GEAR
HANDBOOK

DUDLEY

GEAR HANDBOOK

The Design, Manufacture
and Application of Gears

DARLE W. DUDLEY

EDITOR IN CHIEF

McGRAW-HILL BOOK COMPANY

E7

Property of
J. B. Thomas
GE - S & SE
175 Curtner Ave.
Mail Code 849
San Jose, Ca. 95125

GEAR HANDBOOK

GE-1024.003

OTHER McGRAW-HILL HANDBOOKS OF INTEREST

AMERICAN INSTITUTE OF PHYISCS · American Institute of Physics Handbook
AMERICAN SOCIETY OF MECHANICAL ENGINEERS · ASME Handbooks:
Engineering Tables
Metals Engineering—Design
Metals Engineering—Processes
Metals Properties
AMERICAN SOCIETY OF TOOL AND MANUFACTURING ENGINEERS:
Die Design Handbook
Handbook of Fixture Design
Manufacturing Planning and Estimating Handbook
Tool Engineers Handbook
ARCHITECTURAL RECORD · Time-Saver Standards
BEEMAN · Industrial Power Systems Handbook
BRADY · Materials Handbook
BURINGTON · Handbook of Mathematical Tables and Formulas
BURINGTON AND MAY · Handbook of Probability and Statistics with Tables
CARRIER AIR CONDITIONING COMPANY · Handbook of Air Conditioning System Design
CARROLL · Industrial Instrument Servicing Handbook
CONDON AND ODISHAW · Handbook of Physics
CONSIDINE · Process Instruments and Controls Handbook
CONSIDINE AND ROSS · Handbook of Applied Instrumentation
CROCKER · Piping Handbook
DAVIS · Handbook of Applied Hydraulics
EMERICK · Heating Handbook
FACTORY MUTUAL ENGINEERING DIVISION · Handbook of Industrial Loss Prevention
FLUGGE · Handbook of Engineering Mechanics
HARRIS · Handbook of Noise Control
HARRIS AND CREDE · Shock and Vibration Handbook

HEYEL · The Foreman's Handbook
HUSKEY AND KORN · Computer Handbook
JURAN · Quality Control Handbook
KALLEN · Handbook of Instrumentation and Controls
KING AND BRATER · Handbook of Hydraulics
KNOWLTON · Standard Handbook for Electrical Engineers
KOELLE · Handbook of Astronautical Engineering
KORN AND KORN · Mathematical Handbook for Scientists and Engineers
LASSER · Business Management Handbook
LEGRAND · The New American Machinists' Handbook
MACHOL · System Engineering Handbook
MAGILL, HOLDEN, AND ACKLEY · Air Pollution Handbook
MANAS · National Plumbing Code Handbook
MANTELL · Engineering Materials Handbook
MARKS AND BAUMEISTER · Mechanical Engineers' Handbook
MAYNARD · Industrial Engineering Handbook
MAYNARD · Top Management Handbook
MORROW · Maintenance Engineering Handbook
PERRY · Chemical Engineers' Handbook
PERRY · Engineering Manual
ROSSNAGEL · Handbook of Rigging
ROTHBART · Mechanical Design and Systems Handbook
SHAND · Glass Engineering Handbook
STANIAR · Plant Engineering Handbook
STREETER · Handbook of Fluid Dynamics
TOULOUKIAN · Retrieval Guide to Thermophysical Properties Research Literature
TRUXAL · Control Engineers' Handbook

GEAR HANDBOOK

The Design, Manufacture, and Application of Gears

DARLE W. DUDLEY, Editor

Manager, Advance Gear Engineering
General Electric Company
Lynn, Massachusetts

FIRST EDITION

McGRAW-HILL BOOK COMPANY

New York Toronto London Sydney

GE-1024.006

GEAR HANDBOOK

Library of Congress Catalog Card Number: 61-7304

17902

6 7 8 9 10 11 12 - MAMB - 7 5 4

GE-1024.007

CONTRIBUTORS

MERIWETHER L. BAXTER, Jr., Chief Research Engineer, Gleason Works, Rochester, N.Y. (*Author of Chapter 1, Basic Theory of Gear-tooth Action and Generation*)

ALBERT S. BEAM, Staff Engineer, Vinco Corporation, Detroit, Mich. (*Author of Chapter 23, Gear-inspection Devices*)

SOL BINDER, Chief, Transmission Stress, Vertol Division, Boeing Airplane Company, Morton, Pa. (*Co-author of Chapter 2, Functional Types of Gears*)

F. E. BIRTCH, Manager, Cone-Drive Division, Michigan Tool Company, Detroit, Mich. (*Contribution to Chapter 23, Gear-inspection Devices*)

FRED BOHLE, Manager of Spiroid Division of Illinois Tool Works, Chicago, Ill. (*Contribution to Chapter 5, Gear-tooth Proportions*)

BEN F. BREGI, Vice President—Manufacturing and Research, National Broach and Machine Company, Detroit, Mich. (*Author of Chapter 18, Gear Finishing by Shaving, Honing, and Lapping*)

E. G. CHARTIER, President, Tekniks Engineering Company, Elmhurst, Ill. (*Author of Chapter 17, Gears Produced by Dies*)

WELLS COLEMAN, Chief Gear Analyst, Gleason Works, Rochester, N.Y. (*Contributions to Chapter 5, Gear-tooth Proportions, and Chapter 13, Load Rating of Gears*)

WARNER T. COWELL, Chief Engineer, Gear Engineering Test Center, Gleason Works, Rochester, N.Y. (*Contribution to Chapter 12, Gear Reactions and Mountings*)

S. L. CRAWSHAW, Vice President of Philadelphia Gear Corporation. (*Contribution to Chapter 3, Gear Arrangements*)

PAUL M. DEAN, Jr., Gear Technology Engineer, General Engineering Laboratory, General Electric Company, Schenectady, N.Y. (*Author of Chapter 5, Gear-tooth Proportions, and Chapter 6, Center Distance*)

DARLE W. DUDLEY, Manager, Advance Gear Engineering, General Electric Company, Lynn, Mass. (*Author of Chapter 3, Gear Arrangements. Contribution to Chapter 5, Gear-tooth Proportions. Author of Chapter 10, Gear Materials. Contribution to Chapter 23, Gear-inspection Devices*)

THOMAS W. EDWARDS, Senior Engineer, Engineering Department, Vinco Corporation, Detroit, Mich. (*Contribution to Chapter 23, Gear-inspection Devices*)

M. L. FALLON, Senior Application Design Analyst, Gear Engineering Test Center, Gleason Works, Rochester, N.Y. *(Contribution to Chapter 12, Gear Reactions and Mountings)*

