

---

# **MECHANISMS AND MECHANICAL DEVICES SOURCEBOOK**

**Fourth Edition**

**NEIL SCLATER  
NICHOLAS P. CHIRONIS**

**McGraw-Hill**  
New York • Chicago • San Francisco • Lisbon • London • Madrid  
Mexico City • Milan • New Delhi • San Juan • Seoul  
Singapore • Sydney • Toronto

Copyright © 2007, 2001, 1996, 1991 by The McGraw-Hill Companies, Inc. All rights reserved. Printed in the United States of America. Except as permitted under the United States Copyright Act of 1976, no part of this publication may be reproduced or distributed in any form or by any means, or stored in a data base or retrieval system, without the prior written permission of the publisher.

1 2 3 4 5 6 7 8 9 0 BKM/BKM 0 1 2 1 0 9 8 7 6

ISBN-13: 978-0-07-146761-2

ISBN-10: 0-07-146761-0

*The sponsoring editor for this book was Larry S. Hager and the production supervisor was Pamela A. Pelton. It was set in Times by International Typesetting and Composition. The art director for the cover was Anthony Landi.*

*Printed and bound by BookMart Press.*

This book is printed on acid-free paper.

McGraw-Hill books are available at special quantity discounts to use as premiums and sales promotions, or for use in corporate training programs. For more information, please write to the Director of Special Sales, McGraw-Hill Professional, Two Penn Plaza, New York, NY 10121-2298. Or contact your local bookstore.

Information contained in this work has been obtained by The McGraw-Hill Companies, Inc. ("McGraw-Hill") from sources believed to be reliable. However, neither McGraw-Hill nor its authors guarantee the accuracy or completeness of any information published herein, and neither McGraw-Hill nor its authors shall be responsible for any errors, omissions, or damages arising out of use of this information. This work is published with the understanding that McGraw-Hill and its authors are supplying information but are not attempting to render engineering or other professional services. If such services are required, the assistance of an appropriate professional should be sought.

# PREFACE

This is the fourth edition of *Mechanisms and Mechanical Devices Sourcebook*, a well-illustrated reference book containing a wide range of information on both classical and modern mechanisms and mechanical devices. This edition contains three new chapters: one on basic mechanisms; the second on mobile robots; and the third on new directions in mechanical engineering. The chapter on basic mechanisms provides an overview of the physical principles of mechanics; the chapter on mobile robots examines existing scientific and military mobile robots and the scientific and engineering research in advanced robotics; the chapter on new directions in mechanical engineering reviews the present status and future prospects for microtechnology, highlighting progress in and acceptance of microelectromechanical systems (MEMS). Also included in the chapter are articles on nanotechnology, focused on the role mechanical engineers are playing in this burgeoning science. The field of nanotechnology now involves several branches of engineering as well as the physical, chemical, biological, and medical sciences. A previous section on rapid prototyping has been updated and upgraded as a separate chapter.

This edition contains a large core of archival drawings and text describing and illustrating proven mechanisms and mechanical devices carried over from previous editions. This core has been reorganized to make topics of interest to readers easier to find. Some previously published pages were deleted because their content was deemed to be of little value in future designs, and some figures have been redrawn to make them easier to understand. An extensive and comprehensive index has been provided to make this core a valuable reference resource for engineers, designers, inventors, students, hobbyists, and all enthusiasts for things mechanical. The 11 chapters in this core illustrate practical design solutions that can be recycled into new products.

The first edition of this book, published in 1991, did not mention the influence of electronics and computer science on mechanical engineering and mechanical design. However, since that time a sea change has occurred in the practice of mechanical engineering; today it is difficult to find any contemporary mechanical system or appliance that does not in some way include electronic components or circuits that improve its performance, simplify its operation, or provide for additional safety features. Those components might be as simple as solid-state rectifiers or light-emitting diodes (LEDs) or as complex as microprocessor-based modules that permit the product or system to operate autonomously.

The chapter on basic mechanisms provides the reader with a useful introduction to much of the content of this book; it will also serve as a refresher tutorial for those who have studied mechanical principles in the past and want to get up to speed on the fundamentals again. Topics covered include the inclined plane, screw jack, levers, linkages, gears, cams, and clutches. A previous tutorial chapter on motion control systems that contained illustrations and text describing control schemes and key components has been retained, and a former chapter on industrial robots has been revised and updated with new illustrations and specifications for some of the latest industrial robots. The new chapter on mobile robots extends the book's coverage of robotics and points out their growing economic and technical importance in scientific exploration and research as well as military missions and emergency services.

The new chapter on rapid prototyping discusses the emerging leaders in the field and reports on the trends: increasing popularity of 3-D plastic, paper, and wax models for engineering and design evaluation, and the extrapolation of existing technologies into the fabrication of functional metal and ceramic products. Replacement metal parts for older out-of-production machines are now being made rapidly and cost-effectively by eliminating the high cost and time delay involved in remaking the metal or ceramic dies or casting molds used in mass-production manufacturing.

The earlier articles on MEMS have been revised by reporting on the new developments and significant gains in the complexity of those devices; some MEMS are now being produced in large commercial volumes in established markets. The choices in material alternatives to silicon are discussed, and new microphotographs show more sophisticated multilayer devices.

The impact of electronic controls and communications circuits on mechanical engineering is nowhere more evident than on the latest motor vehicles. Microprocessors and electronics abound: they now control the engines and transmissions in all kinds of motor vehicles, and they have improved vehicle performance and fuel efficiency. Vehicular safety has also been improved by electronically deployed air-bags, antilock braking (ABS), stability or skid control (ESC), traction control (TC), and tire-pressure monitoring. Communication systems summon aid for drivers involved in accidents or breakdowns, and onboard navigation systems now provide map displays of streets to guide drivers.

With the exception of illustrations generously contributed by corporations, and government laboratories (see Acknowledgments), all of the figures in the tutorial Chapters 1 to 4 and 18 and 19 were drawn by this author on a Dell personal computer with software included in the Microsoft Windows XP package. Also, the five illustrations on the front cover of this book were derived from selected figures in those chapters.

Much of the archival core in this edition was first collected from a variety of published sources by Douglas C. Greenwood, then an editor of *Product Engineering* magazine; it first appeared in three volumes published by McGraw-Hill between 1959 and 1964. Nicholas Chironis edited and reorganized much of this content and supplemented it with contemporary technical articles to form the first edition of this book. In subsequent editions this core has been reorganized and new material has been added. References to manufacturers or publications that no longer exist have since been deleted because they are no longer valid sources for further information. The terms *devices* and *mechanisms* used to describe objects in the core pages have been used interchangeably and only some of them have been changed. However, the comprehensive index accounts for these differences in designation. The names of the inventors of these mechanisms and devices have been retained so that readers can research the status of any patents once held by them.

—Neil Sclater

# CONTENTS

|                                                                   |             |
|-------------------------------------------------------------------|-------------|
| <b>PREFACE</b>                                                    | <b>xiii</b> |
| <b>ACKNOWLEDGMENTS</b>                                            | <b>xv</b>   |
| <b>CHAPTER 1 BASICS OF MECHANISMS</b>                             | <b>1</b>    |
| Introduction                                                      | 2           |
| Physical Principles                                               | 2           |
| Inclined Plane                                                    | 3           |
| Pulley Systems                                                    | 3           |
| Screw-Type Jack                                                   | 4           |
| Levers and Mechanisms                                             | 4           |
| Linkages                                                          | 5           |
| Specialized Mechanisms                                            | 9           |
| Gears and Gearing                                                 | 10          |
| Pulleys and Belts                                                 | 14          |
| Sprockets and Chains                                              | 14          |
| Cam Mechanisms                                                    | 14          |
| <b>CHAPTER 2 MOTION CONTROL SYSTEMS</b>                           | <b>21</b>   |
| Motion Control Systems Overview                                   | 22          |
| Glossary of Motion Control Terms                                  | 28          |
| Mechanical Components form Specialized Motion-Control Systems     | 29          |
| Servomotors, Stepper Motors, and Actuators for Motion Control     | 30          |
| Servosystem Feedback Sensors                                      | 38          |
| Solenoids and Their Applications                                  | 45          |
| <b>CHAPTER 3 INDUSTRIAL ROBOTS</b>                                | <b>49</b>   |
| Introduction to Robots                                            | 50          |
| Industrial Robots                                                 | 51          |
| Mechanism for Planar Manipulation with Simplified Kinematics      | 60          |
| Tool-Changing Mechanism for Robot                                 | 61          |
| Piezoelectric Motor in Robot Finger Joint                         | 62          |
| Self-Reconfigurable, Two-Arm Manipulator with Bracing             | 63          |
| Improved Roller and Gear Drives for Robots and Vehicles           | 64          |
| Glossary of Robotic Terms                                         | 65          |
| <b>CHAPTER 4 MOBILE SCIENTIFIC, MILITARY, AND RESEARCH ROBOTS</b> | <b>67</b>   |
| Introduction to Mobile Robots                                     | 68          |
| Scientific Mobile Robots                                          | 69          |
| Military Mobile Robots                                            | 70          |
| Research Mobile Robots                                            | 72          |
| Second-Generation Six-Limbed Experimental Robot                   | 76          |
| All-Terrain Vehicle with Self-Righting and Pose Control           | 77          |
| <b>CHAPTER 5 LINKAGES: DRIVES AND MECHANISMS</b>                  | <b>79</b>   |
| Four-Bar Linkages and Typical Industrial Applications             | 80          |
| Seven Linkages for Transport Mechanisms                           | 82          |
| Five Linkages for Straight-Line Motion                            | 85          |
| Six Expanding and Contracting Linkages                            | 87          |

|                                                                      |            |
|----------------------------------------------------------------------|------------|
| Four Linkages for Different Motions                                  | 88         |
| Nine linkages for Accelerating and Decelerating linear Motions       | 89         |
| Twelve Linkages for Multiplying Short Motions                        | 91         |
| Four Parallel-Link Mechanisms                                        | 93         |
| Seven Stroke Multiplier Linkages                                     | 93         |
| Nine Force and Stroke Multiplier Linkages                            | 95         |
| Eighteen Variations of Differential Linkage                          | 97         |
| Four-Bar Space Mechanisms                                            | 99         |
| Seven Three-Dimensional Linkage Drives                               | 101        |
| Thirteen Different Toggle Linkage Applications                       | 106        |
| Hinged Links and Torsion Bushings Soft-Start Drives                  | 108        |
| Eight Linkages for Band Clutches and Brakes                          | 109        |
| Design of Crank-and-Rocker Links for Optimum<br>Force Transmission   | 111        |
| Design of Four-Bar Linkages for Angular Motion                       | 114        |
| Multi-Bar Linkages for Curvilinear Motions                           | 115        |
| Roberts' Law Helps to Design Alternate Four-Bar Linkages             | 118        |
| Slider-Crank Mechanism                                               | 119        |
| <b>CHAPTER 6 GEARS: DEVICES, DRIVES, AND MECHANISMS</b>              | <b>121</b> |
| Gears and Eccentric Disk Provide Quick Indexing                      | 122        |
| Odd-Shaped Planetary Gears Smooth Stop and Go                        | 123        |
| Cycloid Gear Mechanism Controls Pump Stroke                          | 126        |
| Gears Convert Rotary-to-Linear Motion                                | 127        |
| Twin-Motor Planetary Gears Offer Safety and Dual-Speed               | 127        |
| Eleven Cycloid Gear Mechanisms                                       | 128        |
| Five Cardan-Gear Mechanisms                                          | 131        |
| Controlled Differential Gear Drives                                  | 133        |
| Flexible Face-Gears are Efficient High-Ratio Speed Reducers          | 134        |
| Rotary Sequencer Gears Turn Coaxially                                | 135        |
| Planetary Gear Systems                                               | 136        |
| Noncircular Gears are Balanced for Speed                             | 143        |
| Sheet-Metal Gears, Sprockets, Worms, and Ratchets<br>for Light Loads | 147        |
| Thirteen Ways Gears and Clutches Can Change Speed Ratios             | 149        |
| Gear and Clutch Shifting Mechanisms                                  | 151        |
| Twinworm Gear Drive Offers Bidirectional Output                      | 153        |
| Elastomeric Gear Bearings Equalize Torque Loads                      | 154        |
| Redundant Gearing in Helicopter Transmits Torque                     | 155        |
| Worm Gear Friction Reduced by Oil Pressure                           | 156        |
| Bevel and Hypoid Gear Design Prevents Undercutting                   | 157        |
| Geared Electromechanical Rotary Joint                                | 158        |
| Geared Speed Reducers Offer One-Way Output                           | 159        |
| Design of Geared Five-Bar Mechanisms                                 | 160        |
| Equations for Designing Geared Cycloid Mechanisms                    | 164        |
| Design Curves and Equations for Gear-Slider Mechanisms               | 167        |
| <b>CHAPTER 7 CAM, GENEVA, AND RATCHET DRIVES<br/>AND MECHANISMS</b>  | <b>171</b> |
| Cam-Controlled Planetary Gear System                                 | 172        |
| Five Cam-Stroke-Amplifying Mechanisms                                | 173        |
| Cam-Curve-Generating Mechanisms                                      | 174        |
| Fifteen Different Cam Mechanisms                                     | 180        |
| Ten Special-Function Cams                                            | 182        |
| Twenty Geneva Drives                                                 | 184        |
| Six Modified Geneva Drives                                           | 188        |

