

DAMIR JELASKA

GEARS AND GEAR DRIVES



WILEY

LIBERTY EXHIBIT 1012, Page 1

GEARS AND GEAR DRIVES

GEARS AND GEAR DRIVES

Damir Jelaska

University of Split, Croatia



A John Wiley & Sons, Ltd., Publication

This edition first published 2012
© 2012 John Wiley & Sons Ltd

Registered office

John Wiley & Sons Ltd, The Atrium, Southern Gate, Chichester, West Sussex, PO19 8SQ, United Kingdom

For details of our global editorial offices, for customer services and for information about how to apply for permission to reuse the copyright material in this book please see our website at www.wiley.com.

The right of the author to be identified as the author of this work has been asserted in accordance with the Copyright, Designs and Patents Act 1988.

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording or otherwise, except as permitted by the UK Copyright, Designs and Patents Act 1988, without the prior permission of the publisher.

Wiley also publishes its books in a variety of electronic formats. Some content that appears in print may not be available in electronic books.

Designations used by companies to distinguish their products are often claimed as trademarks. All brand names and product names used in this book are trade names, service marks, trademarks or registered trademarks of their respective owners. The publisher is not associated with any product or vendor mentioned in this book. This publication is designed to provide accurate and authoritative information in regard to the subject matter covered. It is sold on the understanding that the publisher is not engaged in rendering professional services. If professional advice or other expert assistance is required, the services of a competent professional should be sought.

Library of Congress Cataloging-in-Publication Data

Jelaska, Damir.

Gears and gear drives / Damir Jelaska.

p. cm.

Includes bibliographical references and index.

ISBN 978-1-119-94130-9 (cloth)

1. Gearing. I. Title.

TJ184.J415 2012

621.8'33—dc23

2012014164

A catalogue record for this book is available from the British Library.

Print ISBN: 9781119941309

Set in 10/12 pt Times by Thomson Digital, Noida, India

In memory of my mother

Contents

Preface	xv
Acknowledgments	xvii
1 Introduction	1
1.1 Power Transmissions and Mechanical Drives	1
1.2 Classification of Mechanical Drives	3
1.3 Choosing a Mechanical Drive	7
1.4 Multi-Step Drives	9
1.5 Features and Classification of Gear Drives	12
1.5.1 <i>Features of Gear Drives</i>	12
1.5.2 <i>Classification of Gear Drives</i>	12
1.6 List of Symbols	16
1.6.1 <i>Subscripts to Symbols</i>	16
2 Geometry of Cylindrical Gears	17
2.1 Fundamentals of the Theory of Tothing	17
2.1.1 <i>Centroides, Roulettes and Axodes</i>	17
2.1.2 <i>Envelopes, Evolutes and Involute</i>	18
2.1.3 <i>Cycloid and Involute of a Circle</i>	18
2.1.3.1 <i>Cycloid</i>	18
2.1.3.2 <i>Involute of Circle</i>	20
2.1.4 <i>Main Rule of Tothing</i>	21
2.1.4.1 <i>Analytical Determining of Mated Profiles</i>	25
2.1.4.2 <i>Radii of Curvature of Mated Profiles</i>	27
2.2 Geometry of Pairs of Spur Gears	29
2.2.1 <i>Cycloid Tothing</i>	29
2.2.2 <i>Involute Tothing</i>	30
2.3 Involute Teeth and Involute Gears	33
2.4 Basic Tooth Rack	35
2.5 Fundamentals of Cylindrical Gears Manufacture	38
2.5.1 <i>Generating Methods</i>	38
2.5.2 <i>Forming Methods</i>	43

2.5.3	<i>Gear Finishing</i>	45
2.5.4	<i>Basic Rack-Type and Pinion-Type Cutters</i>	48
2.6	<i>Cutting Process and Geometry of Gears Cut with Rack-Type Cutter</i>	49
2.6.1	<i>Profile Shift</i>	49
2.6.2	<i>Meshing of Rack Cutter with Work Piece, Basic Dimensions of Gear</i>	50
2.6.3	<i>Tooth Thickness at Arbitrary Circle</i>	51
2.6.4	<i>Tip Circle Diameter</i>	52
2.6.5	<i>Profile Boundary Point; Tooth Root Undercutting</i>	53
2.6.6	<i>Effect of Profile Shift on Tooth Geometry</i>	55
2.6.7	<i>Gear Control Measures</i>	56
2.6.7.1	<i>Chordal Tooth Thickness on the Arbitrary Circle</i>	56
2.6.7.2	<i>Constant Chord Tooth Thickness</i>	57
2.6.7.3	<i>Span Measurement</i>	58
2.6.7.4	<i>Dimension Over Balls</i>	60
2.7	<i>Parameters of a Gear Pair</i>	62
2.7.1	<i>Working Pressure Angle of a Gear Pair</i>	62
2.7.2	<i>Centre Distance</i>	63
2.7.3	<i>Gear Pairs With and Without Profile Shift</i>	64
2.7.3.1	<i>Gear Pairs Without Profile Shift</i>	64
2.7.3.2	<i>Gear Pairs with Profile Shift</i>	64
2.7.4	<i>Contact Ratio</i>	66
2.7.5	<i>Distinctive Points of Tooth Profile</i>	70
2.7.6	<i>Kinematic Parameters of Toothing</i>	71
2.8	<i>Basic Parameters of Gears Generated by the Fellows Method</i>	74
2.8.1	<i>Pinion-Type Cutter</i>	74
2.8.2	<i>Dimensions of Gears Cut by Pinion-Type Cutter</i>	75
2.8.3	<i>Undercutting the Tooth Root</i>	76
2.8.4	<i>Geometry of Internal Gear Toothing</i>	77
2.9	<i>Interferences in Generating Processes and Involute Gear Meshing</i>	78
2.9.1	<i>Interferences in Tooth Cutting</i>	78
2.9.1.1	<i>Tooth Root Undercutting</i>	78
2.9.1.2	<i>Overcutting the Tooth Addendum (First Order Interference)</i>	79
2.9.1.3	<i>Overcutting the Tooth Tip Corner (Second Order Interference)</i>	80
2.9.1.4	<i>Radial Interference (Third Order Interference)</i>	80
2.9.1.5	<i>Null Fillet</i>	82
2.9.2	<i>Interferences in Meshing the Gear Pair Teeth</i>	83
2.9.2.1	<i>Gear Root Interference</i>	83
2.9.2.2	<i>Interferences of Tooth Addendum</i>	84
2.9.2.3	<i>Radial Interference</i>	84
2.10	<i>Choosing Profile Shift Coefficients</i>	84
2.10.1	<i>Choosing Profile Shift Coefficients by Means of Block-Contour Diagrams</i>	85
2.10.2	<i>Choosing Profile Shift Coefficients by Means of Lines of Gear Pairs</i>	88
2.11	<i>Helical Gears</i>	91
2.11.1	<i>Basic Considerations</i>	91
2.11.2	<i>Helical Gear Dimensions and Parameters of a Gear Pair</i>	97