WAYNE H. GLOVER, Vice President and General Manager, Indiana Gear Works, Inc., a Division of the Buehler Corporation, Indianapolis, Ind. *(Author of Chapter 19, Grinding Spur and Helical Gears)*

HARRY J. HART, Sales Engineer, Gleason Works, Rochester, N.Y. *(Co-author of Chapter 20, Bevel- and Hypoid-gear Manufacturing)*

MARTIN A. HARTMAN, Research Specialist, Rocketdyne Division of North American Aviation, Canoga Park, Calif. *(Author of Chapter 12, Gear Reactions and Mountings)*

RICHARD S. HILDRETH, Chief Engineer, Michigan Tool Company, Detroit, Mich. *(Author of Chapter 22, Gear-cutting Tools)*

CLARENCE JOHNSON, Superintendent, Utility Manufacturing Company, North Wilbraham, Mass. *(Contribution to Chapter 17, Gears Produced by Dies)*

STUART J. JOHNSON, Chief Engineer, Small Tools Division, Barber-Colman Company, Rockford, Ill. *(Author of Chapter 16, Gear Cutting)*

T. W. KHIRALLA, Staff Engineer, Curtiss-Wright Corporation, Propeller Division, Caldwell, N.J. *(Author of Chapter 21, Worm and Wormgear Cutting and Grinding Processes)*

CHARLES B. KING, Associate Chief Research Engineer, Gleason Works, Rochester, N.Y. *(Contribution to Chapter 8, Calculation of Gear-tooth Profiles. Co-author of Chapter 20, Bevel- and Hypoid-gear Manufacturing)*

LEO KINGSTON, Chief, Transmission Design, Vertol Division, Boeing Airplane Company, Morton, Pa. *(Co-author of Chapter 2, Functional Types of Gears)*

EDWIN R. KITCHEN, Gear and Cutter Engineer, Farrel-Birmingham Co., Inc., Rochester, N.Y. *(Contribution to Chapter 16, Gear Cutting)*

THEODORE F. KLEM, Gear Engineer, Gleason Works, Rochester, N.Y., *(Contribution to Chapter 23, Gear-inspection Devices)*

HAROLD KRON, Chief Engineer, Philadelphia Gear Corporation. *(Contribution to Chapter 3, Gear Arrangements)*

ROY E. LANDSTROM, Chief Tool Designer, Michigan Tool Company, Detroit, Mich. *(Contribution to Chapter 22, Gear-cutting Tools)*

A. J. LEMANSKI, Supervisor of Gear Engineering, Lycoming-Avco, Stratford, Conn. *(Contribution to Chapter 17, Gears Produced by Dies)*

GEORGE L. McCAIN, Gear Consultant, Birmingham, Mich. *(Author of Chapter 11, Gear Drawings)*

GEORGE W. MICHALEC, Head of Special Products Department, GPL Division—General Precision, Inc., Pleasantville, N.Y. *(Author of Chapter 9, Gear Tolerances)*

WOODROW NELSON, Chief Engineer, The Spiroid Division of Illinois Tool Works, Chicago, Ill. *(Contribution to Chapter 5, Gear-tooth Proportions)*

CARL PARKER, Assistant Chief Tool Engineer, The Fellows Gear Shaper Company, Springfield, Vt. *(Contribution to Chapter 5, Gear-tooth Proportions)*

ROBERT T. PARSONS, President, The Van Keuren Company, Boston, Mass. *(Author of Chapter 24, Numerical Data Tables)*

HARRY PEDERSEN, Chief Design Analyst, Gleason Works, Rochester, N.Y. *(Co-author of Chapter 20, Bevel- and Hypoid-gear Manufacturing)*

FOLKE RICHARDZ, Consulting Specialist, Gearing Products, Westinghouse Electric Corporation, Buffalo, N.Y. *(Author of Chapter 15, Lubrication of Gears)*

GERALD L. SCOTT, Staff Engineer, American Gear Manufacturers Association, Washington, D.C. *(Author of Chapter 4, Elements of Gears and Basic Formulas. Contribution to Chapter 5, Gear-tooth Proportions)*

JOHN C. SEARS, Executive Director of American Gear Manufacturers Association, Washington, D.C. *(Author of Foreword)*

GEORGE SHIMMIN, Sales Manager, Cone Drive Division, Michigan Tool Company, Detroit, Mich. *(Contribution to Chapter 5, Gear-tooth Proportions)*

EUGENE E. SHIPLEY, Project Engineer—Advance Development, General Electric Company, Lynn, Mass. *(Author of Chapter 14, Loaded Gears in Action)*

GILMORE M. SPEAR, Research Engineer, Gleason Works, Rochester, N.Y. *(Co-author of Chapter 20, Bevel- and Hypoid-gear Manufacturing)*

M. F. SPOTTS, Professor of Mechanical Engineering, Northwestern University, Evanston, Ill. *(Author of Chapter 7, Special Calculations for Spur and Helical Gears)*

ANTONI TABAK, Gear Metallurgist, General Electric Company, Lynn, Mass. *(Contribution to Chapter 10, Gear Materials)*

CLARENCE UNDERWOOD, Gear Consultant, Raytheon Company, Wayland, Mass. *(Contribution to Chapter 10, Gear Materials)*

EDWARD C. VARNUM, Head, Operations Research, Barber-Colman Company, Rockford, Ill. *(Author of Chapter 8, Calculation of Gear-tooth Profiles)*

GARY L. VESEY, Mechanical Engineer, Spiroid Division of Illinois Tool Works, Chicago, Ill. *(Contribution to Chapter 23, Gear-inspection Devices)*

E. J. WELLAUER, Director of Research and Development, The Falk Corporation, Milwaukee, Wis. *(Author of Chapter 13, Load Rating of Gears)*

HAROLD WINZELER, Vice President and General Manager, Winzeler Manufacturing Company, Chicago, Ill. *(Contribution to Chapter 17, Gears Produced by Dies)*

FOREWORD

The design and manufacture of a gear that is no better than the requirement for its successful application is an art rather than a science.

Volumes have been written and said in attempts to attain good gears. Many of them have been rather unsuccessful interpretations of what their authors opined previous authors meant or failed to say.

Thus, much of the information comes under what I call "gear folklore" —just about as obsolete as the cogwheel that sired the quality of gearing we need in modern applications.

The authors of this "Gear Handbook" are seasoned and experienced gear men. They have spent most of their careers in the study of gear phenomena, in tempering theory in the furnace of practicality and economic necessity. Several of them have received the highest honor the American Gear Manufacturers Association has to bestow on contributors to the gear art, its "Edward P. Connell" Award.

These dedicated authors set out to give the gear designer and manufacturer a true and accurate recording of the present state of the art.