|                   |                                                                 |            |
|-------------------|-----------------------------------------------------------------|------------|
|                   | Kinematics of External Geneva Wheels                            | 190        |
|                   | Kinematics of Internal Geneva Wheels                            | 193        |
|                   | Star Wheels Challenge Geneva Drives for Indexing                | 197        |
|                   | Ratchet-Tooth Speed-Change Drive                                | 200        |
|                   | Modified Ratchet Drive                                          | 200        |
|                   | Eight Toothless Ratchets                                        | 201        |
|                   | Analysis of Ratchet Wheels                                      | 202        |
| <b>CHAPTER 8</b>  | <b>CLUTCHES AND BRAKES</b>                                      | <b>203</b> |
|                   | Twelve Clutches with External or Internal Control               | 204        |
|                   | Spring-Wrapped Clutch Slips at Preset Torque                    | 206        |
|                   | Controlled-Slip Expands Spring Clutch Applications              | 208        |
|                   | Spring Bands Improve Overrunning Clutch                         | 209        |
|                   | Slip and Bidirectional Clutches Combine to Control Torque       | 210        |
|                   | Walking Pressure Plate Delivers Constant Torque                 | 211        |
|                   | Seven Overrunning Clutches                                      | 212        |
|                   | One-Way Clutch has Spring-Loaded Pins and Sprags                | 213        |
|                   | Roller Clutch provides Two Output Speeds                        | 213        |
|                   | Seven Overriding Clutches                                       | 214        |
|                   | Ten Applications for Overrunning Clutches                       | 216        |
|                   | Eight Sprag Clutch Applications                                 | 218        |
|                   | Six Small Clutches Perform Precise Tasks                        | 220        |
|                   | Twelve Different Station Clutches                               | 222        |
|                   | Twelve Applications for Electromagnetic Clutches and Brakes     | 225        |
|                   | Roller Locking Mechanism Contains Two Overrunning Clutches      | 227        |
| <b>CHAPTER 9</b>  | <b>LATCHING, FASTENING, AND CLAMPING DEVICES AND MECHANISMS</b> | <b>229</b> |
|                   | Sixteen Latch, Toggle, and Trigger Devices                      | 230        |
|                   | Fourteen Snap-Action Devices                                    | 232        |
|                   | Remote Controlled Latch                                         | 236        |
|                   | Toggle Fastener Inserts, Locks, and Releases Easily             | 237        |
|                   | Grapple Frees Loads Automatically                               | 237        |
|                   | Quick-Release Lock Pin has a Ball Detent                        | 238        |
|                   | Automatic Brake Locks Hoist when Driving Torque Ceases          | 238        |
|                   | Lift-Tong Mechanism Firmly Grips Objects                        | 239        |
|                   | Perpendicular-Force Latch                                       | 239        |
|                   | Two Quick-Release Mechanisms                                    | 240        |
|                   | Ring Springs Clamp Platform Elevator into Position              | 241        |
|                   | Cammed Jaws in Hydraulic Cylinder Grip Sheet Metal              | 241        |
|                   | Quick-Acting Clamps for Machines and Fixtures                   | 242        |
|                   | Nine Friction Clamping Devices                                  | 244        |
|                   | Detents for Stopping Mechanical Movements                       | 246        |
|                   | Twelve Clamping Methods for Aligning Adjustable Parts           | 248        |
|                   | Spring-Loaded Chucks and Holding Fixtures                       | 250        |
| <b>CHAPTER 10</b> | <b>CHAIN AND BELT DEVICES AND MECHANISMS</b>                    | <b>251</b> |
|                   | Twelve Variable-Speed Belt and Chain Drives                     | 252        |
|                   | Belts and Chains are Available                                  |            |
|                   | in Manydifferent Forms                                          | 255        |
|                   | Change Center Distance without Altering Speed Ratio             | 259        |
|                   | Motor Mount Pivots to Control Belt Tension                      | 259        |
|                   | Ten Roller Chains and their Adaptations                         | 260        |
|                   | Twelve Applications for Roller Chain                            | 262        |
|                   | Six Mechanisms for Reducing Pulsations in Chain Drives          | 266        |

|                   |                                                                               |            |
|-------------------|-------------------------------------------------------------------------------|------------|
| <b>CHAPTER 11</b> | <b>SPRING AND SCREW DEVICES AND MECHANISMS</b>                                | <b>269</b> |
|                   | Flat Springs in Mechanisms                                                    | 270        |
|                   | Twelve Ways to Use Metal Springs                                              | 272        |
|                   | Seven Overriding Spring Mechanisms for Low-Torque Drives                      | 274        |
|                   | Six Spring Motors and Associated Mechanisms                                   | 276        |
|                   | Twelve Air Spring Applications                                                | 278        |
|                   | Novel Applications for Different Springs                                      | 280        |
|                   | Applications for Belleville Springs                                           | 281        |
|                   | Vibration Control with Spring Linkage                                         | 282        |
|                   | Twenty Screw Devices                                                          | 283        |
|                   | Ten Applications for Screw Mechanisms                                         | 285        |
|                   | Seven Special Screw Arrangements                                              | 287        |
|                   | Fourteen Spring and Screw adjusting Devices                                   | 288        |
| <b>CHAPTER 12</b> | <b>SHAFT COUPLINGS AND CONNECTIONS</b>                                        | <b>289</b> |
|                   | Four Couplings for Parallel Shafts                                            | 290        |
|                   | Links and Disks Couple Offset Shafts                                          | 291        |
|                   | Disk-and-Link Couplings Simplify Torque Transmission                          | 292        |
|                   | Interlocking Space-Frames Flex as they Transmit Shaft Torque                  | 293        |
|                   | Coupling with Off-Center Pins Connects Misaligned Shafts                      | 295        |
|                   | Universal Joint Transmits Torque 45° At Constant Speed                        | 296        |
|                   | Ten Universal Shaft Couplings                                                 | 297        |
|                   | Nineteen Methods for Coupling Rotating Shafts                                 | 299        |
|                   | Five Different Pin-and-Link Couplings                                         | 303        |
|                   | Ten Different Splined Connections                                             | 304        |
|                   | Fourteen Ways to Fasten Hubs to Shafts                                        | 306        |
| <b>CHAPTER 13</b> | <b>MOTION-SPECIFIC DEVICES,<br/>MECHANISMS, AND MACHINES</b>                  | <b>309</b> |
|                   | Timing Belts, Four-Bar linkage Team Up for Smooth Indexing                    | 310        |
|                   | Ten Indexing and Intermittent Mechanisms                                      | 312        |
|                   | Twenty-Seven Rotary-to-Reciprocating Motion and Dwell Mechanisms              | 314        |
|                   | Five Friction Mechanisms for Intermittent Rotary Motion                       | 320        |
|                   | Nine Different Ball Slides for Linear Motion                                  | 322        |
|                   | Ball-Bearing Screws Convert Rotary to Linear Motion                           | 324        |
|                   | Nineteen Arrangements for Changing Linear Motion                              | 325        |
|                   | Five Adjustable-Output Mechanisms                                             | 329        |
|                   | Four Different Reversing Mechanisms                                           | 331        |
|                   | Ten Mechanical Computing Mechanisms                                           | 332        |
|                   | Seven Different Mechanical Power Amplifiers                                   | 336        |
|                   | Forty-Three Variable-Speed Drives<br>and Transmissions                        | 339        |
|                   | Ten Variable-Speed Friction Drives                                            | 351        |
|                   | Four Drives Convert Oscillating Motion to One-Way Rotation                    | 353        |
|                   | Operating Principles of Liquid, Semisolid, and Vacuum Pumps                   | 355        |
|                   | Twelve Different Rotary-Pump Actions                                          | 359        |
| <b>CHAPTER 14</b> | <b>PACKAGING, CONVEYING, HANDLING,<br/>AND SAFETY MECHANISMS AND MACHINES</b> | <b>361</b> |
|                   | Fifteen Devices that Sort, Feed, or Weigh                                     | 362        |
|                   | Seven Cutting Mechanisms                                                      | 366        |
|                   | Two Flipping Mechanisms                                                       | 368        |
|                   | One Vibrating Mechanism                                                       | 368        |
|                   | Seven Basic Parts Selectors                                                   | 369        |
|                   | Eleven Parts-Handling Mechanisms                                              | 370        |
|                   | Seven Automatic-Feed Mechanisms                                               | 372        |

|                                                                                                |            |
|------------------------------------------------------------------------------------------------|------------|
| Fifteen Conveyor Systems for Production Machines                                               | 375        |
| Seven Traversing Mechanisms<br>for Winding Machines                                            | 379        |
| Vacuum Pickup for Positioning Pills                                                            | 381        |
| Machine Applies Labels from Stacks or Rollers                                                  | 381        |
| Twenty High-Speed Machines for Applying Adhesives                                              | 382        |
| Twenty-Four Automatic Mechanisms for Stopping<br>Unsafe Machines                               | 388        |
| Six Automatic Electrical Circuits for<br>Stopping Textile Machines                             | 394        |
| Six Automatic Mechanisms for Assuring<br>Safe Machine Operation                                | 396        |
| <b>CHAPTER 15 TORQUE, SPEED, TENSION, AND LIMIT<br/>CONTROL SYSTEMs</b>                        | <b>399</b> |
| Applications of the Differential Winch to Control Systems                                      | 400        |
| Six Ways to Prevent Reverse Rotation                                                           | 402        |
| Caliper Brakes Keep Paper Tension in Web Presses                                               | 403        |
| Control System for Paper Cutting                                                               | 403        |
| Warning System Prevents Overloading of Boom                                                    | 404        |
| Lever System Monitors Cable Tension                                                            | 404        |
| Eight Torque-Limiters Protect Light-Duty Drives                                                | 405        |
| Thirteen Limiters Prevent Overloading                                                          | 406        |
| Seven Ways to Limit Shaft Rotation                                                             | 409        |
| Mechanical Systems for Controlling Tension and Speed                                           | 409        |
| Nine Drives for Controlling Tension                                                            | 415        |
| Limit Switches in Machinery                                                                    | 418        |
| Nine Automatic Speed Governors                                                                 | 422        |
| Eight Speed Control Devices for Mechanisms                                                     | 424        |
| <b>CHAPTER 16 INSTRUMENTS AND CONTROLS: PNEUMATIC,<br/>HYDRAULIC, ELECTRIC, AND ELECTRONIC</b> | <b>425</b> |
| Twenty-Four Mechanisms Actuated by Pneumatic or Hydraulic Cylinders                            | 426        |
| Foot-Controlled Braking System                                                                 | 428        |
| Fifteen Tasks for Pneumatic Power                                                              | 428        |
| Ten Applications for Metal Diaphragms and Capsules                                             | 430        |
| Nine Differential Transformer Sensors                                                          | 432        |
| High-Speed Electronic Counters                                                                 | 434        |
| Applications for Permanent Magnets                                                             | 435        |
| Nine Electrically Driven Hammers                                                               | 438        |
| Sixteen Thermostatic Instruments and Controls                                                  | 440        |
| Eight Temperature-Regulating Controls                                                          | 444        |
| Seven Photoelectric Controls                                                                   | 446        |
| Liquid Level Indicators and Controllers                                                        | 448        |
| Applications for Explosive-Cartridge Devices                                                   | 450        |
| Centrifugal, Pneumatic, Hydraulic, and Electric Governors                                      | 452        |
| <b>CHAPTER 17 COMPUTER-AIDED DESIGN CONCEPTS</b>                                               | <b>455</b> |
| Introduction to Computer-Aided Design                                                          | 456        |
| <b>CHAPTER 18 RAPID PROTOTYPING</b>                                                            | <b>461</b> |
| Rapid Prototyping Focuses on Building<br>Functional Parts                                      | 462        |
| Rapid Prototype Processes                                                                      | 462        |
| Rapid Prototyping Steps                                                                        | 463        |
| Commercial Rapid Prototyping Choices                                                           | 463        |

|                   |                                                                 |            |
|-------------------|-----------------------------------------------------------------|------------|
| <b>CHAPTER 19</b> | <b>NEW DIRECTIONS IN MECHANICAL ENGINEERING</b>                 | <b>473</b> |
|                   | The Role of Microtechnology in Mechanical Engineering           | 474        |
|                   | Micromachines Open a New Frontier for Machine Design            | 476        |
|                   | Multilevel Fabrication Permits more Complex and Functional MEMS | 480        |
|                   | Gallery of MEMS Electron-Microscope Images                      | 480        |
|                   | MEMS Chips Become Integrated Microcontrol Systems               | 484        |
|                   | Alternative Materials for Building MEMS                         | 486        |
|                   | LIGA: An Alternative Method for Making Microminiature Parts     | 487        |
|                   | Miniature Multispeed Transmissions for Small Motors             | 488        |
|                   | The Role of Nanotechnology in Mechanical Engineering            | 489        |
|                   | What are Carbon Nanotubes?                                      | 491        |
|                   | Nanoactuators Based on Electrostatic Forces on Dielectrics      | 492        |
| <b>INDEX</b>      |                                                                 | <b>495</b> |

# **CHAPTER 12**

---

# **SHAFT COUPLINGS AND CONNECTIONS**

# FOUR COUPLINGS FOR PARALLEL SHAFTS

**Fig. 1** One method of coupling shafts makes use of gears that can replace chains, pulleys, and friction drives. Its major limitation is the need for adequate center distance. However, an idler can be used for close centers, as shown. This can be a plain pinion or an internal gear. Transmission is at a constant velocity and there is axial freedom.

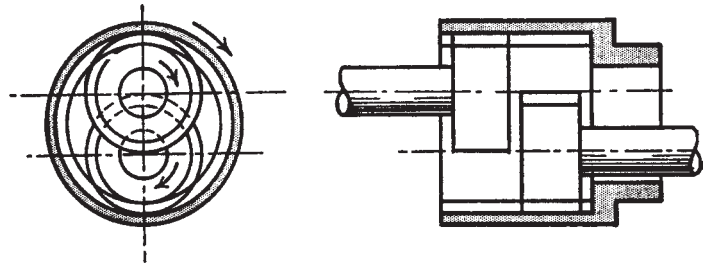


**Fig. 2** This coupling consists of two universal joints and a short shaft. Velocity transmission is constant between the input and output shafts if the shafts remain parallel and if the end yokes are arranged symmetrically. The velocity of the central shaft fluctuates during rotation, but high speed and wide angles can cause vibration. The shaft offset can be varied, but axial freedom requires that one shaft be spline mounted.

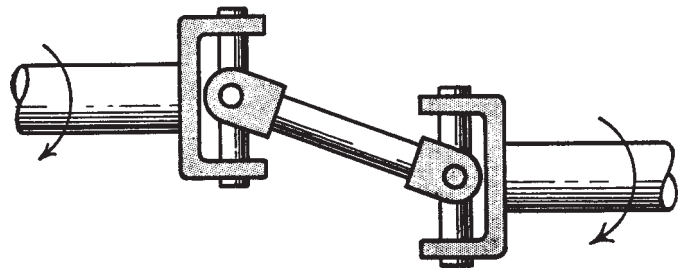


**Fig. 3** This crossed-axis yoke coupling is a variation of the mechanism shown in Fig. 2. Each shaft has a yoke connected so that it can slide along the arms of a rigid cross member. Transmission is at a constant velocity, but the shafts must remain parallel, although the offset can vary. There is no axial freedom. The central cross member describes a circle and is thus subjected to centrifugal loads.

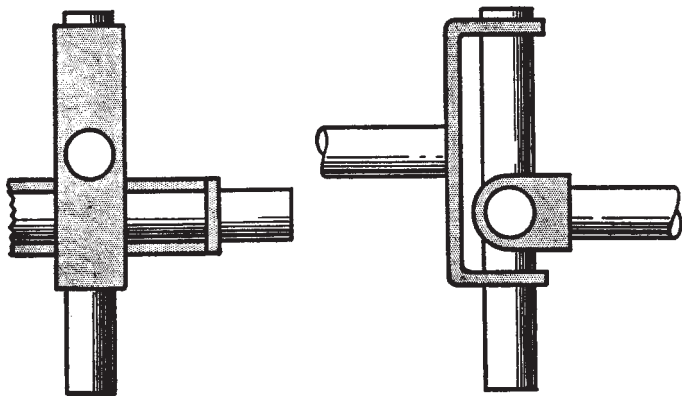
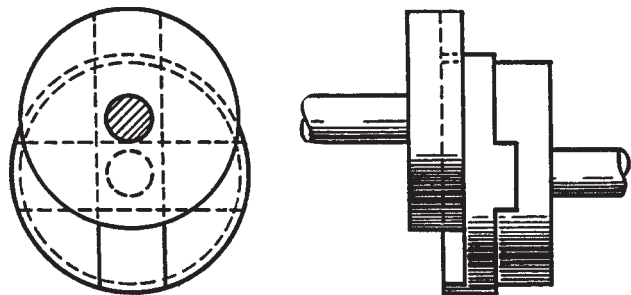


**Fig. 4** This Oldham coupling provides motion at a constant velocity as its central member describes a circle. The shaft offset can vary, but the shafts must remain parallel. A small amount of axial freedom is possible. A tilt in the central member can occur because of the offset of the slots. This can be eliminated by enlarging its diameter and milling the slots in the same transverse plane.



