum cleaner DC brush motor, showing that the drive-side endbell has been integrated with the housing.

high thermal conductivity, dimension precision, high strength, and minimal maintenance. For instance, properly designed and fabricated aluminum extrusions are an excellent choice for electric motor housings. An extruded motor housing with extruded fins and cooling channels on the housing wall can literally be obtained as a finished component by cross-cutting from extruded materials. Even with small or moderate production runs, the manufacturing costs of extruded motor housings are well below those of cast aluminum housings [5.12]. In addition, since cooling fins are designed on the extruded materials in all sides, as shown in Figure 5.11, the extruded housing design can give electric motor an outstanding cooling.

5.1.6 Motor Housing with Composite Materials

The use of composite materials in motor housing facilitates the fabrication of motors with reduced weight, lowered vibration transmission, improved mechanical damping, increased corrosion resistance, enhanced cooling, and improved motor performance. Variation of the thickness and fiber laying geometry in the composite housing may be used to control motor housing strength, stiffness, and damping characteristics. The motor housing contains both metal and composite resin material that may take more than 50%, making it lighter and easier to fabricate [5.13].

5.1.7 Motor Housing Fabricated by 3D Printing Technology

The 3D printing technique was originally proposed by researchers in MIT in the early 1990s [5.14,5.15] as a means to produce relatively inexpensive prototype parts for industrial and automotive design work. Unlike traditional manufacturing processes, this new manufacturing process allows computer-generated 3D models to be directly transformed into physical objects through a layered printing process. Because of its cost-effectiveness and versatility, the advent of this new manufacturing process ultimately has the potential to become as important and game changing as the introduction of the assembly line a hundred years ago [5.16].



FIGURE 5.11
Motor housings are made by cutting from extruded aluminum. The fins are extruded in the axial direction on all sides of the extruded aluminum for enhancing heat dissipation.

In recent years, this technique has been developed to become a manufacturing process for producing solid parts. This technique is an extremely versatile and rapid process accommodating geometry of varying complexity in a wide range of industrial applications. Because this technique can support many types of materials, it is especially suitable for the fabrication of prototypes.

With newly developed 3D printing techniques such as laser sintering and electron beam melting, this manufacturing process has been extended to produce metal parts. Although it has not been used for fabricating motor parts today, the 3D printing techniques, with the continuous improvement and cost reduction, may be applied to the fabrication of motor parts and casting toolings in the near future.

5.1.8 Frameless Motor

Frameless motors (see Figure 5.12) are especially suitable for certain applications such as airplanes and satellites for its reduced volumes and weights. The advantage of frameless motors, as compared with conventional housed motors, resides in its high reliability and efficiency, as well as its capability to form a compact mechatronic system without using redundant parts or components.

A typical type of frameless motors is the frameless DDR motor (Figure 5.13). A DDR motor generally consists of two separate components, that is, rotor (field) and stator (armature). Frameless motors are integrated directly with the load where the same bearings that support the load also support the motor. This configuration eliminates shaft, bearings, endbells, and couplings, which offer reduced volume, weight, and complexity, and also results in improved servo stiffness and quicker response [5.17]. In addition, the frameless DDR motor eliminates mechanical transmissions, such as

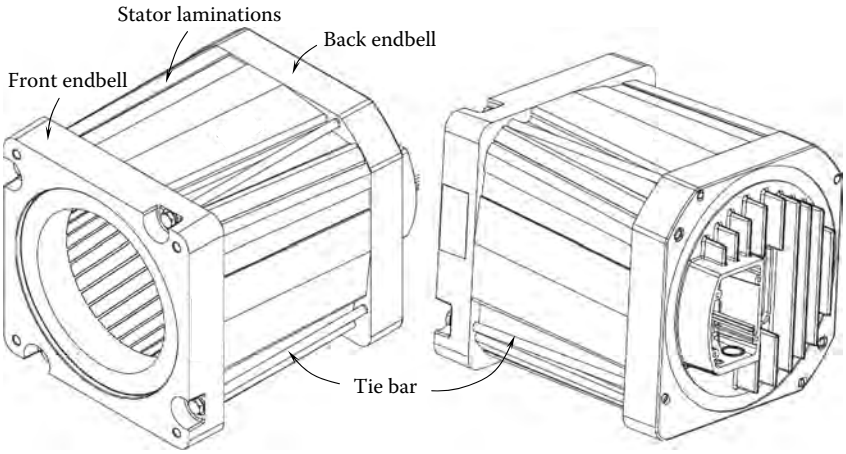


FIGURE 5.12
Frameless motor, assembled with tie bars on endbells.



FIGURE 5.13
Stators of frameless DDR motor.

gearboxes, belts, and pulleys, resulting in zero backlash, quiet operation, and zero-maintenance servo solution.

Frameless motors are attractive to those customers whose applications demand a minimized motor size and weight along with a maximized dynamic performance. Since the load is often supported on its own bearing structure, the frameless motor can be integrated directly into the system/load shaft and be suspended on the same bearings with the load.

The advantages of the frameless motor include the following:

- By eliminating the motor housing, the thermal resistance between the heat sources (e.g., stator and rotor windings) and the environment is greatly reduced.
- Compared with the conventional framed motor, a frameless motor has a smaller volume for a given power rating. Thus, frameless motors offer the most compact and lightweight solution for the motion control industry.

- Frameless motor has a high overall stiffness. Shaft stiffness is a function of the cube of the shaft diameter. For a frameless DDR motor, the shaft diameter can be approximately three times larger than that of a conventional motor.
- Hollow shafts with large diameters can be used in frameless motors for not only reducing rotor inertia but also improving motor cooling.
- Frameless motors can reduce not only the part cost but also assembly time. Therefore, the total motor cost is reduced.

For frameless motors, the motor sealing system must be carefully designed to prevent any potential damages to the motor parts.

5.2 Testing Methods of Casted Motor Housing

Testing machines are commonly used by motor manufacturers to determine mechanical properties of materials, such as yield strength, ultimate strength, strain, and elongation. As shown in Figure 5.14, a typical tension testing machine possesses a movable upper head and a stationary lower head. The grips, which are attached to the heads of the testing machine, hold the specimen firmly at each end. A tensile load is then applied to the test specimen by controlling the separating rate of the two heads until the specimen fractures. This process produces a stress–strain curve showing how the testing specimen behaves throughout the testing process. The test data are recorded by a monitoring/measuring system during the test and are used to determine the mechanical properties of the material.

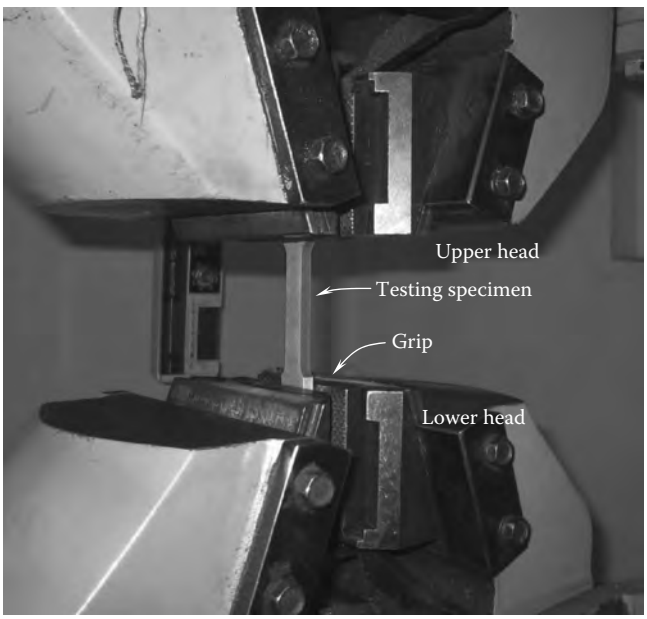


FIGURE 5.14
Standard configuration for tensile testing strip material cutting from casting housing and endbell.

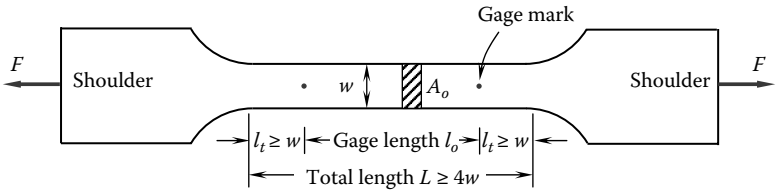


FIGURE 5.15
Typical tensile specimen, showing a reduced gage section at the middle and enlarged shoulders at the ends of the specimen.



FIGURE 5.16
Testing specimens cutting from different locations of a casting housing.

In order to understand the strength distribution over the whole motor housing, the test specimens are specifically sampled at different locations of the housing. The dimension and size of testing specimen are described in the standard [5.18] and are shown in Figure 5.15. The specimen has a reduced gage section in the middle and enlarged shoulders at the ends for being held by serrated grips. To avoid the effect from the shoulders, the length of the transition region x should be at least as big as the specimen width w, and the total length of the reduced section L should be at least four times the width w.

Figure 5.16 illustrates the tested specimens that were cut from the different locations of the same casted aluminum housing. It shows clearly that all fractures occurred in the test area between the gage marks.

The stress σ is determined from the measured load F and the original specimen cross-sectional area A_o:

$$\sigma = \frac{F}{A_o} \tag{5.1}$$

Hesai Exhibit 1031, page 309

The strain ε is calculated as the ratio of the length change Δl during the test to the original specimen length l_o, which is measured between the gage marks before the test:

$$\varepsilon = \frac{l - l_o}{l_o} = \frac{\Delta l}{l_o} \tag{5.2}$$

After fracture, the fractured specimen needs to be put together to measure its gage length l_1. Thus, the elongation of the material is calculated as

$$\varepsilon_e = \frac{l_1 - l_o}{l_o} = \frac{\Delta l_e}{l_o} \tag{5.3}$$

Stress–strain curves are an extremely important graphical measure of a material's mechanical properties. As an example shown in Figure 5.17, for low strains, the material obeys Hooke's law to a reasonable approximation so that stress is proportional to strain with a constant of proportionality (the line *AB*). The Young's modulus *E* is computed as

$$E = \frac{\sigma}{\varepsilon} \tag{5.4}$$

As strain is increased, the material eventually deviates from this linear proportionality and exhibits nonlinear behavior between stress and strain. The highest point attained before the stress–strain line begins to bend is defined as the proportional limit, which is the maximum point at which the material obeys Hooke's law.

Material yield strength refers to the point on the stress–strain curve beyond which the material experiences deformations plastically, as the molecular or microscopic structure undergoes a rearrangement and atom lattice planes slide over each other. This indicates that the yield point and the proportional limit refer to the same point. The only difference is that one is defined from the microscopic perspective and another from the macro perspective.

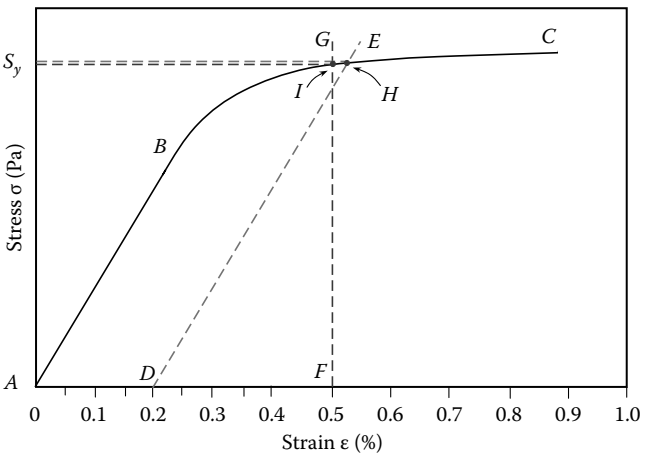


FIGURE 5.17
Determination of yield strength from the offset method and the extension under load method.

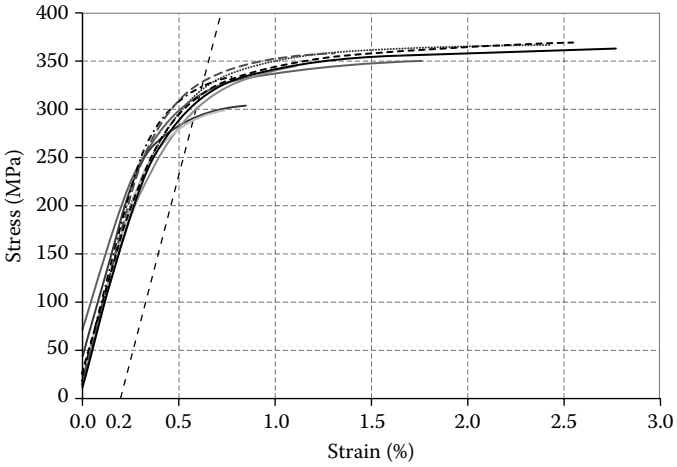


FIGURE 5.18
Stress–strain curves obtained from tested specimens cutting from different locations of the casted aluminum housing. Each curve represents the stress level at that location.

In practice, it is often difficult to pinpoint the exact yield strength at which plastic deformation begins curve. There are primarily two common methods to determine yield strength from the stress–strain curve: the offset method and the extension under load method. The offset is the horizontal distance between the modulus line (the line *AB*) and any line running parallel to it. In industries, tensile yield strengths are often determined by this method at an offset of 0.2% of strain. The intersection of the offset line and the stress–strain curve (the point *H*) thus gives the yield strength.

The extension under load method involves drawing an ordinate line *FG* from the point on the *x*-axis where the strain equals the specified extension, typically 0.5%. The intersection of the line FG and the stress–strain curve (the point *I*) defines the yield strength. From Figure 5.17, it can be seen that the values determined by two methods are very close.

The measured stress–strain curves from the samples being cut from different locations of the housing are displayed in Figure 5.18. It can be seen that the ultimate tensile strength and strain data distribute in the range of 300–370 MPa and 0.7%–2.7%, respectively. These results are very important for optimizing the motor housing design and improving casting processes.

5.3 Endbell Manufacturing

As an integral part of the motor structure, motor endbells are used to cover the ends of motor housing, to house bearings at the ends of a motor for supporting the rotor, to provide paths for cooling air flows, and to protect internal electrical and mechanical parts from moisture and dirt. Motor endbells can be made by many manufacturing processes, such as casting, machining, stamping, forging, extruding, and powder metallurgy sintering. In considering the production efficiency, cost, appearance, and other factors, casting processes have been widely adapted to produce motor endbells.

5.3.1 Casted Endbell

Similar to casting motor housings, motor endbells are often produced through a casting process. As shown in Figure 5.19, an endbell is made of die casting with an aluminum alloy. In any cases, endbells must be designed to provide a rigid and stable support for the stator and rotor assembly. For the consideration of safety, a minimum clearance gap of 1.5 mm must be provided between the coil end turns and the endbell wall.

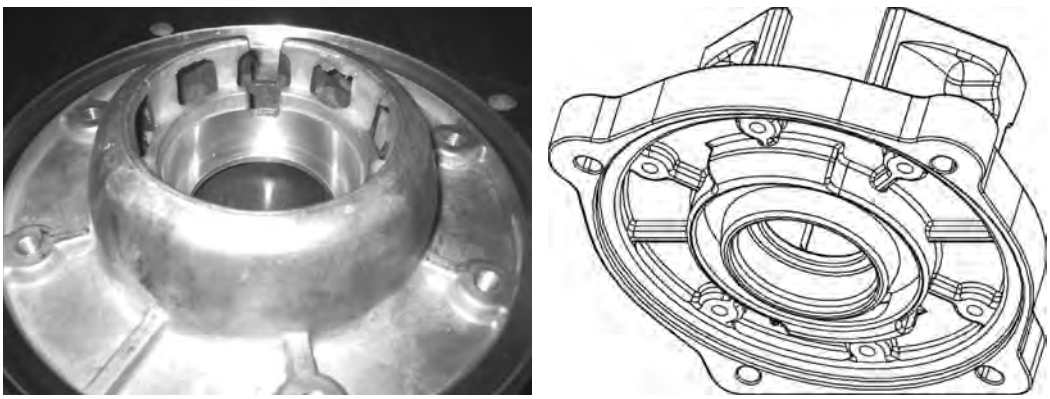
Motor endbells can be made by either die casting or sand casting. Compared with sand casting, die casting produces parts with thinner walls, higher-dimensional accuracy, and smoother surfaces. Production is faster and labor cost per casting is lower.

For some motors with low IP requirements, endbells may be designed to have cooling openings serving as the passage of air over and around the windings. For totally enclosed motors, the endbells must have reliable and tight sealing capabilities. Therefore, no openings are allowed on the endbells.

In the design of casting endbells, it is important to consider the effect of material impurity and porosity on the strength of casting parts. In order to ensure good quality of casting, it is common to cut the casted parts to inspect casting defects, especially porosities and voids, as demonstrated in Figure 5.20. However, due to the multiparameter control in casting processes and the complex geometry of motor components, it is difficult to completely avoid casting defects. As an example, a broken endbell is presented in Figure 5.21.

5.3.2 Stamped Endbell

To provide an economic solution for endbell production, stamped endbells are often used in low- or mid-end motors. As a matter of fact, stamped parts are widely used in many advanced industries such as automotive, aerospace, and energy due to their good mechanical properties (e.g., light weight), low cost, and quick production. In practice, stamped parts have replaced many expensive casted, forged, and machined products in small motors.



(a) (b)

FIGURE 5.19
Aluminum endbells made by die-casting process: (a) with ventilation passages and (b) without ventilation passages.

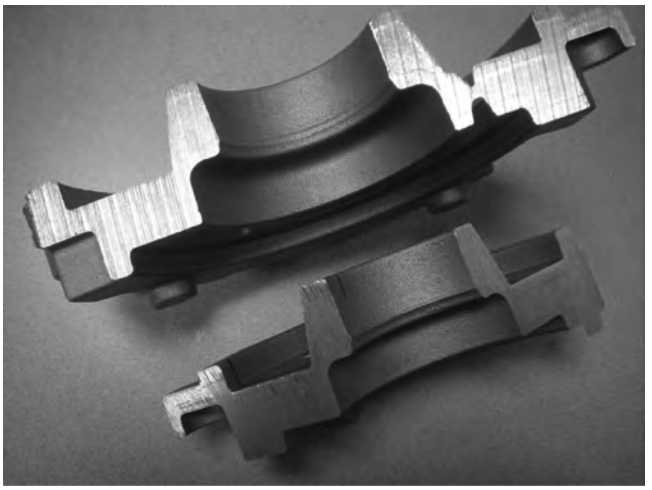


FIGURE 5.20
A casted endbell is cut to inspect the casting defects, particularly for porosities and voids.



FIGURE 5.21
Broken aluminum die-casting endbell due to the porosity and impurities in the part.

5.3.3 Iron Casting versus Aluminum Casting

A number of metal alloys are available for die casting such as aluminum, zinc, magnesium, iron, and copper. In most modern small motors, the endbells are made from aluminum alloys for their good corrosion resistance, light weight, high-dimensional accuracy, stability, and high thermal conductivity. Endbells made from high-grade cast iron are often used for large motors due to its strength, rigidity, and the ability to attenuate motor vibration and noise.

5.3.4 Machined Endbell

Similar to machined housings, machined endbells are often used for motor prototypes or small quantities of products for a short lead time and no need of tooling. It is apparent that machined endbells are more expensive than those made by other manufacturing processes such as die casting for a mass production. With the rapid development of 3D printing technology, it is possible to use the 3D printing technology to replace the machining technology for making motor endbell prototypes in the future.

5.3.5 Forged Endbell

Forging is a manufacturing process involving the shaping of metal by pressing or hammering. Metals can be either cold-forged or hot-forged depending on applications. Forging refines the grain structure in the direction of the deformation and develops the optimum grain flow. The modified structure gives forged parts better mechanical properties than machined parts in which the grain flow is broken by machining. Generally, forging technology is suitable for making parts that have simple geometries. For endbells with complicated structures, casting is more convenient than forging as the manufacturing process.

5.4 Motor Assembly Methods

There are a number of different ways to secure motor assembly, each of which is applicable to certain applications depending on the motor size, motor type, IP, and mounting requirements.

5.4.1 Tie Bar

This method is to use nuts, in coordination with long bolts (or in the alternative key bars), which extend through a distal side of one endbell, the housing and the opposite endbell, to secure the assembly together, as shown in Figure 5.22 [5.13]. The high compressive forces produced by bolts act to maintain the end shields in a static position with respect to the housing. This assembly method is especially suitable for a motor with a low length-to-diameter ratio. For motors with large length-to-diameter ratios, however, this assembly method is prohibitive due to the inherent structural instability caused by the increased bolt length to match the long motor body.

5.4.2 Tapping at Housing End Surface

A frame assembly method incorporates tapping directly at the housing end surfaces, creating attachment sites (i.e., threaded spaces) integrally as part of the housing itself (Figure 5.23). Specifically, short screws are threaded into the tapped housing through the endbells. However, this necessitates either a significant increase in the housing wall thickness or a housing design with complex structures formed with additional material that integrally provide a portion of the housing configured to receive the screws.

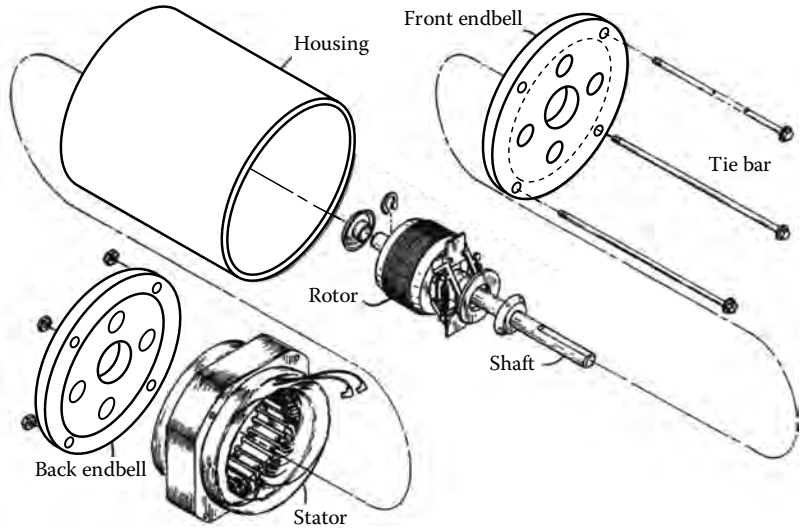


FIGURE 5.22
Motor assembly is secured by tie bars (U.S. Patent 5,412,270) [5.13]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

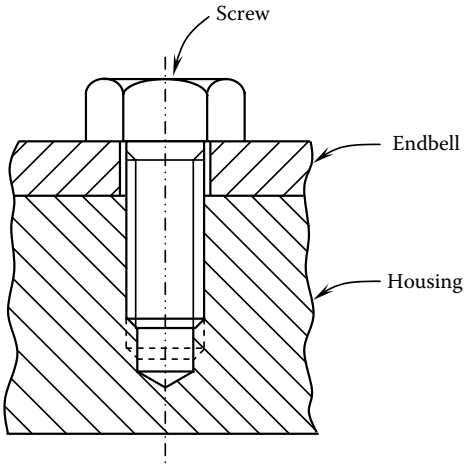


FIGURE 5.23
The motor housing is tapped for receiving short screws, which go through the endbell to complete the construct of the motor frame.

5.4.3 Forged Z-Shaped Fastener

Forging refers to the shaping of metal by plastic deformation. Forging is one of the most economical metal-forming methods for its advantages of significant material saving, simplified production process, high production rate, superior surface quality (especially for cold forging), high shear strength, and high fatigue resistance. During a forging process, internal grain deforms to follow the general shape of the part. As a

result, the grain is continuous throughout the part, giving rise to a piece with improved strength characteristics [5.19].

Cold forging is done at room temperature while conventional hot forging is done at temperatures higher than material recrystallization temperature (for instance, up to 1150°C for steel, 360°C–520°C for aluminum alloys, and 700°C–800°C for copper alloys). Motor manufacturers may choose cold forging over hot forging for a number of reasons, including little or no finishing work, higher-dimensional accuracy, better surface finishing, and improved reproducibility and interchangeability.

One of the most common types of cold forging is a process called impression-die forging, where the metal is placed into a die. Then, the metal is forced into the die by a pressing machine or a descending hammer. In cold-forging operations, pressures as high as 2500 MPa are developed at the forming tool–workpiece interface [5.20]. In order to minimize the friction between the tool and the processed workpieces, phosphate coatings together with lubricants are applied prior to cold forging.

It is widely recognized that surface finish has a significant effect on fatigue behavior. During a forging process, oxide and scaling layers are formed on the surfaces of the forged workpieces. Thus, it requires using either chemical methods such as acid pickling or mechanical methods such as sand blasting and shot peening to remove such oxide and scaling layers. In addition, the residual stress, produced in forgings as a result of inhomogeneous deformation in the forged workpieces, must be relieved through the annealing process following the forging process.

To provide robust motor frame assemblies, Boyland et al. [5.21] invented Z-shaped fasteners for reinforcing the joints between endbells and motor housings (Figure 5.24). Fastening apertures are formed in the lateral ends of the housing. The fastener includes elements that engage the housing through a plurality of contact areas. Specifically, the fastener includes a central portion that establishes a contact area in the plane of the aperture, as well as a

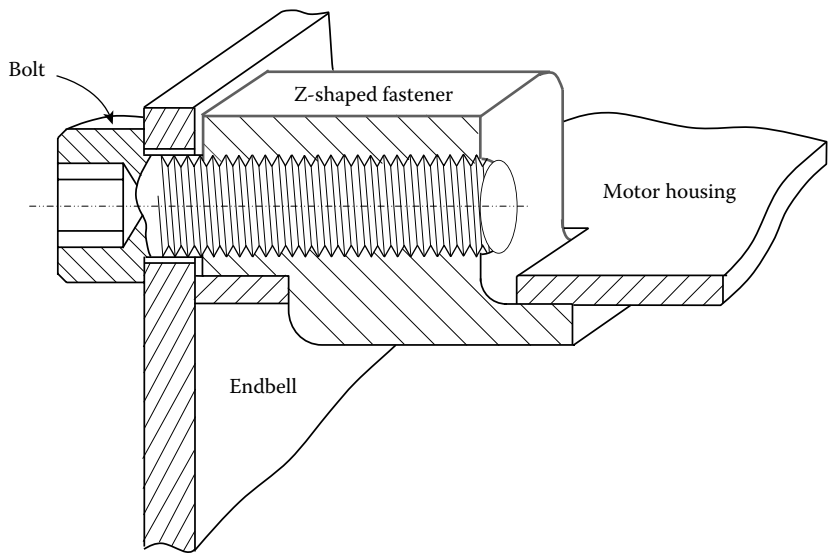


FIGURE 5.24
Jointing motor housing and endbell with Z-shaped fastener.

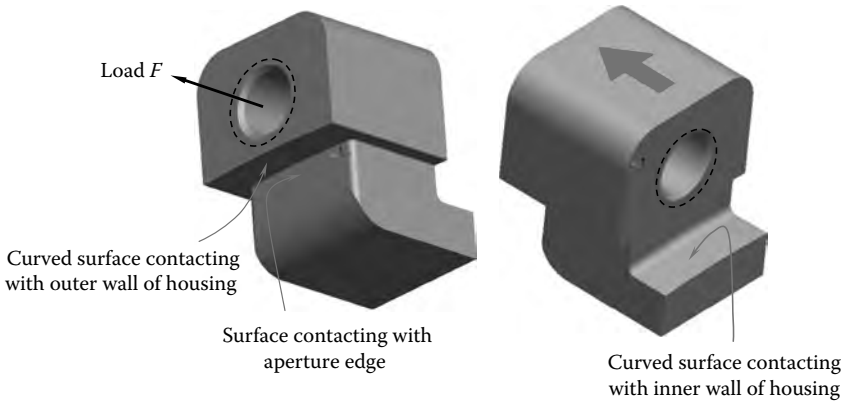
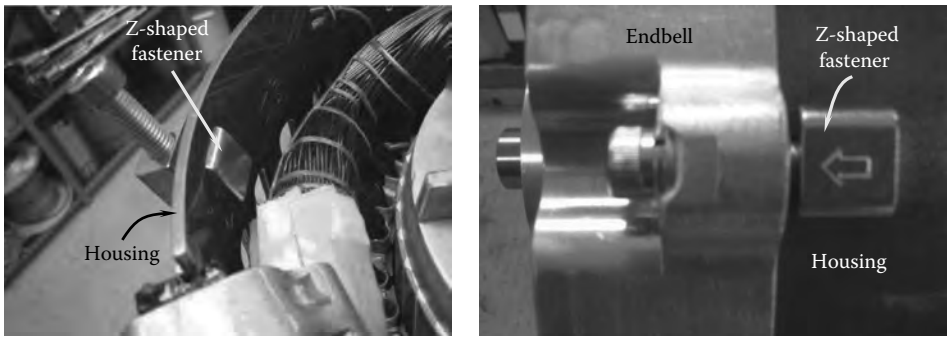


FIGURE 5.25
Contacting surfaces of Z-shaped fastener to housing.



(a) (b)

FIGURE 5.26
Configuration of Z-shaped fastener to housing (a) and housing endbell (b).

base extension tab that establishes at least one contact area with the interior wall of the housing (Figure 5.25). The fastener also includes an extension block that establishes a contact area with the exterior of the housing and engages a securing bolt. The securing bolt threads through an aperture in the endbell and engages the fastener, thereby securing the endbell to the housing. The configuration of Z-shaped fastener to a housing and housing endbell is given in Figure 5.26.

Advantageously, this type of fastener reinforces the motor housing so as to provide additional support to the assembly by reinforcing the joints between the endbell and the motor assembly main body. Furthermore, the invention secures the endbell to the housing by simply engaging the housing and the securing bolt. Accordingly, the fastener acts to simplify housing/endbell manufacturing and the assembly process.

5.4.4 Rotary Fasteners

Rotary fasteners refer to the fasteners that insert through the apertures of joint members and rotate certain angles to secure the joint members together. This type of fasteners is capable of quickly and easily securing and releasing joint members.

5.4.4.1 Triangle-Base Rotating Fastener

Figure 5.27 illustrates a triangle-base rotating fastener with a triangle-shaped base extension tab. The fastener utilizes both horizontal and vertical forces in relation to the housing while engaging the housing's fastening aperture. The extension block and base extension tab may exert forces on the exterior and interior housing walls in a vertical direction. These vertical forces work in coordination with horizontal forces exerted by the central portion of the fastener and the force created by engaging a securing bolt with the extension block.

With regard to engaging the securing bolt with the extension block, the securing bolt used during the fastening process applies a load to the fastener. The load works, in coordination with the fastener contact areas, to maintain the vertical and horizontal fastener position with respect to the housing, the endbell, and the securing bolt. The contact areas apply a retaining force or pressure to the interior and exterior walls of the housing as well as apply a force in the plane of the fastening aperture portion of the housing.

The underside of the extension block joins the central portion to create a second section of the fastener. The central portion lies in the plane of the fastening aperture as the fastener engages the housing. The central portion also separates the base extension tab and the extension block in the fastener. The central portion is formed such that after the fastener is rotated, the central portion provides additional structural support along with an additional contact area between the fastener and the housing. Generally, both the extension block and the base extension tab extend beyond the central portion to engage the surface of the housing. This advantageously allows for the fastener to firmly engage both the interior and exterior sides of the housing wall, as well as a securing bolt.

The cross-sectional view of triangular base extension tab is shown at the right-hand side in Figure 5.27, taken along lines *A*–*A*. It is to be understood that for a given embodiment, both fastening aperture and the base extension tab are designed to match and work together. Generally, the base extension tab is formed with a similar contour as the fastening aperture. The shape of the fastening aperture on the housing also contributes to determining the shape of the central portion (shown shaded). The wedge shape of the central portion allows it to engage the triangular fastening aperture. It can be seen from the figure that making the fastening aperture a triangular shape leads to a complementary contoured triangular base extension tab. Varying the shape of the fastening

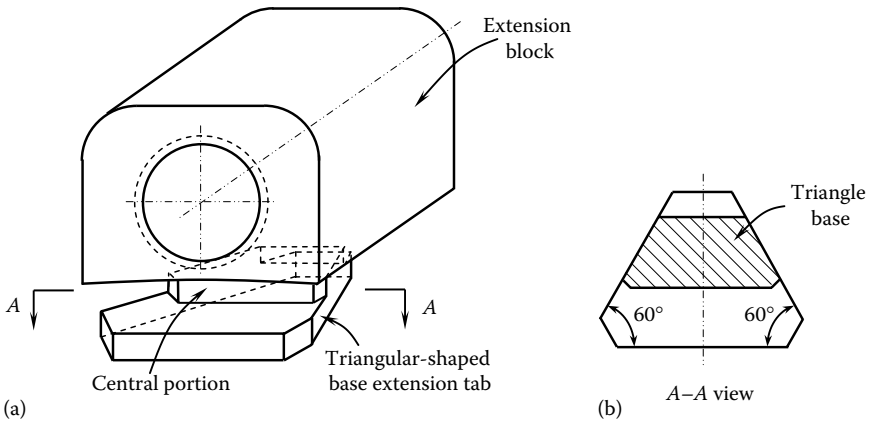


FIGURE 5.27
Triangle-base rotating fastener.

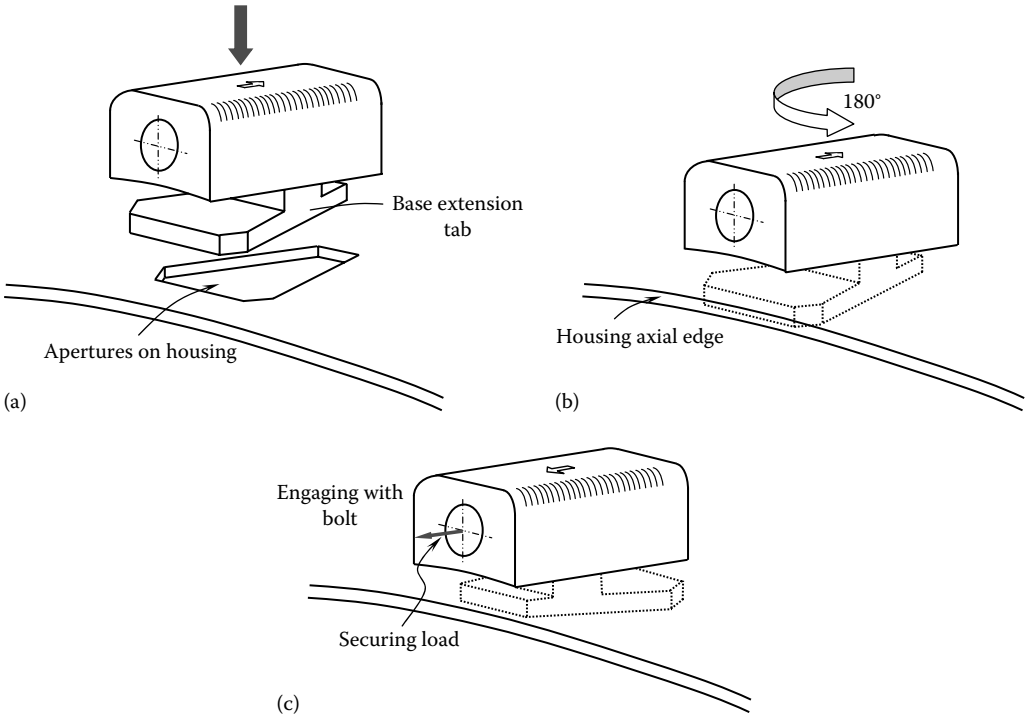


FIGURE 5.28
Fastening mechanism of triangle-base rotating fastener: (a) inserting a fastener into an aperture on the housing, (b) rotating 180° of the fastener, and (c) applying the securing load to the fastener.

aperture leads to subtle implementation variances in the placement of the retention forces securing the elements.

Figure 5.28 illustrates the steps of engaging the fastener having a triangular base extension tab with a motor housing. Initially, as shown in Figure 5.28a, the fastener is positioned above a fastening aperture, thereby aligning the complementary contours of the triangular base extension tab with the fastening aperture. After the elements are aligned, the fastener is inserted into the fastening aperture. In Figure 5.28b through d, the central portion and the triangular base extension tab are represented with dashed lines as they are hidden beneath the fastening aperture and in the interior side of the housing. To ensure a secure connection between the fastener and the housing, the thickness of the central portion is equal to the housing thickness. Figure 5.28b shows the fastener with the central portion, and the base extension tab is inserted into fastening aperture. The extension block prevents the fastener from fully passing through the aperture and advantageously provides additional contact area between the fastener and the housing. As shown in Figure 5.28c, upon insertion, the fastener is rotated by 180° with respect to the axial edge of the housing. The degree of rotation is one of the subtle implementation-specific aspects of the fastening processes referenced earlier.

Figure 5.29 illustrates a fastener viewed from top before and after its rotation. The 180° rotation illustrated creates staggered contact areas for the apex and the base corners of the base extension tab and the triangular aperture. In Figure 5.29, the base extension tab and the complementary aperture are aligned to facilitate insertion of the fastener into the aperture. As shown, the central portion is set back from the leading edge of the base extension tab.

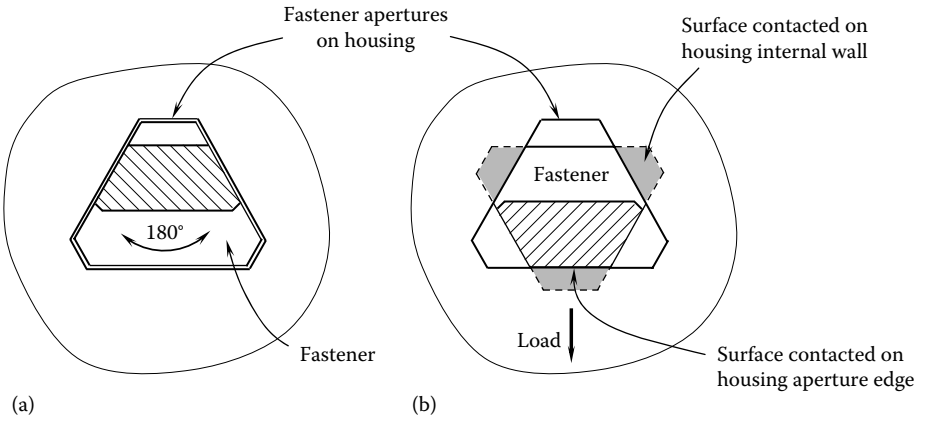


FIGURE 5.29
Installation of triangle-base rotating fastener: (a) inserting the triangle-base fastener and (b) rotating the fastener by 180° so that the three corners of the fastener base come in contact with the interior of the housing (the dark areas).

Rotating the fastener provides three contact areas on the interior surface of the assembly housing, which, in turn, provide additional structural support for the fastening assembly. Specifically, as the hook nut is inserted into the slot and turned by 180°, the three corners of the base extension tab engage the housing's internal wall, thereby providing radial support for balancing the bending moment acting on the fastener as the securing bolt is engaged.

Once the fastener is engaged with the housing, the other elements of the assembly may be secured to the housing. The fastener also includes an extension block that establishes a contact area with the exterior of the housing and engages a securing bolt (not shown). The securing bolt threads through an aperture in the endbell and engages the hollow portion of the extension block. The engaged securing bolt and fastener rigidly secure the endbell between the securing bolt head and the fastener's extension block. It is to be understood that the fastener may work alone or in coordination with other fasteners located at areas along the circumference of the axial end to secure the endbell to the housing.

5.4.4.2 Square-Base Rotating Fastener

A square-base rotating fastener is presented in Figure 5.30. This type of fastener is similar to the triangle-base rotating fastener, except that the central portion and the base extension tab of the fastener are substantially square shaped. As can be seen in Figure 5.30a and b, the central portion of the fastener is set back from the edge of the base extension tab, allowing for an additional contact area when the fastener engages the housing. Figure 5.30c and d illustrates the process of applying the square-base rotating fastener. As discussed previously, varying the geometry of the fastening aperture, the base extension tab, and the central portion leads to subtly different placement of the retention forces. Specifically, as the fastener engages the securing bolt (not shown) and the housing, there are different retaining forces exerted on the interior wall of housing by the square-base extension tab. The square-shaped base extension tab has four independent areas of contact with the interior wall of the housing. Further, due to the geometry associated with this embodiment, the fastener is rotated 45° to engage the housing. The other aspects of engaging the fastener, the housing, the endbell, and the securing bolt are similar to those discussed previously with regard to triangle-base fastener.

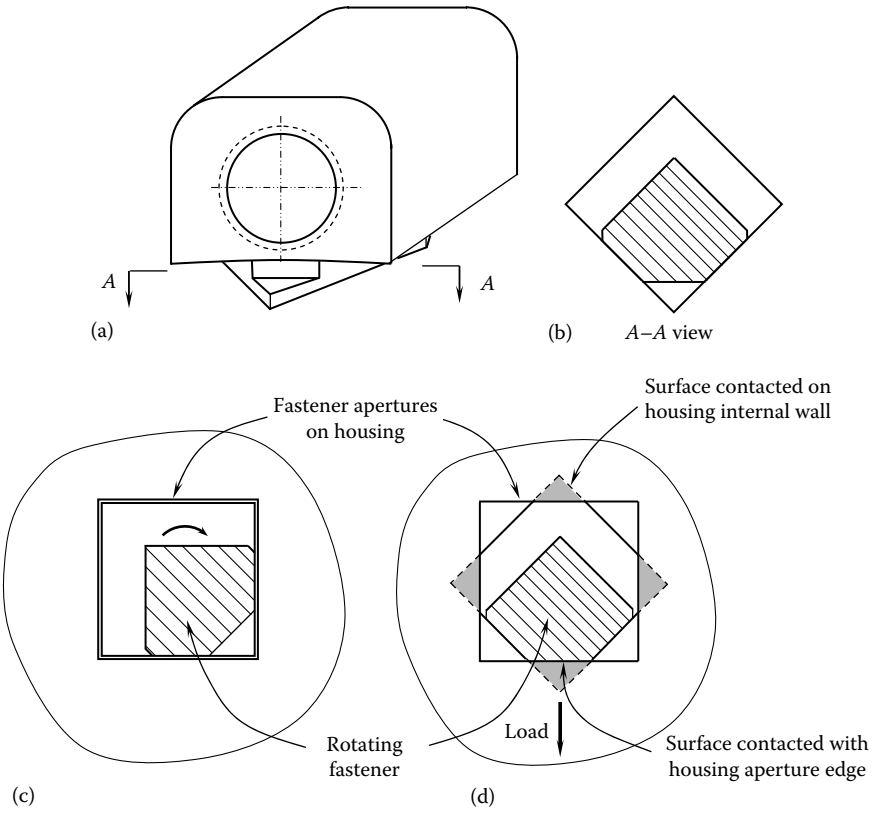


FIGURE 5.30
Installation of square-base rotating fastener: (a) a perspective view of square-base rotating fastener, (b) a cross-sectional view of the square-base extension tab, (c) inserting a fastener into an aperture on the housing, and (d) rotating 45° of the fastener and applying the load on the fastener.

5.4.4.3 Butterfly-Base Rotating Fastener

A butterfly-base rotating fastener is demonstrated in Figure 5.31. It is similar to the triangle-base and square-base fasteners, except the base extension tab is substantially butterfly shaped. The contact surface to the housing is shown at the bottom horizontal wedge of the central portion.

Figure 5.31a illustrated a perspective view of the butterfly-base rotating fastener, and Figure 5.31b presents the cross-sectional view of butterfly-base extension tab taken along lines *A–A* in Figure 5.31a. Figure 5.31c and d presents the top views before and after the rotation of the fastener, respectively. The base extension tab and the complementary aperture are aligned to facilitate insertion of the fastener into the aperture. The final orientation of the fastener after its 90° clockwise rotation is shown in Figure 5.31d. It shows the interior contact areas at the front and the back of the fastener aperture.

5.4.5 Other Types of Fasteners

There are other types of fasteners that may be applied in motor assemblies, each type being appropriate for certain types of motors.

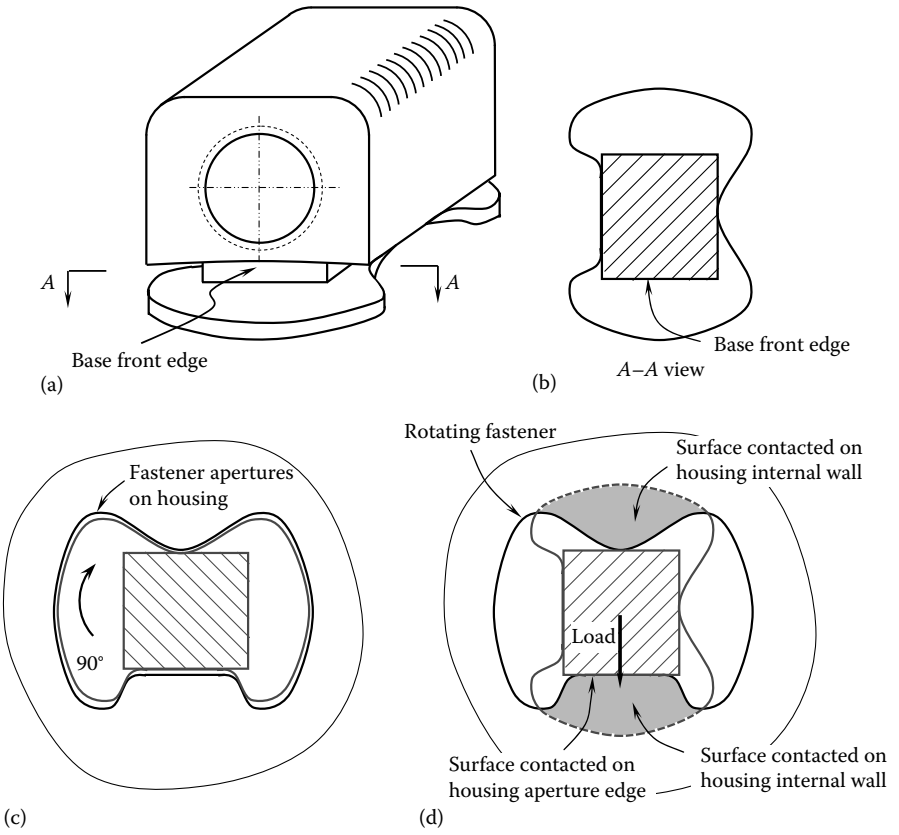


FIGURE 5.31
Installation of butterfly-base rotating fastener: (a) a perspective view of butterfly-base fastener, (b) a cross-sectional view of the butterfly-base extension tab, (c) inserting the fastener into an aperture on the housing, and (d) rotating 90° of the fastener and applying the load on the fastener.

5.4.5.1 Cylinder-Base Fastener Locked with Retaining Ring

Figure 5.32 demonstrates a cylinder-base fastener that is locked with a retaining ring. The fastener has two grooves at the cylindrical base to receive the retaining ring. To ensure normal performance of the fastener, the fastener base is designed to have a sliding fit with the hole on the motor housing, and the surface of the retaining ring must be in contact with the inner wall of the housing. This type of fastener may lead to less production costs associated with the simple elements, as well as a less complex manufacturing and assembly process.

An alternative locking mechanism may also be implemented as a securing pin engaged with a hole through the cylindrical base.

5.4.5.2 Cylinder-Base Fastener with Self-Locking Aperture

This design is essentially similar to the fastener design addressed previously. As shown in Figure 5.33, instead of using a retaining ring to lock the fastener, this type of fastener can be self-locked by engaging an aperture on the motor housing. For this purpose, the

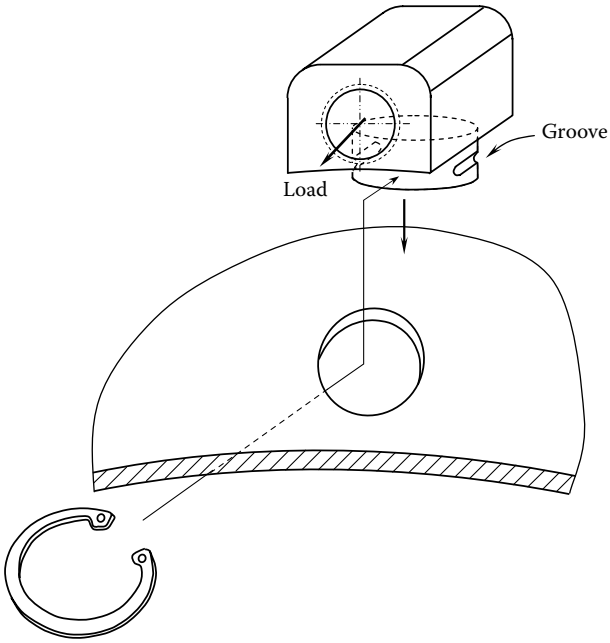


FIGURE 5.32
Cylinder-base faster, locked with a retaining ring at the grooves of the cylindrical base.

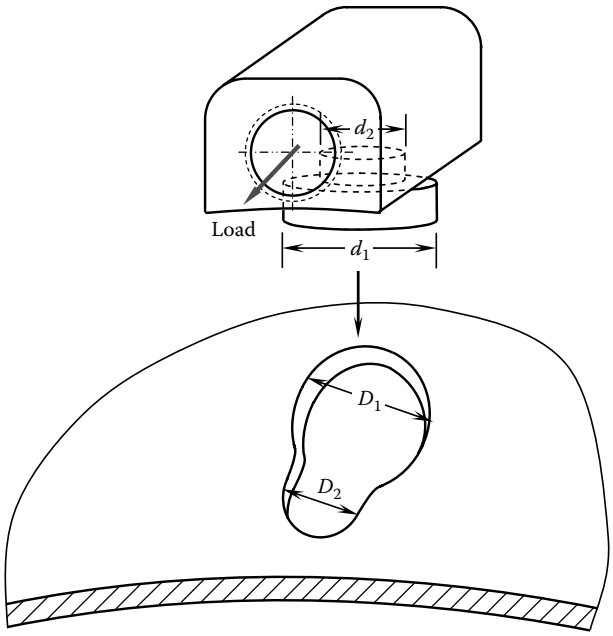


FIGURE 5.33
Cylinder-base fastener with a self-locking aperture on the motor housing.

Hesai Exhibit 1031, page 323

fastener consists of two cylinders having the diameter d_1 and d_2, respectively. The self-locking aperture consists of two circles that have the diameter D_1 and D_2, respectively. The two circles are connected smoothly by two arcs to form a pear-shaped aperture, with the smaller end disposed toward the axial end of the housing. The diameter d_1 is designed to be a little smaller than D_1, and d_2 is a little larger than D_2. This geometry facilitates the fastener's insertion through the larger circle of the aperture and the engagement to a securing bolt (not shown). With the tightening load applied on the fastener, the small cylinder achieves an effective interference fit with the small circle of the aperture.

5.4.5.3 Fastener Engaged with Housing from Housing Interior

Figure 5.34 illustrates a rectangular base fastener inserted into the fastening aperture from the interior side of the housing. As shown in the figure, the extension block of the fastener is inserted along the arrow into the fastening aperture. The continuous base contacts the entire circumference of the interior wall of housing along the fastening aperture. Once the fastener is inserted into the fastening aperture, a securing bolt (not shown) acts to apply a load along the arrow. The securing bolt acts in coordination with the continuous base extension tab to secure both the fastener and the endbell (not shown) to the housing.

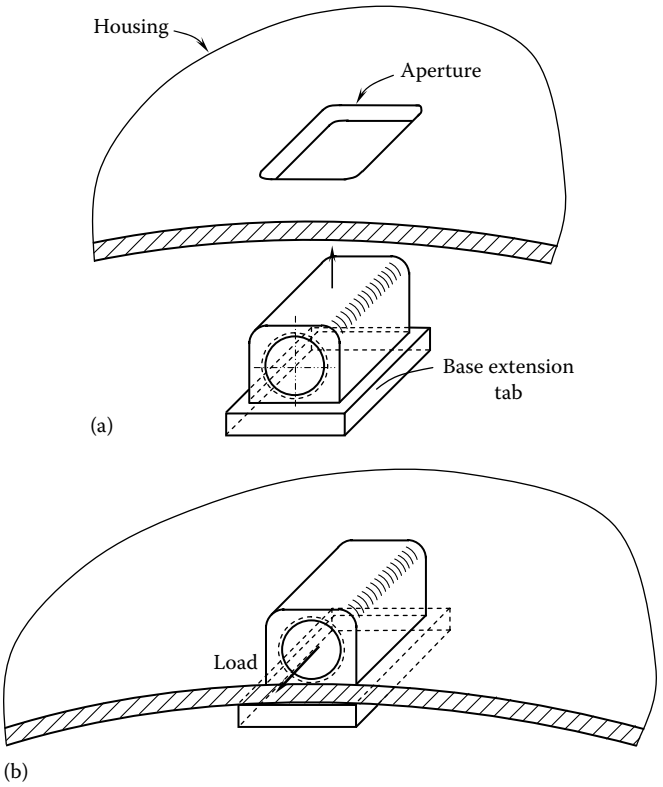


FIGURE 5.34
Installing process of the nut: (a) inserting the nut from the interior of the housing and (b) engaging the nut with the endbell using a bolt.

5.5 Fastening System Design

In the motor industry, it is very common to use fasteners (bolts, nuts, screws, etc.) for joining motor components together and for installing motors into places.

5.5.1 Types of Thread Fasteners

Though a wide variety of threaded fasteners are used in engineering practice, two types of thread fasteners primarily used in motor assemblies and installations are as follows:

- Bolt–nut fasteners, in which a bolt passes through slightly larger holes in two joint members and is engaged with a hexagonal nut. The desired tightness of fastener can be achieved by applying a wrench to either the bolt head or nut. In this way, the two joint members are clamped between bolt head and nut, as shown in Figure 5.35. The grip length l_g is the sum of the thicknesses of both joint members:

$$l_g = l_1 + l_2 \tag{5.5}$$

- Screw-threaded member fasteners, in which the threads of the screw are threaded into an internally threaded hole of the lower joint member, as shown in Figure 5.36. The effective grip length is determined according to the bolt diameter relative to the thickness of the first joint member:

$$l_g = \begin{cases} l_1 + l_2/2 & \text{for } l_2 < d \\ l_1 + d/2 & \text{for } l_2 \geq d \end{cases} \tag{5.6}$$

In both cases, while bolts are subjected to tensile forces, joint members withstand compressive (clamping) forces.

5.5.2 Thread Formation

Fastener threads can be produced by either rolling or cutting operations. Rolled threads are formed by rolling the reduced diameter portion of a shank between two reciprocating serrated dies. The advantages of rolled threads include more accurate and uniform thread dimensions, smoother thread surfaces, higher thread strength (usually 10%–20% in tensile strength), improved fatigue and wear resistance, and material savings. In addition, the grain structure in rolled threads maintains continuous unbroken lines, as opposed to the broken grains in cut threads. Usually, cut threads are used for large diameter or nonstandard fasteners.

5.5.3 Fastener Preload

The preload P_p, or the clamp load, of a fastener is created as a tightening torque is applied on the fastener before receiving any external loads. The tightening process exerts an axial preload tension on the bolt and an axial compression on the joint members, where the generated bolt tension load exactly counterbalances the compression load applied on the assembled

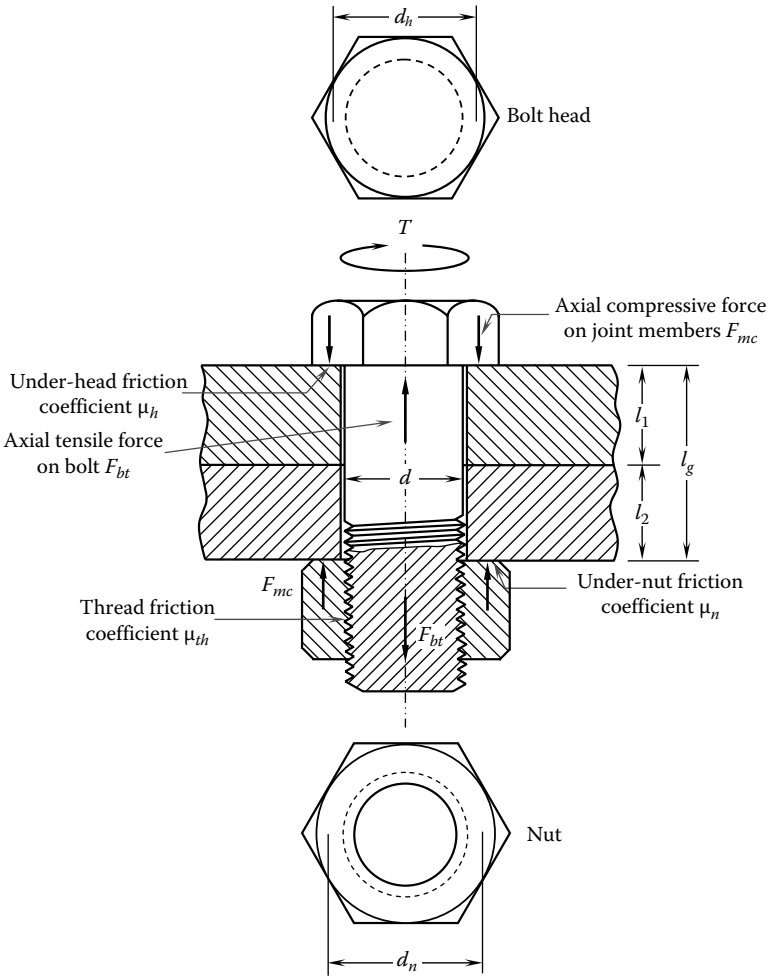


FIGURE 5.35
Bolt–nut fasteners with drilled holes through joint members.

components. It has been reported that when an external load is applied on a preloaded joint, only a portion (usually 80%) of the external load acts on the jointed members [5.22].

In order to achieve high strength and long lifetime of a bolted joint, it is critical to set appropriate bolt tightening torque. Excessive tightening of bolts can cause thread damage or bolted joint failure, and insufficient tightening may lead to separation of jointed members or poor motor performance (such as large vibration). Many surveys have shown that among the possible causes of bolted assembly failure, the most frequent cause is poor assembly. Improper and irregular bolt tightening alone accounts for over 30% of all assembly failures and 45% of all fatigue incidents [5.23].

5.5.4 Fastener-Tightening Process

During a fastener-tightening process, a desired tightening torque is applied to a threaded fastener to stretch the bolt while compressing the clamped joint members elastically according

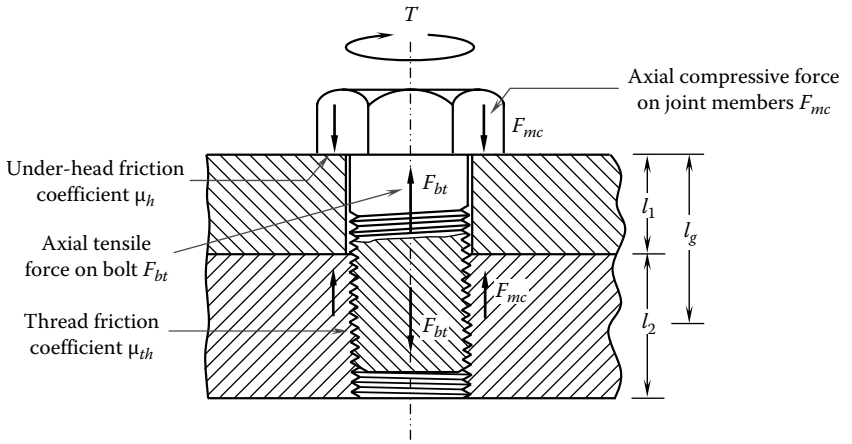


FIGURE 5.36
Screw-threaded member joint. Due to the elimination of a nut, the friction under nut-bearing surface no longer exists.

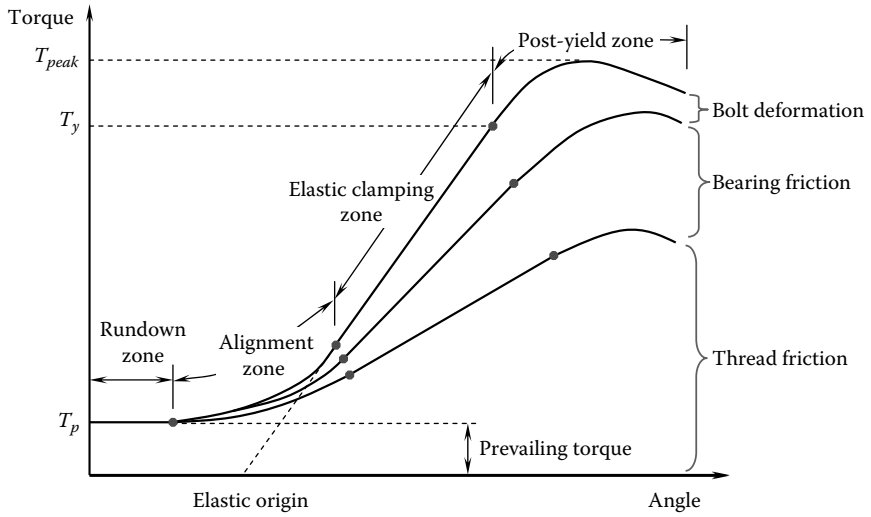


FIGURE 5.37
Four distinct zones in fastener-tightening process.

to the effective spring rates of the bolt and the joint members. This process can be characterized by plotting the input torque against the angular displacement of the bolt. As illustrated in Figure 5.37, four distinct zones can be identified in a fastener-tightening process:

1. Rundown or prevailing torque zone. In this zone, the fastener has not yet touched the bearing surfaces. There is no axial load acting on both the fastener and joint members. The torque needs only to overcome the friction between the fastener threads.
2. Alignment zone. In this zone, the fastener and bearing surfaces are drawn into alignment, and the torque–angle curves is nonlinear. The compressive force

acting on the joint members begins to increase but not at the same rate as the bolt tensile force.

3. Elastic clamping zone. This is the working zone of the fastener. In this zone, the angular displacement of the fastener is proportional linearly to the applied torque on it. The slope of the torque–angle curve at this zone is a function of the stiffness of the assembly and the friction between the threads and on the bearing areas.
4. Post-yield zone. This zone begins with the yield point. Yielding may occur in the bolt or in the joint members or both. Correspondingly, the slope of the torque–angle curve decreases. Continuous increase in the angular displacement will cause the failure of the joint.

The fastener yield torque improvement due to cold working is demonstrated in Figure 5.38. During a tightening process, both the torque applied to the fastener and the angular displacement of the fastener increase simultaneously along the torque–angle curve. Tightening the fastener beyond the yield point but below the ultimate tensile strength can achieve the maximum preload for a given fastener size [5.24]. Due to the plastic deformation of the fastener, releasing the load will not return the fastener to its original state. Then, retightening the fastener again, it is found that the torque and angular displacement increase along the offset curve that is approximately parallel to the first torque–angle curve and the yield torque appearing at a higher level ($T_{y2} > T_{y1}$).

It should be noted that while the yield strength, tensile strength, and hardness of the fastener increase as the consequence of cold working, the ductility decreases along a path opposite to that of hardness. Since the yield strength increases faster than ultimate tensile strength, the yield strength comes closer to the ultimate strength after each cold working cycle. After several repeated load/unload cycles, the fastener exhibits brittle behavior

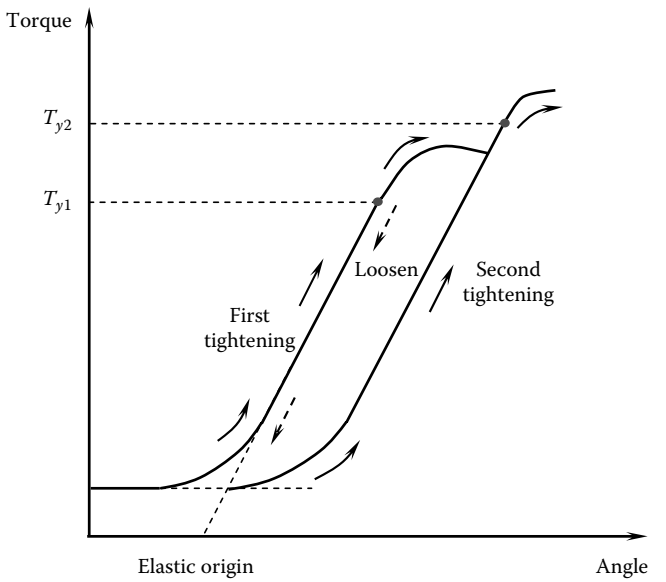


FIGURE 5.38
By loading fastener beyond the yield point, then unloading and reloading again, yield strength, tensile strength, and hardness of the fastener increase, but its ductility decreases.

Hesai Exhibit 1031, page 328

and may suddenly fail without warning signs. Therefore, for critical joints or applications, the reuse of fasteners (bolts, nuts, washers, and other mechanical fasteners) is not recommended.

5.5.5 Tightening Torque

In the elastic clamping zone, when a bolt fastener turns an angle of $\Delta\varphi$ under a tightening torque T, the work done by the tightening torque is (Figure 5.39),

$$W_t = T\Delta\varphi = F_t l\Delta\varphi \tag{5.7}$$

In practice, the work W_t is balanced with three components: the work done by the axial tensile force F_{bt} toward the elastic elongation of the bolt and the clamping force F_{mc} ($F_{bt} = -F_{mc}$) toward the compression of the joint members, W_{def}; the work done by the thread friction between the threads of the fastener system (e.g., bolt and nut), W_{th}; and the work done by the friction on the bearing surfaces under the bolt head and the nut, W_f, as referred to in Figure 5.35.

As shown in Figure 5.40, when a wrench tightens a fastener in a full circle (i.e., 2π in radians) in the elastic clamping range, the corresponding change in the bolt length is the thread pitch p. Thus, as the wrench turns a small angle of $\Delta\varphi$, the change in the bolt length is δ:

$$\frac{\delta}{p} = \frac{\Delta\varphi}{2\pi} \tag{5.8}$$

As a result, the work done due to the bolt deformation becomes

$$W_{def} = F_{bt}\delta = F_{bt}p\frac{\Delta\varphi}{2\pi} \tag{5.9}$$

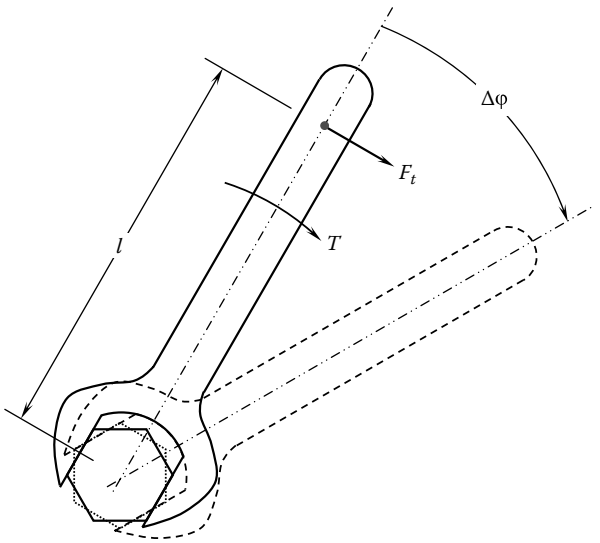


FIGURE 5.39
Work is done by tightening torque over a small angle of $\Delta\varphi$.

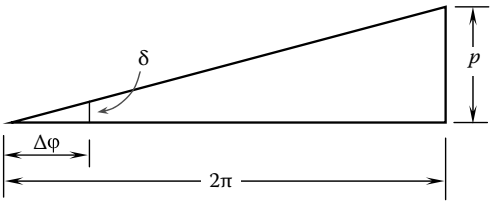


FIGURE 5.40
When a tightening wrench rotates a full circle in the elastic clamping zone, the change in the axial length of bolt is p (thread pitch). As it turns a small angle of $\Delta\varphi$, the change in length is δ.

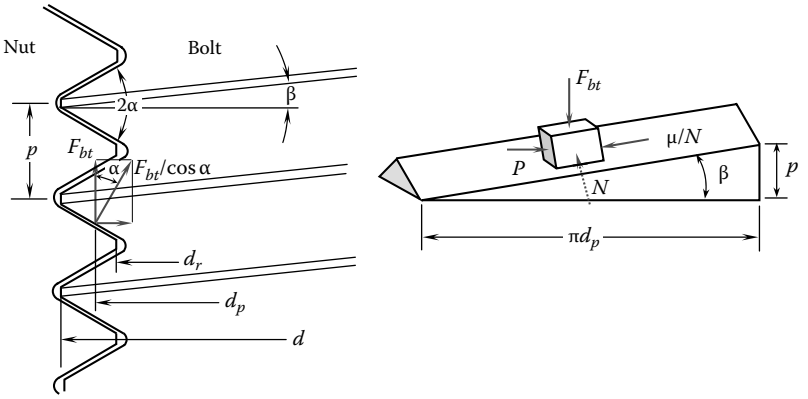


FIGURE 5.41
Work is done by thread friction. Noting that 2α is the thread angle, β is the thread lead angle, d, d_r, and d_p are the major diameter, minor (root) diameter, and pitch diameter, respectively.

During a fastener-tightening process, the thread surfaces of the bolt and nut remain in contact with each other where a relative movement exists. As demonstrated in Figure 5.41, the forces acting on the entire thread surface are the bolt axial compressive force F_{bt}, the tangential force P, the normal force N, and the thread friction force $\mu_{th}N$, where μ_{th} is the coefficient of sliding friction between the bolt and nut threads. It is to be noted that the friction force $\mu_{th}N$ acts on the inclined plane opposite to the direction of motion. In this figure, the inclined angle β denotes the lead angle of the thread, and the angle α is the half of the thread angle. As a function of angle α, the normal contact force N is expressed as $F_{bt}/\cos\alpha$, and hence, the friction force acting on the threads becomes $\mu_{th}F_{bt}/\cos\alpha$. This indicates that an increase in α results in an increase in the friction force.

When the system is in equilibrium, the sum of all forces acting on the system is equal to zero. Therefore, two force–balance equations can be derived as

$$\sum F_h = 0 \quad P - N(\mu_{th}\cos\beta + \cos\alpha\sin\beta) = 0 \tag{5.10a}$$

$$\sum F_v = 0 \quad F_{bt} + N(\mu_{th}\sin\beta - \cos\alpha\cos\beta) = 0 \tag{5.10b}$$

Combining these two equations yields

$$P = F_{bt}\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta} \tag{5.11}$$

Thus, the torque required to overcome thread friction is

$$T_{th} = P\left(\frac{d_p}{2}\right) = \frac{F_{bt}d_p}{2}\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta} \tag{5.12}$$

where d_p is the pitch diameter. Thus, the work done by the thread friction can be expressed as

$$W_{th} = T_{th}\Delta\varphi = \frac{F_{bt}d_p}{2}\left(\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta}\right)\Delta\varphi \tag{5.13}$$

For a very small angle β, tanβ approaches zero; thus,

$$W_{th} = \frac{F_{bt}d_p}{2}\frac{\mu_{th}}{\cos\alpha}\Delta\varphi \tag{5.14}$$

The necessary and sufficient condition for thread self-locking is that the coefficient of thread friction μ_{th} is greater than or equal to the product of the tangent of the thread lead angle β and cosine of the half thread angle α [5.25]:

$$\mu_{th} \geq \tan\beta\cos\alpha \tag{5.15}$$

This indicates that fine-thread fasteners are easier to reach self-locking than coarse-thread fasteners because of their smaller thread lead angles.

Other methods to prevent fastener slackening include using fastener retainers, bolts, nuts, and washers with serrated locking ribs, locked inserts, and adhesive retention materials (such as Loctite).

The work done due to the friction on the bearing surfaces under bolt head and nut becomes

$$W_h = \frac{F_{bt}}{2}(\mu_h d_h)\Delta\varphi \tag{5.16a}$$

$$W_n = \frac{F_{bt}}{2}(\mu_n d_n)\Delta\varphi \tag{5.16b}$$

respectively. In the above equations, μ_h and μ_n are friction coefficients on the bearing surface under the bolt head and nut; d_h and d_n are the effective diameters of the bolt head and nut, respectively:

$$d_h = d_n = \frac{d + 1.5d}{2} = 1.25\ d \tag{5.17}$$

Based on the preceding analysis, it gives that

$$W_t = W_{def} + W_{th} + W_h + W_n \tag{5.18}$$

That is,

$$\begin{aligned} T &= \frac{F_{bt}p}{2\pi} + F_{bt}\frac{d_p}{2}\left(\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta}\right) + 0.625F_{bt}d(\mu_h + \mu_n) \\ &= \left[\frac{p}{2\pi d} + \frac{d_p}{2d}\left(\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta}\right) + 0.625\left(\mu_h + \mu_n\right)\right]F_{bt}d \\ &= KF_{bt}d \end{aligned} \tag{5.19}$$

and

$$K = \sum_{i=1}^{3} K_i \tag{5.20}$$

where K_1, K_2, and K_3 are tightening factors due to the bolt tension, thread friction, and friction on bearing surfaces under the bolt head and nut:

$$K_1 = \frac{p}{2\pi d} \tag{5.21a}$$

$$K_2 = \frac{d_p}{2d}\left(\frac{\tan\beta\cos\alpha + \mu_{th}}{\cos\alpha - \mu_{th}\tan\beta}\right) \tag{5.21b}$$

$$K_3 = 0.625(\mu_h + \mu_n) \tag{5.21c}$$

For metric threads,

$$d_p = d - 0.649519p \tag{5.22}$$

It has been estimated that around 10% of the torque applied to the fastener is to produce the stretch of the bolt and compression of the joint, 40% of the torque is to overcome thread friction, and the remaining 50% is to overcome head- and nut-bearing friction (see Figure 5.37). This implies that approximately 90% of the total torque is to overcome friction, while only 10% does useful work [5.26].