They have eminently succeeded in so doing. Therefore, I personally and on behalf of AGMA enthusiastically commend this "Gear Handbook" to all who have a genuine desire and economic need to acquaint themselves with the gear art as it is now being practiced by forward-looking, down-to-earth gear designers and manufacturers.

J. C. Sears, *Executive Director*
American Gear Manufacturers Association

PREFACE

Although a rather surprising amount of material has already been written on the subject of gears, it has been hard for the average person to quickly find the detailed gear information that he needs. A wealth of gear information has already been published in trade standards, magazine articles, manufacturers' catalogues, private company reports, and gear books. The key information to solve the design of a gear, plan tooling for a gear job, analyze a gear failure, or pick an appropriate manufacturing process may be hidden almost anywhere in current available technical literature.

The "Gear Handbook" was conceived as a means of solving this problem. In approximately one thousand pages of moderately fine print all aspects of the gear art are covered. The questions that the gear designer, the gear builder, or the gear user may have are all covered. Condensed information is given on the many types of gears that are now in use. Specific information is given on the great diversity of manufacturing processes that may be used. Considering that the field of gearing goes all the way from the little mass-produced gears for toys, clocks, instruments, and the like, up to the massive main propulsion units used for major ships or airplanes, it can be appreciated that the manufacturing techniques will vary widely.

The people who design and build gears comprise only a small fraction of the people engaged in manufacturing work. However, the people who use gears include almost everyone. It has been said quite truthfully that you can't tell the time of day without gears, you can't wash your clothes without gears, you can't drive to work without gears, and you can't make anything in your shop without gears. The user of gears is often very interested in things like the performance of gear equipment, the life expectancy of gearing, or the relative cost of different ways of making gears. Subject matter for the "Gear Handbook" has been selected with the aim of covering the whole wide field of interest in the gear art.

Chapters 1 through 4 introduce the reader to the basic theory of gearing, define the types of gear teeth, explain the many possible gear arrangements, and define the nomenclature and basic geometry of gears.

All of this is necessary background material to acquaint a person with the language and definitions peculiar to the gear trade.

Chapters 5 through 15 are the prime engineering-design chapters. These cover the elements of design, such as tooth proportions, involute calculations, choice of materials, stress calculations, load rating, tolerancing of gear dimensions, efficiency calculations, and design of lubrication systems.

Chapter 14 presents material on how to test gears and how to analyze the several possible kinds of gear failure.

Chapters 16 through 21 cover the machine tools and methods of gear manufacture. Gear cutting, gear grinding, die forming of gears, gear shaving, gear lapping, and gear honing are all covered in these chapters.

Chapter 22 covers tools to make gears, and Chapter 23 covers inspection tools. Chapter 24 presents many tables of data useful in the gear trade.

Over three dozen people were direct contributors to the project. These people were selected because of their knowledge and competence in the various aspects of gear work. The technical knowledge that this group of people was able to put together over a four-year working period represents a depth of technical knowledge in this field that is far beyond that of any one individual. From my own standpoint it has been a most rewarding experience to be associated with this project and learn from experts in all phases of gear work. In many cases, information collected for the "Gear Handbook" represents the first time this information has ever been put in public print.

I am very grateful to the many contributors for their unselfish effort in getting the required information and laboring through all the trials of putting it into handbook style. In addition, I feel particularly grateful to certain key individuals who were most helpful in providing technical guidance for the project and in making the resources of their organizations available to ferret out key items of information. My special thanks go to the officers and Headquarters staff of the American Gear Manufacturers Association, the "engineering body for the gear industry," for their assistance in checking many parts of this work. I also wish to give special thanks to Wells Coleman of the Gleason Works for his assistance in the bevel-gear field; to Mr. Gerald Brophy of International Nickel Co. for his advice on gear material developments; to Mr. Fred E. Birtch, Cone-Drive Division, Michigan Tool Co., for his assistance on double-enveloping wormgears; to Mr. S. L. Eastman, Cleveland Worm & Gear Division, Eaton Manufacturing Co., for his help on cylindrical wormgears, and to Mr. F. Bohle and Mr. W. Nelson, Spiroid Division of Illinois Tool Works, for their assistance on the Spiroid family of gears. Mr. Laurence Collins and others in the General Electric Company were very helpful in encouraging me to carry out this project and in making

GE-1024.013

available to me extensive test and field data on spur and helical gears.

In spite of the fact that most of the gear technical material was received from contributors and others associated with this project, I must accept the responsibility for having made the final decisions that determined the technical content of this work. I believe this "Gear Handbook" will represent an important step forward in the development of knowledge in the gear field. It makes this knowledge available for the widest possible benefit of the great host of people who buy, design, build, or use geared equipment.

DARLE W. DUDLEY

Lynnfield, Mass.

CONTENTS

GE-1024.022

GEAR HANDBOOK

Chapter 3

GEAR ARRANGEMENTS*

By

DARLE W. DUDLEY, *Manager, Advance Gear Engineering, General Electric Company, Lynn, Mass.*

Gears may be arranged in a great variety of ways. In the more complex pieces of machinery, the problem of arranging and fitting the gear drives into available space is one of the major gear-design problems.

This chapter makes an over-all survey of the kinds of gear arrangements in common use. The clever designer will appreciate that there are an infinite number of arrangement possibilities for the more complex machinery drives. The arrangements shown in this chapter should be considered as examples of gear arrangements rather than as a complete listing of the possibilities of the field.

Chapter 3 can be best understood if the reader is already familar with the *theory* of gearing given in Chap. 1 and the *kinds* of gear teeth given in Chap. 2.

3-1. Possibilities in Gear Arrangements. In general terms a *gear drive* implies one toothed member that engages another toothed member and transmits motion to the second toothed member. It is usually intended† that the second toothed member should rotate at a uniform angular velocity when the first member is driven at a smooth angular velocity.

The second toothed member may be externally toothed or internally toothed, or it may be a section of a *rack* (a rack is a gear of infinite pitch diameter). A spur or helical pinion can be used to mesh with any of these three gear types. Many of the other gear types cannot mesh with the gear member if it is anything other than externally toothed. In some cases it is the geometry of the gear part that is limiting. In most cases, though, the gear part might be made in theory, but the machinery normally available in the gear trade does not have the capability of making the part.

Some types of gear teeth are *interchangeable* and some are not (see Art. 1-4). In an interchangeable-gear-tooth system, toothed gear wheels *a*, *b*, *c*, and *d* might be made. These might be meshed in a chain with *a* meshing with *b*, *b* meshing with *c*, and *c* meshing with *d*. If these gears were interchangeable, they could be taken apart

* A considerable amount of material for this chapter was furnished by S. L. Crawshaw, Vice President of Philadelphia Gear Corp., and Harold Kron, Chief Engineer, Philadelphia Gear Corp.