2.11.3	<i>Control Measures</i>	100
2.11.4	<i>Helical Gear Overlaps</i>	102
2.11.4.1	<i>Length of Contact Lines</i>	104
2.12	<i>Tooth Flank Modifications</i>	106
2.12.1	<i>Transverse Profile Modifications</i>	107
2.12.1.1	<i>Pre-Finish Flank Undercut</i>	107
2.12.1.2	<i>Tip Corner Chamfering and Tip Corner Rounding</i>	107
2.12.1.3	<i>Tooth Tip Relief</i>	108
2.12.1.4	<i>Tooth Root Relief</i>	113
2.12.1.5	<i>Tooth Tip Relief of the Gear Generated by Pinion-Type Cutter</i>	114
2.12.1.6	<i>Profile Crowning</i>	117
2.12.2	<i>Flank Line Modifications</i>	117
2.12.2.1	<i>Flank Line end Reliefs</i>	117
2.12.2.2	<i>Flank Line Slope Modification</i>	117
2.12.2.3	<i>Flank Line Crowning</i>	118
2.12.3	<i>Flank Twist</i>	119
2.13	<i>Geometry of Fillet Curve</i>	119
2.13.1	<i>Fillet Curve Equation</i>	120
2.13.2	<i>Fillet Curve Radius of Curvature</i>	124
2.13.3	<i>Geometry of Undercut Teeth</i>	125
2.13.3.1	<i>Profile Boundary Point</i>	125
2.13.3.2	<i>Contact Ratio of Gears with Undercut Teeth</i>	126
2.14	<i>Tolerances of Pairs of Cylindrical Gears</i>	127
2.14.1	<i>Control and Tolerances of Gear Body</i>	128
2.14.2	<i>Control and Tolerances of Teeth</i>	128
2.14.2.1	<i>Tooth Profile Control</i>	130
2.14.2.2	<i>Helix Deviations</i>	134
2.14.2.3	<i>Pitch Deviations</i>	135
2.14.2.4	<i>Radial Runout of Teeth</i>	136
2.14.2.5	<i>Tangential Composite Deviation</i>	136
2.14.2.6	<i>Tooth Thickness Tolerances</i>	138
2.14.2.7	<i>CNC Gear Measuring Centre</i>	143
2.14.3	<i>Control of Gear Pair Measuring Values</i>	145
2.14.3.1	<i>Systems of Gear Fits, Centre Distance Tolerances, Backlash</i>	145
2.14.3.2	<i>Contact Pattern Control</i>	149
2.15	<i>Gear Detail Drawing</i>	151
2.16	<i>List of Symbols</i>	153
2.16.1	<i>Subscripts to symbols</i>	154
2.16.2	<i>Combined Symbols</i>	155
3	<i>Integrity of Gears</i>	157
3.1	<i>Gear Loadings</i>	157
3.1.1	<i>Forces Acting on the Gear Tooth</i>	157
3.1.2	<i>Incremental Gear Loadings</i>	159

3.2	Causes of Gear Damage	164
3.2.1	<i>Gear Breakages</i>	164
3.2.2	<i>Active Tooth Flank Damage</i>	166
3.3	Pitting Load Capacity	170
3.3.1	<i>Contact Stresses</i>	170
3.3.1.1	<i>Nominal Value of Contact Stress</i>	170
3.3.1.2	<i>Real Value of Contact Stress</i>	175
3.3.2	<i>Allowable Contact Stresses</i>	181
3.3.3	<i>Dimensioning for Contact Stress</i>	189
3.3.4	<i>List of Symbols for Sections 3.1, 3.2 and 3.3</i>	190
3.3.4.1	<i>Subscripts to Symbols</i>	191
3.3.4.2	<i>Combined Symbols</i>	192
3.4	Tooth Root Load Capacity	193
3.4.1	<i>Tooth Root Stress</i>	193
3.4.2	<i>Tooth Root Permitted Stress</i>	200
3.4.3	<i>Dimensioning for Tooth Root Stress</i>	207
3.5	Gear Load Capacity at Variable Loading	208
3.6	List of Symbols for Sections 3.4 and 3.5	210
3.6.1	<i>Subscripts to Symbols</i>	211
3.6.2	<i>Combined Symbols</i>	212
3.7	Scuffing Load Capacity	213
3.7.1	<i>Safety Factor Against Scuffing for Flash Temperature Method</i>	213
3.7.2	<i>Force Distribution Factor X_G</i>	217
3.7.3	<i>Safety Factor Against Scuffing for Integral Temperature Method</i>	225
3.8	Micro-Pitting Load Capacity	229
3.8.1	<i>Elastohydrodynamic Lubricant Film Thickness</i>	229
3.8.1.1	<i>Calculation of Material Parameter G_M</i>	230
3.8.1.2	<i>Calculation of Speed Parameter U_Y</i>	231
3.8.1.3	<i>Load Parameter W_Y</i>	232
3.8.1.4	<i>Sliding Parameter S_{GF}</i>	232
3.8.2	<i>Safety Factor Against Micro-pitting</i>	232
3.9	List of Symbols for Sections 3.6 and 3.7	236
3.9.1	<i>Subscripts to Symbols</i>	237
3.9.2	<i>Combined Symbols</i>	238
4	Elements of Cylindrical Gear Drive Design	241
4.1	Design Process	241
4.1.1	<i>Design Procedure for a Gear Pair</i>	241
4.1.2	<i>Distribution of Gear Train Transmission Ratio</i>	243
4.1.3	<i>Gear Materials and Heat Treatment</i>	244
4.1.3.1	<i>Metallic Materials and their Heat Treatment</i>	244
4.1.3.2	<i>Sintered Materials</i>	248
4.1.3.3	<i>Polymer Materials</i>	248
4.1.4	<i>Gear Drive Design</i>	249
4.1.4.1	<i>Design of Housing</i>	251
4.1.4.2	<i>Vents</i>	255