5.5.6 Thread Engagement and Load Distribution

In bolted joints, thread engagement refers to the number or length of threads that are engaged between the bolt and nut threads. It has long been recognized that when a bolt is engaged with a nut, the axial load acting on the bolt and nut is distributed extremely

TABLE 5.2

Loading Distribution of Bolt Threads

Thread Number	Loading Distribution (%)		
	Tseng [5.27]	Timoshenko [5.28]	Young [5.29]
1	58.9	54.5	55.0
2	24.2	24.6	24.8
3	10.0	10.7	11.2
4	4.1	4.7	5.1
5	1.8	2.2	2.5
6	1.0	1.3	1.5

nonuniformly between the threads of threaded joints. During a tightening process, the first engaged thread takes the majority of the applied load. This suggests that the first thread and subsequently some of other threads may yield and proceed to plastic deformation. This plastic softness will redistribute the load over other engaged threads and, as a result, make the loading distribution more uniform.

A simple model has been proposed by Tseng [5.27] for determining the loading distribution on bolt threads. The results of his model are in good agreement with those of Timoshenko and Goodier [5.28] and Young et al. [5.29]. The data in Table 5.2 indicate that extra threads engaged do not increase joint strength.

Assuming both the bolt and nut are made with the same material, the minimum thread engagement length required is approximately 65% of the nominal diameter [5.30]. For instance, a M8 × 1.25 bolt needs a minimum of 5.2 mm of thread engagement, that is, 4.16 threads. In addition, it is appropriate to apply a safety margin of approximately 30% to the calculated minimum length to account for high-order effects.

5.6 Anticorrosion of Electric Motor and Components

Many metallic products produced by manufacturing industries may have bare metal surfaces, which require receiving some forms of protection from corrosion. The classification of corrosion protection methods includes active, passive, permanent, and temporary protection. Active anticorrosion methods can alter or control the proceeding reaction during a corrosion process so that corrosion is avoided. Examples of such an approach are the development of corrosion-resistant alloys and the formation of effective passivation layer on metal surfaces for impeding corrosion half-reactions. In passive protection, corrosion is prevented by mechanically isolating metal surfaces from aggressive, corrosive agents. A coating of this type of protection serves as a barrier layer precluding permeation of corrodent to the metal surface. Among various methods, protective layers, films, or other coatings have been usually chosen by motor manufacturers to provide overall protection against corrosion.

5.6.1 Surface Treatment Methods

By coating a thin layer of a certain metal on a motor component (e.g., a fastener) surface, the surface properties of the component can be changed to that of the metal applied. Thus, the

coated component becomes a composite material, consisting of a durable and corrosion-resistant layer and a core having the load-bearing capability. A number of metals are suitable for protective coating, such as chromium, nickel, copper, zinc, and cadmium. However, coating processes may cause inherent pollution problems.

There are many different methods of metal surface treatment for preventing corrosion on motor components. Metal surfaces can be treated chemically, mechanically, or physically. Some of these methods are addressed in the following sections.

5.6.1.1 Electroplating

Electroplating is achieved by passing an electrical current through a solution containing dissolved metal ions and the metal object to be plated. The most widely used metallic coating method for corrosion protection is galvanizing, which involves the application of metallic zinc to iron and steel products requiring protection from corrosion. Its excellent corrosion resistance in most environments leads to its successful use as a protective coating on a variety of products and in many exposure conditions.

5.6.1.2 Electroless Plating

The electroless plating technology has been developed for many decades. Electroless plating is the process of plating a coating with the aid of a chemical reducing agent (e.g., formaldehyde) in solution without the passage of external power. Compared with electroplating, electroless plating has some superior characteristics: (a) Without using electric current in the plating process, it is easier to obtain uniform coatings on parts. This feature is especially suitable for parts with irregular, complex-shaped geometries. (b) It is applicable to non-conductive substrates such as glass and plastic. (c) Electroless plating can deposit particles from different materials to obtain composite coatings readily for adapting to different application requirements. For instance, nickel–phosphorus coatings are used to enhance corrosion resistance. Alternatively, thin cobalt–phosphorus coatings offer superior sliding wear, enhanced lubricity and corrosion resistance, and improved fatigue properties. Because they exhibit some magnetic properties, they are also of interest to the magnetic recording community.

5.6.1.3 Physical Vapor Deposition

Physical vapor deposition (PVD) refers to the techniques used to deposit molecular thin films on solid surfaces in a vacuum environment. As its name implies, the PVD process is a physical process, without involving chemical reactions as in chemical vapor deposition (CVD). There are two common types of PVD processes: electron beam evaporation process and sputtering deposition process. During the electron beam evaporation process, a solid or liquid coating material evaporates into the vapor phase to travel in the vacuum space and then condensates on the solid surfaces, with the transition of its physical morphology to the solid phase. The sputtering deposition process, which is extensively used in the semiconductor industry, involves ejecting coating materials from the sputtering target to the substrates such as silicon wafers. One of its important characteristics is that it can deposit the materials with high melting points. The most outstanding feature of PVD is that the properties of one material can be imparted to the surface of the workpiece which is made from another material, thus creating a wholly different product. However, the PVD techniques require expensive equipment and instruments.

Furthermore, because the PVD process is usually carried out in a vacuum chamber, the size of the coated members is subject to certain restrictions.

5.6.1.4 Inorganic Coating

Inorganic coating can be produced by chemical reaction, with or without electrical assistance. The treatments change the immediate surface layer of metal into a film of metallic oxide or compound that has better corrosion resistance than the natural oxide film and provides an effective base or key for supplementary protection such as paint.

Black oxide is a process that provides a conversion coating on surfaces of ferrous alloys. The coating is formed from chemical reactions between the iron of ferrous alloys and black oxide solutions such as the alkaline aqueous salt solution. When workpieces are immersed in a hot black oxide solution at the operation temperature of 132°C–143°C (270°F–290°F), a layer of magnetite (Fe_3O_4) is produced on the workpiece surfaces about 30 min.

Unlike other coating processes, black oxide has some specific properties and characteristics that make it ideal for certain applications. One of the remarkable characteristics is that for a properly controlled coating process, black oxide has little contribution to hydrogen embrittlement. This is because that the black oxide process does not require an acid activation nor is it an electro process. However, if the workpieces are scaled or rusty such that an acid pickle is required, it may lead to hydrogen embrittlement problems.

5.6.1.5 Phosphate Coating

Phosphate coating is a crystalline conversion coating consisting of an insoluble crystalline metal-phosphate salt formed in a chemical reaction between ferrous metals and a phosphoric acid (H_3PO_4) solution containing metallic ions (e.g., zinc, iron, or magnesium). Phosphate coating can serve either as the foundation for subsequent coatings or the protective coating on ferrous parts for corrosion resistance. In addition, phosphate coating is often employed in cold-forging process for lubricity.

The primary types of phosphate coatings include manganese, iron, and zinc coating. Among them, manganese phosphate coating has the highest hardness and superior corrosion and wear resistance. Therefore, manganese phosphate coating is extensively used to improve friction properties of sliding components such as engine pistons, gears, camshafts, and power transmission systems. Iron phosphate coating inhibits corrosion and improves the adhesion and durability of paint finishes. Due to the low cost and moderate corrosion resistance, this type of phosphate coating is basically suitable for indoor equipment. Zinc phosphate coating can provide the highest corrosion resistance. Hence, it is widely used in the automotive industry and in certain sectors of the appliance and electronics industries.

5.6.1.6 Electropolishing

As a reverse plating technique, electropolishing is an electrochemical process that is typically used for surface polishing and corrosion resistance enhancing. While electroplating deposits a thin metal layer on surfaces on either metallic or nonmetallic parts, electropolishing removes surface metal, beginning with the high points within the microscopic surface texture, by using a combination of chemicals rectified electrical current. As a result, surface imperfections (e.g., surface crack, burr) and embedded contaminants (e.g., rust, oxide scale) of the parts are eliminated. Electropolished parts are left in a homogenous and passive condition which enhances surface resistance to corrosion tarnish or oxidation.

These properties are especially evident on stainless steel but also found on carbon steel, brass, aluminum, copper, and inconel. Even though all these materials are more corrosion resistant, electropolishing is generally applied to stainless steel today.

As a nondistorting process, electropolishing is especially suitable for applications in which mechanical deformation is not allowed. Other benefits of electropolishing include deburring, precise dimension control, ultra cleaning, stress sensitivity reduction, and weldability improvement.

5.6.2 Anticorrosion Treatment of Electric Motor

Almost all metals, with the exception of the common precious metals such as gold, silver, and platinum group metals (iridium, osmium, palladium, platinum, rhodium, and ruthenium), oxidize and corrode in corrosive environments, forming compounds such as oxides, hydroxides, and sulphides. In fact, corrosion has a great effect on motor performance and lifetime and may cause serious damage to motors or components in some severe cases. Therefore, corrosion control and treatment are of vital concern in motor design.

There are many anticorrosion methods in industrial technologies such as cathodic protection, chemical conversion, inhibition of surface reaction, rust preventive treatment, thermal spray, paint, electroplate, and anodic protection. Each of these methods can, in a certain extent, extend the life of electric motors. Among all the anticorrosion methods, the surface protective coatings are often adopted by motor manufacturers for their good rust inhibitive properties, durability, easy operation, and no requirement for special equipment.

The materials of the protective coatings can be generally categorized into two types: inorganic and organic materials. The inorganic materials include metal, glass, ceramics, and plastic. The organic materials include paint, resin, paraffin, ointment, rubber, and asphalt. Each of the protective coatings has its own properties and usage scopes. In selecting protective materials, the pros and cons of the material must be completely taken into account.

In practice, motor manufacturers commonly use thermally conductive paints on the external surfaces of motor frames at room temperature. The painting materials include plastic ferrite polyurethane, polyurethane acrylic varnish, expanded graphite, enamel, alkyd and epoxy-based paints. The paint thickness is typically less than 0.5 mm. If a motor has been over painted, the paint thickness may be greater than 1 mm. In such a case, low thermal conductivity paints could weaken the heat dissipation from the motor to the ambient. Black paint is commonly utilized on motor surfaces to achieve higher thermal emissivity (about 0.96) and thus higher thermal radiation from the motor. It should be noted that the primary thermal energy emitted by the motor is in the invisible infrared portion of the spectrum. All paint colors in the visible spectrum have small differences in thermal emissivity. All paint colors in the visible spectrum have small differences in thermal emissivity. For instance, white paint has an emissivity of 0.90–0.95, blue 0.94, green 0.92 and red 0.91.

The thermal spraying process involves the projection of small molten metal particles onto prepared motor surfaces to form a continuous coating. During the process, a certain amount of heat is transferred to the motor housing. It is thus important to access the distortion of the housing as a result of the heat transfer. This technique has been widely used in constructions, oil and gas plants, petrochemical equipment, and off shore wind turbines.

When a motor operates under harsh environmental conditions such as high temperature, high humidity, corrosive media, high G-shock, high contamination and/or high pressure, adequate precautions shall be taken to prevent the motor from corrosion. Conventionally, the motor is well sealed with sealing devices on the shaft, housing, and endbells to avoid moisture, corrosive media, and other contamination particles to enter the motor.

Hesai Exhibit 1031, page 336

In fact, the use of seals can reduce but not completely eliminate moisture to penetrate into the motor. Usually, thermal cycling can develop tiny gaps in a motor. Thus, as the air inside the motor periodically experiences expansion and contraction, a small amount of moisture may be sucked inside the motor. Under this circumstance, the seals act as a barrier to the moisture removal, causing interior corrosion of the rotor, stator, PM, and other components. According to Lenzing and Chen [5.31], shaft seals typically cause a loss of 3%–15% of shaft torque due to friction.

A new technique for providing corrosion protection is to apply a protective coating to all the motor parts, including rotor and stator cores, PMs, windings, housing, and endbells, both internally and externally [5.31]. Using the standard salt-spray test as defined in MIL-STD-202G [5.32], in which testing machines are subjected to a fine mist of salt solution, a conventional motor develops a locked rotor and is unable to operate after 48 h of the salt-spray test. By contrast, the motor treated with this new technique still runs normally after 140 h of the test.

Alternatively, motor manufacturers often use stainless steel to make motor housing and endbells for some special applications, such as food processing–related applications. Stainless steel has long been thought of as the best material for withstanding caustic chemicals and high pressure. It is thus ideal for applications requiring high corrosion and heat resistance, frequent washdown, and long life. The drawback of stainless steel is its high price.

Kollmorgen has recently launched its AKMH stainless steel servomotors (Figure 5.42) that fulfill the European Hygienic Engineering & Design Group (EHEDG), National Sanitation Foundation (NSF), and 3A standards [5.33]. This type of motor is specially designed for food processing manufacturers with superior performance, long-life operation, low maintenance, and great corrosion protection. These IP69K rated hygienic servomotors feature stainless steel housings and can withstand water pressure up to 100 bars.

5.6.3 Hydrogen Embrittlement Issues

During some surface treatment processes, such as electroplating, acid pickling, phosphating, electrocleaning, and heat treatment, atomic hydrogen can enter into metallic materials



FIGURE 5.42
Stainless steel servomotor for food processing applications. (Courtesy of Kollmorgen Corporation, Radford, VA.)

Hesai Exhibit 1031, page 337

(mainly steel) to degrade the fracture behavior or load-carrying ability of these materials. This phenomenon is often referred to as hydrogen embrittlement or hydrogen-induced brittle failure.

For instance, hydrogen can be produced from the reaction of the cathode with metal deposition in a zinc electroplating process [5.34],

$$Zn^{2+} + 2e^- \rightarrow Zn^0 \tag{5.23}$$

$$2H^+ + 2e^- \rightarrow H_2 \tag{5.24}$$

An acid pickling operation prepares the surfaces of workpieces prior to some processes such as electroplating, black oxide, and phosphate coating. The acid pickling operation is usually completed with sulfuric acid, but hydrochloric acid is also used.

In the acid pickling process, oxide and scaling layers are dissolved in acid pickling solutions without generating hydrogen. Hydrogen is formed as acid attacking the base material [5.35]:

$$Fe + 2H_3PO_4 \rightarrow Fe^{3+} + 2H_2PO_4^- + H_2 \tag{5.25}$$

The absorption and diffusion of hydrogen in steel are quite complicated [5.36]. When hydrogen atoms enter and diffuse through steel surfaces, they may present either as atomic or molecular hydrogen or in combined molecular form. Because these molecules are too large to diffuse through steel, pressure builds at crystallographic detects (dislocations and vacancies) or discontinuities (voids, inclusion/matrix interfaces) causing minute cracks to form [5.37]. The relief of hydrogen embrittlement is partially related to the nature of the coating applied. Simply speaking, if the coating is dense and nonporous, the escape of hydrogen may be difficult; if the coating is open and porous, the release of hydrogen is a matter of time and temperature [5.38].

Even today, hydrogen embrittlement still remains a major cause of fastener failure. In a root cause failure analysis, an essential step is to determine the failure mechanism. More specifically, it is important to distinguish hydrogen embrittlement and stress corrosion cracking from each other because both failures are associated with the interaction between hydrogen and metal workpieces (e.g., fasteners) and occur under tensile stresses well below the yield strength of the material. The major differences include [5.39–5.41] the following:

- While hydrogen responsible for embrittlement comes from manufacturing processes such as plating and cleaning, hydrogen in stress corrosion cracking is supplied from the environment.
- Stress corrosion cracking is associated with high-stress workpieces where the core hardness is larger than 32 HRC, more likely >36 HRC.
- Stress corrosion cracking is not only induced by hydrogen; it could be also resulted from various corrosive substances that attack small cracks, leading to the final brittle fracture.
- Though both hydrogen embrittlement and stress corrosion cracking are delayed failures, the delay time for hydrogen embrittlement is likely from a few minutes to 24 h after installation, whereas the delay time for stress corrosion, as the primary indication of this type of failure, is typically longer than 24–48 h after installation.
- Stress corrosion cracking often exhibits more crack branching and less pronounced dimples than cracking produced by hydrogen embrittlement.

TABLE 5.3

Recommended Baking Process by ASTM B850-98

Tensile Strength				
MPa	**ksi**	**Hardness (HRC)**	**Temperature (°C)**	**Baking Time (h)**
1000–1100	145–160	31–33	190–220	>8
1101–1200	160–174	33–36	190–220	>10
1201–1300	174–189	36–39	190–220	>12
1301–1400	189–203	39–43	190–220	>14
1401–1500	203–218	43–45	190–220	>16
1501–1600	218–232	45–47	190–220	>18
1601–1700	232–247	47–49	190–220	>20
1701–1800	247–261	49–51	190–220	>22

A number of methods can be taken to avoid hydrogen embrittlement failure, including the following:

a. The hydrogen removal process must be carried out following the electroplating process immediately. According to the ASTM standard [5.42], the hydrogen removal requires that the electroplated parts must be baked at 204°C–218°C (400°F–425°F) within 1 h after the plating process for at least 4 h. In ASTM B850-98 [5.43], the baking time varies from >8 to >22 h at 190°C–220°C (374°F–428°F), depending on the tensile strength and hardness of plated parts (see Table 5.3).

   It is worth to note that the baking temperature is limited for some metal coatings. For example, at 225°C, the zinc starts to oxidize in air.

b. A stress relief process is necessary to reduce the residual stress in the parts. For low-carbon steels, it requires to anneal 3–4 h at about 400°C.

c. A more direct and effective way for reducing the hydrogen embrittlement risk is to use low embrittling electroplating processes such as special solution compositions and operating conditions, which result in either a lower pickup of hydrogen or a deposit that allows easier removal of the absorbed hydrogen during the baking treatment.

d. For high-stressed parts, it is recommended to use a pickling-free process such as dry sand blasting or other mechanical means.

e. Some coating techniques can eliminate hydrogen embrittlement. One example is called PVD. This is because PVD processes are done in vacuum, and thus, the chance of embrittlement by hydrogen is precluded. Another example is the mechanical galvanizing process. In this process, zinc coatings are applied to workpieces at room temperature without hydrogen embrittlement, as described in ASTM B695-2000 [5.44]. The coating thickness can be controlled between 5 and 100 μm.

f. It is generally agreed that in plating of Cr, Zn, Cd, Ni, Sn, and Pb, hydrogen tends to remain in steel. By contrast, some elements such as Cu, Mo, Al, Ag, Au, and others lead to lower hydrogen diffusion and solubility and, consequently, lower hydrogen permeation through steel surfaces. Under the condition of honoring the technical requirements, it is preferred to use the coating materials that have low hydrogen permeation.

Hesai Exhibit 1031, page 339

g. Use low-strength steel to replace high-strength steel. This can effectively reduce the risk of hydrogen embrittlement.
h. Because coating layers cover the surfaces of plated fasteners, they play the role as a hydrogen diffusion barrier. In fact, the coating thickness affects significantly the rate of hydrogen diffusion. Usually, the maximum coating thickness for fasteners with the hardness <32 HRC is 12 μm and 8 μm with the hardness ≥32 HRC.

While avoiding hydrogen embrittlement in some manufacturing processes, scientists and engineers have also taken full advantage of hydrogen embrittlement in other manufacturing processes. A typical example is the production of micro- and nanoparticles. This technique has been successfully adopted for making rare-earth PMs.

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Hesai Exhibit 1031, page 341

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Hesai Exhibit 1031, page 342

6

Motor Bearing

The purpose of using bearings in electric motors is to support and locate the rotating rotor in both radial and axial directions, maintain a uniform air gap between the rotor and stator over a wide range of operating speeds, reduce friction and wear between the rotating and stationary components, and transfer loads from the shaft to the motor frame and base. It has been reported that two major root causes of motor failure are bearing and insulation failure. Statistical data have further indicated that more than half of all motor failures can be attributed to the bearing failures [6.1]. Therefore, it is important to understand bearing performance characteristics and bearing failure mechanisms.

The most important bearing rating factors are speed and load because they have a great impact on bearing life, especially load. Bearings should have good dynamic operating characteristics over a wide range of speeds, low noise and vibration levels, minimized friction, high fatigue strength, less maintenance, long service lifetime, and low cost. To ensure motor operation, it is critical to select the appropriate bearing type, lubricant, lubrication method, and suitable bearing arrangement, according to different operational conditions and motor applications.

6.1 Bearing Classification

There are basically three types of bearings used in industrial, military, and commercial applications: journal (sliding) bearings, rolling bearings, and noncontact bearings. Each type has different performing characteristics and is intended for different applications. Besides these, there are also some special bearings, such as air bearings, maglev bearings, and superconducting bearings.

6.1.1 Journal Bearing

A journal bearing commonly uses a grease or mineral oil to form a thin layer on the bearing surface, withstanding the bearing load and separating the rotating shaft from the stationary bearing surface (Figure 6.1). The lubricant film thickness t_f is very small in comparison to the shaft radius R. In most lubricated journal bearings with incompressible lubricants, the ratio (t_f/R) is typically on the order of 10^{-3}. Due to the eccentricity and the difference in radius between the shaft and bearing, a convergent clearance is formed between the two components. Hydrodynamic pressure is thus developed in this film. It is the hydrodynamic pressure that allows journal bearings to tolerate dynamic loads or even momentary shock loads reasonably.

This type of bearing is often used for high-power and high-load industrial machinery such as engines, centrifugal compressors, pumps, and motors. A journal bearing consists of a shaft that rotates freely in a supporting metal sleeve. There are no rolling elements in

Hesai Exhibit 1031, page 344

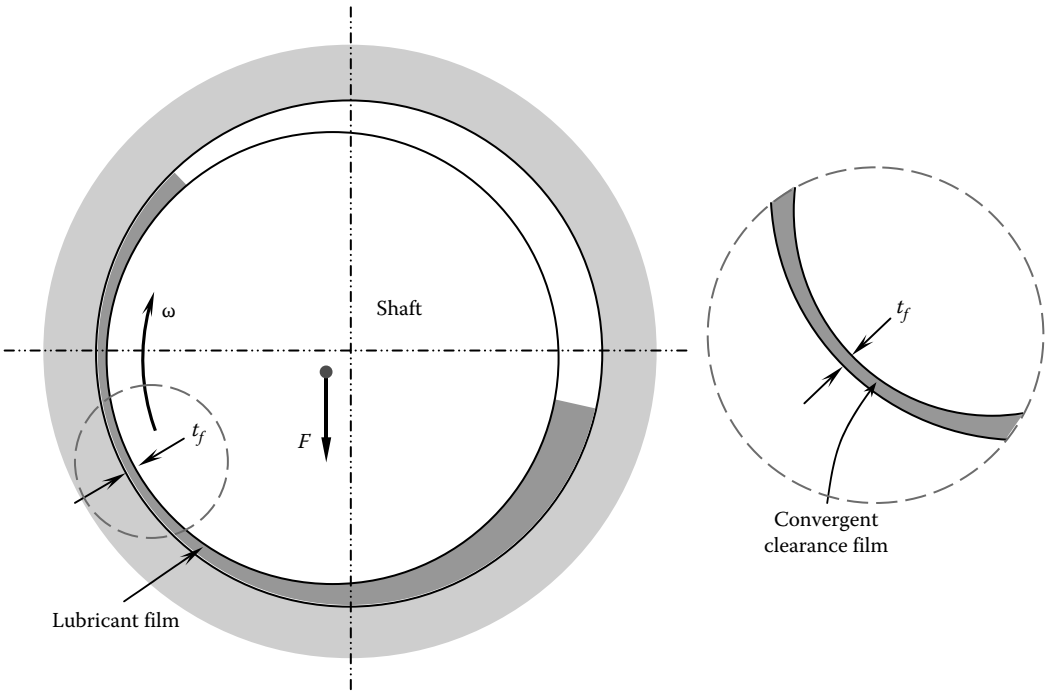


FIGURE 6.1
Lubricant film for supporting motor loads and separating the shaft and stationary bearing surface. Hydrodynamic pressure is developed in the convergent clearance film to lift the shaft.

this type of bearing. Journal bearings have various designs: (1) the self-lubricating bearings, which are made of porous materials and soaked with lubricants. When the temperature rises up due to bearing operation, the lubricant gets out of the pores for playing a role of lubrication. As the temperature decreases, the lubricant is sucked back into the pores due to the capillary effect. The most significant advantage of self-lubricating bearings is lubrication and maintenance-free. (2) The regular lubrication bearings, which require lubricants to form an oil film between the shaft and bearing to support the rotating shaft. (3) The hydrodynamic lubrication bearings, which require lubricants that elevates the rotating shaft by providing the pressurized lubricant from outside. Journal bearings are usually made from cast iron or Babbitt alloys and widely used for engines and large-size, heavy-duty motors.

Journal bearings have some advantages over other types of bearings, including the following:

- Low cost for manufacturing
- Less sensitive to contamination
- Enabling operation under severe conditions
- Requiring small radial space
- Available to work in split halves
- Better shock load–sustaining capacity
- Low operation noise

- Less-precise mounting
- Easy for maintenance

Tribology is a multidisciplinary science to study friction, lubrication, and wear in sliding lubrication systems. In fact, tribological behavior is a complex phenomenon in metallic materials. Many factors, such as loading, lubrication, surface roughness, and relative speed, can significantly affect the interaction between two sliding materials. Stribeck [6.2] had systematically studied the variation of friction between two liquid-lubricated surfaces. Based on the investigations, Stribeck claimed that the journal bearing friction is a function of load, lubricant viscosity, and shaft rotating speed. He showed that the friction in journal-type bearings started with high friction at very low speeds, decreased to a minimum friction when metal-to-metal contact was eliminated, and then increased again at higher speeds, which is well known as *Stribeck curve* today. As depicted in Figure 6.2, there are three friction regions: boundary friction, mixed friction, and fluid friction. In the boundary region, the hydrodynamic pressure between the mating surfaces (i.e., the surfaces of the shaft and journal bearing) has not been developed adequately to support the load. In this region, the lubricant boundary film is partly discontinuous, and the thickness of the film is very thin, approximately 200 Å [6.3]. There are solid contacts in some areas, causing high friction of the sliding system. With the increase in the speed, the film thickness of the lubricant increases rapidly to reduce the direct contact of the shaft and the bearing and consequently reduce the friction until it reaches the minimum value. In the fluid friction region, the film thickness of the lubricant becomes much larger (at least four times) than the surface roughness so that the shaft is completely isolated from the bearing. As a result, the friction between the shaft and the bearing is determined by the dynamic viscosity of the lubricant alone, which increases with the increased speed. The friction factors of different regions are presented in Table 6.1 [6.4] to compare with the dry friction.

The data in Table 6.1 indicate that in any cases, dry friction should be avoided.

Figure 6.2 also provides the comparison of friction between different types of bearings. It exhibits that ball bearings have the lowest friction over a wide range of rotating speeds,

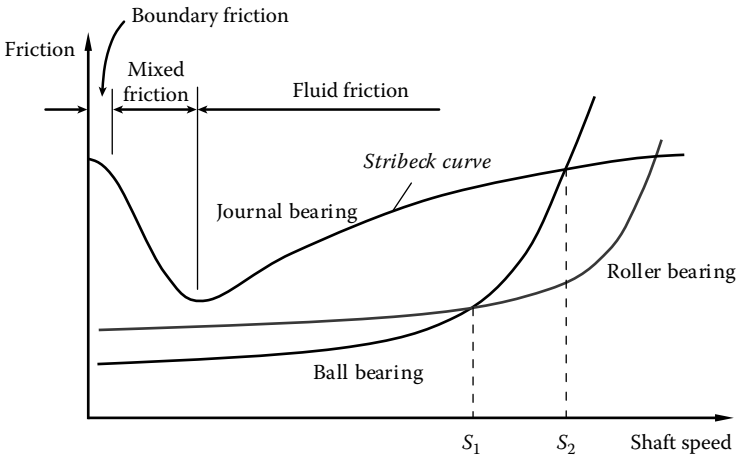


FIGURE 6.2
Comparison of friction between different types of bearings.

TABLE 6.1

Comparison of Friction Factors between Different Friction Regions

Friction Region	Friction Factor, f
Dry friction	0.15–0.8
Boundary friction	0.03–0.15
Mixed friction	0.005–0.03
Fluid friction	<0.005

Source: Mayer, E., *Mechanical Seals*, 3rd edn., Newnes-Butterworth, London, U.K., 1977.

where the rotating speed is lower than a certain value of S_1. Increasing the speed to S_1, the friction of ball bearings increases steeply and thus becomes larger than that of roller bearings. At very high rotating speeds ($>S_2$), the friction of journal bearings becomes lower than that of rolling bearings.

It should be noted that rotating machinery supported by journal bearings presents two kinds of self-excited vibrations, that is, oil whirl and oil whip; both can drive the rotating system to an unstable condition. Oil whirl is commonly masked by the rotor unbalance and occurs near half of the rotating speed, hence being rarely associated with instability problems. Oil whip is a severe vibration that occurs when the oil-whirl frequency coincides with the first flexural natural frequency of the shaft, hence near twice this natural frequency. When operating under this condition, the vibration amplitudes in the bearings are limited by the bearing's clearance. Based on the rotordynamic analysis, two different thresholds of fluid-induced instabilities have been reported by Mendes and Cavalca [6.5].

In designing journal bearings, the bearing static performance characteristics are related to a dimensionless parameter as Sommerfeld number S, which is defined as

$$S = \frac{\mu\omega}{P}\left(\frac{R}{\delta}\right)^2 \tag{6.1}$$

where

μ is the viscosity of the lubricant in Pa-s
ω is the shaft rotating speed in rad/s
R is the shaft radius
δ is the clearance between the shaft and bearing
P is the specific load in Pa

$$P = \frac{F}{ld} \tag{6.2}$$

where

F is the external force acting on the shaft
l and d are the length and diameter of the bearing, indicating the ratio of applied load to bearing projected area

For an unloaded bearing condition, $F=0$ and $P=0$, so that Sommerfeld number S approaches infinity. Thus, a weightless shaft will run concentrically with the bearing.

Either increasing the external load or decreasing the rotating speed will reduce *S*. Under such circumstances, the journal moves away from its concentric position to eccentric positions. As specific load becomes huge or speed approaches zero, *S* approaches zero. Under this condition, metal-to-metal contact occurs between the shaft and bearing.

6.1.2 Rolling Bearing

Rolling bearings have high loading capacity and exhibit very low rolling friction torques. A rolling bearing usually consists of an inner and outer ring, a number of rolling elements, and a cage to hold the rolling elements. Raceways are made on the inner and outer rings, on which the rolling elements rotate freely. The performance of bearing strongly depends on the hardness, surface finish, and accuracy of raceways and other parameters such as internal radial and axial clearance. The ring surfaces must go through grinding without producing waviness. Typically, when the surface roughness is larger than 0.2 μm, the bearing cannot fully play to its carrying capacity. In today's motor industry, the great majority of bearings are rolling bearings.

There are two major types of rolling bearings, distinguished mainly by the rolling element shape: ball bearings and roller bearings. According to the configuration of the bearing rings, ball bearings are further classified as deep-groove ball bearings and angular contact ball bearings. On the other hand, according to the shape of the rollers, roller bearings are classified as cylindrical, spherical, tapered, and needle roller bearings. To distinguish bearings from their functions, the bearing can be divided into radial bearings, which withstand primarily radial loads and axial loads to a certain extent, and thrust bearings, which withstand only axial loads.

6.1.2.1 Ball Bearing

As the most common type of bearing, deep-groove ball bearings are extensively used in electric motors. This type of bearing is not only capable of taking radial loads but also axial loads to some extent. As shown in Figure 6.3, a standard deep-groove ball bearing consists of a number of rolling balls and two concentric steel rings. Each of the concentric steel ring has a hardened ring on which the balls roll. The balls are separated from each other by a cage between the inner and outer rings. The cage is traditionally made of thin steel, but some bearing manufacturers now use plastic materials. To protect against contaminants, the bearing is sealed with plastic covers at the bearing-side surfaces between the inner and outer rings. To reduce the friction and wear in the rolling contacts, bearings are lubricated with greases or mineral oils. From Figure 6.3, it can be seen that a deep-groove ball bearing is symmetric with respect to its center line of cross section and is characterized by a number of primary dimensions, including the following:

- The bearing OD d_o is the fitting dimension with the bearing bore on the motor end-bell or housing, and the bearing ID d_i is the fitting dimension with the rotor shaft.
- d_{ir} is the inner ring diameter at the lowest point of the raceway.
- d_{or} is the outer ring diameter at the highest point of the raceway.
- d_p is the bearing pitch diameter.
- d_b is the ball diameter.
- δ_r is the radial clearance of the bearing.
- w is the bearing width.

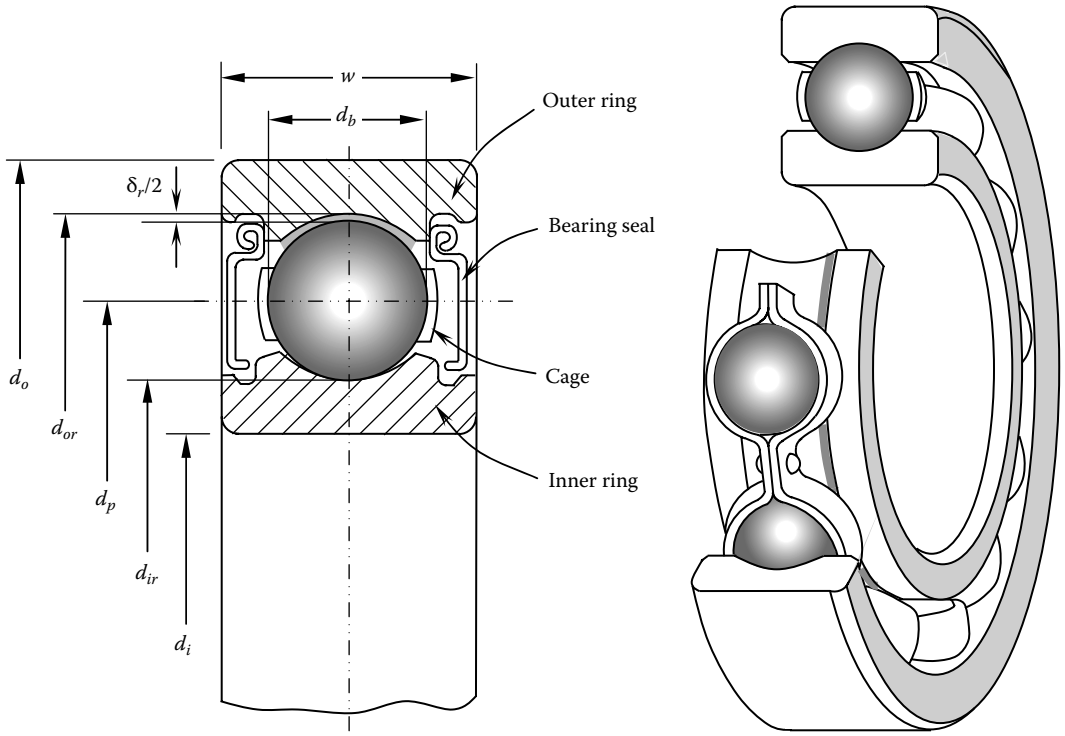


FIGURE 6.3
Deep-groove ball bearings (seals are removed from the right figure).

The bearing pitch diameter d_p is defined as the diameter going through the ball centers. From Figure 6.3, it can be easily derived that

$$d_p = d_{ir} + d_b \tag{6.3}$$

An alternative expression of the bearing pitch diameter d_p can be related to the bearing internal radial clearance δ_r as

$$d_p = \frac{d_{ir} + d_{or}}{2} - \frac{\delta_r}{2} \tag{6.4}$$

One of the key design features of the deep-groove ball bearing is the curvature of the outer and inner rings because they have a strong impact on the bearing stress, friction, and fatigue life. The curvatures of the outer and inner rings are r_o and r_i, as depicted in Figure 6.4. The ratio of the ring radius to the ball diameter is defined as race conformity, which is a measure of the geometrical conformity of the race to the ball in a plane passing through the bearing axis. Therefore, the outer and inner race conformities are expressed as

$$f_o = \frac{r_o}{d_b} \tag{6.5a}$$

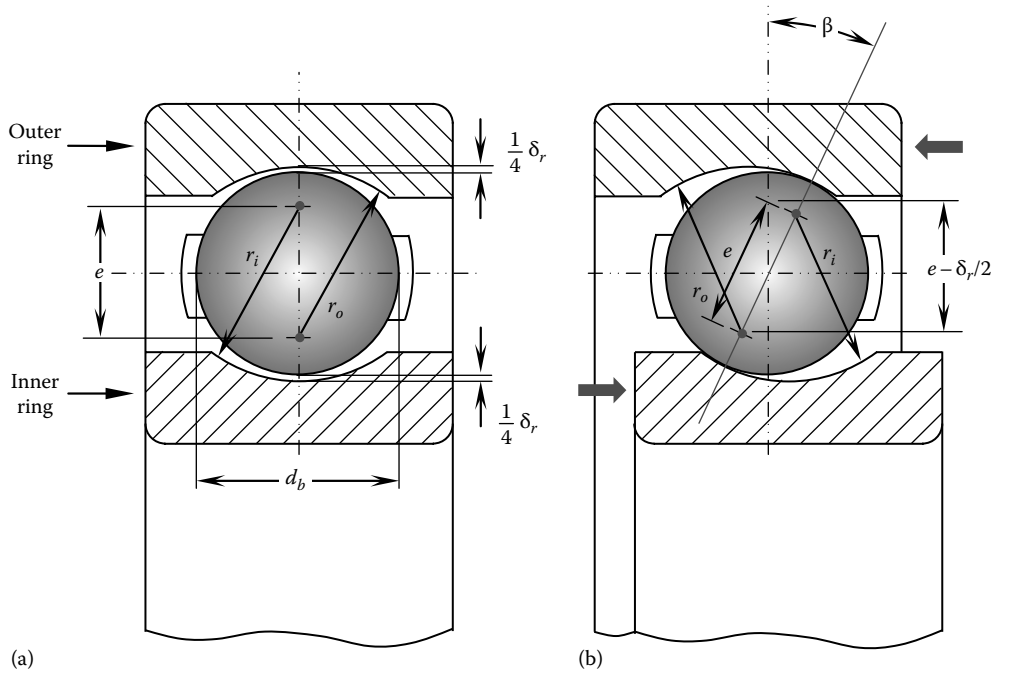


FIGURE 6.4
Cross section of a radial ball bearing, showing inner and outer ring radius (r_o and r_i) and the distance e between the centers of the radii r_i and r_o: (a) initial position and (b) shifted position under axial load.

$$f_i = \frac{r_i}{d_b} \tag{6.5b}$$

respectively. Usually, the outer and inner race conformities are around 52%.

Ball bearings are designed with a specific internal radial clearance because it provides free rotation of balls, compensation for thermal expansion, and optimum load distribution (Figure 6.4a). This parameter can significantly influence bearing operating characteristics. When the bearing is subject to axial load, the inner and outer rings shift relatively in the axial direction. As a result, the internal radial clearance no longer exists (Figure 6.4b).

The bearing contact angle β is defined as the angle between the bearing radial line and the contact line, which passes through the contact points between the rolling balls and the inner and outer raceways. From Figure 6.4b, the contact angle β can be expressed as

$$\cos\beta = \frac{e - \delta_r/2}{e} \tag{6.6}$$

where e is the distance between the radius centers of r_i and r_o and can be determined as

$$e = (r_o + r_i) - d_b \tag{6.7}$$

Combining Equations 6.5a and 6.5b with Equation 6.7, it follows that

$$e = (f_o + f_i - 1)\, d_b \tag{6.8}$$

where (f_o+f_i − 1) is known as the total conformity ratio, which is the measure of the combined conformity of both the outer and inner rings to the balls [6.6].

By optimizing the contact geometry between the balls and rings, the misalignment between the outer and inner rings can be minimized, providing 35% less friction and 50% less noise [6.7]. These optimized bearings are ideal for applications that require precise rotation with large range of torques, such as motors and machine tools. Other design features impacting bearing loss include the ring finish, cage material, and bearing lubrication.

Angular contact ball bearings are specially designed to accommodate combined radial and axial loads. Since this type of bearing uses axially asymmetric rings, the contact line that connects the contact points of the ball and the rings is no longer perpendicular to the axis of the bearing rotation. Rather, it forms an angle with the bearing radial line, called contact angle (Figure 6.5). The standard contact angles are 15°, 20°, and 25°, but the contact angles of 30° and 40° are also available for using in different loading conditions. Generally, the larger the contact angle, the higher the axial load that bearing can handle. It is very important to note that single-row angular contact ball bearings can take axial loads only in one direction. Under reverse loading conditions, the elliptical contact area on the outer ring is truncated by the low shoulder on that side of the outer ring, causing excessive stress and temperature rise at the contact area, and thus leading to high

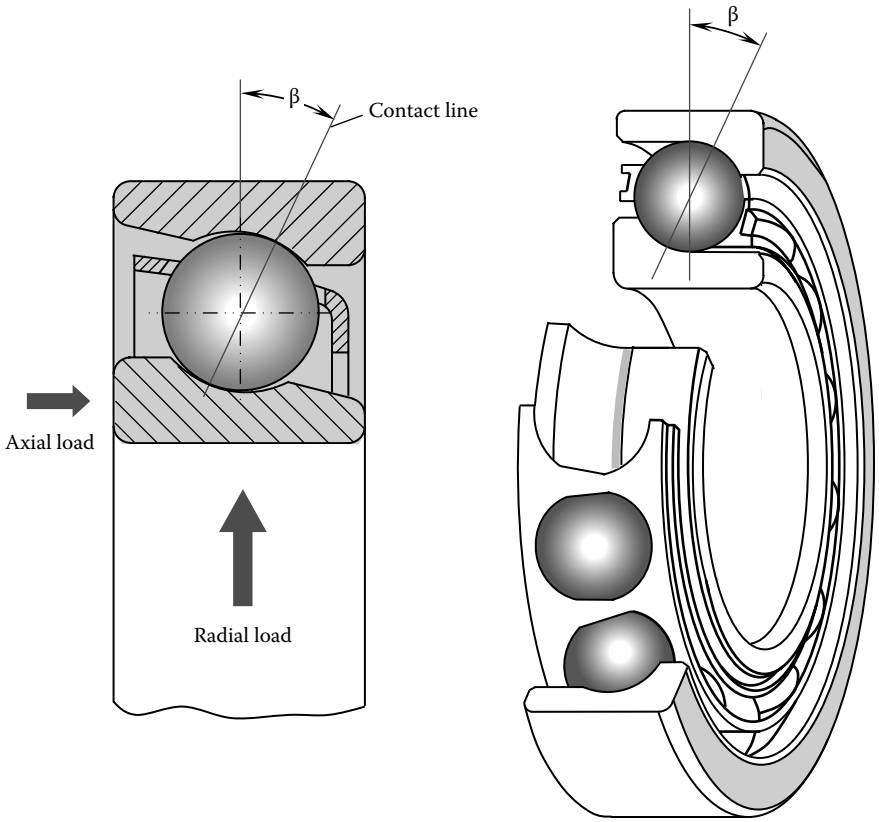


FIGURE 6.5
Angular contact ball bearing with the contact angle β. The bearing shown here has the outer ring cut away reveling the balls and ball cage.

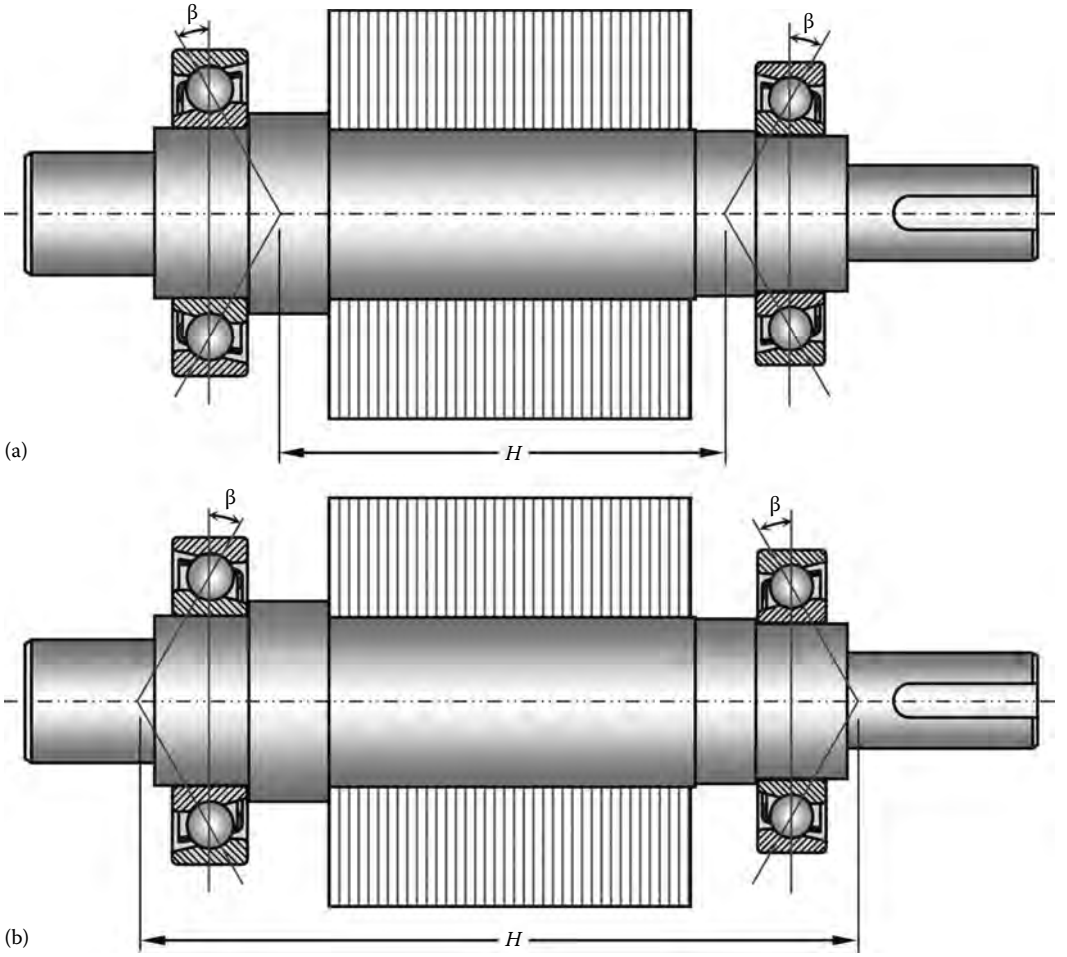


FIGURE 6.6
A pair of angular contact ball bearings used together to support rotor in a motor with two bearing arrangements: (a) contact line inwards (face-to-face); (b) contact line outwards (back-to-back).

vibration and bearing failure. Therefore, at least a pair of bearings must be used together, as shown in Figure 6.6. As can be observed, there are two bearing arrangements: face-to-face (contact line inwards) and back-to-back (contact line outwards). In the face-to-face arrangement (Figure 6.6a), the cones formed by the contact lines point inwards. The axial loads converge toward the bearing axis and may cancel each other out. However, because the support base length H is shorter than the back-to-back arrangement (Figure 6.6b), this arrangement provides a higher tilting clearance (which is defined as the differential mutual ring displacement in the axial direction). As a result, the face-to-face arrangement is not as stiff as the back-to-back arrangement and is not appropriate for the tilting moment loading.

The performance of the angular contact ball bearings is similar to that of the deep-groove ball bearings. However, this type of bearing is limited to lower rotating speeds and has slightly higher noise. For some applications, matched bearing sets that combine two,

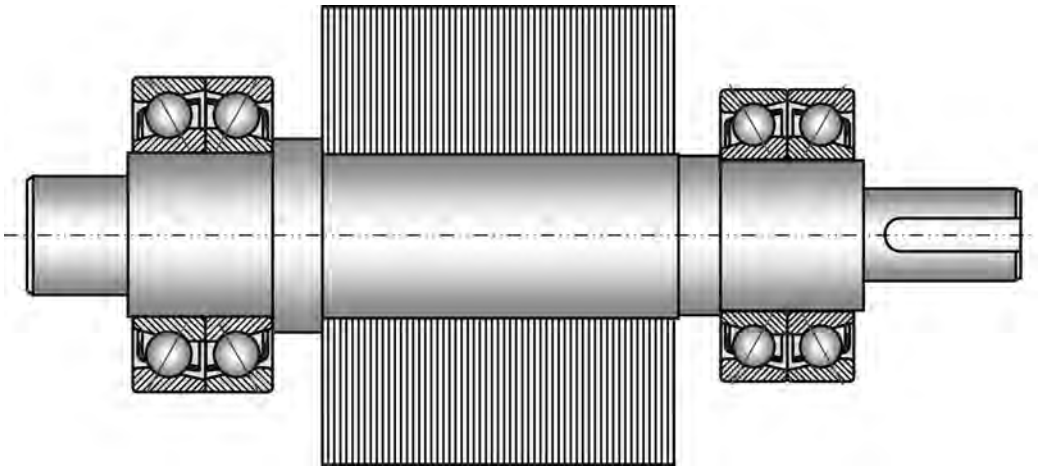


FIGURE 6.7
Matched angular contact ball bearings in a specific bearing arrangement for supporting heavy rotor.

three, or four angular contact ball bearings together may be used to meet special requirements for load capacity and rigidity (Figure 6.7).

The load analysis of an angular contact ball bearing is shown in Figure 6.8. The radial load F_r acting on the bearing is perpendicular to the bearing axis of rotation, while the thrust load F_a is parallel to the axis of rotation. Thus, the resultant force is formed from these two bearing loads. The load angle α is defined as the angle between the resultant load and the radial load, determined as

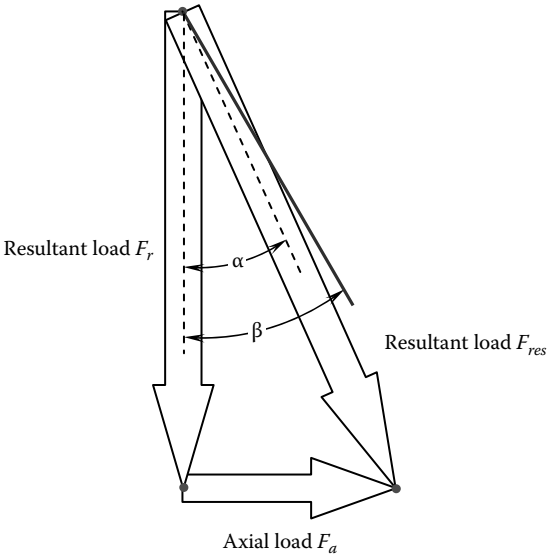


FIGURE 6.8
Loads acting on angular contact ball bearing. The load angle α is determined from the radial and axial load.

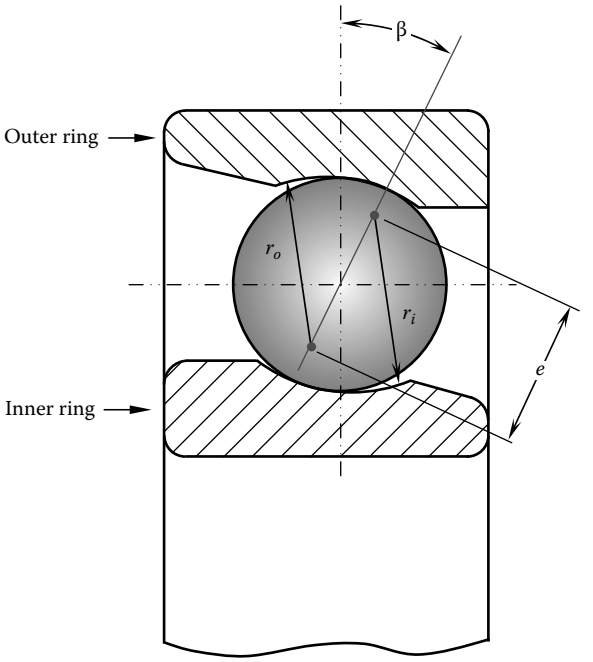


FIGURE 6.9
Cross section of an angular contact ball bearing, showing the outer and inner ring radius (r_o and r_i) and the distance e between the centers of the radii r_r and r_o.

$$\alpha = \arctan\left(\frac{F_a}{F_r}\right) \tag{6.9}$$

The radii of the outer and inner rings in an angular contact ball bearing are shown in Figure 6.9. It is noted that the centers of the radii are located at the bearing contact line.

6.1.2.2 Roller Bearing

In roller bearings, contact rollers may come in a variety of different shapes, including cylindrical, spherical, and conical (tapered roller). In general, the bearing running performance and noise generation depend to a large extent on the dimension precision and surface finish of rollers. Compared with ball bearings, roller bearings have much greater load-carrying capacity but also higher rolling resistance (typically 0.0015 for roller bearings and 0.0008 for ball bearings).

Cylindrical roller bearings improve load-carrying capacity and have been designed to support primarily high radial loads and, to a lesser extent, thrust loads. They have exceptionally low-friction torque characteristics that make them suitable for high-speed applications. The critical design features of this type of bearing include the surface finish, roundness, and waviness of the tracks and cylindrical rollers. As depicted in Figure 6.10, a single-row cylindrical roller bearing consists of an outer and an inner ring and a number of cylindrical rollers for enhancing the load-carrying capacity. The cylindrical rollers are parallel to the bearing axis of rotation and held in position by a bearing cage.

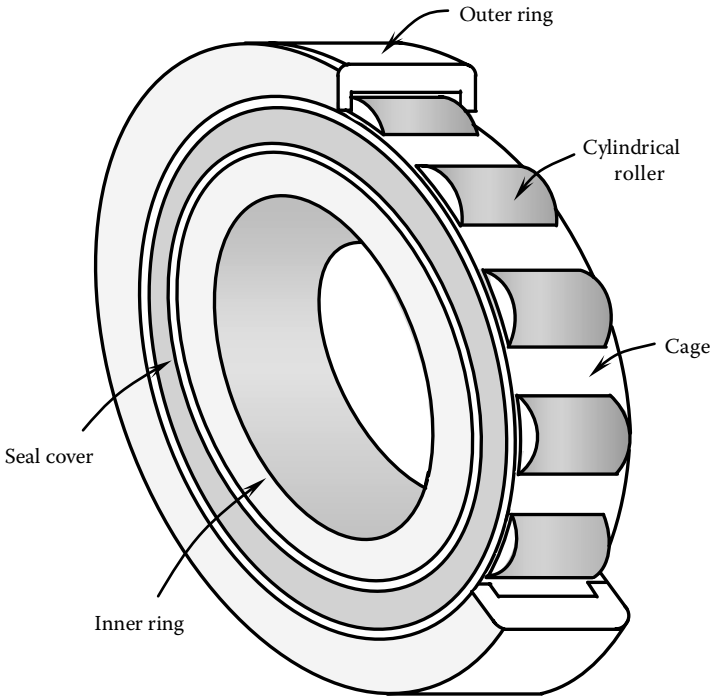


FIGURE 6.10
A single-row cylindrical roller bearing.

Tapered roller bearings have the ability to carry combinations of large thrust and radial loads. As shown in Figure 6.11, a taper angle is formed between the inner and outer ring. Nominal taper angles of 10° and 17° are commonly used in tapered roller bearings.

Thrust bearings are a particular type of rotary bearing, designed to support primarily thrust (axial) loads in a variety of applications (Figure 6.12). There are several thrust bearing types such as ball, crossed roller, cylindrical roller, tapered roller, needle, and spherical roller. Thrust bearings are often used in vertical motors. However, they are typically noisier than ball bearings.

6.1.3 Noncontact Bearing

Noncontact bearings, including air and magnetic bearings, are referred to as floating bearings that maintain a small gap with moving surfaces to completely avoid any kind of solid-to-solid contact under typical operating conditions. Unlike conventional bearings, noncontact bearings prevent problems of friction. Air bearings use a thin film of pressurized air to support a load (Figure 6.13) and offer much higher stiffness than rolling bearings because the air film fully supports the rotating rotor load, as opposed to rolling elements (i.e., balls or rollers) that have point or line contact. As air has a much lower viscosity than liquid, pressurized air bearings have lower load capacity and operate with essentially zero static and dynamic frictions as opposed to the liquid film bearings, which have much higher frictions and pumping losses within the bearings. These features, as well as their high precision capabilities, high damping, and silent operation, make air bearings excellently suitable for high-speed, low-load, and low-noise applications such as

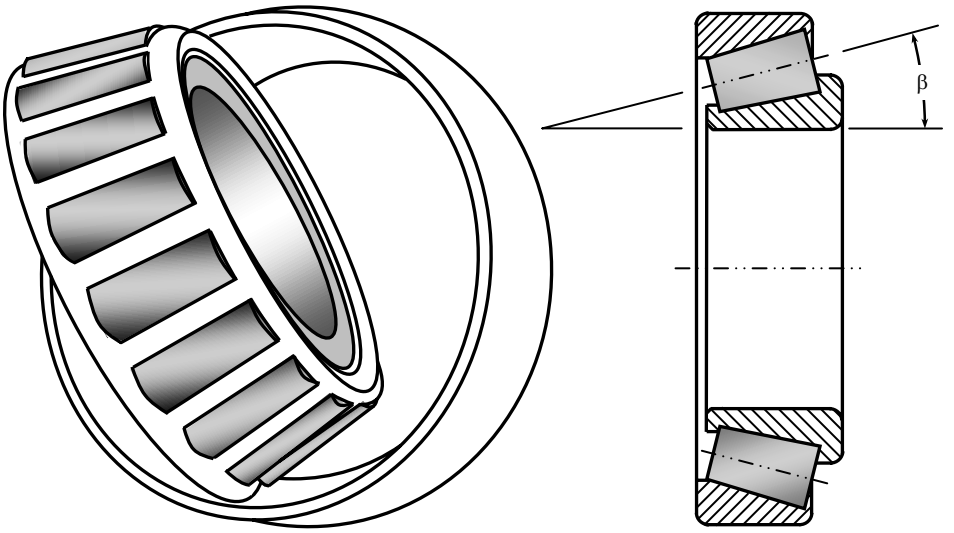


FIGURE 6.11
Tapered roller bearing, which can withstand both radial and axial loads. The cross-sectional view shows the specially designed tapered rollers and demonstrates their angular mounting, which gives their dual load ability.

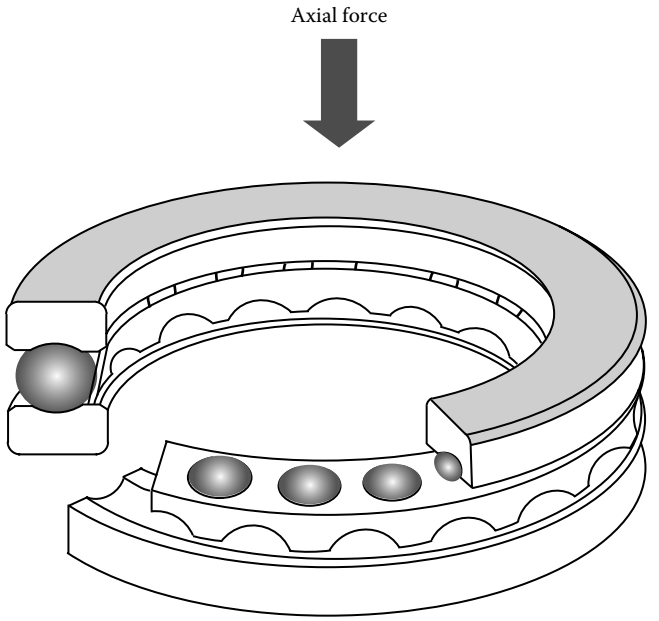


FIGURE 6.12
Thrust ball bearing designed for receiving thrust (axial) loads.

Hesai Exhibit 1031, page 356

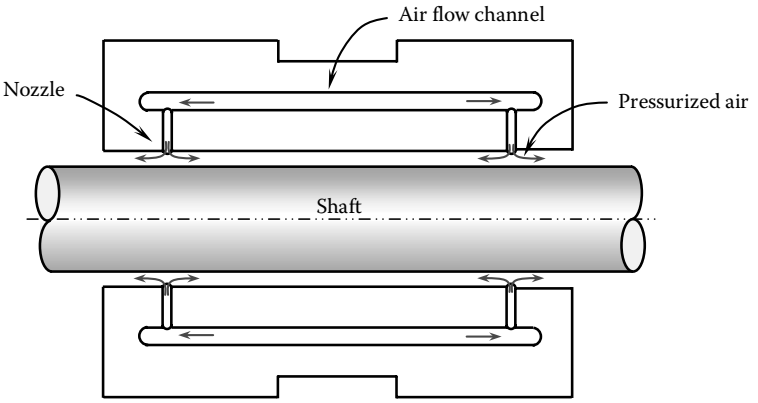


FIGURE 6.13
In air film bearing, load is supported by pressurized air.

dental drills and CMMs [6.8]. However, air bearings require very tight bearing gaps (about 10 μm compared 100 μm for liquid film bearings) and very high manufacturing accuracy. This directly leads to extremely tight tolerances, typically less than ±2.54 μm (±0.0001 in.) for the bearings with air gaps of 12.7 μm (0.0005 in.).

Magnetic bearings can provide superior performance over fluid film bearings and roller-element bearings. They generally have lower drag losses, higher damping properties, and moderate load capacity. The steady-state stiffness of magnetic bearings can be essentially infinite, depending on how the close-loop control system is designed. The dynamic stiffness depends on the frequency of applied load and the bandwidth of the control system [6.9]. Unlike other types of bearings, magnetic bearings do not require lubrication, thus eliminating lubricant, valves, pumps, filters, coolers, and other related components, which are typically responsible for adding system complexity and reducing machine's operating reliability.

Although magnetic bearings can use either attractive or repulsive magnetic forces to elevate the rotating shaft, magnetic bearings are efficient only in the attraction mode. In order to obtain high performance in systems with randomly oriented force components, magnetic bearings should be used in an opposed mode design. Air bearings use air pressure to support the rotating shaft. For this type of bearings, the ratio of δ_r/R is typically 0.0001. Noncontact bearings are often used for extremely high-speed applications such as gyroscopes and dental drills and for precision machinery such as CMM.

6.1.4 Sensor Bearing

In some motor applications, it is required to monitor the operational status of rotor, including rotor position (number of revolutions), rotating speed and direction, as well as acceleration and deceleration. A sensor bearing is made by integrating shielded Hall effect sensors, a magnetic impulse ring, and a versatile ball bearing to form a sensor-bearing unit to detect the required motor operation information and convert it into electrical signals.

As shown in Figure 6.14, while a magnetic impulse ring divided into a sequence of north and south poles is attached to the inner ring of the bearing, small sensors are attached to the outer ring of the bearing. When the bearing inner ring rotates, the impulse ring moves

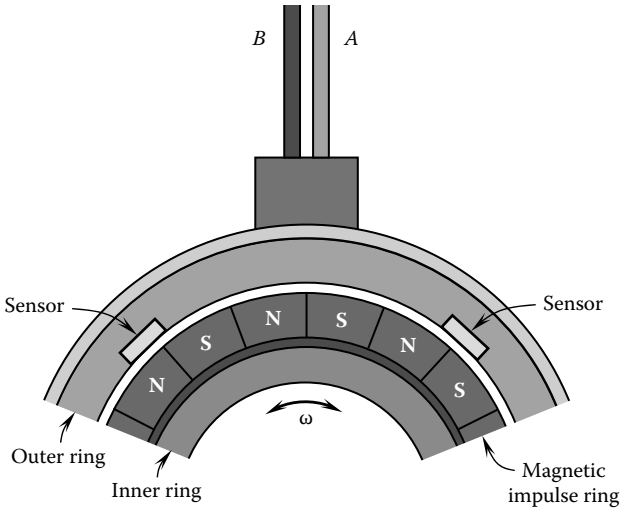


FIGURE 6.14
Operating principle of sensor bearing.

past the stationary sensor ring, generating a magnetic field of alternating polarity. The sensor outputs a pulse whose frequency depends on the number of polarity changes per second. Via the sensor-bearing connection cable, the sensor output signal is transmitted to an electronic unit normally developed by the user. This unit analyzes the signal and provides application-specific information [6.10].

6.2 Bearing Design

Bearing design involves several fundamental elements: selecting correct bearing materials, designing proper bearing lubrication and cooling approach, understanding bearing service environment and operating conditions, and choosing suitable values of bearing design parameter.

6.2.1 Bearing Materials

Bearings materials are classified as through-hardened materials used largely for ball bearings and case-hardened materials used largely for roller bearings. The commonly accepted minimum hardness for steel bearing components is 58 Rockwell, and the hardness for ceramic bearings ranges from 78 to 81 Rockwell [6.11]. For normal applications, Society of Automobile Engineers (SAE) 52100 (100Cr6 in Germany and GCR15 in China) is an excellent general purpose bearing steel, which has a carbon content of 0.95%–1.1% and a chromium content of 1.3%–1.6% [6.12]. This steel grade is generally suitable for applications in which high-strain strength and high resistance to wear under high alternate loads are required. Therefore, the steel is manufactured by induction vacuum melting process for minimizing porosity that might be caused by gas released from smelting process.

Hesai Exhibit 1031, page 358

High-temperature and high-load applications often use hybrid bearings with rings made of SAE 52100 and balls made of ceramic (e.g., silicon nitride Si_3N_4). Under corrosive environment conditions, stainless steel AISI 440C (SUS440C in Japan and 9Cr18 in China) hardened to 58 Rockwell and above is used as the standard rolling bearing material. Due to the lower hardness, the bearings made from this material have a load-carrying capacity 20% lower than those made from SAE 52100.

Nickel–chrome alloys are often used for making bearings. The addition of elements nickel and chrome can increase the hardenability of steel. In addition, chromium brings high temperature strength and resistance to corrosion, oxidation, and abrasion. As a nickel–chrome alloy, SAE 8620 contains about 0.2% of carbon, 0.5% of chrome, and 0.55% of nickel.

Another alternative material is high-nitrogen steel (HNS), which is a new type of alloy with up to 0.9% nitrogen. This material has a good size stability, high strength, and enhanced resistance to pitting corrosion.

Because cylindrical roller bearings can take much higher loads than ball bearings, these bearings are widely used in large motors with heavy loads. In addition, the cylindrical roller bearing allows for floating of the inner race relative to the outer race. In some motor applications, the external load acting on a motor is far from uniform, that is, the motor drive end takes much higher load than the nondrive end. Thus, for such an application, it is common to use a cylindrical roller bearing on the drive end and a deep-groove ball bearing on the nondrive end.

For some special applications, spherical roller bearings may also be used for heavy radial load application. This type of bearing has a good bearing misalignment capability and can carry even more radial loads. However, the performance of sphere bearings is very sensitive to their lubrication conditions, requiring three to five times relubrication than similarly sized cylindrical roller bearings. Statistic results have shown that lubrication failures with these types of bearings are common. Consequently, these bearings are rarely found in motors today [6.13].

6.2.2 Bearing Internal Clearances

Bearing clearance is defined as the total free distance that one bearing ring can be shifted relative to the other either in the radial direction, which is defined as the internal radial clearance δ_r, or in the axial direction, which is defined as the internal axial clearance δ_a, as demonstrated in Figure 6.15. Both internal radial and axial clearances are important parameters that significantly affect bearing operating characteristics such as vibration, noise, lubrication, heat generation, friction, wear, and bearing fatigue life. If the radial clearance of a bearing is too large, it may lead to bearing vibration and inaccurate motion. If the radial clearance is too small, the lubrication film may break down, leading to metal-to-metal contact between rolling balls and raceways.

The axial internal clearance is established by ball diameter, inner and outer raceway radius, and the radial internal clearance. Depending on the bearing geometry, the axial clearance is 8.5–10 times larger than the radial clearance [6.14].

Bearing clearance is categorized as unmounted and mounted (effective) clearance. Unmounted clearance is referred to as the initial bearing clearance, which is provided by bearing manufacturers. As a reference value, the initial bearing clearance is chosen during the motor design and bearing selection process. Mounted clearance is referred to as the operational clearance as the bearing is fitted onto the shaft and into the housing bore and

Hesai Exhibit 1031, page 359

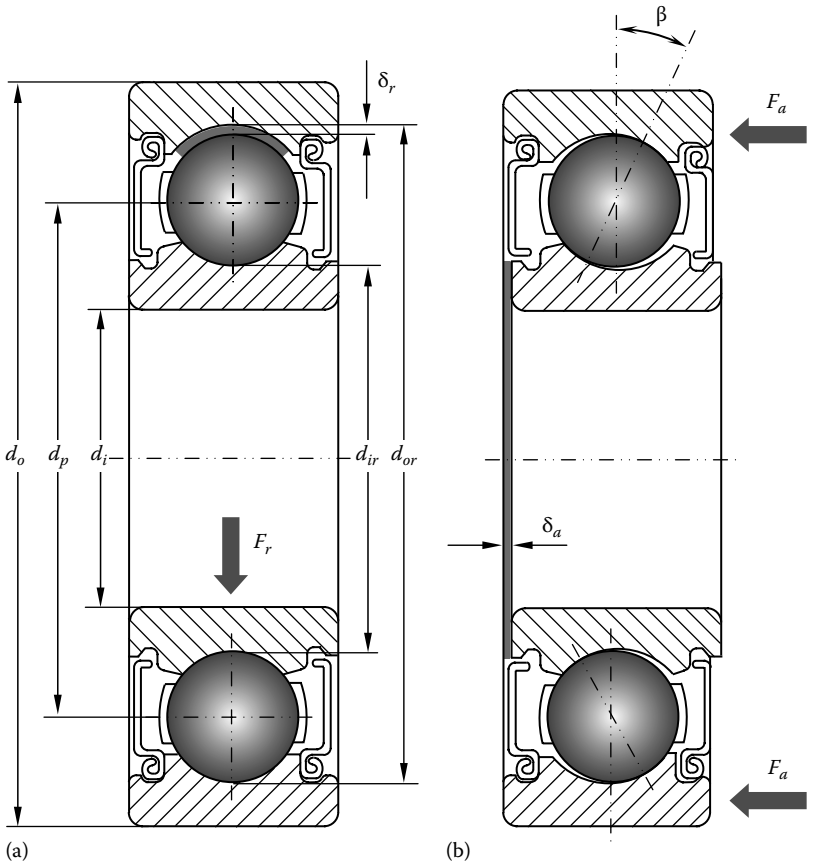


FIGURE 6.15
Bearing clearance: (a) when a bearing is subject to a radial load F_r, a radial clearance δ_r is formed between the inner and outer ring; (b) when a bearing is subject to axial load F_a, the axial clearance δ_a and contact angle β are formed, but the radial clearance disappears.

when the bearing reaches steady-state operating temperature. In practical applications, mounted clearance is difficult to measure in real time. Usually, the operational clearance is smaller than the bearing initial clearance due to several factors including bearing fit and the temperature difference between the inner and outer rings.

Because cooling the stator in a motor is typically easier than the rotor, the outer ring is often cooler than the inner ring by about 5°C–10°C. For water-cooled stator cases, the temperature difference between the two rings can even be greater to significantly reduce the bearing radial clearance. In some cases, the radial clearance can reach zero or even negative values. When the interference between the rolling elements and rings become quite large, it may damage bearing elements, accelerate bearing wear, or even lead to bearing failure. For each size of bearing, there is an optimum range of radial clearance depending on bearing applications.