† In gearing for watches a nonuniformity in motion is often tolerated to cut down the friction when a gear drives a pinion of small tooth numbers. In computing machines elliptical or other noncircular gears are sometimes used to create a motion corresponding to some mathematical formula.

and assembled and run properly with a meshing with c or b meshing with d. Generally speaking *involute* profile gears have the property of interchangeability and noninvolute gears do not. Spur and helical gears are normally made with involute teeth, and when so made they have interchangeability. An exception to this is the single-enveloping wormgear where interchangeability does not exist even when the worm member is made to an involute profile.

In some gear types it is geometrically rather impractical to make the pinion member with as few teeth as one. A spur pinion with only one tooth would not be capable of continuous gear action. On the other hand, a worm with one thread is very practical and is often used. Some gear types can be made with the pinion member having as few as five teeth. Most gear types can have pinions with over 16 teeth. A single-enveloping worm, however, is somewhat unhandy to manufacture at 16 threads, and there is seldom any good design reason to use this many threads on a worm.

Table 3-1 shows in tabular form the meshing possibilities of all the major types of gear teeth. This table is based on normal trade practice and should not be considered as absolutely firm. For instance, an *internal* bevel gear is theoretically possible and has been used in a few special cases. However, normal bevel-gear-manufacturing machines will not make such a gear.

Table 3-1. Gear-meshing Possibilities

Type of gear teeth	Pinion and gear	Pinion and rack	Pinion and internal gear	Interchange-ability *a-b-c-d* or *a-c, b-d*	One-tooth pinion	Pinion of 5 teeth	Pinion of 16 or more teeth
Spur	Yes	Yes	Yes	Yes	No	No*	Yes
Helical	Yes	Yes	Yes	No	No*	No*	Yes
Straight bevel	Yes	No*	No	No*	No	No*	Yes
Zerol®† bevel	Yes	No	No	No	No	No*	Yes
Spiral bevel	Yes	No	No	No	No*	No*	Yes
Hypoid	Yes	No	No	No	Yes	Yes	Yes
Face gear	Yes	No	No	No	No	No*	Yes
Crossed-helical	Yes	Yes	No	Yes	Yes	Yes	Yes
Single-enveloping worm	Yes	No*	No*	No	Yes	Yes	No*
Double-enveloping worm	Yes	No	No	No	Yes	Yes	No*
Beveloid®†	Yes	Yes	No	Yes	No	No*	Yes
Spiroid®†	Yes	No	No	No	Yes	Yes	No*
Planoid®†	Yes	No	No	No	No*	Yes	Yes
Helicon®†	Yes	No	No	No	Yes	Yes	Yes

* Items with asterisk indicate that it is mechanically possible to have this design but that the design is not normally used in the gear trade.

† Trade-marks: Zerol, registered trade-mark of Gleason Works, Rochester, N.Y.; Beveloid, registered trade-mark of Vinco Corp., Detroit, Mich.; Spiroid, Planoid, Helicon, registered trade-marks of Illinois Tool Works, Chicago, Ill.

A worm and an internal gear might be construed as simply a nut and bolt. Nuts and bolts are, of course, very practical. However, they are not normally considered as *gear* parts.

3-2. How the Arrangement May Drastically Affect Ratio, Power, Efficiency, and Gearbox Volume. The arrangement of the gear drive can play a very important part on how easy or hard it is to achieve things like high ratio, high power capacity, high efficiency, or a very compact unit.

In ordinary spur- or helical-gear practice, it is reasonable to handle ratios from 1:1 to about 8:1 in a single reduction. Ratios of even 10:1 are quite good. Suppose a ratio of 120:1 was needed. This would require a triple-reduction unit of spur or helical gears. It would also take a triple-reduction bevel-gear unit. A triple-reduction unit has three sets of pinions and gears, or a total of six toothed parts. With a wormgear set it is possible to handle this much ratio in just a single reduction (only two toothed parts).

For each toothed part in a gearset, a shaft and generally two bearings are required. Where large ratios are needed, the choice of gear arrangement is often determined by finding the arrangement with the fewest number of parts that will do the job adequately.

Table 3-2 gives some general information about how different ratios can be achieved and the number of gear parts required.

Table 3-2. How to Obtain Ratios

Kind of arrangement	Min. No. of toothed parts	Ratio range		
		5:1	50:1	100:1
Single reduction:				
Spur	2	Yes	No	No
Helical	2	Yes	No	No
Bevel	2	Yes	No	No
Hypoid	2	Yes	Yes	Yes
Face	2	Yes	No	No
Worm	2	Yes	Yes	Yes
Spiroid	2	No	Yes	Yes
Planoid	2	Yes	No	No
Simple planetary	3	Yes	No	No
Fixed differential	5	No	Yes	Yes
Planocentric	2	No	Yes	Yes
Harmonic Drive	2	No	Yes	Yes

The power-transmitting capacity of different gear arrangements is quite variable. For instance, there has never been a wormgear set that would handle 5,000 hp as a continuous duty rating. On the other hand, many helical gearsets are in regular service carrying more than 10,000 hp. The upper limit of power for gear arrangement is about as hard to define precisely as it is to define the altitude at which the earth's atmosphere ceases. The upper limit of a gear's capacity is determined by such things as

1. How large a gear can be made by available gear-manufacturing equipment?
2. What is the upper limit on tooth-surface loading and pitch-line velocity for the best material available?
3. Are the job requirements of ratio, input speed, life required, and lubrication favorable for a maximum horsepower?

Table 3-3 shows some nominal maximum horsepower capacities for different gear arrangements. These should be used only as a general guide. In all cases there are a few gearsets in use (or that could be put into use) at considerably higher horsepower ratings. Conversely, the job requirements of many applications make it completely impractical to achieve the ratings shown for the specific application. No horsepower limits are shown for the fixed differential, the planocentric, or the Harmonic Drive. These kinds of units have been used mostly at low horsepowers (less than 50 hp), but they can probably be extended up into fairly high horsepower.

Table 3-3. General Survey of Power and Efficiency

Kind of arrangement	Nominal max. hp	Typical efficiency, %		
		5:1 ratio	50:1 ratio	100:1 ratio
Single reduction:				
Spur	3,000	98		
Helical	30,000	98		
Straight bevel	500	98		
Zerol bevel	1,000	98		
Spiral bevel	5,000	98		
Hypoid	1,000	95	80	60
Crossed-helical	100	95	80	60
Cylindrical worm	750	95	80	60
Double-enveloping worm	1,000	95	80	60
Spiroid	500	95	80	60
Planoid	1,000	95		
Helicon	100	95	80	60
Double reduction:				
Spur	3,000	97	96	94
Helical	30,000	97	96	94
Spiral bevel	5,000	97	96	
Simple planetary	10,000	97		
Fixed differential	......	...	80	60
Planocentric	......	...	90	85
Harmonic Drive	......	...	90	85

For specific data on load rating for a given gear design, Chap. 13 should be consulted.