4.1.4.3	<i>Lubricant Drain</i>	255
4.1.4.4	<i>Design of Bearing Locations</i>	257
4.1.4.5	<i>Design of Ribs</i>	257
4.1.5	<i>Design of Gears</i>	258
4.2	<i>Gear Drive Lubrication</i>	262
4.2.1	<i>Selection of Lubricant</i>	262
4.2.2	<i>Ways of Gear Lubrication</i>	263
4.2.2.1	<i>Bath Lubrication</i>	263
4.2.2.2	<i>Spray Lubrication</i>	265
4.3	<i>Power Losses and Temperature of Lubricant</i>	266
4.3.1	<i>Power Losses in Mesh</i>	266
4.3.1.1	<i>Power Losses in Mesh, Under Load, for a Single Gear Pair</i>	266
4.3.1.2	<i>Power Losses in Idle Motion</i>	267
4.3.2	<i>Power Losses in Bearings</i>	268
4.3.2.1	<i>Rolling Bearings</i>	268
4.3.2.2	<i>Sliding Bearings</i>	269
4.3.3	<i>Power Losses in Seals</i>	270
4.3.4	<i>Power Efficiency of Gear Drive</i>	270
4.3.5	<i>Temperature of Lubricant</i>	271
4.4	<i>List of Symbols</i>	275
4.4.1	<i>Subscripts to Symbols</i>	276
4.4.2	<i>Combined Symbols</i>	276
5	<i>Bevel Gears</i>	279
5.1	<i>Geometry and Manufacture of Bevel Gears</i>	279
5.1.1	<i>Theory of Bevel Gear Genesis</i>	279
5.1.2	<i>Types and Features of Bevel Gears</i>	280
5.1.3	<i>Application of Bevel Gears</i>	283
5.1.4	<i>Geometry of Bevel Gears</i>	284
5.1.4.1	<i>Fundamentals of Geometry and Manufacture</i>	284
5.1.4.2	<i>Virtual Toothing and Virtual Gears</i>	287
5.1.4.3	<i>Basic Parameters of Straight Bevels</i>	289
5.1.4.4	<i>Design of Bevel Teeth</i>	291
5.1.4.5	<i>Undercut, Profile Shift</i>	291
5.1.4.6	<i>Sliding of Bevels</i>	292
5.1.4.7	<i>Contact Ratio of Straight Bevels</i>	293
5.1.5	<i>Geometry of Helical and Spiral Bevels</i>	293
5.1.6	<i>Manufacturing Methods for Bevel Gears</i>	294
5.1.6.1	<i>Straight Bevels Working</i>	294
5.1.6.2	<i>Spiral and Helical Bevel Working</i>	301
5.2	<i>Load Capacity of Bevels</i>	306
5.2.1	<i>Forces in Mesh</i>	306
5.2.2	<i>Pitting Load Capacity</i>	307
5.2.3	<i>Tooth Root Load Capacity</i>	310
5.2.4	<i>Scuffing and Micro-Pitting Load Capacities</i>	311

5.3	Elements of Bevel Design	311
5.4	Control and Tolerances of Bevel Gears	316
5.4.1	<i>Pitch Control</i>	316
5.4.2	<i>Radial Runout Control of Tothing</i>	318
5.4.3	<i>Tangential Composite Deviation</i>	319
5.4.4	<i>Tooth Thickness Control</i>	319
5.4.5	<i>Bevel Gear Drawing</i>	321
5.5	Crossed Gear Drives	321
5.5.1	<i>Basic Geometry</i>	323
5.5.2	<i>Speed of Sliding</i>	324
5.5.3	<i>Loads and Load Capacity</i>	325
5.5.3.1	<i>Forces Acting on Crossed Gears</i>	325
5.5.3.2	<i>Efficiency Grade</i>	325
5.5.3.3	<i>Load Capacity of Crossed Gear Pair</i>	326
5.6	List of Symbols	327
5.6.1	<i>Subscripts to Symbols</i>	328
5.6.2	<i>Combined Symbols</i>	328
6	Planetary Gear Trains	331
6.1	Introduction	331
6.1.1	<i>Fundamentals of Planetary Gear Trains</i>	331
6.1.2	<i>Rotational Speeds and Transmission Ratio</i>	334
6.1.3	<i>Features of Planetary Gear Trains</i>	341
6.1.4	<i>Mating Conditions</i>	342
6.1.4.1	<i>Condition of Coaxiality</i>	342
6.1.4.2	<i>Condition of Neighbouring</i>	342
6.1.4.3	<i>Assembly Condition</i>	343
6.1.5	<i>Diagrams of Peripheral and Rotational Speeds</i>	344
6.1.6	<i>Wolf Symbolic</i>	347
6.1.7	<i>Forces, Torques and Power of Planetary Gear Trains</i>	347
6.1.7.1	<i>Peripheral Forces and Torques</i>	347
6.1.7.2	<i>Power and Efficiency</i>	349
6.1.7.3	<i>Branching of Power</i>	352
6.1.7.4	<i>Self-Locking</i>	353
6.2	Special Layouts of Simple Planetary Gear Trains	356
6.2.1	<i>Bevel Differential Trains</i>	356
6.2.2	<i>Planetary Gear Trains with Single Gear Pair</i>	358
6.2.3	<i>Harmonic Drive</i>	359
6.2.4	<i>Differential Planetary Gear Trains</i>	361
6.2.5	<i>Planetary Gear Train of a Wankel Engine</i>	362
6.3	Composed Planetary Gear Trains	364
6.3.1	<i>Compound Planetary Gear Trains</i>	364
6.3.2	<i>Parallel Composed Planetary Gear Trains</i>	364
6.3.3	<i>Coupled Planetary Gear Trains</i>	364
6.3.4	<i>Closed Planetary Gear Trains</i>	366
6.3.5	<i>Reduced Coupled Planetary Gear Trains</i>	368