The internal radial clearance δ_r is determined by the outer and inner ring diameters and ball diameter:

$$\delta_r = d_{or} - d_{ir} - 2d_b \tag{6.10}$$

Hesai Exhibit 1031, page 360

where d_{or} and d_{ir} are the outer and inner ring diameter, respectively. From Equation 6.6, the internal radial clearance δ_r can be also expressed as

$$\delta_r = 2e(1-\cos\beta) \tag{6.11}$$

The interference fits of an inner ring to a shaft and an outer ring to a bearing bore can reduce the bearing radial clearance. As described by Harris [6.15], the increase in the inner ring diameter Δd_{ir} due to the interference fit between a bearing inner ring and a hollow shaft (with the ID of d_2) is given as

$$\Delta d_{ir} = \frac{2\delta d_{ir}/d_i}{[(d_{ir}/d_i)^2-1]\left\{\frac{(d_{ir}/d_i)^2+1}{(d_{ir}/d_i)^2-1}+\nu_b+\frac{E_b}{E_s}\left[\frac{(d_i/d_2)^2+1}{(d_i/d_2)^2-1}\right]-\nu_s\right\}} \tag{6.12}$$

where

δ is the interference between the inner ring and the hollow shaft
v_b and v_s are Poisson's ratio
E_b and E_s are modulus of elasticity of the bearing and shaft, respectively

If both the bearing and shaft are made of the same material, v_b and E_b should be equal to v_s and E_s, respectively. The above equation can be simplified as

$$\Delta d_{ir} = \delta\frac{d_{ir}}{d_i}\left[\frac{(d_i/d_2)^2-1}{(d_{ir}/d_2)^2-1}\right] \tag{6.13}$$

Furthermore, for a solid shaft, $d_2=0$. By means of L'Hôpital's rule, it can be derived that

$$\lim_{d_2\to 0}\left[\frac{(d_i/d_2)^2-1}{(d_{ir}/d_2)^2-1}\right]=1 \tag{6.14}$$

Thus,

$$\Delta d_{ir} = \delta\frac{d_{ir}}{d_i} \tag{6.15}$$

Similarly, the decrease in the outer ring diameter $-\Delta d_{or}$ due to the interference fit between a bearing outer ring and a bearing bore (with the diameter of d_1) on the endbell or housing is given as

$$\Delta d_{or} = \frac{2\delta d_o/d_{or}}{\left[(d_o/d_{or})^2-1\right]\left\{\frac{(d_o/d_{or})^2+1}{(d_o/d_{or})^2-1}-\nu_b+\frac{E_b}{E_h}\left[\frac{(d_1/d_o)^2+1}{(d_1/d_o)^2-1}\right]+\nu_h\right\}} \tag{6.16}$$

where v_h and E_h are Poisson's ratio and modulus of elasticity of the bearing bore, respectively.

If both the bearing outer ring and the bearing bore are made of the same material, $v_b=v_h$ and $E_b=E_h$. The above equation can be simplified as

$$\Delta d_{or} = \delta \frac{d_o}{d_{or}} \left[\frac{(d_1/d_o)^2 - 1}{(d_1/d_{or})^2 - 1} \right] \tag{6.17}$$

Taking into account the influence of interference fits, the bearing internal radial clearance becomes

$$\delta_r = (d_{or} - \Delta d_{or}) - (d_{ir} + \Delta d_{ir}) - 2d_b \tag{6.18}$$

When selecting motor bearings, thermal expansion must be taken into consideration because it can significantly change the internal clearances. A temperature differential between the shaft and the motor endbell/housing always leads to a change in the internal clearances. Ricci [6.6] has investigated the influence of the radial temperature gradient between the inner and outer rings on the internal radial clearance. Assuming the temperature rises in the outer and inner rings over the ambient temperature are $(T_o - T_a)$ and $(T_i - T_a)$, respectively, the increases in d_{or} and d_{ir} are Δd_{or} and Δd_{ir}, respectively

$$\Delta d_{or} = \beta_{or} d_{or} (T_o - T_a) \tag{6.19a}$$

$$\Delta d_{ir} = \beta_{ir} d_{ir} (T_i - T_a) \tag{6.19b}$$

where β_{or} and β_{ir} are the coefficient of thermal expansion of outer and inner rings, respectively. Generally, $\beta_{or} = \beta_{ir} = \beta$.

Assuming the ball temperature takes the mean value of the outer and inner rings, the increase in the ball dimension is

$$\Delta d_b = \beta_b d_b \left(\frac{T_o + T_i}{2} - T_a \right) \tag{6.20}$$

Thus, the change in the bearing internal radial clearance due to the thermal expansion is

$$\Delta d_r = \beta d_{or} (T_o - T_a) - \beta d_{ir} (T_i - T_a) - \beta_b d_b \left(\frac{T_o + T_i}{2} - T_a \right) \tag{6.21}$$

The internal clearance is grouped into different classes, that is, *C*2, *C*3, *C*4, and *C*5, for which the lower number indicates the lower clearance. Some bearing manufacturers designate *CN* as the normal clearance which is between *C*2 and *C*3.

The clearance *C*3 is the most popular clearance used in most motor applications because it permits a sufficiently large tilting clearance even at high speeds. A bearing with a risk of increased thermal stressing for any reasons needs to select the clearance of *C*4.

It is noted from Figure 6.15 that when a ball bearing is subject to an axial load, the inner and outer rings are dislocated axially, resulting in an axial clearance δ_a. Correspondingly, the contact line is no longer perpendicular to the axis of bearing rotation; instead, the contact angle β is formed between the bearing contact line and the bearing normal line. Under this circumstance, the radial clearance δ_r no longer exists. Generally, internal axial clearance δ_a can be established by internal radial diameter δ_r, ball diameter d_b, and outer and inner ring radius r_o and r_i, respectively

$$\delta_a = 2\sqrt{\delta_r (r_o + r_i - d_b)} \tag{6.22}$$

6.2.3 Allowable Bearing Speed

High bearing speed often leads to temperature rise in the bearing due to the friction heating effect. When the temperature becomes high enough, it may cause the decomposition and degradation of the lubricant, thereby deteriorating bearing performance. Therefore, it is a common practice for most bearing manufacturers to set a maximum bearing rotating speed to avoid such detrimental situations. The allowable speed of a bearing varies with the bearing type, size, material, lubrication method, manufacturing precision, sealing type, loading condition, noise level, operation configuration, both bearing and cage structures, cooling condition around the bearing, etc. As the bearing speed exceeds 70% of its allowable speed, high-speed lubricants (grease or oil) of high quality must be applied to the bearing. Generally, ball bearings have higher allowable speed than roller bearings due to their lower friction.

As radial bearings are used in a vertically mounted motor, lubricant retention and cage guidance are not favorable compared to horizontal motor mounting. Thus, the allowable speed should be reduced to approximately 80% of the listed speed [6.16].

6.2.4 Bearing Fitness

For the majority of bearings used in electric motors, the rotating inner rings of bearings are fitted tightly with the rotor shafts via the interference fitting method for ensuring reliable torque transmissions. Meanwhile, the stationary outer rings are fitted less tightly (usually slip fitting) with the stationary bearing bores at the motor frame. This ensures that at least one bearing in the motor is able to slide when the shaft expands or shrinks due to the change in operation temperature. Also, it facilitates the motor assembling and disassembling processes.

It is recommended by bearing manufacturers [6.17] that tight interference fits be used when (1) the operating conditions involve large vibration or shock loads, (2) hollow shafts or housings with thin walls are used, and (3) housings made of light alloys or plastic are used. Similarly, loose interference fits are preferable for (1) requiring high running accuracy and (2) using small-sized bearings or thin-walled bearings.

The fit selection for bearings with shafts and housings is graphically presented in Figure 6.16. The bars designated by lower case letters and numbers (e.g., g6, h6) represent shaft diameter and tolerance ranges to achieve various fitness conditions (i.e., loosen and interference fits) required for various load and bearing rotation conditions. Similarly, the bars designated by upper case letters and numbers (e.g., F7, G7) represent housing bore diameter and tolerance ranges. The recommendations for bearing fits are usually provided by bearing manufacturers in their catalogs for different operating conditions and a variety of bearing types and sizes.

The selection of proper bearing fits is crucial for some motor applications. The selection depends on both bearing and motor operating conditions (such as rotating speed, operation temperatures, load conditions), shaft and housing materials, manufacturing accuracy of mating surfaces, among other factors. Too tight bearing fits can cause excessive stress, reduced radial clearance, and elastic deformation of rolling elements and raceways that may adversely influence the bearing performance. In a worse case, the bearing radial clearance may vanish, turning a loose interference into a tight interference. For such circumstances, the rolling elements may be excessively loaded, leading to rapid wear and temperature rise. On the other hand, too loose bearing fits can cause relative motion between mating parts, resulting in fretting corrosion on the matting surfaces.

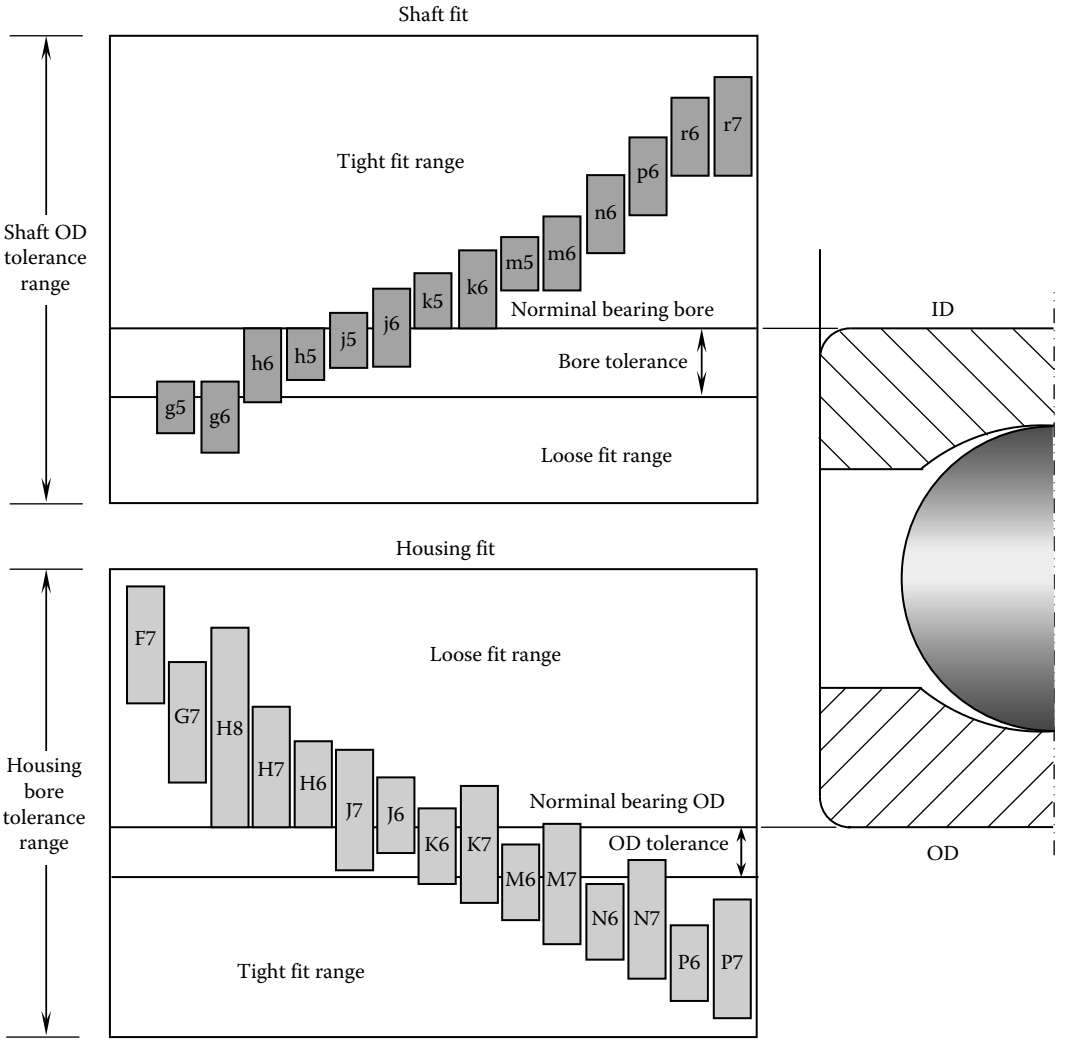


FIGURE 6.16
Fit section for bearings with shafts (top) and housings (bottom).

6.2.5 Prevention of Bearing Axial Movement

In normal motor operation, bearings must be held in their positions firmly. The axial movement of bearing may alter the load distribution on the bearings and thus possibly overload one of them, leading to the damage of the bearing, wave spring washer, endbells, or windings.

A motor shaft is generally supported at its ends by suitable bearings contained in motor endbells or motor frame. Typically, the inner ring of the bearing is pressed onto the shaft, and the outer ring is pressured into a bearing seat defined within an endbell/housing. A number of mechanical methods have been employed to prevent bearing axial movement. The most common way is to use shaft shoulders and retaining rings, which are inserted in machined grooves on the motor shaft to position the bearings precisely.

Hesai Exhibit 1031, page 364

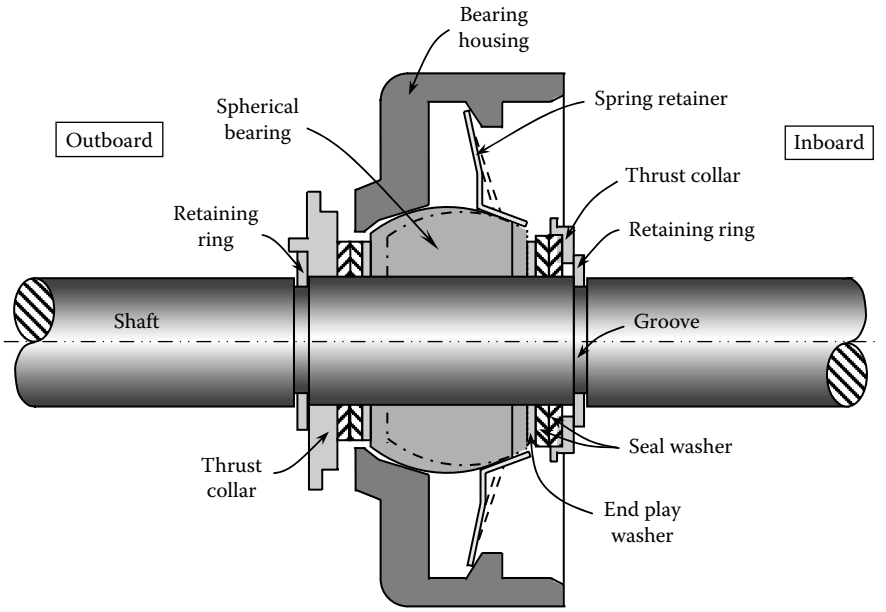


FIGURE 6.17
Using a simplified end play control system for preventing excessive axial movement of shaft and bearing.

In addition, proper fits of the bearing inner ring on the shaft and the outer ring in the bearing bore of the endbell/housing also help hold the bearings in their positions against axial movement.

Bischoff [6.18] proposed a method that uses a simplified end play control system to prevent excessive axial movement of the motor shaft in response to forces applied to the shaft. Figure 6.17 shows an endbell hub assembly, which consists of a rotor shaft, a self-aligning bearing carried on the shaft, and a bearing support structure surrounding the shaft and journaling the self-aligning bearing member. A resilient retainer biases the self-aligning bearing into engagement with its journal. The end play control system includes a thrust collar, a retaining ring, and washers at the outboard (left) and inboard (right) side, respectively. While the thrust collar and retaining ring at the outboard side are for preventing inward movement (i.e., movement to the right), the thrust collar and retaining ring at the inboard side are for preventing outward movement (i.e., movement to the left) of the shaft. It is desirable to insert a seal washer and a pair of washers between the thrust collar and the bearing to position the bearing properly. The seal washer also serves to dampen axial vibration that may occur during motor operation.

An alternative approach was proposed by Grund [6.19], as presented in Figure 6.18. This method uses a bearing holder to fix a bearing to a support structure such as a motor endbell or housing. As can be seen in the figure, the single platelike bearing holder is made from a single metal plate. A plurality of butt straps as a holding structure is formed on the side of the panel opposite to the bottom hooks. The butt straps are arranged to retain the bearing within the bearing holder and ideally to press the bearing axially against the bottom hooks.

Once the bearing has been inserted into the bearing holder with the bearing seated in contact with the bottom hooks, the butt straps are radially deformed to move the fingers radially inwardly so as to axially cover the bearing preventing its removal from the

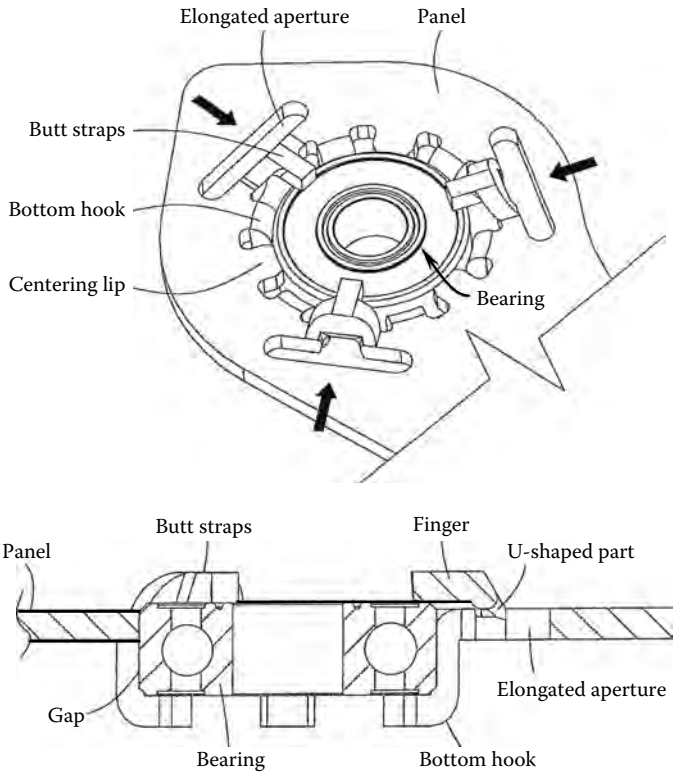


FIGURE 6.18
Bearing holder for fixing a bearing to a support structure using a single platelike body (U.S. Patent Application 2013/0169092) [6.19]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

bearing holder. Centering lips extending inwardly from the hole edge may be provided to fix the lateral position of the bearing.

6.2.6 Bearing Load

Bearing load and speed are the most critical factors in bearing selection and bearing design. To ensure normal operation, both bearing loads and speeds must be within their limits.

6.2.6.1 Bearing Preload Arrangement

Wave spring washers are normally used to provide a preload on the outer ring of one of the bearings for compensating dimensional variations, absorbing impulse forces, and lowing motor operating vibration and noise (Figure 6.19). Many design variations have evolved to best serve these basic functions. The deflection rate for wave spring washers usually range from 20% to 80%, depending on the spring type, structure, material, and number of spring waves and turns (Figure 6.20).

The radial preload is often from an interference fit, usually between the motor shaft and the bearing inner ring and between the bearing bore and the outer ring. In some cases, improper fit can cause relative movements between the mating parts, leading to severe wear on either shaft or bearing born and reduced bearing life.

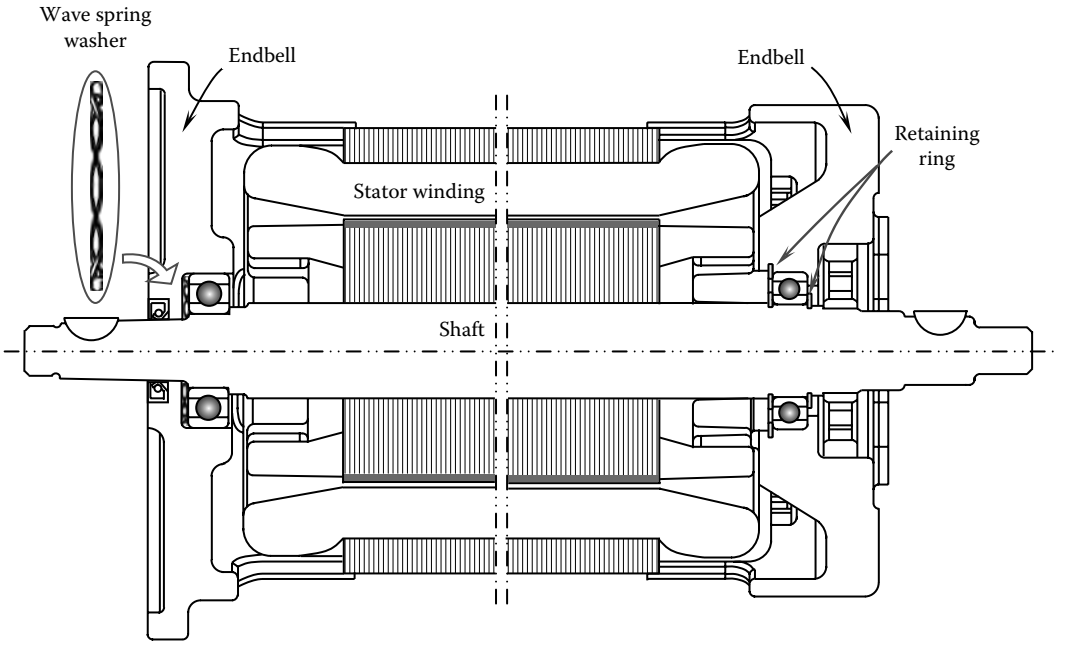


FIGURE 6.19
Using wave spring washer and retaining rings for providing preload to the shaft and positioning bearings axially.

6.2.6.2 Radial and Axial Bearing Load

Even though most radial rolling bearings used in electric motors are designed to take primarily radial load, they can also withstand smaller axial load, as a certain portion of their rated radial load.

The most important parameters of bearing loads are the load magnitudes and their directions. The external load, which comes from the interaction of the rotor and stator, acts firstly on the inner ring and is subsequently transferred to the outer ring through the rolling elements and finally to the bearing bore on the motor endbell or housing.

6.2.6.3 Load Distribution

When a rolling bearing is subject to radial load, the rolling elements are not equally loaded. Usually, at any given time, only less than half of the rolling elements are loaded. The actual load on each rolling element depends on its position in the bearing and other bearing characteristics such as motor internal clearance. Stribeck [6.20] conducted pioneering work on investigating the variation of loads on rolling elements of a radially loaded ball bearing. He found that the load zone for a radially loaded bearing with an internal clearance is less than $\pi/2$, and the load on the most heavily loaded ball is 4.37 times the average ball load for zero internal clearance. Later, Palmgren [6.21] determined that the theoretical value of Stribeck's constant for roller bearings at zero internal clearance is 4.08.

Oswald et al. [6.22] investigated the effect of motor internal clearance on the load distribution and the life of radially loaded ball and roller bearings. Their analysis was extended



FIGURE 6.20
Various types of wave spring washers.

to negative clearance (interference) conditions for producing a curve of life factor versus internal clearance. It was found that for a small negative clearance, rolling-element loads can be optimized and bearing life maximized. To compare with the results from previous investigators, the concept of Stribeck number is introduced as

$$St = \frac{P_{\max}}{P_r / z} \tag{6.23}$$

where
$P_{\max}$ is the maximum element load
P_r is the applied radial load
z is the number of rolling elements

Their analysis confirmed that $St = 4.37$ for ball bearings at zero internal clearance and $St = 4.08$ for roller bearings at zero clearance.

The load distribution of a rolling bearing is primarily determined by the load factor ε, the load angle, and the bearing internal clearance.

$$F(\phi) = F_{\max}\left[1 - \frac{1}{2\varepsilon}(1 - \cos\phi)\right]^n \tag{6.24}$$

where
$F_{\max}$ is the maximum load
ε is the load distribution factor
ϕ is the angle from the bearing lower center line (as shown in Figure 6.21)
$n = 10/9$ for ball bearings and $n = 3/2$ for roller bearings

For a bearing with a normal radial clearance, the maximum load $F_{\max}$ can be determined approximately as

$$F_{\max} = \frac{5F_r}{Z\cos\alpha} \tag{6.25}$$

where
F_r is the applied radial load
Z is the number of rolling element
α is the mounted contact angle

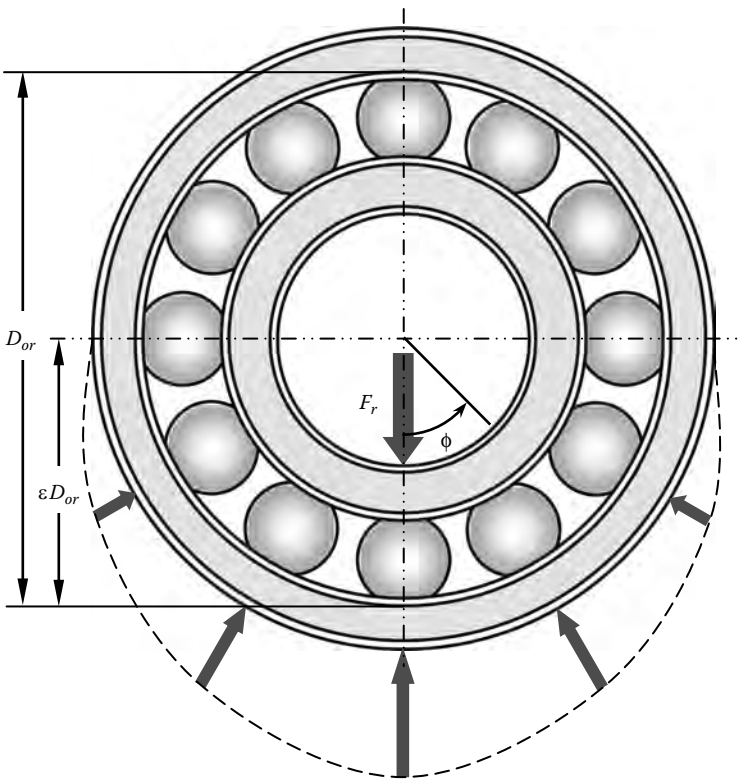


FIGURE 6.21
Load distribution of rolling bearing, $\varepsilon = 0.5$ in this case.

6.3 Bearing Fatigue Life

Each bearing has a limited life of service. The factors that determine bearing fatigue life include (1) bearing rotating speed, (2) external radial and axial loads, (3) bearing radial and axial clearances, (4) lubrication condition (e.g., formation of lubricant film, contamination), (5) lubricant type, (6) operation condition (e.g., temperature, corrosive environment, machine vibration), (7) bearing type and size, (8) bearing material, (9) fitting condition, (10) manufacturing process, (11) bearing installation condition, and (12) motor shaft misalignment. All of these factors must be taken into account in the bearing section and application.

Bearing operating temperature has a strong impact on bearing life by affecting bearing radial and axial clearances, viscosity of lubricant, rate of lubricant volatilization, and bearing sealing performance. Motor vibration and shock loading can not only bring additional loading to motor bearings to deteriorate bearing operation but also alter bearing lubrication condition and thus are detrimental to bearing life.

6.3.1 Calculation of Bearing Fatigue Life

Bearing life is defined as the length of time, or the number of revolutions, until a fatigue spall of a specific size develops. This spall size is usually defined as an area of 0.01 in^2 (6.5 mm^2). This life depends on many factors as mentioned above. Due to all these factors, the life of an individual bearing is impossible to predict precisely. Also, bearings that may appear to be identical can exhibit considerable life scatter when tested under the same conditions.

Basic bearing rating fatigue life or L_{10} is defined as the life that 90% of a sufficiently large group of apparently identical bearings is expected to reach or exceed. By contrast, bearing average life is the life that 50% of the bearings exceed.

Either in theory or in fact, with sound and robust design, correct bearing material selection, accurate manufacture, proper lubrication, and diligent maintenance, bearing life can be extremely long. A calculation was made by Orlowski [6.23] for a medium-duty ball bearing with a dynamic load ration of 7000 lb at an operating speed of 1800 rpm; the bearing life can be as high as 362 years!

Though there are several bearing fatigue life calculation methods available, the formula that is widely used by bearing manufacturers is given as

$$L_{10} = a\left(\frac{C}{P}\right)^b \frac{B}{N} \tag{6.26}$$

where

C is the radial-rated bearing load rating
P is the dynamic equivalent radial load applied on the bearing
B is the factor that depends on the method and units used in the equation
N is the bearing rotating speed
a is the life adjustment factor
$b = 3$ for ball bearings and $b = 10/3$ for roller bearings

In 2007, ISO [6.24] specifies methods of calculating the basic bearing rating life L_{10} of rolling bearings in its standard ISO 281: 2007, with commonly used high-quality material, good

manufacturing quality, and conventional operating conditions. By introducing a correction factor a_{ISO}, which provides an estimation of the influence of lubrication and contamination on the bearing life, the adjusted rating life (in one million revolutions) for a different level of reliability is given as

$$L_{n,a} = a_1 a_{ISO} L_{10} \tag{6.27}$$

where a_1 is the life adjustment factor for reliability. The value of a_1 is entirely determined by L_n, where the subscript n represents the failure probability, that is, the difference between 100% and the requisite reliability. For instance, $a_1 = 1$ for L_{10} (i.e., 90% reliability) and $a_1 = 0.21$ for L_1 (i.e., 99% reliability). The introduction of a_{ISO} is based on the assumption that the fatigue load is directly linked with the bearing's static capacity and the contamination factor is constant whatever the lubrication conditions.

Another popular bearing life formula was also released by ISO in 1978, which is still used today in industrial applications. This model uses three adjustment factors for taking into account reliability, bearing properties, and operating conditions, respectively [6.25]:

$$L_{n,a} = a_1 a_2 a_3 L_{10} \tag{6.28}$$

where a_1 is the adjustment factor for reliability as in Equation 6.27. a_2 is the adjustment factor for bearing special properties such as material processing, forming method, heat treatment, and other manufacturing methods. For standard bearing material and manufacturing method, $a_2 = 1$. As special steel or advanced manufacturing method (e.g., vacuum degassed or vacuum melt) is adopted for improving bearing quality, $a_2 > 1$. a_3 is the adjustment factor for operating condition, which takes into account a wide range of operating, mounting, and application conditions, as well as bearing features and design. For instance, it adjusts the impact of lubrication, contamination, mounting conditions, misalignment, etc., on fatigue life. Under ideal lubrication where the rolling elements and raceways are completely isolated by lubricant films and normal operation conditions, $a_3 = 1$. For poor lubrication cases, such as incomplete lubrication or low lubricant viscosity, and unusual loading or mounting conditions, $a_3 < 1$. Under especially excellent lubrication condition with very smooth operation, a_3 becomes larger than 1. Logically, for a poor operating case where $a_3 < 1$, the life adjustment factor for bearing special properties a_2 cannot exceed unity.

In addition to these standards, some major bearing manufacturers such as Timken and SKF have developed their own bearing life models to best predict bearing life of their products. For example, SKF has introduced the SKF life modification factor a_{SKF} to adjust the bearing rated life for each type of bearings (e.g., ball bearings, roller bearings, thrust ball bearings, and thrust roller bearings).

In recent years, with the advances in manufacturing, tribology, materials, bearing operation monitoring, and computational technology, the bearing life calculations continue to evolve and become increasingly accurate over time, reflecting the collective experience of the bearing industry.

6.3.2 Bearing Failure Probability Distribution

Bearing reliability is the probability of bearing performing without failure over a certain period of time under the operation conditions encountered. Reliability is intimately linked

with failure, failure rates, and component and system mortality. Depending on the type of application machines, there are various models available for dealing with bearing failure probability distributions. These models can be briefly categorized as age-related failure models (e.g., bathtub-shaped distribution, normal distribution, monotone distribution) and random failure models (e.g., exponential distribution).

The Weibull probability distribution is one of the most widely used lifetime distributions in industries for modeling failure data. The main advantages of the Weibull distribution include the following: (1) it is used extensively in many different disciplines due to its ability to fit a variety of data and to represent a variety of distributions; (2) it can easily handle suspensions (nonfailure points); and (3) it can provide simple graphical solutions. The cumulative distribution function (CDF) of the Weibull distribution $F(t, k, \lambda)$ is

$$F(t,k,\lambda)=\begin{cases}1-e^{-(t/\lambda)^k} & t\geq 0\\ 0 & t<0\end{cases} \tag{6.29}$$

where

t is the bearing service time
λ ($\lambda>0$) is the scale parameter
k ($k>0$) is the shape parameter

By differentiating $F(t, k, \lambda)$ with respect to t, the probability density function (PDF) $f(t, k, \lambda)$ is obtained as

$$f(t,k,\lambda)=\begin{cases}k\left(\dfrac{t^{k-1}}{\lambda^k}\right)e^{-(t/\lambda)^k} & t\geq 0\\ 0 & t<0\end{cases} \tag{6.30}$$

The Weibull probability distribution is a versatile distribution that becomes either the normal or exponential probability distribution, based on the value of the shape parameter k. Figure 6.22 illustrates the variations of the PDF versus time t. When $k=2$ and 3.5, the PDF $f(t, k, \lambda)$ becomes the log-normal and normal distributions, respectively. By setting the shape parameter k to 1, Equation 6.29 becomes the exponential probability distribution, which has a constant failure rate

$$F(t,\lambda)=\begin{cases}1-e^{-(t/\lambda)} & t\geq 0\\ 0 & t<0\end{cases} \tag{6.31}$$

It can be observed in Figure 6.22 that the PDF curves exhibit obviously different behaviors for the $0<k\leq 1$ and $k>1$ cases. At $t=0$, $f(t, 0<k\leq 1, \lambda)$ approaches infinity and $f(t, k>1, \lambda)=0$. The increase in t results in the decrease of $f(t, 0<k\leq 1, \lambda)$ but the increase of $f(t, k>1, \lambda)$ until it reaches the peak value. At the peak point, any increase of t causes a

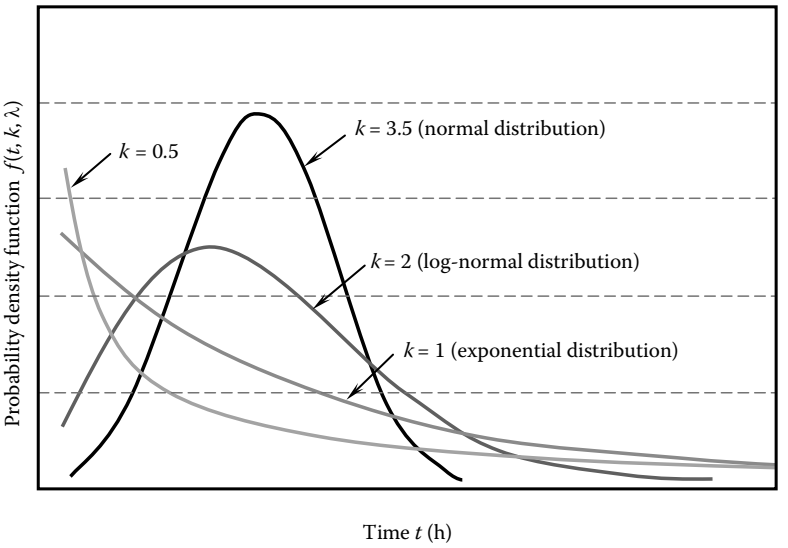


FIGURE 6.22
Weibull probability distributions with different values of the shape parameter k.

sharp decrease of $f(t, k>1, \lambda)$. As the time t becomes very large, both $f(t, 0<k\leq 1, \lambda)$ and $f(t, k>1, \lambda)$ approach zero.

Many studies suggested that bearing failures due to absent lubrication or other negligence generally conform to a Weibull exponential distribution. As an extension of Weibull distribution, this model has wide applications in many disciplines, such as bearing fatigue and windshield break. As addressed by Brake [6.26], the exponential distribution is an excellent model for describing bearing fatigue failures. Based on this model, the bearing failure rate is constant. It indicates that the bearing failure occurs randomly, regardless of the age of bearings. According to Nowlan and Heap [6.27] and Moubray [6.28], among all failure patterns, random failure patterns account for 89%. The detailed information can be found in Table 6.2.

As a real example, the bearing failure probability distribution versus the bearing operating time is presented in Figure 6.23, based on the real statistical data from a bearing manufacturer [6.29]. At the beginning of the bearing service, the bearing failure probability is rather low. Continuous bearing service results in the increase of bearing failure probability until it reaches the peak. Then, the failure probability decreases exponentially and asymptotically approaches a constant over a long time.

6.3.3 Influence of Unbalance on Bearing Fatigue Life

Excessive vibration in rotating machinery such as motors can cause unacceptable levels of noise and, more importantly, substantially reduce the life of shaft bearings. Vibration essentially results from residual imbalance, which is caused by an effective displacement of the mass center line from the true axis in a unit. Hence, it is desired to remove all inherent causes of vibration. Where vibrations are reduced, the size, mass, and/or complexity of the mounting structures can often be reliably reduced.

TABLE 6.2

Failure Patterns of the Conditional Probability

Type	Failure Patterns	Share (%)	Note
Age-related failure (11%)	A	4	The bathtub curve consists of an infant mortality region, a constant or gradually increasing failure probability region, and a pronounced wear-out region. An age limit may be desirable.
	B	2	Constant or gradually increasing probability, followed by a pronounced wear-out region. An age limit may be desirable.
	C	5	Gradually increasing failure probability, but with no identifiable wear-out age. It is usually not desirable to impose an age limit in such cases.
Random failure (89%)	D	7	Low failure probability when the item is new or just out of the shop, followed by a quick increase to a constant level.
	E	14	Constant probability of failure at all ages (exponential distribution).
	F	68	Infant mortality, followed by a constant or very slowly increasing failure probability (particularly applicable to electronic equipment).

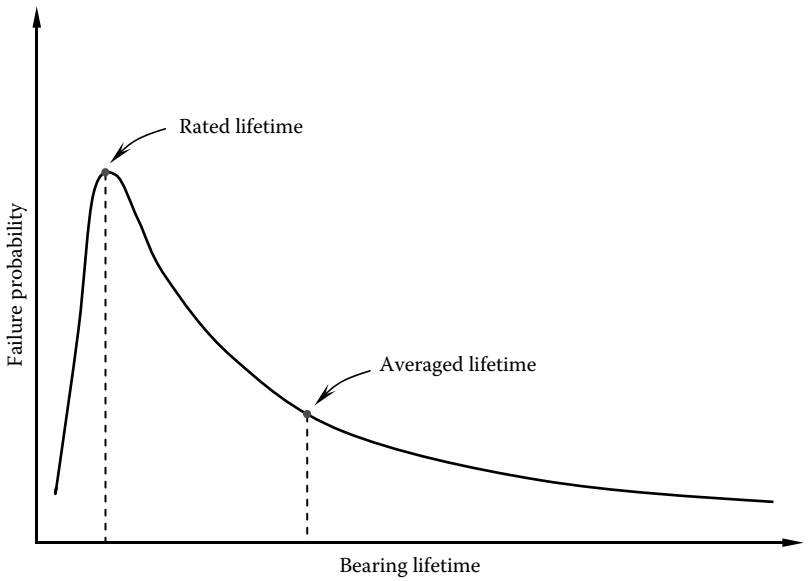


FIGURE 6.23
The bearing lifetime of motor as a function of bearing failure probability, suggested by NSK [6.29].

Mass unbalance in a motor commonly causes vibration of rotor-bearing systems. Usually, unbalance is caused by an asymmetry in the rotor that results in an offset between the shaft center line and the center of mass. Asymmetry can be attributed to an off-center weight distribution, or a thermal mechanism that produces uneven heating and bowing of the rotor, or an electrical effect that produces uneven magnetic field.

One purpose for a rotor to be balanced is to reduce the force at the bearings. This is because that under an unbalanced condition, an additional centrifugal force will be added to the bearing load. The centrifugal force can be calculated as

$$F_c = 1.77\left(\frac{n}{1000}\right)^2 WR \tag{6.32}$$

where

n is the rotational speed in rpm
W is the unbalance weight in ounce (oz)
R is the radius of unbalance weight in inch

For example, for $N = 600$ rpm, $W = 0.42$ oz (11.9 g), and $R = 1.26$ in. (32.0 mm), the calculated centrifugal force at 1g is 0.58 lb_f (2.58 N). With the increase of the centrifugal force to the radial load applied on the bearing (i.e., P) in Equation 6.32, the bearing fatigue life will be reduced for both 1g and high-g environments. The calculated results are given in Table 6.3.

For the present example, the reduction of the bearing life is less than 5% because of the small magnitude and radius of the unbalance weight. Increasing either the magnitude or radius of the unbalance weight results in a more decrease in the bearing fatigue life. It shows that bearing fatigue life is very sensitive to the radial load acting on the bearing. For instance, for the sensor bearing, as the radial load increases from 46 lb_f (205 N) to 460 lb_f (2050 N), the bearing life reduces from 33,847 to 15.7 years. The calculated results show that the reductions in bearing fatigue life make a little difference between 1g and high-g environments.

TABLE 6.3

The Impact of Rotor Unbalance on the Bearing Life

	Sensor Bearing at Rear Motor	Torque Bearing at Fount Motor	Sensor Bearing at Rear Motor	Torque Bearing at Fount Motor
Bearing Condition	**Hours**		**Year**	
Bearing life at balance condition at 1g	296,500,581	109,264,650	33,847.1	12,473.1
Bearing life at unbalance condition 1g	284,411,241 (−4.08%)	107,185,338 (−1.94%)	32,467.0 (−4.08%)	12,235.8 (−1.94%)
Bearing life at balance condition at 10g	137,623	50,716	15.71	5.79
Bearing life at unbalance condition 10g	132,012 (−4.08%)	49,751 (−1.90%)	15.07 (−4.08%)	5.68 (−1.90%)

Hesai Exhibit 1031, page 375

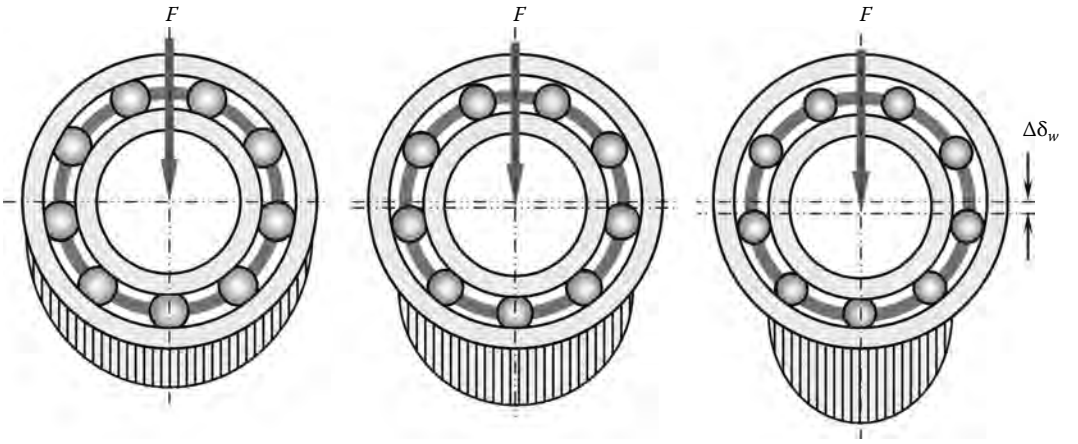


FIGURE 6.24
Influence of bearing wear on load distribution between rolling elements, where $\Delta\delta_w$ is the increased radial clearance due to bearing wear.

6.3.4 Influence of Wear on Bearing Fatigue Life

During bearing operation, the frictional sliding and rolling motion of rolling elements along raceways usually cause rolling bearing to wear. The wear allows the rolling elements to move out of position, increases the internal radial clearance, and makes the load distribution on both the rolling elements and rings even more uneven. As a result, it significantly reduces the life of rolling bearings.

Mitrović and Lazović [6.30] investigated the influence of wear on bearing life for rolling bearings. Their results indicate that the bearing internal geometries are changed due to wear, that is, the diameters of the balls and inner raceway decrease while the diameter of the outer raceway increases. The change of these dimensions leads to the increase in the internal radial clearances of the bearing and the changes in bearing operational characteristics (friction torque, vibration, noise, static and dynamic load rating, etc.). With increasing internal radial clearance, load distribution between rolling elements becomes more unequal (Figure 6.24), causing the reduction of the bearing life. Base on the bearing life theories, a mathematical model was proposed to analyze the wear influence on bearing service time using a newly developed load distribution factor.

6.3.5 Influence of Internal Radial Clearance on Bearing Fatigue Life

The internal radial clearance of radial loaded bearings can significantly alter the load distribution on rolling elements and thus impact the bearing operation reliability and fatigue life. The calculation of the bearing fatigue life is usually based on the assumption that the bearing internal clearance is zero. With a nonzero radial clearance δ_r, the function of load factor $f(\varepsilon)$ can be related to the bearing fatigue life with clearance L_ε [6.22]

- For deep-groove ball bearing

$$f(\varepsilon) = \frac{\delta_r d_b^{1/3}}{0.00044(F_r/Z)^{2/3}} \quad \text{in unit of Newton (N)} \tag{6.33}$$

- For cylindrical roller bearing

$$f(\varepsilon) = \frac{\delta_r l_{er}^{0.8}}{0.000077(F_r / Zi)^{0.9}} \quad \text{in unit of Newton (N)} \tag{6.34}$$

where
δ_r is the radial clearance (mm)
F_r is the radial load (N)
Z is the number of rolling elements
i is the number of rows of rolling elements
d_b is the ball diameter (mm)
l_{er} is the effective roller length (mm)

Under some circumstances, bearing radial internal clearance becomes zero or even negative (interference) at operating conditions. In fact, a slight interference helps reduce bearing vibration and noise, improve the load distribution on rolling elements, and achieve optimum bearing life and motor reliability. However, when the bearing radial interference becomes considerably large, high radial loads can act on rolling elements, causing damages to both rolling elements and raceways.

The bearing life factor is defined as the ratio of the bearing life L at any clearance condition to the life at zero clearance L_o

$$LF = \frac{L(\delta_r)}{L_o(\delta_r = 0)} \tag{6.35}$$

Hence, $LF = 1$ indicates zero clearance on any bearing.

ANSI/ABMA Standard 20-1996 [6.31] defines five ranges of radial clearance for various radial loaded bearings. Based on the investigation of 210-size, deep-groove ball bearings radially loaded to produce 1720 MPa inner race maximum Hertz stress under the zero clearance condition, Oswald et al. [6.22] provided radial clearance and corresponding *LF* values for four ANSI/ABMA clearance groups, as shown in Table 6.4.

The influence of internal radial clearance and interference on bearing life is presented in Figure 6.25 (using the data from [6.22]) under the same loading condition mentioned earlier. It can be seen that bearing life increases with the decrease in radial clearance. At zero clearance, $LF = 1$. With a continuously increasing radial interference (i.e., negative clearance), the bearing life increases accordingly until reaching its maximum at interference of about 0.009. Beyond this point, any increase in interference will cause a sharp decrease in bearing life.

TABLE 6.4

Life Factors and Internal Clearances in Different ANSI/ABMA Clearance Group

Group	Clearance Range (mm)	Corresponding *LF* Range
2	0.001–0.011	0.99–0.85
N	0.006–0.023	0.91–0.77
3	0.018–0.036	0.80–0.69
4	0.03–0.051	0.74–0.63

Hesai Exhibit 1031, page 377

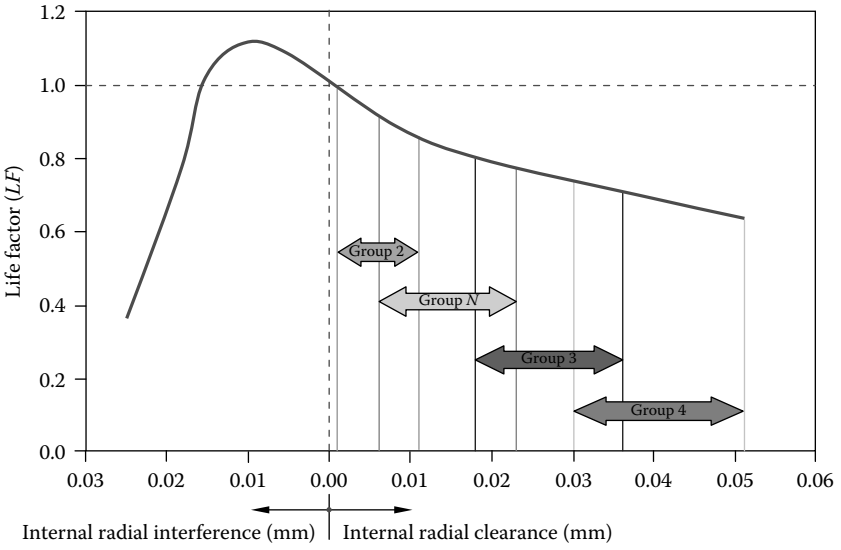


FIGURE 6.25
Effect of internal radical clearance on life factor for deep-groove bearings.

6.4 Bearing Failure Mode

Bearings are considered to be the most critical components of electric motors for maintaining motor normal operation and ultimately reducing unnecessary motor downtime. Many factors can impact bearing life, including operating condition, loads, environment condition, bearing type, bearing structure and quality (seal, surface finishing, dimension accuracy, etc.), bearing materials, heat treatment, lubrication condition, installation, and maintenance. It is commonly accepted that bearing failures that resulted from normal rolling fatigue are less than five percent [6.32].

The failure of a bearing may result in a considerable amount of consequential damage to other motor components. There are tremendous expenses associated with bearing failure in industrial and military motor applications each year. Because the rotor is fully supported by the motor bearings, any bearing defect may directly lead to the variation of the rotor position in the radial direction. This mechanical movement causes the motor air gap to vary in a manner that can be described by a combination of rotating eccentricities moving in both directions [6.33].

Bearing failure can be caused by a variety of factors, including manufacturing defects, unsuitable lubrication, improper bearing selection, inefficient sealing, incorrect fits, defective installation, system misalignment, and careless shipment. Statistical data show that bearing failure due to material fatigue accounts for only a small portion among all bearing failures. In fact, the prediction of theoretical bearing life is based on material fatigue. Most bearing failures are caused by some avoidable factors.

In diagnosing root causes of bearing failure, a fishbone diagram is a useful tool for identifying the potential suspects. It can help understand systematically how system components/factors interact and how they lead to undesirable results such as bearing failure. A fishbone diagram allows engineers to identify, explore, and graphically display all

of the possible causes related to a problem to reveal its real root causes. In fact, the fishbone diagrams work very well in many different areas.

As an example presented in Figure 6.26, the fishbone diagram has been successfully used in the root cause analysis for customers' bearing failure.

6.4.1 Major Causes of Premature Bearing Failure

Fatigue of rolling elements and raceways is an extremely complex and unpredictable process. Therefore, the analysis of bearing fatigue is based on statistical prediction depending on many different parameters, including steel type, steel processing, heat treatment, bearing precision, bearing type, lubricant, and operating conditions.

Bearing root cause analysis involves a structured investigative process that aims to identify the true cause of bearing failure and the actions necessary to mitigate and eliminate the failure. Among the factors that may lead to bearing failure are inadequate lubrication, improper bearing selection, excessive loading, overheating, contamination, high vibration, corrosion, misalignment, and loose or tight bearing fits. Among all failure modes for electric motor bearings, the lubrication-related bearing failure is the primary mode, as summarized in Table 6.5.

This table shows that the lubrication-related failure modes are the top three and account for 55% of all motor failures. The issues related to motor manufacturing and mechanical engineering account for about one-third.

Some more recent results of bearing failure modes have also been released, as shown in Table 6.6 [6.34]. Unlike the results in Table 6.5, the main bearing failure modes are corrosion, manufacturing defects, and bearing precision. The failure mode categorized under *other* includes bearing misalignment, true and false brinelling, excessive thrust/bearing overload, lubrication, heat and thermal preload, roller edge stress, cage fracture, element or ring fracture, skidding damage, and electric arc discharge. Some of these major causes will be addressed in the following sections.

For journal engine bearings, eight major causes of premature bearing failure are listed in Table 6.7 [6.35]. It is important to note that in many cases, a premature bearing failure is collectively caused by a combination of several of these causes.

6.4.2 Lubricant Selection

From the reference [6.1], it is clearly shown that the vast majority of bearing failures in electric motors are associated with lubrication-related problems. All of the rolling bearing systems require appropriate lubrication to ensure normal operation. The metal-to-metal contact of the rolling elements against the raceways necessitates the presence of grease or oil at all times. If the external lubricant is not present, the balls or rollers will begin to make direct contact with the rail material, resulting in galling and brinelling damage. Many manufacturers attempt to overcome this weakness in the design by adding oil-impregnated seals to the ends of the bearing for extending the bearing life [6.36].

In the vast majority of cases, correct lubrication is the most important factor in obtaining good performance of bearings. As a matter of fact, the selection of lubricant and lubrication method is as important as the selection of the bearing itself.

In most cases, greases are considered as the best lubricant for motor bearings. Today, grease manufacturers offer multitudes of greases suitable for a wide variety of bearing

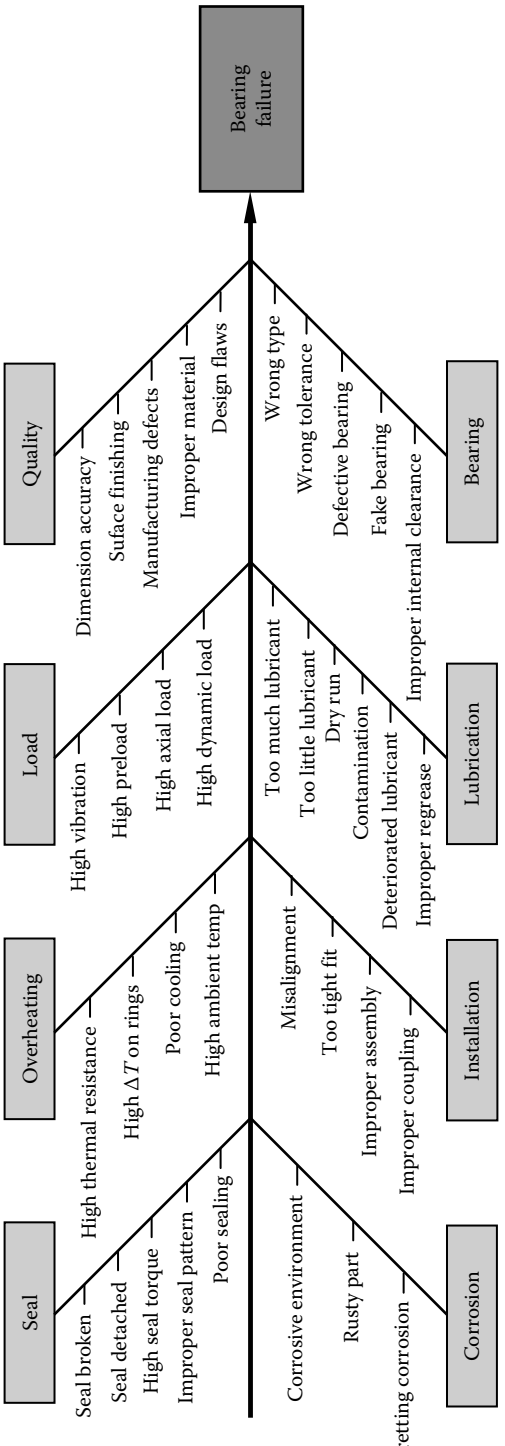


FIGURE 6.26
Fishbone diagram for root cause analysis of bearing failure.

TABLE 6.5

Common Reasons for Bearing Failures

Ranking	Reason of Bearing Failure	Weight (%)
1	Incorrect or aged lubricant	22
2	Dirty bearing, coarse particles, or liquid	19
3	Too much or too little grease	14
4	Incorrect alignment	11
5	Bearing currents/bearing insulation	9
6	Vibration from motor or load machine	8
7	Installation/maintenance failure	7
8	Manufacturing/dimensioning	6
9	Consequential damages/others	4

Source: Data from Lawrie, R., *Electrical Construction & Maintenance (EC&M) Magazine*, February Issue, 2001.

TABLE 6.6

Bearing Failure Mode

Probable Cause for Bearing Failure	Weight (%)
Fatigue (surface and subsurface origin)	3
Cage	3
Bearing wear	6
Handling damage	7
Dimensional discrepancies	17
Debris denting/contamination	20
Corrosion pitting	27
Other	17

TABLE 6.7

Major Causes of Premature Bearing Failure

Cause of Premature Bearing Failure	Weight (%)
Dirt	45.4
Misassembly	12.8
Misalignment	12.6
Insufficient lubrication	11.4
Overloading	8.1
Corrosion	3.7
Improper journal finish	3.2
Other	2.8

applications and operation conditions. The characteristic feature of greases is that they possess a high initial viscosity. The correct choice of grease depends on many factors, including grease viscosity, base oil type, additive requirements, thickener, and some other factors such as dropping point and operating temperature range.

For very high-speed applications such as spindle motors, the bearing temperature can be considerably higher due to friction between deformed rolling elements, raceways,

and cage. For these circumstances, bearing cooling becomes more important than bearing lubrication. Obviously, the use of greases is no longer appropriate and oil lubrication should be provided. Oil, as an efficient coolant, can provide good solutions not only for bearing lubrication but also for bearing cooling. There are three types of oil lubrications:

1. Oil mist lubrication. Oil mist is produced in an atomizer and conveyed to the bearings by an air current.
2. Oil–air lubrication. Oil droplets are carried by compressed air to lubricate/cool bearings.
3. Oil jet lubrication. Injection technique is often used in cooling of high heat flux devices. Oil jet can carry away a large amount of heat from the bearings and is cooled by an oil-to-water or oil-to-air heat exchanger. This lubrication method is especially suitable for high-load, high-speed bearings.

6.4.3 Improper Bearing Lubrication

Among all causes of bearing failure, a primary cause is over or under bearing lubrication. Improper lubrication may result in high temperature buildup in bearings, causing the grease to break down prematurely and thus, leading to the excessive wear of the bearing until its being completely scrapped.

Lubricants are usually made from various oils and additives. There are many different types of motor lubricants, including mineral oils (pure and refined) and synthetic oils for higher temperatures. Animal and vegetable oils are not normally used for bearing lubrication due to the risk of acid formation after a short operating period.

With too much added lubricant such as grease, bearings consume energy as the rolling elements begin churning the lubricant. The energy is then converted into heat that raises the temperature of the lubricant. As a result, the viscosity of the lubricant is reduced and, at elevated temperatures, varnish or coke may form. Furthermore, too much lubricant can effectively reduce the bearing internal clearance. On the other hand, too little lubricant cannot guarantee a sufficient lubrication of the bearing. With insufficient lubricant, an oil film with needed load-carrying capacity cannot fully form. This may result in metal-to-metal contact between rolling elements and raceways, leading to adhesive wear and bearing failure. As a matter of fact, more motor bearing failures are due to overgreasing rather than undergreasing [6.37].

6.4.4 Lubricant Contamination

One of the leading causes of premature bearing failure can be attributed to lubricant contamination, which increases energy consumption by causing abrasive friction at bearing surfaces, raising lubricant viscosity, generating dents on the raceway and rolling element surfaces, and consequently reducing the bearing operation life. Water from condensation of moisture-contained air can enter the lubricant via penetration through the bearing seal or some other ways. It has been reported [6.38] that as little as 0.002% water in the bearing lubricant will reduce bearing life by 48%, while 6% water content in the lubricant will reduce bearing life by as much as 83%.

Microscopic particles are the most harmful form of lubricant contamination as they initiate irreversible microscopic damages on the bearing contact surfaces. Over time, these



FIGURE 6.27
A rusty bearing with 2-year service in an electric mold nonsinusoidal oscillation system for slab caster in a steel plant.

microscopic damages may accumulate and progress into macroscopic damages and finally lead to premature bearing failure and breakdown of motors.

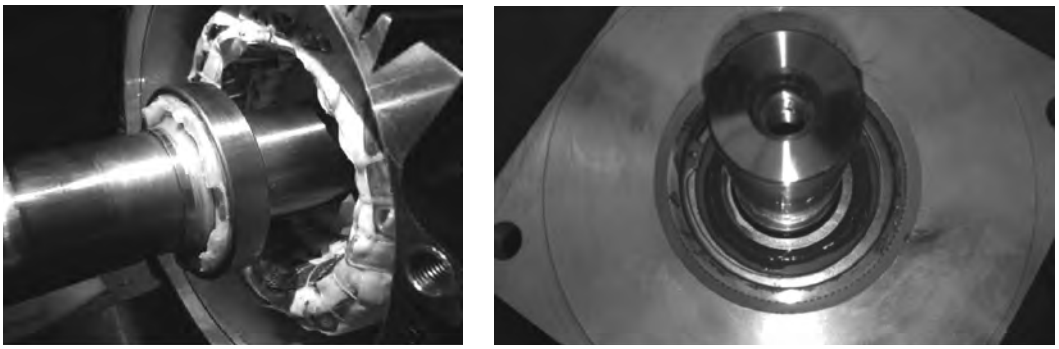
When bearings work under certain hazard conditions such as high temperature, high moisture, and other corrosive environments (e.g., steel or glass plants), corrosion on some bearing materials would be hardly avoidable. Figure 6.27 shows a rusty bearing after a 2-year service in an electric mold nonsinusoidal oscillation system for slab caster in a steel plant. Ferric oxide is an inorganic compound with the formula Fe_2O_3. During motor operation, Fe_2O_3 powder may enter critical areas of the bearing where sliding or rolling contact takes place, causing fretting damage on the bearing.

6.4.5 Grease Leakage

In a field motor inspection, it is common to encounter grease leakage problems. Grease may leak from a sealed bearing between the inner seal groove and the seal, especially for vertically installed motors. While a small amount of grease leakage may exert little impact on motor performance, an aggravated level of grease leakage can significantly shorten bearing life, degrade bearing performance, or even cause bearing failure.

When grease leakage occurs, it is important to check the questions listed in the following:

- Is the selected grease appropriate? The properties of the grease (e.g., viscosity, the allowable operation temperature) must be checked carefully.
- What is the operation temperature when the leakage occurs? As the operation temperature goes higher, the grease viscosity becomes significantly lower.
- What is the condition of the bearing seal? An improper or damaged bearing seal always results in grease leakage.
- What is the bearing load condition? An overloaded bearing tends to cause grease leakage.



(a) (b)

FIGURE 6.28
Grease leakage from the bearings in the vertically mounted motors: (a) with a 2-year service in an electric mold nonsinusoidal oscillation system for slab caster and (b) with a 3600 h service in a CNC laser cutting machine.

- Is the bearing overfilled with grease? Grease overfilling can cause excess churning of the grease during operation and generate extra resistance in the bearing.
- What is the level of motor vibration? A high motor vibration can accelerate grease leakage from the motor bearings.

As demonstrated in Figure 6.28, lubricating grease leaks out of the bearings and accumulates on the bearing seals. It is noted that in both cases, the grease purge occurs from the vertically mounted motors with low-frequency vibration.

In order to improve the sealing capability and avoid grease purge from sealed deep-groove ball bearings, NTN engineers have designed new geometrical cages that differ in shape from that of the standard case [6.39]. The new type of cages has been confirmed experimentally to suppress the adherence of grease to the inner ring seal groove, improving the resistance to grease purge substantially.

6.4.6 Bearing Sealing

Bearing seals and shields are commonly used to retain the bearing lubricant and prevent ingress of external contaminations. Both seals and shields are available in a variety of materials and configurations. Briefly, a seal is retained to the bearing outer ring, but also makes light contact with the rotating inner ring. This provides a high degree of bearing sealing capability with a modest increase in friction and torque. Generally, when the sealing performance is enhanced, the torque required to turn the bearing increases with the friction caused by the seal. It is noted that some types of seals (e.g., lip seals) may generate a high pressure at the lip, resulting in greatly increased torque and friction losses, especially in high-speed applications.

In contrast, a shield is a noncontact closure piece, being staked to the bearing outer ring with a small gap between the shield ID and the bearing inner ring. Thus, bearing shields have virtually little compact on bearing performance. As a result, in torque-sensitive applications, it is advantageous to use shields rather than seals.

Retention of bearing seals/shields has a critical impact on achieving the best bearing sealing performance. This is especially true for vertically installed motors subject to high levels of vibration. It is interesting to note that even for the same model and bearing



(a) (b)

FIGURE 6.29
Different retention methods used for securing bearing seals for the same model and same brand bearings: (a) with a number of knocks equally distributed along the circumference of the outer ring ID and (b) a simple retention method without using knocks. It is obvious that the retention method used in (a) can provide better retaining capability than that in (b).

brand, the retention methods could vary depending on in which countries they are manufactured. As shown in Figure 6.29, one bearing uses 19 knocks that are equally distributed along the circumference of the outer ring ID to retain the seal firmly. However, the other bearing has no such feature. Obviously, the former has a much better seal retaining capability.

Figure 6.30 shows an example of a bearing failure due to the bearing seal being pushed down in a vertically mounted motor as a consequence of improper sealing. The field inspection has revealed that as the seal was dropped off, the grease was completely lost from the bearing. Under a dry run condition, the bearing temperature was raised dramatically, leading to the disintegration of bearing cage into pieces.

6.4.7 Excessive Load

Bearing load and speed are the most common rating factors impacting bearing service life. Of the two, bearing load has a greater influence on its lifetime. Generally, by doubling the load of a bearing, the bearing life can be reduced by a factor of ten for the ball bearing and eight for the roller bearing. As a contrast, doubling the speed reduces the bearing life by half.

Excessive loads usually lead to premature fatigue failure of the contact surfaces of the rolling elements. As excessively heavy loads repeatedly act on the contact rolling elements, microcracks start to develop on the surfaces of raceways or rolling elements (balls or rollers). After a period of time, these cracks grow and propagate on the fracture surfaces and finally cause flaking on the raceway or rolling element surfaces. When bearings are subject to very high loads or shock loads, it may lead to large cracks or fractures of bearing components.

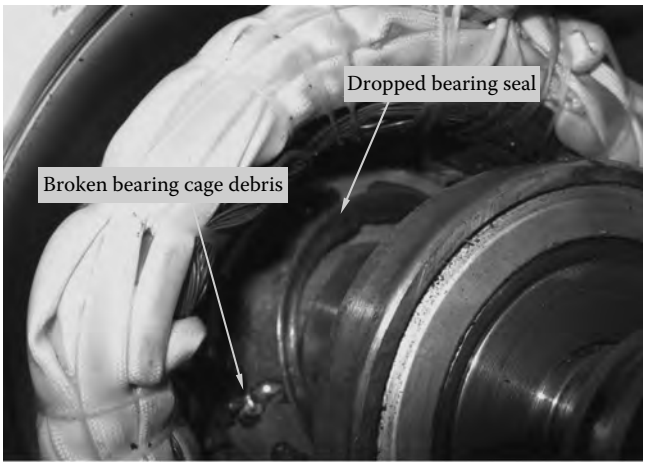


FIGURE 6.30
The bearing seal is dropped off from the bearing. The cage is severely destroyed, broken into pieces to scatter inside the motor.

6.4.8 Internal Radial Interference Condition

Regularly, bearing manufacturers always design bearings with certain internal radial clearances. However, under some operating conditions, the internal radial clearance in a bearing may be reduced or even completely eliminated. In some cases, the radial clearance may be further turned into the interference. With the bearing internal radial interference, the rolling elements and raceways are tightly fitted together, applying large frictions and loads to the bearing components.

The bearing effective radial operating clearance δ_{eff} is defined as the amount of radial internal free movement during bearing normal operation. It can be determined by three clearance components: (1) the initial bearing radial internal clearance before mounting, δ_o; (2) the reduced amount of radial clearance due to interference fit, δ_f; and (3) the reduced amount of radial clearance due to temperature difference between the outer and inner rings, δ_t [6.16], that is,

$$\delta_{eff} = \delta_o - (\delta_f + \delta_t) \tag{6.36}$$

When bearings are mounted on shafts and in bearing bores with interference fits, the inner ring will expand and the outer ring will contract, thus reducing the bearings' radial internal clearance to approximately 70%–90% of the effective interference.

$$\delta_f = (0.7 \sim 0.9)\Delta\varepsilon_{eff} \tag{6.37}$$

where $\Delta\varepsilon_{eff}$ is the effective interference

During motor normal operation, the bearing outer ring is typically about 5°C–10°C cooler than the inner ring due to the smaller thermal resistance from the stator to the surrounding environment. However, as the motor housing is subjected to forced convective cooling, the temperature difference between the outer and inner bearing rings

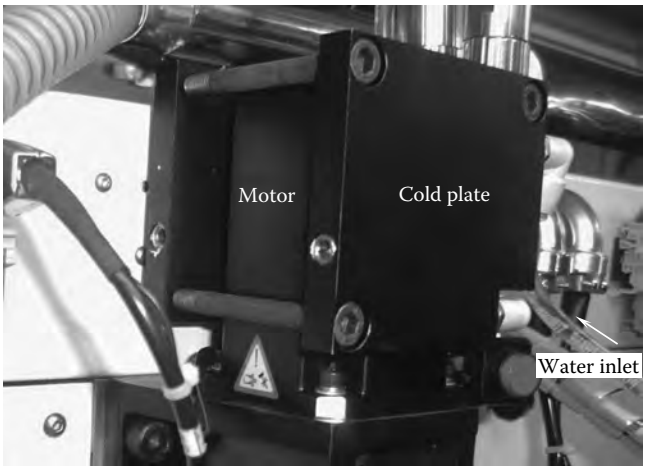


FIGURE 6.31
A motor is sandwiched by two water cold plates, causing a large temperature difference between the bearing outer ring and inner ring.

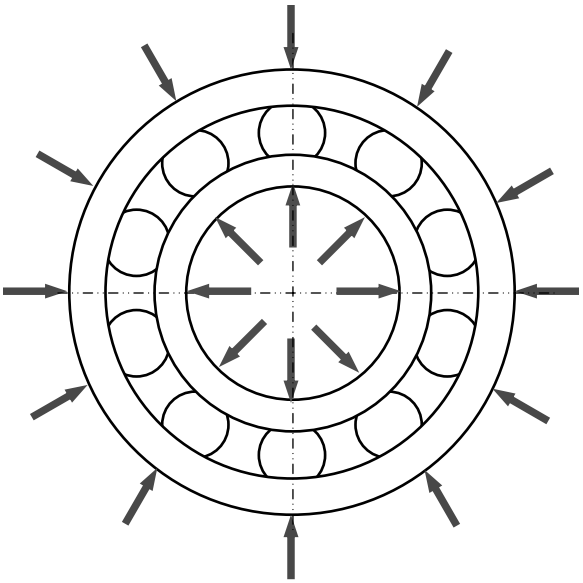


FIGURE 6.32
Due to the temperature difference on the bearing's outer and inner rings as a result of motor cooling, shrinkage occurs on the outer ring and expansion on the inner ring.

can be considerably larger. As shown in Figure 6.31, a motor is sandwiched by two water cold plates. A much lower temperature is measured at the bearing outer ring when the cooling systems start operating. Thus, the amount of radial internal clearance is further reduced due to the shrinkage of the outer ring and the expansion of the inner ring (Figure 6.32).

The decrease in radial internal clearance due to the temperature difference ΔT of the outer and inner rings δ_t can be calculated as

$$\delta_t = \alpha(\Delta T)d_{or} \tag{6.38}$$

where

α is the bearing material expansion coefficient, which, for stainless still, $\alpha = 12.5 \times 10^{-6}$
ΔT is the temperature differential between the inner and outer rings
d_{or} is the outer ring raceway diameter. For ball bearings, $d_{or} = 0.2\ (d_i + 4d_o)$

Using the above equations, the bearing effective radial operating clearance δ_{eff} can be displayed as a function of the temperature difference between the inner and outer rings. As an example, the results for a 6802 bearing with a normal clearance class CN (its radial clearance ranges of 3–18 μm) are shown in Figure 6.33.

It can be seen that for $\Delta T < 7.2$°C, the bearing has a positive radial clearance. At the point of $\Delta T = 7.2$°C, the radial clearance vanishes. As ΔT becomes larger than 7.2°C, it turns to the radial interference. Note that small interference is allowed and desirable for some applications for increasing bearing stiffness and reducing vibration and noise. However, excessive interferences result in extra rolling contact pressure, which can greatly reduce the fatigue life of the bearing. When the interference becomes high enough, the bearing balls may be completely held by the inner and outer raceways. As a result, these balls are unable to rotate freely but forcibly slide between the raceways, causing overheating, overwearing, and eventually the failure of bearing. It is to be noted that since large bearings usually have large radial clearances, they are unlikely experiencing overheating and overwearing situations. Even for the same bearing size, if the bearing has a higher radial clearance class,

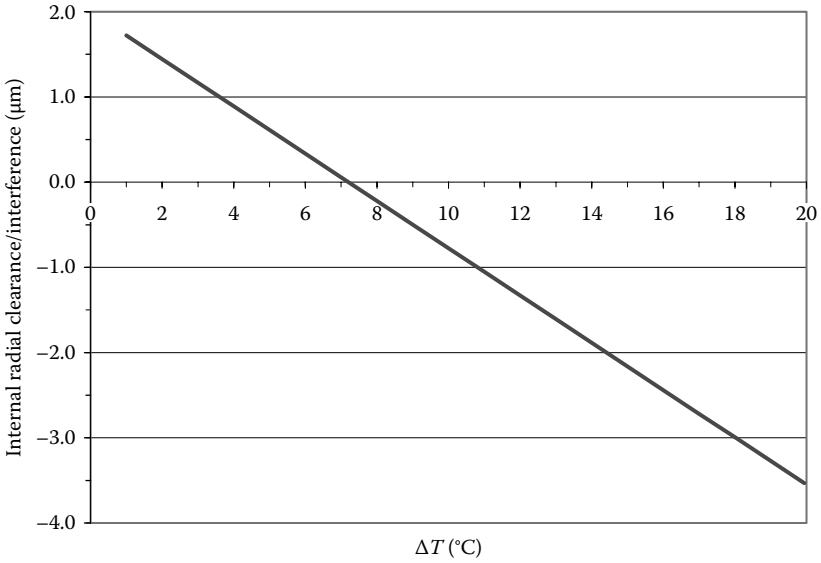


FIGURE 6.33
Effect of the temperature difference of the bearing inner and outer rings on the bearing radial clearance/interference for a specific bearing with the clearance class of *CN*.