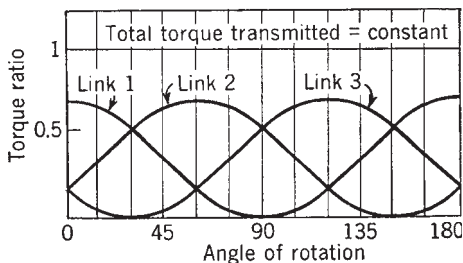
# LINKS AND DISKS COUPLE OFFSET SHAFTS

An unorthodox yet remarkably simple arrangement of links and disks forms the basis of a versatile parallel-shaft coupling. This coupling—essentially three disks rotating in unison and interconnected in series by six links (see drawing)—can adapt to wide variations in axial displacement while it is running under load.

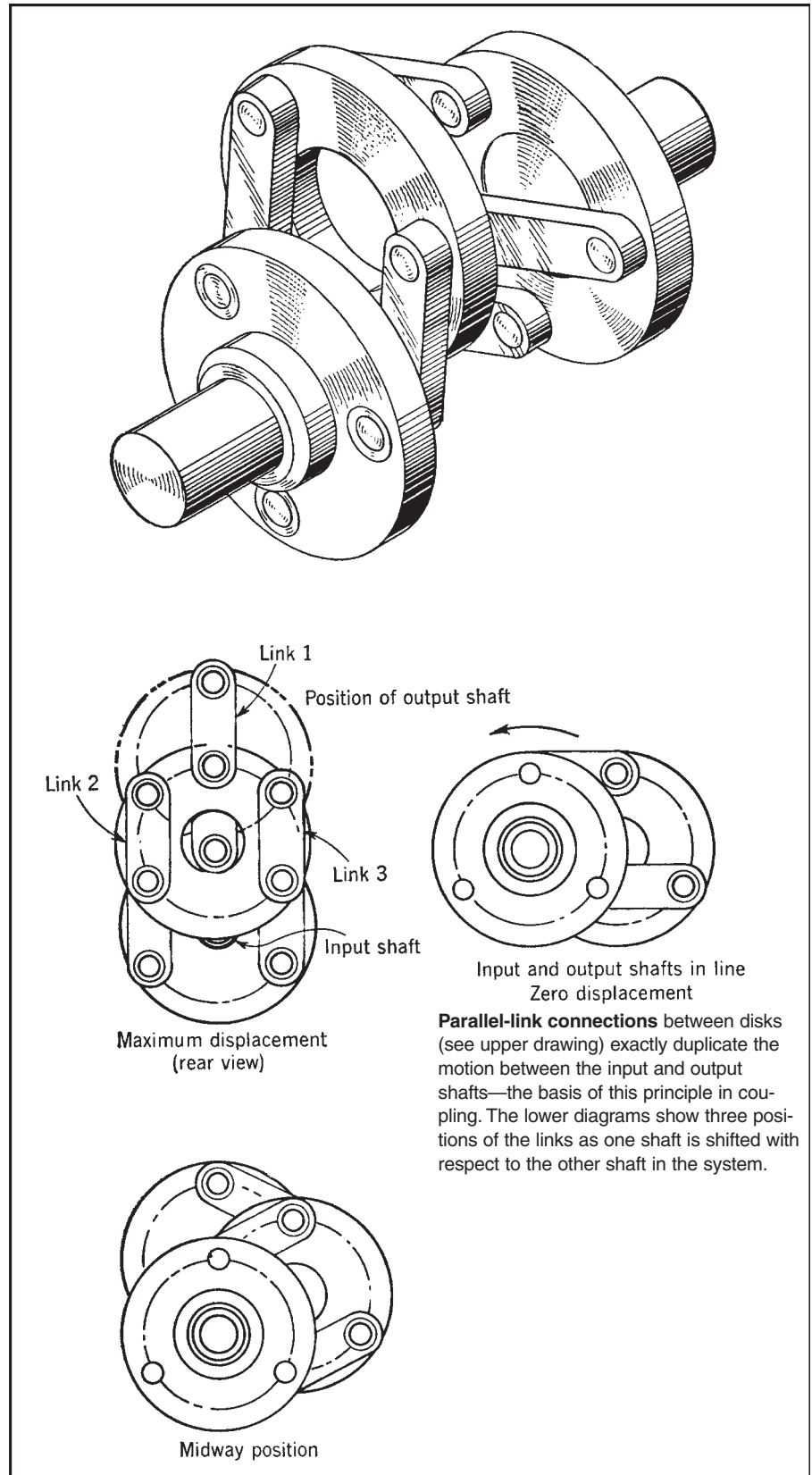
Changes in radial displacement do not affect the constant-velocity relationship between the input and output shafts, nor do they affect initial radial reaction forces that might cause imbalance in the system. Those features open up unusual applications for it in automotive, marine, machine-tool, and rolling-mill machinery (see drawings).

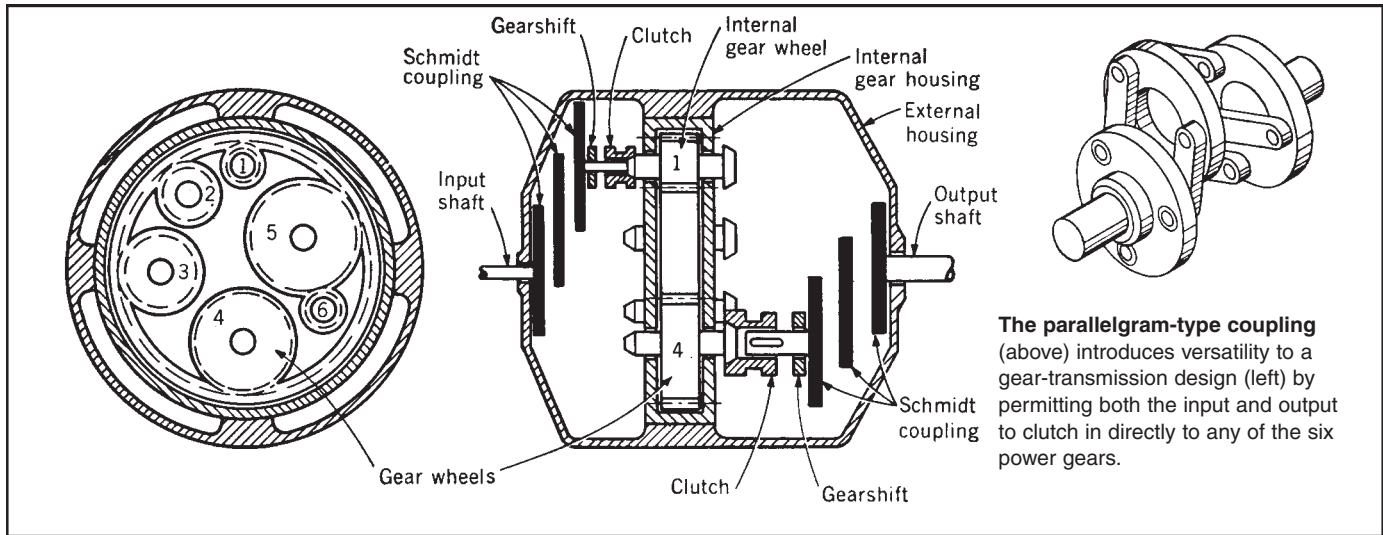
**How it works.** The inventor of the coupling, Richard Schmidt of Madison, Alabama, said that a similar link arrangement had been known to some German engineers for years. But those engineers were discouraged from applying the theory because they erroneously assumed that the center disk had to be retained by its own bearing. Actually, Schmidt found that the center disk is free to assume its own center of rotation. In operation, all three disks rotate with equal velocity.

The bearing-mounted connections of links to disks are equally spaced at  $120^\circ$  on pitch circles of the same diameter. The distance between shafts can be varied steplessly between zero (when the shafts are in line) and a maximum that is twice the length of the links (see drawings.) There is no phase shift between shafts while the coupling is undulating.



**Torque transmitted** by three links in the group adds up to a constant value, regardless of the angle of rotation.





## DISK-AND-LINK COUPLINGS SIMPLIFY TORQUE TRANSMISSION

A unique disk-and-link coupling that can handle large axial displacement between shafts, while the shafts are running under load, has opened up new approaches to transmission design. It was developed by Richard Schmidt of Madison, Alabama.

The coupling (drawing, upper right) maintains a constant transmission ratio between input and output shafts while the shafts undergo axial shifts in their relative positions. This permits gear-and-belt transmissions to be designed that need fewer gears and pulleys.

**Half as many gears.** In the internal-gear transmission shown, a Schmidt

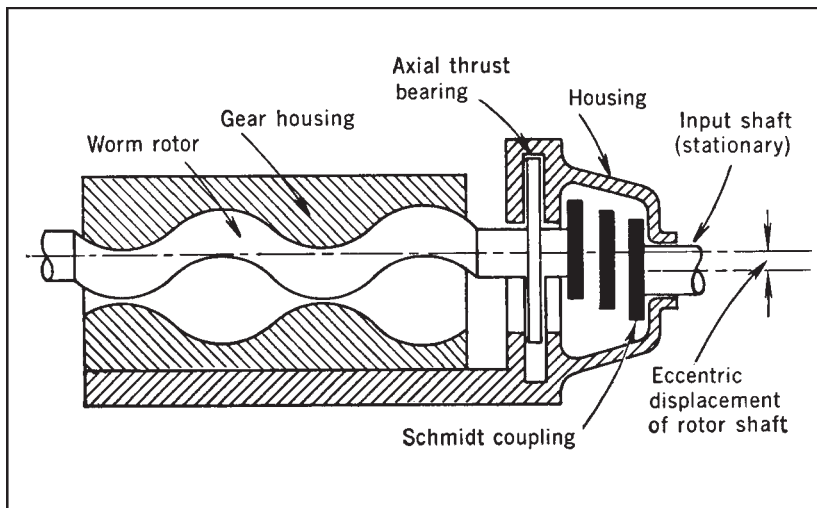
coupling on the input side permits the input to be plugged in directly to any one of six gears, all of which are in mesh with the internal gear wheel.

On the output side, after the power flows through the gear wheel, a second Schmidt coupling permits a direct power takeoff from any of the same six gears. Thus, any one of  $6 \times 6$  minus 5 or 31 different speed ratios can be selected while the unit is running. A more orthodox design would require almost twice as many gears.

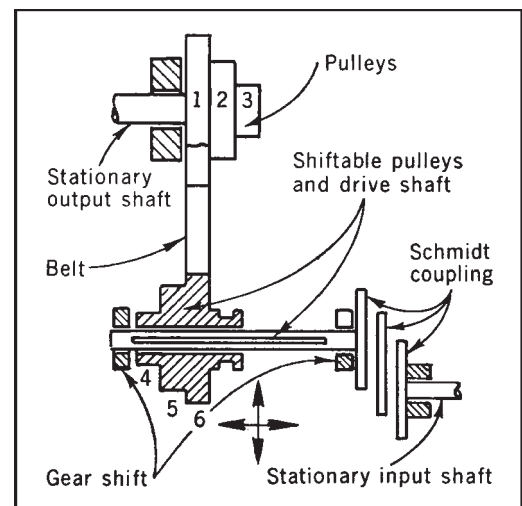
**Powerful pump.** In the worm-type pump (bottom left), as the input shaft rotates clockwise, the worm rotor is

forced to roll around the inside of the gear housing, which has a helical groove running from end to end. Thus, the rotor center-line will rotate counterclockwise to produce a powerful pumping action for moving heavy liquids.

In the belt drive (bottom right), the Schmidt coupling permits the belt to be shifted to a different bottom pulley while remaining on the same top pulley. Normally, because of the constant belt length, the top pulley would have to be shifted too, to provide a choice of only three output speeds. With this arrangement, nine different output speeds can be obtained.



**The coupling** allows a helically-shaped rotor to oscillate for pumping purposes.



**This coupling** takes up slack when the bottom shifts.

# INTERLOCKING SPACE-FRAMES FLEX AS THEY TRANSMIT SHAFT TORQUE

---

This coupling tolerates unusually high degrees of misalignment, with no variation in the high torque that's being taken from the shaft.

A concept in flexible drive-shaft couplings permits unusually large degrees of misalignment and axial motion during the transmission of high amounts of torque. Moreover, the rotational velocity of the driven member remains constant during transmission at angular misalignments; in other words, cyclic pulsations are not induced as they would be if, say, a universal coupling or a Hooke's joint were employed.

The coupling consists essentially of a series of square space-frames, each bent to provide offsets at the diagonals and each bolted to adjacent members at alternate diagonals. The concept was invented by Robert B. Bossler, Jr. He was granted U.S. Patent No. 3,177,684.

Couplings accommodate the inevitable misalignments between rotating shafts in a driven train. These misalignments are caused by imperfect parts, dimensional variations, temperature changes, and deflections of the supporting structures. The couplings accommodate misalignment either with moving contacts or by flexing.

Most couplings, however, have parts with moving contacts that require lubrication and maintenance. The rubbing parts also absorb power. Moreover, the lubricant and the seals limit the coupling environment and coupling life. Parts wear out, and the coupling can develop a large resistance to movement as the parts deteriorate. Then, too, in many designs, the coupling does not provide true constant velocity.

**For flexibility.** Bossler studied the various types of couplings in the market and first developed a new one with a moving contact. After exhaustive tests, he became convinced that if there were to be the improvements he wanted, he had to design a coupling that flexed without any sliding or rubbing.

Flexible-coupling behavior, however, is not without design problems. Any flexible coupling can be proportioned with strong, thick, stiff members that easily transmit a design torque and provide the stiffness to operate at design speed. However, misalignment requires flexing of these members. The flexing produces alternating stresses that can limit coupling life. The greater the strength and stiffness of a member, the higher the alternating stress from a given misalignment. Therefore, strength and stiffness provisions that transmit torque at speed will be detrimental to misalignment accommodation capability.

The design problem is to proportion the flexible coupling to accomplish torque transmission and overcome misalignment for the lowest system cost. Bossler looked at a drive shaft, a good example of power transmission—and wondered how he could convert it into one with flexibility.

He began to evolve it by following basic principles. How does a drive shaft transmit torque? By tension and compression. He began paring it down to the important struts that could transmit torque and found that they are curved beams. But a curved beam in tension and compression is not as strong as a straight beam. He ended up with the beams straight in a square space-frame with what might be called a *double helix arrangement*. One helix contained elements in compression; the other helix contained elements in tension.