The efficiency that can be achieved in a gearset is quite variable. The kind of oil, the amount of pitch-line velocity, and the kind of bearings enter quite directly into the efficiency. In a spur gearset, the best designs made under favorable conditions will achieve as much as 99 per cent efficiency. Some spur sets under poor conditions may not even achieve 90 per cent efficiency.

Table 3-3 shows the nominal order-of-magnitude efficiencies for different gear arrangements. These represent general trade experience with good designs but are not the best possible efficiency that can be achieved with an optimum design and a favorable application.

Table 3-3 shows that the efficiency drops off rather rapidly as the ratio increases. Generally this is caused by the fact that the sliding velocity increases rather rapidly as ratio increases. A worm, for instance, at high ratio has a very large amount of sliding.

The fixed-differential gear is capable of high ratio, but it too has a drop-off in efficiency. This is caused by the fact that a large amount of tooth meshing occurs at high torque (see Art. 3-11).

Suppose a 100:1 ratio is needed. So far as arrangement goes the double-reduction approach using either spur, helical, or bevel gears is the easiest way to achieve high efficiency. With a favorable application and a good design an efficiency of 96 per cent can be achieved with helical gears. It is quite impossible to obtain this kind of efficiency with any of the single-reduction gears that can reach a 100:1 ratio.

Chapter 14 covers the general subject of gear efficiency and shows curves for many gear types. This chapter should be consulted for detail information on gear efficiency.

The volume or weight of a gearbox is also considerably affected by the arrangement. The weight of a gearbox to do a given job may vary as much as 50:1! For instance, a single-reduction helical-gear unit made of low-hardness material in industrial practice might weigh 10,000 lb. An aircraft-style planetary made of fully hardened gears in aircraft practice might handle the same horsepower, for the same design life, and with the same degree of reliability as the industrial gear and weigh only 200 lb.

In very general terms, a 50:1 spread in gear weight can be broken down as follows:

Weight ratio for low tooth loads on low-hardness material vs. high loads on high-hardness material	7:1
Weight ratio for heavy construction using thick walls and solid shafts vs. thin walls and hollow shafts	3:1
Weight ratio of best arrangement to save weight vs. normal arrangement	2½:1

The opportunity to save weight in gear designs depends somewhat on the ratio required. For instance, a worm gearset is not particularly lightweight at a low ratio of 5:1. At a high ratio of 50:1 the wormgear becomes rather light because it can still do the job with just two toothed parts, whereas a spur or helical set would take at least twice as many parts to do the job.

Table 3-4. Gearbox Relative Size and Weight

Kind of arrangement	Ratio range			
	5:1	20:1	50:1	100:1
Single reduction:				
Spur, helical, bevel	Small			
Worm	....	Small	Small	Small
Hypoid	Small	Small	Small	Small
Spiroid	....	Small	Small	Small
Planoid	Small			
Double reduction:				
Single power path, helical gears	....	Med. size		
Multiple power path, helical gears	....	Small	Very small	
Epicyclic gears:				
Simple planetary	Very small			
Compound planetary	....	Very small		
Double-reduction planetary	....	Very small	Very small	
Fixed differential	....	Small	Very small	Very small

Table 3-4 shows a general picture of what arrangements might be small for different gear ratios. In using this table, it should be kept in mind that intensity of tooth loading is often more important than arrangement in reducing weight. For instance, a casehardened gearset used in a nominal arrangement might give a lighter unit than a set of low-hardness cut gears used in the most weight-saving arrangement.

In using Table 3-4 it should also be kept in mind that the amount of horsepower must be favorable for the design. Some designs cannot handle high horsepower. These designs are small in weight only in comparison with other designs when the horsepower required is well within their capability.

Generally speaking, the volume of a gearbox is fairly proportional to the weight—provided that the type of construction is the same. Aircraft-type gearboxes with magnesium casings will have more volume for their weight than industrial boxes with cast-iron casings.

3-3. Parallel-axis Arrangements, Simple Mesh. Perhaps the most common gear arrangement is that of a pinion and gear meshing on parallel axes. The teeth for this kind of arrangement may be either spur, helical, or herringbone.*

A single pinion and gear make a *single*-reduction unit, or an S unit in popular terminology. If two such units are in series, it is a *double*-reduction unit or a D unit. A *double*-reduction unit may have all the gears mounted in a common casing, or the unit may be made up of two somewhat separate casings which are attached to each other. In a like manner it is possible to have a *triple*-reduction unit or a *quadruple*-reduction unit.

The ratio for such units is figured quite simply

$$m_G = m_{G_1} m_{G_2} m_{G_3} \text{ (etc.)} \tag{3-1}$$

where

$$m_{G_1} = \frac{N_{G_1}}{N_{P_1}}$$

N_{G_1} = number of gear teeth, first reduction
N_{P_1} = number of pinion teeth, first reduction

Quite a variety of arrangements are possible with parallel axes and simple meshing of external teeth. Besides multiple reductions, there may be multiple power inputs or multiple power outputs. There may also be multiple power paths through the unit. The latter type is analogous to parallel circuits in electrical wiring.

An example of the multiple power path is the *double-reduction twin* unit, or DT gear. In this case a first-reduction pinion meshes with two first-reduction gears. These first-reduction gears in turn are connected to two second-reduction pinions. The second-reduction pinions drive a common second-reduction gear.

The reduction ratio of a multiple-power-path gear unit is calculated by following through any one power path and using Eq. (3-1).

Figure 3-1 shows schematic arrangements of many of the more popular arrangements used in turbine-drive applications. These same arrangements are used frequently in lower-speed motor drives; but the lower-speed units frequently have fewer bearings and flexible couplings than shown in Fig. 3-1. The number of toothed gear elements would be the same in either case.

Figure 3-2 shows a typical high-speed single-reduction unit. Units like this are often used with steam-turbine drives. They may be used with electric motors, diesel engines, or gasoline engines. Many sets are also used with gas turbines.

High-speed gears used for power in industrial plants, public-utility plants, or for drives on large ships have horsepower ratings going from less than 1,000 to over 50,000 hp.

Figure 3-3 shows a DT-type high-speed gear rated at 4,000 hp. Note the close connection of the turbine and the gear unit.