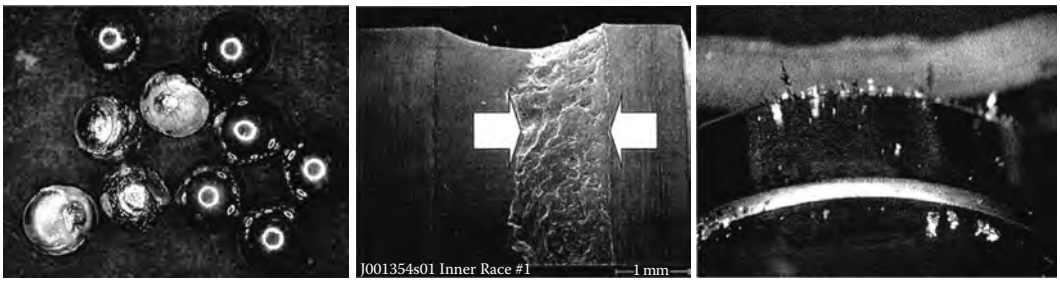


FIGURE 6.34
Damaged bearing balls, inner and outer raceways due to large axial load. It can be clearly seen that spalling occurs at one side of the inner raceway.

for instance, *C*3 or *C*4, it will significantly improve bearing operation condition under such circumstances.

Most bearings are designed to withstand primarily radial loads, and only certain extent of axial loads. As an example, for deep-groove ball bearings, the maximum axial load is typically about 50% of the maximum radial road. However, for small bearings, this value can be lowered to 25%. When a ball bearing is subjected to a large axial load, the internal radial clearance vanishes as the result of the shift of the inner ring axially relatively to the outer ring (see Figure 6.15). Correspondingly, the balls and raceways develop elastic deformation, resulting in an increase in the contact angle β (defined as the angle between the bearing contact line and bearing normal line) and contact width. This changes the bearing from the point contact to ellipse contact, which in turn causes asymmetric wear of the ring at their sides of the raceways, as demonstrated in Figure 6.34.

6.4.9 Bearing Current

The problem of bearing currents and shaft voltages has been recognized for almost a century [6.40]. A shaft voltage can cause the bearing current that flows through the bearings. A number of surveys have indicated that 30% of all motor failures operated with 60 Hz sine wave voltage are due to bearing current damage [6.41]. In recent years, there is an increase in motor failures due to bearing current.

The most common underlying cause of bearing currents in three-phase IMs is alternating magnetic flux, which is a result of asymmetric magnetic properties of the stator or rotor core. The asymmetrical flux through the steel results in time-varying flux lines that enclose the shaft. This can drive a current down to the bearings, endbell, housing, and back again to the bearings [6.42]. In addition, the use of variable frequency drives can induce the shaft voltage because of the extremely high-speed switching of the insulated gate bipolar transistors, which produce the pulse width modulation used to control AC motors. This shaft voltage can exceed the dielectric strength of the lubricant film in bearings, causing the bearings to discharge [6.43].

When electric current flows through IM bearings along internal closed-loop paths (Figure 6.35), it may generate electric arc, which tends to melt and damage internal bearing surfaces, just like a series of small lightning strikes. This can cause bearing surfaces to flake away and spall out, leaving behind microcraters on the bearing surfaces. As rolling elements roll on these damaged surfaces with contact pressures, the microcraters will

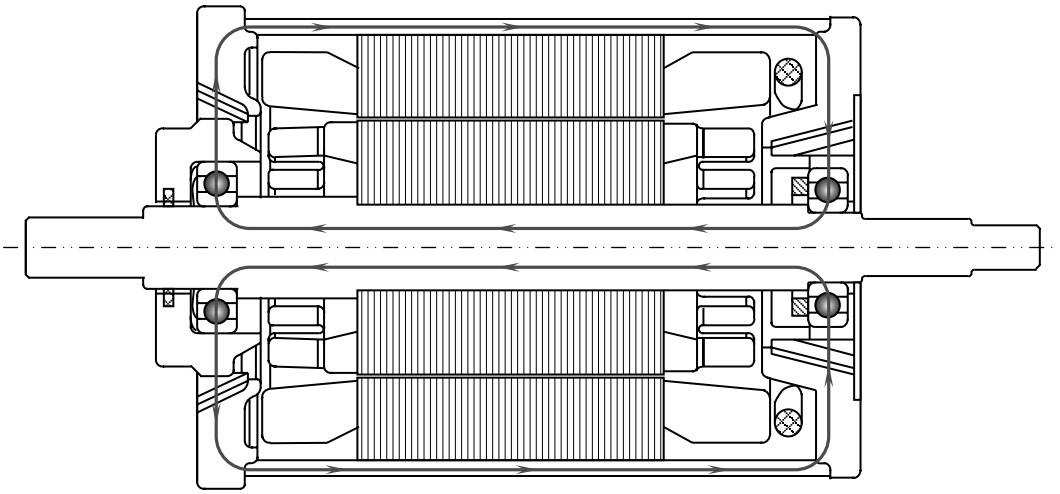


FIGURE 6.35
Bearing current resulted from time-varying magnetic flux in a motor.

be increasingly enlarged, further leading to the deterioration of bearing contact surface, acceleration of wear, elevation of noise and vibration levels, and eventually bearing failure. In addition, wear debris may be introduced into the lubricant, causing lubricant contamination problems.

There are a number of strategies and methods available for reducing/eliminating bearing currents:

- An effective way to eliminate bearing currents is to break the current path by insulating either bearings or endbells from the motor frame. Insulating bearings may be accomplished by the following [6.44]: (1) adding an insulation material to the bearing ring or bearing journal; (2) coating a ceramic material on bearing components; or (3) adopting ceramic bearing, which has been experimentally investigated.
- Grounding shaft is another effective way to eliminate the shaft voltage. This can be done via the usage of metal brushes or grounding rings [6.45].
- Another solution to the bearing current problems is to improve the high-frequency grounding connection from the motor to the drive and from the motor to the driven equipment.
- Establishing an alternative current path directly from rotor to stator or from rotor to frame by means of bushes can effectively reduce bearing current.
- Theoretically, using a conductive Faraday shielded between the rotor and stator would prevent the current from being induced in the shaft. However, this method is difficult to implement in practice.

6.4.10 Impact of High Temperature on Bearing Failure

The maximum permitted temperature in rolling element bearings depends on many factors, including bearing material, bearing type, lubrication type, heat treatment, manufacturing

and assembly accuracy, and bearing operation condition. Temperatures in excess of this limit can permanently cause bearing failure.

The standard stabilization temperatures are 120°C for deep-groove ball bearings and 150°C for angular contact ball bearings. Overheating bearings above their stabilization temperatures can anneal the ring and ball materials, leading to the rapid softening of the bearing steel, the reduction of load-carrying capacity, and possibly the subsequent failure.

High temperatures can lead to the decomposition, oxidation, or deterioration of some lubricates. In addition, bearing overheating also lowers viscosity of bearing lubricates and thus makes it easier for lubricates to leak off from the bearings. It was reported [6.39] that grease purge from a greased and sealed bearing occurs between the seal and the inner seal groove that rotate against each other. An increase in bearing temperature causes a pressure on grease located on the inner ring seal groove, resulting in grease purge. In order to improve the prevention of grease leakage, the cage was redesigned with optimized geometrical shape. The promising results were achieved experimentally using the improved cage.

6.4.11 Bearing Failure Associated with Motor Vibration and Overloading

Vibration can occur at anytime when a motor is in normal or abnormal operation. Vibration can worsen motor performance and reduce the bearing life due to the alternating force producing both impact force and stress reversal. An excessive level of vibration may lead to premature failure of bearings.

Motor vibration can be activated by the combination of a variety of forces, including the following: (1) mechanical, electrical, and electromagnetic unbalancing forces, which can severely reduce the bearing life; (2) misalignment forces due to shaft misalignment; (3) belt/gear driving tension forces; (4) looseness forces due to looseness of motor components such as a large radial clearance in motor bearings; (5) gravitational force of rotor; and (6) reaction forces from other devices such as gearboxes. Increasing the dynamic forces caused by unbalance, misalignment, and looseness can remarkably result in higher vibration levels and thus significantly reduce the bearing life.

The impact of increased load/force on the bearing life is pronounced. As demonstrated in Figure 6.36, doubling the load/force on bearing can reduce the bearing life by 87% for ball bearings and 90% for other rolling element bearings (e.g., cylindrical, spherical, tapered, and needle bearings) [6.46].

On the contrary, reducing the vibration level can greatly extend the bearing life. According to the data provided by Berry [6.46], by assuming the dynamic load is the major force component, and reducing the vibration level by half will extend the bearing life by 700% for ball bearings and 908% for other rolling element bearings, as shown in Figure 6.37.

As one of the leading causes of bearing failure, overloading often results in overstresses of bearing components. Overstress creates the possibility of motor component damage caused by a single extreme loading event. A common catalyst for this type of damage is improper belt tensioning. Excessive load introduced by belt can drastically reduce the expected bearing life. It has reported that a 10% increase in tension may reduce bearing life by up to 50%, depending on the system [6.47].

A tight fit between the bearing and the mating parts (i.e., shaft and bearing bore) can eliminate the radial clearance of the bearing. Under this circumstance, the bearing balls and raceways become excessively loaded, leading to rapid wear on these components and resulting in a rapid temperature rise accompanied by high torque.

Hesai Exhibit 1031, page 391

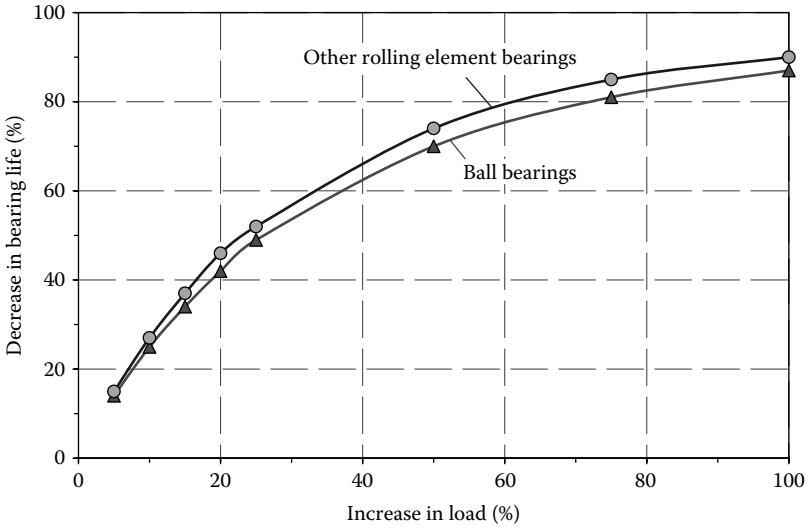


FIGURE 6.36
Impact of increased load on bearing life.

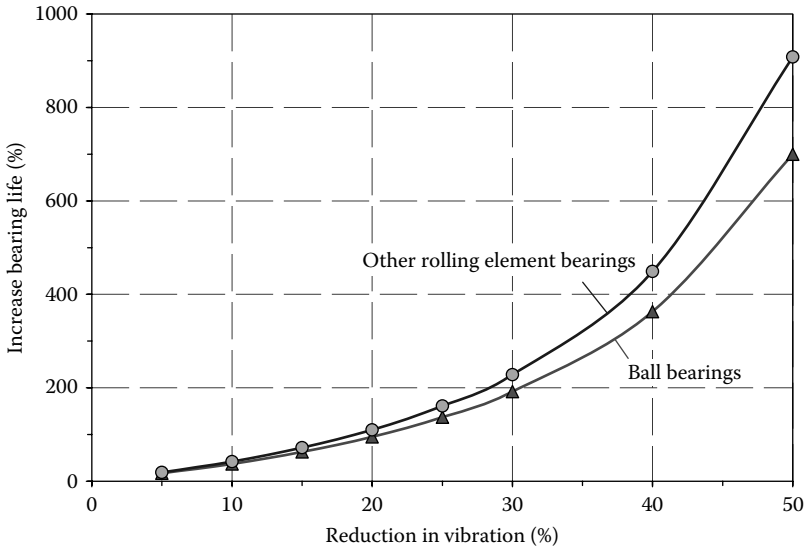


FIGURE 6.37
Impact of reduction in vibration level on bearing life.

6.4.12 Improper Bearing Installation and Bearing Misalignment

Bearing failure may be also caused by improper bearing installation. In an installation process, bearings are forced onto shafts or in bearing bores of endbells/frames. This may produce physical damage in the form of brinelling or false brinelling of the raceways, which leads to premature failure. Bearing misalignment is another common consequence

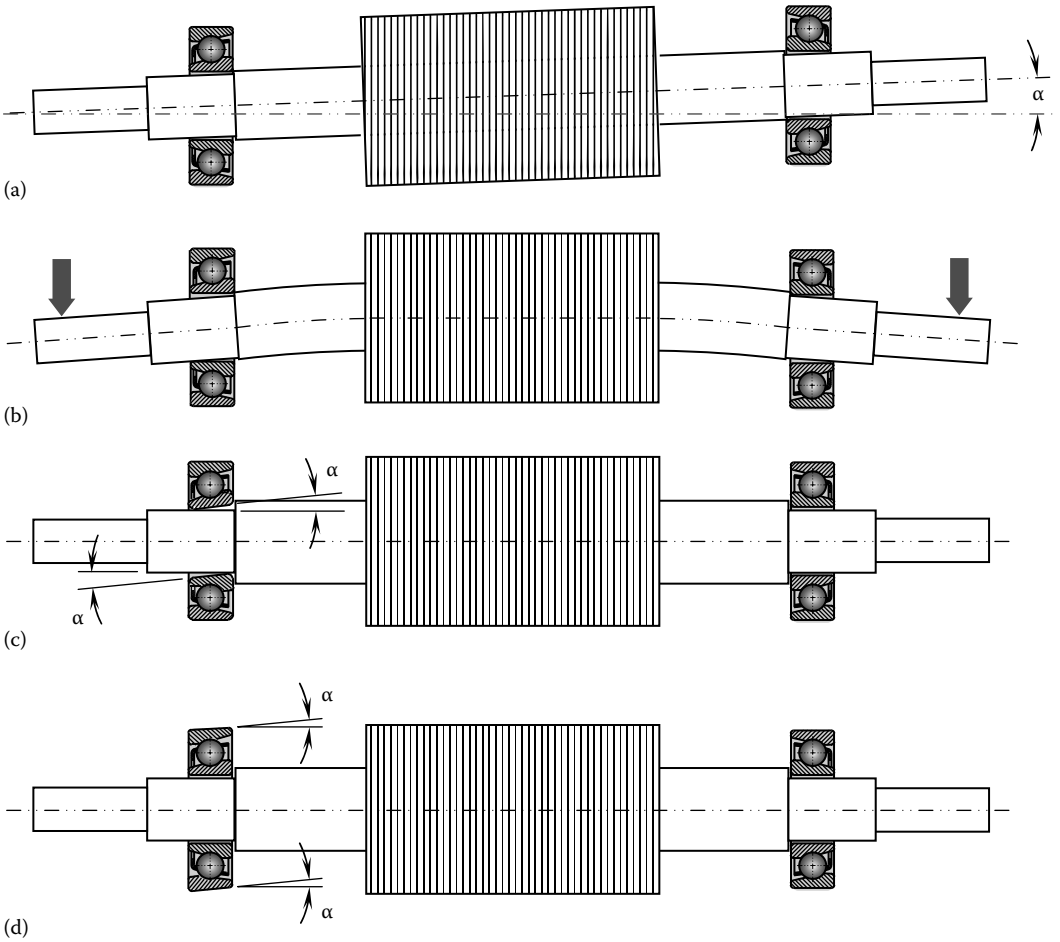


FIGURE 6.38
Four types of rolling element bearing misalignment: (a) out-off-line; (b) shaft deflection; (c) cocked or tilted inner raceway; and (d) cocked or tilted outer raceway.

of improper bearing installation. As depicted in Figure 6.38, there are four types of bearing misalignment: out-off-line, shaft deflection, cocked or tilted inner raceway, and cocked or tilted outer raceway [6.48]. Each type of misalignment can cause abnormal forces acting on the bearing that may accelerate wear of the bearing raceways, cage, and rolling elements.

6.4.13 Vertically Mounted Motor

As a motor is mounted vertically, the weight of the rotor itself as well as the possible external loads will act on the bearings in the motor axial direction. The increasing axial load can significantly change the bearing operation characteristics, for instance, resulting in larger pressure angle, pressure ellipse, and stresses on bearing rings and rolling elements.

Although many motor manufacturers claim in their motor manuals that their motors can be installed either horizontally or vertically, it must be recognized that in most cases, electric motors are primarily designed for horizontal operation. Motor bearing surveys have revealed that a considerable number of motors use deep-groove ball bearings. This type of bearing can withstand large radial loads but relatively small axial loads. For a vertically mounted motor, it may consider using tapered roller bearings or dual bearing assembles (which are similar to the front wheel bearing on a car) that have the ability to take loads in both radial and axial directions.

Lubricant retention is critical for vertically installed motor for normal operation. Due to the gravity force, vibration, and temperature effects, the lubricant in motor bearings can gradually leak out from the bearings. Therefore, the sealing capability of bearing becomes one of the key parameters in the bearing selection for reducing the risk of premature bearing failure and enhancing bearing life. In addition, better bearing sealing can prevent external contaminants from entering the bearing interior.

For a vertically mounted motor, any movement of the shaft relative to the motor bearings, as well as to the motor endbells, is not allowed, especially under vibration and high-temperature operating conditions. To avoid this problem, it is critical to use interference fit between the shaft and bearings and tight fit between the bearings and bearing bores. In addition, retaining rings or snap rings must be added to the shaft to avoid the axial movement of the bearings and shaft in any direction with respect to the motor frame. Failing to do so will cause axial movement of either the shaft or bearing relative to the motor, resulting in severe bearing damages. For any cases, the axial load on motor bearings should not exceed the maximum allowed value.

6.5 Bearing Noise

Bearing noise comes from many sources, including cage noise, race noise, seal noise, waviness noise, and lubricant noise. Some types of noise are unavoidable and acceptable, such as lubricant noise. However, some types of noise are related to the manufacturing process and can be readily avoided, such as waviness and click noise. The various types of noise in rolling bearings are grouped into four categories [6.49]: structural (cage, squeal, race, and click noise), manufacturing (waviness noise), handling (flaw and contamination noise), and others (seal and lubricant noise).

Manufacturing precision has a strong impact on bearing noise. There are a large number of parameters affecting the bearing noise level, including roundness of bearing rings and balls, shape accuracy, surface finishing, dimensional tolerance, internal clearance control, and assembly accuracy. In fact, even a slight defect in shape may cause noise and vibration. Generally, noise due to manufacturing defects can be lowered with the improved manufacturing process. For instance, waviness noise can be reduced by decreasing the waviness in the circumferential direction on the finished surfaces of the bearing raceways.

Major bearing components are heat treated to render the hardness larger than 60 HRC. Lower quenching temperatures may give bearing components lower hardness or develop soft points on them. This reduces the wear resistance of the bearing components sharply. Other defects due to improper heat treatment include quenching microstructure,



FIGURE 6.39
Bearing is cut for inspecting bearing damage and wear condition.

deformation, surface decarburization, and quenching crack. All these defects raise bearing noise levels.

Compared with ball bearings, rolling bearings are manufactured and processed according to higher precision standards, and therefore generally produce relatively low level of noise. For motors requiring particularly low-noise operation, the selection of deep-groove ball bearings or cylindrical roller bearings can provide appropriate solution.

In some motor manufacturers, bearing noise can be simply detected with modified medical stethoscopes. This has been proven to be an effective and convenient method, particularly to operators on assembly lines. In order to find root causes of bearing noise, the suspected bearing is often sliced into halves to inspect manufacturing defects, abnormal wear, and other damages (Figure 6.39). It has been found that even very tiny manufacturing defects, which cannot be identified by naked eyes, can generate significant click or waviness noise. Thus, a microscope is a necessary apparatus for the bearing inspection.

6.6 Bearing Selection

The primary factors in the bearing selection are the total system reliability during the design life and the cost-effectiveness. The selection of appropriate bearings has become crucial to the overall performance of electronic motors. The selection process must carefully consider all factors that affect bearing performance and cost, including the following:

- Radial and axial load conditions. In fact, bearing fatigue life is sensitive to the magnitude of load. Generally, if the load is doubled, the fatigue life is reduced by a factor of ten for the roller bearing and a factor of eight for the ball bearings.

- The change in the contact condition between the rolling elements and rings. When a bearing is running under loads, the forces are transmitted through the rolling elements from one ring to another. The magnitude of the load carried by an individual rolling element depends on the location of the element at any instant, the number of rolling element, and the bearing type. Due to the plastic deformation of the bearing components, the contact condition of the bearing components can alter dramatically. For instance, for a ball bearing, the contact between the balls and rings is no longer a point contact rather, it forms a contact area called contact ellipse. For a roller bearing, instead of a line contact, the contact area is a narrow rectangular.
- Rotor rotating speed range—The bearing speed has a strong impact on the selection of the bearing lubricant. Generally, the higher the bearing speed, the lower the lubricant viscosity.
- Expected bearing life and motor operation life—Usually, the life expectancy of motor can exceed 20 years. This requires an extended bearing life. However, under some harsh operation conditions, the bearing life is much shorter than expected. Hence, it is important to adjust it according to the real situation. For instance, bearings used in oscillation systems for slab caster experience high temperature, moisture, and dust level. These bearings should be replaced after 2-year service, regardless of whether the bearings are damaged or not.
- Operating temperature—The variation in operating temperature may result in the expansion or contraction of rings and rolling elements, which alters the bearing internal clearance. In some cases, the radial clearance may be eliminated due to the temperature difference between the inner and outer ring. In the bearing selection, a wide operation temperature range is highly desired.
- Sealing requirement—Under harsh environmental operating conditions, good sealing is necessary for motor bearings to prevent dust, moisture, and other foreign matters to get in the bearings, as well as the leakage of lubricant from the bearings. This is especially true for outdoor motor applications.
- Lubrication method—Bearings typically use grease or oil as lubricant. Grease lubrication is often used in applications where frequent replenishment of the lubricant is undesirable or impossible. Oil lubrication is suitable for applications that require low torque or narrow range of torque variations. There are a number of oil lubrication methods for different applications, including oil bath, circulation oil, oil jet, oil drop, and oil mist. Because bearing lubrication is one of the major causes for bearing failure, it is highly desired that bearings can be self-lubricated for extending the bearing life and reducing the maintenance cost.
- Bearing internal clearances in both radial and axial directions—For precisely controlled motion systems, bearings with small internal clearances are preferred for minimizing motor vibration and maintaining motor smooth operation. In the bearing section, it is important to take into account the variations of bearing internal clearances with the operating temperature.
- Bearing noise level—Bearing noise can be a critical factor for some applications such as medical devices. Specifically designed precision bearings generally produce very low levels of noise. However, low noise bearings are more expensive than regular bearings.
- Installation/disassembly requirements—In some harsh operating conditions, the bearing life is significantly shorter than the motor lifetime. This indicates that the

motor bearings must be changed for a certain period of time during the motor lifetime. Easy bearing installation/disassembly is the basic requirement to ensure the changes of motor bearings.

- Maintenance requirements—Low-maintenance bearing is a prime advantage for some applications such as pitch control motors in offshore wind turbines. With continuous developments in bearing material, better designs, more advanced manufacturing process, lubrication technology, precise production methods, and improved surface finishing in the bearing industry, bearing life has been greatly increased with the reduced maintenance in the last several decades. A longer maintenance cycle and lower maintenance cost mean a reduced life-cycle cost of bearing.
- Shaft and endbell/housing designs—In an electric motor, the shaft and bearings, as well as the endbell/housing, constitute a rotating system, and thus, the selection of bearing can directly affect the designs of the shaft and endbell/housing. As a consequence, it is highly desired to take into account such influences in the earlier stage of the motor design.
- Allowable misalignment of inner/outer rings—Some types of bearing have a self-aligning capability, such as self-aligning ball bearings and spherical roller bearings. Using these types of bearing can help reduce loads that resulted from the misalignment between the motor and driven machine.

However, it is very difficult to satisfy all requirements listed earlier. Each application may have its specific requirements. It is critical for motor engineers to prioritize all these factors for a specific application, make trade-offs, and identify the relatively important factors as the design factors.

6.6.1 Bearing Type Selection Based on Load

Rolling bearings are available in a variety of types, configurations, and sizes. The correct selection of bearing type is mainly determined by the direction, magnitude, and characteristics of loads acting on the bearing. Generally, a thrust ball bearing can only withstand axial loads. By contrast, needle bearings are designed to take only radial loads. In practice, motor bearings have to take both radial and axial loads.

Bearings for practical applications often need to withstand both dynamic and static loads in a variety of magnitudes and directions. Usually, many bearing manufacturers use the equivalent load to determine the bearing load ratings. When both dynamic radial loads and dynamic axial loads act on a bearing simultaneously, the dynamic equivalent radial load P_{dr} (in N) can be determined as

$$P_{dr} = X_d F_{dr} + Y_d F_{da} \tag{6.39}$$

where

F_{dr} and F_{da} are the actual dynamic radial and dynamic axial load (in N)
X_d and Y_d are dynamic radial and dynamic axial load factor, respectively.
X_d and Y_d can be found in bearing catalogs.

Similarly, the static equivalent radial load P_{sr} (in N) can be found from the following equation:

$$P_{sr} = X_s F_{sr} + Y_s F_{sa} \tag{6.40}$$

where

F_{sr} and F_{sa} are the static radial and static axial load (in N)
X_s and Y_s are static radial and static axial load factor, respectively

All values of X_d, Y_s, X_s, and Y_s are provided by manufacturers. The calculated values of P_{dr} and P_{sr} can be used to select the bearing for specific applications.

Couplings are used to connect the motor and driven machine together. The selection of flexible or rigid couplings can strongly influence bearing operation. Improper couplings may introduce additional loads to motor bearings. In addition, using belts or gears as the driving mechanism can produce the transverse force acting on the shaft, which is in turn transmitted to the motor bearings, especially on the drive end bearing. In this case, using roller bearings at the drive end is commonly recommended because of their higher load-carrying capability.

In real bearing applications, there are always some small misalignments between the motor shaft and driven machine shaft. Using flexible couplings can help accommodate the misalignments to some extent.

Different types of bearings have different load capacities. Deep-groove ball bearings can take more axial loads than ball bearings. Cylindrical roller bearings have a stable axial load-carrying capacity. This type of bearings can be safely used as long as the radial load exceeds the axial load by 2.5 times or more. In general, rolling bearings are capable of withstanding greater vibration and shock loads than ball bearings.

6.6.2 Bearing Type Selection Based on Speed

The ball bearing has relatively lower rolling resistances over a wide range of rotating speed than a roller bearing, making it perfectly suitable for medium-/high-speed and low-torque applications. Furthermore, compared with most other types of bearings, ball bearings have superior acoustic characteristics.

By contrast, a roller bearing is in line contact with the inner and outer raceways. Therefore, it can withstand much larger radial loads than the ball bearing. It is interesting to note that at very high speeds, the friction of roller bearing becomes lower than that of ball bearings.

For extremely high-speed motors, noncontact bearings such as magnetic bearings may be selected due to their extremely low friction and wear.

6.6.3 Selection of Bearing Size

Bearing size can be initially selected by the load ratings to determine the required load-carrying capacity with a given factor of safety n_s. Both dynamic and static loads must be evaluated prior to the selection. Dynamic loads are the loads related to the rotor rotation or system displacement (e.g., vehicle). Shock loads are also dynamic loads, which are frequently encountered from heavy impact to earthquake. Usually, dynamic loads are higher than static loads for motors.

In order to ensure bearing normal operation, it has been recommended [6.50] by bearing manufacturers that bearing size should be selected on the basis of static load ratings for the following conditions: (1) the bearing is stationary and is subjected to continuous or intermittent loads, (2) the bearing makes slow oscillating or alignment movements under load, (3) the bearing rotates under load at very slow speed (<10 rpm)

Hesai Exhibit 1031, page 398

and only operates for short life, and (4) the bearing rotates and, in addition to the normal operating loads, has to sustain heavy shock loads.

6.7 Bearing Performance Improvement

The most recent significant advancement in bearing technology is the introduction of ceramic ball bearings. A ceramic bearing is usually made with ferrous inner and outer rings as well as ceramic balls made of silicon nitride steel, Si_3N_4. The advantages of ceramic bearings over steel bearings include lower friction, higher hardness, finer finish, lesser heat generation, and better acceleration capability. Ceramic bearings are usually used in high-speed and high-temperature (up to 500°C–800°C) applications. High-speed electric motors that require voltage isolation often utilize ceramic bearings [6.51].

Other advantages of ceramic bearings over steel bearings include the following:

- They can withstand wear and tear better compared to steel bearings.
- The modulus of elasticity of silicon nitride is 1.5 times higher than steel, indicating that ceramic balls have a smaller contact area and thus lower rolling friction and heat generation.
- The smaller mass of the ceramic balls results in less inertia and gravitational force on the bearing borne.
- Ceramic balls are less vulnerable to lubricating film breakdowns and starved lubrication conditions than steel balls. They need less lubricant and exhibit less lubrication degradation, which contribute to an increased operation lifetime.
- Rotating bearing components may experience electric charging. This leads to superfluous noise and wear out the bearings. Because ceramic bearings usually have poor electrical conductance, electric charging hardly occurs in this type of bearing.

Bearings under contaminated lubrication condition suffer often from indentations that are generated on the bearing rolling surfaces by foreign particles. During bearing operation, high stress concentrations occur in the vicinities of dents and eventually lead to cracking and surface-originating flaking. The stress concentration is expressed as

$$\frac{\sigma}{\sigma_o} \propto \left(\frac{r}{c}\right)^{-0.24} \tag{6.41}$$

where

r is the radius at the shoulder of the dent
c is the shoulder-to-shoulder half width
σ_o is the stress far away from the dent

As shown in Figure 6.40, the greater the ratio of r/c, the smaller the stress concentration and the longer the bearing life.

In order to reduce the stress concentration around surface dents, NSK [6.52] has developed super-TF bearings using the TF technology. This technology is a heat treatment process for optimizing the level of retained austenite in bearing materials [6.53]. It has been

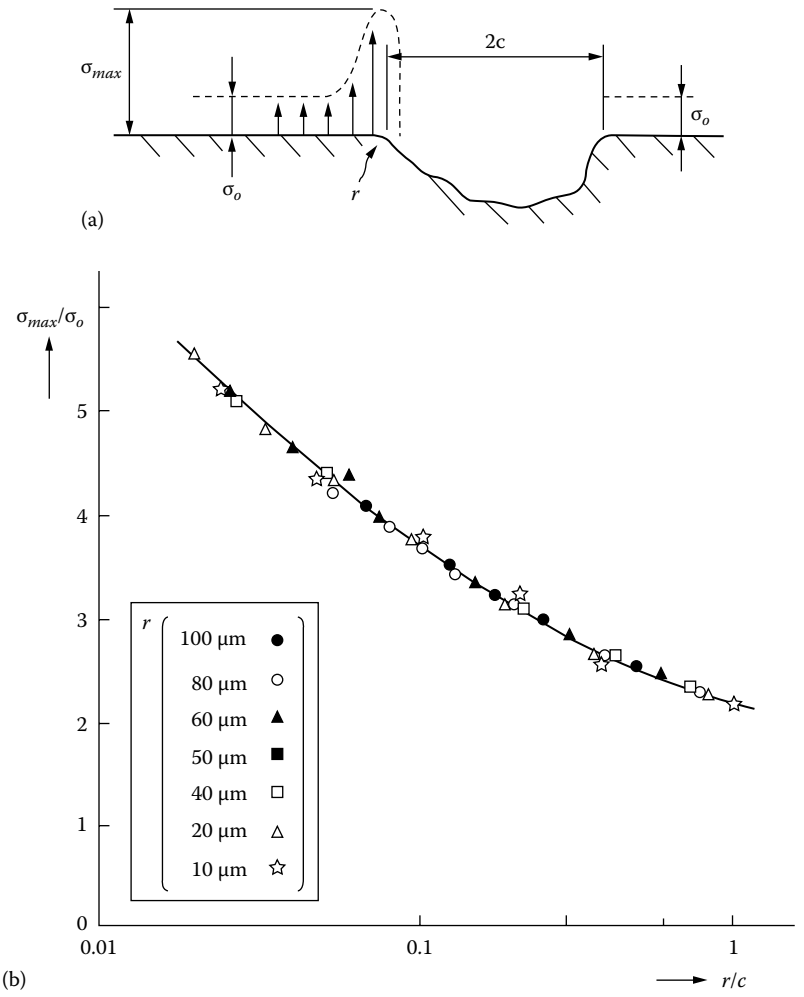


FIGURE 6.40
(a) Stress concentration around a surface dent and (b) the relationship between the stress ratio and the ration of *r/c* (U.S. Patent 4,904,094) [6.52]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

found that the increase in the retained austenite level can significantly raise the ratio of *r/c* and thus greatly improve bearing performance and extend the bearing life.

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Hesai Exhibit 1031, page 403

7

Motor Power Losses

Motor power losses refer to the consumption of electrical energy not converted to useful mechanical energy. Motor power losses are important in motor design for a number of reasons: (1) Power losses in electric motor can substantially increase the temperature rise in the windings and deteriorate the performance characteristics of motor. The vast majority of power losses are converted into heat energy, which must be eventually dissipated from the motor to the surrounding environment. Consequently, it is highly desired to understand the mechanisms of various power losses for providing the best suitable solution of motor cooling. (2) The motor efficiency is defined as the ratio of the power output (which equals the power input minus the power losses) to the power input. Thus, the higher the power losses, the lower the motor efficiency. From an economic standpoint, higher power losses are always associated with increased motor costs. (3) The conversion of power losses into heat energy may cause an excessive internal temperature of motor. Thus, power losses have a strong impact on the selection of cooling modes that are required to keep the motor temperature below the maximum allowed value. (4) Temperature is a major cause of degradation of insulation materials. Excessive motor temperature can accelerate the aging of winding insulations and thus reduce their lifetime. (5) High temperature can significantly reduce magnetic properties of PMs to lower the performance of PM motors. As the temperature becomes higher than the Curie temperature of the magnet, all magnetic properties are eventually lost. (6) High temperature may cause bearing grease to deteriorate, leaving motor bearings without adequate lubrication. (7) A small portion of the total power loss may be converted to sound energy, leading to high acoustic noise emitted from the motor.

In recent years, with a continuous and ever-increasing demand for high-performance and high-efficiency electric motors, a great deal of effort has been directed toward determining power losses of rotating electric machines.

The motor power losses can be briefly defined as no-load and load losses. The major no-load losses, which do not require load currents, include (1) core losses, also called magnetic losses, because alternating magnetic flux produces both hysteresis losses and eddy-current losses in the stator and rotor cores, magnets, and other motor components and (2) mechanical losses that include friction and windage losses. The primary load losses are (1) resistive losses in the stator windings, (2) resistive losses in the rotor windings, and (3) stray load losses that consist of various kinds of losses that are not mentioned previously.

Benhaddadi et al. [7.1] have provided the estimation on the contribution of each type of losses to the total power loss. Their results show that the largest contribution comes from the stator winding (35%). The following are the rotor winding and magnetic losses; each accounts for 20%. The mechanical losses are the lowest, only accounting 10%. GE [7.2] recently released the typical power loss distribution data for a four-pole motor. The largest contribution is still found from stator windings, taking 33% of the total loss. This is consistent with the value of Benhaddadi as 35%. The mechanical (14%) and stray load losses (22%) are moderately higher than but the rotor windings (15%) and magnetic losses (16%) slightly lower than the data in [7.1]. The comparison of the contribution data from these two references is presented in Figure 7.1.

Hesai Exhibit 1031, page 404

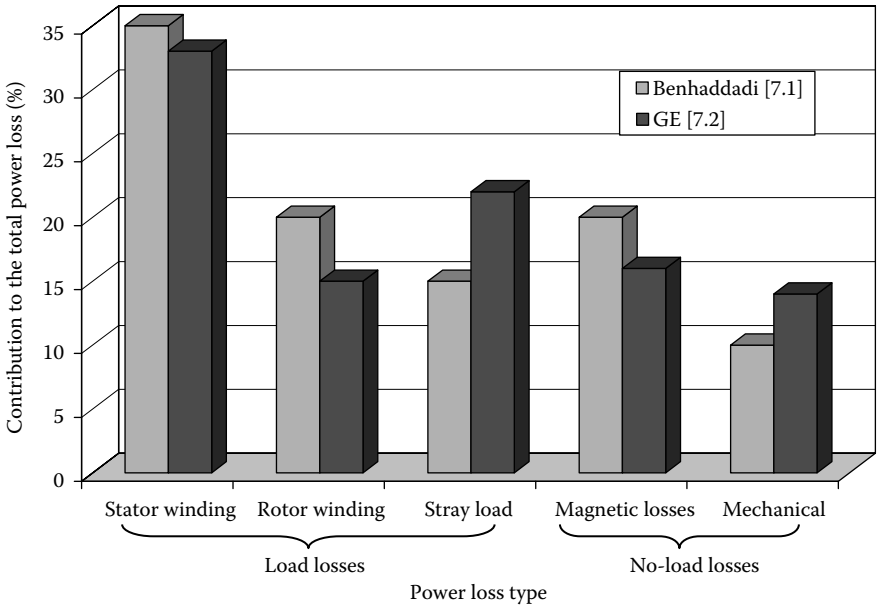


FIGURE 7.1
Comparison of power loss distributions in electric motors.

7.1 Power Losses in Windings due to Electric Resistance in Copper Wires

The most significant energy loss in a motor comes from electrical resistivity of the stator winding and either the rotor winding for a wound-rotor induction motor or the squirrel cage for a squirrel cage induction motors. For common conducting materials such as copper and aluminum, electric current is produced by the movement of electrons from one atom to another under the influence of an electric field. During this process, the migrations of free electrons cause many collisions with the captive electrons in the conducting materials. These collisions consume some energy of electrons as the basic cause of resistance.

In a DC motor, current flows uniformly across the entire cross section of the winding wires. The electrical resistance R_{DC} of a conductor is proportional to the resistivity of the material ρ (in Ω·m) and the conductor length l (in m) and inversely proportional to the cross-sectional area of the conductor A (in m^2), that is,

$$R_{DC} = \frac{\rho l}{A} \tag{7.1}$$

It must be noted that material resistivity is strongly influenced by temperature. A higher temperature typically results in more collisions between electrons, and hence, the higher resistance to electric current. For a common conducting material such as copper and aluminum, electric resistivity is approximately expressed as

$$\rho(T) = \rho(T_o)[1 + \alpha(T - T_o)] \tag{7.2}$$

where
T_o is the reference temperature
α is the temperature coefficient of resistance of the material

Thus, the resistive winding loss can be simply calculated as

$$P_r = I^2 R_{DC} = \frac{V^2}{R_{DC}} \tag{7.3}$$

where
I is the electric current flowing through the winding
V is the voltage across the winding

If there is enough space in the winding slots, the resistive winding loss can be reduced by adopting a conductor with a larger diameter.

For an AC motor, Equation 7.3 can be expressed as

$$\overline{P}_r = i_{rms}^2 R_{AC} = \frac{v_{rms}^2}{R_{AC}} \tag{7.4}$$

where
$\overline{P}_r$ is the averaged power loss
R_{AC} is the AC electrical resistance
i_{rms} and v_{rms} are the RMS current and voltage, respectively

For AC current with sine waves, i_{rms} and v_{rms} can be expressed as

$$i_{rms} = \frac{i_{peak}}{\sqrt{2}}; \quad v_{rms} = \frac{v_{peak}}{\sqrt{2}} \tag{7.5}$$

respectively. In Equation 7.5, i_{peak} and v_{peak} are the peak value of AC current and voltage, respectively.

The resistance loss in an AC motor increases with frequency due to the skin effect, which refers to the tendency of AC current to distribute in the thin layer at the conductor's outer surface. The skin depth is defined as the distance below the conductor surface where the current density has decayed to e^{-1} ($e^{-1} = 0.368$) of its value at the surface (Figure 7.2). The skin depth δ (in m) varies as the square root of the electrical resistivity and the inverse square root of frequency and relative permeability:

$$\delta = \sqrt{\frac{\rho}{\pi f \mu_r \mu_o}} \tag{7.6}$$

where
f is the frequency (Hz)
μ_r is the relative permeability
$\mu_o = 4\pi \times 10^{-7}$ (H/m) is the permeability of free space

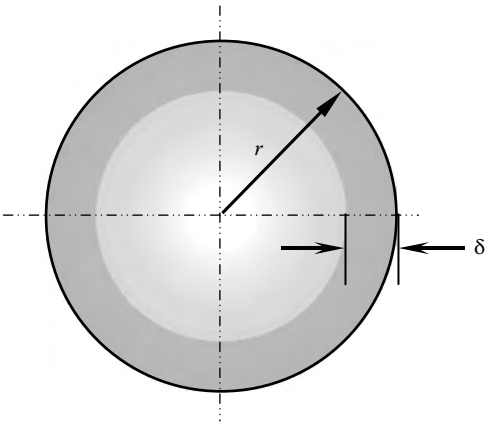


FIGURE 7.2
Skin depth of a conducting magnet wire, where $\delta < r$.

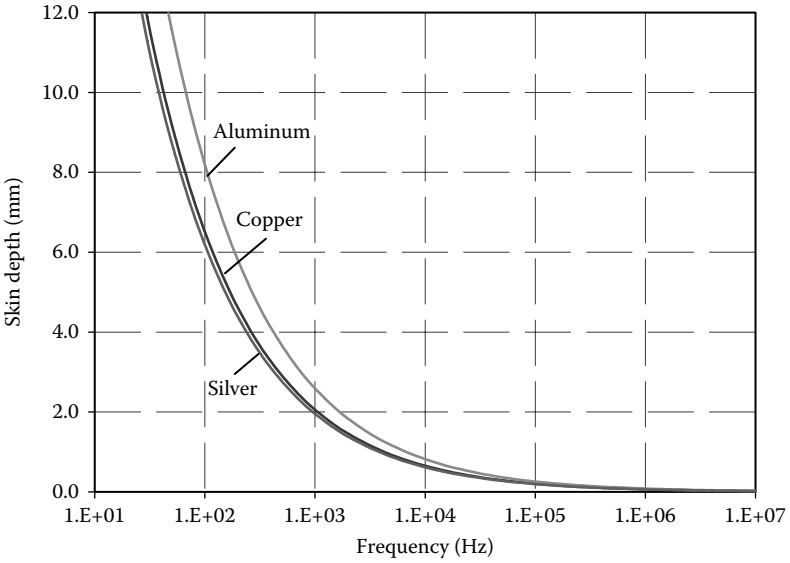


FIGURE 7.3
Variations of skin depth with frequency for different materials.

The variations of skin depth with frequency for different materials are displayed in Figure 7.3. It has shown that the skin depths for all materials reduce sharply for $f < 1$ kHz. As f becomes larger than 100 kHz, the changes in the skin depth can be ignored. For good conducting materials, skin depth varies as the inverse square root of the material conductivity. This indicates that better conducting materials have a reduced skin depth.

The ratio of AC resistance to DC resistance can be simply determined as

$$\frac{R_{AC}}{R_{DC}} = \frac{1}{2\delta/r - (\delta/r)^2} \tag{7.7}$$

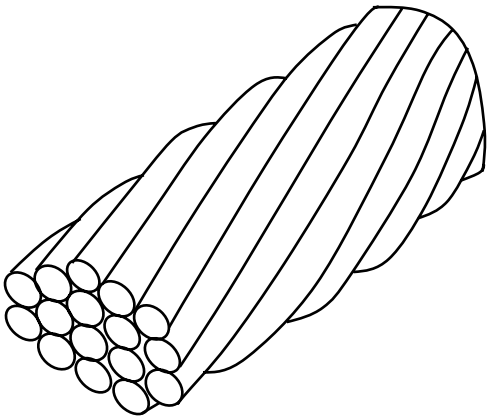


FIGURE 7.4
Litz wires used for motor windings for minimizing skin effect.

In case that $\delta \ll r$, Equation 7.7 can be simplified as

$$\frac{R_{AC}}{R_{DC}} \approx \frac{r}{2\delta} \tag{7.8}$$

This equation indicates that the smaller the skin depth, the higher the ratio of AC to DC resistance. More advanced and complete expression of the resistance ratio was derived by Ramo et al. [7.3].

The skin effect appears as the skin depth δ is smaller than the wire radius r. In order to minimize the resistance loss for high-frequency applications, the copper windings are made with multiple strands, insulated from each other and in a weaving or twisting pattern (so-called Litz wire, see Figure 7.4). It was found that the slotless structure combined with a 75-strand Litz wire in a superhigh-speed cryogenic PM motor can significantly reduce iron loss in the rotor without increase in eddy-current loss in the winding [7.4].

The resistive winding loss is the loss that is proportional to the motor load. With the increase in the motor torque, the electric current through the motor windings increases, so does the resistive winding loss. In most applications, the resistive winding loss is predominant among all losses, except motors having very high rotating speeds.

7.2 Eddy-Current and Magnetic Hysteresis Losses

7.2.1 Eddy-Current Loss

From Faraday's law of induction, when magnetic flux changes through a loop of area due to any reasons (the magnetic field change, the loop area change, or the orientation of the loop relative to the magnetic field change), an induced EMF is developed along the loop. Thus, circulating currents, that is, eddy currents, are induced in bulk conductors moving through a magnetic field.

Eddy currents are often undesirable because they consume a considerable amount of energy to internal energy without making useful work. For electric motors, eddy-current loss is caused by local circulating currents induced in conductive core components and results in the rise in temperature.

Eddy currents are affected by the electrical resistance of ferromagnetic materials in which eddy currents flow. In order to reduce eddy-current loss, both stator and rotor cores are made of a stack of laminations, which are insulated by nonconducting materials such as lacquer or metal oxide. This layered structure prevents large current loops and effectively confines eddy currents to small loops in individual layers. The sum of the individual eddy current in each piece of the laminated core is much less than that in the solid iron core. Furthermore, the addition of the silicon element into the core steel increases electrical resistance, leading to the further decrease of eddy current.

7.2.2 Magnetic Hysteresis Loss

Magnetic hysteresis phenomena occur in ferromagnetic materials. As an external alternating magnetic field is applied to a ferromagnetic material, it forces the atomic dipoles in the material to align themselves with the magnetic field. When the magnetic field suddenly changes its orientation relatively with the material, the atomic dipoles must realign themselves to accommodate such change. Because the ferromagnetic material tends to retain some degree of magnetization, known as the magnetic hysteresis, it must take a certain amount of energy to overcome such a hysteresis to adjust the atomic dipoles according to the change of the external magnetic field. In fact, hysteresis losses depend on several factors including the power frequency, the peak flux density, the material of core steel, and the orientation of the magnetic flux with respect to the grain structure of the steel.

7.2.3 Calculations of Eddy-Current and Magnetic Hysteresis Losses

Maxwell's equations, which perfectly describe classical electromagnetic phenomena, consist of four equations: Faraday's law of induction, Ampère–Maxwell's law, and Gauss's laws for the electric and magnetic fields, that is,

Faraday's law of induction

$$\nabla \times \mathbf{E} = -\frac{\partial \mathbf{B}}{\partial t} \tag{7.9}$$

Ampère–Maxwell's law

$$\nabla \times \mathbf{H} = \mathbf{J} + \frac{\partial \mathbf{D}}{\partial \mathbf{t}} \tag{7.10}$$

Gauss's law for electric field

$$\nabla \cdot \mathbf{D} = \rho_e \tag{7.11}$$

Gauss's law for magnetic field

$$\nabla \cdot \mathbf{B} = 0 \tag{7.12}$$

Hesai Exhibit 1031, page 409

In Maxwell's equations, **E** and **H** are the electric and magnetic field intensities, **D** and **B** are the electric and magnetic flux densities, respectively, **J** is the electric current density, and ρ_e is the electric charge density of any external charges.

The relationships between the current density **J** and the electric field intensity **E**, electric flux density **D** and the electric field intensity **E**, and the magnetic flux density **B** and magnetic field intensity **H** are given as

$$\mathbf{J} = \sigma\mathbf{E} \tag{7.13a}$$

$$\mathbf{D} = \varepsilon\mathbf{E} \tag{7.13b}$$

$$\mathbf{B} = \mu\mathbf{H} \tag{7.13c}$$

where

σ is the electrical conductivity
ε is the electric permittivity
μ is the magnetic permeability

Using Equations 7.9 and 7.10, it can be derived that

$$\nabla\cdot(\mathbf{E}\times\mathbf{H}) = \mathbf{H}\cdot(\nabla\times\mathbf{E}) - \mathbf{E}\cdot(\nabla\times\mathbf{H}) = -\mathbf{H}\cdot\frac{\partial\mathbf{B}}{\partial t} - \mathbf{E}\cdot\left(\mathbf{J}+\frac{\partial\mathbf{D}}{\partial\mathbf{t}}\right) \tag{7.14}$$

Integrating both sides over a closed volume *V* surrounded by the surface *S*, Equation 7.14 is expressed in the integrated form,

$$\int_V \nabla\cdot(\mathbf{E}\times\mathbf{H})\,dV = -\int_V\left[\mathbf{H}\cdot\frac{\partial\mathbf{B}}{\partial t}+\mathbf{E}\cdot\left(\mathbf{J}+\frac{\partial\mathbf{D}}{\partial t}\right)\right]dV \tag{7.15}$$

Using the divergence theorem, it yields

$$\int_S(\mathbf{E}\times\mathbf{H})\,d\mathbf{S} = -\int_V\left[\mathbf{H}\cdot\frac{\partial\mathbf{B}}{\partial t}+\mathbf{E}\cdot\left(\mathbf{J}+\frac{\partial\mathbf{D}}{\partial t}\right)\right]dV \tag{7.16}$$

The left side of Equation 7.16 represents the energy flow into the volume per unit time. The first and second terms at the right-hand side represent the energy stored and dissipated per unit time, respectively. Thus, the total energy loss in the volume over a complete cycle of time becomes

$$P = \int_V\int_t\left[\mathbf{H}\cdot\frac{\partial\mathbf{B}}{\partial t}+\mathbf{E}\cdot\left(\mathbf{J}+\frac{\partial\mathbf{D}}{\partial t}\right)\right]dt\,dV \tag{7.17}$$

Since the displacement current **D** is typically relevant only at radio frequencies (in the MHz regime), the term $\partial\mathbf{D}/\partial\mathbf{t}$ can be ignored in Equation 7.17.

Using Equation 7.13a, it gives that

$$\mathbf{E}\cdot\mathbf{J} = \frac{J^2}{\sigma} \tag{7.18}$$

Equation 7.17 can be rewritten as

$$P = \int_V \left(\int_B HdB \right) dV + \int_V \left[\int_t \left(\frac{J^2}{\sigma} \right) dt \right] dV = P_h + P_e \tag{7.19}$$

where P_h and P_e are the power losses due to hysteresis and eddy current, respectively.

Using a lumped circuit approach and assumed eddy-current paths, eddy-current loss of magnetic core (in unit of W) can be derived as [7.5]

$$P_e = \frac{\pi^2}{6} V_c B^2 f^2 a^2 \sigma = K_e V_c B^2 f^2 \tag{7.20}$$

$$K_e = \frac{\pi^2 a^2 \sigma}{6} \tag{7.21}$$

where

V_c is the volume of magnetic core in unit of m^3
a is the lamination thickness in unit of m
B is the peak flux density

Equation 7.20 has shown that eddy-current loss is proportional to the square of lamination thickness a. This indicates that eddy-current loss can be greatly reduced if thinner laminations are adopted. For instance, by reducing the lamination thickness by 20%, eddy-current loss can be reduced by 36%.

Some textbooks use eddy-current loss per unit volume, in the unit of W/m^3:

$$p_e = K_e B^2 f^2 \tag{7.22}$$

Hysteresis refers to the phenomenon in which the magnetic induction of a ferromagnetic material lags behind the changing magnetic field. When a ferromagnetic material is subjected to a magnetic field, the magnetic particles in the material tend to line up with the magnetic field. As the magnetic field keeps changing its direction, the magnetic particles try to align themselves with the magnetic field. The continuous movement of the magnetic particles thus produces molecular friction, resulting in energy loss.

Hysteresis loss is strongly affected by material electromagnetic properties. The empirical formula expressing hysteresis loss per unit volume was developed by Steinmetz [7.6–7.8] as follows:

$$P_h = K_h V_c B^n f \tag{7.23}$$

where

K_h is the hysteresis coefficient
n is the Steinmetz coefficient, which has a value between 1.6 and 2.3 for most modern magnetic materials (most commonly is around 2)
f is the frequency of magnetization

The hysteresis coefficients of some materials are listed in Table 7.1 [7.9].

TABLE 7.1

Hysteresis Coefficients of Some Metallic Materials

Material	Hysteresis Coefficient K_h (J/m³) × 10⁻²
Cast iron	27.63–40.2
Sheet iron	10.05
Cast steel	7.54–30.14
Hard cast steel	63–70.34
Silicon steel (4.8% in Si)	1.91
Hard tungsten steel	145.7
Good dynamo sheet steel	5.02
Mild steel castings	7.54–22.61
Nickel	32.66–100.5
Permalloy	0.25

From Table 7.1, it can be seen that permalloy has the lowest hysteresis coefficient, and, thus, it has been extensively used in transformer laminations and magnetic recording heads with medium or high frequencies. Silicon steel is the material having the second lowest hysteresis coefficient. As a result, hysteresis loss can be mitigated by making the motor cores with silicon steel. In practice, industrial applications of silicon steel vary in quantities from the few ounces used in small relays or pulse transformers to tons used in generators, motors, and transformers.

It is apparent that eddy-current loss P_e increases with the square of the frequency ($P_e \propto f^2$), and hysteresis loss P_h increases linearly with the frequency ($P_h \propto f$). This indicates that for high frequencies, eddy-current loss P_e becomes dominant among the total energy loss P.

Magnetic hysteresis and eddy-current losses are sometimes collectively known as iron losses or core losses. This type of losses is a significant fraction of total losses of electric motors. With advanced FEA tools, the contributions of the hysteresis and eddy-current losses in motor components can be determined fairly well. As an example, Figure 7.5 plots the data from a small motor, showing the rotor's hysteresis and eddy-current losses

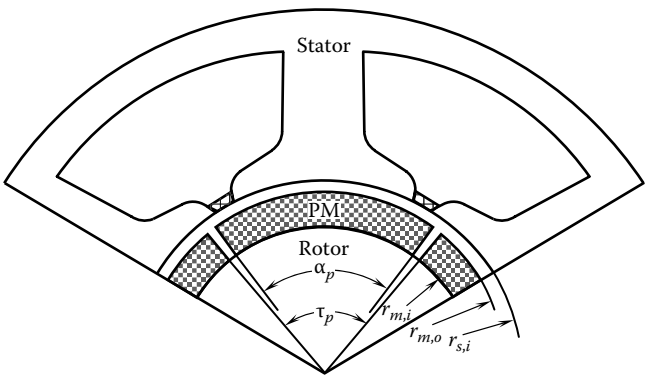


FIGURE 7.5
Eddy-current losses of PMs in PM motors having a fractional number of slots per pole.

as a function of rotor rotating speed (or equivalently frequency). The eddy-current trend shows a parabolic behavior, while the hysteresis loss follows a power law behavior as expected.

7.2.4 Losses in Stator and Rotor Iron Cores

The power losses in the stator and rotor cores consist of both hysteresis and eddy-current losses. In electric motors, the core or iron losses usually take about 20%–25% of the total power losses [7.10]. For PM motors, interior PM machines have significantly higher full-load iron losses than SPM machines.

By dividing eddy current into classical and excess eddy currents for more accurate analysis, power losses in stator and rotor iron cores can be determined as [7.11,7.12]

$$P_{core} = k_h B^2 f + K_c (Bf)^2 + K_e (Bf)^{3/2} \tag{7.24}$$

where

K_h, K_c, and K_e are the coefficients of hysteresis loss, classical eddy-current loss, and excess eddy-current loss, respectively

B is the peak flux density

The coefficients can be obtained using the curve fitting of the iron loss data from manufacturers.

There are a number of ways to reduce the core losses: (1) Utilizing a *soft* magnetic material. This is because the hysteresis loss is proportional to the shaded area enclosed by the hysteresis loop on the B–H curve. A *soft* magnetic material has a smaller shaded area within the hysteresis loop than that of a *hard* magnetic material (see Figure 7.6). (2) Using silicon grain-oriented steel. (3) Using thinner steel laminations. (4) Designing longer cores to reduce magnetic flux density.

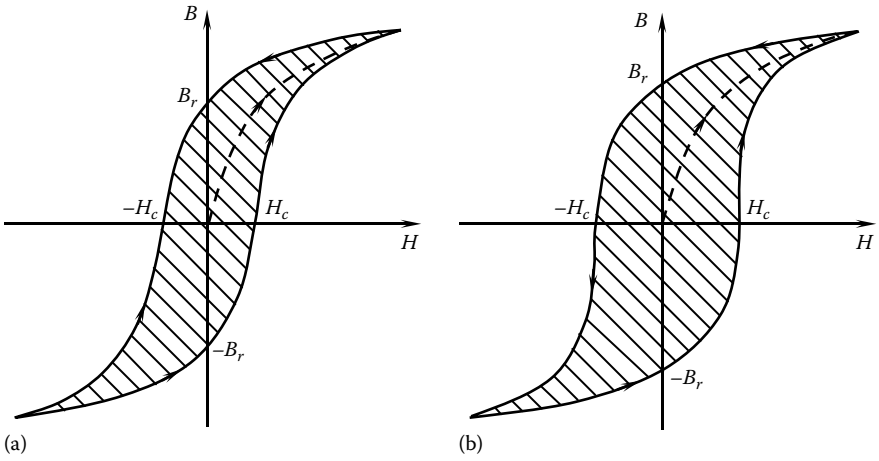


FIGURE 7.6
Magnetization hysteresis loops for different ferromagnetic materials: (a) *soft* magnetic materials with low hysteresis loss, which are desirable for motor cores to minimize power losses, and (b) *hard* magnetic materials with high hysteresis loss.

7.2.5 Losses in PMs

PM motors are extensively used in many applications for their excellent characteristics such as high power density, high efficiency, and good dynamic performance. However, eddy-current loss in PMs is often overlooked. For high-speed PM motors, eddy-current loss in PMs cannot be ignored. Eddy-current loss can be estimated as [7.13]

$$P_m \approx \frac{V_m b_m^2 B^2 f^2}{12\rho_m} \tag{7.25}$$

where
V_m is the magnet volume
b_m is the magnet width
ρ_m is the magnet resistivity ($\rho = 1/\sigma$)

Correspondingly, eddy-current loss becomes

$$P_m = K_m V_m B^2 f^2 \tag{7.26}$$

where

$$K_m = \frac{b_m^2}{12\rho_m} \tag{7.27}$$

An analytical model for predicting eddy-current losses in SPMs of PM motors was presented by Ishak et al. [7.14]. As shown in Figure 7.7, PMs are attached on the surface of the rotor that rotates at the speed of ω_r. The magnet has the inner radius $r_{m,i}$ and outer radius $r_{m,o}$.

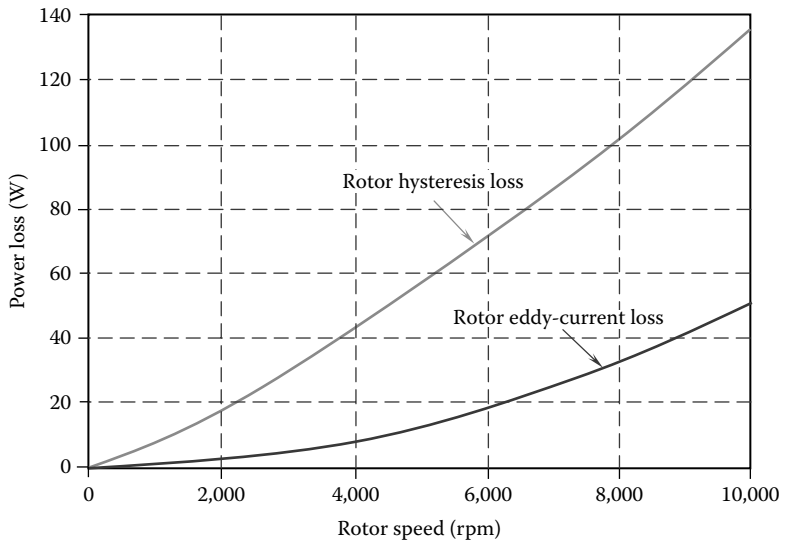


FIGURE 7.7
Rotor hysteresis and eddy-current losses as functions of rotor rotating speed from a sampler motor.

The inner stator bore radius is $r_{s,i}$. Thus, the air gap is given as the difference between $r_{s,i}$ and $r_{m,o}$. The magnet central angle corresponding to the magnet width denotes α_p.

For a PM motor with a slot number N_{slot} and pole number $2p_r$ (where p_r is the number of pole pairs), the number of slots per pole gives

$$n_{slot} = \frac{N_{slot}}{2p_r} \tag{7.28}$$

When n_{slot} is a noninteger number, the eddy-current loss per unit axial length in each magnet can be predicted as

$$P_m = 2p_r \frac{\omega_r}{2\pi} \int_0^{2\pi/\omega_r} \int_{r_{m,i}}^{r_{m,o}} \int_{-(\alpha_p/2)}^{\alpha_p/2} \rho_m J_m^2 r dr d\theta dt = \sum_u \sum_v (P_{cuv} + P_{auv}) \tag{7.29}$$

where

u is the time-harmonic order in the phase current waveform
v is the space-harmonic MMF order
ρ_m is the electrical resistivity of the magnets

The variables P_{cuv} and P_{auv} are given by

$$P_{cuv} = \frac{\mu_0^2 \alpha_p}{\rho_m} p_r \sum_u \sum_v \frac{J_{uv}^2 (up_r \pm vp_s)^2 \omega_r^2}{\left[1 - (r_{m,i}/r_{s,i})^{2vp_s}\right]^2 (vp_s)^2}$$
$$\times \left\{ \left(\frac{r_{m,o}}{r_{s,i}}\right)^{2vp_s} \frac{r_{s,i}^2 r_{m,o}^2}{2vp_s + 2} \left[1 - \left(\frac{r_{m,i}}{r_{m,o}}\right)^{2vp_s+2}\right] + \left(\frac{r_{m,i}}{r_{s,i}}\right)^{2vp_s} r_{s,i}^2 (r_{m,o}^2 - r_{m,i}^2) + \left(\frac{r_{m,i}}{r_{s,i}}\right)^{2vp_s} r_{s,i}^2 r_{m,i}^2 F_v \right\} \tag{7.30}$$

$$P_{auv} = \left[-\frac{8\mu_0^2}{\alpha_p \rho_m} p_r \sum_u \sum_v \frac{J_{uv}^2 H_v^2}{\left[1 - (r_{m,i}/r_{s,i})^{2vp_s}\right]^2 (vp_s)^4} \frac{(up_r \pm vp_s)^2 \omega_r^2}{(r_{m,o}^2 - r_{m,i}^2)} \sin^2\left(vp_s \frac{\alpha_p}{2}\right) \right] \tag{7.31}$$

where p_s is the number of pole pairs in the stator winding MMF. In the above equations,

$$F_v = \begin{cases} \dfrac{\left(\dfrac{r_{m,o}}{r_{m,i}}\right)^{-2vp_s+2} - 1}{-2vp_s + 2} & \text{for } vp_s \neq 1 \\ \ln\left(\dfrac{r_{m,o}}{r_{m,i}}\right) & \text{for } vp_s = 1 \end{cases} \tag{7.32}$$

$$H_v = \left(\frac{r_{m,o}}{r_{s,i}}\right)^{vp_s} \frac{r_{s,i} r_{m,o}^2}{vp_s + 2} \left[1 - \left(\frac{r_{m,i}}{r_{m,o}}\right)^{vp_s+2}\right] + \left(\frac{r_{m,i}}{r_{s,i}}\right)^{vp_s} r_{s,i} r_{m,i}^2 E_v \tag{7.33}$$

where

$$E_v = \begin{cases} \dfrac{\left(\dfrac{r_{m,o}}{r_{m,i}}\right)^{-vp_s+2} - 1}{-vp_s + 2} & \text{for } vp_s \neq 2 \\ \ln\left(\dfrac{r_{m,o}}{r_{m,i}}\right) & \text{for } vp_s = 2 \end{cases} \tag{7.34}$$

This model has been validated by FEA.

7.2.6 Power Losses in Other Core Components

Eddy-current and hysteresis losses may occur in other core components such as stator end plates. In most cases, the contributions of the losses from these components to the total system power loss are relatively small and thus can be ignored in design.

7.3 Mechanical Friction Losses

Mechanical losses in an electric motor refer to the losses due to mechanical friction between two or among more motor components as they contact each other and have relative movements.

Whenever there is relative motion of two surfaces in contact, there is frictional resistance between two surfaces. The friction force opposes the relative motion of two bodies. The friction losses are independent of load and dependent on the speed of the machine. For the electric rotating machinery, a minimum of friction is highly desired.

7.3.1 Bearing Losses

Bearings are key components for rotating machinery to support rotating components with as little friction as possible. In normal operation, bearing rolling elements (balls or rollers) spin and slide between the inner and outer raceways (Figure 7.8). Therefore, the frictional resistance is produced between the rolling elements and their contact components such as the raceways, the cage, the rubbing seal (if present), and the lubricant (Figure 7.9). It has been estimated that among the total power losses of motor-driven equipment (e.g., pumps, fans, and compressors), about 20% of the loss is due to various mechanical losses, and of this, roughly 20% is attributed to bearings. This indicates that the bearing loss counts for about 4% of the total power losses [7.15].

There are a number of factors impacting on the resistance to rotation of the bearing, for instance, the bearing load, the shaft rotating speed, the bearing dimensions, the roughness of the rolling elements and raceways, the lubrication condition, and the lubricant properties.

As an important performance index, the friction torque, defined as a resistance torque when a rolling bearing rotates, has a strong impact on the bearing's working condition and lifetime. The early approach to calculate the friction torque was proposed by

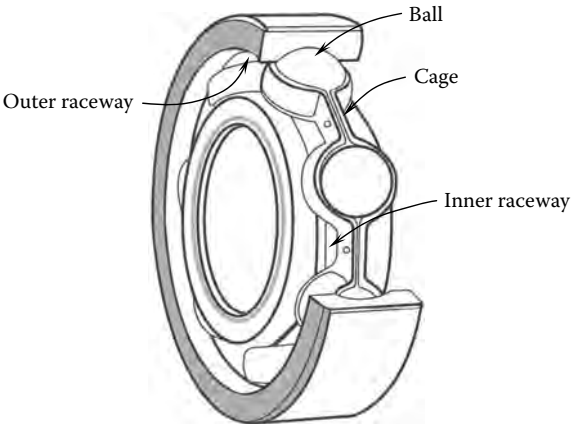


FIGURE 7.8
Ball bearing structure.

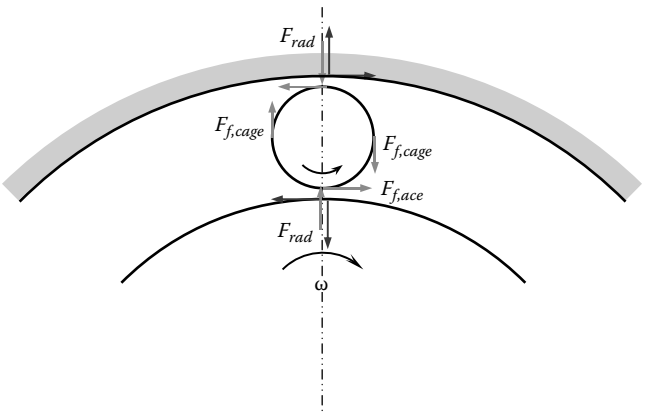


FIGURE 7.9
Bearing losses due to friction between balls and raceways, balls and the cage, and balls and the lubricant.

Palmgren [7.16,7.17]. In this approach, the total friction torque is the sum of the load friction torque T_l and the viscous friction torque T_v, that is,

$$T = T_l + T_v \tag{7.35}$$

The load friction torque is determined empirically as

$$T_l = f_1 F_\beta D_m \tag{7.36}$$

where

F_β depends on the magnitude and direction of the applied load
D_m is the bearing mean diameter, $D_m = (D_i + D_o)/2$
f_1 is a factor depending on the bearing type and bearing load expressed as

$$f_1 = z\left(\frac{F_s}{C_s}\right)^y \tag{7.37}$$

where
F_s is the static equivalent load
C_s is the basic static load rating
z and y are constants, depending on the bearing type

For radial deep-groove ball bearings, $z = 0.0004–0.0006$ and $y = 0.55$.

For bearings that operate at moderate speeds, the viscous friction torque can be estimated as

$$T_v = 10^{-7} f_o(\nu_o n)^{2/3} D_m^3 \quad \nu_o n \geq 2000 \tag{7.38}$$

$$T_v = 160 \times 10^{-7} f_o D_m^3 \quad \nu_o n < 2000 \tag{7.39}$$

where
ν_o is the kinematic viscosity of lubricant in centistokes
n is the bearing rotating speed in rpm
f_o is a factor depending on the bearing type and the lubrication method

In recent years, based on more advanced computational models, the new approach has been developed by SKF to calculate the total frictional moment (in units of N-mm) with the identified four sources of friction in every contact occurring in the bearing [7.18]:

$$M = \phi_{ish}\phi_{rs} M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{7.40}$$

where
ϕ_{ish} is the inlet shear heating reduction factor
ϕ_{rs} is the kinematic replenishment/starvation reduction factor
M_{rr} is the rolling frictional moment
M_{sl} is the sliding frictional moment
M_{seal} is the frictional moment of seal
M_{drag} is the frictional moment of drag losses, churning, splashing, etc.

The reduction factors ϕ_{ish} and ϕ_{rs} are introduced in the new friction model to account for the effects of inlet shear heating reduction and high-speed replenishment/starvation and of rolling friction, respectively.

The rolling frictional moment M_{rr} can be estimated as

$$M_{rr} = G_{rr}(vn)^{0.6} \tag{7.41}$$

where
n is the bearing rotating speed in rpm
v is the kinematic viscosity of the lubricant at the operating temperature
G_{rr} depends on the bearing type, the bearing mean diameter D_m [where $D_m = (D_i + D_o)/2$], the radial and axial loads, F_r and F_a, respectively.

TABLE 7.2

Sliding Friction Coefficient μ_{sl}

Bearing Type	Lubrication	μ_{sl}
Ball	Mineral oils	0.05
	Synthetic oils	0.04
	Transmission fluids	0.1
Cylindrical roller	—	0.02
Tapped roller	—	0.002

The sliding frictional moment M_{sl} is given as

$$M_{sl} = \mu_{sl} G_{sl} \tag{7.42}$$

where

μ_{sl} is the sliding friction coefficient (see Table 7.2)

G_{sl} depends on the bearing type, the bearing mean diameter D_m, and the radial and axial loads, F_r and F_a, respectively.