6.3.6	<i>Reverse Reducers</i>	373
6.3.7	<i>Planetary Gear Boxes</i>	374
6.4	Elements of Planetary Gear Train Design	377
6.4.1	<i>Issues of Planetary Gear Train Design</i>	377
6.4.2	<i>Calculations for Central Gears and Planets</i>	382
6.5	List of Symbols	384
6.5.1	<i>Subscripts to Symbols</i>	385
6.5.2	<i>Combined Symbols</i>	386
7	Worm Gear Drives	387
7.1	Concept, Features, Classification	387
7.2	Geometry and Working of Worm Gear Pair	389
7.2.1	<i>Geometry and Working of Worm</i>	389
7.2.1.1	<i>Dimensions of Worm</i>	390
7.2.1.2	<i>Worm Sections</i>	390
7.2.1.3	<i>Worm Working and Shape of Flanks</i>	392
7.2.2	<i>Geometry and Working of Wormwheels</i>	392
7.2.2.1	<i>Wormwheel Geometry</i>	394
7.2.2.2	<i>Wormwheel Working</i>	397
7.2.3	<i>Calculation Values of Worm Gear Pair</i>	399
7.2.3.1	<i>Centre Distance of Worm Gear Pair</i>	399
7.2.3.2	<i>Transmission Ratio and Gear Ratio</i>	399
7.2.3.3	<i>Tip Clearance of Worm Gear Pair</i>	399
7.2.3.4	<i>Contact Ratio of Worm Gear Pair</i>	399
7.2.3.5	<i>Worm Gear Pair Speeds</i>	400
7.3	Control Measures and Tolerances of Worm Gear Pair	400
7.3.1	<i>Control of Worm Measuring Values</i>	401
7.3.1.1	<i>Pitch Control</i>	401
7.3.1.2	<i>Thread Profile Control</i>	401
7.3.1.3	<i>Radial Runout Control</i>	402
7.3.2	<i>Control of Wormwheel Measuring Values</i>	402
7.3.2.1	<i>Pitch Control</i>	402
7.3.2.2	<i>Tooth Profile Control</i>	402
7.3.2.3	<i>Radial Run-Out Control</i>	402
7.3.2.4	<i>Tooth Thickness Control</i>	403
7.3.2.5	<i>Composite Deviation Control</i>	403
7.3.3	<i>Measuring Values Control of Worm Gear Pair</i>	403
7.3.3.1	<i>Centre Distance Control</i>	403
7.3.3.2	<i>Backlash Control</i>	404
7.4	Forces, Power Losses and Efficiency of Worm Gear Drives	404
7.4.1	<i>Forces Acting on Worm Gear Pair</i>	404
7.4.2	<i>Power Losses and Efficiency of Worm Gear Pair</i>	406
7.5	Load Capacity of Worm Gear Pair	409
7.5.1	<i>Wear Load Capacity</i>	409
7.5.1.1	<i>Calculation of Expected Wear</i>	410
7.5.1.2	<i>Permitted Wear</i>	413

7.5.2	<i>Pitting Load Capacity</i>	414
7.5.3	<i>Heating Load Capacity</i>	415
7.5.3.1	<i>Heating Load Capacity at Bath Lubrication</i>	416
7.5.3.2	<i>Heating Load Capacity at Spray Lubrication</i>	416
7.5.4	<i>Wormwheel Bulk Temperature</i>	417
7.5.4.1	<i>Wormwheel Bulk Temperature in Bath Lubrication</i>	417
7.5.4.2	<i>Wormwheel Bulk Temperature in Spray Lubrication</i>	417
7.5.5	<i>Wormwheel Tooth Root Load Capacity</i>	418
7.5.5.1	<i>Shear Stress in Wormwheel Tooth Root</i>	418
7.5.5.2	<i>Shear Fatigue Limit of Wormwheel Tooth</i>	419
7.5.6	<i>Load Capacity for Worm Shaft Deflection</i>	420
7.6	<i>Elements of Worm Gear Drive Design</i>	421
7.6.1	<i>Design Procedure</i>	421
7.6.1.1	<i>Previous Choices</i>	421
7.6.1.2	<i>Dimensioning the Worm Gear Pair</i>	422
7.6.2	<i>Design Details of Worm Gear Drive</i>	424
7.7	<i>List of Symbols</i>	427
7.7.1	<i>Subscripts to Symbols</i>	428
7.7.2	<i>Combined Symbols</i>	429
Further Reading		433
Index		437

Preface

Since gear drives operate with a power efficiency significantly higher than any other mechanical drive, or any electrical, hydraulic or pneumatical power transmission, they have the widest use in transforming rotary motion from the prime mover to the actuator, and their importance is growing day by day. Although efficiency is not the only criterion for choosing the type of transmission, the gear drive, due to its robustness and operational reliability, presents an inevitable component of most mechanical engineering systems. Gear drives are known to be highly demanding in design, manufacture, control and maintenance.

The entire field is well provided with standards, books and journal and conference papers. Thus, why a necessity for this book? There are three main reasons:

1. Much knowledge has lost its validity through the statute of limitations, so it needs to be renewed. This book incorporates up-to-date knowledge.
2. Despite the body of data available through the Internet, there is obviously still a lack of real knowledge. Namely, a basic knowledge is necessary for one to be able to apply the data. By collecting the data and by using the gear standards, a designer can get all the necessary information for gear drive design. Nevertheless, if someone wants to become a gear drive designer, he must primarily have basic knowledge. This book is conceived to enable both the basic knowledge and the data necessary to design, control, manufacture and maintain gear drives.
3. There is no single book so far which incorporates almost all types of gears and gear drives: spur, helical, bevel and worm gear drives and planetary gear trains.

This book is written with the presumption that the reader has a basic knowledge of mechanics and general mechanical engineering. It is primarily addressed to graduate and undergraduate students of mechanical engineering and to professionals dealing with the manufacturing of gears and gear drives. For all of these, it is supposed to be a primary text. Groups with an occasional need for this material are students of industrial engineering, technology, automotive engineering, students of marine engineering, aviation engineering and space engineering and professionals in control and maintenance. The objective of this book is to provide all of these with everything they need regarding the subject matter in a single book: (i) a background for dealing with gears and gear trains (classification, power, torque, transmission ratio distribution), (ii) a complete geometry and kinematics for almost any type of gears and gear drives, (iii) assessments of load capacities in accordance

with recent standards, including the calculation of micro-pitting load capacity, (iv) directions and suggestions for the practical design of gears and gear drives, (v) detailed instructions and formulae for determining the tolerances and procedures for measuring and controlling the accuracy of drives and their members in accordance with the latest standards. The reading matter is accompanied with a large number of figures and every important formula is derived and discussed.

This book consists of seven chapters. The first chapter introduces the reader to the fundamental parameters of mechanical drives – transmission ratio, power, efficiency, torque and rotational speed – and explains the way for determining them. The classification of mechanical drives and gear drives is also included. The second chapter explains in depth the geometry of cylindrical gear toothing as the basis of the entire field of gear drives, beginning with the idea of rolling, through the manufacturing of gears, the mesh and interference of teeth, tooth modifications, to the gear tolerances. The third chapter deals with the integrity of cylindrical gears, presenting the ways of calculating the load capacities for pitting, tooth root strength, scuffing and micro-pitting. In the fourth chapter the cylindrical gear drive design process is suggested and the selection of gear materials and their heat treatment are explained in depth, as well as gear drive lubrication and the efficiency and temperature of the lubricant. The fifth chapter deals with bevel gear drives: geometry, manufacturing, control, tolerances and load capacity checks. Crossed gear drives are also explained. In the sixth chapter simple planetary gear trains are first presented: transmission ratio, torques, efficiency of power and branching. Special trains, like harmonic and composed trains and also coupled, closed and reduced coupled trains are explained, as well as planetary reducers. The seventh chapter deals with worm gear drives: their geometry, manufacture, deviation control and load capacity assessments for the wear, pitting, heating, wormwheel tooth root and worm shaft deflection.