**Flattening the helix.** The total number of plates should be an even number to obtain constant velocity characteristics during misalignment. But even with an odd number, the cyclic speed variations are minute, not nearly the magnitude of those in a Hooke's joint.

Although the analysis and resulting equations developed by Bossler are

based on a square-shaped unit, he concluded that the perfect square is not the ideal for the coupling, because of the position of the mounting holes. The flatter the helix—in other words the smaller the distance  $S$ —the more misalignment the coupling will tolerate.

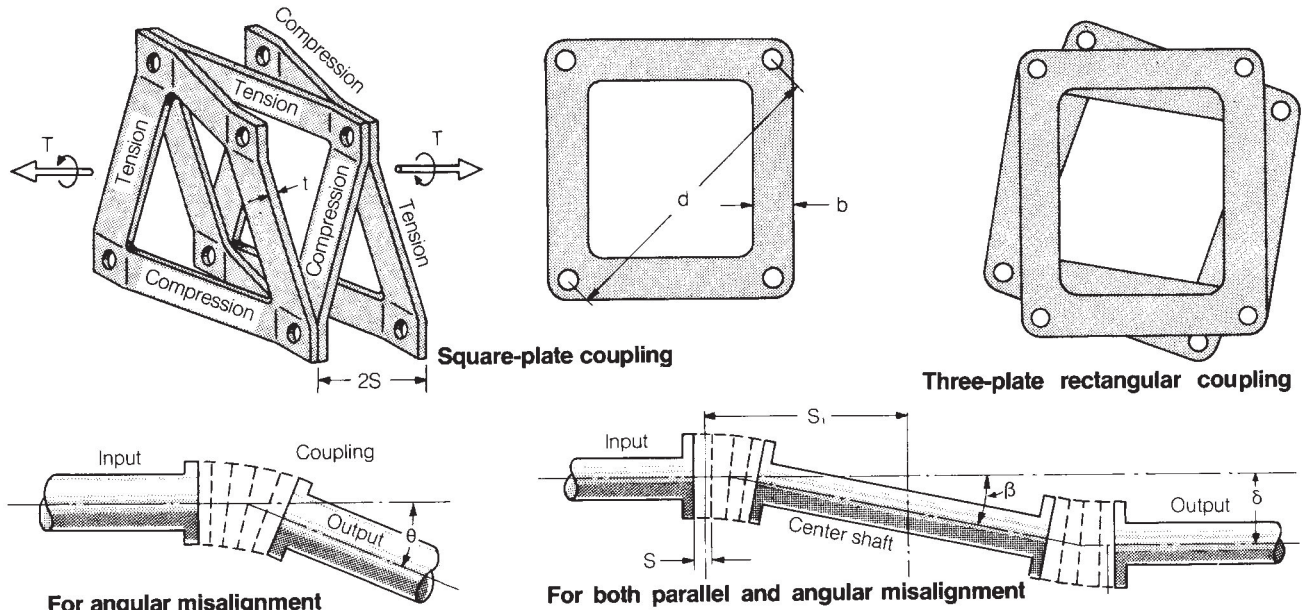
Hence, Bossler began making the space-frames slightly rectangular instead of square. In this design, the bolt-heads that fasten the plates together are offset from adjoining pairs, providing enough clearance for the design of a "flatter" helix. The difference in stresses between a coupling with square-shaped plates and one with slightly rectangular plates is so insignificant that the square-shape equations can be employed with confidence.

**Design equations.** By making a few key assumptions and approximations, Bossler boiled the complex analytical relationships down to a series of straightforward design equations and charts. The derivation of the equations and the resulting verification from tests are given in the NASA report *The Bossler Coupling*, CR-1241.

**Torque capacity.** The ultimate torque capacity of the coupling before buckling that might occur in one of the space-frame struts under compression is given by Eq. 1. The designer usually knows or establishes the maximum continuous torque that the coupling must transmit. Then he must allow for possible shock loads and overloads. Thus, the clutch should be designed to have an ultimate torque capacity that is at least twice as much, and perhaps three times as much, as the expected continuous torque, according to Bossler.

**Induced stress.** At first glance, Eq. 1 seems to allow a lot of leeway in selecting the clutch size. The torque capacity is easily boosted, for example, by picking a smaller bolt-circle diameter,  $d$ , which

### Bent space frames join to form couplings



### Design equations for the Bossler coupling

Ultimate torque capacity

$$(1) \quad T = 11.62 \frac{Ebt^3}{dn^{0.9}}$$

Maximum stress per degree of misalignment.

$$(2) \quad \sigma_{\max} = 0.0276 Et/L$$

Minimum thickness to meet required torque strength

$$(3) \quad t = 0.4415 \left( \frac{dT}{bE} \right)^{1/3} n^{0.3}$$

Weight of coupling with minimum-thickness plates

$$(4) \quad W = 1.249 w \left( \frac{T}{E} \right)^{1/3} d^{4/3} b^{2/3} n^{1.3}$$

Maximum permissible misalignment

$$(5) \quad \theta_{\max} = 54.7 \left[ \frac{bd^2}{TE^2} \right]^{1/3} \sigma_c n^{0.7}$$

Maximum permissible misalignment (simplified)

$$(6) \quad \theta/d = 10.9 \frac{n^{0.7}}{T^{1/3}}$$

Maximum permissible offset-angle

$$(7) \quad \beta = 54.7 \left[ \frac{bd^2}{TE^2} \right]^{1/3} \frac{\sigma_c C}{n^{0.3}}$$

$$\text{where: } \sum_{x=1}^{x=n} \left[ 1 - (x-1) \frac{S}{S_1} \right]^2$$

Maximum permissible offset-angle (simplified)

$$(8) \quad \beta/d = \frac{10.9C}{T^{1/3} n^{0.3}}$$

Critical speed frequency

$$(9) \quad f = \frac{60}{2\pi} \left( \frac{k}{M} \right)^{1/2}$$

$$\text{where: } k = \frac{24(EI)_e}{(nS)^3} \text{ and } (EI)_e = 0.886Ebt^3S/L$$

### List of symbols

- b = Width of an element
- d = Diameter at the bolt circle
- E = Modulus of elasticity
- f = First critical speed, rpm
- I = Flatwise moment of inertia of an element =  $bt^3/12$
- k = Spring constant for single degree of freedom
- L = Effective length of an element. This concept is required because joint details tend to stiffen the ends of the elements.  $L = 0.667 d$  is recommended
- M = Mass of center shaft plus mass of one coupling with fasteners
- n = Number of plates in each coupling
- S = Offset distance by which a plate out of plane
- t = Thickness of an element
- T = Torque applied to coupling, useful ultimate, usually taken as lowest critical buckling torque
- w = Weight per unit volume
- W = Total weight of plates in a coupling
- $(EI)_e$  = Flexural stiffness, the moment that causes one radian of flexural angle change per unit length of coupling
- $\beta$  = Equivalent angle change at each coupling during parallel offset misalignment, deg
- $\nu$  = Total angular misalignment, deg
- $\sigma_c$  = Characteristic that limits stress for the material: yield stress for static performance, endurance limit stress for fatigue performance

makes the clutch smaller, or by making the plates thicker. But either solution would also make the clutch stiffer, hence would restrict the misalignment permitted before the clutch becomes overstressed. The stress-misalignment relationship is given in Eq. 2, which shows the maximum flat-wise bending stress produced when a plate is misaligned  $1^\circ$  and is then rotated to transmit torque.

**Plate thickness.** For optimum misalignment capability, the plates should be selected with the least thickness that will provide the required torque strength. To determine the minimum thickness, Bossler found it expedient to rearrange Eq. 1 into the form shown in Eq. 3. The weight of any coupling designed in accordance to the minimum-thickness equation can be determined from Eq. 4.

**Maximum misalignment.** Angular misalignment occurs when the centerlines of the input and output shafts intersect at some angle—the angle of misalignment. When the characteristic limiting

stress is known for the material selected—and for the coupling's dimensions—the maximum allowable angle of misalignment can be computed from Eq. 5.

If this allowance is not satisfactory, the designer might have to juggle the size factors by, say, adding more plates to the unit. To simplify eq. 5, Bossler made some assumptions in the ratio of endurance limit to modulus and in the ratio of  $dsb$  to obtain Eq. 6.

**Parallel offset.** This condition exists when the input and output shafts remain parallel but are displaced laterally. As with Eq. 6, Eq. 7 is a performance equation and can be reduced to design curves. Bossler obtained Eq. 8 by making the same assumptions as in the previous case.

**Critical speed.** Because of the noncircular configurations of the coupling, it is important that the operating speed of the unit be higher than its critical speed. It should not only be higher but also should avoid an integer relationship.

Bossler worked out a handy relationship for critical speed (Eq. 9) that employs a somewhat idealized value for the spring constant  $k$ .

Bossler also made other recommendations where weight reduction is vital:

- **Size of plates.** Use the largest  $d$  consistent with envelope and centrifugal force loading. Usually, centrifugal force loading will not be a problem below 300 ft/s tip speed.
- **Number of plates.** Pick the least  $n$  consistent with the required performance.
- **Thickness of plates.** Select the smallest  $t$  consistent with the required ultimate torque.
- **Joint details.** Be conservative; use high-strength tension fasteners with high preload. Provide fretting protection. Make element centerlines and bolt centerlines intersect at a point.
- **Offset distance.** Use the smallest  $S$  consistent with clearance.

## COUPLING WITH OFF-CENTER PINS CONNECTS MISALIGNED SHAFTS

Two Hungarian engineers developed an all-metal coupling (see drawing) for connecting shafts where alignment is not exact—that is, where the degree of misalignment does not exceed the magnitude of the shaft radius.

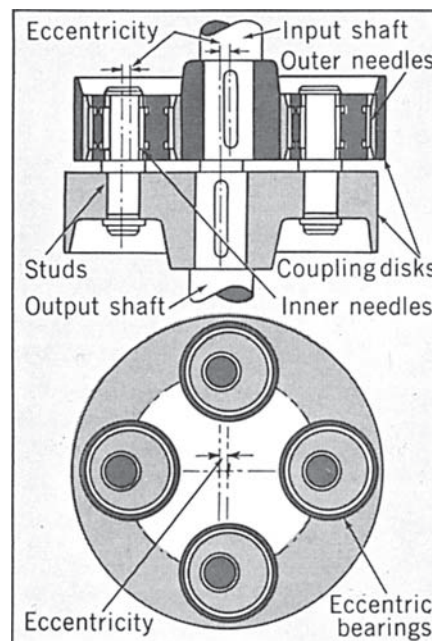
The coupling is applied to shafts that are being connected for either high-torque or high-speed operation and that must operate at maximum efficiency. Knuckle joints are too expensive, and they have too much play; elastic joints are too vulnerable to the influences of high loads and vibrations.

**How it's made.** In essence, the coupling consists of two disks, each keyed

to a splined shaft. One disk bears four fixed-mounted steel studs at equal spacing; the other disk has large-diameter holes drilled at points facing the studs.

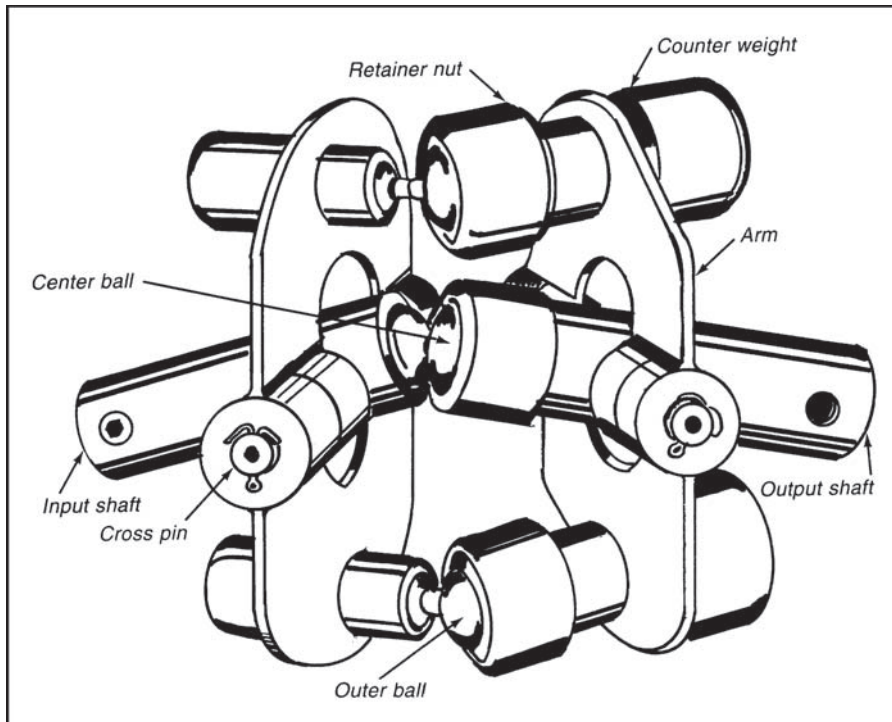
Each large hole is fitted with a bearing that rotates freely inside it on rollers or needles. The bore of the bearings, however, is off-center. The amount of eccentricity of the bearing bore is identical to the deviation of the two shaft centerlines.

In operation, input and output shafts can be misaligned, yet they still rotate with the same angular relationship they would have if perfectly aligned.



**Eccentrically bored bearings** rotate to make up for misalignment between shafts.

# UNIVERSAL JOINT TRANSMITS TORQUE 45° AT CONSTANT SPEED



A novel arrangement of pivots and ball-socket joints transmits uniform motion.

A universal joint that transmits power at constant speeds through angles up to 45° was designed by Malton Miller of Minnesota.

Models of the true-speed drive that can transmit up to 20 hp have been developed.

It had not been possible to transmit power at constant speeds with only one universal joint. Engineers had to specify an intermediate shaft between two Hooke's joints or use a Rzappa-type joint to get the desired effect.

**Ball-and-socket.** Basically, the True-Speed joint is a system of ball-and-socket connections with large contact areas (low unit pressure) to transmit torsional forces across the joint. This arrangement minimizes problems when high bearing pressures build up against running surfaces. The low-friction bearings also increase efficiency. The joint is balanced to keep vibration at high speeds to a minimum.

The joint consists of driving and driven halves. Each half has a coupling sleeve at its end of the driveshaft, a pair of driving

arms opposite each other and pivoted on a cross pin that extends through the coupling sleeve, and a ball-and-socket coupling at the end of each driving arm.

As the joint rotates, angular flexure in one plane of the joint is accommodated by the swiveling of the ball-and-socket couplings and, in the 90° plane, by the oscillation of the driving arms about the transverse pin. As rotation occurs, torsion is transmitted from one half of the joint to the other half through the swiveling ball-and-socket couplings and the oscillating driving arms.

**Balancing.** Each half of the joint, in effect, rotates about its own center shaft, so each half is considered separate for balancing. The center ball-and-socket coupling serves only to align and secure the intersection point of the two shafts. It does not transmit any forces to the entire drive unit.