The bearing seal frictional moment M_{seal} can be estimated as

$$M_{seal} = K_1 D_s^{\alpha} + K_2 \tag{7.43}$$

where

K_1 is a constant depending on the bearing type

K_2 is a constant depending on bearing and seal type

D_s is the seal counterface diameter

α is an exponent depending on bearing and seal type

The friction torque of drag losses T_{drag} are

$$M_{drag} = V_M K_{ball} D_m^5 n^2 \quad \text{for ball bearing} \tag{7.44a}$$

$$M_{drag} = 10 V_M K_{roll} w D_m^4 n^2 \quad \text{for roller bearing} \tag{7.44b}$$

where

V_M is a function of the oil level

w is the bearing width

K_{ball} and K_{roll} are the ball- and roller-related constants, respectively,

$$K_{ball} = \frac{i_{rw} K_z (D_o + D_i)}{D_o - D_i} \times 10^{-12} \tag{7.45}$$

$$K_{roll} = \frac{K_L K_z (D_o + D_i)}{D_o - D_i} \times 10^{-12} \tag{7.46}$$

where

i_{rw} is the number of ball rows

K_z and K_L are the bearing-type and roller-bearing-type related geometry constants, respectively

The total power loss in a bearing as a result of bearing friction can be obtained using the empirical formula [7.19]

$$P_b = (1.05 \times 10^{-4}) Mn \tag{7.47}$$

where

n is the bearing rotating speed (in rpm)

M is the total frictional moment of the bearing (in N m)

7.3.2 Sealing Losses

Some motors work under harsh environmental conditions. In order to ensure motor's normal operation, mechanical seals are installed on motor shafts to prevent foreign contamination such as dusts and wetness from the outside. When a shaft rotates, the seal tips must always exert a certain radial load on the shaft surface, resulting in friction losses at the sealing locations.

The friction force on the shaft is obtained by integrating the shear stress τ over the shaft surface:

$$F_f = R \int_0^L \int_0^{2\pi} \tau \, d\theta \, dz \tag{7.48}$$

Thus, the sealing losses are

$$P_{seal} = 2\pi R F_f \mu n = 2\pi R^2 \mu n \int_0^L \int_0^{2\pi} \tau d\theta \, dz \tag{7.49}$$

where

R is the shaft radius

μ is the friction coefficient

n is the shaft rotating speed in units of rpm

7.3.3 Brush Losses

In a brushed motor such as an induction DC motor, brushes are in contact with the commutator to carry current to coils. The commutator is the rotary electrical switch, which reverses the direction of electric current periodically between the rotor and the external circuit. In general, brush losses consist of two parts. One is the electric resistance, consisting of the contact resistance occurring at the brush–commutator interface and the electric resistance through the graphite brushes. The power loss due to the contact resistance depends on the contact pressure, contact surface area, surface smoothness, and contamination condition. The I^2R power loss depends on the brush material and brush temperature. Another is the mechanical friction resistance due to the sliding contact of the brushes on the commutator, causing the friction loss. This type of power loss is proportional to the brush contact pressure p_c, kinetic friction coefficient between the brush and commutator μ_f, contact area A_c, and commutator tangential speed $r\omega$, that is,

$$P_b = \mu_f p_c A_c (r\omega) \tag{7.50}$$

where ω is the commutator rotating speed in units of rad/s.

The combination of these two effects results in heating at the brush contact interface and in the brush. Brush wear is resulted from both the mechanical friction and electrical erosion.

7.4 Windage Losses

Windage in a rotating electric machine is defined as the resisting influence of fluid (air or liquid) against not only rotating but also stationary components, creating power losses. In an electric motor, the windage power loss P_w increases exponentially as a function of the rotor rotating speed ω and the rotor radius r_r. In practice, windage losses of electric motors are very important in the motor design and optimization, primarily consisting of the following components:

- The friction loss due to the viscous shear effect in the fully developed turbulent flow. It was reported that such a loss could be up to 30% of the total windage loss.
- The dynamic loss (or rotor body loss in some references) due to driving air in the air gap. The dynamic loss depends strongly upon the thermophysical properties of air such as density and viscosity and, in turn, the air temperature in the rotor–stator air gap.
- The loss associated with the stator slots with unfilled openings. The penetration of the air flow into the slot openings and the formation of the flow recirculations in such slot openings can have complicated influence on energy loss.
- The loss associated with the roughness on rotor surfaces.
- The windage loss of some motor components such as fans.
- The ventilation path loss.

In practical engineering design, it is highly desired to minimize these losses to enhance the machine performance and efficiency.

Windage losses can be very significant for high-speed motors due to the high windage friction. There are three components of velocity in the air gap of electric motors: (1) tangential flow due the rotor rotation, (2) axial cooling flow passing through the air gap, and (3) Taylor vortices due to centrifugal forces. It is the reaction among these velocity components that determines the complex velocity field in the air gap. The importance of each velocity component depends on rotor rotating speed, cooling air flowrate, thermophysical properties of coolant (particularly density, specific heat, and thermal conductivity), rotor and stator geometries, and air-gap dimension. The velocity and pressure fields at the end-winding regions are extremely complex because of a number of factors: a large number of components, windings with irregular geometries and existing voids inside the windings, complexity in cooling flow paths, splitting and mixing of coolant, and interaction between rotor-induced flows and nonrotating flows from stationary sections [7.20]. Modern motors are often designed to small-volume, high-speed machines to increase power design and reduce overall weight. These high-speed machines may lead to a small air gap between the rotor and stator. All these factors can result in high windage loss.

Windage losses vary on a case-by-case basis, depending largely on machine type, size, rated power, coolant type, cooling methods, and many others. As an example shown in

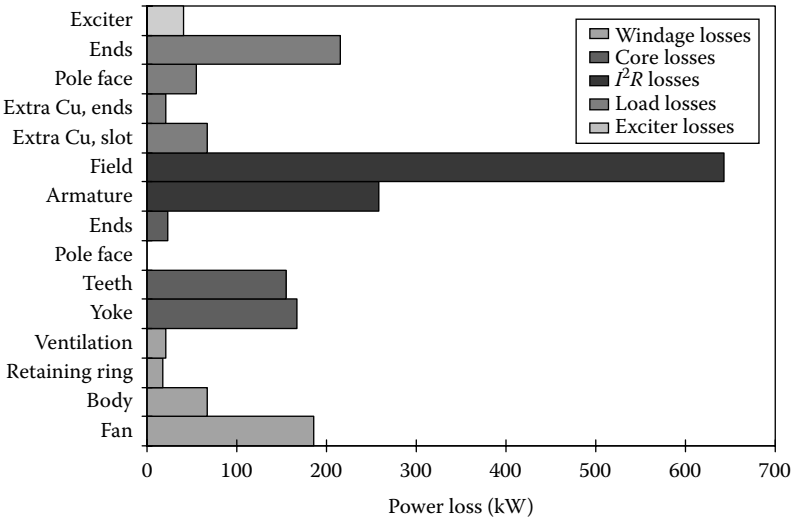


FIGURE 7.10
Various power losses for a high-power electric machine (machine *A*).

Figure 7.10 for a large-size, high-power electric machine, the overall windage loss accounts for 15.1% to the total power loss of the machine. However, for some improperly designed machines, the contribution of windage losses to the total power loss can be very large. As an example shown in Figure 7.11, the overall windage loss accounts for more than one-third, up to 37.6%. The comparison of power losses of these two machines is presented in Table 7.3.

Because of the complicated nature of the flow field inside a motor, it becomes more popular today to use CFD tools to determine the windage losses of electric motors.

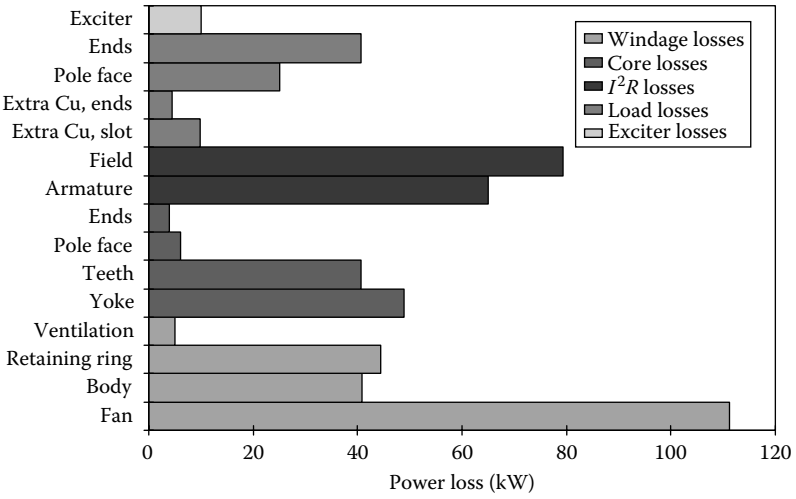


FIGURE 7.11
Various power losses for a high-power electric machine (machine *B*).

TABLE 7.3

Comparison of Power Losses in Two Large-Size, High-Power Electric Machines

Component Loss	Machine 1 Power Loss (kW)		Machine 1 Power Loss (kW)	
Fan	186 (9.6%)	Windage 291.7 (15.1%)	111.3 (20.8%)	Windage 201.5 (37.6%)
Rotor body	67.2 (3.5%)		40.8 (7.6%)	
Retaining ring	17.5 (0.9%)		44.5 (8.3%)	
Ventilation	21 (1.1%)		4.9 (0.9%)	
Yoke	167.5 (8.7%)	Iron (core) 345.4 (17.8%)	48.9 (9.1%)	Iron (core) 99.6 (18.6%)
Teeth	155.3 (8.0%)		40.6 (7.6%)	
Pole face	0 (0%)		6.1 (1.1%)	
Ends	22.6 (1.2%)		4 (0.8%)	
Armature	258 (13.3%)	Copper (I^2R)	65 (12.1%)	144.4
Field	643 (33.2%)	901 (46.5%)	79.4 (14.8%)	(27.0%)
Extra Cu, slot	66.6 (3.4%)	Load 358 (18.5%)	9.9 (1.9%)	Load 80.1 (15.0%)
Extra Cu, ends	21 (1.1%)		4.5 (0.8%)	
Pole face	54.7 (2.8%)		25.1 (4.7%)	
Ends	215.7 (11.1%)		40.6 (7.6%)	
Exciter	41 (2.1%)	Exciter (2.1%)	10 (1.9%)	Exciter (1.9%)
Total	1937.1		535.6	

7.4.1 Windage Loss due to Rotating Rotor

In a high-speed rotating system, windage loss is associated with pumping forces of the rotor in imparting energy to air at the rotor–stator annular gap. The magnitude of windage loss in electric motors varies widely from motor type, rotor geometry, rotor surface speed, air-gap depth, fluid properties, and rotor and stator surface finishes. In practice, windage loss due to rotating rotor is accounted for a large portion of the overall windage loss in the machine, and thus, it must be addressed carefully.

7.4.1.1 Taylor Vortex

Taylor [7.21–7.23] was the first who studied the flow between two coaxial smooth cylinders of which the inner is rotating. He found that when Taylor number reaches a certain value, a steady distinctive vortex structure is developed symmetrically grouped around the axis of rotation (Figure 7.12). Since then, considerable research on the Taylor vortex flow has been devoted to the investigations for better understanding of the flow phenomena and energy losses between concentric rotating cylinders [7.24–7.30]. Ustimenlo and Zmeikov [7.28] experimentally investigated the hydrodynamics of a flow in an annular channel with an inner rotating cylinder. Based on their work, the comprehensive data of shear stress, velocity, total and static pressure were presented and correlated. A theoretical analysis of the vortex flow and heat transfer was conducted by Leont'ev and Kirdyshkin [7.29]. The comparison of the theoretical formula with the experimental data on the friction coefficient has shown good agreement in the range of Taylor numbers between 4×10^3 and 10^7.

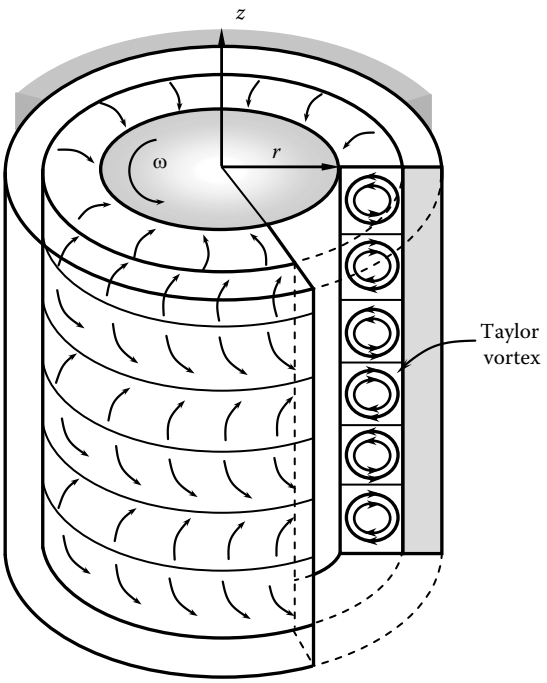


FIGURE 7.12
Flow configuration of Taylor vortices in the annular gap between two coaxial cylinders, with inner cylinder rotating and outer cylinder at rest.

As shown in Figure 7.13, radially oriented stator slots are used to mount stator windings. The stator winding in each slot is secured by a dovetail wedge, which sustains forces from the stator winding. Conventionally, the wedge is short in height, leaving the slot partially unfilled near the slot mouth with the slot opening height h_s. The stator slots distribute uniformly on the stator circumference with the slot width w_s and tooth width w_t. The air-gap depth in the radial direction is denoted by d_g. The rotor radius and stator inner radius are represented by r_r and r_s, respectively.

In fluid dynamics, Taylor number characterizes the relative importance of inertia forces to viscous force in rotating viscous fluids. As a dimensionless quantity, Taylor number is related to Couette Reynolds number Re_d as,

$$Ta = Re_d^2\left(\frac{d_g}{r_r}\right) = \frac{(\rho\omega)^2 r_r d_g^3}{\mu^2} \tag{7.51}$$

where Re_d is defined as

$$Re_d = \frac{\rho \bar{u}_\theta d_g}{\mu} \tag{7.52}$$

where

ρ and μ are the density and dynamic viscosity of air, respectively
d_g is the air-gap depth in the radial direction

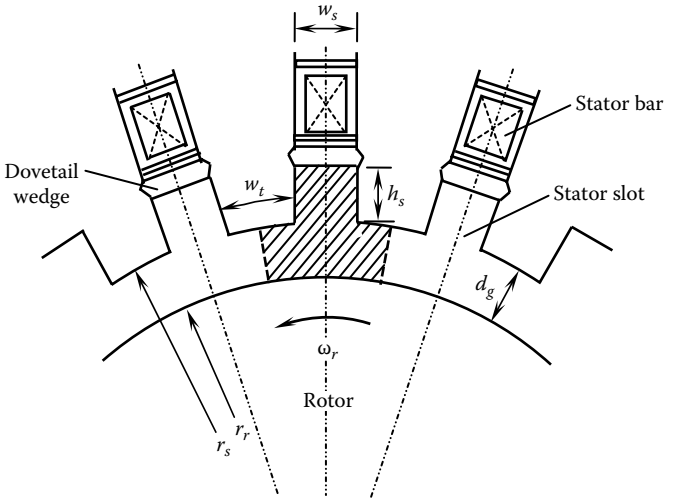


FIGURE 7.13
Geometries and configurations of air gap, state slots and teeth, dovetail wedges, and stator slot openings. The shaded area is the computational domain.

It is important to note that $\bar{u}_\theta$ is the mean swirl velocity across the stator–rotor gap, which can be obtained by integrating u_θ along the radial direction,

$$\bar{u}_\theta = \frac{1}{d_g}\int_{r_r}^{r_s} u_\theta(r)\,dr \tag{7.53}$$

The data and correlations in references [7.28] and [7.29] were combined and represented in a useful chart for friction factor for smooth rotating cylinders in a smooth coaxial enclosure without axial flow [7.30]. According to the Taylor number values, four regimes have been distinguished with a formula in each regime: laminar ($Ta \le 41$), transition ($41 < Ta \le 63$), vortex flow ($63 < Ta \le \mathrm{Ta}_{V\text{–}T}$, where $Ta_{V\text{–}T}$ is the value at which the flow transits from the vortex-dominant flow to the turbulence-dominant flow), and turbulent ($Ta_{V\text{–}T} < Ta < \infty$).

Taylor–Couette flows between concentric rotating cylinders have been extensively studied as rotational power loss and flow instability problems. Due to the complexity of the flows in rotating machinery, a majority of the studies have assumed uniform rotor–stator gaps and smooth rotor and stator walls. In practice, windage losses and friction characteristics in electric machines are generally determined through experimental tests and numerical simulations. Analytical models that precisely predict flow fields are rather limited.

Bruckner [7.31] studied experimentally windage losses in the rotor–stator gap and gas foil bearings in electric machines. The results show an exponential rise in power loss as mean operating density is increased for both the machine windage and gas foil bearings. These losses can become increasingly significant for high-speed machines operating in high-pressure environments, especially in narrow rotor–stator gaps. Bruckner also reported that the machine windage can be nondimensionalized in terms of Taylor number and moment coefficient. Rough stator surfaces have higher torque than the smooth-walled stator.

With the CFD technique, Wild et al. [7.32] have performed experimental and computational investigations of Taylor–Couette flow. The computed flow field shows significant

variations in the axial distribution of the azimuthal shear stress due to the secondary flow associated with Taylor vortices. Three functional relations, developed by other researchers for windage torque of infinite cylinders, are found to be in good agreement with the experimental results when coupled with a correction for end wall effects. These relations are the most useful with respect to the design of rotating equipment.

It is noteworthy that rotors and stators of electric motors are not so smooth. They generally have grooves, slots, cavities, etc., on their walls. For instance, the rotor of a switch reluctance motor is shaped with several poles on the rotating cylinder. This geometry can result in a more complex mechanism of windage loss. Thus, it is important to understand and predict the effect of such structural textures on windage losses and friction characteristics.

Recently, with an increase in demands for higher performance and higher efficiency of turbomachinery and other rotary systems, a great deal of effort has been directed toward the determination of friction factors on rotating and stationary surfaces.

7.4.1.2 Friction Factor

There are two types of friction factor used in the literature: Darcy–Weisbach friction factor and Fanning friction factor. Darcy–Weisbach friction factor is commonly used in the Moody diagram for pipe friction flows, and Fanning friction factor is primarily used for rotating or swirling flows [7.33]. Obviously, for electric rotating machines, Fanning friction factor is more appropriate.

Fanning friction factor is defined as the ratio of the shear stress acting on the solid surface to the mean dynamic pressure at the same location, that is,

$$f = \frac{\tau_o}{\rho \bar{u}_\theta^2 / 2} \tag{7.54}$$

The shear stress τ_o can be evaluated from the gradient of the swirl velocity profile at the wall:

$$\tau_o = \mu \left| \frac{\partial u_\theta}{\partial r} \right|_{r=r_o} \tag{7.55}$$

The friction factor in an annular gap is a function of slot geometry, stator and rotor radii, rotational speed, and fluid properties, that is,

$$f = F(w_s, w_t, h_s, d_g, r_s, r_r, \omega, \rho, \mu) \tag{7.56}$$

To reduce the number of variables, two length ratios, the slot width-to-pitch ratio η and the slot aspect ratio A_s, are introduced as

$$\eta = \frac{w_s}{w_s + w_t} \tag{7.57}$$

Hesai Exhibit 1031, page 426

$$A_s = \frac{h_s}{w_s} \tag{7.58}$$

Thus, applying Equations 7.57 and 7.58 in Equation 7.56 gives

$$f = F(\eta, A_s, Re_d) \tag{7.59}$$

7.4.1.3 Prediction of Windage Loss due to Rotating Rotor

As friction factor is determined, windage power loss P_w can be related to friction factor as

$$P_w = \pi\rho\,\bar{u}_\theta^2 r_r^2 f\omega l \tag{7.60}$$

where l is the length of the rotor.
As proposed by Saari [7.34], the windage loss of radial flux motor is associated with the resisting drag torque on the rotor:

$$P_w = k_r \pi f \rho \omega^2 r_r^4 l \tag{7.61}$$

where

k_r is the roughness coefficient (for a perfect smooth surface, $k_r = 1$; for a lamination stacked surface, $k_r = 2$–4)
f is the friction coefficient
ρ is the air density
r_r and l are the rotor radius and axial length, respectively

Equation 7.61 is essentially derived from the definition of the friction coefficient. The roughness coefficient k_r is expected to increase with the Couette Reynolds number Re_d.

Assumed that the fluid velocity varies proportionally to the distance from the air-gap midpoint with the power of 1/7, Yamada [7.35] developed an equation for the friction coefficient:

$$f = \frac{0.0152}{Re_d^{0.24}} \quad 800 < Re_d < 6 \times 10^4 \tag{7.62}$$

Based on their own measurements, as well as the experimental data from other investigators, Bilgen and Boulos [7.36] presented equations for the friction coefficient on smooth surfaces:

$$f = 0.515 \frac{(d_g / r_r)^{0.3}}{Re_d^{0.5}} \quad 500 < Re_d < 10^4 \tag{7.63a}$$

$$f = 0.0325 \frac{(d_g / r_r)^{0.3}}{Re_d^{0.2}} \quad 10^4 < Re_d \tag{7.63b}$$

The experimental friction coefficients are found to deviate less than 8.35% from the results calculated by above equations.

For the turbulent flow that $Ta >> 400$, the friction coefficient f is [7.37]

$$f = \frac{0.0095}{(Ta)^{0.2}} \tag{7.64}$$

The windage loss of axial flux motor is given by [7.38]

$$P_w = \frac{\pi\mu\omega^2(r_o^4 - r_i^4)}{2l_g} \tag{7.65}$$

where
- μ is the air viscosity
- ω is the rotor rotating speed
- l_g is the axial length of the air gap
- r_o and r_i are the shaft outer and inner radii, respectively

7.4.2 Windage Loss due to Entrance Effect of Axial Air-Gap Flow

Assuming that the air cooling flow does not have a tangential velocity component before entering the air gap, Polkowski [7.39] underlined that a rather large friction torque may be associated with the entrance effects of the axial air-gap flow. He suggested that the torque needed to accelerate the axial cooling flow into a tangential movement is presented as

$$T_a = \frac{2}{3}\pi\rho(r_s^3 - r_r^3)v_m u_m \tag{7.66}$$

where
- r_s and r_r are the radii of the stator and rotor, respectively
- v_m and u_m are the mean axial and tangential air velocities, respectively

Thus, windage losses may be associated with the cooling flow through the air gap of a high-speed motor:

$$P_a = \frac{2}{3}\pi\rho(r_s^3 - r_r^3)v_m u_m \omega \tag{7.67}$$

According to Saari [7.34], the power losses predicted by the above equations agree well with the measured results. The surface roughness caused by the stator slot openings does not significantly increase the friction losses in the air gap.

7.4.3 Windage Loss due to Stator Surface Roughness

The opposite observation to the Saari's conclusion [7.34] that the stator roughness has little effect on the rotor body windage loss was presented by Tong and Gott [7.40]. They investigated the influence of stator slot openings on the rotor windage loss in large-sized electric machines. Their results suggest that the roughness of the stator surface could significantly alter the velocity field in the air gap and thereby change windage losses.

Referring to Figure 7.10, when the rotor rotates with angular velocity ω_r, the rotor rotation-driven flow may penetrate into these unfilled slots so that the flow field in the stator–rotor air gap is distorted and leads to the variations in the friction factor on the rotor surface and correspondingly the rotor body windage loss.

Friction factor is defined as the ratio of the shear stress acting on the surface to the dynamic pressure applied on the same surface. For a rotating electric machine, friction factor is associated with the pumping action of a rotor in imparting energy to the cooling medium at the rotor–stator annular gap. An important characteristic of rotor rotation-driven flows is the tendency of fluid with high angular momentum to be flung radially outward.

With the defined unfilled aspect ratio A_s ($A_s = h_s/w_s$) and friction factor ratio f_r/f_{ro}, where f_r is the friction factor for the rotor and f_{ro} is the friction factor for the smooth stator wall (i.e., $h_s = 0$), the effect of the aspect ratio A_s on the friction factor ratio can be obtained by numerical simulations using a commercial CFD code FLUENT. As depicted in Figure 7.14, the friction factor ratio is zero as the wedge is flush with the adjacent stator surfaces. The friction factor ratio exhibits a rapid rise by increasing in A_s and then declines sharply with the further increase of A_s. When A_s approaches approximately 0.8, it reaches its local minimum. Thus, the preferred way to reduce windage loss is to redesign the wedge with either $A_s = 0$ or $A_s = 0.8$ for this specific machine. It has been found that the selection of correct A_s values can reduce rotor body windage loss by about 25% (Figure 7.15).

This may be similar to the golf ball design. As a simple, passive means of drag reduction, small dimples are made on the golf ball surface. In such a way, the dimpled surface manipulates the flow and creates vortices that can prevent or delay the boundary-layer separation on the ball surface [7.41].

When both the stator and rotor surfaces are smooth, the friction factors on the rotor and stator have shown a relative difference of about 12%, mainly resulted from the curvature effect and rotor pumping effect (Figure 7.16).

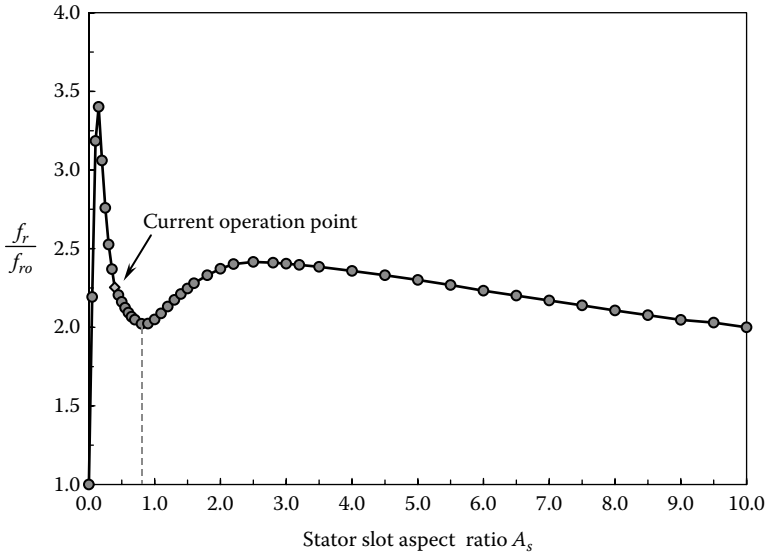


FIGURE 7.14
Effect of stator slot aspect ratio on rotor friction factor ratio.

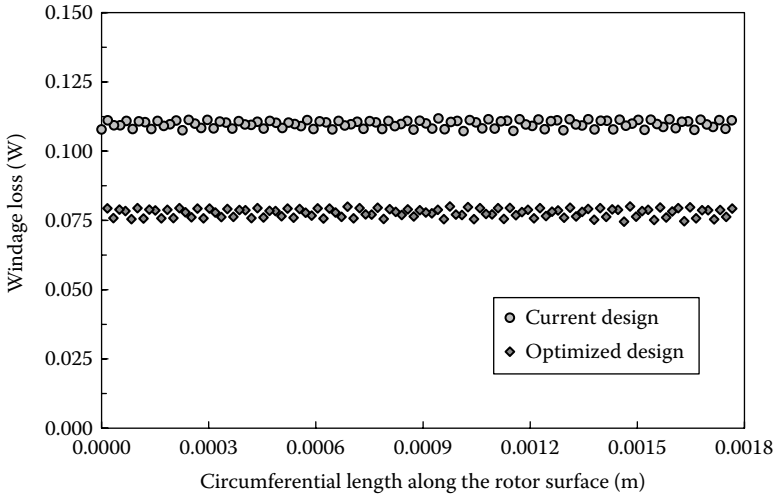


FIGURE 7.15
Comparison of rotor windage loss data between the current and optimized design.

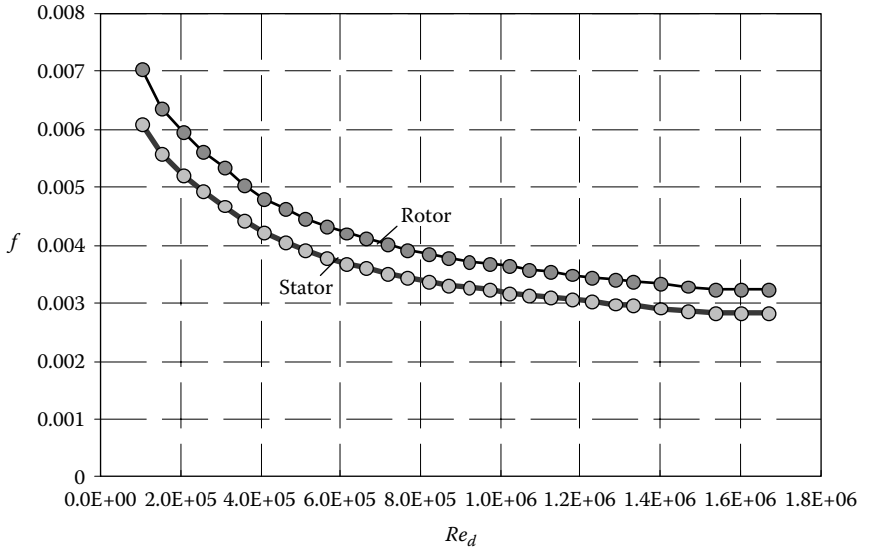
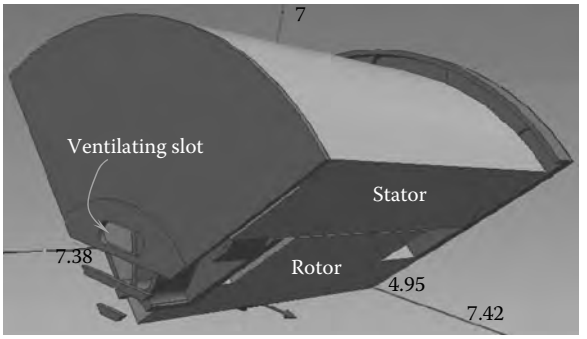


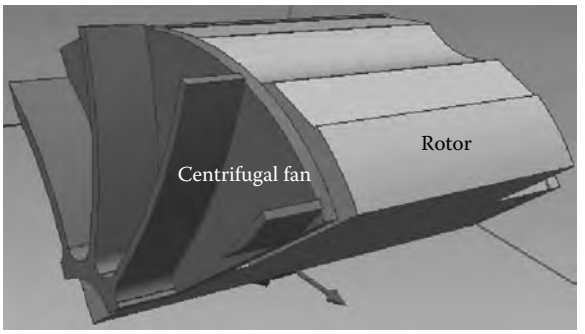
FIGURE 7.16
Comparison of friction factor on the smooth rotor and stator surfaces.

7.4.4 Fan Losses

Fans are devices that utilize electric energy to drive fan blades rotating to achieve both the air movement and the increased air total pressure. Motor cooling fans are often attached to rotor shafts at one or two ends to cool motors. As shown in Figure 7.17, a centrifugal cooling fan is mounted on one end of a rotor. When the rotor rotates, it drives the fan rotating to suck the cooling air into the machine through the ventilating slots on the endbell. Due to the centrifugal effect, the cooling air flows radially to cool the stator



(a)



(b)

FIGURE 7.17
A centrifugal fan is mounted on one end of the rotor to suck the cooling flow into the motor: (a) the motor assembly; and (b) the centrifugal fan and rotor assembly, where the rotor with scallop-shaped structure for providing ventilation paths of axial cooling flow.

winding and then turns 180° to flow toward the rotor. Then, the cooling air turns 90° to get into the scallop-shaped cooling channels in the axial direction and finally exhausts at the other end of the machine (Figure 7.18). The exhausted air is guided by the endbell at the other end to flow over the outer surface for cooling the motor housing.

The pressure field in the machine has shown that the high static pressure zone occurs at the vicinity of the fan (Figure 7.19). A large static pressure differential is observed across the fan. In practice, the cooling fan represents a significant load on the machine, reducing available power. Generally, the fan power loss increases with the increase in the rotating speed of the fan. In addition, the fan power loss is a function of the duty cycle of the fan. As a duty cycle is larger than 50%, the fan power loss is about a constant regardless of its rotating speeds. As the duty cycle is less than 40% (i.e., the *on time* is less than 40%), the fan power loss reduces sharply in the reduction of the duty cycle [7.42].

The loss in pressure between any two points in a system is always equal to the difference between the total pressures at these points. Therefore, the total pressure loss of fan ΔP_t is the difference between the total pressure at the fan inlet $P_{t,in}$ and outlet $P_{t,out}$:

$$\Delta P_t = P_{t,in} - P_{t,out} \tag{7.68}$$

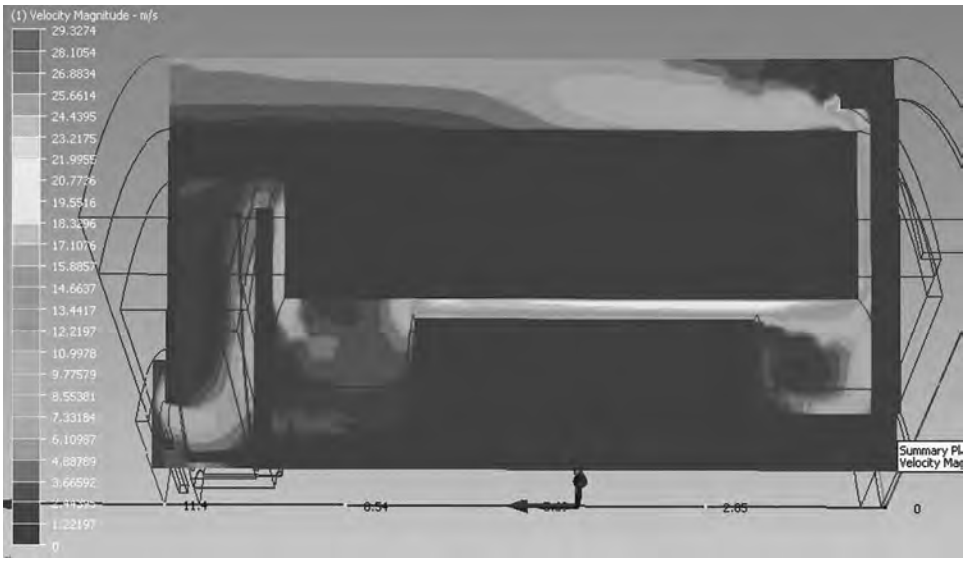


FIGURE 7.18
Velocity field through the internal cooling paths.

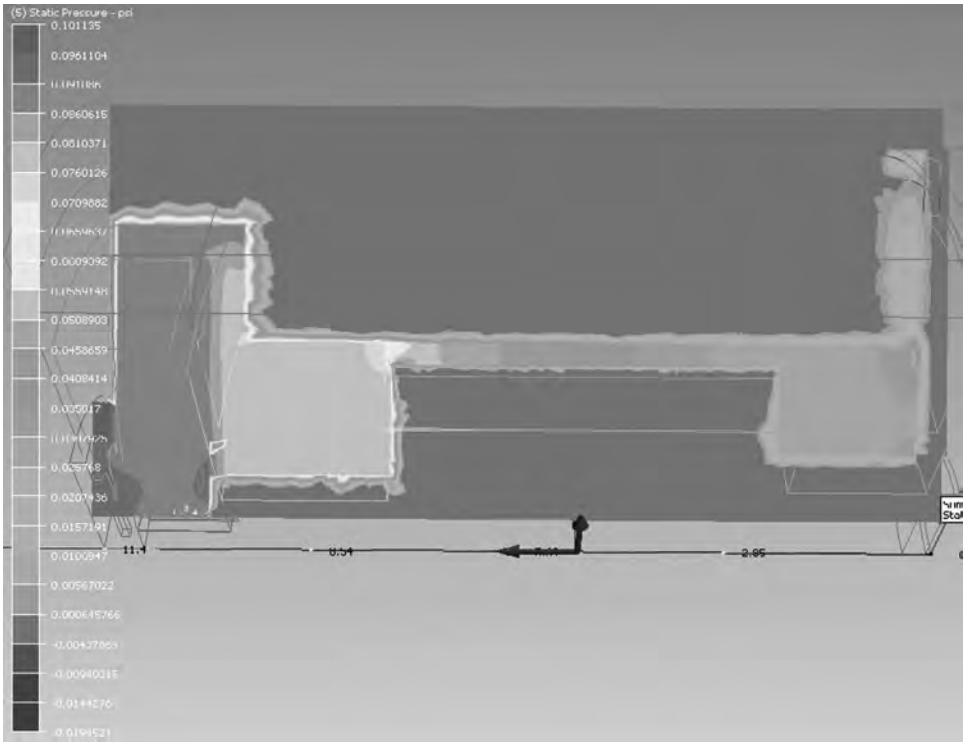


FIGURE 7.19
Pressure distribution inside the machine. The high-pressure zone is shown at the vicinity of the fan.

The total pressure at any point consists of two components: static pressure P_s and dynamic pressure P_d. Consequently,

$$P_{t,in} = P_{s,in} + P_{d,in} = P_{s,in} + \frac{1}{2}\rho V_{in}^2 \tag{7.69a}$$

$$P_{t,out} = P_{s,out} + P_{d,out} = P_{s,out} + \frac{1}{2}\rho V_{out}^2 \tag{7.69b}$$

where
$P_{s,in}$ and $P_{s,out}$ are the static pressures
$P_{d,in}$ and $P_{d,out}$ are the dynamic pressures at the fan inlet and outlet
V_{in} and V_{out} are the velocities at the fan inlet and outlet, respectively

Substituting these two equations into (7.68) yields

$$\Delta P_t = (P_{s,in} - P_{s,out}) + \frac{1}{2}\rho(V_{in}^2 - V_{out}^2) = \Delta P_s + \Delta P_d \tag{7.70}$$

All these pressures are defined in ANSI/AMCA Standard 210-07 [7.43]. The relations of these pressures and pressure differentials at the inlet and outlet of a fan are presented in Figure 7.20. With a ducted inlet and outlet, the air flow velocity along the ducted inlet is constant, and thus, the dynamic pressure remains unchanged. The static pressure decreases (relative to the pressure datum) as the air flow approaches the fan. Hence, the total pressure reaches its minimum (relative to the pressure datum) at the point of the fan inlet. Similarly, along the ducted outlet, both the air flow velocity and dynamic pressure remain constants. The static pressure decreases when the air flow leaves away from the

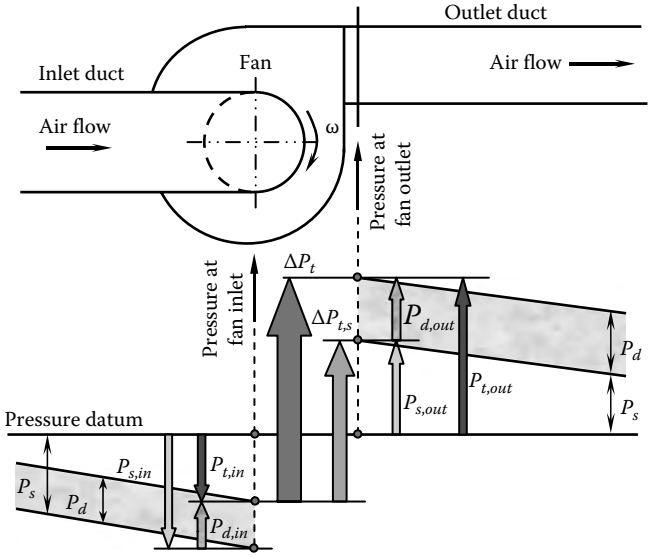


FIGURE 7.20
Relations of various pressures and pressure differentials at the inlet and outlet of a fan.

Hesai Exhibit 1031, page 433

fan, causing the drop in the total pressure along the ducted outlet. It is worth to note that the total pressure of the air flow experiences a big jump when air flows from the fan inlet to the outlet due to gained mechanical energy from the fan. If the total pressure is plotted against the flow path, it can be seen that the total pressure decreases everywhere except across the fan.

Fan losses may come from (1) head loss due to the changes in flow area and direction; (2) energy lost in some components such as filters, sound attenuators, and grills; (3) friction of the duct and fan walls; (4) air flow recirculation and reentrance; and (5) windage loss of the fan itself.

7.4.5 Ventilating Path Losses

Ventilating paths inside an electric rotating machine is considerably complex due to irregular shapes, rough surfaces, and internal obstructions. Pressure losses occur at sharp turns, sudden contractions and expansions, and any torturous paths in the end-winding regions and in the gap between the rotor and stator, resulting in large ventilation windage losses [7.44].

Because ventilating paths are so complicated and vary case to case, no theoretical solutions or empirical equations are available. In order to predict the windage loss resulted from ventilating paths, it is recommended to use CFD tools.

7.4.6 Methods for Reducing Windage Losses

The increase in electric motor efficiency is extremely important for applications in modern society because any power losses ultimately imply the consumption of energy. As motor power densities decrease and output power levels increase to meet high demand of high-performance, high-speed, and cost-effective motors, power loss due to windage in some motion control systems can become significantly large. The excessive windage loss can effectively lower the machine capacity and performance. In some cases, the derating due to the total windage loss may consume over half of the available shaft power in the machine.

One effective way to reduce windage loss in a motor/generator, which is flooded with fluid, is to apply a so-called film divider [7.45]. The film divider is positioned within the air gap and freely rotatable within the gap. As the embodiment shown in Figure 7.21, the film divider is essentially a thin sleeve, which may be made of any magnetically permeable material such as fiberglass or epoxy composite. It is concentric to both the rotor and stator. During machine operation, the rotor rotates with an angular velocity of ω. As the rotor spins, the fluid in the air gap causes the film divider to spin. Ideally, the angular velocity of the film divider is $\omega/2$. Thus, the differential vectors in each half of the gap are half of the vector flow without the film divider. Because windage loss is proportional to the 2.7 power of the velocity in the turbulent flow regime (i.e., $P_w \propto \omega^{2.7}$), the overall windage loss is lower with the film divider in place.

The comparison of windage loss in high-speed motors with and without the film divider is presented in Figure 7.22. It can be seen that the motor with the film divider has much lower windage loss than that without the film divider.

In a high-power, high-speed electric machine, windage losses are created by a large rotor body. The peripheral velocity on the rotor's surface causes significant windage losses. In order to reduce windage losses, it was proposed to optimize the rotor construction [7.46]. As shown in Figure 7.23, the rotor has a central cylindrical body provided with

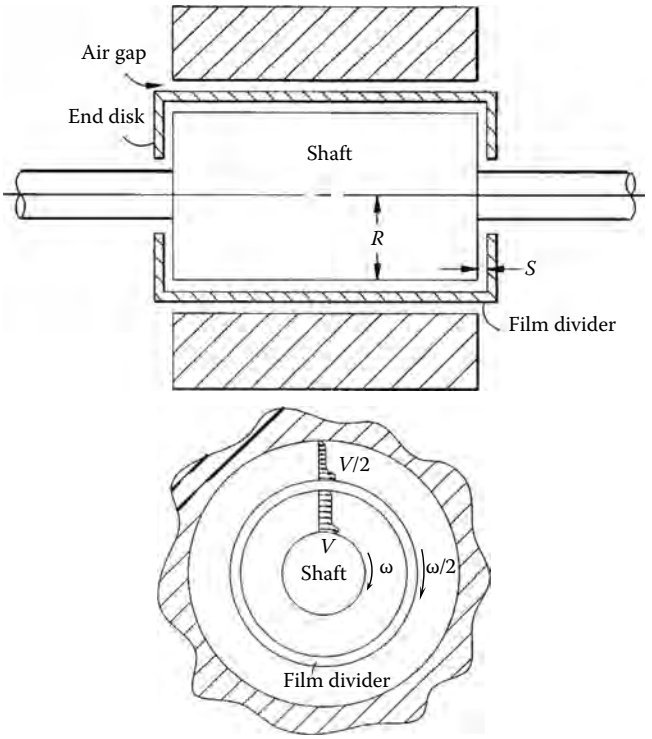


FIGURE 7.21
Using a film divider to lower windage loss in the air gap (U.S. Patent 5,828,148) [7.45]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

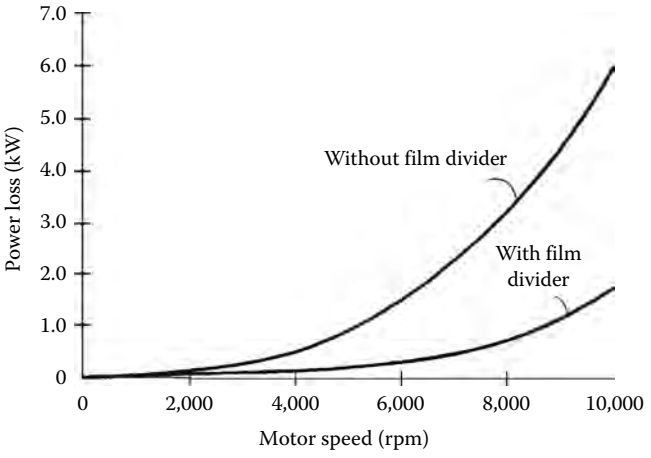


FIGURE 7.22
Comparison of windage power loss of motors with and without film divider in high-speed machine (U.S. Patent 5,828,148) [7.45]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

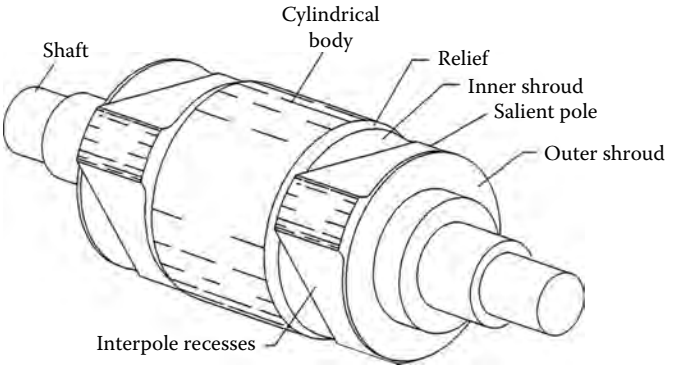


FIGURE 7.23
Optimized rotor construction for reducing windage loss (EP0489859) [7.46]. (Courtesy of European Patent Office, Munich, Germany.)

circumferentially spaced, axially extending surface recesses at the ends of the rotor, which define a desired number of salient poles at each end of the rotor. The inner and outer shrouds are provided on either side of salient poles to reduce windage losses. The outer shrouds block axial air flow into the recesses and thus reduce windage losses. Furthermore, with the reduced diameter of the central cylindrical body, the peripheral velocity of the cylindrical body is reduced, and, thereby, the windage loss is also reduced.

To further minimize windage losses, an outer cylindrical surface can be produced in the pole region of the rotor by filling interpole recesses with nonmagnetic material.

A similar idea is to use a rotor containment shell surrounding the rotor to lower windage losses [7.47]. The outer cylindrical surface of the shell is formed with a plurality of concavities in the form of annular turbulator grooves axially spaced from one another along the rotor length (Figure 7.24). These grooves will enhance the local air mixing, improve the axial heat transfer, reduce windage and friction losses, and reduce the heat penetration from the heat generated by windage into the rotor body. The optimal depth of the groove depends on a trade-off between increased resistance to tangential motion and decreased resistance to axial flow.

Smith et al. have proposed several methods for reducing power losses in motor [7.48]. One of them is to maintain constant pressure throughout the motor cavity. A pressure valve can be placed within the motor cavity to release higher-pressure air built up during operation. The maintenance of constant pressure in the cavity increases the motor efficiency due to the reduction of windage loss.

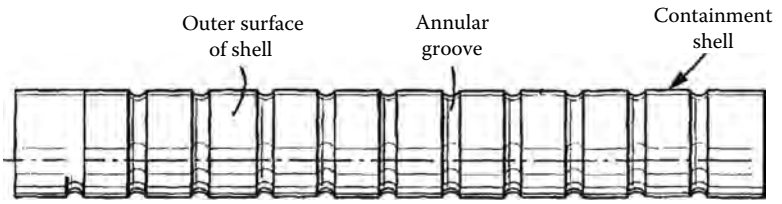


FIGURE 7.24
Rotor containment shell with a plurality of surface concavities (U.S. Patent 7,057,326) [7.47]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 436

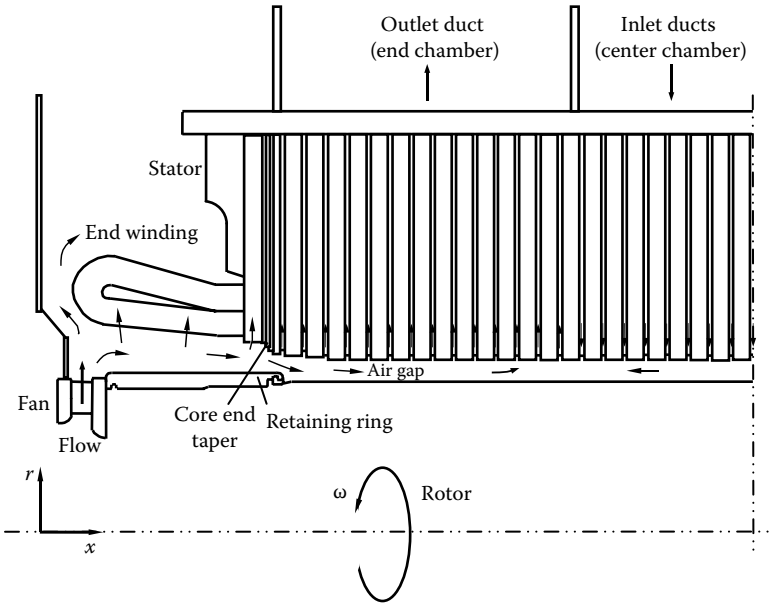


FIGURE 7.25
Schematic diagram of a ventilating cooling system in a large-size, high-power electric machine. Air flow is distributed to cool the end windings and the stator core.

One obvious way for windage loss reduction is to smooth cooling flow paths inside the motor [7.49]. All sharp edges, rapid expansions and contractions, and obstacles in flow paths should be avoided. A cooling air ventilation circuit in a large-size, high-power electric machine is presented in Figure 7.25. The inlet flow is accelerated in the radial direction by a ventilating fan, which is mounted on the end of the rotor. About one-third of the total volumetric air flowrate enters into the air gap. The remainder proceeds through and around the armature end winding. This flow allocation is thus designed and controlled to ensure sufficient air flow for the heat removal. It can be seen that at the air-gap entrance, a *bottle neck* is formed between the retaining ring inboard end and the end-core taper on the stator. As cooling gas passes through the gap entrance, the significant flow acceleration causes a large local pressure drop that is only partially recovered downstream.

By optimizing the retaining ring with the preferable spline profile and other components, the flow recirculation is eliminated, and thus, local windage loss is remarkably reduced (Figure 7.26). In addition, the enlarged air-gap entrance allows more flows to get into the air gap.

More recently, Pal [7.50] has proposed a new idea for reducing windage loss in liquid-cooled electric machines. This method uses injecting oil flow to cool the machine windings. The machine has two rotors and two stators. A number of baffles are coupled with the shaft at the center and each end of the shaft. The baffles extend radially outward from the shaft toward baffle cavities at the back iron. As the shaft rotates, oil is urged toward the baffle cavities along the baffles via centrifugal forces (Figure 7.27). The baffle directing oil to the baffle cavity substantially improves the scavenge performance. As a result, it reduces the oil level in a sump and lowers windage and friction losses.

Hesai Exhibit 1031, page 437

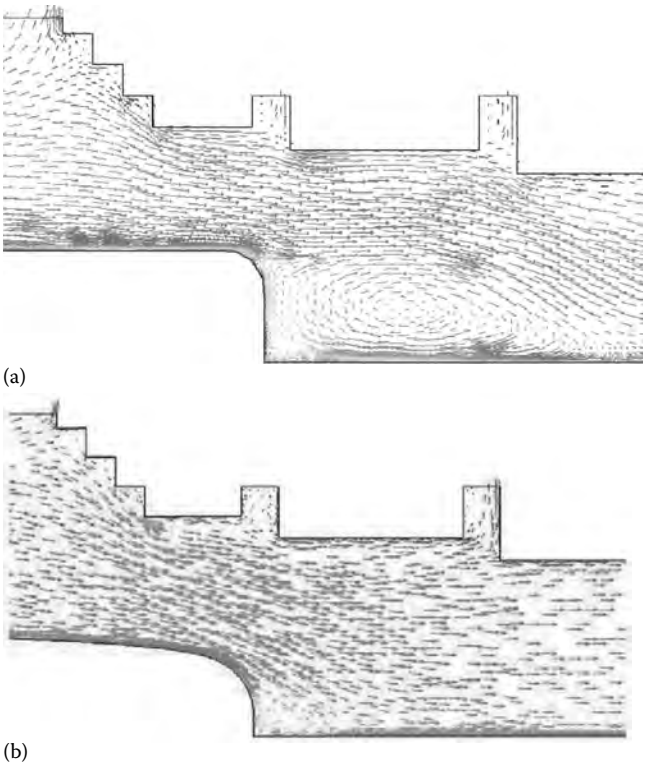


FIGURE 7.26
Reducing windage loss at the air-gap entrance by smoothing the cooling flow path: (a) original design with a large flow recirculation and (b) improved design with optimized flow path to eliminate the flow recirculation and thus reduce windage loss.

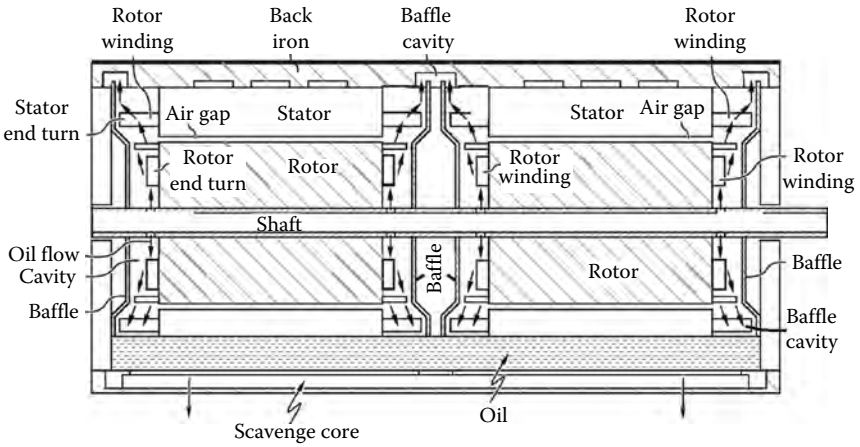


FIGURE 7.27
A liquid-cooled electric machine is cooled by injecting oil radially along a plurality of baffles via centrifugal force toward rotor and stator end windings [7.50]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 438

An effective method to reduce windage losses is to use hydrogen gas to cool electric machines. The high thermal conductivity of hydrogen has proven to be a key advantage in its use as a cooling medium in electric power machines. Hydrogen has a thermal conductivity of nearly seven times higher than that of air, and its ability to transfer heat through forced convection is about 50% better than air [7.51]. At a given temperature and pressure, the density of hydrogen is only 1/14 the density of air. Due to its superior thermophysical properties, the use of hydrogen can not only reduce windage friction losses but also greatly enhance heat transfer superior properties.

Using hydrogen as coolant can be traced back to the late 1930s. Since then, this technique has been developed and successfully used in generator cooling but still not applied to electric motors.

7.5 Stray Load Losses

Stray load losses consist of a variety of minor losses not accounted previously, for instance, the losses referring to the sources such as stray flux, the cage in the slots, interbar currents, and harmonic rotor bar currents from the nonsinusoidal distribution of the slotted stator winding [7.52]. Stray load losses appear only when motors operate under load conditions and are very difficult to predict analytically and to measure accurately. The largest contribution to stray losses is harmonic energies. These energies are dissipated as currents in the copper windings, harmonic flux components in the iron parts, and leakage fluxes in the laminate core. The detailed discussions and analyses of stray losses have been provided by Aoulkadi [7.53] and Englebretson [7.54].

It is commonly accepted that the investigation of stray load losses in electric rotating machines is far from complete. Most motor engineers rely on empirical equations for the estimation of stray load losses.

Stray load losses represent a small portion of the total motor power loss. The ratio of stray load loss to motor rated power decreases with the increased motor rated power. IEEE Standard 112-2004 [7.55], GB 18613-2002 [7.56], and IEC Standard 60034-2-1 [7.57] provide assumed values of stray load losses under different power ratings. Some of them are shown in Tables 7.4 and 7.5.

TABLE 7.4

Assumed Values of Stray Losses

Machine Rating (kW)	Stray Load Loss (% of Rated Power)
1–90	1.8
91–375	1.5
376–1850	1.2
>1850	0.9

Source: IEEE Power Engineering Society, IEEE 112-2004 Standard test procedure for polyphase induction motors and generators, 2004.

TABLE 7.5

Values of Stray Losses Proposed by GB 18613-2002

Machine Rating (kW)	Stray Load Loss (% of Rated Power)
0.55–1.1	2.5
1.5	2.4
3	2.3
4	2.2
5.5–7.5	2.1
11	2.0
15–18.5	1.9
22–30	1.8
37–45	1.7
55–75	1.6
90–110	1.5
132–160	1.4
200–315	1.3

Source: Standardization Administration of China (SAC), GB 18613-2002, Limited values of energy efficiency and evaluating values of energy conservation of small and medium three-phase asynchronous motors, 2002.

For other than rated load, stray losses are assumed to be proportional to the square of the rotor current [7.55].

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Hesai Exhibit 1031, page 441

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Hesai Exhibit 1031, page 442

8

Motor Cooling

The power output rating of an electric motor is often limited by the ability to provide additional electric current through stator and rotor windings. However, the increase of current in the windings is constrained by temperature limitations imposed on the electric conductor insulation and PMs (for a PM motor). According to the Arrhenius equation [8.1,8.2], the failure rate of an electric/electronic device is exponentially related to the reciprocal of the operating temperature. A fairly accurate approximation of the Arrhenius equation states that an operating temperature exceeds the thermal limit by 10°C and reduces the lifetime of the device by half. Therefore, effective cooling of stator and rotor windings contributes directly to the output capability, operation reliability, and lifetime of electric motors. This is especially true in motor end-winding regions where the configuration of cooling flow paths is extremely complicated and direct forced cooling is very difficult [8.3,8.4]. In addition, the relatively poor heat dissipation from the rotor can cause the increase in rotor core losses, leading to an excessive temperature rise in the rotor and magnets. This may result in partially irreversible demagnetization of the magnets, particularly of sintered Nd–Fe–B magnets [8.5] and thus a decline in the performance of the motor.

In recent years, the development and production of high-performance, high-efficiency, and high-power-density electric motors have been accompanied by increasing heat fluxes at both the component and system levels. Over the years, significant advances have been made in the applications of air cooling and liquid cooling techniques to manage increased heat fluxes. As the prevailing market demands electric motors with better performance, lower power consumption, and smaller size, thermal management has emerged as an increasingly important aspect in motor design.

8.1 Introduction

Modern electric motors generate a large amount of heat during their normal operation as the results of various electric, electromagnetic, and mechanical losses. The great majority of motor losses are converted into heat, causing the temperature of windings, cores, magnets, etc., to rise. Excessive temperatures are generally detrimental to motor performance and reliable operation. Hot spots in a motor due to nonuniform heat fluxes and improper cooling may result in damage to winding insulation or premature bearing failure and, in turn, greatly reduce the motor lifetime. For some special applications, thermal management is critical to motor performance. For instance, a telescope tracking motor requires an extremely tight control in the temperature rise relative to the ambient temperature because any sharp and excessive temperature changes may significantly influence the telescope's tracking accuracy. Therefore, to ensure safe and reliable operation of electric motors, all heat generated in the motor must be efficiently dissipated to the ambient air.

Hesai Exhibit 1031, page 444

A variety of different cooling mechanisms may be used to cool electric motors to maintain the temperatures of stator and rotor windings, as well as PMs and bearings, below the maximum allowable limit. Today, cooling of an electric motor still remains a major challenge for motor engineers and designers because of a number of factors: (1) a great number of motor parts with irregular geometries, (2) the nonuniform heat generation in different heat sources (stator and rotor windings, cores, bearings, etc.) due to various power losses, (3) the complexity in cooling flow paths, (4) cooling flow splitting and mixing, (5) interactions between the rotor-induced rotating flows and the nonrotating flows at the stationary components, (6) a variety of thermal exchange modes (conduction, convection, radiation, etc.) that are involved simultaneously, and (7) the compact and lightweight motor design, as a result of growing global demand for high-power-density motors. As a consequence, the thermal phenomena inside an electric motor are considerably complex.

Cooling of high-heat-flux heat sources can be generally divided into two steps [8.6]: (1) heat extraction, in which heat is removed from a high thermal density region (e.g., motor windings) and reformatted to a larger area, and (2) heat rejection, in which the heat is dissipated convectively to the surrounding environment.

Remarkable progresses in heat extraction and heat rejection have been made in the last several decades, including adoptions of advanced cooling techniques such as heat pipe, microchannel, heat pump, high-conductivity material, thermoelectric cooler, carbon nanotube, and recently proposed ionic wind generator in electric/electronic cooling. However, in cooling of electric motors, technologies have not yet made significant progress in past several decades. As a result of the technology stagnation, the design and development of high efficient cooling systems still remain challenges today to motor engineers and manufacturers.

8.1.1 Passive and Active Cooling Techniques

Cooling techniques are often divided into two categories: passive cooling and active cooling, depending on what the heat transfer mode would be or how cooling flows are generated. Passive cooling refers to cooling with no active mechanism such as fan or pump. Passive cooling techniques have been developed for a long time for motors that have low power losses. In a passive cooling mode, motor cooling primarily relies on heat conduction to transport the generated heat from the heat sources (e.g., stator winding) to the motor frame (e.g., motor housing and endbells) and then to dissipate the heat to the environment by natural convection and radiation. By eliminating fans or pumps, motor operation reliability is significantly increased. Compared with active cooling techniques, passive cooling techniques have lower costs, lower noise emission, and no power consumption. However, passive cooling techniques are restricted to low thermal load conditions. For large-size, heavy-duty motors, active cooling is the only choice.

In contrast, active cooling techniques require the use of external devices such as fans, blowers, and pumps to force a coolant (air or liquid) flowing through cooling channels either inside or outside, or both, of the motor. With the development of modern electric motors, which have high power density and consequently high power losses, active cooling is required to remove the waste heat produced by motor components to keep these components within permissible operating temperature limits.

The magnitude of the heat transfer coefficient is determined by a number of parameters such as heat transfer mode, geometry, flow path dimensions, coolant type, flowrate, and flow condition (laminar or turbulent). The comparison of heat transfer coefficients for different heat transfer modes is presented in Table 8.1.

Hesai Exhibit 1031, page 445

TABLE 8.1

Heat Transfer Coefficients of Different Heat Transfer Modes with Different Coolants

Heat Transfer Mode	Heat Transfer Coefficient (W/m²-K)	References
Conduction	5–25	
Natural convection (air)	5–30	[8.7,8.8]
Natural convection (FC)	10–100	[8.7,8.8]
Natural convection (water)	30–300	[8.7,8.8]
Forced convection (air)	20–200	[8.7,8.8]
Forced convection (FC)	50–300	[8.7,8.8]
Forced convection (water)	300–8,000	[8.7,8.8]
Forced jet (air)	200–800	[8.7,8.8]
Forced jet (FC)	2,000–20,000	[8.7,8.8]
Forced jet (water)	8,000–50,000	[8.7,8.8]
Pool boiling (water)	2,500–3,500	[8.9]
Forced boiling (water)	5,000–100,000	[8.9]
Condensing (water)—vertical surface	4,000–11,300	[8.9]
Condensing (water)—Outside of horizontal tube	9,500–25,000	[8.9]
Radiation	5–30	

Note: Conductive heat transfer coefficient and radiative heat transfer coefficient are defined as $h_c = \frac{-\nabla(kT)}{(T_h - T_c)}$ and $h_r = \sigma\varepsilon\frac{T_h^4 - T_c^4}{T_h - T_c}$, respectively. They are estimated as the same order as natural convection.

The selection of appropriate and feasible cooling techniques depends basically on the required cooling load, cooling efficiency, cooling system operation reliability, cost of cooling system, temperature difference between the heat source and the environment, coolant thermophysical properties, geometries of heated sources, noise, and vibration. All these determining factors must be taken into account. The trade-off between these factors is highly desired in order to choose the best cooling technique for each particular application. Three liquids, namely, air, fluorocarbon (FC), and water, are chosen to compare their heat transfer coefficients for natural and forced convection and forced jet.

The results show that all cooling modes that involve a phase change can make very high heat transfer coefficients. Water is always the best coolant. However, for motor cooling applications, water has to be restricted in indirect cooling application. In all cases, air cooling has the lowest heat transfer rate, compared with FC and water.

8.1.2 Heat Transfer Enhancement Techniques

A race for increasing power density of electric motors has provided the driving force behind the scientific research and technological advancement in the field of heat transfer enhancement. It is now widely accepted that the thermal management in high-power-density motors becomes critically important. Efficient cooling plays an increasingly key role that controls the performance and stability of electric motors.

It has been recognized that the primary limitation to performance of a convective cooling system is the boundary layer of motionless air that adheres to and envelops all surfaces of the cooling system. Within this boundary layer region, diffusive transport is the dominant mechanism for heat transfer. The resulting thermal bottleneck largely determines the thermal resistance and convective heat transfer coefficient and thus cooling efficiency of the cooling system. The heat transfer enhancement can be achieved by disturbing cooling flows to lower the thickness of the boundary layer and increasing the convective contact area.

A variety of heat transfer enhancement techniques have been developed for adapting to the rapid increase in thermal load in electric and electronic devices [8.10]. For passive techniques, heat transfer enhancements are achieved by employing special surface geometries or fluid additives. Common surface geometries include coated surfaces, rough surfaces, and extended surfaces to increase the film heat transfer coefficient and to reduce the thermal resistance. Additives for liquid include solid particles or gas bubbles in single-phase flows and liquid trace additives for boiling systems.

Active techniques employ turbulators such as displaced inserts, pin fins, dimpled surfaces, surfaces with arrays of protrusions, and swirl chambers. All of these turbulators act to increase rotating/secondary flows or vortices on heat surfaces for augmenting forced convective heat transfer. These rotating/secondary flows and vortices not only increase rotating/secondary advection of heat away from surfaces but also increase 3D turbulence production by increasing shear and creating gradients of velocity over significant flow volumes.

The selection and application of heat transfer enhancement techniques are core concerns for modern motor engineers and designers. Many researchers have contributed their efforts toward better understanding of heat transfer techniques.

8.2 Conductive Heat Transfer Techniques

Heat conduction is a mechanism of heat transfer, taking place if there is a temperature gradient in a solid body or between contacted solid bodies. From the microscopic standpoint, conduction is relevant to the movement of particles (e.g., atoms or molecules) in a material. The particles vibrate randomly about fixed positions. Since the rise in temperature will result in the increase in the particle kinetic energy, the higher the temperature, the greater the particle vibration. When particles collide with their neighbors, part of the kinetic energy will be transferred from the particles that have higher energy to the particles that have lower energy until thermal equilibrium is reached. At the state of thermal equilibrium, there are an equal number of collisions resulting in an energy gain as there are collisions resulting in an energy loss. On average, there is no net energy transfer resulting from the collisions of particles.

8.2.1 Conductive Heat Flux and Energy Equations

The conductive heat flux vector $\mathbf{q}''$ is proportional to the thermal conductivity k_n and the temperature gradient:

$$\mathbf{q}'' = -\nabla(k_n T) \tag{8.1}$$

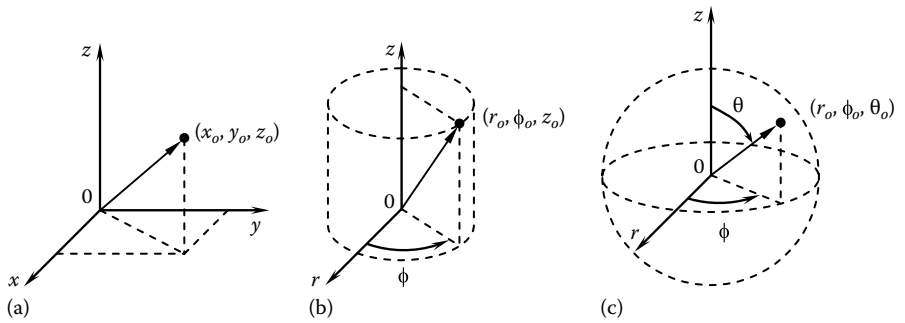


FIGURE 8.1
Coordinate systems: (a) Cartesian, (b) cylindrical, and (c) spherical.

where T is the temperature. Generally, thermal conductivity is a function of temperature. In anisotropic materials such as insulations and silicon steel laminations, thermal conductivity k_n may have different values in different directions of the material spatial orientation at a specific temperature T, that is,

$$k_n = k_n(n_1, n_2, n_3, T) \tag{8.2}$$

The negative sign in Equation 8.1 denotes that the heat flow is always from the region of high temperature to the region of low temperature. Equation 8.1 in 3D Cartesian, cylindrical, and spherical coordinate systems (see Figure 8.1) can be expressed as

$$q'' = -\left[\frac{\partial(k_x T)}{\partial x} + \frac{\partial(k_y T)}{\partial y} + \frac{\partial(k_z T)}{\partial z}\right] \tag{8.3a}$$

$$q'' = -\left[\frac{\partial(k_r T)}{\partial r} + \frac{1}{r}\frac{\partial(k_\phi T)}{\partial \phi} + \frac{\partial(k_z T)}{\partial z}\right] \tag{8.3b}$$

$$q'' = -\left[\frac{\partial(k_r T)}{\partial r} + \frac{1}{r\sin\theta}\frac{\partial(k_\phi T)}{\partial \phi} + \frac{1}{r}\frac{\partial(k_\theta T)}{\partial \theta}\right] \tag{8.3c}$$

respectively.

Energy equation governing conductive thermal energy transport in solids is based on energy conservation in a differential control volume through which energy transfer is exclusively by conduction and can be generally expressed as

$$\rho c_p \frac{\partial T}{\partial t} = \nabla^2(k_n T) + q''' \tag{8.4}$$

where

ρ is the density
c_p is the specific heat
q''' is the heat generation in the control volume

Similarly, Equation 8.4 can be expressed in Cartesian, cylindrical, and spherical coordinate systems as

$$\rho c_p \frac{\partial T}{\partial t} = \frac{\partial^2 (k_x T)}{\partial x^2} + \frac{\partial^2 (k_y T)}{\partial y^2} + \frac{\partial^2 (k_z T)}{\partial z^2} + q''' \tag{8.5a}$$

$$\rho c_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left[r \frac{\partial (k_r T)}{\partial r} \right] + \frac{1}{r^2}\frac{\partial^2 (k_\phi T)}{\partial \phi^2} + \frac{\partial^2 (k_z T)}{\partial z^2} + q''' \tag{8.5b}$$

$$\rho c_p \frac{\partial T}{\partial t} = \frac{1}{r^2}\frac{\partial}{\partial r}\left[r^2 \frac{\partial (k_r T)}{\partial r} \right] + \frac{1}{r^2 \sin^2\theta}\frac{\partial^2 (k_\phi T)}{\partial^2 \phi} + \frac{1}{r^2 \sin\theta}\frac{\partial}{\partial \theta}\left[\sin\theta \frac{\partial (k_\theta T)}{\partial \theta} \right] + q''' \tag{8.5c}$$

For isotropic materials, thermal conductivity is identical in all directions (i.e., $k_x = k_y = k_z = k$, $k_r = k_\phi = k_z = k$, $k_r = k_\phi = k_\theta = k$). Taking this condition and assuming k is a constant, the aforementioned equations can be greatly simplified:

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + \frac{q'''}{k} \tag{8.6a}$$

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \left[\frac{1}{r}\frac{\partial}{\partial r}\left(r \frac{\partial T}{\partial r} \right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \phi^2} + \frac{\partial^2 T}{\partial z^2} \right] + \frac{q'''}{k} \tag{8.6b}$$

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \left[\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin^2\theta}\frac{\partial^2 T}{\partial \phi^2} + \frac{1}{r^2 \sin\theta}\frac{\partial}{\partial \theta}\left(\sin\theta \frac{\partial T}{\partial \theta} \right) \right] + \frac{q'''}{k} \tag{8.6c}$$

where α is the thermal diffusivity of material, $\alpha \equiv k/\rho c_p$.

8.2.2 Encapsulation and Impregnation of Electric Motor

As discussed in Chapter 4, the objectives of encapsulation and/or impregnation of electric motors are as follows:

- The encapsulation/impregnation of electric motor can remarkably enhance electrical insulation of motor windings from damage during normal operation and thus extend the motor lifetime.
- By filling voids in motor windings with epoxy resin and creating a rigid mass with optimum dielectric strength, an encapsulated and/or impregnated motor can significantly lower the thermal resistance from internal heat sources (e.g., the stator winding) to the motor housing.
- The encapsulation/impregnation increases the overall protection to motors, especially chemical and environmental protections.
- By integrating stator components (e.g., stator winding, laminations, and insulations) into a rigid mass, it increases the stator stabilization and thus reduces motor vibrations and noise levels.