The book assumes that the reader is familiar with the metric (SI) system of units. However, some remarks are given herein: since standard modules are given in millimetres, all gear dimensions should be expressed in millimetres as well. Hence, in all equations where only length units appear, all physical quantities are to be substituted in millimetres. The exceptions are the allowance equations, where gear dimensions are to be substituted in millimetres to obtain the allowance in microns. In other equations, where the dimensions of physical quantities are not only their lengths, the SI scale of units should be applied and gear dimensions should be substituted in metres, regardless of being marked in millimetres in the list of symbols at the end of each chapter. Relationship equations make an exception where the units of both sides of the equation are not the same. In each such equation the units of physical quantities (those which are to be substituted) are specified, as well as the unit of physical value obtained on the left side of the equation.

Acknowledgments

Boris Obsieger, Technical Faculty, University of Rijeka, Croatia

Irma R. Sharma, Hindustan Motors, Noida, India

Josip Obsieger, Tehnical Faculty, University of Rijeka, Croatia

Jože Flašker, Faculty of Mechanical Engineering, University of Maribor, Slovenia

Srđan Podrug, Faculty of Mechanical Engineering, Electrical Engineering and Naval Architecture, University of Split, Croatia

Srečko Glodež, Faculty of Physical Sciences, University of Maribor, Slovenia

Stanislav Pehan, Faculty of Mechanical Engineering, University of Maribor, Slovenia

Zoran Ren, Faculty of Mechanical Engineering, University of Maribor, Slovenia

1

Introduction

1.1 Power Transmissions and Mechanical Drives

Mechanical power transmissions¹ consist of units which, in distinction from electrical, pneumatic and hydraulic ones, transfer power from the prime mover to the actuator (operational machine or operational member) with the assistance of *rotary motion*. These units are called mechanical drives and are situated between the prime mover and the actuator (Figure 1.1). The drive is connected with both the prime mover and the actuator by couplings or clutches forming an entirety whose function is defined by the purpose of the actuator.

The embedding of a power transmission to link the prime mover and the machine operating member can be due to a number of reasons:

- The required speed of the machine operating member very often differs from the speeds of the standard prime movers.
- One prime mover has to drive several actuators.
- The driven side speed has to be frequently changed (regulated), whereas the prime mover cannot be used to full advantage for this purpose.
- Certain periods of the driven side operation may require torques far from those obtained on the motor shaft.
- As a rule, standard motors are designed for uniform rotary motion, while operating members have sometimes to move with varying speed or periodic halts.
- If a resonant vibration of some member in the chain of power transmission cannot be solved in any other way, the frequency of rotary motion can be changed by building-in a drive.
- Sometimes considerations of safety, convenience of maintenance or the dimensions of the machine, especially if the prime mover and operational machine shaft axes are not coaxial, do not allow the direct coupling of the prime mover shaft with operating member.

¹ The power is a feature of some machine or device and cannot be transmitted. Actually, only the energy is transmitted, but it is globally common to say that the power is transmitted.

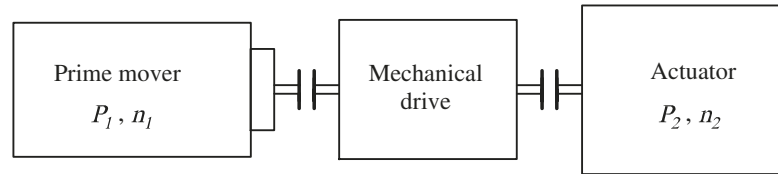


Figure 1.1 Schematic account of a mechanical drive application

The capital task of the designer is to select such an assembly ‘prime mover – transmission (drive)’ which should optimally meet the needs of the operational machine or member. This act of choosing is a complex task, whose solution depends on: (i) accessibility of the energy source and its price, (ii) efficiency of the entirety of prime mover – transmission – operational machine, (iii) investment costs, (iv) operational machine features, primarily the (v) variability of its speed of rotation, (vi) service conditions, (vii) drive maintainability and so on. Within the framework of this task, a particularly complex problem is defining the transmission: mechanical or some other? This question is beyond the scope of this book, but generally it may be affirmed that the basic advantage of mechanical drives in relation to all the others is their very high efficiency, which is becoming more and more important day by day.

The comparative advantages offered by possible transmissions and drives are outlined in Table 1.1 which gives only a general illustration. Recently, a prominent feature in power transfer has been the extensive employment of electric, hydraulic and pneumatic transmissions. Frequently, such transmissions together with mechanical drives are simultaneously used to actuate various mechanisms. The proper choice of a drive for each specific case can be made only by comparing the technical and economical features of several designs.

The mechanical drive driving shaft receives power P_1 at speed of rotation n_1 from the prime mover driven shaft, and the mechanical drive driven shaft supplies power $P_2 < P_1$ at

Table 1.1 Advantages of transmissions and drives

Advantage	Transmission		Mechanical drive		
	Electric	Hydraulic	Pneumatic	Friction	Mesh
Centralized power supply	+		+		
Simplicity of power transmission over large distances	+				
Easy accumulation of power			+		
Step by step speed change over a wide range	+			+	+
Stepless change over a wide range	+	+		+	
Maintaining accurate transmission ratio					+
High speed of rotation	+		+		
Simplicity of machine designed for rectilinear motion		+	+	+	+
No effect of ambient temperature	+		+		+
Comparatively high practically obtainable loads acting upon actuators of machine		+			+
Easy control, automatic and remote	+				

speed of rotation n_2 to the operational machine driving shaft. The difference $P_1 - P_2 = P_L$ is called power loss and the ratio:

$$\eta = \frac{P_2}{P_1} = \frac{P_1 - P_L}{P_1} = 1 - P_L/P_1$$

is called efficiency; it takes a special place amongst power transmission characteristics because it shows unproductive power expenditure and so indirectly characterizes the wear of the drive and its warming up – the capital problems in power transmissions. Warming up causes strength and lifetime decrease of drive parts. Their corrosion resistance and the functional ability of lubricant are also imperilled. The importance of efficiency is raised to a power by the global lack of increasingly expensive energy and its value also decisively affects the price of the drive.

The power loss consists of constant losses which on the whole do not depend on load, and variable losses which on the whole are proportional to the load. The value of constant losses approximates the power of idle run, that is, the power needed to rotate the drive at $P = 0$ on the driven shaft. It depends on the weight of the drive parts, the speed of rotation and the friction in the bearings and on other surfaces of contact.