Figure 3-4 shows a typical MD-type marine gear unit with a 17,500-hp rating. In this case two turbines drive a single propeller.

Articulated Gear Unit. In a double-reduction design the first-reduction gear may be solidly mounted on the second-reduction pinion shaft with only two bearings supporting the assembly. Or the pinion and gear may be each mounted on a pair of bearings and the two connected by a flexible quill shaft. The latter arrangement is called an "articulated design." The articulated design is quite popular for the higher-speed higher-horsepower gear units. It has the advantage of providing flexibility between a first-reduction gear and a second reduction which will reduce dynamic† tooth loading, and the fact that both the gear and pinion have their own separate

* Chapter 2 defines the *kinds* of gear teeth. Chapter 4 defines and explains all kinds of gear nomenclature.
† See Chap. 14 for information on dynamic tooth loading.

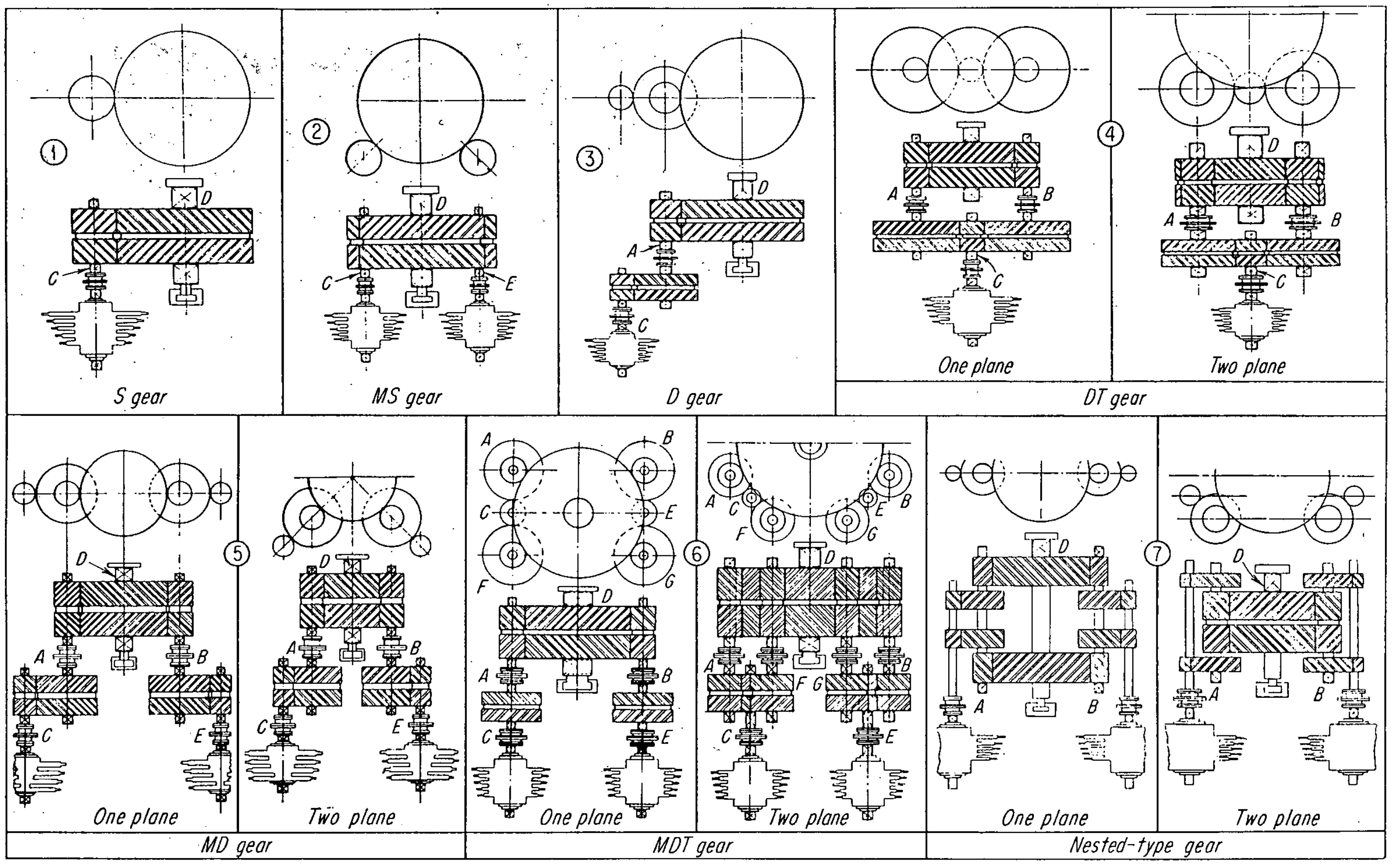


FIG. 3-1. Some schematic arrangements for parallel-axis gears.

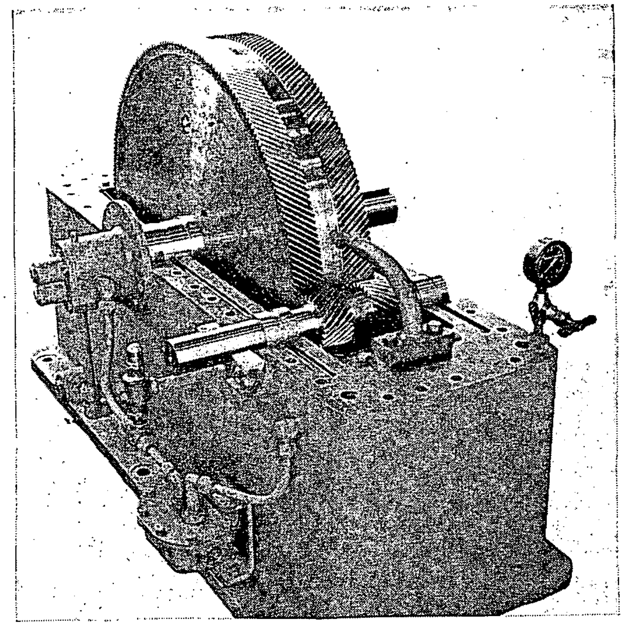


FIG. 3-2. High-speed single-reduction unit with oil pump and force-feed system. (*Courtesy of General Electric Co., Lynn, Mass.*)