Hesai Exhibit 1031, page 449

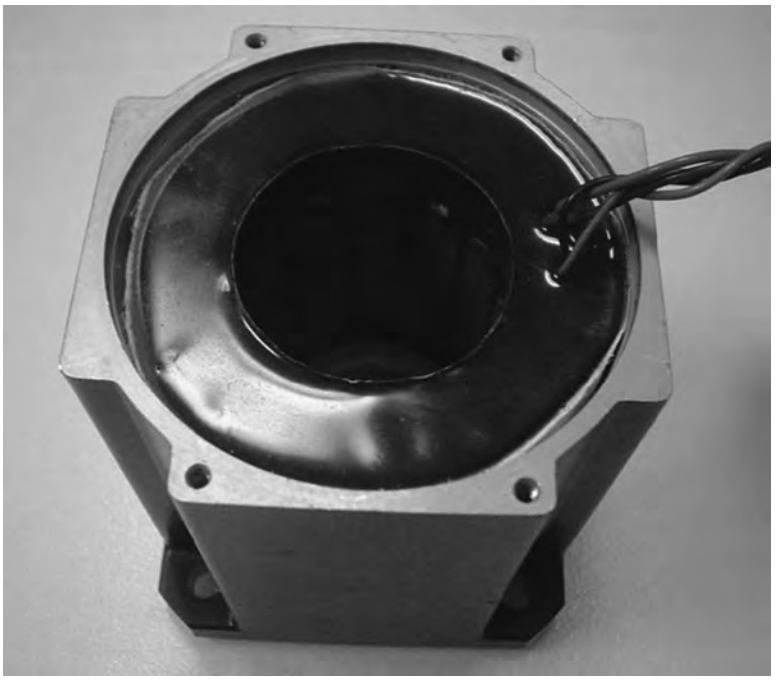


FIGURE 8.2
Encapsulation of stator winding.

Though many materials can be used in motor encapsulation/impregnation, the most popular encapsulation material is epoxy resin, used by many motor manufacturers in a wide range of applications. In the selection of encapsulation materials, thermal conductivity is one of the most important parameters. An epoxy-encapsulated stator is shown in Figure 8.2.

8.2.3 Enhanced Heat Transfer Using High-Thermal-Conductivity Material

One of the effective ways in motor winding cooling is to place high-thermal-conductivity materials between winding coils inside the winding slots and/or to interleave these materials into stacked motor laminations [8.11]. As shown in Figure 8.3a, the thermally conductive strips provide low thermal-resistant paths through which heat is transferred from internal high-temperature components (i.e., stator winding coils) to the stator laminations and housing and then dissipated to the environment. In Figure 8.3b, the thermally conductive strips are sandwiched between stacked stator laminations. In this way, heat generated inside the stator core due to the losses from eddy currents and hysteresis is conducted directly from the interior of the stator to the motor outer portions by means of these strips. Because these thermally conductive strips are directly contacted with heat sources, the cooling efficiency is considerably high. As a fact, this cooling technique can result in a significant reduction of weight and volume of motor, along with a substantial increase in the power density while operating at a moderate temperature above ambient.

There are a variety of advanced materials having exceptionally high thermal conductivities. Among them, the monolithic carbonaceous material are well known as the best

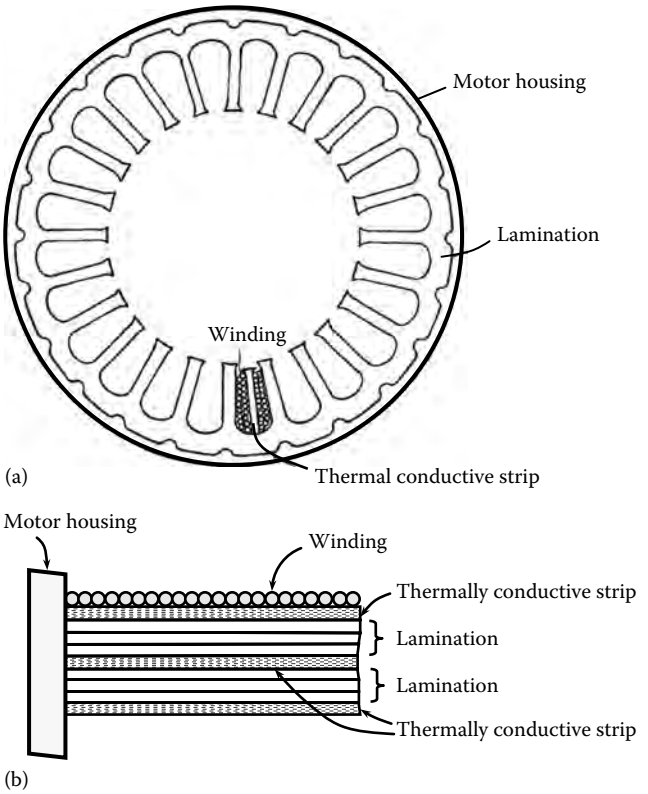


FIGURE 8.3
Using thermally conductive strips in motor cooling: (a) placed between winding coils, and (b) interleaved into stacked motor laminations (U.S. Patent 6,777,835) [8.11]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

thermally conductive materials, such as diamonds that are made from chemical vapor deposition (CVD) at low pressures and low temperatures. One of the remarkable properties of diamond is its unsurpassed thermal conductivity. With a value of 1800 W/m-K and higher at room temperature, it exceeds that of copper (approximately 400 W/m-K at room temperature) by a factor of 4.5 [8.12]. Furthermore, diamond had been the world's hardest material until recent years [8.13].

In addition to thermal conductivity, thermal stress arising from differences in CTEs between mating parts is a key issue in motor design. In general, materials with low CTE are highly desirable for their inherent dimensional stabilities. At temperature of 300 K (26.85°C), the CTE of CVD diamond is 1×10^{-6} K^{-1}, which is much lower than that of copper (17×10^{-6} K^{-1} at 20°C) and aluminum (24×10^{-6} K^{-1} at 20°C).

The superior properties of carbon fibers such as high thermal conductivity, low thermal expansion, high strength, lightweight, and corrosion resistance have made these materials especially useful for applications where heat removal is important. Some commercial carbon fibers have nominal thermal conductivities as high as 1100 W/m-K. Experimental discontinuous fibers reportedly have thermal conductivities of 2000 W/m-K [8.14]. The development of carbon nanotubes with extremely high thermal conductivities is a major breakthrough and will be discussed in later sections.

More recently, a new advanced material called Pyriod® HT pyrolytic graphite has been developed for enhancing heat transfer in electronic cooling. With a single crystalline structure and a high purity (99.999%), the thermal conductivity of the material can be as high as 1700 W/m-K [8.15], which is about four times higher than that of copper and seven times of aluminum (205–250 W/m-K at room temperature). Moreover, this material exhibits four times the ability to sustain tensile load than natural graphite material, nearly five times the flexural load and six times Young's modulus. The material can work at extremely high temperature, up to 3300°C. Because of its superior thermal and mechanical properties, Pyriod HT can be used for thermal management applications in a variety of industries including aerospace, defense, electronic, automobile, medical device, and power generation.

8.2.4 Using Self-Adhesive Magnet Wire for Fabricating Stator Winding

The use of self-adhesive magnet wires to make stator windings has a long history in linear motor manufacturing. In a linear motor, each stator coil is produced by winding self-adhesive wire into a coil shape, subjecting the coil to rapid electric heating to activate a bounding layer on the wire and pressing the coil into its final shape. When the coil is cooled down, it forms the self-supported robust coil (Figure 8.4).

This winding technique can be easily adopted into rotating electric motors. The use of self-adhesive wires may offer advantages over conventional wires in certain winding applications: (1) As a semisolid body with minimized voids inside the coil, conductive heat transfer in the self-adhesive winding is greatly enhanced, and in turn, hot spots of the winding are mitigated or eliminated. (2) Using of self-adhesive wires allows coils to be self-supported, so that bobbins as well as coil taping are no longer necessary. (3) A stator winding made from self-adhesive wires has much smaller volume compared with a conventional winding, leading to smaller motor size and high power density.

8.3 Natural Convection Cooling with Fins

With very low heat removal capacity, natural convection cooling techniques are limited to applications with low thermal loads. Natural convection flows are induced by buoyancy forces due to density gradients caused by temperature variations in the air layer adjacent to the heated surfaces. As shown in Table 8.1, without applying heat transfer enhancement

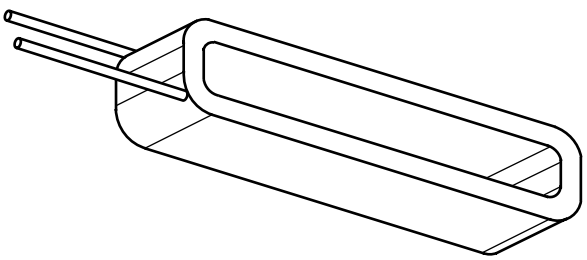


FIGURE 8.4
Stator coil made of self-adhesive magnet wire.

techniques, the heat transfer coefficient of natural air convection only ranges from 5 to 30 W/m^2-K. As a result, natural convection cooling has a limited role in motor cooling.

8.3.1 Cooling Fin

The rapid increase in power density of advanced motors and the miniaturization of motion control devices have created a need for improved cooling technologies to achieve high heat dissipation rates. Under low or moderate heat flux conditions, the most common approach in passive cooling techniques is to utilize natural convective flows that are induced and developed along hot surfaces due to density gradients in fluids (air or liquid).

In order to achieve reliability and optimal performance of motion systems, appropriate thermal management is imperative. Thus, it is desirable to increase the overall convective surface area exposed to the cooling fluid for enhancing heat transfer rates between the heat-dissipating surface and the convective cooling fluid. One of the effective methods to accomplish this objective is to directly employ various cooling fins on the external surfaces of the motor housing and endbells. Such augmentation can result in as much as an order-of-magnitude increase in the heat transfer rate without altering other motor components.

There are many techniques available to enhance heat dissipating from electric machines. Heat sinks are widely employed in electronic systems where space is limited. The use of passive natural convection–cooled longitudinal straight plate-fin heat sinks offers substantial advantages in cost and reliability but is often accompanied by relatively low heat transfer rates.

Cooling fins can be classified as either plate fins or pin fins. Plate fins are usually made of aluminum or copper by extrusion or casting processing. As shown in Figure 8.5, there are various types of plate fins, including (a) straight plate fin, (b) wavy plate fin, (c) radial–helical plate fin, (d) wavy protruding strip plate fins [8.16], (e) flared plate fin, (f) 3D radial plate fin, and (g) constructive fin [817,8.18]. The use of straight plate fins offers substantial advantages in cost and manufacturing but is often accompanied by relatively low heat transfer rates. For a plate-fin heat sink, cooling channels are formed with adjacent plate fins. When cooling flow passes through these channels, the velocity profile is identical except at the flow inlet regions. This indicates that the thickness of the flow boundary layers remains a constant both along the channel and across the channels. Bar-Cohen and Jelinek [8.19] developed guidelines and design equations for optimum plate-fin arrays.

With wavy plate fins, the cooling air flow changes its flowing direction periodically along the channel, causing the local flow separation and reattachment with the passage side surfaces. The disturbance between the cooling flow and the plate fins reduces boundary layer thickness and, as a result, increases the heat transfer coefficient. Because flared plate fins and 3D radial plate fins can effectively disturb local cooling flows, they can provide even higher heat transfer enhancement than other types of plate fins. In all designs, rough surfaces of heat sink help generate local turbulent flows that have higher heat transfer coefficient.

8.3.2 Fin Optimization

The optimization of motor cooling plate fins can be achieved by choosing appropriate values from the following design parameters: (1) fin thickness t, (2) fin spacing between adjacent fins b, (3) fin height H, (4) number of fins n, and (5) thickness of the base plate, d.

An effective way to optimize fin design is to use web-based tools. Though these tools were originally developed for the design of heat sinks, not quite the same for motor cooling

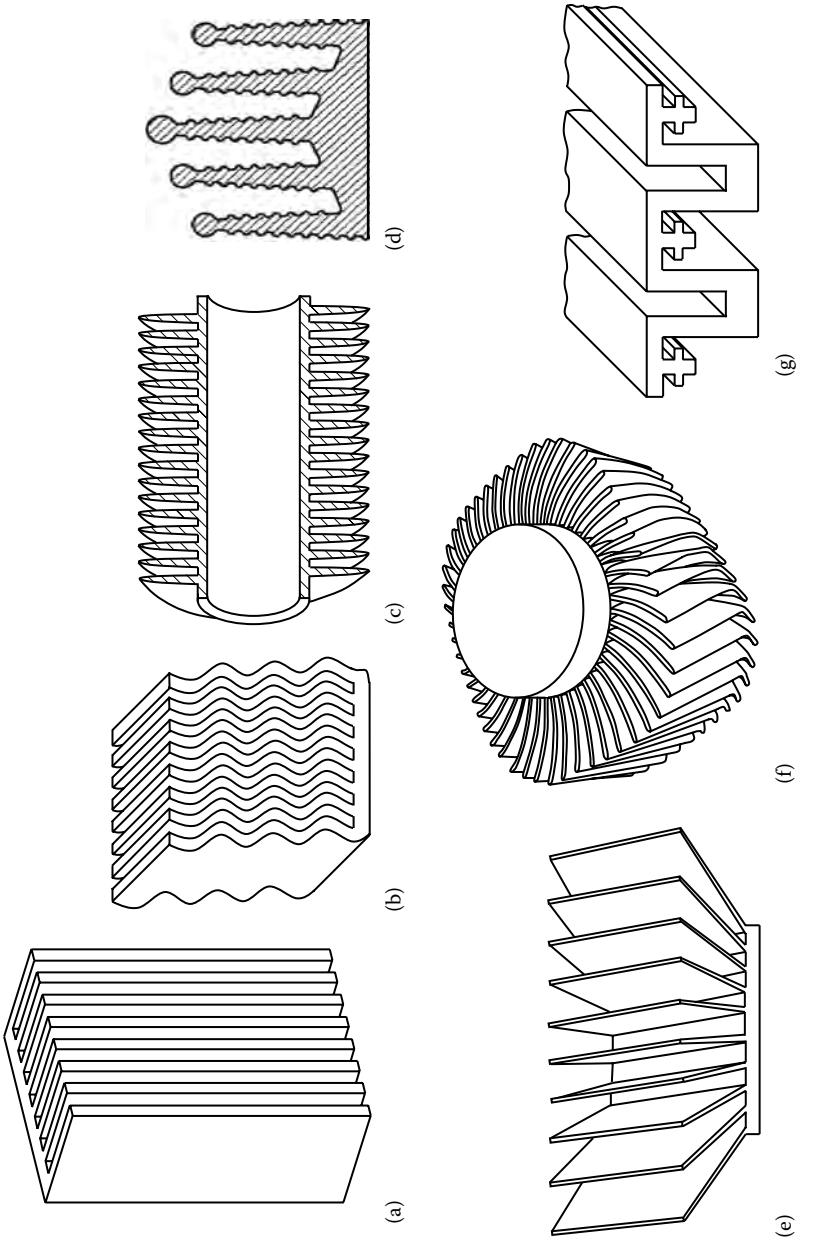


FIGURE 8.5
Types of plate fins: (a) straight plate fin, (b) wavy plate fin, (c) radial–helical plate fin, (d) wavy protruding strip plate fins, (e) flared plate fin, (f) 3D radial plate fin, and (g) constructive plate fin.

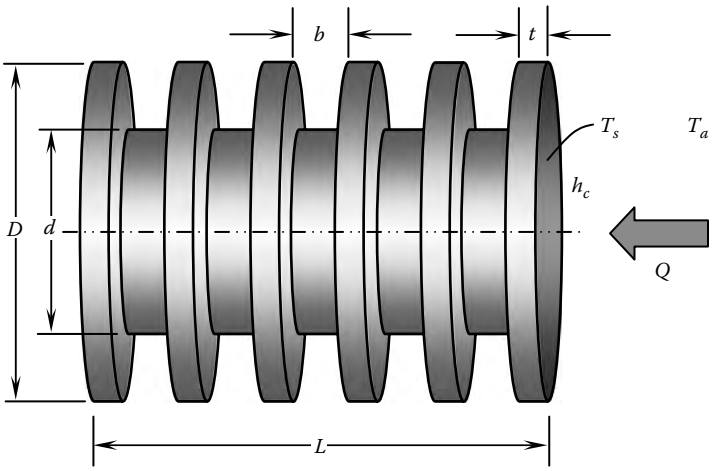


FIGURE 8.6
Thermal model of external fins on the motor's outside surface.

(e.g., the thermal load Q is generated inside the motor rather than entered at the end of the motor), it can provide a starting point for optimizing fin parameters such as fin thickness t, fin spacing b, and fin OD D, as shown in Figure 8.6. With the model in this figure, natural convection heat transfer can be calculated based on the geometry, material properties, and the boundary and ambient conditions [8.20]. Options are available to calculate both the total heat flowrate Q and the isothermal base plate temperature T_s through the independent control of the geometric parameters.

The straight plate-fin efficiency η_f is calculated as [8.21]

$$\eta_f = \frac{\tan h(mH)}{mH} \tag{8.7}$$

where
H is the fin height
m is the fin parameter, which is given as

$$m = \sqrt{\frac{2h}{k_f t}} \tag{8.8}$$

where
h is the convective heat transfer coefficient
k_f is the thermal conductivity of the fin material
t is the fin thickness

For radial plate fins, the fin height is calculated as

$$H = \frac{D - d}{2} \tag{8.9}$$

In calculating the heat transfer of heat sink, two nondimensional parameters are important, the Nusselt number Nu and the Raleigh number Ra, defined as [8.22]

$$Nu = \frac{Q}{A_t \Delta T} \frac{b}{k_a} \tag{8.10}$$

$$Ra = \frac{g\beta \Delta T b^3}{\alpha \nu} \frac{b}{D} \tag{8.11}$$

where

A_t is the total area of the outer surface exposed to air
k_a is the thermal conductivity of air
β is the thermal expansion coefficient
g is the acceleration of gravity
α is the thermal diffusivity
ν is the kinematic viscosity
$\Delta T = T_s - T_a$, the temperature differential between the heat source and the ambient

The overall motor-to-ambient thermal resistance R_{th} (in °C/W) is represented by

$$R_{th} = \frac{\Delta T}{Q} \tag{8.12}$$

To demonstrate how the web-based tools work, use the motor and fin parameters as follows: $d = 250$ mm, $t = 5$ mm, $D_1 = 340$ mm (for $H = 45$ mm), $D_2 = 330$ mm (for $H = 40$ mm), $L = 293$ mm, $Q = 1000$ W, and $T_a = 25$°C. The calculated results are displayed in Figure 8.8. It can be seen that with the increase of the fin ratio (defined as the ratio of the fin spacing b to the fin thickness t), the total thermal resistance decreases at lower fin ratios and increases at higher fin ratios. This is because that for a small fin ratio, a flow channel, which is formed between adjacent fins, is too narrow to allow enough cooling air to pass through it. Therefore, increasing the channel width (i.e., the fin ratio) will significantly improve the thermal transport between the cooling air and the fins. In contrast, as the fin ratio becomes very large, since a portion of the cooling air at the channel center region has a weak influence on the heat transport, the thermal performance will become worse for a very large fin spacing.

From Figure 8.7, it can be seen that the lowest thermal resistance occurs at the fin ratio of 1.6, regardless of the fin height. This indicates that for $t = 5$ mm, the best fin spacing is 8 mm. The corresponding fin number is 22.

Increasing the fin height will result in the decrease of the thermal resistance and in turn the increase of the heat transfer rate. As shown in Figure 8.8, when the fin height increases from 30 to 45 mm, the overall thermal resistance reduces about 22.8%. However, the fin height may be limited by several factors such as the motor's available space, manufacturing restrictions, and costs.

The calculated results can be used for estimating the improvement of a finned housing over a bald housing. The overall thermal resistance is 0.115 W/°C for the finned housing and 0.18 W/°C for the bald housing, indicating a 36% thermal improvement.

8.3.3 Applications of Various Fins in Motor Cooling

As shown in Figure 8.9, oblique plate fins are distributed oppositely on the surfaces of a totally enclosed nonventilated motor. The use of oblique plate fins can effectively reduce boundary layer thickness and improve the overall thermal performance.

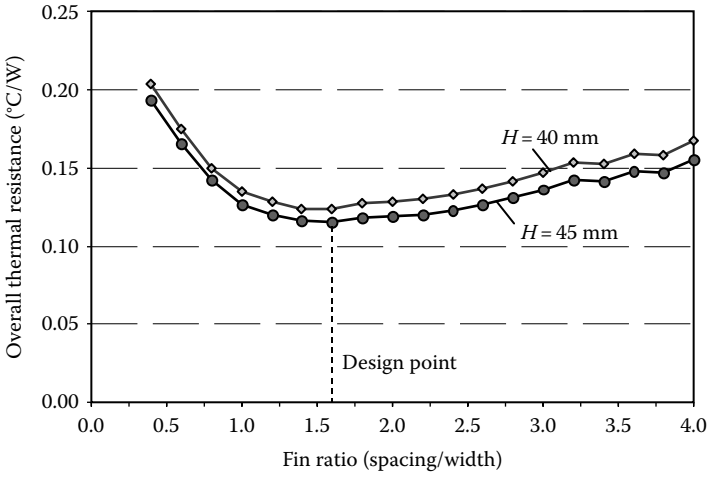


FIGURE 8.7
Effect of fin ratio on overall thermal resistance for a motor with finned housing.

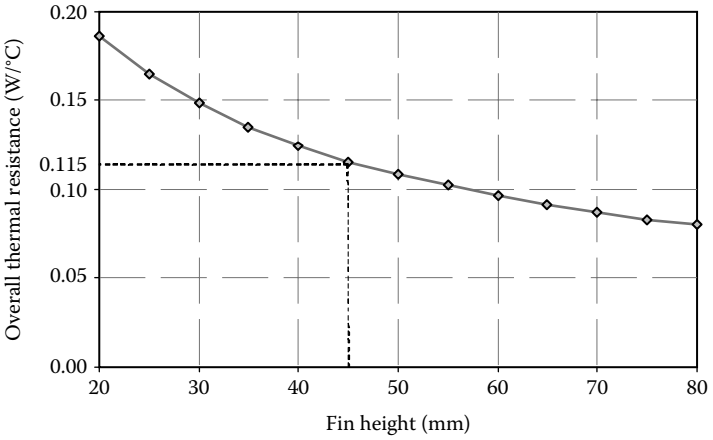


FIGURE 8.8
Effect of fin height on overall thermal resistance of a motor with finned housing.

Figure 8.10 is a perspective view of the motor frame with external and internal fins [8.23]. Because of the extended surfaces, heat generated in the motor during motor operation is dissipated into the environment at a much greater rate, thereby permitting the motor to operate under more severe load and environmental conditions without detrimental effects. This technique can be applied on totally enclosed or open externally ventilated motors. Motors that are totally enclosed and nonventilated depend entirely on the removal of heat from the frame surfaces by natural convection and radiation. Motors that have an open construction depend mainly on the moving of external cooling air into the interior of the motor and discharge the heated air out of the ventilating exits.

In order to increase the heat dissipation from the motor, cooling fins may be used at the outer surface of the endbell, as shown in Figure 8.11.



FIGURE 8.9
Oblique plate fins on the motor outer surface.

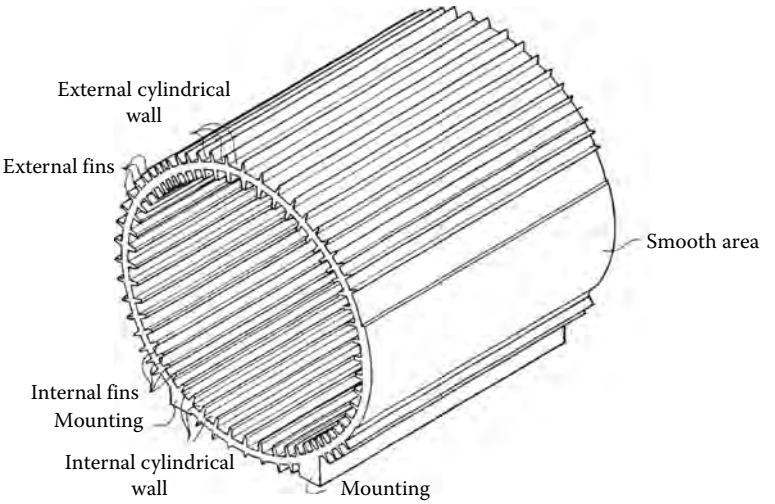


FIGURE 8.10
Outer and inner fins on the motor frame (U.S. Patent 4,839,547) [8.23]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

For some frameless motors, the stator laminations, all of one design, are designed in irregular shapes (e.g., asymmetric rhombus or parallelogram). When these laminations are positioned relative to one another, they form radially oriented external cooling fins. In such a way, motor cooling is benefited from not only the reduction in thermal resistance due to the elimination of the housing but also the heat transfer enhancement due to the effective increase in the heat-dissipative surface area and the disturbance of the cooling air flow resulting from the fins.

Stator fins may be fabricated by prestamping a number of radial slots around the OD of the stator laminations. As the laminations are stacked together as a stator core, a plurality of plate fins are formed in the axial direction [8.24], as shown in Figure 8.12.



FIGURE 8.11
Cooling fins on a motor endbell.

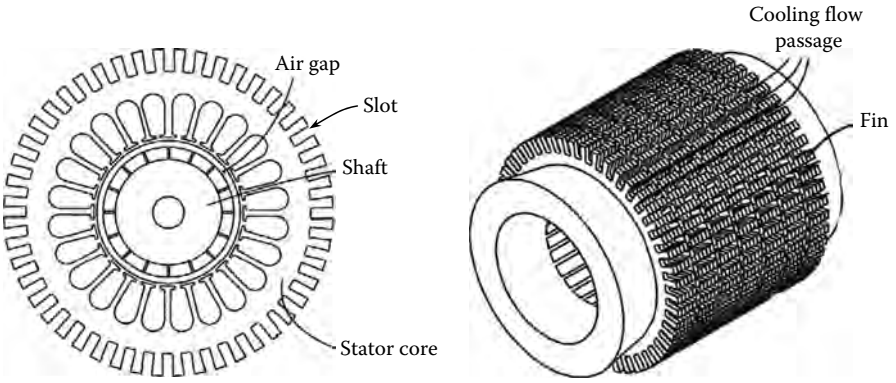


FIGURE 8.12
Stator laminations are stacked to form extending fins on the stator outer surface (U.S. Patent 8,053,938) [8.24]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

This design can be adopted to both frame and frameless motors. For a frame motor, adjacent plate fins define a plurality of longitudinal cooling air channels that extend the entire stator. This approach requires an axial fan to drive cooling flows passing through these cooling channels. As to a frameless motor, heat dissipates directly from the fins to the environment.

A similar design of stator lamination fins is presented in Figure 8.13 [8.25]. The stator consists of a plurality of groups of juxtaposed laminations. When stacking the laminations in one group, each cooling fin portion on each lamination is aligned with a respective cooling fin portion in each lamination such that the aligned cooling fin portions form a cooling

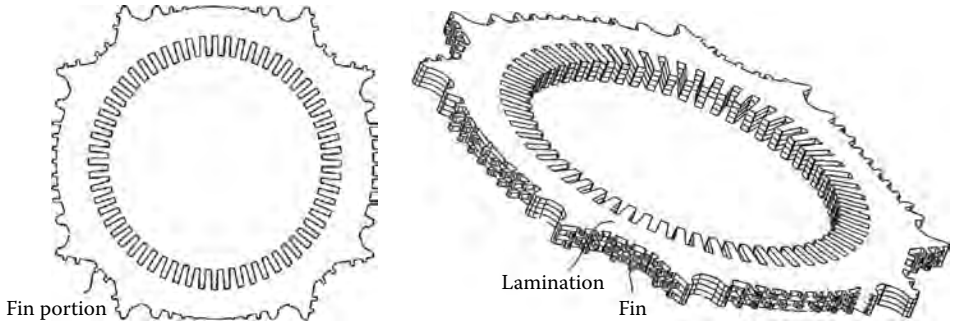


FIGURE 8.13
Fins formed from stacked stator laminations at the outer surface of the stator (WO 2005/022718) [8.25]. (Courtesy of World Intellectual Property Organization, Geneva, Switzerland.)

fin and a series of cooling fins in side-by-side arrangement are formed. Preferably, juxtaposed pairs of groups are arranged such that the cooling fins of one of the groups are staggered with respect to the cooling fins of the other group. With such shaped and arranged cooling fins, air flows between the cooling and adjacent fins are disrupted to reduce the thickness of the thermal boundary layers on the stator surfaces and hence increase conventional heat transfer coefficient.

8.3.4 Pin-Fin Heat Sink

To further improve the thermal performance of heat sinks, an array of pin fins has been used to replace longitudinal straight plate fins. Employment of such pin fins can result in better convective heat transfer characteristics because of the promotion of turbulence in the coolant passages and the increased convective heat transfer surfaces. Heat sinks using pin fins can minimize thermal resistance to improve convective cooling over traditional heat sink designs. It has been reported that with a plate–pin fin heat sink, which is the combination of plate fins and pin fins (as shown in Figure 8.14), thermal resistance can be 30% lower than the conventional plate-fin heat sinks [8.26].

Pressure drop across a pin-fin heat sink is one of the key variables that govern the thermal performance of the heat sink. In general, the total heat sink pressure drop depends on several design parameters and operation conditions, including the pin-fin geometry, pin-fin density and configuration arrangement, pin-fin size, heat sink orientation, and approach velocity (in forced convection). The design of heat sink is intended to decrease the impedance of the fluid flow through the heat sink and, thereby, reduce pressure losses.

Depending on different applications, many shapes of pin fins at the cross-sectional area can be adopted, for instance, round, square, rectangular, oval, rhombic, crescent, and raindrop shaped (Figure 8.15). Among them, the raindrop-shaped pin fins generate the lowest pressure drop across the whole heat sink by minimizing the drag force acting on fins and maintaining large exposed surface area available for heat transfer [8.27]. Square and rectangular pin fins are not preferred because they usually generate relative large flow resistances and thus high pressure drops.

Figure 8.16 depicts the flow pattern between raindrop-shaped pin fins. As cooling flow passes the pin fins, horseshoe vortices are generated in the stagnation area at the front edge of the pin fin by virtue of the staggered fin arrangement, and the flow separates

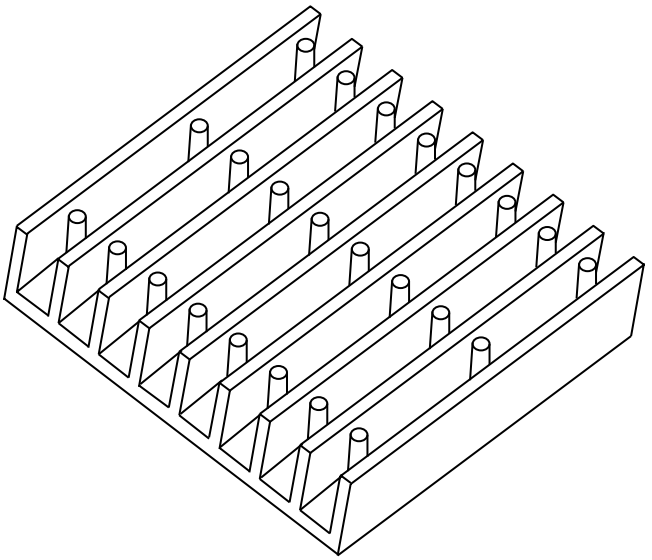


FIGURE 8.14
Plate–pin fin heat sink, which is the combination of plate fins and pin fins.

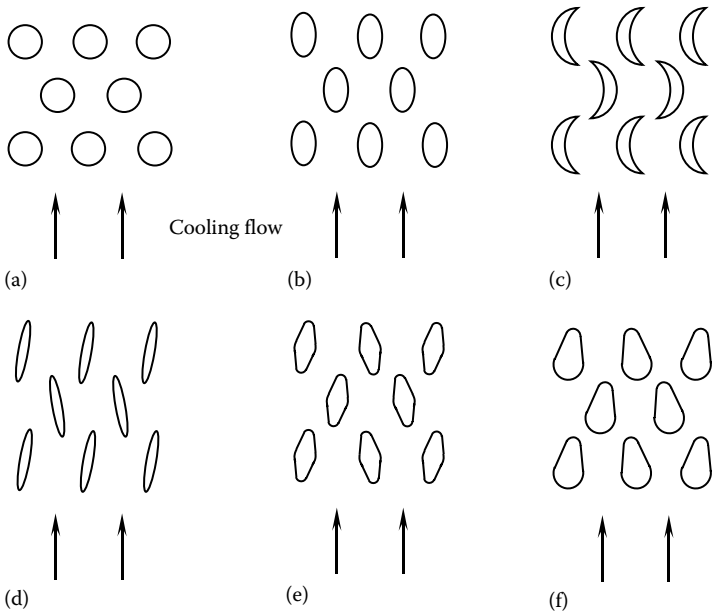


FIGURE 8.15
A variety of pin-fin shapes and arrangements: (a) round pin fins, (b) oval pin fins, (c) crescent pin fins, (d) tilted narrow oval pin fins, (e) tilted rhombic pin fins, and (f) tilted raindrop-shaped pin fins.

Hesai Exhibit 1031, page 461

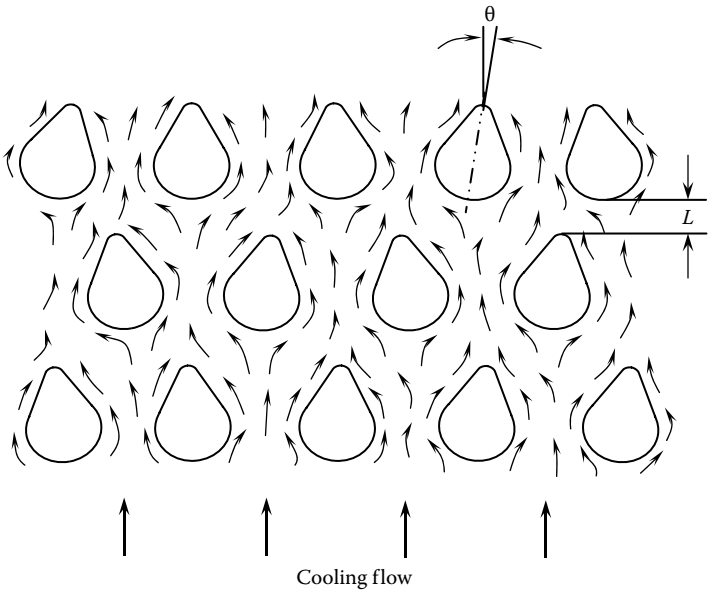


FIGURE 8.16
The flow patterns between raindrop-shaped pin fins. The horseshoe vortices generated in the stagnation areas of the pin fins and the flow separation and reattachment significantly increase the flow turbulence level and thus enhance convective heat transfer.

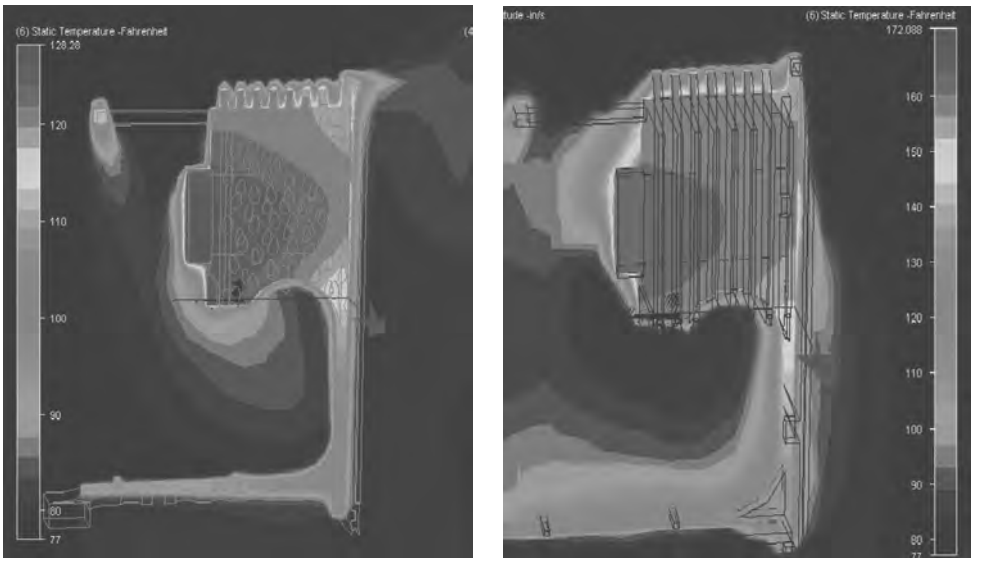
around the pin fins. The horseshoe vortices can essentially enhance the heat transport. The staggered and tilted arrangement of the pin fins increases the local turbulent level and reduces the size of the wakes, thus tending to reduce the local thermal boundary layer thickness and augment heat transport in these regions. As such, the pin fins serve as turbulence promoters to enhance heat transfer. The combination of the increased heat transfer coefficient and enlarged flow contact area considerably improves heat convection in the flow channel, thereby reducing or even eliminating hot spots.

The fin angle θ is defined as the angle between the fin centerline and the cooling flow direction. In Figure 8.16, a symmetry line is shown with a dash-dotted line. The angle θ of each pin fin may preferably vary between +20° and –20° for maximizing the heat transfer rate. Thus, the fin centerlines are nonparallel with respect to one another.

Another variable that can be used to maximize the heat transfer rate is the fin distance L between adjacent rows of pin fins. At the cooling flow upstream, because the temperature differential ΔT between the convective air and the heat sink is relatively large, the fin distance L can be large. As the temperature of cooling air increases along its path, ΔT becomes smaller and smaller. Thus, the fin distance L may vary accordingly for achieving higher cooling flow velocity and, in turn, higher heat dissipation rate. Regularly, the distance between adjacent rows of pin fins decreases along the cooling flow path.

There are several benefits of using raindrop-shaped pin fins. Firstly, the fins function as turbulent promoters to create local turbulence for enhancing heat transport. Secondly, the fins are designed to minimize the fin-induced wakes at the fin tails, whereby they provide the lowest resistance to the convective air flow. Thirdly, the local heat transfer can be easily controlled by varying the fin angle θ and the fin distance L. Fourthly, the local heat transfer can be alternatively controlled by changing the fin density and fin size along the cooling flow path.

Hesai Exhibit 1031, page 462



(a) (b)

FIGURE 8.17
Numerical analysis results show that under the identical operating condition, the maximum temperature for the raindrop-shaped heat sink (a) is 53.49°C (128.28°F), which is much lower than that for the conventional heat sink with straight plate fins (b) at 77.83°C (172.09°F).

A CFD model has been carried out to compare the thermal performance between the straight plate-fin heat sink and raindrop-shaped pin-fin heat sink. As shown in Figure 8.17a, under identical thermal conditions, the straight plate-fin heat sink has the maximum temperature of 77.83°C, whereas as shown in Figure 8.17b, the raindrop-shaped pin-fin heat sink results in a maximum temperature of 53.49°C. Thus, the heat sink with raindrop-shaped pin fins can efficiently enhance the heat dissipation and improve the heat sink thermal performance by approximately 10%–30% due to not only the extended surface area in contact with the cooling fluid but also the increased local turbulence level of the fluid flow.

The pin-fin configuration can significantly affect the cooling flow pattern and consequently the thermal performance of pin fins. For example, a crescent-shaped pin fin has a concave and a convex surface. As illustrated in Figure 8.18a, crescent-shaped pin fins are configured in a concave–convex pattern at fin rows. The orientation of pin fins between the adjacent rows is opposite. This configuration forces the cooling flow to change its flow direction periodically as it makes its way through the pathway. This causes local flow separation disturbances and subsequent reattachment of flow in the fin boundary layer and correspondingly increases the flow turbulence level and thus enhances convective heat transfer. In addition, the redeveloping boundary layer from the reattachment point also contributes to heat transfer enhancement.

Referring now to the embodiment illustrated in Figure 8.18b, the crescent-shaped pin fins are configured in a concave–concave and convex–convex pattern for all fin rows. This fin arrangement forms a large number of small convergent–divergent (convex–convex pattern) and divergent–convergent (concave–concave pattern) flow channels, configured in a staggered pattern along the flow path. As a cooling flow passes convex–convex fin

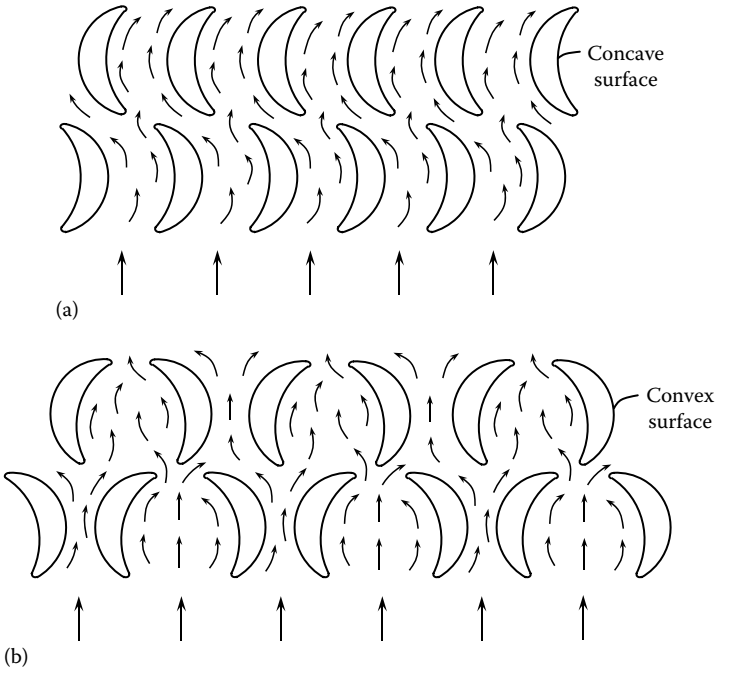


FIGURE 8.18
The flow patterns between crescent-shaped pin fins: (a) configured in a concave–convex pattern at fin rows and (b) configured in a concave–concave and convex–convex pattern for all fin rows.

channels, the flow is compressed at the channel entrance and expanded at the channel exit. Contrarily, as the cooling flow passes concave–concave fin channels, the flow is expanded at the channel entrance and compressed at the channel exit. Thus, the variations in the flow velocity, flow direction, and local flow pressure will increase flow reactions and turbulence level in the flow field, resulting in the high heat transfer performance between the fins and cooling flow.

Using an entropy generation minimization method, an optimal design of cylindrical pin-fin heat sinks is obtained for both the in-line and staggered arrangements [8.28,8.29].

Because of the thermally efficient structure of pin fins, motors with pin fins may provide excellent cooling solutions. Benefited from higher thermal contact areas and the promotion of turbulence in cooling flows, pin fins can provide much higher heat transfer rates than plate fins. However, pin fins have not been used in motor cooling yet, primarily due to the high cost of pin fins and some technical difficulties in fabricating pin fins.

8.3.5 Thermal Interface Materials

Thermal interface materials are extensively used in electronic cooling applications for reducing thermal resistance across a contact interface formed by any two solid bodies. Thermal contact resistance occurs due to the voids created by surface roughness effects, material defects, and misalignment of the interface. The voids on the interface are filled with air, which has very low thermal conductivity. Thus, the voids make a significant

contribution to the total thermal resistance. This is especially true for the systems with high thermal loads, a large number of contact parts, and complex surface contacts.

From the microscopic point of view, solid surfaces made by regular manufacturing processes are never perfectly smooth. Rather, the surfaces are constituted of asperities of different sizes and shapes. Therefore, when two solid surfaces are brought together, only a small portion of the matching surfaces make physical contact and the most of the matching surfaces are separated by a layer of air. The voids/gaps formed between the contacted surfaces thus result in high thermal resistance as heat is transferred across the interface. Therefore, a variety of thermal interface materials have been developed for filling voids/gaps, thus improving surface contact and heat conduction across the interface. Because thermal interface materials generally have much higher thermal conductivities than that of air, the interfacial thermal resistance can be significantly reduced.

There are several types of thermal interface materials [8.30]:

- Thermal greases typically offer the best thermal performance to serve as the most popular thermal interface materials. Thermal greases are made by dispersing thermally conductive fillers in viscous silicone or hydrocarbon oils. The use of thermal greases can effectively eliminate the interstitial air in voids or gaps and thus provide a low thermal resistance across the contacted surfaces. Thermal greases tend to wet the matching surfaces well while allowing retention of the high-thermal-conductivity asperity microcontacts. The conductivity of thermal greases is about 0.3 W/(m-K) [8.31], which is about 10 times higher than that of air.

  The characteristics of five greases were studied by Gowda et al. [8.32]. The thermal performance of these greases was measured using the laser flash thermal diffusivity method. The primary filler in these greases is spherical boron nitride (BN) with an average filler size of 60 μm. The testing results are summarized in Table 8.2. The results show that the increase in thermal conductivity is mainly due to the increase in filler loading for all five greases under different pressures. It also shows a significant reduction in thermal resistance with increasing pressure due to a combination of reduced bondline thickness and interfacial thermal resistance.
- Conductive-particle-filled silicone- or acrylic-based thermal tapes and pads offer high thermal conductivity, ranging from 0.7 to 7.3 W/(m-K), and low thermal resistance, ranging from 0.11 to 1.0°C/W, depending on the material and its thickness.
- As indicated by their name, phase-change materials can change their phase when temperature changes. At room temperature, the material is a film. As the temperature increases to a certain value, the viscosity of the materials becomes very small, allowing them to freely flow throughout the joint and fill the voids/gaps.
- Soft metal foils have been developed for years as compressible metallic shims in many applications, especially power devices. Soft metal foils are very thermally conductive, reliable, and easily adopted.
- Thermally conductive elastomers are silicone elastomer pads filled with conductive ceramic particles.
- Thermal adhesives, such as thermal adhesive pads, thermal adhesive tapes, and thermal adhesive films, provide low thermal impedance with long-term reliability.

Each type of thermal interface materials may display different levels of efficiency for reducing the thermal contact resistance, depending on the nature of the contacting

TABLE 8.2

Characteristics of Five Greases and Their Thermal Performance

					$P = 0.1$ MPa		$P = 0.2$ MPa		$P = 0.3$ MPa	
Grease	**Filler**	**Filler Loading (%)**	**Viscosity (Pa-s)**	**Density (kg/m³)**	**Bondline Thickness (μm)**	**Thermal Conductivity (W/m-K)**	**Bondline Thickness (μm)**	**Thermal Conductivity (W/m-K)**	**Bondline Thickness (μm)**	**Thermal Conductivity (W/m-K)**
A1	60 μm spherical BN	35.8	57.5	1200	56.4	4.2	51.3	4.3	48.3	4.4
A2		41.6	98	1300	71.1	5.2	67.3	5.4	59.2	5.2
A3		46.4	191	1400	92.7	5.7	74.9	7.9	69.9	8.3
B1	60 μm spherical BN + secondary filler	71.7	330	2200	50.0	6.5	47.0	7.9	43.9	8.4
B2		80.1	690	2400	57.2	7.9	51.3	9.8	51.8	10.3

Source: Tonapi, S. et al., *Electron. Cool.* November Issue, 2007.

Hesai Exhibit 1031, page 466

materials, contact conditions (e.g., applied contact pressure, temperature), morphological and crystallographic characteristics of the mating surfaces (e.g., surface roughness, hardness, and wettability), and the process parameters for the thermal interface material application technique.

When the interfacial gaps are filled with a thermal interface material while the surface asperity microcontacts are allowed to form, the heat transfer between two contact bodies can take place through both the microcontacts and microgaps with filled or unfilled interface material. Thus, the overall thermal contact resistance R_c can be determined by treating the microcontact resistance R_{mc} and microgap resistance R_{mg} in parallel:

$$R_c = \frac{R_{mc}R_{mg}}{R_{mc} + R_{mg}} \tag{8.13}$$

When the thermal interface material essentially fills most microgaps, there is little direct contact between two bodies, implying $R_{mc} \gg R_{mg}$ and $R_c \approx R_{mg}$. This indicates that the thermal contact resistance is generally determined by the thermal interface material.

To improve cooling performance of electric motors, thermal interface materials can also play an important role to enhance heat transfer between contacting motor components, such as the stator core and housing, the housing and endbells, winding coils and slots, and bearings and bearing bores.

8.4 Forced Air Cooling Techniques

Forced air cooling has long been applied in motor cooling. Over recent several decades, significant advances have been made in the application of air cooling techniques to manage increased thermal loads. Today, forced air cooling techniques are extensively used for various types of motors for their simplicity, low cost, and high reliability.

Typically, a motor is constructed to have either a totally enclosed architecture or an open architecture. A totally enclosed motor is designed to prevent external contaminating particles and other foreign matter entering into the motor. Therefore, for a totally enclosed motor, there is no air ventilation between the inside and outside of the motor. In contrast, an open motor architecture permits ventilating air entering and leaving the motor freely.

8.4.1 Thermophysical Properties of Air

Air is a mixture of gases, containing approximately 78% of nitrogen and 21% of oxygen, as well as traces of water vapor, carbon dioxide, argon, and various other components. The thermophysical properties of air are functions of temperature and pressure.

Figure 8.19 displays the specific heat of air as a function of temperature. It can be seen that both constant pressure specific heat c_p and constant volume specific heat c_v increase with temperature in a nonlinear manner.

Figure 8.20 presents the thermal conductivity of air as a function of temperature. As the temperature is raised, air molecules move more vigorously, and thus, heat energy is converted into kinetic energy of air. This will lead to more collisions between air molecules per unit time and, consequently, to the increase of the thermal conductivity of air.

Hesai Exhibit 1031, page 467

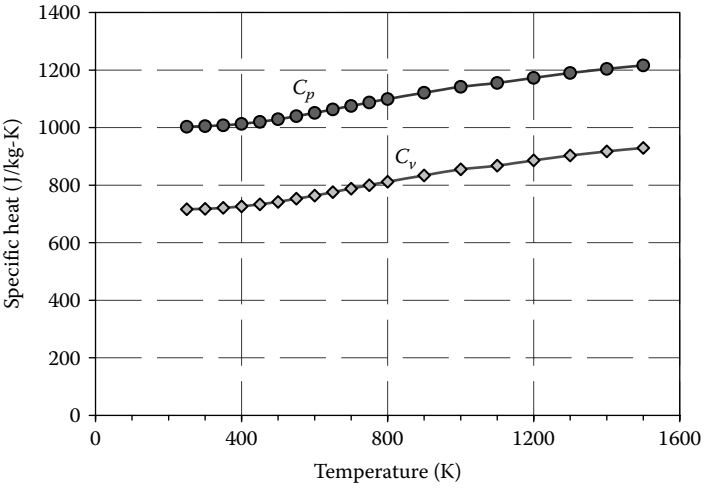


FIGURE 8.19
Specific heat of air as a function of temperature.

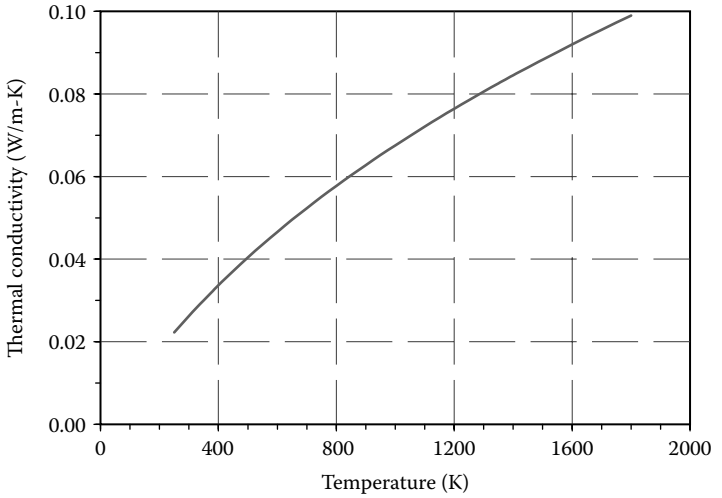


FIGURE 8.20
Thermal conductivity of air as a function of temperature.

Air density is an important parameter in determining the required mass flowrate, system pressure loss, and power. It can be seen from Figure 8.21 that air density decreases with the increase of temperature.

8.4.2 Direct Forced Air Cooling Techniques

Direct forced air cooling is the most popular cooling method for electric motors. In this cooling mode, cooling air is forced to contact directly with heat sources and exhausted out of the motor. It is to be noted that because the winding coils are always wrapped with

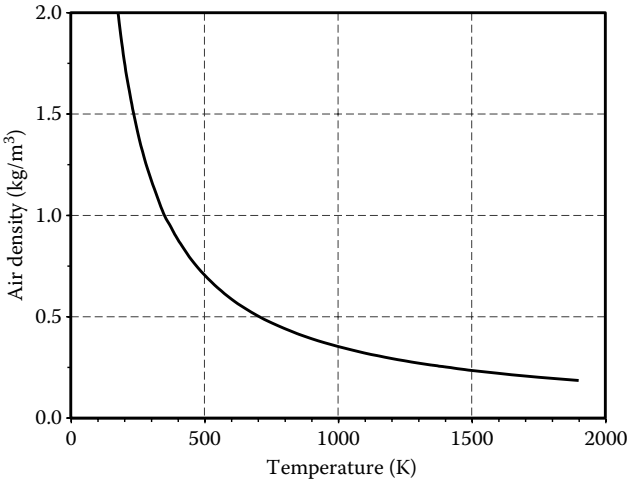


FIGURE 8.21
Air density as a function of temperature.

insulation materials that form a significant thermal barrier, the heat generated within the winding coils must first pass through the electric insulation by conduction.

8.4.2.1 Forced Air Cooling at End-Winding Regions

Hot spots may occur at the stator end-winding regions due to inappropriate cooling of end windings. This is because at the central winding region, the winding coils are in contact with the stator core. The heat generated by the windings is thus conducted through the stator core to the environment. In contrast, the heat generated at the end windings is transferred through two routes. One route travels along the copper coils to the stator core and the other route dissipates the heat to the air surrounding the end windings. Therefore, the thermal resistance of the end windings is considerably higher than that of the central winding. This is especially true for motors with a high *L/D* ratio. In practical motor design, special considerations often put on cooling at the end-winding regions.

According to the fan and cooling path arrangement, several patterns of direct air cooling systems can be identified for cooling at the end-winding regions. A simplest method is to cast a number of fan blades on the end rings during the casting process (Figure 8.22). As the rotor rotates, these blades generate turbulent swirling air flows surrounding the end windings and then force the exhaust hot air exiting the motor through the ventilation slots on the motor housing. For entirely enclosed units, because the heat carried by circulating air must be dissipated to the housing/endbells by convection with a relatively high thermal resistance, the use of this cooling mode may be only suitable for small motors with low thermal loads.

8.4.2.2 Forced Air Flowing through Internal Cooling Channels across the Motor

In this cooling mode, cooling air is introduced from one end of the motor and forced along the internal cooling channels across the motor and exhausted at another end of the motor. As shown in Figure 8.23, the internal fan is mounted at an end of the shaft and cooling air

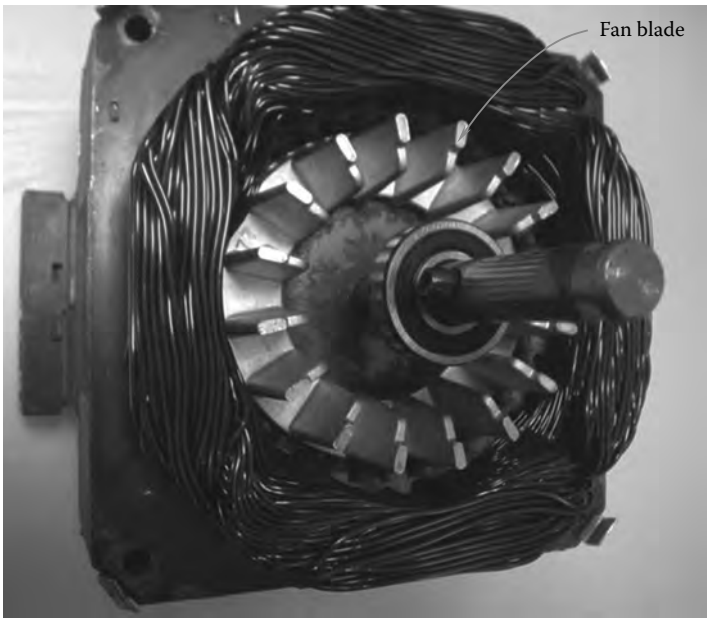


FIGURE 8.22
Rotor fan is casted together with the rotor conducting bars and end rings at each end of the rotor assembly for cooling motor windings, particularly winding end turns.

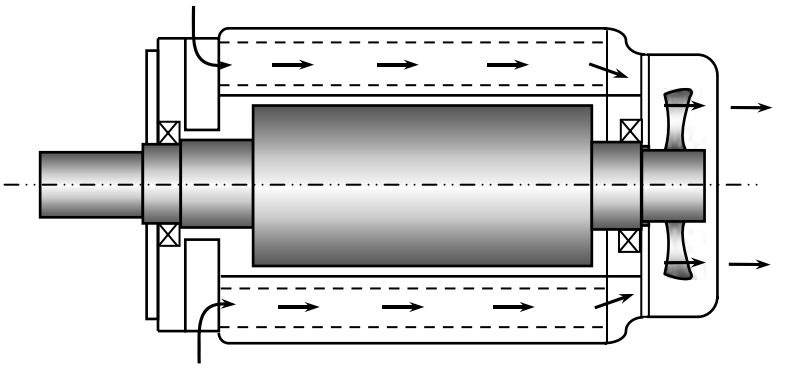


FIGURE 8.23
Open ventilating cooling system with sucked air forcing through stator cooling channels. In this design, a rotor fan is mounted at the nondrive end.

is sucked radially through the ventilating slots from the motor's outer surface and is forced to flow through the cooling channels within the stator core.

In order to provide enough cooling flows through electric motors, the arrangement of cooling channels can be rather flexible. The internal cooling channels may be formed and arranged in a variety of ways: (1) between the stator core and the motor housing (Figure 8.24), (2) through the motor housing (Figure 8.25), (3) through the stator core (Figure 8.26) [8.33], (4) between the state winding coils in the slots (Figure 8.27) [8.34], (5) within the rotor core (Figure 8.28), (6) between the rotor and stator cores (Figure 8.29) [8.35], and (7) a combination of any or all of the aforementioned (Figure 8.30).

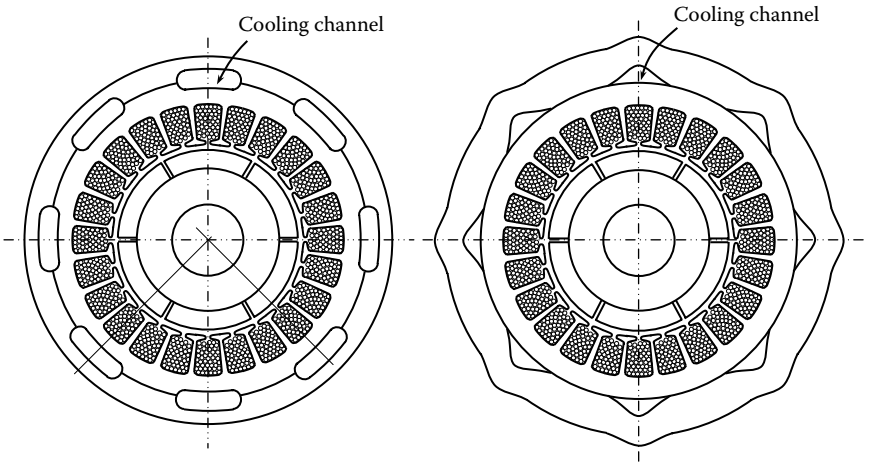


FIGURE 8.24
Arrangement of air cooling channels between the stator core and the motor housing.

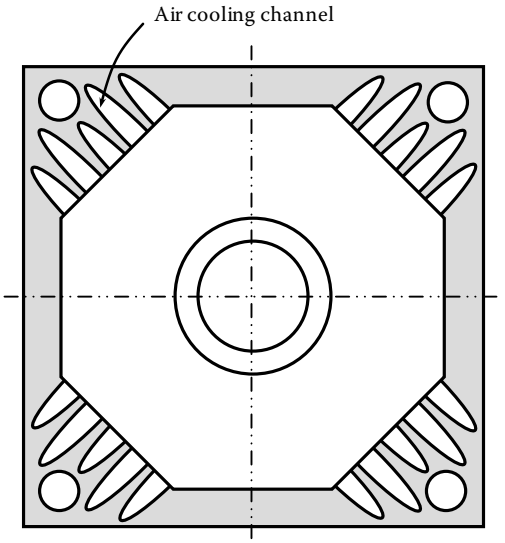


FIGURE 8.25
Air cooling channels through the motor housing.

It must be noted that as cooling air goes through the air gap or the channels through the rotor core, the cooling flow is subjected to a pumping pressure gradient in the radial direction as a consequence of the centrifugal effect, causing a complicated helical flow pattern and resulting in higher pressure drops across the flow channels but better cooling performances.

In addition, the formation of cooling channels through rotor cores or scallop-shaped channels on the rotor surface not only benefits motor cooling but also reduces the rotational inertia of the rotor for shortening the motor start time and prevents the PM between poles from magnetic leaking.

Hesai Exhibit 1031, page 471

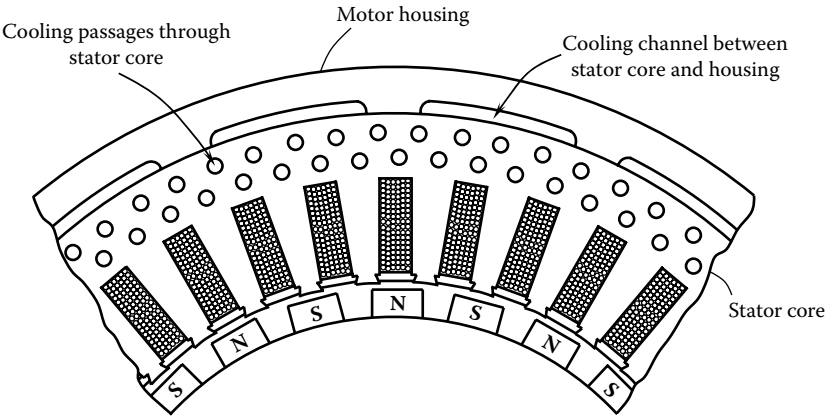


FIGURE 8.26
Cooling passages through the stator core and between the stator core and the housing (U.S. Patent 7,288,870) [8.33]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

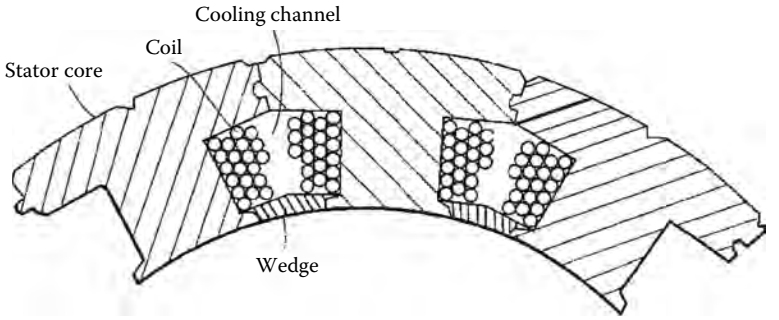


FIGURE 8.27
Cooling channels formed between stator winding coils (U.S. Patent 6,713,927) [8.34]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

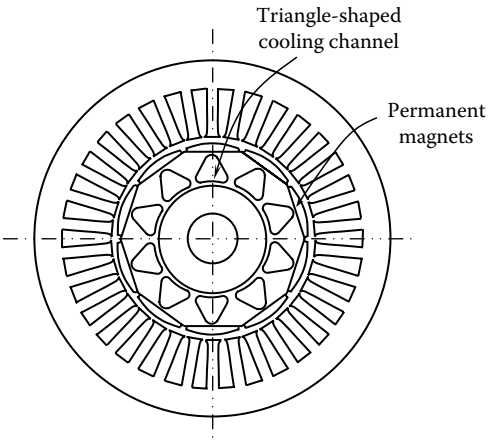


FIGURE 8.28
Triangle-shaped cooling channels through the rotor core.

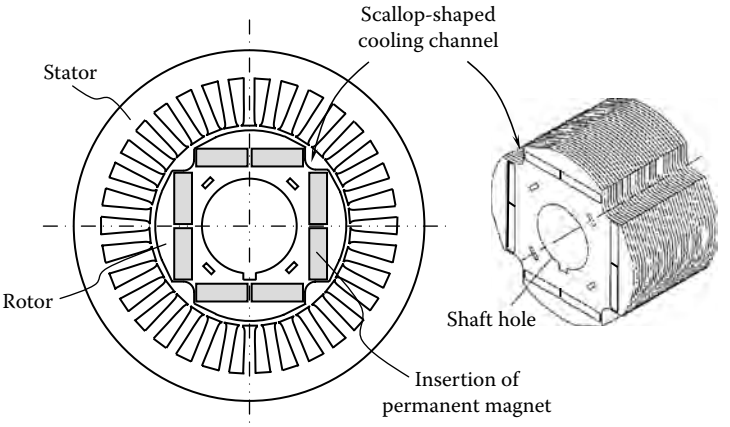


FIGURE 8.29
Scallop-shaped cooling channels between the rotor and stator (U.S. Patent 6,703,745) [8.35]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

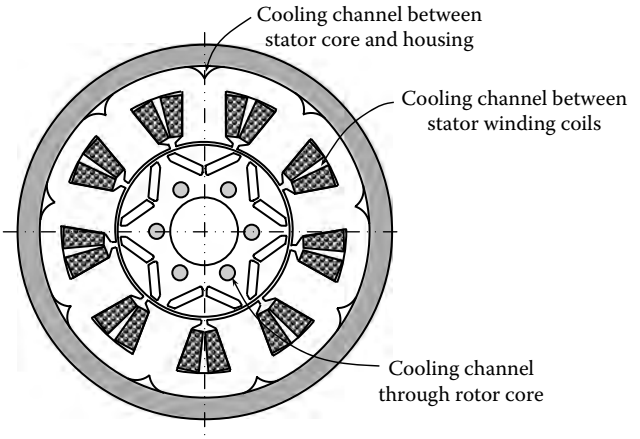


FIGURE 8.30
Combinations of different cooling channel arrangements.

8.4.2.3 Forced Air Flowing over Motor Outer Surfaces

For a totally enclosed motor, because external cooling air cannot enter into the motor, the motor may be cooled by forcing cooling air through the motor outer surface along axial plate fins (Figure 8.31). In this cooling mode, the fan is mounted on the rotor shaft outside of the motor so that the cooling air flowrate depends on the rotor rotating speed. Heat from the stator winding primarily relies on conduction from the stator to the motor housing and is dissipated to the motor surroundings by forced convection. It is noted that in this cooling mode, no internal fans are used inside the motor.

In some circumstances, mounting a cooling fan directly on the motor shaft of a totally enclosed motor is not practical. An alternative solution for motor cooling is to use an additional fan that is installed at the end of the motor. Thus, cooling air is blown over the motor

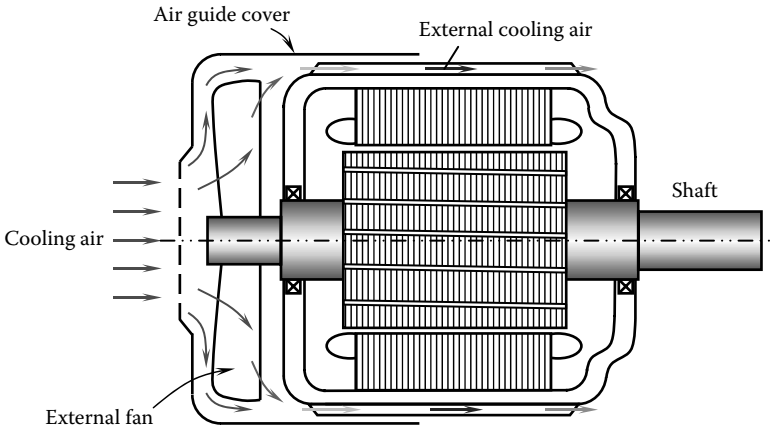


FIGURE 8.31
Cooling air is blown over the totally enclosed motor surface along axial plate fins by an external fan mounted on the shaft. No internal fan is used.

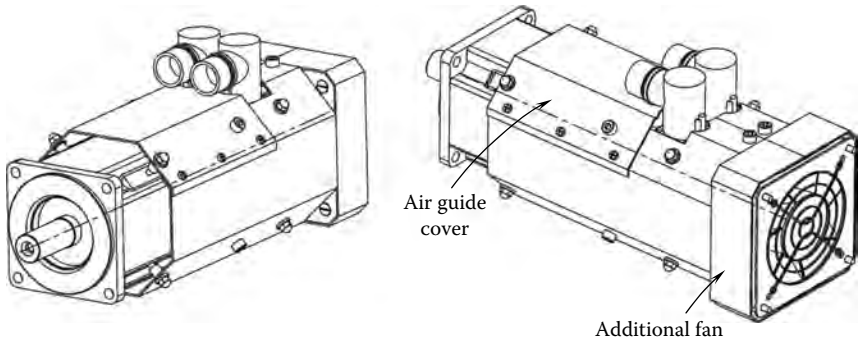


FIGURE 8.32
An additional cooling fan is mounted at the end of motor, blowing cooling air over the motor external surface. (Courtesy of Kollmorgen Corporation, Radford, VA.)

surface guided by the air guide cover (see Figure 8.32). This cooling mode can be flexibly adapted to meet different motor cooling requirements.

8.4.2.4 Forced Air Flowing through Both Motor Outer and Inner Surfaces

This cooling scheme is very similar to the one mentioned previously. To enhance heat transfer for totally enclosed motors, two sets of wafters are cast on the rotor end rings and applied to produce turbulent swirl circulating flows inside the motor. In such a way, the motor housing functions as a heat exchanger. As shown in Figure 8.33, heat generated by motor components is carried by the internal flow and is transferred to the housing wall. At the same time, cooling flow passing through the motor outer surfaces dissipates heat into the environment.

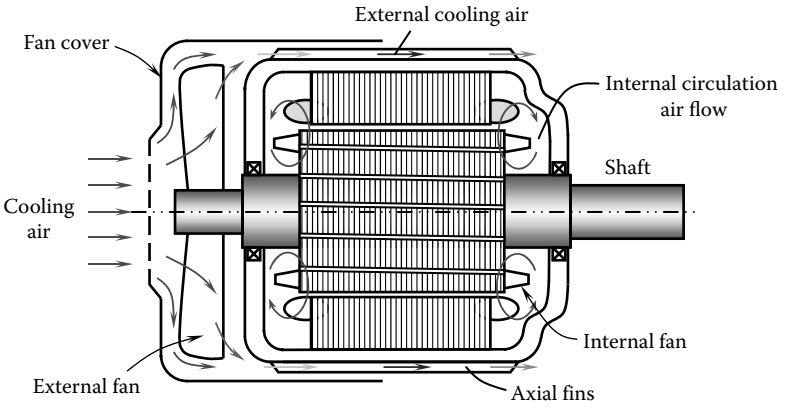


FIGURE 8.33
Motor is cooled by both external and internal air flows; the motor housing functions as a heat exchanger.