Balancing for rotation is achieved by equalizing the weight of the two driving arms of each half of the joint. Balancing the acceleration forces due

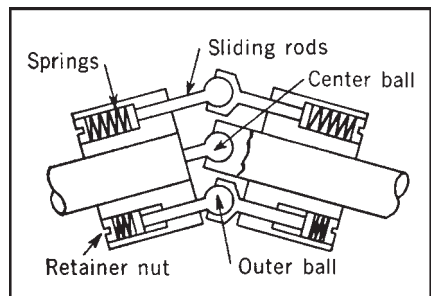
to the oscillation of the ball-and-socket couplings, which are offset from their swiveling axes, is achieved by the use of counterweights extending from the opposite side of each driving arm.

The outer ball-and-socket couplings work in two planes of motion, swiveling widely in the plane perpendicular to the main shaft and swiveling slightly about the transverse pin in the plane parallel to the main shaft. In this coupling configuration, the angular displacement of the driving shaft is exactly duplicated in the driven shaft, providing constant rotational velocity and constant torque at all shaft intersection angles.

**Bearings.** The only bearing parts are the ball-and-socket couplings and the driving arms on the transverse pins. Needle bearings support the driving arms on the transverse pin, which is hardened and ground. A high-pressure grease lubricant coats the bearing surfaces of the ball-and-socket couplings. Under maximum rated loadings of 600 psi on the ball-and-socket surfaces, there is no appreciable heating or power loss due to friction.

**Capabilities.** Units have been laboratory-tested at all rated angles of drive under dynamometer loadings. Although the first available units were for smaller capacities, a unit designed for 20 hp at 550 rpm, suitable for tractor power take-off drive, has been tested.

Similar couplings have been designed as pump couplings. But the True-Speed drive differs in that the speed and transfer elements are positive. With the pump coupling, on the other hand, the speed might fluctuate because of spring bounce.



An earlier version for angled shafts required spring-loaded sliding rods.

# TEN UNIVERSAL SHAFT COUPLINGS

## Hooke's Joints

The commonest form of a universal coupling is a *Hooke's joint*. It can transmit torque efficiently up to a maximum shaft alignment angle of about  $36^\circ$ . At slow speeds, on hand-operated mechanisms, the permissible angle can reach  $45^\circ$ . The simplest arrangement for a Hooke's joint is two forked shaft-ends coupled by a cross-shaped piece. There are many variations and a few of them are included here.

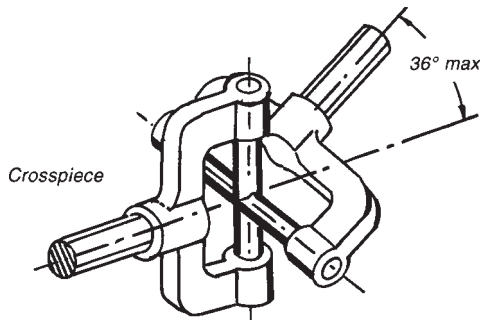


**Fig. 1** The Hooke's joint can transmit heavy loads. Anti-friction bearings are a refinement often used.

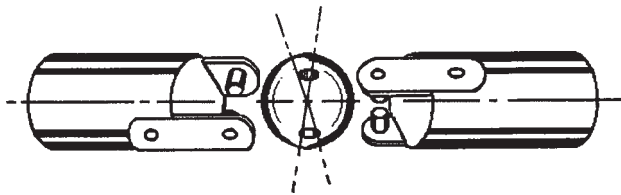


**Fig. 2** A pinned sphere shaft coupling replaces a cross-piece. The result is a more compact joint.

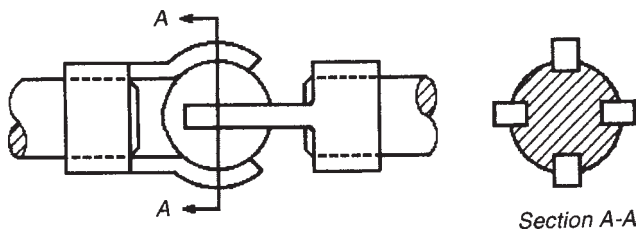


**Fig. 3** A grooved-sphere joint is a modification of a pinned sphere. Torques on fastening sleeves are bent over the sphere on the assembly. Greater sliding contact of the torques in grooves makes simple lubrication essential at high torques and alignment angles.

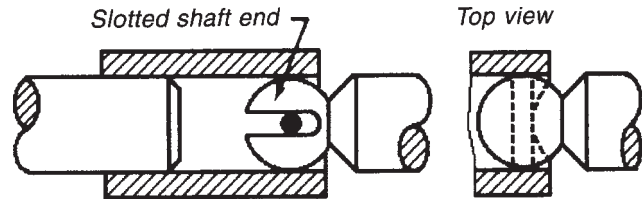


**Fig. 4** A pinned-sleeve shaft-coupling is fastened to one shaft that engages the forked, spherical end on the other shaft to provide a joint which also allows for axial shaft movement. In this example, however, the angle between shafts must be small. Also, the joint is only suitable for low torques.

## Constant-Velocity Couplings

The disadvantages of a single Hooke's joint is that the velocity of the driven shaft varies. Its maximum velocity can be found by multiplying driving-shaft speed by the secant of the shaft angle; for minimum speed, multiply by the cosine. An example of speed variation: a driving shaft rotates at 100 rpm; the angle between the shafts is  $20^\circ$ . The minimum output is  $100 \times 0.9397$ , which equals 93.9 rpm; the maximum output is  $1.0642 \times 100$ , or 106.4 rpm. Thus, the difference is 12.43 rpm. When output speed is high, output torque is low, and vice versa. This is an objectionable feature in some mechanisms. However, two universal joints connected by an intermediate shaft solve this speed-torque objection.

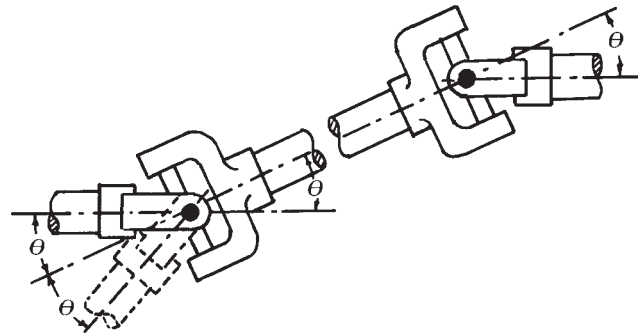


**Fig. 5** A constant-velocity joint is made by coupling two Hooke's joints. They must have equal input and output angles to work correctly. Also, the forks must be assembled so that they will always be in the same plane. The shaft-alignment angle can be double that for a single joint.

**This single constant-velocity coupling** is based on the principle (Fig. 6) that the contact point of the two members must always lie on the homokinetic plane. Their rotation speed will then always be equal because the radius to the contact point of each member will always be equal. Such simple couplings are ideal for toys, instruments, and other light-duty mechanisms. For heavy duty, such as the front-wheel drives of military vehicles, a more complex coupling is shown

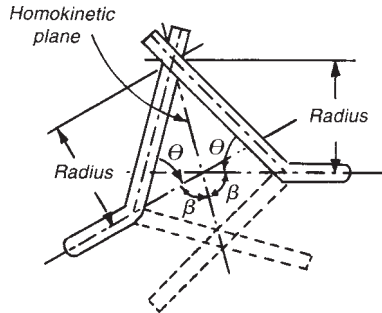


Fig. 6



**Fig. 8** This flexible shaft permits any shaft angle. These shafts, if long, should be supported to prevent backlash and coiling.

diagrammatically in Fig. 7A. It has two joints close-coupled with a sliding member between them. The exploded view (Fig. 7B) shows these members. There are other designs for heavy-duty universal couplings; one, known as the *Rzeppa*, consists of a cage that keeps six balls in the homokinetic plane at all times. Another constant-velocity joint, the *Bendix-Weiss*, also incorporates balls.

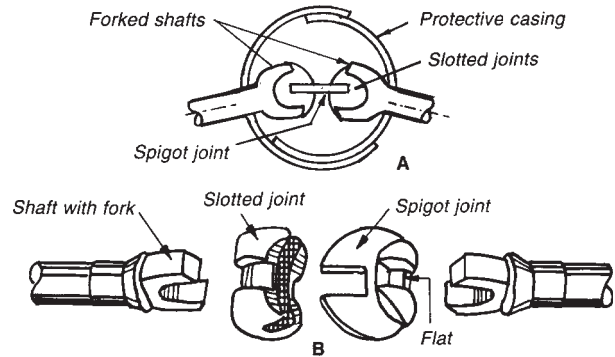


Fig. 7

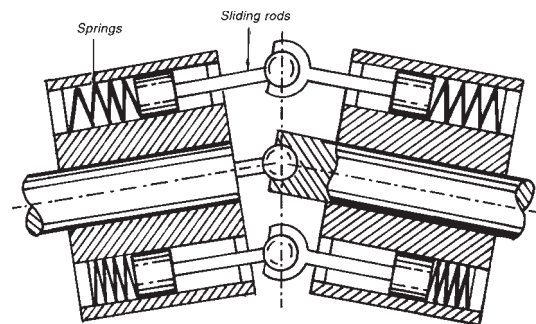


**Fig. 9** This pump-type coupling has the reciprocating action of sliding rods that can drive pistons in cylinders.

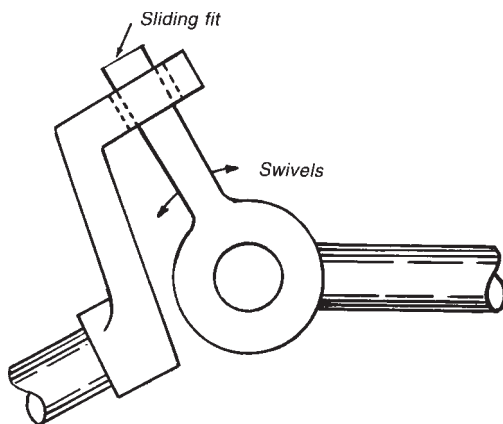


**Fig. 10** This light-duty coupling is ideal for many simple, low-cost mechanisms. The sliding swivel-rod must be kept well lubricated at all times.

# NINETEEN METHODS FOR COUPLING ROTATING SHAFTS

Methods for coupling rotating shafts vary from simple bolted flange assemblies to complex spring and synthetic rubber assemblies. Those including chain belts, splines, bands, and rollers are shown here.

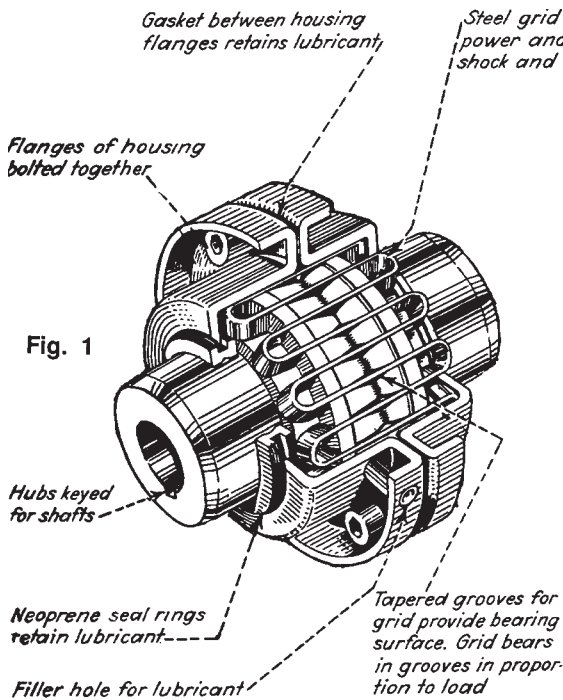


Fig. 1

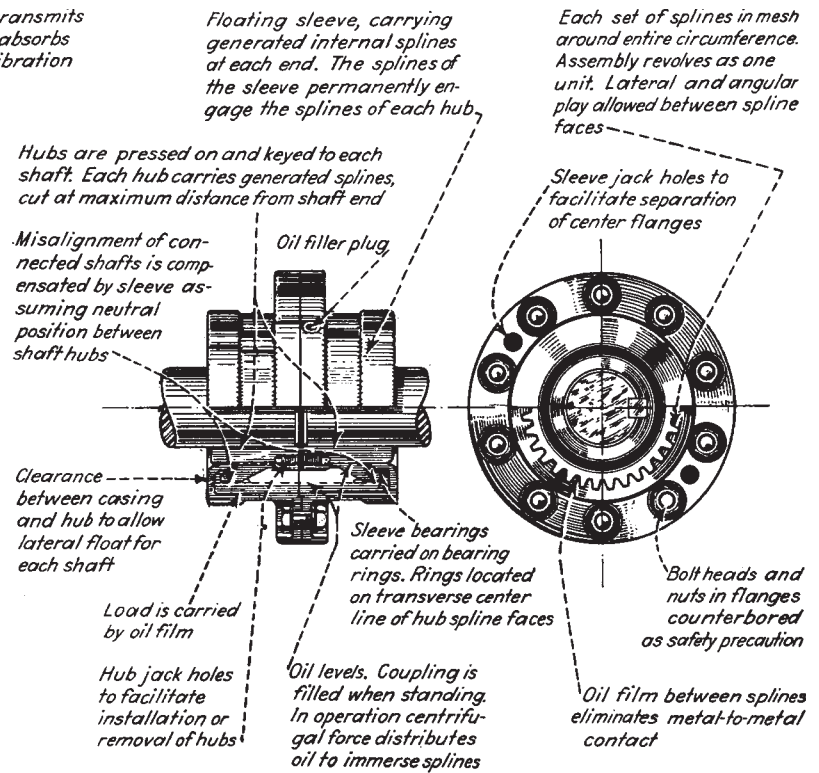


Fig. 2

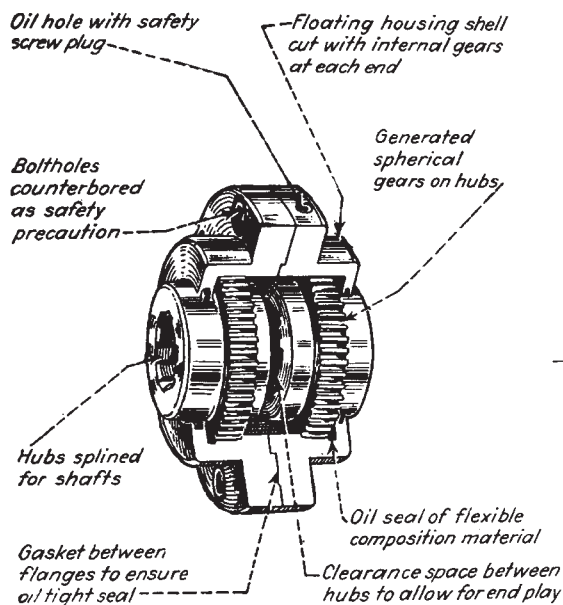


Fig. 3

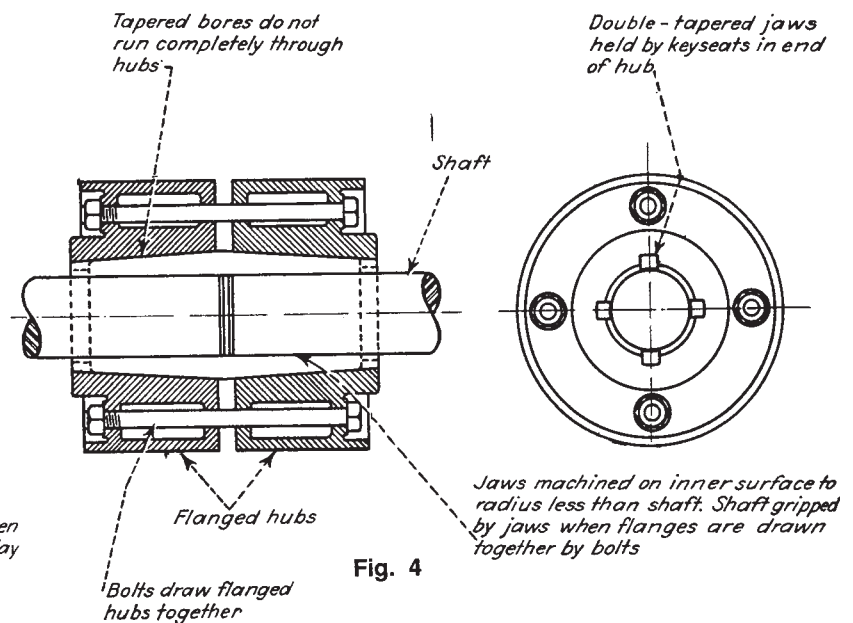
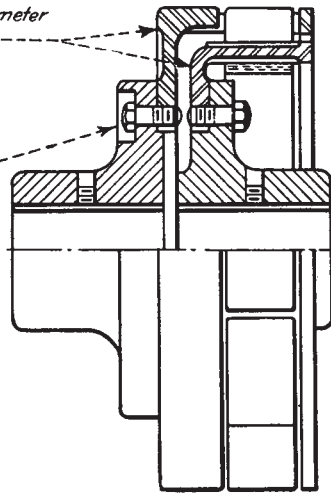