The second fundamental parameter of a mechanical drive is the *transmission ratio* i defined as the ratio of its driving n_1 and driven n_2 shaft speeds of rotation or angular speeds:

$$i = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2}. \quad (1.1)$$

If $i > 1$ ($n_1 > n_2$) the mechanical drive is called an *underdrive* and its member is called a *reducer*. It reduces the speed of rotation and the transmission ratio is also called a speed reducing ratio. If $i < 1$ the mechanical drive is called an *overdrive*, its member is called a *multiplicator* and the transmission ratio is also called a speed increasing ratio. It multiplies the speed of rotation. An overdrive usually works less efficiently than an underdrive. This is especially true for a toothed wheel gearing.

1.2 Classification of Mechanical Drives

The basic division of mechanical drives falls into:

- *Drives with a constant transmission ratio.*
- *Drives with a variable transmission ratio.*

In **constant transmission ratio drives**, the constant speed of driving shaft rotation results in a constant speed of driven shaft rotation, $n_2 = n_1/i$. Their design should, as a rule, include at least the following data: (i) transmitted power of the driving (P_1) or driven (P_2) shaft or related torques, (ii) speed of rotation (rpm) of the driving (n_1) and driven (n_2) shaft, mutual location of the shafts and distance between them, (iii) overall dimensions and drive operating conditions, especially the dependence of driven shaft rpm or torque on time.

In general, this design has several solutions, that is, given conditions can be used to develop drives of various types. All possible designs should be compared according to their efficiency, weight, size, original and operational costs in order to select the most advantageous one. Some general considerations, mainly the available experience of design,

Table 1.2 Limit parameters of mechanical drives

Drive type	Transmission ratio	Efficiency	Power (MW)	Rotational speed (min^{-1})	Peripheral speed (m/s)	Ratio mass/power (kg/kW)
Cylindrical gears	45	0.99	35	100 000	40	0.2 . . . 1.0
Planetary gear trains	1000	0.996	65	150 000	200	0.4 . . . 1.8
Bevel gears	8	0.98	4	50 000	130	0.6 . . . 2.5
Hypoid gears	50	0.90	1	20 000	50	0.7 . . . 3.0
Crossed gears	100	0.95	0.08	20 000	50	1.5 . . . 3.0
Flat belt	20	0.98	3.6	200 000	120	1.5 . . . 6.0
V-Belt	15	0.94	4	8000	40	1.0 . . . 5.0
Worm drives	100	0.98 ^a	1.5	50 000	70	0.2 . . . 4.5
Friction drives	8	0.98 ^b	0.25	10 000	50	8 . . . 30
Chain drives	15	0.99	5	30 000	40	6 . . . 10

^a Only for small transmission ratios.

^b Values are related to planetary friction drives.

manufacture and operation of various drives enables us to outline generally the limits of priority application of these drives.

Therefore, the designer must take into account the limit values of main parameters which the mechanical drive can reach. These parameters are the maximum values of transmission ratio, efficiency, power and speed of rotation. For underdrives, they are presented in Table 1.2.

However, these limits are of a temporary nature: as new materials are produced and manufacturing methods are improved, our knowledge of processes taking place in transmission becomes deeper and designs are perfected to suit broader fields of application.

For overdrives, these values are considerably smaller; such drives have a poor performance. It primarily refers to efficiency and transmission ratio, but vibrations and noise levels are also much greater, especially in gear drives. It is due to the reasonable and experimentally approved fact that, with a similar error in the manufacture of the two meshing wheels, the driving wheel of greater diameter (in overdrive) causes larger angular accelerations (causing dynamic impacts) on the smaller driven wheel, while the reverse is applied to an underdrive.

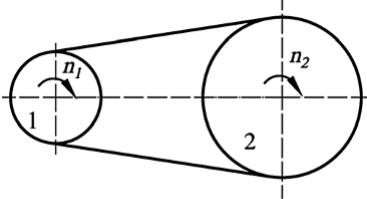
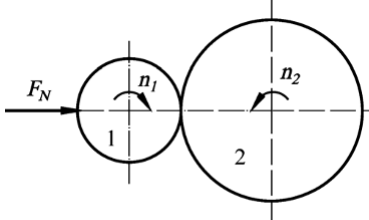
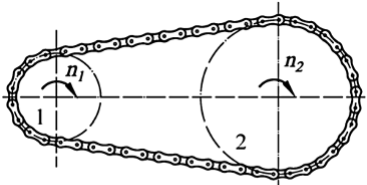
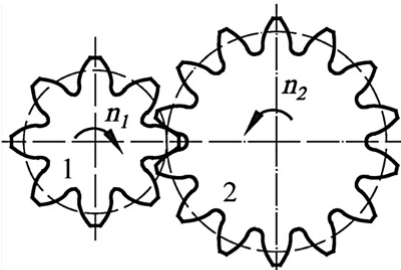
Mechanical drives with a constant transmission ratio are divided into: (i) drives with immovable axes – classical mechanical drives – and (ii) drives with movable axes – planetary mechanical trains.

According to the mode in which they transmit motion from the driving wheel to the driven one, **mechanical drives with immovable axes** fall into the following types:

- **Transmissions using friction:** with direct contact of wheels (friction drives) or with a flexible connection (belt drives).
- **Transmissions using mesh:** with direct contact (toothed and worm gears) or with flexible connection (chain and gear-belts drives).

This classification is clearly presented in Table 1.3.

Table 1.3 Main classification of classical mechanical drives with constant transmission ratio and immovable axes

		Mutual position of driving wheel and driven wheel	
		Indirect	Direct
Mode of motion transmission	Friction	 belt drive	 friction drive
	Mesh	 chain drive	 gear drive

Any individual drive is basically composed of a pair of wheels: gear wheels, rollers, belt pulleys and sprockets. It can operate on its own or can be built into mechanical trains of various machines and instruments and made in the form of an individual drive enclosed in a special housing. It can be also joined with other individual drives, making a multi-stage mechanical drive (see Section 1.4).

It is appropriate to mention that $i = \text{const}$ classical friction drives are no longer in use. They are included in the classification because they can be still found in machinery plants.

Figure 1.2 presents simple examples of the three main types of mechanical drives with immovable axes: gear drive with a constant transmission ratio (Figure 1.2a), belt drive with a stepless change of transmission ratio by the help of an axially movable belt (Figure 1.2b), gear drive with a step by step change of transmission ratio by the help of an axially movable set of gears 1 and 3 (Figure 1.2c).