8.4.2.5 Air Jet Impingement Cooling

Jet impingement is an attractive cooling mechanism for its capability of achieving high local heat transfer rate in the region of stagnation point. This cooling technique has been extensively used in cooling of gas turbine blades. As a heat transfer enhancement technique, it has been widely used in the cooling of high-heat-flux components such as gas turbine blades and photovoltaic cells. With the increase in power output and the decrease in motor size, interest has been expressed by the motor industry in exploring this technique for cooling motor components. As shown in Figure 8.34 [8.36], the stator end windings are cooled using a jet impingement assembly, which includes a temperature-controlled fluid-generating device for bringing compressed air or other fluids onto the end windings. When the jet impinges on the end-winding surfaces, very thin hydrodynamic and thermal boundary layers form in the impingement region. Consequently, very high heat transfer

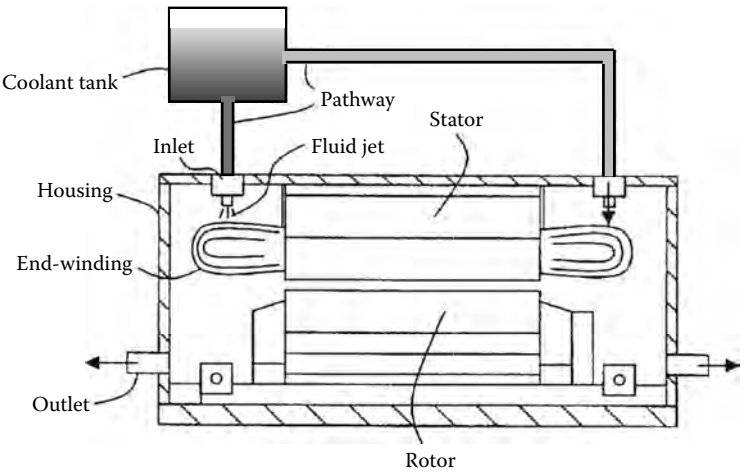


FIGURE 8.34
Cooling of end windings with an impinging jet cooling technique (U.S. Patent 6,639,334) [8.36]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

TABLE 8.3

Cooling Efficient Improvement over Natural Convection

Cooling Technique	Cooling Efficiency Improvement (%)
Forced air	14
Impinging air jet	48
Impinging liquid jet	77

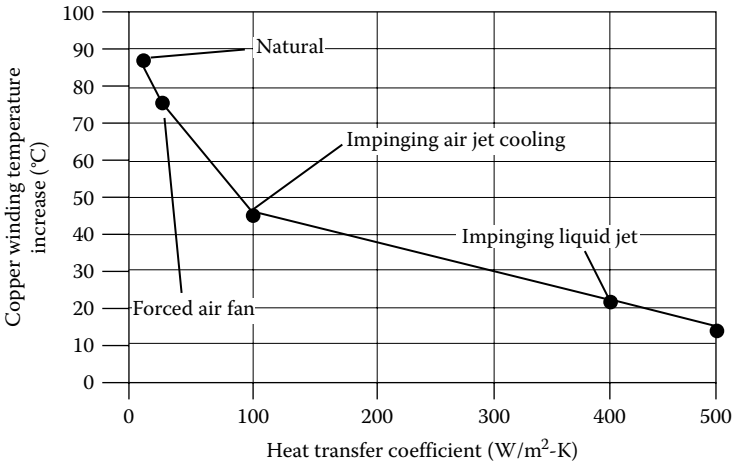


FIGURE 8.35
Temperature rise at end winding with different cooling methods (U.S. Patent 6,639,334) [8.36]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

coefficients are obtained within a stagnation zone. The experimental results have shown that impinging jet cooling offers up to 48% higher cooling efficiency for air impinging jet and 77% for liquid impinging jet over natural convection (see Table 8.3).

Figure 8.35 displays the end-winding temperature rise over the coolant temperature for different cooling methods. Impinging jet cooling, using either air or liquid, can provide a better cooling efficiency by comparing with natural convection and forced air fan cooling. The heat transfer coefficients of jet impingement cooling are much greater than that of natural convection.

The key parameters determining the heat transfer characteristics of a single impinging jet are the Reynolds number, Prandtl number, jet diameter, and jet to target spacing [8.37]. The optimization of an impinging jet system depends on the determination of these parameters through numerical analyses and experiments.

8.4.3 Indirect Forced Air Cooling Techniques

Although forced direct air cooling of a motor can provide a higher cooling efficiency, indirect forced air cooling techniques are often encountered on large motors. For indirect air cooling, air does not directly contact with heated components; the heat generated in a motor is transferred to the ambient by means of heat-exchanging devices.

8.4.3.1 Indirect Forced Air Cooling with Heat Exchangers

Indirect forced air cooling with either air-to-air or air-to-liquid heat exchangers is generally applied to large-size, high-powered electric motors. As depicted in Figure 8.36, the motor is equipped with two axial fans at the two ends of the rotor shaft to internally circulate cooling air within an enclosed system containing the motor and the air-to-air heat exchanger. One centrifugal fan is mounted on the shaft at the outside of the motor frame to force ambient air going through the small staggered tubes of the heat exchanger. In such a way, the energy balance holds between the heat energy dissipated from the internal circulation flows Q_h and the energy carried away by the ambient cooling flow Q_c:

$$Q_h = Q_c \tag{8.14}$$

It follows that

$$\frac{\dot{m}_h}{2}\sum_{i=1}^{2}\left[c_p(T_{hi,out})T_{hi,out} - c_p(T_{h,in})T_{h,in}\right] = \dot{m}_c\left[c_p(T_{c,out})T_{c,out} - c_p(T_{c,in})T_{c,in}\right] \tag{8.15}$$

where

$\dot{m}$ is the mass flowrate

$T_{h1,out}$ and $T_{h2,out}$ are the outlet temperatures of the internal circulation flow at the motor nondrive and drive side, respectively

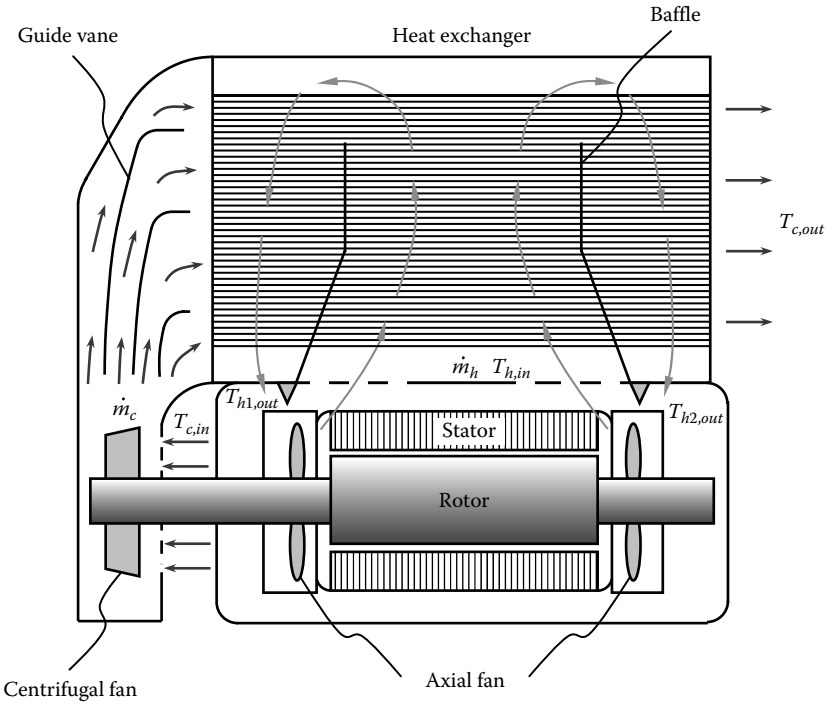


FIGURE 8.36
Forced air cooling with an air-to-air heat exchanger.

Hesai Exhibit 1031, page 477

This cooling method has been successfully used for a 2350 kW totally enclosed air–air cooled motor [8.38]. With the total loss of 78.93 kW in the motor, the measured temperatures at the copper winding and the iron core are 122°C–138°C and 111°C, respectively. The measured data, including the fan performance curves and the temperature profiles of the stator and the heat exchanger, have shown good agreement with the simulated results.

8.4.3.2 Indirect Forced Air Cooling via Indirect Evaporative Air Cooler

As shown in Figure 8.37, indirect forced air cooling is achieved via an indirect evaporative cooler with the orthogonal flow configuration. The principle of heat transfer in an indirect evaporative cooler involves the use of two types of air, namely, primary air and secondary air [8.39]. The primary air is isolated from the ambient air and is circulated in a loop to cool electric motor. The secondary air is induced from the ambient into the evaporative cooler. Water is sprayed into the passage of the secondary air so that heat and mass transfer takes place. Due to the latent heat absorption during the liquid–vapor phase transition, the temperature of the secondary air reduces and thus lowers the temperature of the primary air. Assuming the same spacing is used for the primary and secondary air, it has been found that the optimum heat transfer of an indirect evaporative cooler occurs at the velocity ratio of the primary to secondary air at approximately 1.4 [8.40].

Khmamas [8.41] has proposed an indirect evaporative cooling (forced or natural) method. With this cooling method, the air cooler is modified to operate as a cooling tower to produce cooling water by the evaporation process. The cooled water is then pumped to the indoor unit passing through a radiator that is cooled by a fan. The experimental results show that the evaporation cooling effectiveness (ECE) reduces by 15% for indirect forced evaporation case and by 22% for the indirect natural case, as compared with the direct cooling case. Although it is seen that the indirect method is less effective than the direct method, some advantages can be gained such as noise reduction and low power consumption.

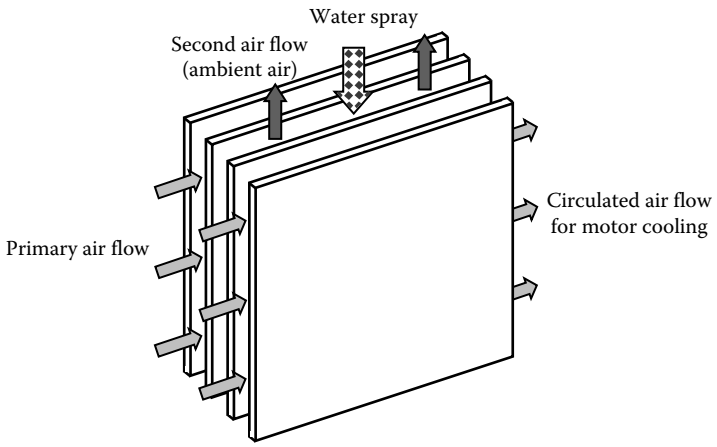


FIGURE 8.37
Forced air cooling with an indirect evaporation air cooler.

8.4.4 Fan and Blower

Fans and blowers are commonly used to provide cooling air flows to cool electric and electronic devices. They are differentiated by the method used to produce air flow and by the system pressure they must operate against. As a general rule, fans typically operate at pressures up to 13.79 kPa (2 psi) and blowers from 13.79 kPa (2 psi) to 193.1 kPa (28 psi), though some special designed fans and blowers may fall out of these ranges. In addition, blowers have much higher specific ratio (usually 1.11–1.20), which is defined as the ratio of the discharge pressure to the suction pressure, than fans (up to 1.11) [8.42].

To deal with the increasing heat loads in high-power-density motors, high-speed fans with noise, energy consumption, reliability, and weight penalties are often used.

8.4.4.1 Fan Types

Based on air flow discharge patterns, several generic fan types with varying air volume and pressure capacities can be identified such as axial fans, centrifugal fans, mixed-flow fans, and cross-flow fans. Axial fans, in which air flow is parallel to the axis of the blade rotation, are the most common type used in electric device cooling. This type of fans is the best choice for applications with high flowrates and low pressure drops. Therefore, the axial flow fans have been used in a wide variety of industrial applications, ranging from small fans for motor drive cooling to the huge fans employed in large wind tunnels. A typical axial fan used for motor cooling is shown in Figure 8.38.

A centrifugal fan has a fan wheel composed of a number of helical impellers or blades (Figure 8.39), mounted on a motor shaft. When the shaft rotates, the cooling air enters at the central part of the fan along the motor axis and then changes its flowing direction by 90° and finally is discharged in the radial direction of the impeller. During the process, the cooling air gains kinetic energy from the rotating impellers to increase its total pressure (i.e., the sum of velocity pressure and static pressure).

At a constant fan speed, a centrifugal fan can drive a constant volume of air rather than a constant mass. Centrifugal fans are the most prevalent fans used in ventilation and air conditioning applications.

A mixed-flow fan combines the high flow of an axial fan and a high pressure of a centrifugal fan. In practice, it uses a modified axial flow impeller to produce a radial component

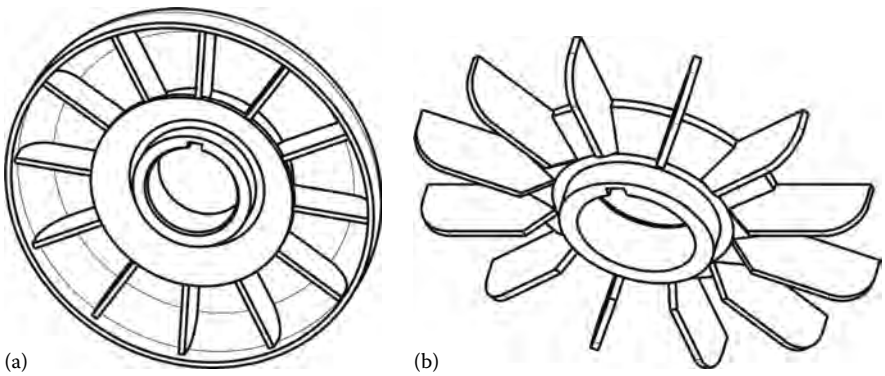


FIGURE 8.38
Casted or molded cooling fan: (a) fan structure (blades, hub, and shroud) and (b) blades on fan hub.

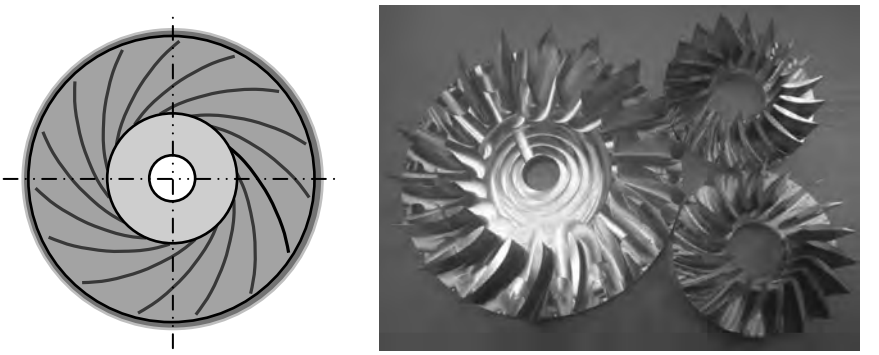


FIGURE 8.39
Helical impeller design in centrifugal fans.

of flow to add to a spiral flow. This type of fans has less outlet losses than axial fans. This provides more flexibility than with either axial or centrifugal fans in relation to fan position, fan outlet, and design of other components [8.43]. It is capable of operating throughout its performance curve without stalling. This type of fans produce less noise compared to axial fans.

Cross-flow fans, sometimes known as tangential fans or tubular fans, have been extensively used in the HVAC industry due to their silent operation and high-volume ventilating air flows at relatively low pressures. The cross-flow fan has a large *L/D* ratio and a drum-type impeller with multiple forward-curved blades. When the impeller rotates, a vortex stream is produced transversely across the impeller. By simply increasing the longitudinal axial length of the impeller, a high discharge flowrate can be achieved.

As shown in Figure 8.40, the air flows pass between the blades on one side of the impeller, through the internal space of the runner, and then through the blade passages for a second time to discharge on the other side of the impeller [8.44]. Thus, the efficiency of this type

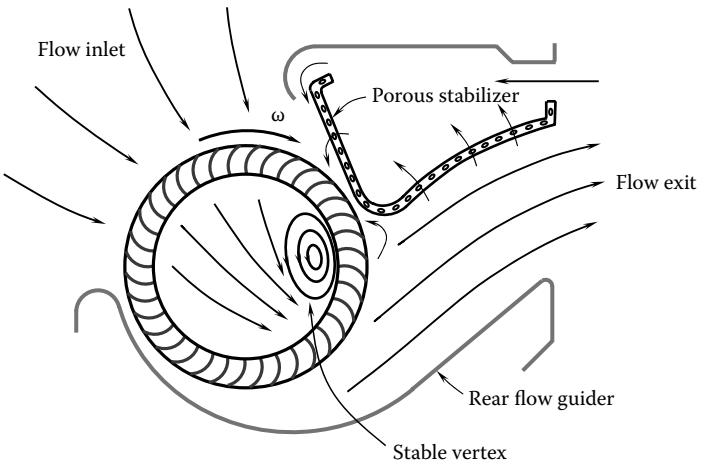


FIGURE 8.40
Cross section and flow pattern of a cross-flow fan.

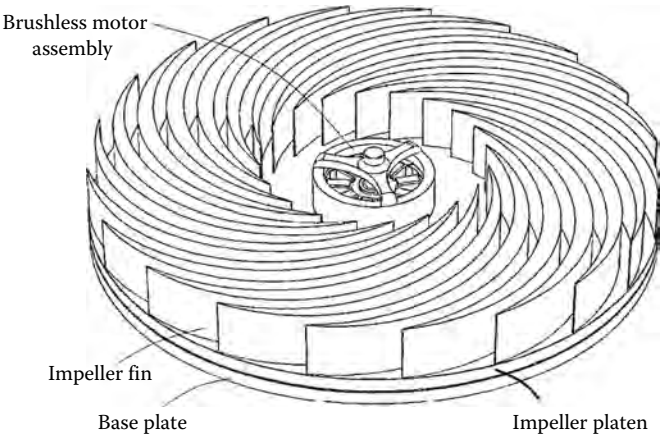


FIGURE 8.41
Combination of fan blades and heat fins as an effective cooling device (U.S. Patent 8,288,675) [8.46]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

of fans is relatively low (<40%) due to the air passing through the blades twice. The porous structure of stabilizer is used for attenuating the fan noise. The numerical results show that the porous stabilizer does affect the emitted noise from the fan. The weak air flows moving in or out through the porous wall of the stabilizer are believed to have played an important role in uniformizing the pressure in the zone of the stabilizer and the vortex and thus are helpful for weakening the vertex flow impingement on the stabilizer wall and reducing the relevant pressure oscillations [8.45].

An innovative device architecture invented at a Sandia National Laboratory [8.46] is presented in Figure 8.41. This device, called *heat sink-impeller*, combines an impeller and rotating fins together to achieve high cooling efficiency and degradation thermal resistance. It consists of a disk-shaped heat spreader populated with helical fins on its top surface and functions like a hybrid of a conventional finned metal heat sink and an impeller. It is a cast metal impeller that floats on a hydrodynamic air bearing just a thousandth of an inch (0.03 mm) above a metal heat pipe spreader, powered by a high-efficiency brushless motor in the middle. Air is drawn in the downward direction into the central region and expelled in the radial direction through the dense array of fins.

The prototype device has shown to provide a severalfold reduction in boundary layer thickness, intrinsic immunity to heat sink fouling, and drastic reductions in noise. This type of cooler is quiet and 30 times more efficient than fan-and-heat sink solutions. It was estimated that if every conventional heat sink in the United States is replaced with this device, the country would use 7% less electricity [8.47].

Though this device was invented mainly for electronic cooling, its fundamental can be adopted for motor cooling easily.

8.4.4.2 Forward-Curved, Backward-Curved, and Straight Blades of Centrifugal Fans

For centrifugal fans, fan blades can be made forward curved, backward curved, or straight (see Figure 8.42). Forward-curved blade fans have blades with leading edges curved forward to the DIR. The fan requires a scroll housing to convert the fan kinetic energy to useful static pressure. However, the efficiency of this type of fans is rather low, about

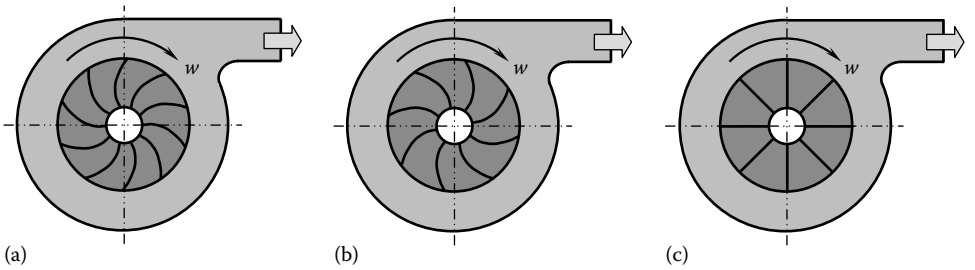


FIGURE 8.42
Three types of centrifugal fan blades: (a) forward-curved blade, (b) backward-curved blade, and (c) straight blade.

55%–65%. This type of fans requires more blades than other centrifugal fans and is best suited for applications that require high flowrates with low static pressures. Since the fans operate at relatively lower speeds, they generate relatively lower noise.

Backward-curved blade fans have blades with leading edges curved backward with respect to the rotation of the impeller. This type of fans runs faster than forward-curved type and is characterized by the most efficient fan, up to 70%–80%. The backward-curved blade fans are normally used for high air flow capacity and high static pressure applications. It should be noted that this type of fans can be designed either using or not using scroll housings, depending on the applications. For a nonscroll housing fan, the impeller can work effectively by itself.

As the simplest and least efficient centrifugal fans, straight blade radial fans have blades that extend straight from the shaft. This type of fans runs at medium speed but generates higher pressures than other types of centrifugal fans. They are often used in corrosive applications and in high-temperature environments.

8.4.4.3 Fan Performance Curve and Operation Point

Fan performance can vary widely among fan models and manufacturers. One of the most valuable pieces of information supplied by fan manufacturers is the fan performance curve, which gives the relationship between the volumetric flowrate $\dot{Q}$ delivered by the fan and the pressure difference Δp. Ventilating fans operate against pressure in their function of moving air through a motor. The fan pressure difference is normally expressed in either total pressure or static pressure (usually in the United States), and the fan volumetric flowrate is in either cubic meters per second (m^3/s) in the *SI* system or cubic feet per minute (*CFM*, ft^3/min) in the English system. Static pressure difference is defined as the difference between the fan inlet and the atmospheric pressure. The total pressure produced by a fan is the sum of the static pressure and the dynamic pressure.

When air flows through a cooling system (i.e., motor), a resistance is produced in the system against the air flow. This resistance, expressed as the sum of all static pressure losses, approximately varies as the square of the flowrate near the operation point of the fan. Thus, as the air flowrate through the system is doubled, the static pressure required to drive air flow is increased about four times. As illustrated in Figure 8.43, the operation point of the fan is determined by the intersection of the fan curve and the system resistance curve. A change in fan rotating speed and the use of the fan in different cooling systems will result in the shift in the operation point of the fan.

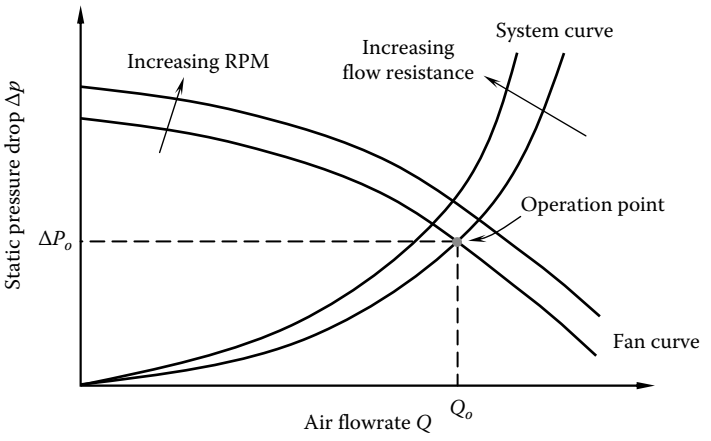


FIGURE 8.43
Fan performance characteristics.

8.4.4.4 Fan Selection

The selection of fans or blowers is critical to long-term successful operation of a motor cooling system. The factors impacting the fan or blower selection include the air flow capacity, efficiency, pressure, type of fluids, size, noise level, and cost.

As discussed previously, centrifugal fans have been frequently used in motor cooling and other high-pressure and low-flow applications. In some cases, they can be also used for low-pressure and low-flow general-purpose applications. Axial fans provide an economic solution in high flow volume and low-pressure systems (e.g., ventilation fans). The type of mixed-flow fan is a hybrid of centrifugal and axial fans, which are capable of greater air flow than centrifugal fans and higher pressures than axial fans. With more complex design, mixed-flow fans typically have higher efficiency, lower noise, and higher cost, compared with other types of fans.

The efficiency of a fan is largely determined by the design of fan blades. Specially designed airfoil blades can drive flow more efficiently and lower fan losses. Therefore, they are suitable for use in compact motors that require high cooling efficiencies with limited spaces. However, airfoil blades are specifically designed for a certain rotating direction. For motors that require bidirectional rotation (e.g., wheel-driving motors), this type of blade is no longer applicable.

8.5 Liquid Cooling Techniques

While traditional air cooling techniques have provided the lowest cost options and served small- or medium-sized electric motors well for a long time, liquid-cooled solutions have been recognized as the best means for cooling high-powered and large-sized modern electric machines. In recent years, liquid cooling is accepted as an attractive cooling method for an increasing number of applications due to higher heat transfer coefficients achieved as compared to air cooling. By moving from air cooling to liquid cooling, thermal restrictions on the design of large-size and high-power-density motors can be removed.

In fact, liquid cooling effectively increases the continuous torque rating of the motor. Application of liquid cooling may be categorized as either direct or indirect, depending upon the liquid direct or indirect contact with the heated motor components. Liquid cooling is accepted as an attractive cooling method for an increasing number of applications.

8.5.1 Thermophysical Properties of Coolants

In the selection of a suitable coolant in liquid cooling, a number of parameters must be carefully considered, including (1) thermophysical properties; (2) electrical properties (especially dielectric strength); (3) chemical compatibility of the coolant with the motor components; (4) freezing, boiling, and flash points; (5) toxicity; (6) erosion; (7) flammability; (8) impact on human health; and (9) cost. Thermophysical properties of some selected coolants including FC series coolants (which are extensively used in electronic cooling) are given in Table 8.4, compared with water. As can be seen from the table, water has the highest specific heat and the lowest viscosity. However, because water is not a dielectric liquid, it may be used for indirect cooling cases.

TABLE 8.4

Thermophysical Properties of Liquid Coolants

Coolant	Density (kg/m^3) at 1 atm	Specific Heat (J/kg-K)	Thermal Conductivity (W/m-K)	Latent Heat of Vaporization (kJ/kg)	Dynamic Viscosity (Pa-s) $\times 10^{-3}$ at 20°C	Freezing Point (°C)	Boiling Point (°C)
Aerosol	1100			216.7	0.205	−101.0	−26.1
Ammonia	609	4740	0.521	1369	0.138	−77.75	−33.34
Aromatic	860	1750	0.14		1.0	<−80	80
Dynalene HC-30	1275	3077	0.519		2.5	<−40	112
Engine oil (SAE 40)	887	1765	0.138		81.3	−12	> 310
Ethanol/water (44/56 by weight)	927	3500	0.38	1636	3.0	−26.6	
Ethylene glycol/water (50/50 by volume)	1069	3350	0.39	1.528	3.4	−37	107.2
Methanol/water (40/60 by weight)	935	3560	0.4	1.344	2.0	−38	78.9
Potassium formate/water (40/60 by weight)	1250	3200	0.53		2.2	−35	110
Propylene glycol/water (50/50 by volume)	1041	3559	0.37	1.585	5.4	−34	105.6
Silicate ester (Coolanol 25R)	910	1990	0.135		4.6	−50	
Silicone (Syltherm XLT)	852	1647	0.11	172	1.4	−111	173
FC-43	1880	1100	0.066	71	5.264	−50	174
FC-72	1680	1088	0.055	87.9	0.672	−90	56
FC-77	1780	1172	0.057	83.7	1.424	−95	97
FC-87	1650	1100	0.056	103	0.66	−115	30
R134a	1188	1420	0.013	215.9	0.209	−96.6	−26.1
Water	997	4181	0.613	2.257	1.002	0	100

For natural and forced convection, the most important parameter is the specific heat of the coolant. This is because it determines how much heat energy can be carried away by the coolant as the coolant mass flowrate $\dot{m}$ and the temperature difference ΔT are specified. A high specific heat is aimed to transfer a large amount of heat with a minimum fluid flow. A low viscosity is also desired because it helps to reduce the resistance of fluid flow. For phase-change heat transfer mechanisms such as evaporative cooling and boiling, the latent heat of vaporization is most important. Boiling and freezing points define the working range of coolants.

8.5.2 Direct Liquid Cooling Techniques

In direct liquid cooling techniques, liquid coolants are directly brought into contact with stator and rotor windings/cores. Therefore, this form of cooling offers excellent opportunities for efficiently removing heat directly from these heat sources without intervening thermal conduction resistance. As a result, direct liquid cooling has a higher cooling efficiency than other convective heat transfer modes such as air cooling and indirect liquid cooling.

The heat transfer coefficient in direct liquid cooling not only depends on the thermal physical properties of the coolant but also on the mode of convective heat transfer (i.e., natural convection, forced convection, or boiling). As shown in Figure 8.44 [8.48], due to the phase change during heat transfer processes, boiling heat transfer offers the highest heat transfer coefficient among all combinations of coolant and heat transfer mode. Though water has showed excellent heat transfer performance, it may be only used for indirect motor cooling for its poor dielectric strength.

Closed-loop systems are normally applied in direct liquid cooling (as well as in indirect liquid cooling) due to the benefits including reducing the consumption of expensive liquid coolant, reducing operation and maintenance costs, avoiding environmental issues associated with the coolant, and lowering noise levels.

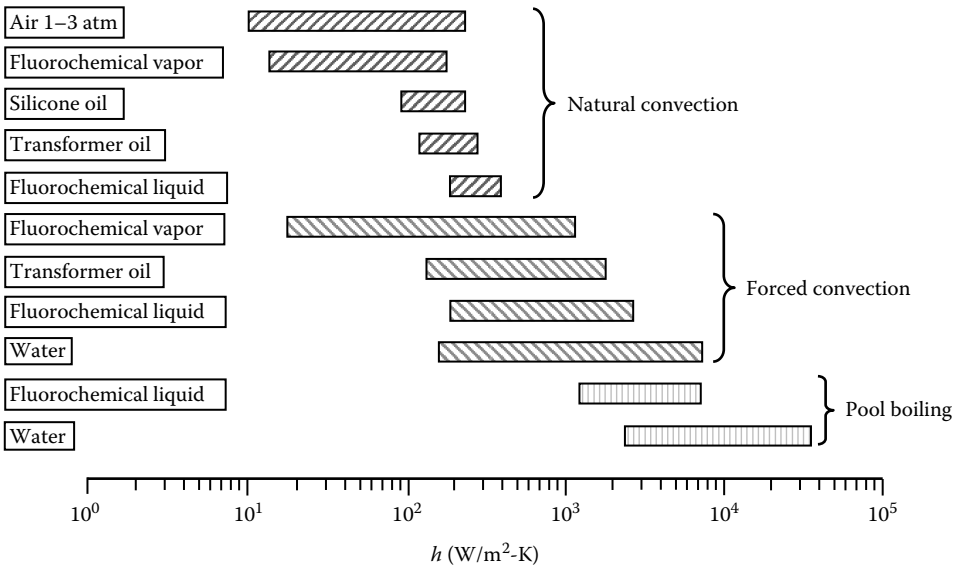


FIGURE 8.44
Relative magnitude of heat transfer coefficients for various coolants and modes of convection. (From Simons, R.E., *Electron. Cool.*, 2(2), 24, 1996.)

Hesai Exhibit 1031, page 485

There are a number of direct liquid cooling techniques available in motor cooling, such as liquid spray, impinging jet, mist spray, and splash cooling. These cooling techniques are addressed in more detail in the following.

8.5.2.1 Direct Liquid Cooling of Bundled Magnet Wires

This innovative method employs shrink wraps to make magnet wires in bundles and thus liquid cooling channels are formed surrounding the magnet wires [8.49]. As shown in Figure 8.45, a plurality of magnet wires pass through each of the lamination slots. A shrink wrap is inserted into the lamination slots. The stator laminations, shrink wrap, and the magnet wires are heated to shrink the shrink wrap onto the outside of the magnet wire. Therefore, the magnet wires are secured in bundles by the shrink wrap, allowing dielectric oil to move freely inside the lamination slots surrounding the wire bundles. Since fluid can pass between and around the windings, the overall winding temperature will be normalized, essentially reducing hot spots that occur in conventional designs.

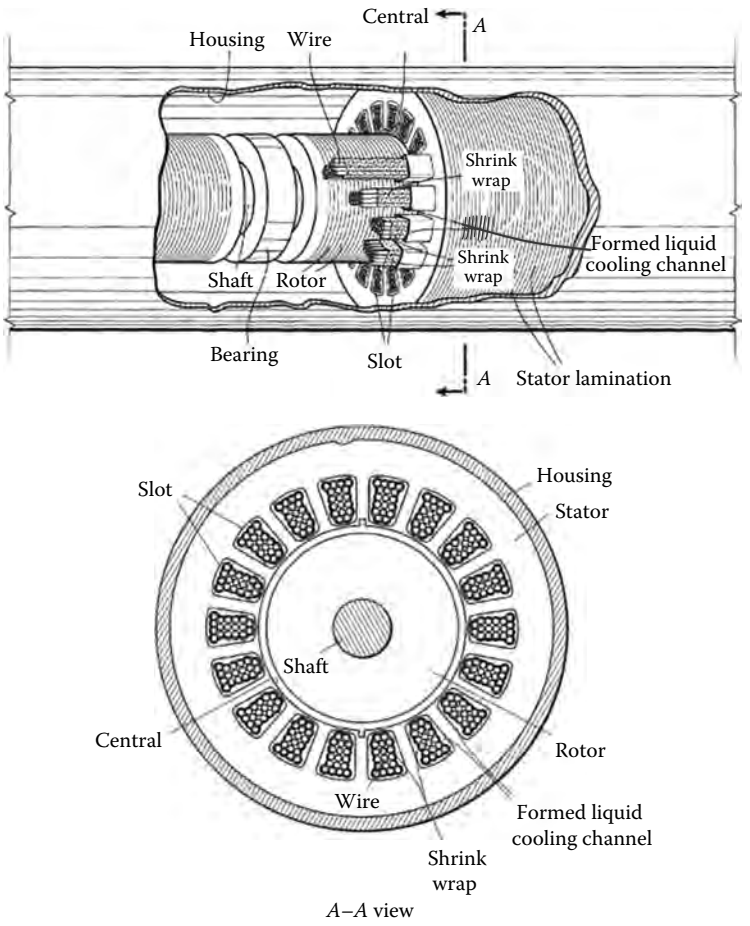


FIGURE 8.45
Enhanced heat transfer using shrink wrap surrounding magnet wires (U.S. Patent 7,679,242) [8.49]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

Hesai Exhibit 1031, page 486

8.5.2.2 Spray Cooling

Spray cooling is an efficient form of liquid cooling for applications having low to moderate thermal loads. This cooling mode offers an economical and efficient approach for extracting heat from hot surfaces through the use of latent heat of vaporization of a liquid. This approach is to spray liquid droplets onto hot surfaces to create a thin film on the surfaces. As the liquid evaporates, it absorbs heat from its surroundings and lowers the temperature on the surfaces. Hence, it has been widely used in cooling military electronics. As an efficient cooling method, this approach may be promisingly applied to motor cooling.

As demonstrated in Figure 8.46, a spray cooling system primarily consists of a liquid tank, a flow control valve, a pump, a filter, a heat exchanger, and a number of nozzles. The cooling liquid is pumped from the liquid tank and becomes droplets after passing through spray nozzles. By wetting the motor surface with liquid droplets, heat is transferred from heat sources inside the motor to the liquid film and finally dissipated to the environment via forced convection due to the liquid spray, as well as air impinging flow induced by the liquid spray, evaporation, and/or nucleation (in the case that boiling occurs). Obviously, the spray cooling process mainly relies on the phase change from liquid to vapor. Correspondingly, the exceptional cooling capacity is greatly enhanced due to the latent heat of vaporization.

It should be noted that special machining capabilities are required to manufacture spray nozzles to the precision and tolerance required. It has been revealed the variations in nozzle performance due to contamination, corrosion, and long-term wear. Placement of spray nozzles is fairly critical to assure adequate cooling.

The internal spray cooling technique has been used in motors subject to high heat fluxes. As a rotor rotates, liquid coolant such as engine oil is flung off the rotor and/or shaft surfaces in small droplets due to the centrifugal force acting on the coolant. The droplets create a fine mist and impinge on heated motor components. The droplet impingement enhances the spatial uniformity of the heat removal.

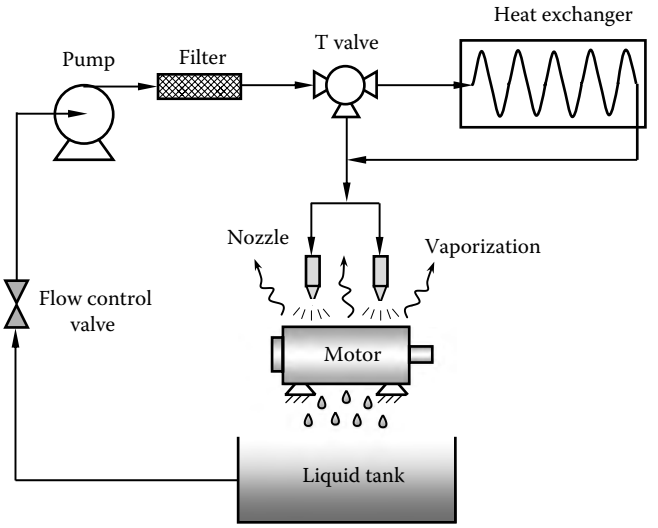


FIGURE 8.46
Spray cooling technique applied on the outer surface of a totally enclosed motor.

In a rotating machinery, an important characteristic of wall rotating–driven flows is the tendency of fluid with high angular momentum to be flung radially outward. For a motor, the rotor rotating-driven flow, usually referred to as the rotating pumping flow, plays an important role in cooling motor windings. Spray cooling occurs when liquid is forced through an array of tiny nozzles to be shattered into dispersive fine droplets. These droplets can spread on the surface and evaporate, removing large amounts of energy through the latent heat of evaporation in addition to substantial conventional effects. In a closed-loop system, the vapor is then condensed and pumped back to the nozzles. Due to its high efficiency, spray cooling can be used for high-heat-flux applications.

The heat transfer rate of spray cooling strongly depends on temperature difference between coolants and heated surfaces. At low temperature differences, heat transfer occurs primarily through single-phase convection. As the temperature difference increases, the evaporation on the liquid film surface, as well as boiling on the heated surface, can significantly augment the heat transfer rate.

Numerous spray cooling techniques have been developed and used in industrial applications. As a promising technique, the internal spray cooling technique has been proven as an effective way for electric machine cooling. The diagram of the spraying cooling system is shown in Figure 8.47. To avoid electric short circuits, dielectric liquids such as mineral oils are commonly selected as coolants. However, the selection of liquid coolant involves the trade-off between performance, functionality, cost, chemical stability, maintenance, safety, and environmental protection. The thermophysical properties of engine oil are listed in Table 8.5 as a function of temperature [8.50].

As can be seen in the figure, the cooling oil is pumped into the hollow shaft and ejected from the tiny nozzles uniformly distributed on the shaft circumference by means of the rotor centrifugal force. The droplet-like oil hits the motor end windings to carry heat away from them. The hot oil is then pumped back into the oil tank and cooled by an oil–water heat exchanger.

A CFD analysis has been carried out to predict the temperature distribution and flow field of the rotor assembly with a similar spray cooling system. An assembled view of a casted rotor assembly is shown in Figure 8.48. The rotor assembly consists of the rotor core, shaft, rotor end rings, and rear endbell. During motor operation, heat is generated in the rotor assembly due to three types of power losses: (1) mechanical power losses such as frictional loss of bearings and windage loss, (2) eddy current and hysteresis losses in the rotor core and other components, and (3) stray losses in the rotor end rings. The cooling oil enters form the rear endbell into the axial cooling duct at the shaft center, turns 90° to get into the four small radial holes near the rotor axial center, then ejects from the nozzles into the cooling ducts between the shaft and rotor core, and finally sprays from the end rings into the interior of the motor.

The mass flow field of the rotor is presented in Figure 8.49. For a one-inlet and multiple-outlet system under a steady-state condition, the inlet mass flowrate must equal to the total mass flowrate from all outlets:

$$\dot{m}_{in} = \sum_i \dot{m}_{out,i} = \sum_i \rho A_{out,i} u_{out,i} \tag{8.16}$$

where

$\dot{m}_{in}$ is the mass flowrate at the inlet

$\dot{m}_{out,i}$ is the mass flowrate at the ith outlet

$A_{out,i}$ and $u_{out,i}$ are the area and velocity at the ith outlet, respectively

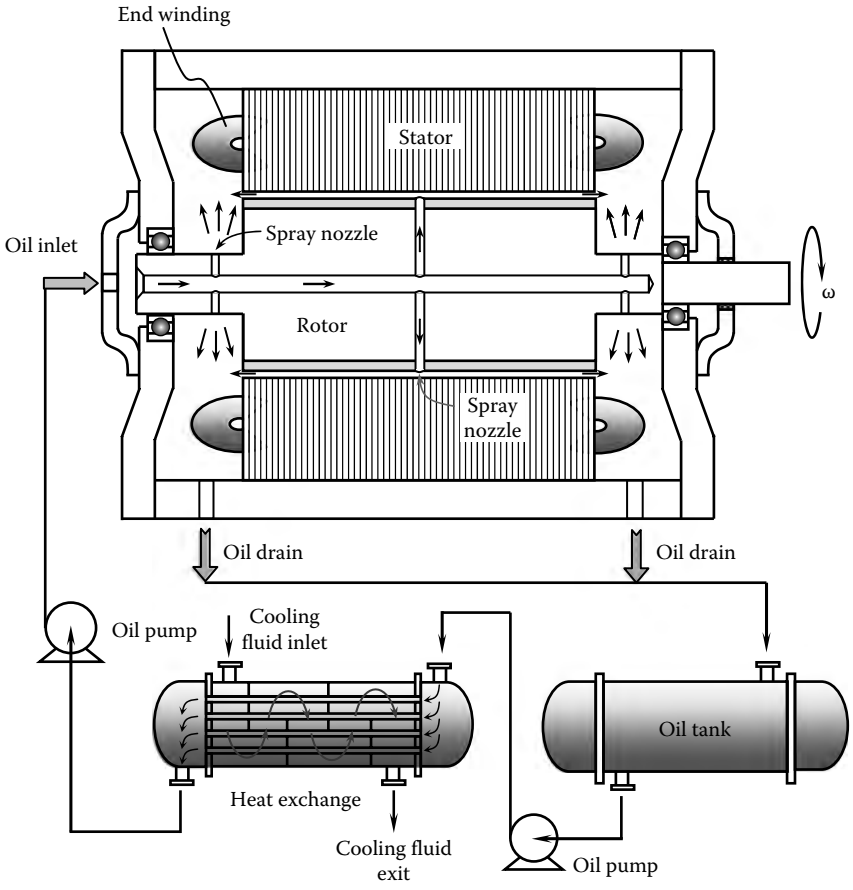


FIGURE 8.47
The internal spray cooling system by utilizing rotor centrifugal force.

TABLE 8.5

Thermophysical Properties of Engine Oil

Temperature (K)	Density (kg/m³)	Specific Heat (kJ/kg-K)	Dynamic Viscosity (Pa-s)	Thermal Conductivity (W/m-K)	Thermal Diffusivity ($m^2/s) \times 10^{-8}$	Prandtl Number	Volume Expansion Coefficient $(1/K) \times 10^{-4}$
260	908	1.76	12.23	0.149	9.324	144,500	7
280	896	1.83	2.17	0.146	8.904	27,200	7
300	884	1.91	0.486	0.144	8.529	6,450	7
320	872	1.99	0.141	0.141	8.125	1,990	7
340	860	2.08	0.053	0.139	7.771	795	7
360	848	2.16	0.025	0.137	7.479	395	7
380	836	2.25	0.014	0.136	7.230	230	7
400	824	2.34	0.009	0.134	6.950	155	7

Hesai Exhibit 1031, page 489

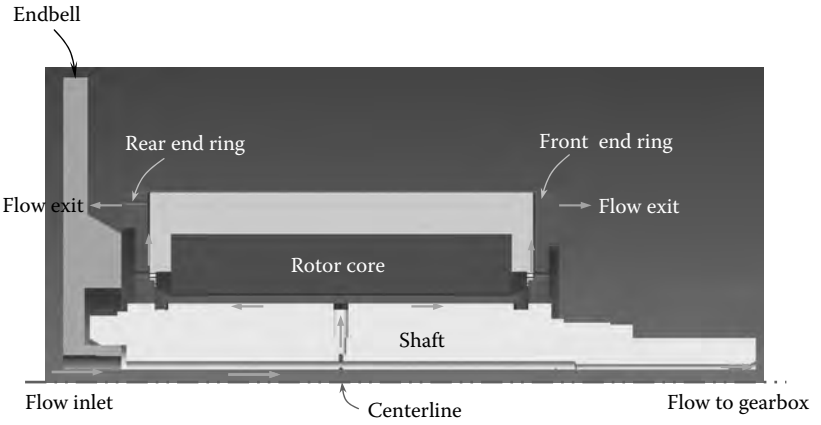


FIGURE 8.48
Cooling flow paths in the rotor assembly (only a half model shown here).

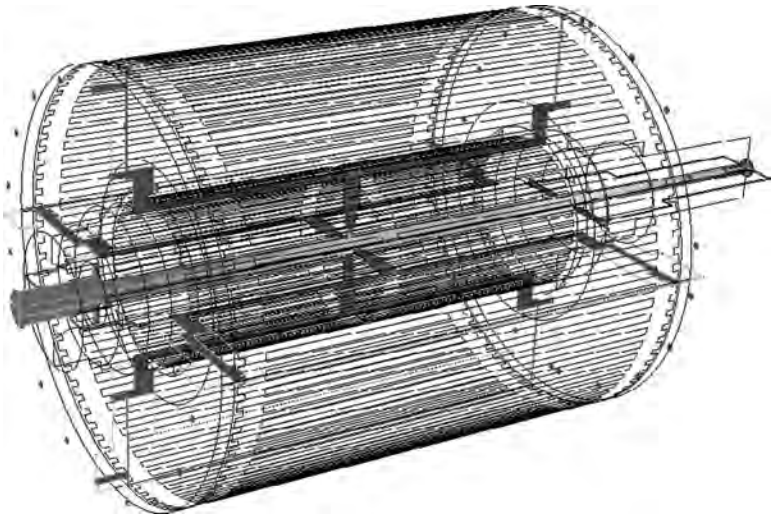


FIGURE 8.49
Cooling flow paths inside the rotor assembly.

The mass flowrate ratio at the *i*th outlet is defined as the ratio of the mass flowrate at that outlet to the total mass flowrate, that is,

$$\eta_i = \frac{\dot{m}_{out,i}}{\dot{m}_{in}} \tag{8.17}$$

Obviously, $\sum_i \eta_i = 1$.

The determination of the η value at each outlet is critical for obtaining relative uniform temperature distribution and elimination of hot spots. It requires distributing the cooling flowrate at each exit appropriately. For this purpose, the dimensions of nozzle and cooling ducts must be iteratively adjusted according to the CFD results.

Hesai Exhibit 1031, page 490

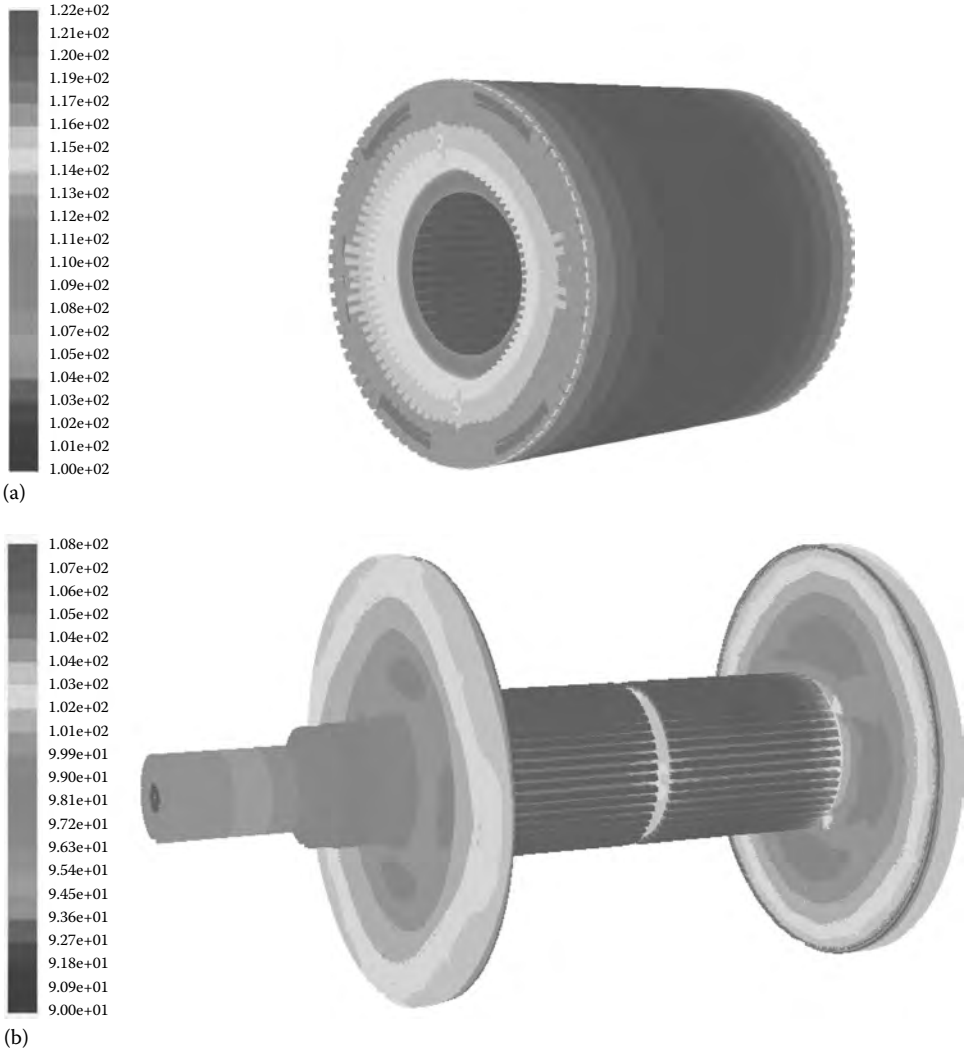


FIGURE 8.50
Temperature distributions in the rotor assembly: (a) rotor core and (b) shaft and end rings, showing the temperature generally decreasing from the center to the two rotor ends symmetrically.

The temperature distribution in the rotor assembly is demonstrated in Figure 8.50, showing that the highest temperature occurs on the outer rotor surface near the rotor axial center and the temperature generally decreases from the axial center to the rotor's two ends.

More detailed information about the temperature distribution inside the rotor assembly is given in Figure 8.51. This information can be adopted to help optimize the rotor cooling design for gaining a higher cooling efficiency and eliminating hot spots in the rotor.

8.5.2.3 Liquid Jet Impingement Cooling

Liquid jet impingement is an attractive cooling option in a number of industries over the past decades. In this cooling method, a liquid is delivered through an array of tiny nozzles

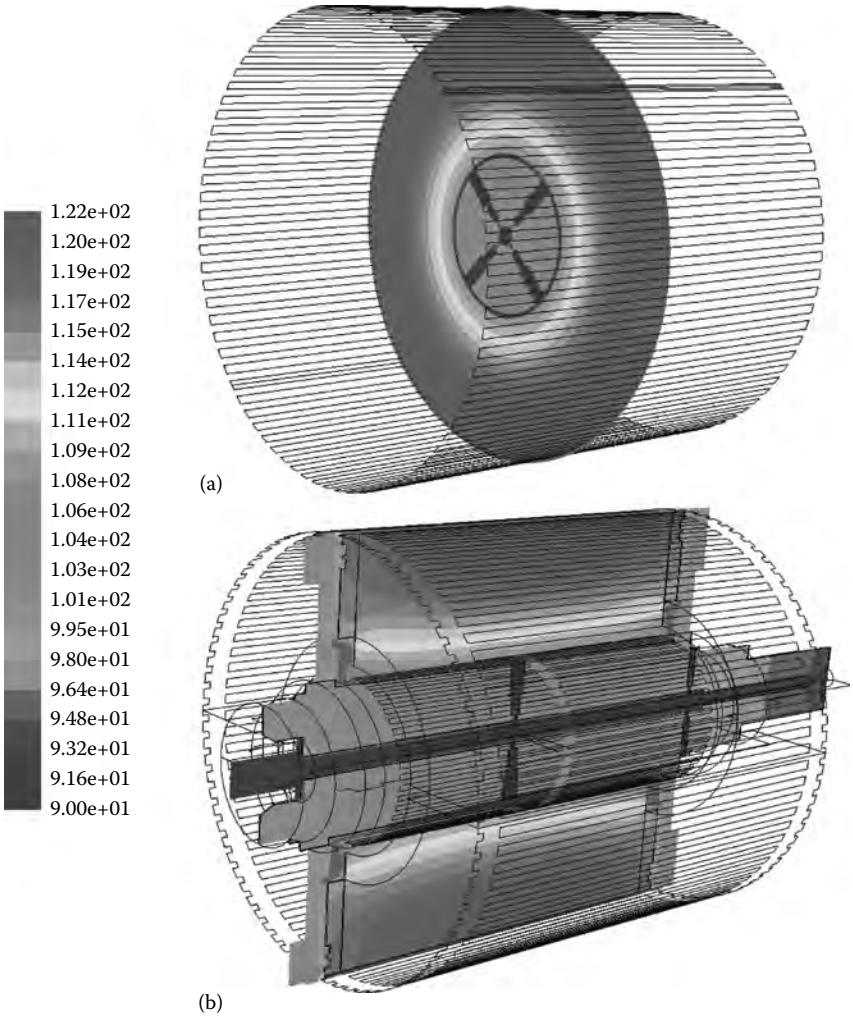


FIGURE 8.51
Temperature distributions in the rotor assembly: (a) the cross-sectional area perpendicular to the rotor axis, the highest temperature occurring at the rotor outer surface, and (b) the cross-sectional area through the rotor axial centerline.

to hit motor windings and other heat source components. For direct liquid jet impingement cooling, a dielectric liquid is required in such an approach to prevent an electrical short from the motor windings and leads unless they are properly protected. For indirect liquid cooling, water may be adopted as the best coolant due to its superior thermophysical properties. Liquid jet impingement cooling offers lower thermal resistance and improves the temperature uniformity.

8.5.3 Liquid Immersion Cooling

Liquid immersion cooling becomes very popular today in electronic cooling applications. The first computer designed to be directly cooled by a liquid was Cray-2 supercomputer,

back to 1985. Since it used a dielectric coolant that does not conduct electricity, the supercomputer could be submersed in the coolant without causing any short circuit. In addition, because the dielectric coolant has much better thermophysical properties than air, it offers the prospect of dramatically more energy-efficient cooling than those in the conventional air cooling approaches. Besides high cooling efficiency, the potential benefits of liquid immersion cooling include the high cooling capacity, temperature uniformity, reliability, and not needing to power internal fans/pumps to assist fluid cooling flows. Compared with air cooling techniques, liquid immersion cooling can help design much more dense devices without the need for flow aisles.

This cooling mode can be successfully transplanted into motor cooling. Some special dielectric coolants such as FC are particularly suitable for motor immersion cooling due to their low toxicity, stable viscosity, UV resistance, and high dielectric strength. In order to control the friction losses, this method may be limited to low rotating speed motors.

8.5.4 Indirect Liquid Cooling Techniques

In an indirect liquid cooling scheme, a liquid coolant does not come into contact with cooling components. Instead, cooling is implemented by using heat exchangers, cold plates, coolers, and other cooling products. In such cases, a good thermal conduction path is needed from heat sources (winding, core, etc.) to the coolant. Since there is no direct contact with the electric windings and connectors, water or water-based antifreezes (e.g., ethylene glycol/water, propylene glycol/water, methanol/water, and ethanol/water), even some liquid metals, can be used as the coolant, taking advantage of their superior thermophysical properties and relatively low costs. This heat transfer mode is in fact encountered in a wide spectrum of engineering systems including energy–power conversion, material procession, internal combustion, electric and electronic devices, and manufacturing process.

8.5.4.1 Indirect Liquid Cooling via Cold Plates Attached to Motor Walls

Cold plates are typically made of copper or aluminum tubes that are bonded into aluminum plates to optimize thermal conductance. Liquid-cooled cold plates may be mounted on motor external surfaces to provide very effective contact cooling for various motor applications having different thermal requirements (Figure 8.52). The dimensions of a cold plate are designed to fit the size of a motor. In practice, cold plates use either water or water-based antifreezes as a coolant. As an example, thermophysical properties of water/glycol mixtures with different mixing ratios are given in Table 8.6.

8.5.4.2 Indirect Liquid Cooling via Helical Copper Pipes Casted in Motor Housing

This cooling method uses helical copper pipes that are directly casted inside the motor housing (Figure 8.53). Water flowing through these helical copper pipes ensures more uniform cooling of the stator due to the generation of the secondary flows in these curved pipes by the centrifugal effect. The advantages of this cooling method are as follows: (1) Because the copper pipes are perfectly integrated with the housing by casting, the thermal contact resistance between the copper pipes and the housing is minimized. (2) There is no leakage from the copper pipes. However, the fabrication cost for such a cooling system is usually higher than other cooling methods, as listed in the following.

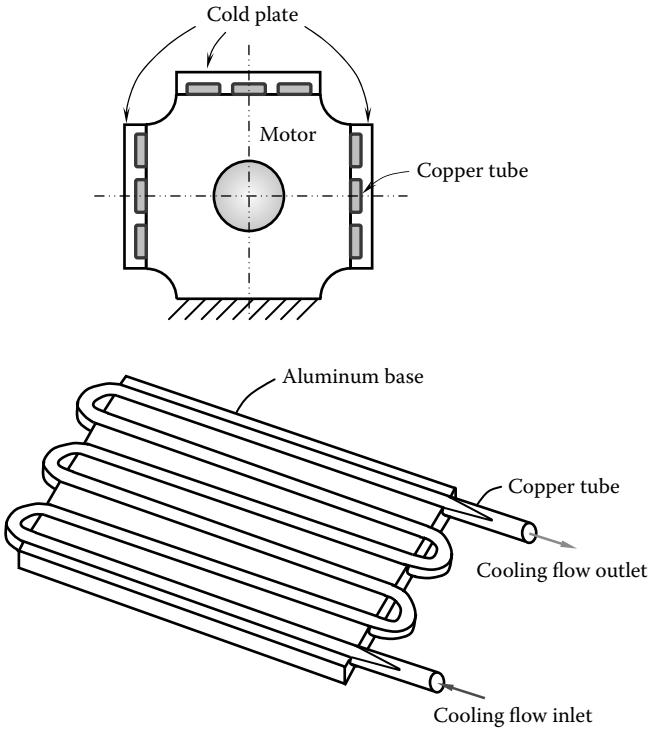


FIGURE 8.52
Cold plates used for motor cooling, mounted on the three sides of the motor.

The thermal energy dissipated to the coolant Q can be determined as

$$Q = \dot{m} C_p \Delta T \tag{8.18}$$

where $\Delta T = T_o - T_i$, the temperature difference between the outlet and inlet of the coolant.

Several correlations are available in the literature for calculating the friction factor as flows through helical pipes. Ito [8.51] proposed a model of friction factor for a turbulent flow:

$$f_c = 0.00725 + 0.076\left[Re\left(\frac{r}{0.5d} \right)^{-2} \right]^{-0.25} \left(\frac{r}{0.5d} \right)^{-0.5} \tag{8.19}$$

which is valid for

$$0.034 < Re\left(\frac{r}{0.5d} \right)^{-2} < 300$$

Srinivasan et al. [8.52] developed a model for flows in a helical pipe:

$$f_c = 0.084\left[Re\left(\frac{r}{0.5d} \right)^{-2} \right]^{-0.2} \left(\frac{r}{0.5d} \right)^{-0.5} \tag{8.20}$$

TABLE 8.6

Thermophysical Properties of Water/Glycol Mixtures with Different Mixing Ratios

Substance	Volume in Mixture (%)	Minimal Working Temperature (°C)	Temperature (°C)	Density (kg/m^3)	Specific Heat (kJ/kg-K)	Thermal Conductivity (W/m-K)	Dynamic Viscosity × 10^3 ($W\text{-}s/m^2$)	Kinematic Viscosity × 10^6 (m^2/s)
Monoethylene glycol $[C_2H_4(OH)_2]$	20	−10	−10	1038	3.85	0.498	5.19	5.0
			0	1036	3.87	0.500	3.11	3.0
			20	1030	3.90	0.512	1.65	1.6
			40	1022	3.93	0.521	1.02	1.0
			60	1014	3.96	0.531	0.71	0.7
			80	1006	3.99	0.540	0.523	0.52
			100	997	4.02	0.550	0.409	0.41
	34	−20	−20	1069	3.51	0.462	11.76	11.0
			0	1063	3.56	0.466	4.89	4.6
			20	1055	3.62	0.470	2.32	2.2
			40	1044	3.68	0.473	1.57	1.5
			60	1033	3.73	0.475	1.01	0.98
			80	1022	3.78	0.478	0.695	0.68
			100	1010	3.84	0.480	0.515	0.51
	52	−40	−40	1108	3.04	0.416	110.8	100
			−20	1100	3.11	0.409	27.5	25
			0	1092	3.19	0.405	10.37	9.5
			20	1082	3.26	0.402	4.87	4.5
			40	1069	3.34	0.398	2.57	2.4
			60	1057	3.41	0.394	1.59	1.5
			80	1045	3.49	0.390	1.05	1.0
			100	1032	3.56	0.385	0.722	0.7

Hesai Exhibit 1031, page 495

1,2-Propylenglycol $[C_3H_6(OH)_2]$	25	−10	−10	1032	3.93	0.466	10.22	9.9
			0	1030	3.95	0.470	6.18	6.0
			20	1024	3.98	0.478	2.86	2.8
			40	1016	4.00	0.491	1.42	1.4
			60	1003	4.03	0.505	0.903	0.90
			80	986	4.05	0.519	0.671	0.68
			100	979	4.08	0.533	0.509	0.52
	38	−20	−20	1050	3.68	0.420	47.25	45
			0	1045	3.72	0.425	12.54	12
			20	1036	3.77	0.429	4.56	4.4
			40	1025	3.82	0.433	2.26	2.2
			60	1012	3.88	0.437	1.32	1.3
			80	997	3.94	0.441	0.897	0.9
			100	982	4.00	0.445	0.687	0.7
	47	−30	−30	1066	3.45	0.397	160	150
			−20	1062	3.49	0.396	74.3	70
			−10	1058	3.52	0.395	31.74	30
			0	1054	3.56	0.395	18.97	18
			20	1044	3.62	0.394	6.264	6
			40	1030	3.69	0.393	2.987	2.9
			60	1015	3.76	0.392	1.624	1.6
			80	999	3.82	0.391	1.10	1.1
			100	984	3.89	0.390	0.807	0.82

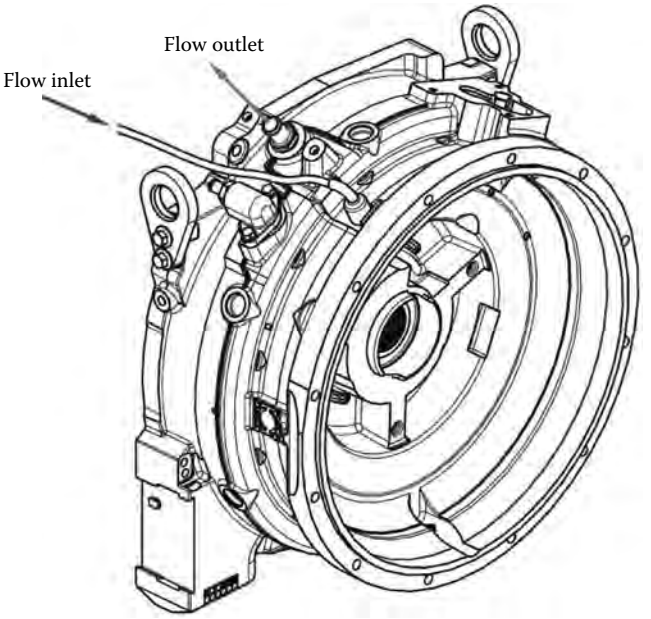


FIGURE 8.53
Assembled stator with a dual-cooling pipe casted inside the motor housing.

which is valid for

$$Re\left(\frac{r}{0.5d}\right)^{-2} < 700 \quad \text{and} \quad 7 < \frac{r}{0.5d} < 10^4$$

With the friction factor, the pressure drop across a helical pipe can be determined as

$$\Delta p = \pi d L \tau_w A \tag{8.21}$$

To reduce the pressure drop across the cooling pipe, a helical dual-cooling pipe may be adopted in the design. As shown in Figure 8.54, the adaptor is designed to distribute the coolant with an equal flowrate into the two pipes.

8.5.4.3 Indirect Liquid Cooling with Cooling Channels on Casted Motor Housing

Figure 8.55 provides another cooling scheme in which cooling water flows through sealed cooling channels. The cooling channels are formed by integrating the casted motor housing and the channel cover. Because water is distributed more uniformly into all channels, the pressure drop across the channels is much lower than that across helical copper pipes. To prevent water leakage, an interference fit (press fit or shrink fit) must be adopted between the housing and the channel cover.

8.5.4.4 Indirect Liquid Cooling via Copper Pipe in Spacer

In this cooling approach, a motor is cooled by water passing through a copper pipe in a special designed spacer near the front endbell (Figure 8.56). The stator winding is cooled by

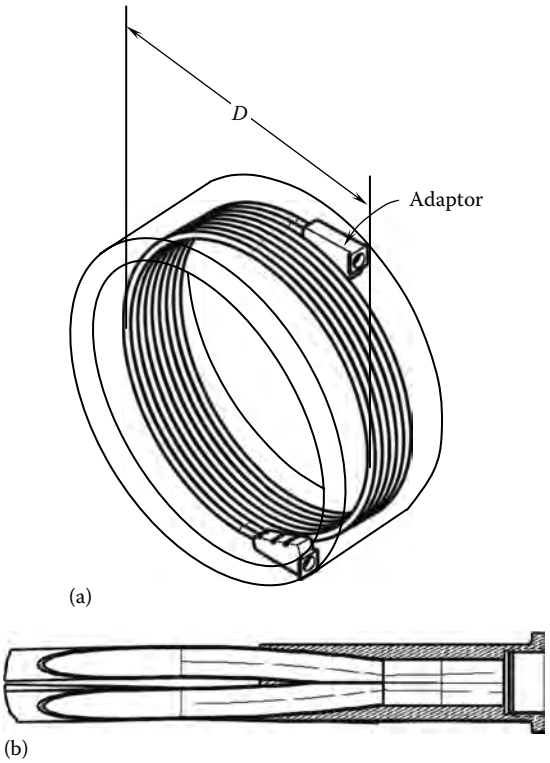


FIGURE 8.54
Components of the indirect liquid cooling system: (a) the helical dual-cooling pipe, casted directly into the motor housing for reducing thermal resistance, and (b) the adaptor at the flow inlet and outlet to connect with the helical dual-cooling pipe.

transferring heat from the winding to the motor housing and spacer, and then to cooling water. However, the present arrangement of the cooling pipe may cause nonuniform cooling on the motor, resulting in a temperature gradient along the motor axis. Furthermore, since the cooling pipe is close to the front bearing, it may shrink the outer race of the bearing to significantly reduce the bearing radial clearances, leading to the bearing lock or even failure.

8.5.4.5 Indirect Liquid Cooling through Stator Winding Slots

As discussed in Chapter 7, the biggest power loss is the resistive loss in copper windings. Therefore, in order to cool a motor more efficiently, a logical thinking is to cool these windings directly. An innovative approach was proposed by Tan et al. [8.53]. In their approach, water pipes are inserted in the stator slots (see Figure 8.57). In such a way, water flows through these intercoil pipes to carry away directly the heat generated in the winding due to the copper loss. Obviously, by this method, the thermal resistance is minimized compared with conventional cooling methods.

8.5.4.6 Indirect Liquid Cooling through Microscale Channels

With the rapid increase in power density of modern motion control systems, the traditional cooling methods such as conduction and natural/forces convection may

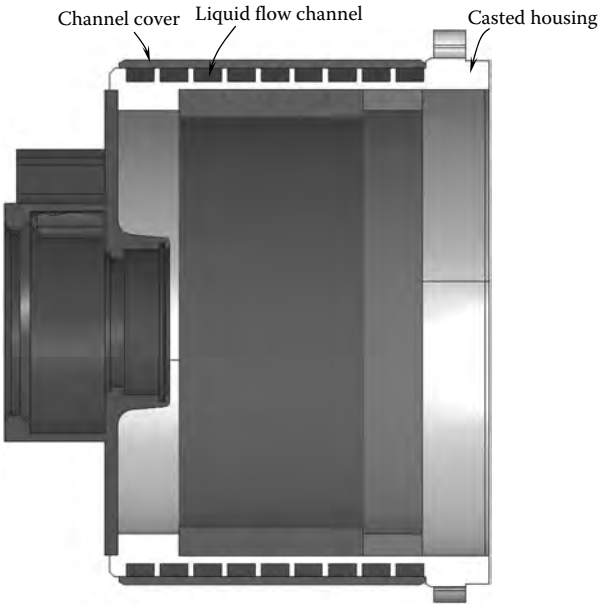


FIGURE 8.55
Liquid flow channels formed between the casted housing and the channel cover.



FIGURE 8.56
Indirect water cooling via copper pipe integrated in the spacer near the front endbell. (Courtesy of Kollmorgen Corporation.)

Hesai Exhibit 1031, page 499

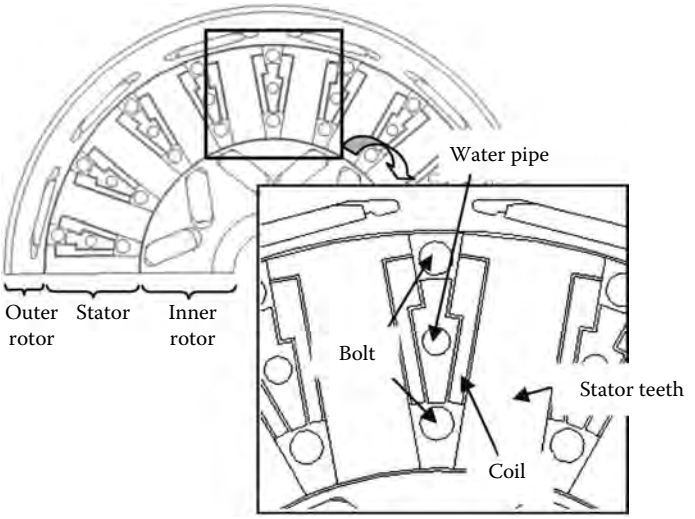


FIGURE 8.57
Water pipes inside stator slots for effectively cooling stator windings. (From Tan, N. et al., Cooling performance design for super motor and its experimental validation, *Proceedings of the 2005 International Power Electronic Conference*, pp. 1492–1488.)

be insufficient for these systems. The need for new technologies capable of dissipating high heat fluxes is of critical importance to motion industry. The use of pumped liquid through microscale channels is a promising technique in various industries for high-heat-flux applications.

As pointed out by Upadhya et al. [8.54], the performance of a liquid-cooled system depends on several factors, including (1) feature size of the microscale channel, (2) flowrate of the liquid through each channel, (3) surface area of radiator fins, and (4) air flow available for heat rejection.

8.6 Phase-Change Cooling Techniques

Phase-change cooling is an extremely effective way to cool energy systems. By accommodating large heat fluxes with relatively small driving temperature differences, heat in this approach is primarily removed through the vaporization of the working fluid utilizing the large latent heat capacity. As a consequence, the heat transfer augment is achieved and much less coolant is required as compared with single-phase cooling.

8.6.1 Cooling with Heat Pipes

Continuously increasing power densities of electric motors make using conventional conduction-based thermal management materials a problem, because heat dissipation requirements often exceed material conductivities. That makes heat pipes and other advanced thermal management materials ideal choices for direct conduction cooling.

During normal operation of an electric motor, the temperature distribution inside the motor is far from uniform. Hot spots may occur locally in some components that have higher power losses. In order to eliminate such hot spots, in addition to the overall cooling system, individual components may have their own cooling devices in place. The motor components that can be individually cooled include, but are not limited to, stator winding, bearings, electric connectors, and encoder. For the purpose of specific local cooling, the use of heat pipes may be more efficient and suitable.

Heat pipes are two-phase heat transfer devices with extremely high thermal conductivities. The world's first heat pipe was originally invented by Gaugler in 1944 and patented in 1946 [8.55]. However, it did not gain any significant attention within the heat transfer community until the space program resurrected the concept in the early 1960s [8.56]. Heat pipes rely on latent heat of evaporation of a working fluid (usually thermal refrigerant) to transmit heat from one hot point to another. Because it involves both boiling (or evaporation) and condensation, this cooling technique is generally categorized as a two-phase cooling technique. Based on its cooling mechanism, heat pipes can operate in any orientations. Furthermore, by carefully selecting coolants, heat pipes can be designed to operate over a broad range of temperatures. In motor cooling applications where it is typically desirable to maintain the motor temperature below 120°C–155°C, depending on the insulation class, copper/water heat pipes can be applied.