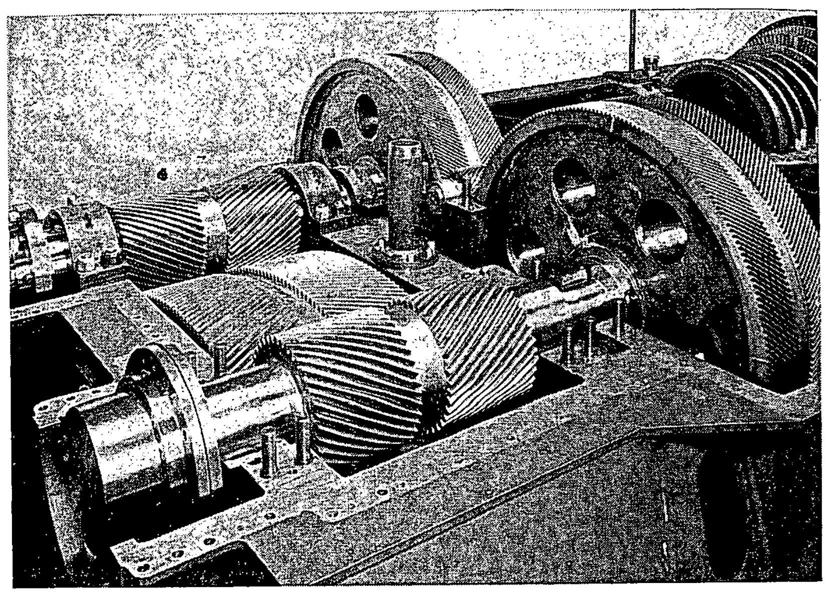


FIG. 3-3. Partly disassembled DT-type gear unit for ship propulsion rated at 4,000 hp. (*Courtesy of General Electric Co.*)

pair of bearings makes it possible to provide a better control on the contact across the face width at each mesh.

Figure 3-5 shows a schematic arrangement for an articulated MDT unit. Further details of an articulated design are shown in Fig. 3-6. Figure 3-7 shows an MDT unit of articulated design that was used as a main drive unit on a super aircraft carrier.

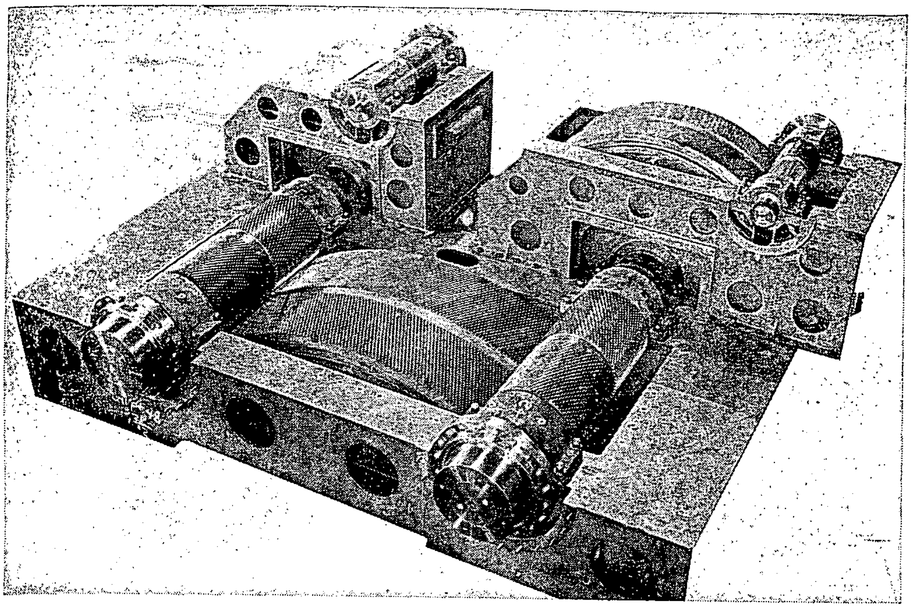


FIG. 3-4. Marine MD-type gear unit with covers off. Rating 17,500 hp. (*Courtesy of General Electric Co.*)

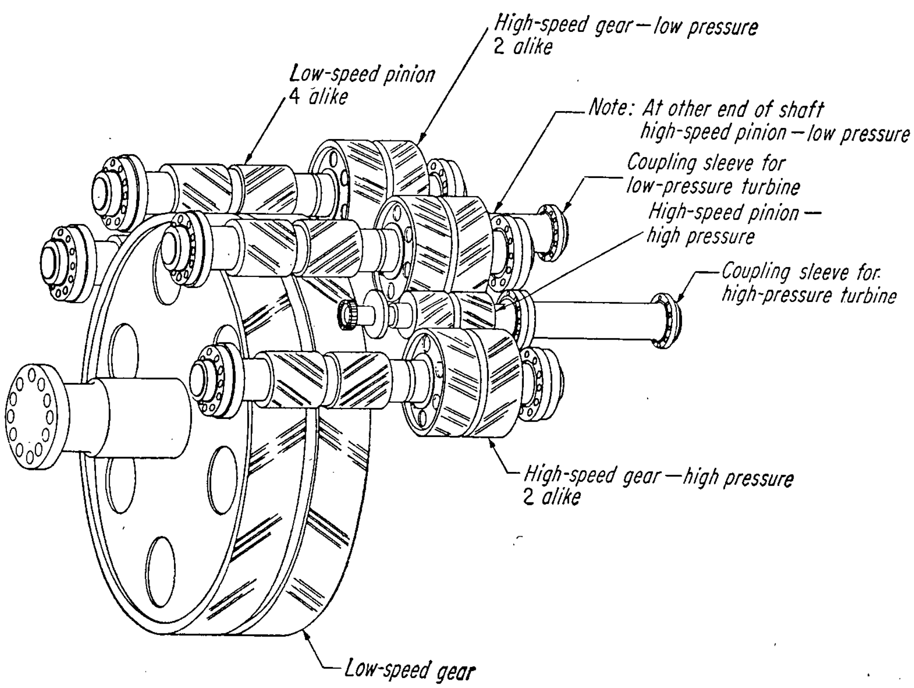


FIG. 3-5. Schematic arrangement of articulated MDT unit.

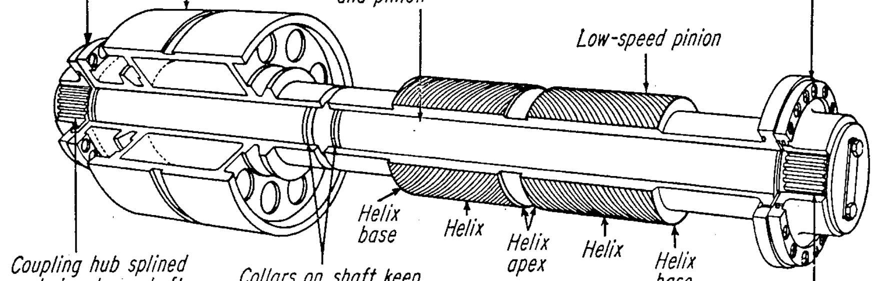


FIG. 3-6. Detail of articulated design of high-speed gear and low-speed pinion. Note intermediate shaft passing through hollow pinion and gear and the fact that both pinion and gear have their own separate bearings.