Fig. 4

Steel rims, one of which is of smaller diameter than the other

Rims mounted on flanges



Rims

Endless leather belt laced through alternate rim slots

Axial slots on each rim

Flanges keyed to shaft

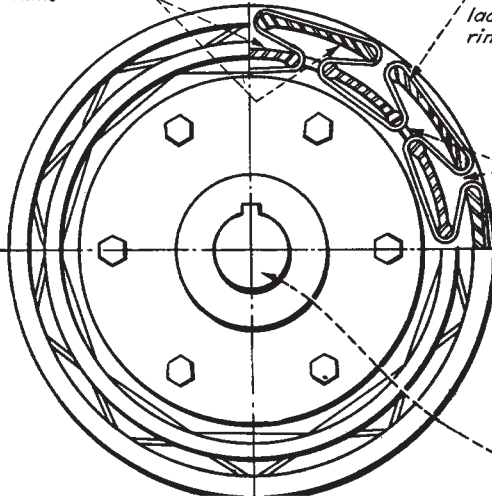


Fig. 5

Casing provided with lugs that fit loosely in the twin-arms of the spider and bear against spring plugs

Removable access plate to springs

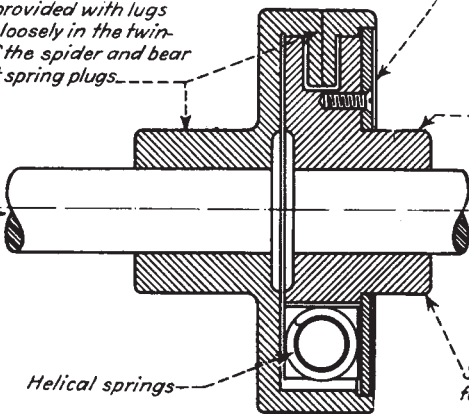
Spider

Shaft

Helical springs

Fig. 6

Smooth exterior for safety

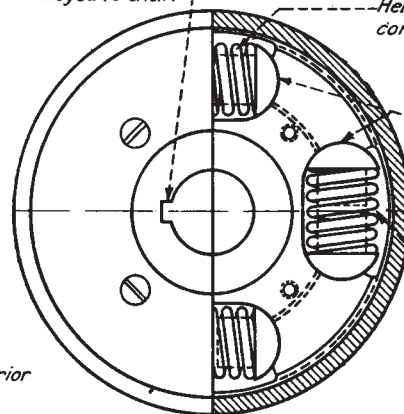


Casing and spider keyed to shaft

Helical springs in constant compression

Case-hardened plugs fitted into pockets between twin-arms of spider

Clearance between ends of spring plugs less than maximum deflection of spring. Sudden overload can not break springs



Cylindrical sleeve with eccentric chambers on inside

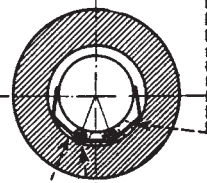
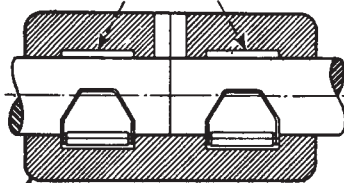
Turning coupling forces rollers up inclined sides of eccentric chamber to lock coupling to shaft

Smooth exterior for safety

Fig. 7

With rollers located in largest part of eccentric chamber, coupling can be slipped over end of shaft

Two steel rollers held parallel by light wire frame



Side clearance provided between chain and teeth for accommodation of angular displacement between shafts

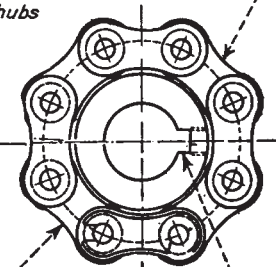
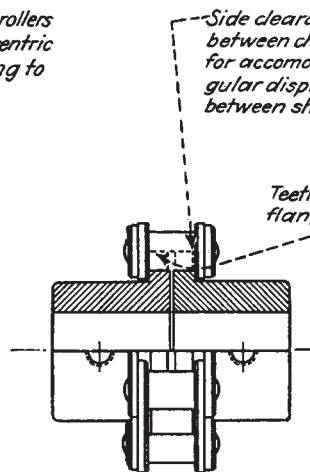
Teeth cut on flanges of hubs

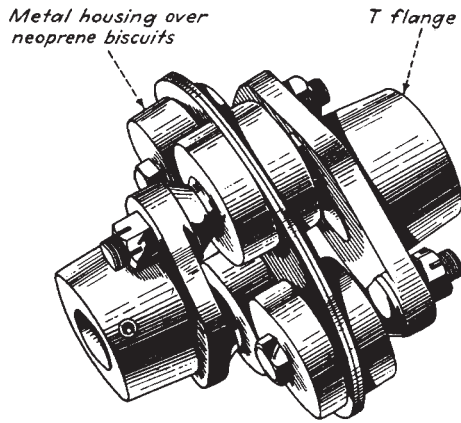
Roller chain over teeth on hub flanges. All rollers in contact with teeth for equal distribution of transmitted load

Fig. 8

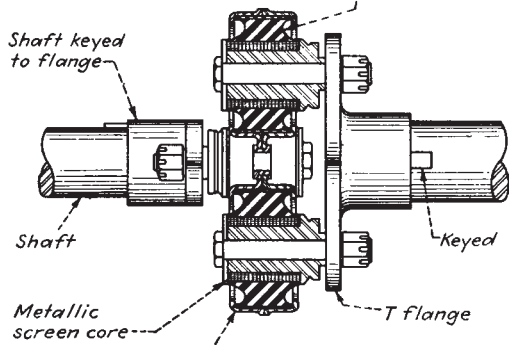
Chain provided with master link for removal

Hubs keyed to shafts





Neoprene center designed for uniform stress, linear deflection and absorption of vibration