Planetary mechanical trains are mechanical drives where at least one wheel (planet) has movable axes and, when meshed with one or more (central) wheels with immovable (common) axes, it turns around them. The planets are joined together into the planet carrier which has its own shaft. By the mode in which they transmit the motion, that is, by the type of wheel, planetary trains are divided into planetary gear trains (see Chapter 6) and planetary friction trains. Both of them have the same kinematics.

Except for their high price and the noise they produce, gear drives have an advantage over all other drives in all features, especially in operational safety and endurance, efficiency and smaller dimensions. This is why gear drives, although very exacting in design and

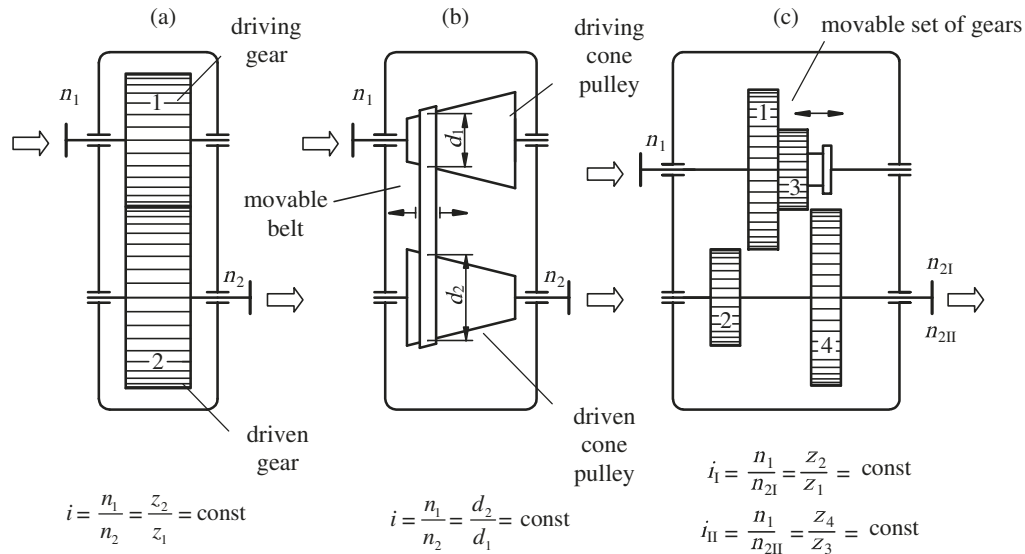


Figure 1.2 Sample types of mechanical drives with immovable axes: (a) constant transmission ratio, (b) stepless change of transmission ratio, (c) step by step change of transmission ratio

technology of production, make up approximately 80% of all mechanical drives (see diagram in Figure 1.3).

In **variable transmission ratio drives**, the constant speed n_1 of driving shaft rotation results in a variable speed n_2 of driven shaft rotation. If n_2 changes are *stepless*, such drives are called *continuous variable speed drives* or *variators*. The ratio drive is changed by a change in the mutual position of the wheels (in friction variators) or by a change in the belt position (in *belt variators*; see Figure 1.2b). Modern *gear planetary variators* are able to achieve a practically infinite range of regulation besides reverse.

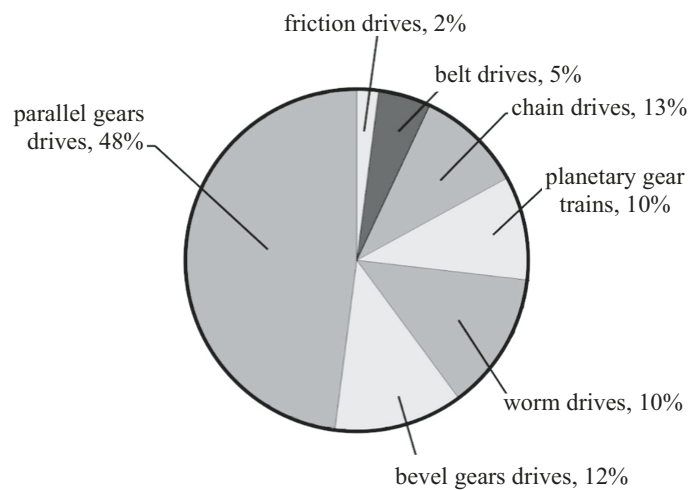


Figure 1.3 Global spread of mechanical drives towards the end of 2005

Mechanical drives with a *cyclically variable transmission ratio* are realized mostly by means of gear pairs with gears of an elliptic or hypocycloid form (see Figure 1.6). A cyclic rotary motion can also be achieved by a segment gear drive (see Figure 1.7a); and an alternate rectilinear motion can be obtained by engaging two toothed racks with driving segment gear (see Figure 1.7b).

There are also *gear drives with cyclically variable centre distance* which, beside a variable transmission ratio, enable a change in the direction of rotation of the driven shaft (reverse) (see Figure 1.10).

Mechanical drives with a *step by step ratio change* are mostly appropriate for the gearboxes of automotive vehicles and machine tools. They function by the principle of changeable or axially movable gears which produce a transmission ratio depending on which of them is in mesh, that is, depending on which of them transfers the motion (see Figure 1.2c).

There are also *planetary gear variators* which, coupled with friction variators, can reach a practically limitless regulation range and change in the direction of revolution (reverse). Planetary gear drives with a step by step change of transmission ratio are known as *planetary gear-boxes*.

The application of mechanical drives with a changeable transmission ratio is always adequate when the prime mover is not able to change the speed of rotation in the manner required by the operational machine. An accustomed application is found in constant power drives at a larger range of speed of rotation and always when it is necessary to obtain the designed high precision transmission ratios. It is important to mention that regulation and control of speed at constant torque is more efficient by electric motor regulation than by mechanical drive.

1.3 Choosing a Mechanical Drive

A basic presumption in choosing a mechanical drive is the designer's good knowledge of all types of drives: their features and applications. Only in such a case he can correctly predict what type of drive should be the most appropriate one when the prime mover and operational machine are already determined.

The first step in the process of mechanical drive design is to select as much data as possible in order to outline the drive and to dimension its parts correctly. This information refers to nine basic fields: basic function data of the drive, prime mover data, actuator data, drive manufacture data, load data, user requirements, lubrication, environment and assembly and maintenance.

Basic function data of a drive:

- Driving and driven axes rotational speeds and nature of transmission ratio: constant or not; stepped or not?
- Transmission ratio variation.
- Reverse: yes or no?
- Types of prime mover and operational machine and their power and torque characteristics.
- Ordering party requests with regard to type of drive, built in position, rotational speed limits, prime mover, establishments and so on.
- Operational machine position in regard to prime mover: is it changeable; what are the limits?