The thermal resistance represents the effectiveness of the heat pipe, which is defined as

$$R_{th} = \frac{T_e - T_c}{Q}$$

where

Q is the heat input

T_e and T_c are average evaporator and condenser surface temperatures, respectively

An experimental investigation of heat transfer in heat pipes with three working fluids was conducted by Mozumder et al. [8.57]. Their study reveals that the dominating parameters for heat transfer in heat pipes are average evaporator temperature T_e, saturated boiling temperature of working fluid T_{sat}, latent heat of vaporization h_{fg}, specific heat of working fluid $c_{p,l}$, and the fluid fill ratio F. The correlation for the heat transfer coefficient is given as

$$Q = hA_e(T_{sat} - T)_e = 39(Ja)^{4/5} F^{1/4} \left(\frac{T_{sat}}{T_e} \right)^5 \tag{8.22}$$

where

h is the overall heat transfer coefficient in W/(m^2-K)

A_e is the surface area at the heat pipe evaporation region in m^2

Jakob number is defined as

$$Ja = \frac{C_{p,l}(T_e - T_{sat})}{h_{fg}} \tag{8.23}$$

It must be noted that Q in Equation 8.22 has a unit of watt and the right-hand side is dimensionless.

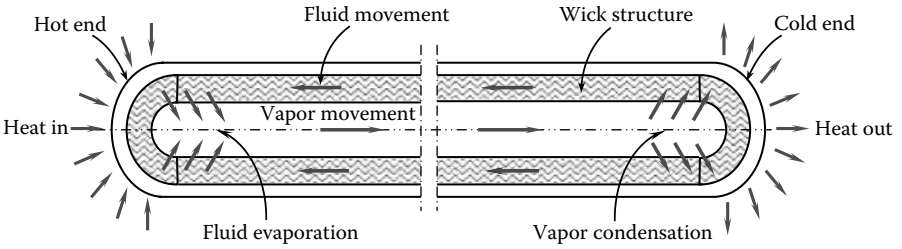


FIGURE 8.58
Performing principle of a heat pipe. It relies on both boiling (or evaporation) and condensation of working fluid.

Thus, the overall heat transfer coefficient h is given as

$$h = \frac{Q}{A_e(T_{sat} - T_e)} \tag{8.24}$$

where A_e is the surface area at the heat pipe evaporation region.

The correlation listed previously for predicting heat transfer coefficient fairly agrees with the experimental results. The investigation also shows that 85% fill ratio can be regarded as an optimum value for a heat pipe. Later, Manimaran et al. [8.58] found that the lowest thermal resistance is obtained when the heat pipe is operated at 75% fill ratio.

As illustrated in Figure 8.58, at the hot end of the heat pipe, the fluid evaporates into vapor, thereby absorbing the heat from the heat source. As the vapor migrates to the cold end through the central cavity, it condenses back into the liquid state to dissipate the heat to the environment. Then, the liquid returns to the hot end via the wick structure due to the capillary effect. With the repeat of the process, heat is continually transported from the hot end to the cold end. This indicates that heat pipes operate independently of gravity and can be effectively used in gravity-free conditions.

Since heat pipes involve the phase change of the working fluid, they have much higher effective thermal conductivity than solid materials. For instance, in some applications and orientations, heat pipes have a thermal conductivity in exceeds of 20,000 times than that of a solid copper bar of the same geometry. Also, heat pipes require no maintenance due to the lack of mechanical moving parts.

8.6.2 Cooling with Vapor Chambers

Vapor chambers can offer far superior thermal performance than traditional solid metal heat spreaders at the same thermal load and device size. With much lower thermal resistance, vapor chambers provide evenly spread temperatures on all of their surfaces, regardless of the location and thermal density of the heat source. Therefore, vapor chambers are able to deal with high-heat-flux motor components, extending the life of components and motors.

During the last several decades, boiling heat transfer techniques have been developed for cooling high-power systems, leading to the design of highly efficient vapor chambers. Since nucleate boiling attains heat flux during phase change, it has been an attractive design choice for electric motors and generators.

Although a vapor chamber looks like a heat pipe in its appearance, it has a fundamentally different cooling mechanism than that of the heat pipe. In a vapor chamber, the

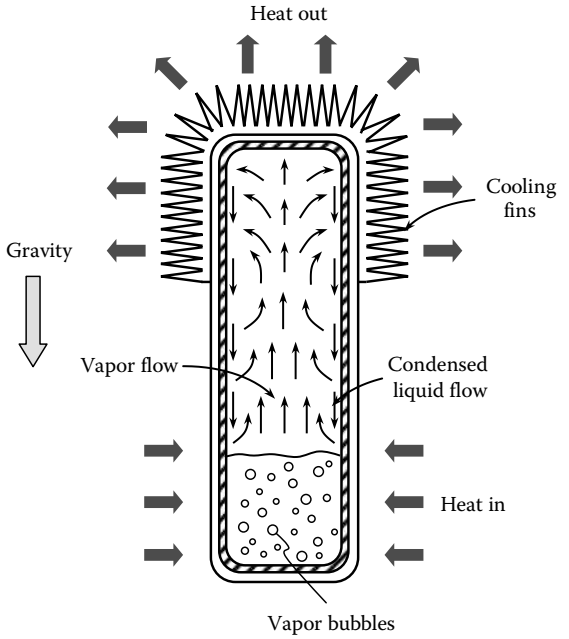


FIGURE 8.59
Boiling and condensation of coolant in a vapor chamber. The movements of vapor and liquid coolant rely on gravitational force. Fins are added on the vapor chamber at the condensation end to enhance heat dissipation.

movements of vapor and liquid coolant rely on gravitational force. This indicates that the orientation of a vapor chamber has a strong impact on its performance. To increase the cooling efficiency of vapor chamber, cooling fins may be added at the condensation end of the vapor chamber to improve the heat dissipation from it to the environment (Figure 8.59). As a result, the heat transfer enhancement can be achieved not only by increasing convective heat transfer surface area but also by creating local flow turbulence, both resulting in a higher heat convective transfer coefficient.

According to these superior thermal characteristics of heat pipes, motors with heat pipes can be made for higher power density and higher efficiency, allowing more current passing through motor windings. Heat pipes may be used to cool different motor components. As shown in Figure 8.60 [8.59], a set of heat pipes is inserted in the stator slots to directly cool the windings. Because most of the heat in an electric motor is generated in the motor winding, putting the heat pipes in close proximity to the copper winding will make the heat transfer more efficient. In addition, another set of heat pipes is incorporated into the stator core to absorb heat generated in the core due to the eddy-current losses. A perspective view of a motor with these two sets of heat pipes is provided in Figure 8.61.

Similarly, heat pipes may be also implemented in rotors to assist in dissipating heat. Referring to Figure 8.62, heat pipes can be inserted into rotor conducting bars.

8.6.3 Evaporative Cooling

A compelling evaporative cooling system has been developed to address challenges of cooling high-power devices [8.60]. This cooling system employs a noncorrosive, nonconductive

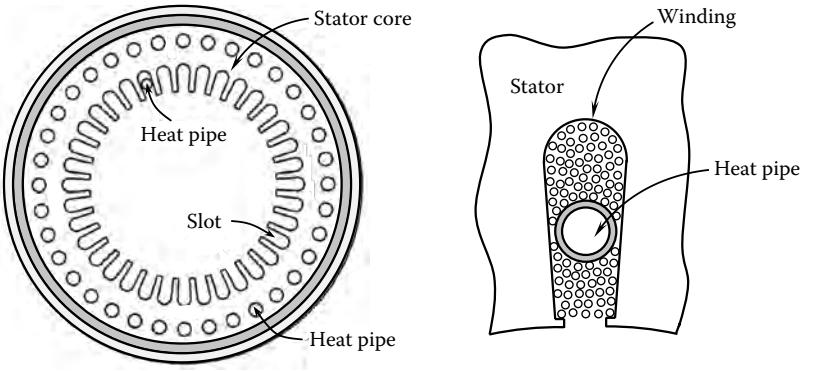


FIGURE 8.60
Heat pipes inserted in stator slots and stator core (U.S. Patent 7,569,955) [8.59]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

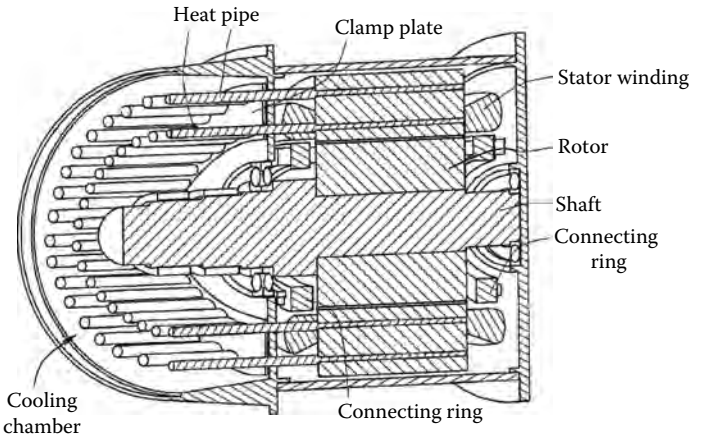


FIGURE 8.61
Cross-sectional view of motor cooling with heat pipes (U.S. Patent 7,569,955) [8.59]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

refrigerant that evaporates on contact with hot surfaces of the high-power device, in a small, lightweight, and highly efficient closed-loop system. By taking advantage of the latent heat absorbed during phase change of the refrigerant from liquid to vapor, as well as the latent heat released during phase change from vapor to liquid, this system has a very high cooling efficiency.

As shown in Figure 8.63, the refrigerant is pumped into one or more cold plates that are optimized to acquire the heat from the device. The refrigerant vaporizes to maintain a cool uniform temperature on the device surfaces. The resulting refrigerant is then pumped to a heat exchanger where it rejects the heat to the ambient and condenses back into the liquid, completing the cycle.

The comparisons of thermal performance of the evaporative cooling system with various cold plates, as well as the air-cooled system, are plotted in Figure 8.64. It can be seen that

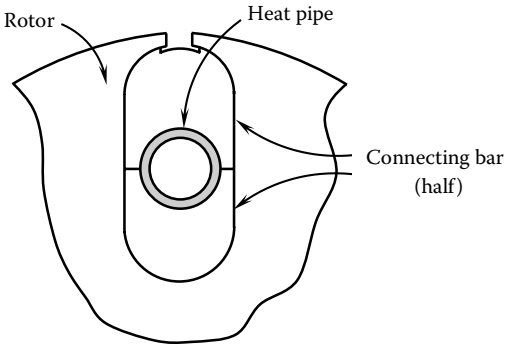


FIGURE 8.62
Heat pipe inside the rotor connection bar (U.S. Patent 7,569,955) [8.59]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

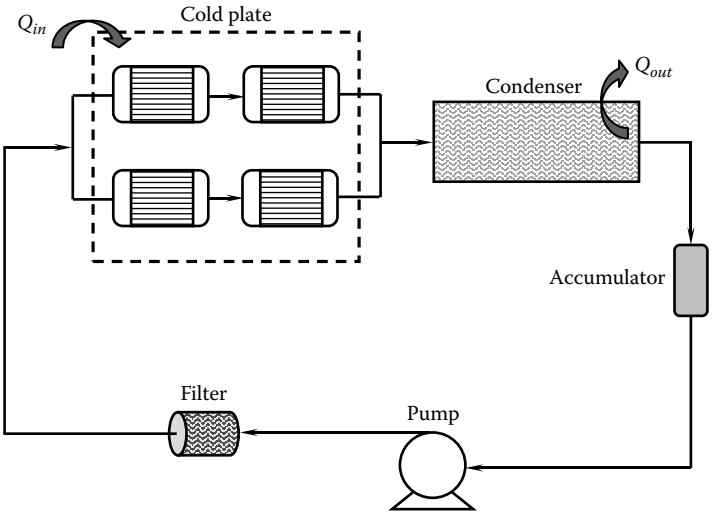


FIGURE 8.63
Evaporative cooling system.

under identical ambient conditions, the air-cooled system has the highest thermal resistance (0.094°C/W) and the evaporative cooling system (two-phase evaporative cooling) has the lowest (0.008°C/W). This indicates that by taking advantage of the highly efficient evaporation process that occurs when the liquid refrigerant changes phase to vapor, up to 11.8 times the amount of heat can be removed for the same given temperature difference by comparing with the air-cooled system. Therefore, the two-phase evaporative approach can increase the system power density (or reduce the system volume for the same power output) and improve performance reliability due to the elimination of the safety and maintenance issues associated with water cooling. In addition, the isothermal nature of the two-phase cooling system reduces thermal cycling.

By changing the material of cold plates from aluminum to copper, the thermal resistance can be lowered approximately 27%.

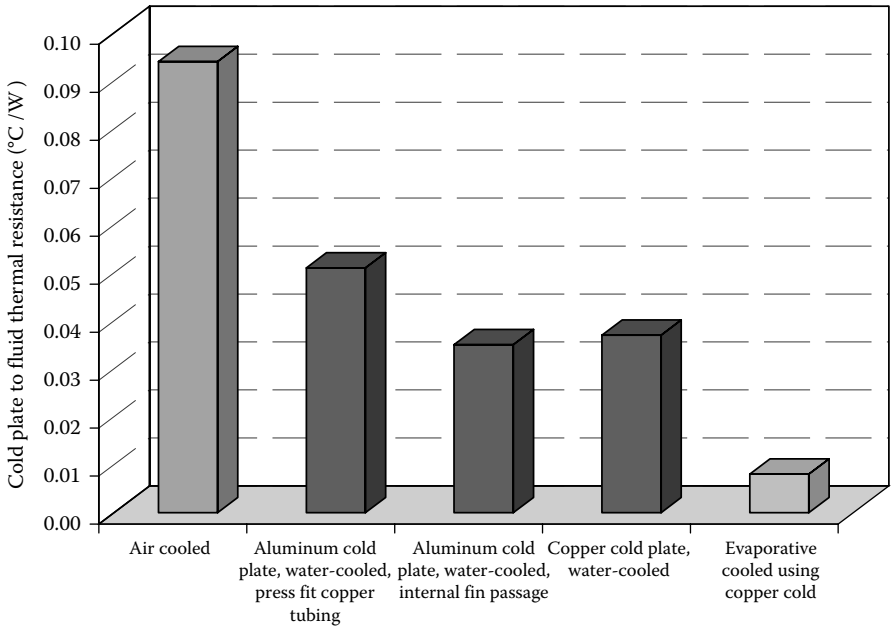


FIGURE 8.64
Comparisons of thermal performance of evaporative cooling system with various cold plates and air-cooled system.

8.6.4 Mist Cooling

Mist cooling continues to be a viable solution for a wide range of residential and industrial cooling applications for creating safer and more comfortable working environment. Mist cooling has demonstrated to remove high heat energy with relatively low coolant flowrates. In a mist cooling system, a high-pressure pump pushes water through a plurality of mist nozzles, producing microscopic droplets of water. As fine water mists hit hot air or surfaces, they evaporate to cool the surrounding space.

Totally enclosed nonventilated motors may be suitable to use the mist cooling technique. For this type of motors, all motor components are contained within a motor frame to prevent moisture, duct, and other harmful contaminates into the motor. To effectively cool this type of motors, very fine water mists are used to directly hit the outer surface of the motor. Thus, heat generated by the motor is absorbed by evaporation of water mists. Furthermore, the mixing flow of mists and air also has a favorable contribution in motor convection cooling.

8.7 Radiative Heat Transfer

Radiative heat transfer of rotating electric machines involves thermal radiation between the stator and rotor and between the stator (or rotor for the outer rotor machines) and the environment. According to the Stefan–Boltzmann law, the radiative energy E_r from a hot object at temperature T_h to its cooler surroundings at temperature T_c is given as

$$E_r = e\sigma A(T_h^4 - T_c^4) \tag{8.25}$$

where
e is the emissivity ($e = 1$ for a black body) of the object
A is the radiating area
σ is Stefan's constant ($\sigma = 5.6703 \times 10^{-8}$ W/m^2-K^4)

The impact of radiation heat transfer on electric motor cooling has been studied numerically by Chirilă et al. [8.61]. Their results show that for an internally air-cooled motor, the average temperature of the motor housing is lower when the radiation is considered for all inlet flow velocities. The maximum temperature difference obtained is about 9°C, indicating that the radiation is important for cooling the motor house at low inlet air velocities. Increase in the cooling air velocity can always significantly lower the housing temperature, from 118°C at $u = 2$ m/s to less than 80°C at $u = 8$ m/s. However, with the increase in the flow velocity, the temperature difference due to the radiation effect becomes very small, from about 9°C at $u = 2$ m/s to about 2°C at $u = 8$ m/s.

In most motor cooling schemes, a cooling system is disposed within the stator. As the result, the rotor temperatures are generally higher than those of stators. The heat generated by the rotor is thus dissipated by forced convection due to its rotation, conduction via bearings and radiation emitted to the motor housing and stationary components surrounding the rotor.

The radiation resistance between a motor and its surrounding environment depends both on the geometrical and thermal characteristics of the motor and the characteristics of the environment. This resistance is much higher than the forced convection resistance and thus is often neglected. For example, for the motors in refrigeration compressors, heat radiation is so weak that it can be neglected [8.62]. However, in some situations, particularly when totally enclosed fan-cooled (TEFC) motors are used in variable speed drives, radiation heat transfer can play a nonnegligible role at low speeds. It is also true for motors that use noncontact bearings. For these situations, radiation must be taken into account.

For outdoor electric motors, it is important in the design of a motor cooling system to consider not only the heat dissipation from the motor but also the heat from sunlight. In some circumstances, the solar loading effect could be substantial, depending on the motor size and its orientation toward the sun.

8.8 Other Advanced State-of-the-Art Cooling Methods

8.8.1 Microchannel Cooling Systems

Increasing heat densities in motion control systems cause high heat fluxes at both the component and system levels. The advancement of high-performance microchannel cooling has stimulated interest in this technique to substantially improve thermal energy removal capability. Though the microscale regime is not strictly defined, it is accepted that the characteristic length scale of microchannel ranges from a micrometer (10^{-6} m) to approximately 0.08 mm (8×10^{-5} m). Microchannels can provide a much larger contact area with the cooling fluid per unit volume and thus much higher heat transfer coefficients than conventional channels under similar thermohydraulic conditions.

According to previous studies, the microchannel cooling technique can remove heat densities up to 10 MW/m^2 [8.63].

In this technique, a cooling fluid is forced to pass through a microchannel heat sink by an external microscale pump for achieving very high heat removal rates. Because of its high heat removal capability, the researchers in Oregon Stator University are currently working with a motor manufacturer to design microchannel cooling systems in electric motors [8.64]. However, before this technique is successfully transplanted from electronic device cooling to high-power-density motor cooling, several difficulties must be overcome. Firstly, high heat-removing capability of a microchannel system is always associated with high pressure drop across the microchannels. Thus, it requires high power for driving cooling fluid through the microchannels. Secondly, due to the complex geometries of motor components, it is a challenge to achieve uniform cooling of the motor. Thirdly, heat transfer processes in microchannel are somewhat different from those in ordinary size channels. The capillarity of fluid in microchannels has a strong influence on flow patterns and heat transfer rates. This mechanism must be fully understood. Finally, it is noted that two-phase flows can be quite different from single-phase flows in microchannels. It is risky to extrapolate macroscale two-phase flow maps, flow boiling heat transfer methods, and pressure drop models to microchannels. More research is necessary to understand two-phase microchannel flows and heat transfer mechanisms.

8.8.2 Metal Foams

One of the principles of heat transfer enhancement is to extend heat dissipation surfaces. A typical example is the application of a variety of fins in heat exchangers. As the promising alternative, metal foams have been used for enhancing heat transfer in a variety of engineering applications. With a large number of open cell foams, metal foams have shown higher heat transfer enhancement than pin-fins [8.65]. Open cell foams consist of cells that are all interconnected, allowing a fluid to pass through them and thus have unique beneficial characteristics, including excellent surface-to-volume ratio (from 500 to over 10,000 m^2/m^3), high thermal conductivity, high fluid permeability and mixing, low flow resistance, low density, good thermal shock resistance, high temperature tolerance, excellent noise attenuation, corrosion resistance and other favorable properties [8.66–8.68]. It was reported that with the same pressure drop, the performance of the metal foam exchanger is superior when compared to that of conventional finned designs [8.69,8.70]. T'Joen et al. [8.71] have experimentally investigated the impacts of various parameters of metal foams on the thermohydraulic performance of a metal foam heat exchanger. Their results indicate that in all cases, heat transfer is greatly enhanced by coating metal foam on metal tubes. A good metallic bonding between the foam and the tube can significantly reduce the thermal resistance. A comparison of the performance of foam-covered tube with different foam heights has clearly showed that increasing the foam height reduces convective resistance. This technique is suitable for motor cooling by coating metal foams on the motor external surfaces, as described in Figure 8.65.

Metal foams can be fabricated from many different manufacturing processes, including investment casting, chemical vapor decomposition, electroforming, and metallic sintering, to name a few. Each fabrication process has its pros and cons and is used for certain applications. A comprehensive review of various metal foam fabrication processes was provided by Banhart [8.72].

As an indirect method, investment casting uses space-holding filler materials or melting of powder compacts that contain a blowing agent to produce cellular metal foams.



FIGURE 8.65
Enhanced motor cooling by coating metal foam with microporous structure on motor external surface. (Courtesy of Ghent University, Gent, Belgium.)

In a chemical vapor decomposition process, metal foams are made from gaseous metal. The condensed metal vapor in a vacuum chamber coats the surface of the polymer precursor to generate a film of metal foam.

Electroforming can generate a thin porous layer on a metal skin through an electrodeposition process. The metallic sintering process consists of various steps, including powder fractioning and preparation, compaction or molding, and sintering. The achievable porosity can range between 20% and 80%. In addition, using metal fibers to replace powders sheds new light on the fabrication of porous metal structures since a variety of metals can be produced as fibers.

8.8.3 Heat Transfer Enhancement with Nanotechnology

Nanotechnology deals with the control of matter at dimensions between approximately 1 and 100 nanometers (nm). The emergence and development of nanotechnology has affected/altered many aspects of engineering designs and manufacturing. It has the potential to create new materials and devices with a vast range of applications and thus has been considered as a revolutionary technology. As one of the cutting edge in innovative cooling methods, it is expected that nanotechnology can feature innovative cooling solution for electric machine in the near future.

Hesai Exhibit 1031, page 509

8.8.3.1 Nanofluid

Due to the recent developments in nanotechnology, a new class of heat transfer fluid called nanofluid has attracted considerable attention from researchers and engineers. A nanofluid is a solid–liquid mixture produced by dispersing nanoparticles in a liquid (usually water, ethylene glycol, or minerals oil) to display enhanced heat transfer due to the combination of convection and conduction and additional energy transfer by particle dynamics and collision [8.73]. The studies of nanofluids with phase change have shown that the presence of nanoparticles in liquid can enhance critical heat flux (CHF) in boiling heat transfer. The mechanism of the CHF enhancement is attributed to the deposition of nanoparticles on the boiling surfaces with foamed porous layers [8.74].

The nanoparticles used in nanofluids are typically made of metals, oxides, titanate nanotubes, or carbon nanotubes with the typical particle size of less than 100 nm (or less than 50 nm in some applications). It is such tiny sizes that make the nanoparticles interact with liquids at the molecular level. Shima and Philip [8.75] have found that by varying the magnetic field strength and its orientation, the thermal properties of magnetic nanofluids (e.g., using Fe_3O_4, γ-Fe_2O_3, or Fe nanoparticles) can be tuned to vary at high values. Their results show that the nanoparticles with larger size exhibit larger thermal conductivity enhancement due to the enhanced dipolar interaction effect. Because the thermal properties of these nanofluids are perfectly reversible, these magnetic nanofluids are ideal for applications in smart devices.

8.8.3.2 Carbon Nanotube

Carbon nanotubes are well known for their outstanding thermophysical and mechanical and properties, particularly their extraordinary thermal conductivities. Measurements have shown that the thermal conductivity of single-wall carbon nanotubes (with the length of 2.6 μm and diameter of 1.7 nm) along their longitudinal axes can reach as high as 3500 W/m-K at the room temperature, which is 9.1 times higher than that of copper [8.76]. However, nanotubes exhibit a very poor thermal performance in the lateral direction. From the same test, the thermal conductivity in the lateral direction was measured as 1.52 W/m-K, which is only 0.4% of copper.

Carbon nanotubes are promising for use in various fields as a thermal material [8.77]. One potential application is to use carbon nanotubes in a layer coated onto motor surfaces for enhancing conductive heat transfer (Figure 8.66).

Another interesting property of nanotubes is that they have the highest strength-to-weight ratio of any known materials. This property may help produce lightweight motor components, for instance, combining nanotubes with other materials into composites that can be used to build lightweight fan, endbell, and housing.

8.8.3.3 NanoSpreader™ Vapor Cooler

NanoSpreader vapor coolers are patented copper-encased, two-phase vapor chambers into which pure water is vacuum sealed [8.78]. In fact, this type of vapor cooler has the same working principle as heat pipes. The heat transfer process consists of four steps: (1) At the high-temperature end, liquid evaporates to vapor by absorbing thermal energy from the heat source. (2) Vapor migrates through microperforated copper sheets toward the low-temperature end. (3) Vapor condenses back to liquid, releasing thermal energy. (4) Liquid is absorbed by a copper-mesh wick and flows back to the higher-temperature end. However,

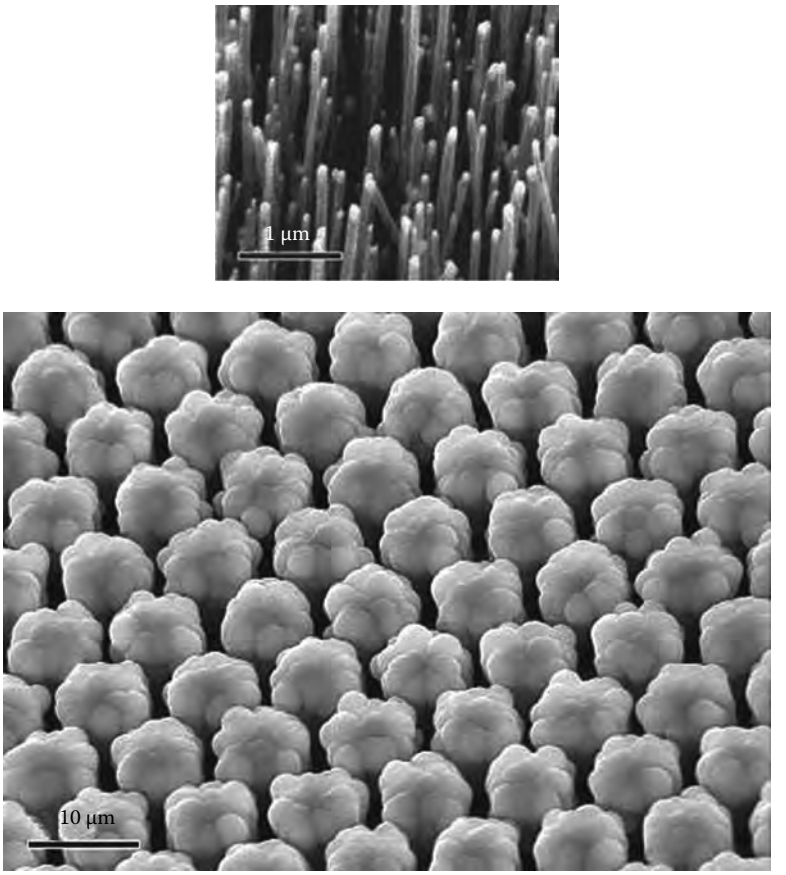


FIGURE 8.66
An array of vertically aligned carbon nanotubes. (Courtesy of NASA Ames Research Center, Moffett Field, CA.)

because of the design optimization of the NanoSpreader vapor cooler, this type of cooler has some advantages over heat pipes, including the following: (1) The reduction in thermal resistance allows to increase cooling efficiency up to 30% over the heat pipes. (2) Heat is spread in a more uniform pattern across a large thin surface area. (3) Due to the uniform heat spreading, hot spots from high-heat-flux devices are minimized/eliminated. (4) The weight can be reduced up to 30% over the heat pipes.

8.8.3.4 CarbAl™ Heat Transfer Material

CarbAl thermal material has been recognized as one of the 100 most significant product innovations in 2009 by *R&D Magazine* [8.79]. CarbAl material is a carbon-based metal nanocomposite composed of 80% carbonaceous matrix and a dispersed metal component of 20% aluminum, with a unique combination of high thermal diffusivity, high thermal conductivity, a low CTE, and low density [8.80]. As one of the passive thermal management materials, its thermal conductivity is about two times higher than that of aluminum. The speed to remove heat from a heat source (i.e., thermal diffusivity) is two and three times higher than copper and aluminum, respectively. The CTE of CarbAl material is only 29% and 41% of aluminum and copper, respectively.

Hesai Exhibit 1031, page 511

8.8.3.5 Ionic Wind Generator

This innovative cooling device is known in industries as the ionic wind generator [8.81]. This type of cooling system does not use a fan or heat sink. It generates air flow based on the ionization of air molecules. A limitation of currently proposed ionic wind generator cooling systems for such devices (and for other conventional cooling devices as well) is that the generated air flow, from a first electrode toward a second electrode, is limited to a linear path that is essentially static and thus could only cool a specific region of an electronic system; particularly, only the regions that are in, or immediately adjacent, the path of the air flow could be cooled.

Because this cooling device was invented for cooling computer CPUs, with the current design, it is only able to cool limited areas within a computer. However, this type of cooling device may be good for eliminating some heat spots in electric motors.

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9

Motor Vibration and Acoustic Noise

Vibration refers to mechanical oscillating movements of an object from its equilibrium position. Sound to human ears is a physiological response to a physical phenomenon of periodically oscillating pressure waves in a certain range. In fact, vibration and sound are closely related to each other. Vibrating objects such as electric motor with random oscillatory movement can produce sound and it travels as acoustic waves through various mediums such as air and water. Sound waves radiate outward from the sound sources in all directions.

Often, electric rotary machines produce vibration and noise, which not only cause gradual hearing loss of operators but also deteriorate machines' operating conditions. In the present era, the use of electric rotary machinery is ubiquitous, and therefore, it is necessary to effectively control and reduce sound and vibration intensities.

Frequency is defined as the number of mechanical oscillations per unit time. Among all vibrational frequencies emitted from a vibrating object, the audible sound frequencies range approximately from 20 to 20,000 Hz. From the viewpoint of physics, noise is called an inconsistent tone that is irregularly composed of many different frequencies and sound waves. From the standpoint of physiology, noise is an unpleasant and unwanted sound.

The characteristic parameters normally used to assess vibration include natural frequency, frequency ratio, damping ratio, phase angle, dynamic displacement, velocity, acceleration, and vibration mode. Frequency is often used in studies of both vibration and sound.

9.1 Vibration and Noise in Electric Motor

When a motor is running under a full-load condition, unwanted vibration is produced due to various reasons that ultimately affect the performance of the motor. These reasons can be divided into two categories: one is dynamic and another is kinematic. The possible causes under dynamic categories are unbalanced electromagnetic reluctance forces, asymmetric air gap, bearing wear and damage, unbalanced rotor, looseness, and improper motor installation. Excessive motor vibration may damage the motor windings, develop high stress on the rotor bearings, reduce the critical speed, and deteriorate motor performance, which may lead to poor motor reliability, increased unexpected downtime, and even motor failure, while acoustic noises pollute the environment and can also harm human's health. Nonetheless, vibration and noise are important parameters for assessment of motor quality and operation reliability. Therefore, the motor design must take into account the resonance frequencies of the motor to shift the critical frequencies away from the motor's operating speed to at least ±10%. This will effectively reduce motor vibration and noise levels at motor normal operation.

Hesai Exhibit 1031, page 516

It has been recognized that about one-third of motor failures are due to mechanical breakages. The largest single contributor to the mechanical breakages is high vibration, which may be attributed to defective motor designs, rough manufacturing processes, improper assemblies and installations, inadequate maintenances, and faulty operations.

The interaction between the rotor and stator is a predominant feature of vibration and noise of an electric motor. A large number of studies [9.1–9.3] have shown that the vibration model $\upsilon(t, \phi, z)$ can be expressed by three terms, each of which has a different physical origin:

$$\upsilon(t,\phi,z) = \upsilon_e(t,\phi,z) + \upsilon_m(t,\phi,z) + \upsilon_a(t,\phi,z) \tag{9.1}$$

where
t is time
ϕ is the angular position
z is the axial position on the stator frame

In Equation 9.1, $\upsilon_e(t, \phi, z)$, $\upsilon_m(t, \phi, z)$, and $\upsilon_a(t, \phi, z)$ are electromagnetic, mechanical, and aerodynamic vibrational terms, respectively. These vibrations produce forces on associate components of the motor and severely affect its performance.

Similarly, sound emitted from a motor can be expressed as the sum of these three components:

$$p(t,\phi,z) = p_e(t,\phi,z) + p_m(t,\phi,z) + p_a(t,\phi,z) \tag{9.2}$$

where $p_e(t, \phi, z)$, $p_m(t, \phi, z)$, and $p_a(t, \phi, z)$ are electromagnetic, mechanical, and aerodynamic sound pressure terms, respectively. Noise from the electromagnetic term dominates at low and medium speeds. At high speeds, the mechanical and aerodynamic noises are the most influential among the three [9.4].

The electromagnetic vibration and resulting noise resulted from various reasons such as phase imbalance, broken rotor bar, asymmetric air gap, eccentric rotor, slot openings, magnetic saturation, space/time harmonics associated with electrical and magnetic frequency, and magnetostrictive expansion of the core lamination. An electromagnetic noise is a low-frequency noise generated in the motor.

The mechanical vibration and resulting noise are associated with motor parts, which are subjected to unbalanced mechanical forces due to oscillatory motions about their equilibrium positions. The causes of mechanical vibration include unbalanced mass of rotating components, shaft misalignment, bearing deterioration, and frame distortion. The method to distinguish between an electromagnetic vibration and a mechanical vibration is to disconnect the electricity supply from the motor. In the absence of the electrical power, the electromagnetic vibration does not exist [9.5], but the mechanical vibration exists till rotary motion is sustained. Here, it should be noted that this method is applicable to IMs and not to PMMs.

As ventilating air flows through a motor, the aerodynamic vibration and noise may be generated due to the interactions of cooling air to both rotating parts (e.g., fan and rotor) and stationary parts (e.g., stator and housing). Usually, noise generated by the aerodynamic vibration is due to turbulent flow, which is proportional to the 6th to 8th power of the flow velocity (i.e., $W \propto u^{6-8}$). This indicates that a doubling of the flow velocity u can increase the sound power W by 18–24 dB, respectively [9.6]. For small-sized motors, the aerodynamic vibration and resulting noise are relatively low compared to the mechanical and electromagnetic components.

Excessive vibration can cause damages in electric motor in different ways [9.7]: (1) It can accelerate bearing failure by causing indentations on the bearing raceways at the ball or roller spacing. (2) It can loosen windings and cause mechanical damage to insulation by fracturing, flaking, or eroding of the material. (3) The excessive movement can generate high temperature, and as a result, the lead wires can become brittle. (4) It can cause brush sparking at commutators or current collector rings. As a result of these problems, motor vibration can harm motor performance, generate high noise, and even cause motor failure.

9.2 Fundamentals of Vibration

It is very important to understand vibrational phenomena such as resonance, harmonics, and damping. Thus, in order to reduce vibration and noise in electric machines, it is essential to understand the fundamentals of vibration.

Generally, vibration in rotordynamic systems can be categorized into synchronous or subsynchronous vibrations depending on the dominant frequency and source of the disturbance forces. The first type is the synchronous vibrations that have a dominant frequency component that matches the rotating speed of the rotor. This type of vibration is usually caused by the imbalance or other synchronous forces in the system. The subsynchronous vibration or whirling has a dominant frequency below the operating speed. This type of vibration is mainly caused by fluid excitation from the cross coupling stiffness [9.8].

Most of the mechanical systems in the real world are considerably very complicated. Therefore, the first step in dealing with difficult problems is to appropriately simplify problems. Generally, the fundamental equations employed in a vibration analysis of multiple-degree-of-freedom systems can be expressed in matrix notation:

$$[m]\{\ddot{x}\}+[c]\{\dot{x}\}+[k]\{x\}=\{F\} \tag{9.3}$$

where

$[m]$ is the mass matrix
$[c]$ is the damping matrix
$[k]$ is the stiffness matrix
$\{\ddot{x}\}$ is the acceleration vector
$\{\dot{x}\}$ is the velocity vector
$\{x\}$ is the displacement vector
$\{F\}$ is the acting force vector

In Equation 9.3, the physical interpretations of the terms from the left-hand side are inertial force, damping force, and spring force (or restoring force), respectively. The term on the right-hand side is external exciting or disturbing force. All these forces balance each other to maintain a dynamic equilibrium state.

9.2.1 Simple Harmonic Oscillating System

A harmonic motion refers to the oscillations that are symmetrical about a position of equilibrium. The motion may have either one frequency or amplitude, defined as simple

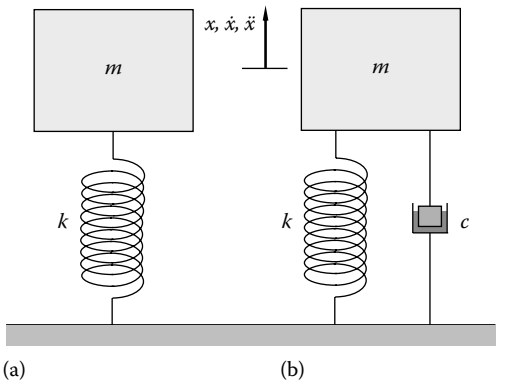


FIGURE 9.1
Oscillation systems: (a) simple harmonic oscillation system without damping and (b) damped harmonic oscillation system.

harmonic motion, or a combination of two or more components of harmonics, defined as complex harmonic motion. Simple harmonic motion oscillates with only the restoring force acting on the system. For this system, the restoring force is directly proportional to the displacement. As an example, a pendulum swings in a small arc, and in its periodic motion, the tangential component of gravity acts as the restoring force of the pendulum. It is the restoring force that interacts with the inertia property of mass (kinetic energy) to perpetuate the oscillation.

For 1D simple harmonic motion (see Figure 9.1a), the equation of motion can be obtained by means of Newton's second law and Hooke's law:

$$F = m\frac{d^2x(t)}{dt^2} = -kx(t) \tag{9.4}$$

where
F is the restoring force
m is the inertial mass of an oscillating body
$x(t)$ is the displacement from an equilibrium position as a function of time t
k is the spring constant

This equation indicates that for simple harmonic motion, the inertial force is balanced by the spring force.

Thus, applying a sinusoidal form of solution, this differential equation can be solved as

$$x(t) = A\cos(\omega t - \varphi) \tag{9.5}$$

where
A is the maximum displacement from the equilibrium position, which is also defined as the oscillating amplitude
φ is the phase angle
ω is the oscillating frequency

$$\omega = 2\pi f = \sqrt{\frac{k}{m}} \tag{9.6}$$

Accordingly, velocity and acceleration of the oscillating body can be found as

$$v = \frac{dx(t)}{dt} = -A\omega\sin(\omega t - \varphi) \tag{9.7}$$

$$a = \frac{d^2x(t)}{dt^2} = -A\omega^2\cos(\omega t - \varphi) \tag{9.8}$$

The oscillating frequency can be expressed as

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{9.9}$$

Since the oscillating period T is reciprocal of frequency, it follows that

$$T = \frac{2\pi}{\sqrt{k/m}} \tag{9.10}$$

The kinetic energy $E_k(t)$ and the potential energy $E_p(t)$ in an oscillating system can be described, respectively, as

$$E_k(t) = \frac{1}{2}m\left[v(t)\right]^2 = \frac{1}{2}mA^2\omega^2\sin^2(\omega t - \varphi) = \frac{1}{2}kA^2\sin^2(\omega t - \varphi) \tag{9.11a}$$

$$E_p(t) = \frac{1}{2}k[x(t)]^2 = \frac{1}{2}kA^2\cos^2(\omega t - \varphi) \tag{9.11b}$$

Therefore, the total mechanical energy E_t is the sum of the energies $E_k(t)$ and $E_p(t)$:

$$E_t = E_k + E_p = \frac{1}{2}kA^2 \tag{9.12}$$

It is worth to note that the total mechanical energy in an oscillating system is constant.

9.2.2 Damped Harmonic Oscillating System

Damping is a measure of an ability of a vibrating system to dissipate mechanical vibratory energy so that the amplitude and duration of vibration can be effectively reduced. In other words, damping creates a force that acts in the opposite direction to the object travel. It can be broadly classified into two categories: passive and active damping. Passive damping can be achieved by using mechanical properties of damping materials and/or by designing vibration-absorbing structures. In active damping, an actuator is used to generate force opposing the motion, regardless of the relative velocity across it [9.9]. Active damping employs sophisticated closed-loop control techniques to minimize the effect of

vibration. The active controllers continuously regulate the damping characteristics and require measurements of response and feedback from the vibrating system.

For a damped simple harmonic oscillating system (see Figure 9.1b), the governing equation is

$$m\frac{d^2x(t)}{dt^2} + c\frac{dx(t)}{dt} + kx(t) = 0 \tag{9.13}$$

where c is the damping coefficient, given in units of N s/m or kg/s.

Introducing a dimensionless quantity called damping ratio ζ, it can be derived as

$$\zeta = \frac{c}{2\sqrt{mk}} \tag{9.14}$$

while the natural frequency of an undamped system ω_n is

$$\omega_n = \sqrt{\frac{k}{m}} \tag{9.15}$$

Equation 9.13 can be rewritten as

$$\frac{d^2x(t)}{dt^2} + 2\zeta\omega_n\frac{dx(t)}{dt} + \omega_n^2 x(t) = 0 \tag{9.16}$$

The initial conditions are given as

$$x(t=0) = X \tag{9.17a}$$

$$\dot{x}(t=0) = V_o \tag{9.17b}$$

The solution of Equation 9.16 is in the form

$$x(t) = e^{st} \tag{9.18}$$

Substituting (9.18) into (9.16), it yields

$$s^2 + 2\zeta\omega_n s + \omega_n^2 = 0 \tag{9.19}$$

The two roots of Equation 9.19 are

$$s_{1,2} = \omega_n\left(-\zeta \pm \sqrt{\zeta^2 - 1}\right) \tag{9.20}$$

Thus, a general solution is given by

$$x(t) = Ae^{s_1 t} + Be^{s_2 t} = Ae^{\omega_n t\left(-\zeta+\sqrt{\zeta^2-1}\right)} + Be^{\omega_n t\left(-\zeta-\sqrt{\zeta^2-1}\right)} \tag{9.21}$$

where the constants A and B are determined by the initial conditions.

According to the value of damping ratio ζ, four cases can be identified as follows:

1. When $\zeta = 1$, the system is in the critical damping condition, giving the fastest return of the system to its equilibrium condition without much oscillation. For this case, the general solution can be written as

$$x(t) = (A + Bt)e^{-\omega_n t} \tag{9.22}$$

2. When $\zeta > 1$, the characteristic roots are positive, real, and distinct. Under such a circumstance, the system is overdamped. Like a critical damping, an overdamped system does not oscillate but takes a longer time to converge to its equilibrium position than a critically damped system. It is important to note that $\zeta > 1$ indicates both roots s_1 and s_2 are negative:

$$x(t) = Ae^{s_1 t} + Be^{s_2 t} \tag{9.23}$$

   Therefore, every solution in this case goes asymptotically to zero.

3. When $0 < \zeta < 1$, the term under the square root is negative and the characteristic roots are not real. Under this condition, the system is underdamped. Introducing $\omega = \omega_n\sqrt{\zeta^2 - 1}$, the characteristic roots are $(-\zeta\omega_n \pm i\omega)$, leading to the general solution as

$$x(t) = Ae^{(-\zeta\omega_n + i\omega)t} + Be^{(-\zeta\omega_n - i\omega)t} = Ae^{-\zeta\omega_n t}\cos(\omega t) + Be^{-\zeta\omega_n t}\sin(\omega t) \tag{9.24}$$

   Alternatively, Equation 9.24 can be expressed as

$$x(t) = Ce^{-\zeta\omega_n t}\cos(\omega t - \varphi) \tag{9.25}$$

   The term $\cos(\omega t - \varphi)$ in Equation 9.25 reflects the oscillating behavior of the system.

4. When $\zeta = 0$, the system is undamped, just as the simple harmonic oscillating system.

   The comparisons of four cases, critical damping, overdamped, underdamped, and undamped, with different damping ratios ζ are presented in Figure 9.2. As can be seen from the figure, for the undamped case ($\zeta = 0$), the system oscillates sustainably with a constant amplitude. Increase in the damping ratio in the range of $0 < \zeta < 1$ leads to vibration decays, accompanied with the reduction of the oscillation amplitude. The reduction in the oscillation amplitude is inversely proportional to the damping ratio ζ. When it reaches critical damping at $\zeta = 1$, the system no longer oscillates, but it monotonically reaches equilibrium in the shortest time without overshooting. For overdamped cases where $\zeta > 1$, the system experiences large damping that it does not oscillate at all. In fact, if the system is displaced from equilibrium, it takes a long time for it to return to its initial position because the damping force is so severe.

9.2.3 Forced Vibration with Damping

In many mechanical systems, it is often encountered that an external time-dependent force is applied to the system to start forced vibration. The applied exciting force may be harmonic,

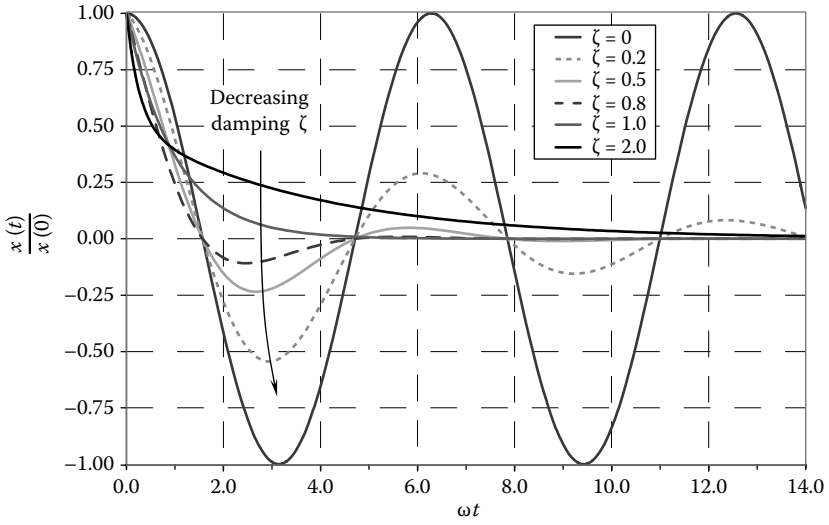


FIGURE 9.2
Comparison of damped and undamped harmonic oscillating systems with various damping ratios: (a) $\zeta = 1$, critical damping; (b) $\zeta > 1$, overdamped; (c) $0 < \zeta < 1$, underdamped; and (d) $\zeta = 0$, undamped.–

nonharmonic (periodic or nonperiodic), or random in nature. The dynamic response of a system to a suddenly applied nonperiodic excitation is called transient response.

Considering a dynamic response of a damped system to a harmonic exciting force $F(t)$ shown in Figure 9.3, this single-degree-of-freedom system has the mass m that is supported by a spring and a damper and is subject to an external exciting force in the form

$$F(t) = F_o \sin(\omega t - \varphi) \tag{9.26}$$

where
F_o is the force magnitude
ω is the excitation frequency
φ is the phase angle

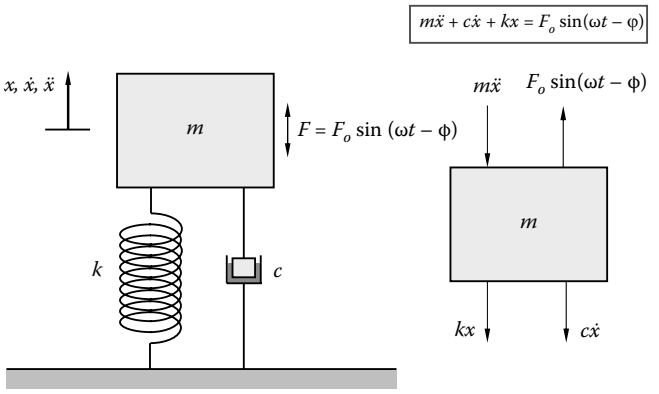


FIGURE 9.3
Forced vibration of single-degree-of-freedom system with damping.

The value of φ depends on $F(t)$ at $t=0$. The dynamic equation of motion is given as

$$m\frac{d^2x(t)}{dt^2}+c\frac{dx(t)}{dt}+kx(t)=F_o\sin(\omega t-\varphi) \tag{9.27}$$

where
c is the viscous damping coefficient
k is the spring coefficient

The general solution of this nonhomogeneous second-order differential equation consisted of two parts: the complementary solution $x_c(t)$ and the particular solution $x_p(t)$, that is, $x(t)=x_c(t)+x_p(t)$. The complementary solution is obtained by setting the right-hand side zero:

$$m\frac{d^2x_c(t)}{dt^2}+c\frac{dx_c(t)}{dt}+kx_c(t)=0 \tag{9.28}$$

This equation is identical to (9.13). Therefore, the general solution is given by

$$x_c(t)=Ae^{\omega_o t\left(-\zeta+\sqrt{\zeta^2-1}\right)}+Be^{\omega_o t\left(-\zeta-\sqrt{\zeta^2-1}\right)} \tag{9.29}$$

where constants A and B are determined from the specific boundary conditions and ζ values.

To obtain the particular solution $x_p(t)$, it is assumed that

$$x_p(t)=D\cos(\omega t-\varphi)+E\sin(\omega t-\varphi) \tag{9.30}$$

where D and E are constants. The associated velocity and acceleration are

$$\begin{cases}\dfrac{dx_p(t)}{dt}=-D\omega\sin(\omega t-\varphi)+E\omega\cos(\omega t-\varphi)\\[2ex]\dfrac{d^2x_p(t)}{dt^2}=-D\omega^2\cos(\omega t-\varphi)-E\omega^2\sin(\omega t-\varphi)\end{cases} \tag{9.31}$$

Substituting Equation 9.31 into 9.27, it yields

$$\cos(\omega t-\varphi)(-mD\omega^2+cE\omega+kD)+\sin(\omega t-\varphi)(-mE\omega^2-cD\omega+kE)=F_o\sin(\omega t-\varphi) \tag{9.32}$$

Comparing both sides of Equation 9.32, it leads to

$$D(k-m\omega^2)+cE\omega=0 \tag{9.33a}$$

$$E(k-m\omega^2)-cD\omega=F_o \tag{9.33b}$$

Defining the frequency ratio β as the ratio of excitation frequency ω over natural frequency ω_n,

$$\beta=\frac{\omega}{\omega_n}=\frac{f}{f_n} \tag{9.34}$$

Equation 9.34 can be rewritten as

$$D(1-\beta^2)+2\zeta\beta E=0 \tag{9.35a}$$

$$-2\zeta\beta D+E(1-\beta^2)=\frac{F_o}{k} \tag{9.35b}$$

By solving Equations 9.35a and 9.35b, constants D and E can be determined as

$$D=-\frac{F_o}{k}\frac{2\zeta\beta}{(1-\beta^2)^2+(2\zeta\beta)^2} \tag{9.36a}$$

$$E=\frac{F_o}{k}\frac{1-\beta^2}{(1-\beta^2)^2+(2\zeta\beta)^2} \tag{9.36b}$$

The particular solution $x_p(t)$ becomes

$$x_p(t)=-\frac{F_o}{k}\frac{2\zeta\beta}{(1-\beta^2)^2+(2\zeta\beta)^2}\cos(\omega t-\varphi)+\frac{F_o}{k}\frac{1-\beta^2}{(1-\beta^2)^2+(2\zeta\beta)^2}\sin(\omega t-\varphi) \tag{9.37}$$

Alternatively, the particular solution $x_p(t)$ can be expressed as

$$x_p(t)=X\sin\left[(\omega t-\varphi)-\Phi\right]=X\left[\sin(\omega t-\varphi)\cos\Phi-\cos(\omega t-\varphi)\sin\Phi\right] \tag{9.38}$$

Comparing Equations 9.37 and 9.38, it follows that

$$X\cos\Phi=\frac{F_o}{k}\frac{1-\beta^2}{(1-\beta^2)^2+(2\zeta\beta)^2} \tag{9.39a}$$

$$X\sin\Phi=\frac{F_o}{k}\frac{2\zeta\beta}{(1-\beta^2)^2+(2\zeta\beta)^2} \tag{9.39b}$$

These two equations can be combined to yield the desired equation for X:

$$X=\frac{F_o}{k}\frac{1}{\sqrt{(1-\beta^2)^2+(2\zeta\beta)^2}} \tag{9.40}$$

and

$$\tan\Phi=\frac{2\zeta\beta}{1-\beta^2} \tag{9.41}$$

Thus,

$$x_p=\frac{F_o}{k}\frac{1}{\sqrt{(1-\beta^2)^2+(2\zeta\beta)^2}}\sin\left[(\omega t-\varphi)-\Phi\right] \tag{9.42}$$

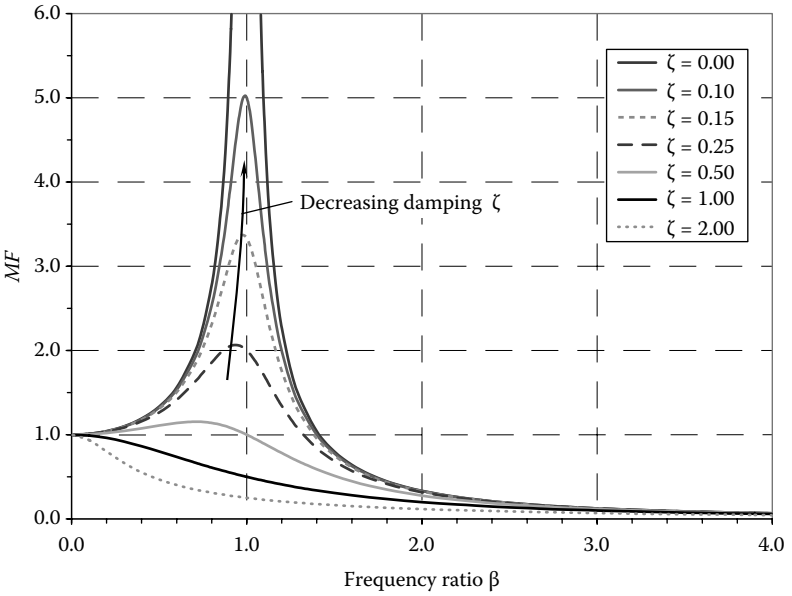


FIGURE 9.4
Variations of the *MF* with frequency ratio β and damping ratio ζ.

The ratio of the amplitude of the steady-state response to the static deflection under the action of force F_o is known as the *MF*:

$$MF = \frac{1}{\sqrt{\left(1-\beta^2\right)^2 + (2\zeta\beta)^2}} \tag{9.43}$$

Thus, the *MF* depends upon the frequency ratio β and the damping ratio ζ. The variations of the *MF* as functions of ζ and β are displayed in Figure 9.4. It can be observed that for all cases, if β is much less than 1, the *MF* approaches 1, regardless of the damping ratio. Actually, exceeding excitation frequencies cause the amplitude of forced vibration to be sufficiently small.

If β becomes large enough (i.e., $\omega \gg \omega_n$), the *MF* approaches zero for all ζ values. This is because the increase in excitation frequency always causes the increase in system inertia force, resulting in the reduction of the amplitude of forced vibration. The system exhibits resonance when $\beta = 1$, that is, the excitation frequency ω is exactly equal to the system natural frequency ω_n. At this point, the amplitude of vibration increases without bound if no damping is applied on the system.

For very small damping ratios (i.e., $\zeta \to 0$), the *MF* can go extremely high near $\beta = 1$. Theoretically, when $\zeta = 0$, the *MF* goes infinity at $\beta = 1$. Increasing ζ can reduce sharply the peak values of the *MF*, particularly at $\beta = 1$ due to the damping effect. As ζ becomes considerably large, the amplitude of the *MF* is reduced for all β values. At the critical damping where $\zeta = 1$, the *MF* decreases monotonically with the increase in the frequency ratio β. For overdamped cases that $\zeta > 1$, the *MF* approaches zero much faster than all other cases.

Figure 9.5 shows the variations of the phase angle with frequency ratio β and damping ratio ζ. At the resonance point where $\omega = \omega_n$, the phase angle is 90° for all damping ratios.

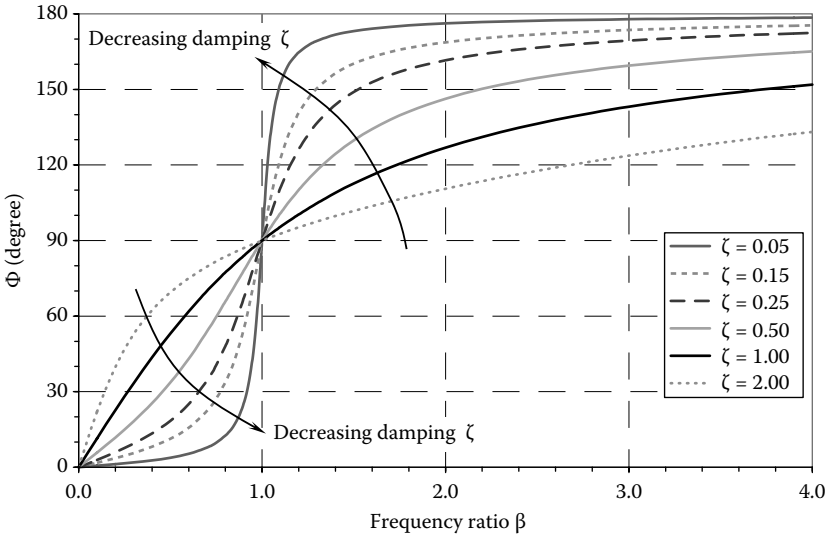


FIGURE 9.5
Variations of phase angle Φ with frequency ratio β and damping ratio ζ.

For an undamped system (where $\zeta = 0$), the phase angle $\Phi = 0°$ for $0 \leq \beta < 1$ and $\Phi = 180°$ for $\beta \geq 1$. Increasing the damping ratio ζ leads to a longer time for the phase angle to approach 180°. For $0 < \beta < 1$ and $\zeta > 0$, the response lags the excitation. Contrarily, for $\beta > 1$ and $\zeta > 0$, the response leads the excitation.

9.2.4 Forced Vibration due to Mass Unbalance

Mass unbalance is one of the most common sources of vibration excitation in rotating machinery because it develops a centrifugal force along with dynamic eccentricity. The total unbalanced force is the sum of the centrifugal force and unbalanced electromagnetic force. Rotor eccentricity increases due to this unbalanced force that tries to pull apart the rotor further away from the stator bore center, varying the air gap. This unbalanced force is proportional to the rotor rotating speed [9.10,9.11].

Although a motor is designed, manufactured, assembled, and installed with great care to ensure proper balance under working conditions, some degree of mass unbalance is always present in the motor. Under normal circumstances, slight unbalance in major rotating components with relatively low rotating speeds should not notably affect motor operation. However, if the deviation in mass symmetry about the rotating axis becomes significant or if the rotating speed of motor is considerably high, large unbalanced forces are produced to exert on the rotor, bearings, support structure, and ancillary equipment, leading to high vibration and high stresses of these components. In fact, vibration is a major cause of downtime and reliability problems for electric motors. Many sources highlight the impact of vibration on the motor performance, noise, fatigue lifetime, and failure. In a worse case, a rotor unbalance may cause the rotor and stator contact and rub each other, resulting in a severe motor failure. The vibration frequency of unbalance f_r is equal to the rotor rotating speed in the unit of revolutions per second (rps):

Hesai Exhibit 1031, page 527

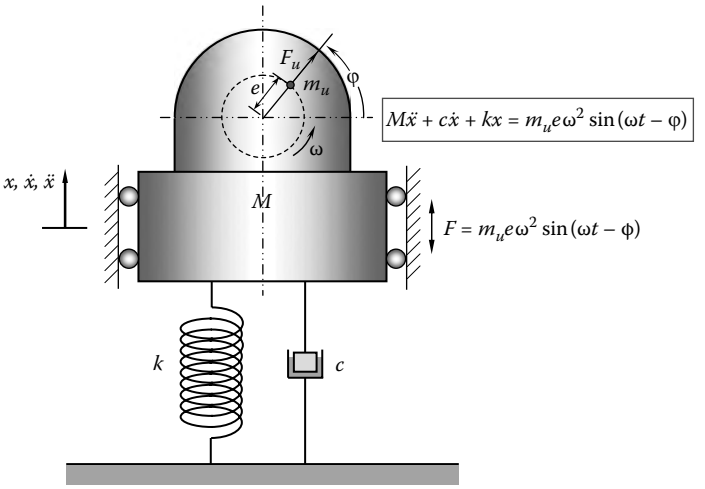


FIGURE 9.6
Forced vibration induced by unbalanced mass.

$$f_r = \frac{n}{60} = \frac{\omega}{2\pi} \tag{9.44}$$

where
the unit of rotor rotating speed n is rpm
the unit of the angular speed ω is radians per second (rad/s)

Considering the rotating system in Figure 9.6, the unbalance is represented by an unbalanced mass m_u with eccentricity e from the system rotating axis and angular velocity ω. The centrifugal force resulting from the eccentric mass is given by

$$F_u = m_u e\omega^2 \sin(\omega t - \varphi) \tag{9.45}$$

This force serves as the exciting force to induce forced vibration of the system. The total mass of the system is M. The equation of motion is

$$M\frac{d^2x(t)}{dt^2} + c\frac{dx(t)}{dt} + kx = (m_u e\omega^2)\sin(\omega t - \varphi) \tag{9.46}$$

This equation is identical to (9.27) as F_o is replaced by $(m_u e\omega^2)$. Because the complementary solution $x_c(t)$ is independent of the exciting force, it remains the same as shown in Equation 9.29. The particular solution $x_p(t)$ can be obtained by replacing F_o by $(m_u e\omega^2)$ in Equation 9.42, that is,

$$x_p = \frac{m_u e\omega^2}{k}\frac{1}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}}\sin\left[(\omega t - \varphi) - \Phi\right] \tag{9.47}$$

It is noted that the damping ratio ζ in this case is based on the system mass M:

$$\zeta = \frac{c}{2\sqrt{Mk}} \tag{9.48}$$

Similarly, the system national frequency becomes

$$\omega_n = \sqrt{\frac{k}{M}} \tag{9.49}$$

The steady-state amplitude X is given by

$$X = \frac{m_u e\omega^2}{k} \frac{1}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}} = \frac{m_u e}{M} \frac{\beta^2}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}} \tag{9.50}$$

This equation can be alternatively rearranged as

$$\frac{MX}{m_u e} = \frac{\beta^2}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}} \tag{9.51}$$

and

$$\tan\Phi = \frac{2\zeta\beta}{1-\beta^2} \tag{9.52}$$

The variations of $MX/m_u e$ are plotted in Figure 9.7 as functions of frequency ratio β and damping ratio ζ. It shows all curves start at zero amplitude. Resonances occur for various damping levels at β = 1, that is, when the excitation frequency ω is identical to the system natural frequency ω_n. At very large β values, all curves approach 1 regardless of damping.

Since the phase angle Equation 9.52 is identical to 9.41, the variation of the phase angle with β and ζ is the same as in Figure 9.5.

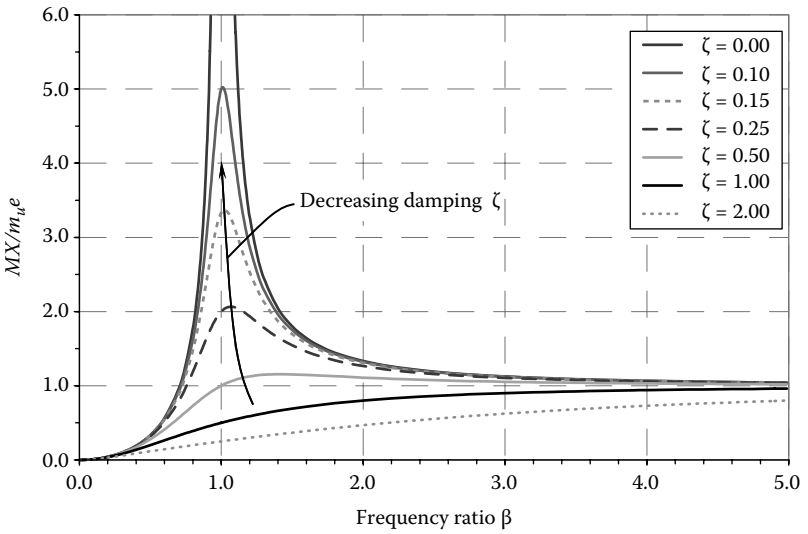


FIGURE 9.7
Variations of factor $MX/m_u e$ with frequency ratio β and damping ratio ζ.

Due to mass unbalance in rotor condition, mutual inductances between stator and rotor loops become unsymmetrical, inducing characteristic harmonics as given in the following:

$$f_{ubm} = f\left[\frac{k(1-s)}{p}+1\right], \quad k = 1, 2, 3, \ldots \tag{9.53}$$

where
p is the number of pole pairs
s is the slip
f is the supply frequency

Among all the causes of motor vibration, rotor unbalance is the most common one. In fact, unbalanced mass in a rotor creates large forces that often result in motor failure and downtime. For high-speed motors, even a very small amount of unbalance may cause a severe vibration problem. Mass unbalance in a rotor may come from manufacturing errors such as porosity in casting, improper machining process, manufacturing tolerances, and gain or loss of material during operation [9.12].

9.2.5 Vibration Induced by Support Excitation

Many motor applications involve vibrations that are induced by the excitation of motor support. A typical example is driving a vehicle on a bumpy road. The vehicle shakes due to the uneven surfaces such as broken pavement, potholes, troughs, and small stones. Vibration is transmitted through wheels to the vehicle body and other components including motors.

Another example is related to seismic waves generated by earthquakes. Earthquakes generate three types of seismic waves: primary (longitudinal) wave, secondary (shear) wave, and surface wave (mixed wave of primary and secondary waves). Due to the long wavelength and high amplitude, surface wave is the main cause of building destruction. When a building sways and oscillates during an earthquake, all objects attached to the building vibrate accordingly. This will seriously affect the safe operation of elevator motors. Therefore, it is very important to understand the excitation mechanism of motor vibration induced by support excitation because it helps design safer and more reliable elevator motors.

As demonstrated in Figure 9.8, a spring–damper system undergoes harmonic motion, where $x(t)$ denotes the displacement from the mass equilibrium position and $y(t)$ the displacement of the motor support at time t. The damping force in this case is the product of the relative velocity between the system and its support $(\dot{x}-\dot{y})$ and the damping coefficient c, that is, $c(\dot{x}-\dot{y})$, and the spring force is the product of the net spring elongation $(x - y)$ and the spring coefficient k, that is, $k(x - y)$.

The motion equation is expressed as

$$m\frac{d^2x(t)}{dt^2}+c\left(\frac{dx(t)}{dt}-\frac{dy(t)}{dt}\right)+k(x-y)=0 \tag{9.54}$$

Introducing the relative displacement $x_R = x - y$, Equation 9.54 can be rewritten as

$$m\frac{d^2x_R(t)}{dt^2}+c\frac{dx_R(t)}{dt}+kx_R(t)=-m\frac{d^2y(t)}{dt^2} \tag{9.55}$$

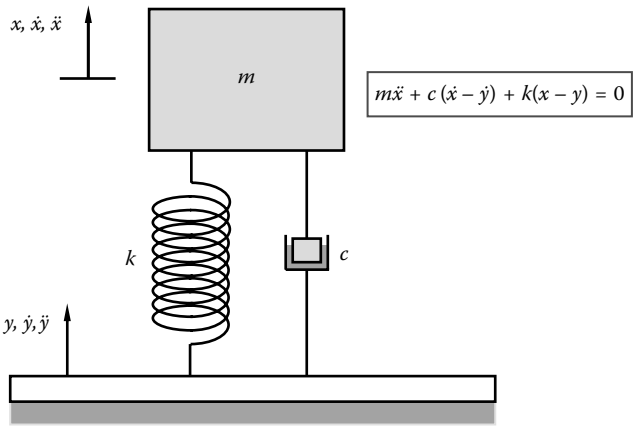


FIGURE 9.8
Vibration induced by support excitation in a two-degree-of-freedom mechanical system.

Following the same procedure as discussed previously, this equation is solved for x_R:

$$x_R(t) = \frac{\beta^2 y(t)}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}} \sin(\omega t - \Phi) \tag{9.56}$$

The vector diagram shows that the relationship between three vectors is (Figure 9.9)

$$\mathbf{x} = \mathbf{x}_R + \mathbf{y} \tag{9.57}$$

It follows that

$$x^2 = x_R^2 + y^2 + 2x_R y \cos\Phi \tag{9.58}$$

The phase angle is obtained as follows:

$$\tan\Phi = \frac{2\zeta\beta}{1-\beta^2} \tag{9.59}$$

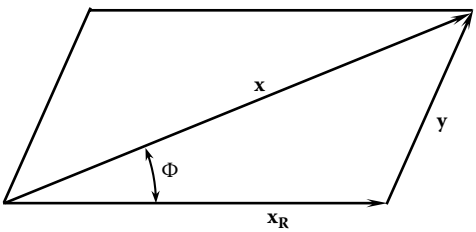


FIGURE 9.9
The relationship of three vectors given that $\mathbf{x} = \mathbf{x}_R + \mathbf{y}$. The vector $\mathbf{x}_R$ lagging behind $\mathbf{x}$ by an angle Φ.

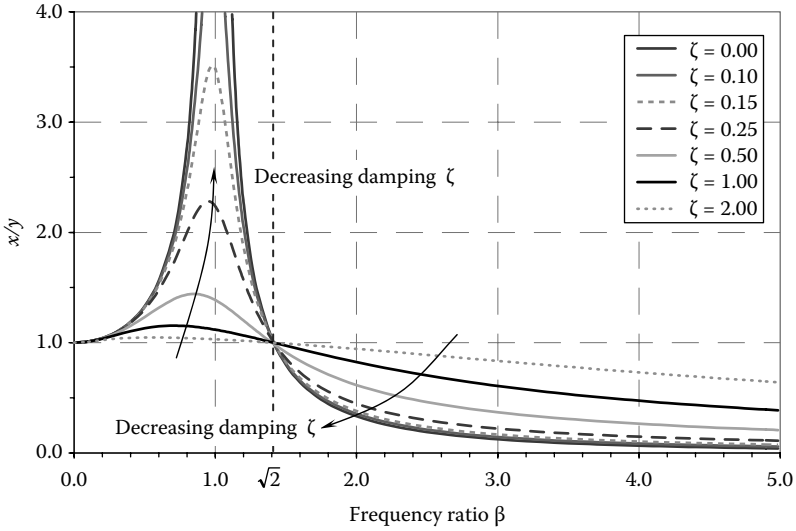


FIGURE 9.10
Variations of displacement transmissibility x/y with frequency ratio β and damping ratio ζ in a two-degree-of-freedom mechanical system where vibration is induced by the excitation of support.

The ratio of the amplitude of the response $x(t)$ to that of the support motion $y(t)$ is defined as the displacement transmissibility. Thus,

$$\frac{x(t)}{y(t)} = \frac{\sqrt{1+(2\zeta\beta)^2}}{\sqrt{(1-\beta^2)^2+(2\zeta\phi)^2}} \tag{9.60}$$

The displacement transmissibility $x(t)/y(t)$ is plotted in Figure 9.10 with various values of the damping ratio ζ and frequency ratio β. It can be seen that the value of x/y is unity for all curves at $\beta = 0$ and $\beta = \sqrt{2}$, regardless of the damping levels. At the system resonance point where $\beta = 1$, x/y becomes infinity for an undamped spring–mass system ($\zeta = 0$). Increasing the ζ value will greatly reduce the peak values of the displacement transmissibility x/y. The values of x/y are larger than 1 for $0 < \beta < \sqrt{2}$ and less than 1 for $\beta > \sqrt{2}$. Moreover, for $\beta < \sqrt{2}$, a small change in the damping ratio leads to a large change in x/y. As a contrast, for $\beta > \sqrt{2}$, the change in the damping ratio is less sensitive to the magnitude of x/y.

9.2.6 Directional Vibration

According to the direction of vibration, there are basically three types of vibration associated with the motion:

1. Radial vibration is the vibration of rotor in the radial direction. This type of vibration can be caused by several reasons: (a) Rotating magnetic field produces a traveling force wave that distorts stator and causes vibration in the stator frame and base (see Figure 9.11). (b) Any unbalance on the rotor can result in an unbalanced centrifugal force, leading to the rotor distortion. (c) Large bearing radial clearance provides free space for the rotor in the radial direction. (d) Bearing misalignment can directly lead to motor vibration.