Outer fabric ring impregnated with neoprene, provides support for center section

Fig. 9

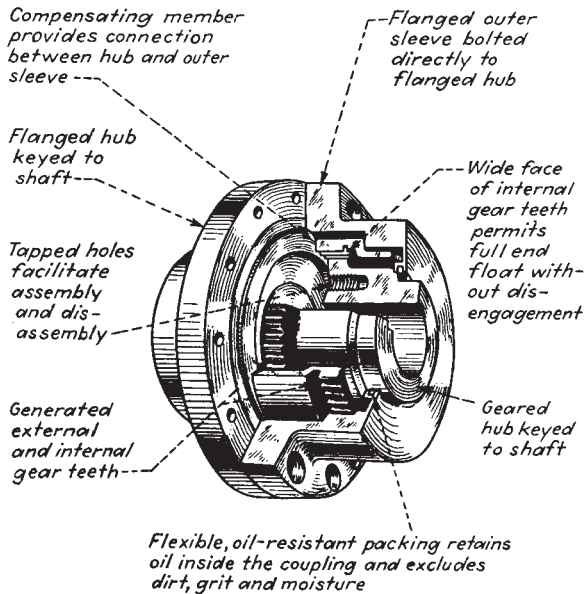


Fig. 12

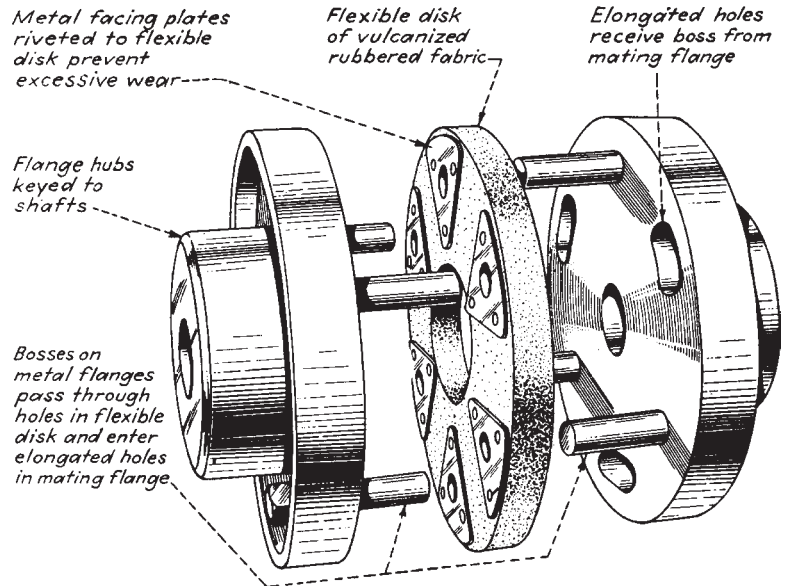


Fig. 10

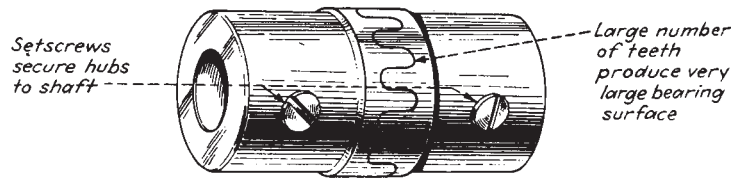


Fig. 11

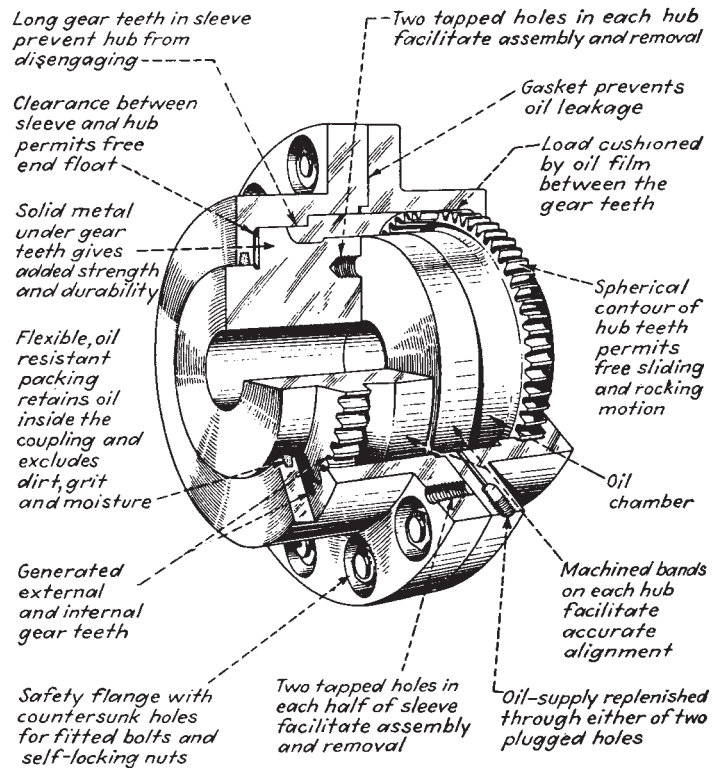


Fig. 13

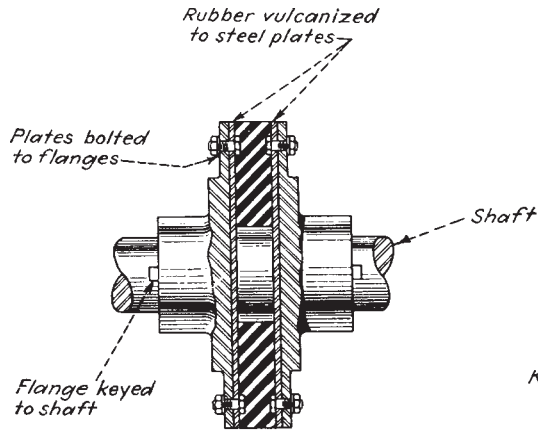


Fig. 14

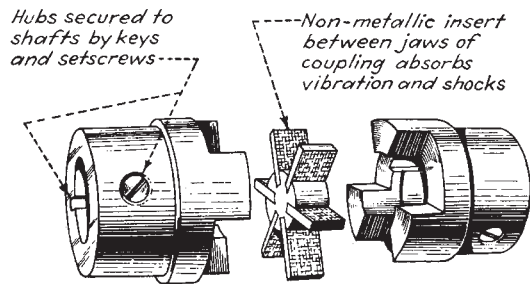


Fig. 16

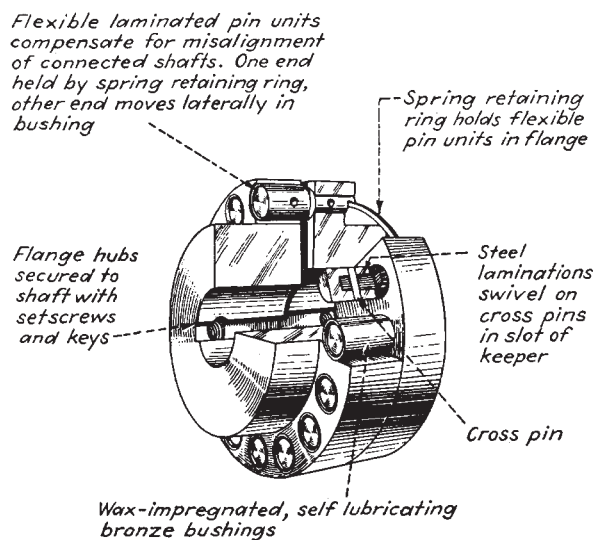


Fig. 18

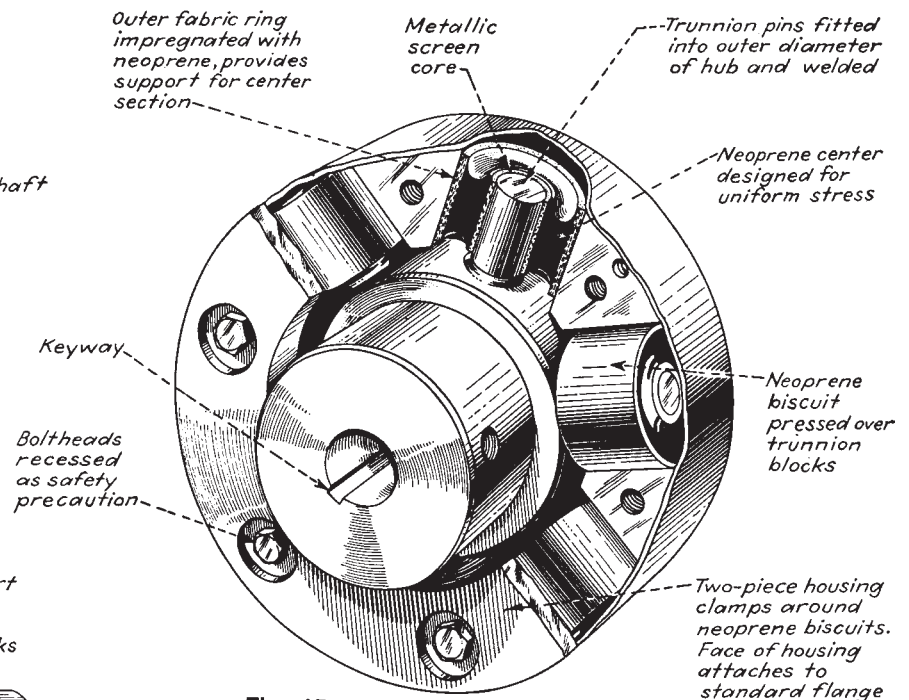


Fig. 15

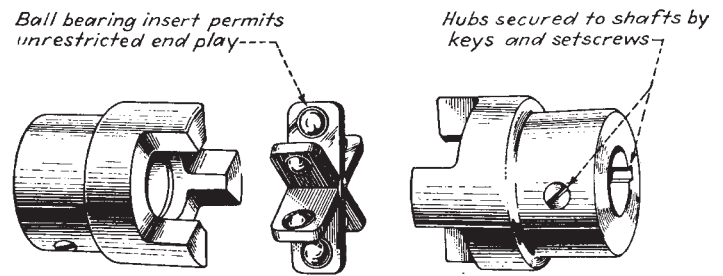


Fig. 17

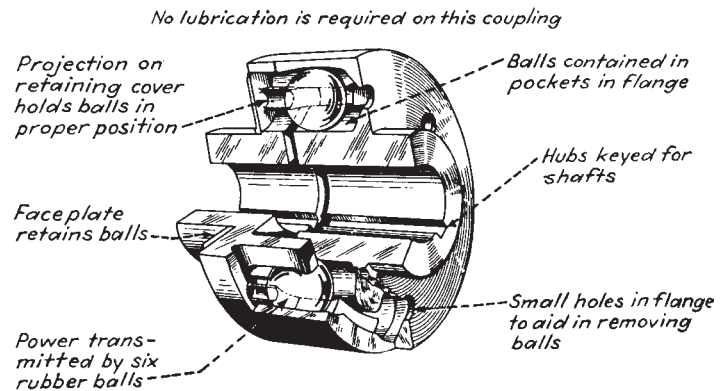


Fig. 19

# FIVE DIFFERENT PIN-AND-LINK COUPLINGS

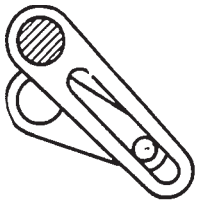


Fig. 1

**Fig. 1** If constant velocity is not required, a pin and slot coupling can be used. Velocity transmission is irregular because the effective radius of operation is continually changing. The shafts must remain parallel unless a ball joint is placed between the slot and pin. Axial freedom is possible, but any change in the shaft offset will further affect the fluctuation of velocity transmission.

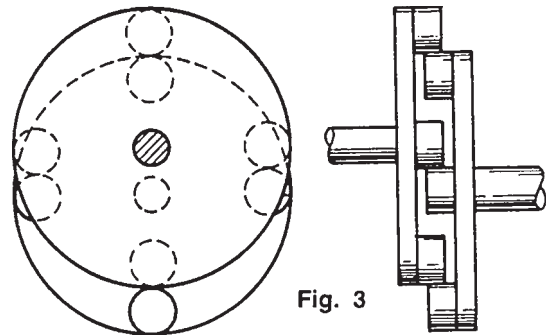


Fig. 3

**Fig. 3** This coupling is kinematically equivalent to Fig. 2. It can be made by substituting two circular and contacting pins for each link. Each shaft has a disk carrying three or more projecting pins. The sum of the radii of the pins is equal to the eccentricity of offset of the shafts. The center lines between each pair of pins remain parallel as the coupling rotates. The pins need not have equal diameters. Transmission is at a constant velocity, and axial freedom is possible.

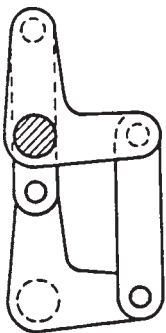


Fig. 2

**Fig. 2** This parallel-crank coupling drives the overhead camshaft on engines. Each shaft has at least two cranks connected by links. Each must have full symmetry for constant velocity action and to avoid dead points. By attaching ball joints at the ends of the links, displacement between the crank assemblies is possible.

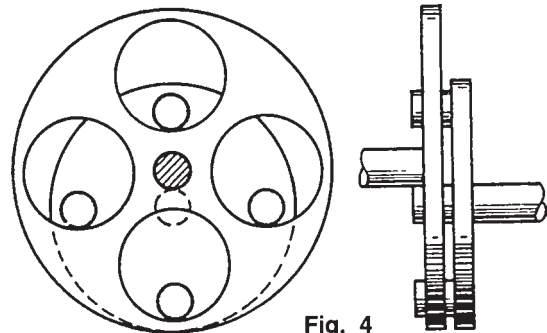


Fig. 4

**Fig. 4** This coupling is similar to the coupling shown in Fig. 3. However, holes replace one set of pins. The difference in radii is equal to the eccentricity or offset. Velocity transmission is constant; axial freedom is possible, but as in Fig. 3, the shaft axes must remain fixed. This type of coupling can be installed in epicyclic reduction gear boxes.

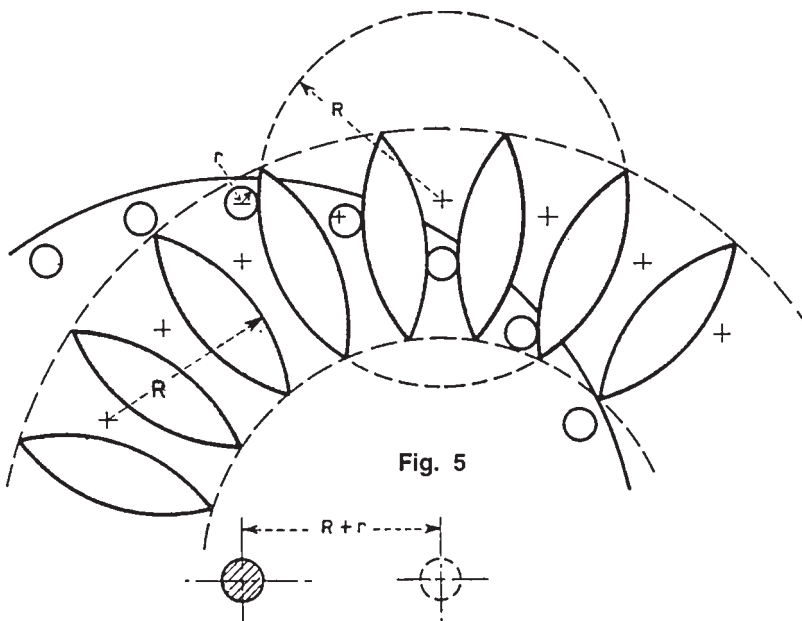
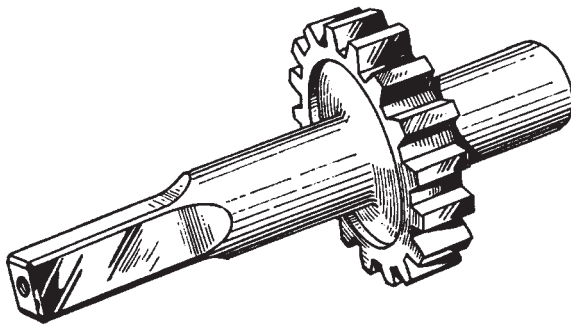


Fig. 5

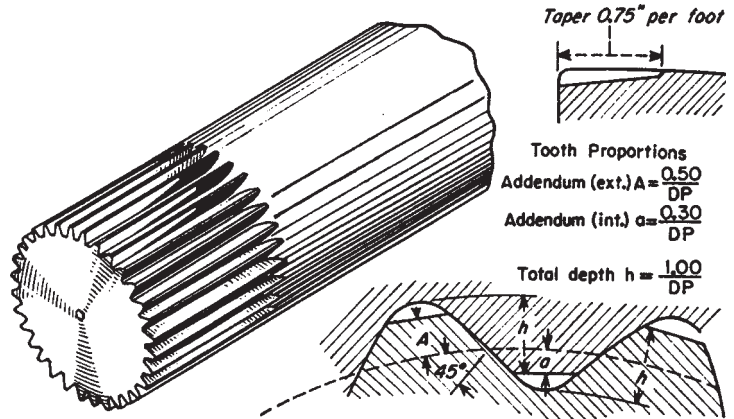
**Fig. 5** An unusual development in pin coupling is shown. A large number of pins engages the lenticular or shield-shaped sections formed from segments of theoretical large pins. The axes forming the lenticular sections are struck from the pith points of the coupling, and the distance  $R + r$  is equal to the eccentricity between the shaft centers. Velocity transmission is constant; axial freedom is possible, but the shafts must remain parallel.

# TEN DIFFERENT SPLINED CONNECTIONS

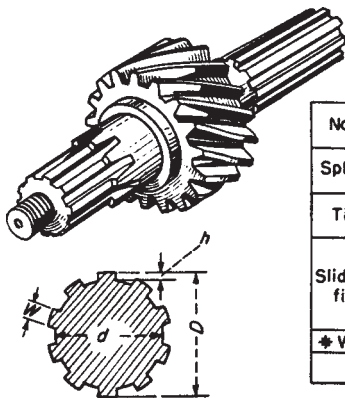
## CYLINDRICAL SPLINES



1. **SQUARE SPLINES** make simple connections. They are used mainly for transmitting light loads, where accurate positioning is not critical. This spline is commonly used on machine tools; a cap screw is required to hold the enveloping member.



2. **SERRATIONS** of small size are used mostly for transmitting light loads. This shaft forced into a hole of softer material makes an inexpensive connection. Originally straight-sided and limited to small pitches, 45° serrations have been standardized (SAE) with large pitches up to 10 in. dia. For tight fits, the serrations are tapered.

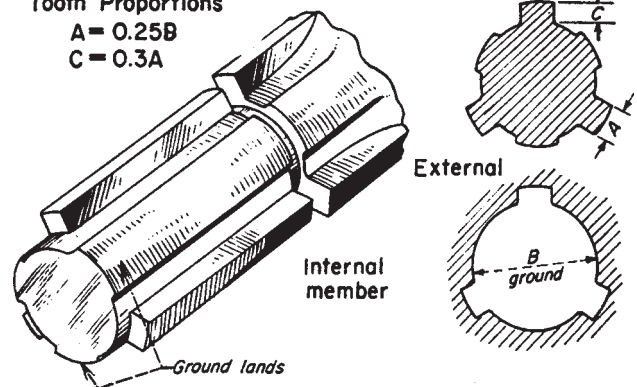


SAE STANDARD  
SPLINE PROPORTIONS

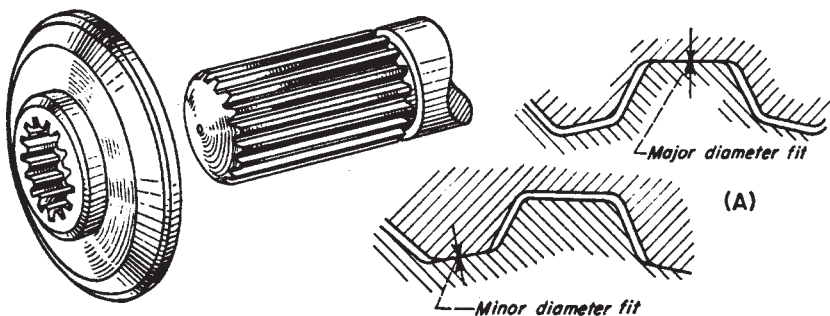
| No. of splines                       | 4                     | 6      | 10 or 16 |
|--------------------------------------|-----------------------|--------|----------|
| Spline width                         | $W = 0.241D$          | 0.250D | 0.156D   |
| Tight fit                            | $h = 0.075D$          | 0.050D | 0.045D   |
| Sliding fit                          | Unloaded $h = 0.125D$ | 0.075D | 0.070D   |
|                                      | loaded $h = 0.100D$   | 0.095D |          |
| * Width of 16-tooth spline is 0.098D |                       |        |          |
| Root dia, $d = D - 2h$               |                       |        |          |

3. **STRAIGHT-SIDED** splines have been widely used in the automotive field. Such splines are often used for sliding members. The sharp corner at the root limits the torque capacity to pressures of approximately 1,000 psi on the spline projected area. For different applications, tooth height is altered, as shown in the table above.

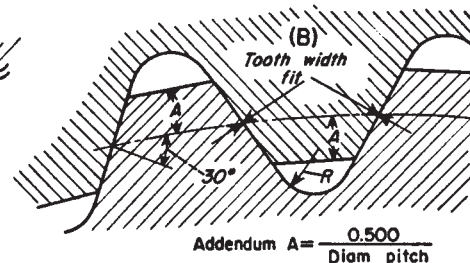
Tooth Proportions  
 $A = 0.25B$   
 $C = 0.3A$



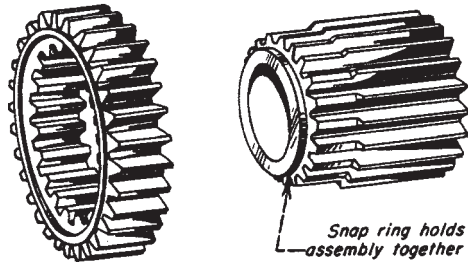
4. **MACHINE-TOOL** splines have wide gaps between splines to permit accurate cylindrical grinding of the lands—for precise positioning. Internal parts can be ground readily so that they will fit closely with the lands of the external member.