Prime mover data:

- Speed of rotation.
- Consequences of machine damage or breakdown.
- Is there a need to take into account the machine overthrowing?
- What are the sizes of intensity and frequency and duration of service impact?
- What maximum torque can the prime mover reach; and could it cause a catastrophe?
Is there an overload safeguard?

Operational machine (member) data:

- Speed of rotation and its change,
- Intensity, frequency and duration of service impact; load spectrum,
- Intermittency, that is, operational hours per day,
- Consequences of machine damage or breakdown.

Data needed for drive manufacture:

- Material selection limits,
- Tools and machines needed for drive manufacture,
- Limits in usage of machine tools, heat treatment furnaces and the like,
- Are there the technologies predicted that we do not master?

Load data:

- Is there any axial force on the driving or driven shaft?
- How is the drive fixed and what are the values of the forces arising from that?
- Other forces transferring to environment.

Ordering party requests, conditions of taking over:

- Driving and driven axes clutches,
- Design standards,
- Noise level, minimum efficiency, testing.
- Drive design requests: forged or welded gear body, cast or welded housing, joint between gear body and rim,
- Safety requests,
- Special directions in drive design,
- Conditions of drive transfer,
- Reserve price,
- Terms of delivery.

Lubrication:

- Is the designer limited in lubricant selection?
- Mineral or synthetic lubricant?

- Independent or central lubrication?
- Cooling requests: fresh- or sea-water, air or water emitted into the environment? What is the temperature of operation? Central cooling equipment or an independent unit?
- Drive operation control.

Environment, drive placing:

- Placing of the drive: open or enclosed space?
- Installation limits.
- Is there enough space around the drive?
- Substructures: are they taken together for the prime mover and actuator?
- Can there be difficulties in transferring the drive at the installation place?
- Room temperature.

Maintenance:

- Predicted type of maintenance.
- Lifetime of the drive and its components.
- Which diagnostic method is predicted?
- Is there a request for special access to certain components of the drive?
- Is there a request for a list of critical parts?
- Is it required to deliver the most necessary spare parts?
- How the drive should be coloured?

When there is agreement between the designer and the ordering party on the listed data and dilemmas, or at least on most of them, it is customary to write it in the form of a document. It is often only a morally required document but, by arrangement, it can also become a materially required document – a constituent part of the contract for the manufacture and delivery of the drive.

1.4 Multi-Step Drives

Simple mechanical drives with a single pair of wheels, like those depicted in Table 1.1, are called one-stage drives. Their transmission ratio is limited and a larger one is often required. The reduction is then obtained in a few stages in the way that the driven shaft of each particular stage is at the same time the driving shaft of the next stage. Such a drive is then called a multi-step drive.

It is understandable that each shaft can, in principle, drive one operational machine. In Figure 1.4 a fictitious, general multi-step mechanical drive is presented, which consists of prime mover PM, n wheels with corresponding shafts, that is, with $n/2$ stages and k operational machines. The first stage consists of the pair of spur gears 1, 2 and operational machine OM₁ connected to gear 2 shaft; the second stage is a pair of spur helical gears 3, 4 and operational machine OM₂ connected to gear 4 shaft and so on; the last stage is the V-belt drive with two V-belt pulleys, each with three V-belts transferring the power to the OM_k operational machine. The OM_{k-1} operational machine is driven by the $(n - 1)^{\text{th}}$ wheel.

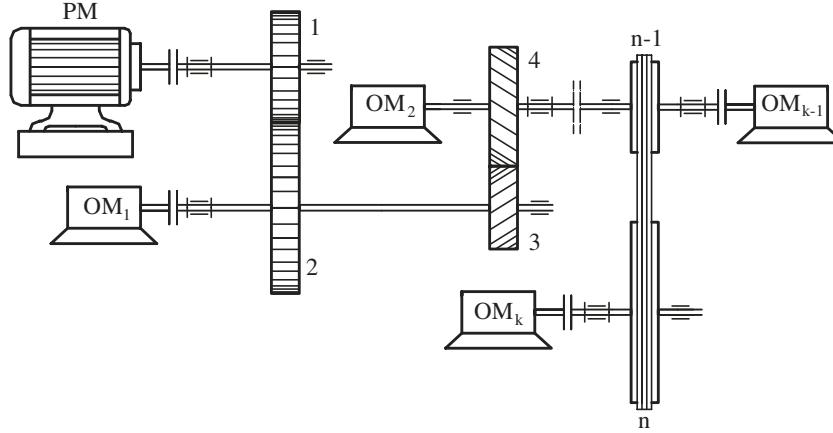


Figure 1.4 Schematic account of a multi-step drive

When designing multi-step drives it is necessary to determine the required power of the prime mover for the given power of the operational machine, the power losses (i.e. efficiency) and a transmission ratio. Also, the torques of any shaft must be determined in order to dimension all the related components. For such a purpose, it is necessary to identify the relation between the total transmission ratio and the partial ones within each stage, the relation between the total efficiency and the partial ones within each stage and the power balance, that is, the relation between the powers of the prime mover and the operational machine.

Total transmission ratio i_{tot} can be written as:

$$i_{tot} = i_{1,n} = \frac{n_1}{n_n} = \frac{\omega_1}{\omega_n} = \frac{\omega_1}{\omega_2} \cdot \frac{\omega_2}{\omega_4} \cdot \frac{\omega_4}{\omega_6} \cdot \dots \cdot \frac{\omega_{n-2}}{\omega_{n-1}} \cdot \frac{\omega_{n-1}}{\omega_n} \quad (1.3)$$

where the angular speeds of particular wheels are signed with ω . It follows that:

$$i_{tot} = i_{1,n} = i_{1,2} \cdot i_{3,4} \cdot i_{5,6} \cdot \dots \cdot i_{n-1,n} = i_I \cdot i_{II} \cdot i_{III} \cdot \dots \cdot i_{n/2} = \prod_{j=I}^{n/3} i_j \quad (1.4)$$

where the partial transmission ratios are signed with i_j . It is clear now that the total transmission ratio equals the product of the particular stages' partial transmission ratios.

In the same way the total efficiency is obtained:

$$\eta_u = \eta_{1,n} = \eta_{1,2} \cdot \eta_{3,4} \cdot \eta_{5,6} \cdot \dots \cdot \eta_{n-1,n} = \eta_I \cdot \eta_{II} \cdot \eta_{III} \cdot \dots \cdot \eta_{n/2} = \prod_{j=I}^{n/2} \eta_j \quad (1.5)$$

that is, the total efficiency equals the product of the particular stages' partial efficiencies.

From the mechanical drive energy balance, which declares that the energy produced in the prime mover equals the sum of energies spent by all operational machines plus their power losses, it follows that the *power balance* of a drive is the power of the prime mover P_{PS} which