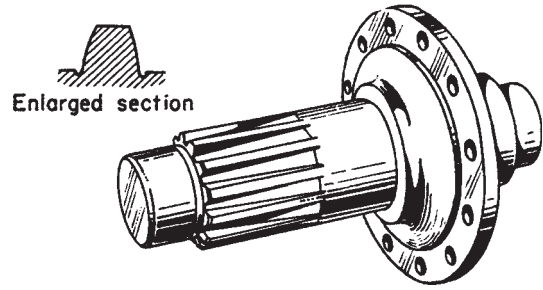
5. **INVOLUTE-FORM** splines are used where high loads are to be transmitted. Tooth proportions are based on a 30° stub tooth form. (A) Splined members can be positioned either by close fitting major or minor diameters. (B) Use of the tooth width or side



positioning has the advantage of a full fillet radius at the roots. Splines can be parallel or helical. Contact stresses of 4,000 psi are used for accurate, hardened splines. The diametral pitch shown is the ratio of teeth to the pitch diameter.

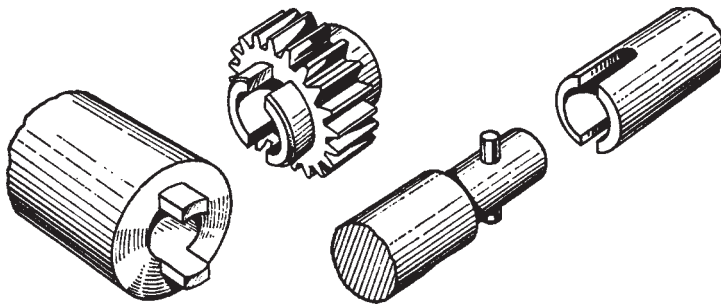


6. SPECIAL INVOLUTE splines are made by using gear tooth proportions. With full depth teeth, greater contact area is possible. A compound pinion is shown made by cropping the smaller pinion teeth and internally splining the larger pinion.

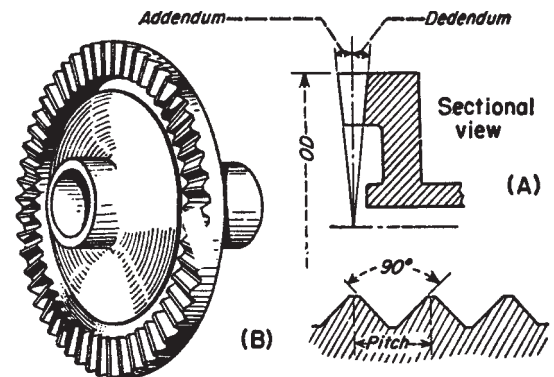


7. TAPER-ROOT splines are for drivers that require positive positioning. This method holds mating parts securely. With a  $30^\circ$  involute stub tooth, this type is stronger than parallel root splines and can be hobbled with a range of tapers.

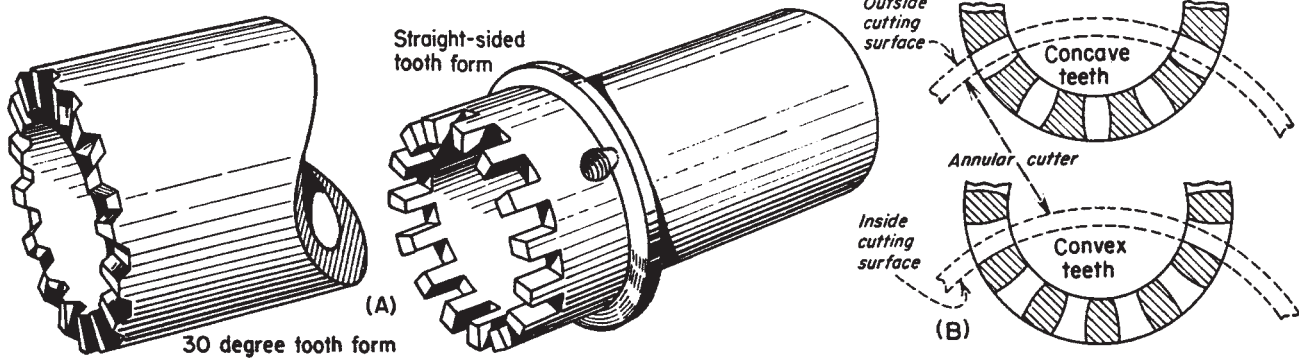
## FACE SPLINES



8. MILLED SLOTS in hubs or shafts make inexpensive connections. This spline is limited to moderate loads and requires a locking device to maintain positive engagement. A pin and sleeve method is used for light torques and where accurate positioning is not required.



9. RADIAL SERRATIONS made by milling or shaping the teeth form simple connections. (A) Tooth proportions decrease radially. (B) Teeth can be straight-sided (castellated) or inclined; a  $90^\circ$  angle is common.



10. CURVIC COUPLING teeth are machined by a face-mill cutter. When hardened parts are used that require accurate positioning, the teeth can be ground. (A) This process produces teeth with uniform depth. They can be cut at any pressure angle,

although  $30^\circ$  is most common. (B) Due to the cutting action, the shape of the teeth will be concave (hour-glass) on one member and convex on the other—the member with which it will be assembled.

# FOURTEEN WAYS TO FASTEN HUBS TO SHAFTS

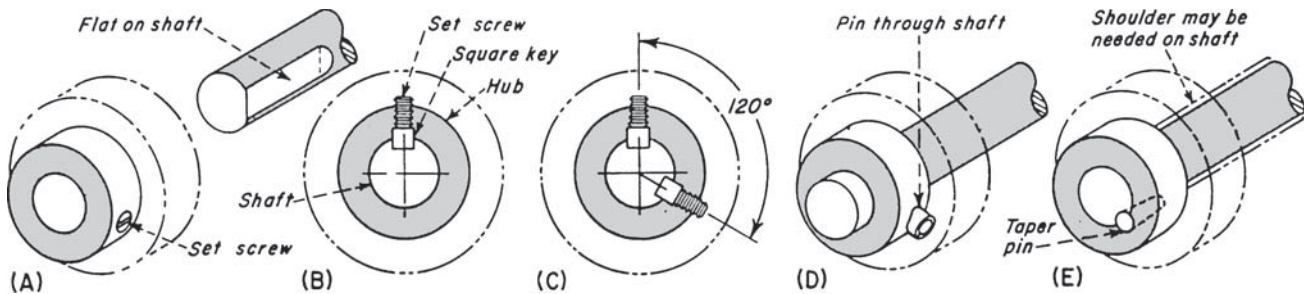


**Fig. 1** A cup-point setscrew in hub (A) bears against a flat on a shaft. This fastening is suitable for fractional horsepower drives with low shock loads but is unsuitable when frequent removal and assembly are necessary. The key with setscrew (B) prevents shaft marring from frequent removal and assembly.

It can withstand high shock loads. Two keys 120° apart (C) transmit extra heavy loads. Straight or tapered pin (D) prevents end play. For experimental setups an expanding pin is suitable yet easy to remove. Taper pin (E) parallel to shaft might require a shoulder on the shaft. It can be used when a gear or pulley has no hub.

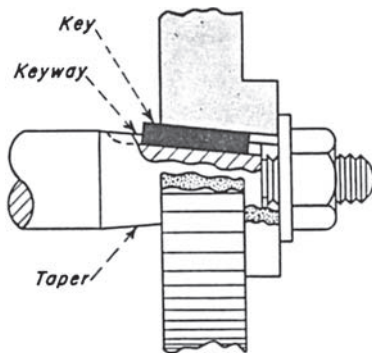


**Fig. 2** A tapered shaft with a key and threaded end is a rigid concentric assembly. It is suitable for heavy-duty applications, yet it can be easily disassembled.

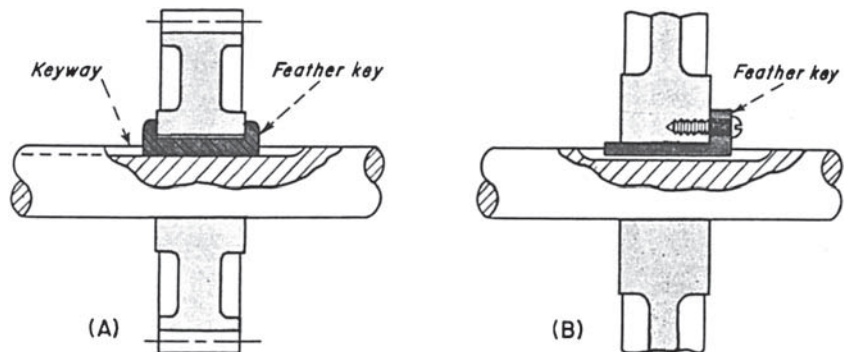


**Fig. 3** A feather key (A) allows axial gear movement. A keyway must be milled to the end of the shaft. For a blind keyway (B) the hub and key must be drilled and tapped, but the design allows the gear to be mounted anywhere on the shaft with only a short keyway.

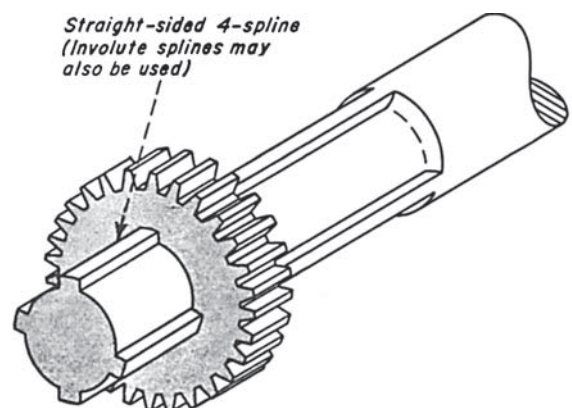


**Fig. 4** Splined shafts are frequently used when a gear must slide. Square splines can be ground to close minor diameter gaps, but involute splines are self-centering and stronger. Non-sliding gears can be pinned to the shaft if it is provided with a hub.

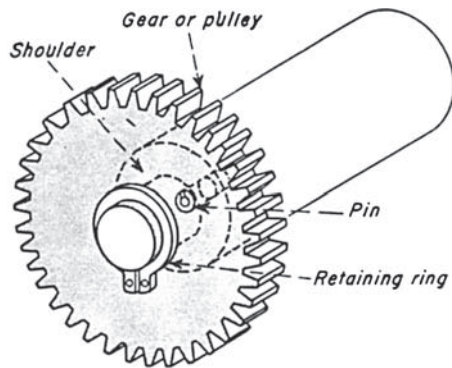


**Fig. 5** A retaining ring allows quick gear removal in light-load applications. A shoulder on the shaft is necessary. A shear pin can secure the gear to the shaft if protection against an excessive load is required.

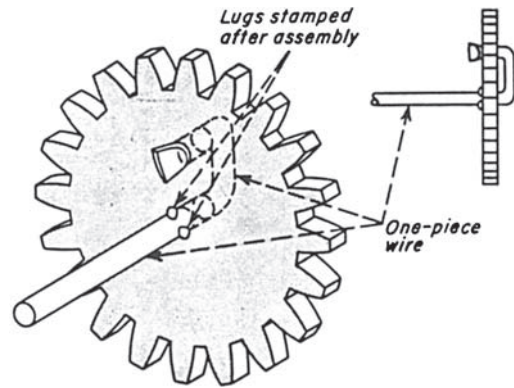


**Fig. 6** A stamped gear and formed wire shaft can be used in light-duty application. Lugs stamped on both legs of the wire prevent disassembly. The bend radii of the shaft should be small enough to allow the gear to seat.

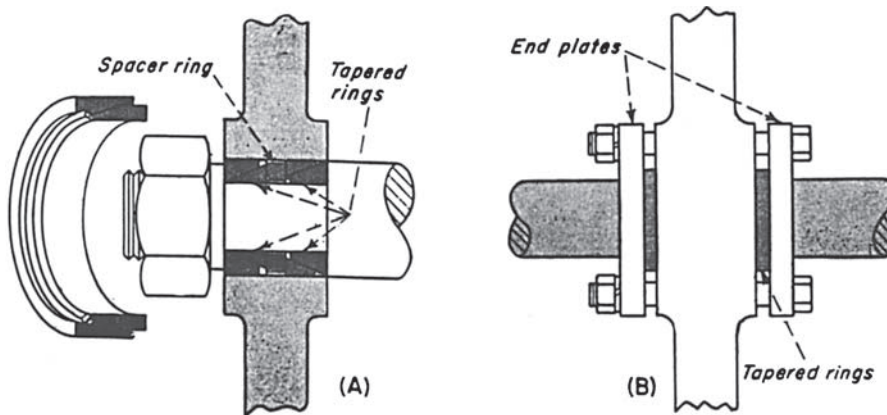


**Fig. 7** Interlocking tapered rings hold the hub tightly to the shaft when the nut is tightened. Coarse machining of the hub and shaft does not affect concentricity as in pinned and keyed assemblies. A shoulder is required (A) for end-of-shaft mounting. End plates and four bolts (B) allow the hub to be mounted anywhere on the shaft.

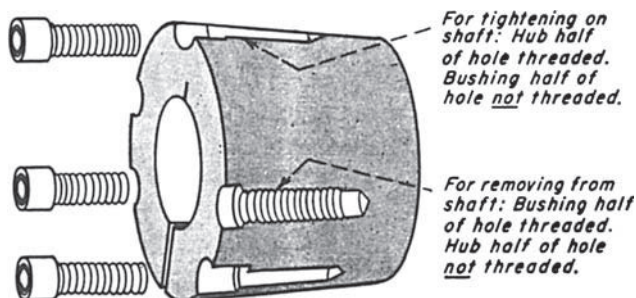


**Fig. 8** This split bushing has a tapered outer diameter. Split holes in the bushing align with split holes in the hub. For tightening, the hub half of the hole is tapped, and the bushing half is un-tapped. A screw pulls the bushing into the hub as it is tightened, and it is removed by reversing the procedure.

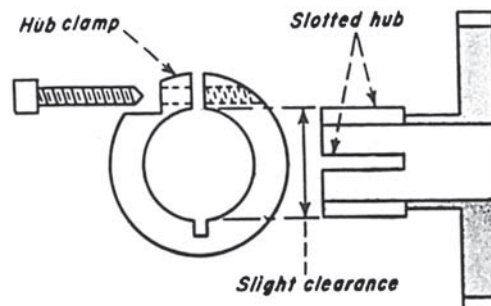


**Fig. 9** The split hub of a stock precision gear is clamped onto a shaft with a separate hub clamp. Manufacturers list correctly dimensioned hubs and clamps so that they can be efficiently fastened to a precision-ground shaft.