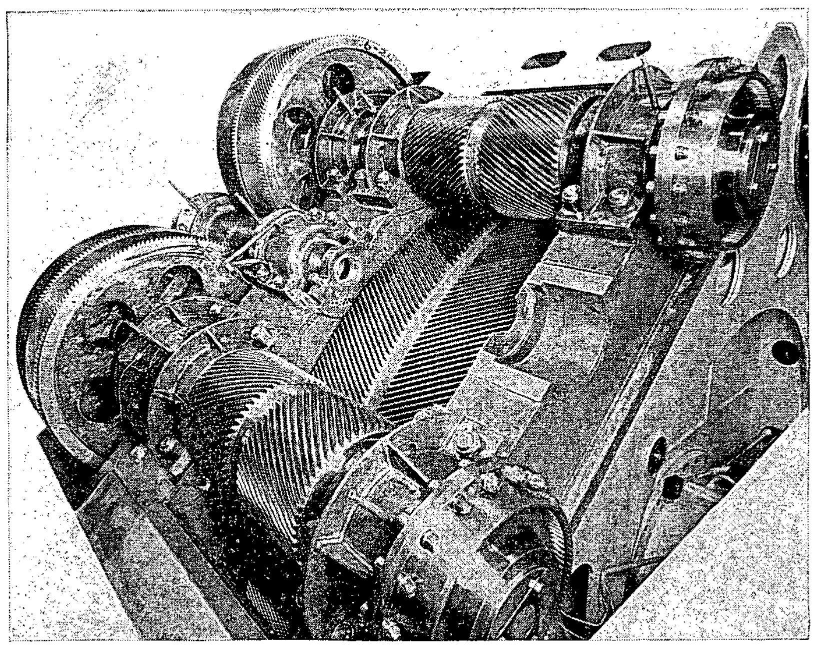


FIG. 3-7. Side view of MDT unit of type used on super aircraft carrier *Saratoga*. Note articulated design. Rating in excess of 50,000 hp. (*Courtesy of U.S. Navy and General Electric Co.*)

Nested Design. It is possible to make a unit somewhat more compact and use fewer bearings by having one reduction straddle another reduction. This style of gear is called the "nested design." Figure 3-1 in part 7 shows the schematics of a nested design. Figure 3-8 shows a typical double-reduction nested unit.

The nested unit has been widely used for low-speed industrial gearing, and it has been used for high-speed high-horsepower marine units.

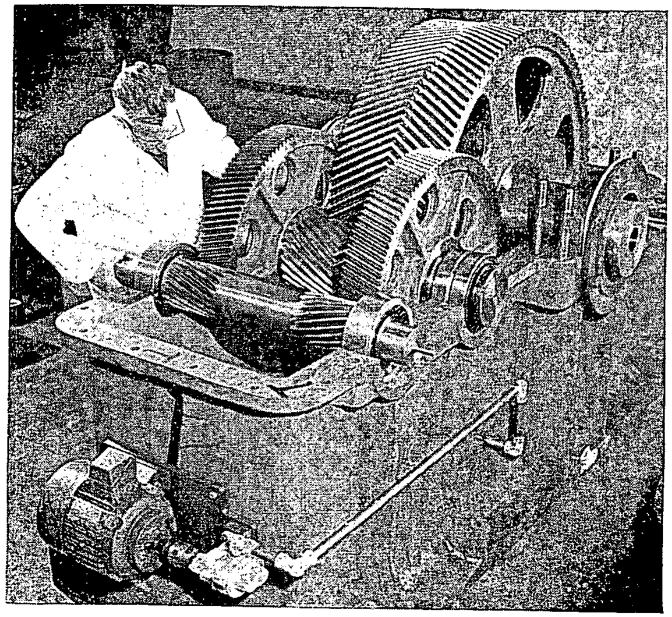


FIG. 3-8. Double-reduction nested gear unit. (*Courtesy of Philadelphia Gear Corp., Philadelphia, Pa.*)

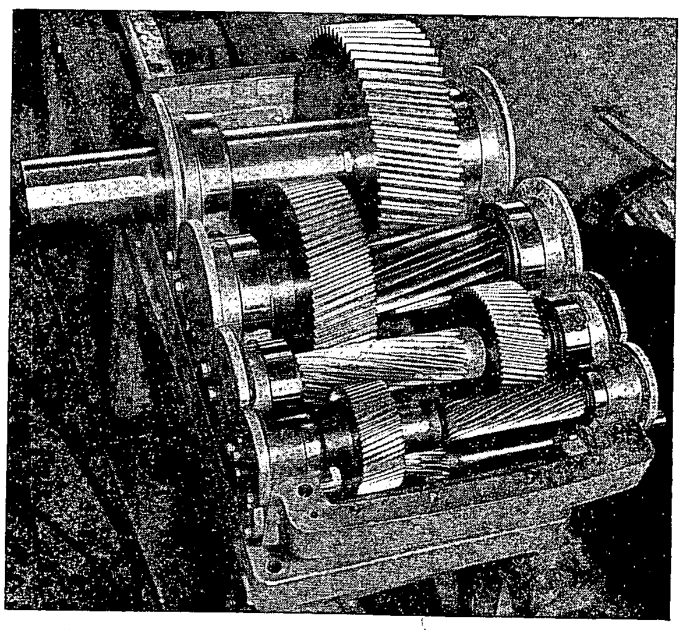


FIG. 3-9. Compact arrangement of a quadruple-reduction gear unit. (*Courtesy of Philadelphia Gear Corp.*)

Figure 3-9 shows a very compact quadruple-reduction unit. This style of unit is quite popular in general gear work. This design is not nested.

Parallel-axis Gears Combined with Right-angle Gears. In many cases it works out well to take a basic parallel-axis gear unit and add a first or last reduction of right-angle gearing. This will permit the drive to turn a right-angle corner, and it may have merit from the standpoint of making the drive more compact or using fewer bearings and other parts.

Figure 3-10 shows a triple-reduction unit with a bevel first reduction. Figure 3-11 shows a double-reduction unit with a bevel second reduction. Both these units demonstrate how a right-angle drive can be added to parallel-axis gearing and a compact drive achieved.

Figure 3-12 shows a double-reduction unit with a cylindrical-worm second reduction. A unit like this can handle over-all ratios of 50:1 to 200:1 with ease. The worm set operates at the slower speed where it has its best torque capacity for its size. A unit of this type is quite popular in industrial work where the output speed is quite low and the horsepower handled is less than 500.

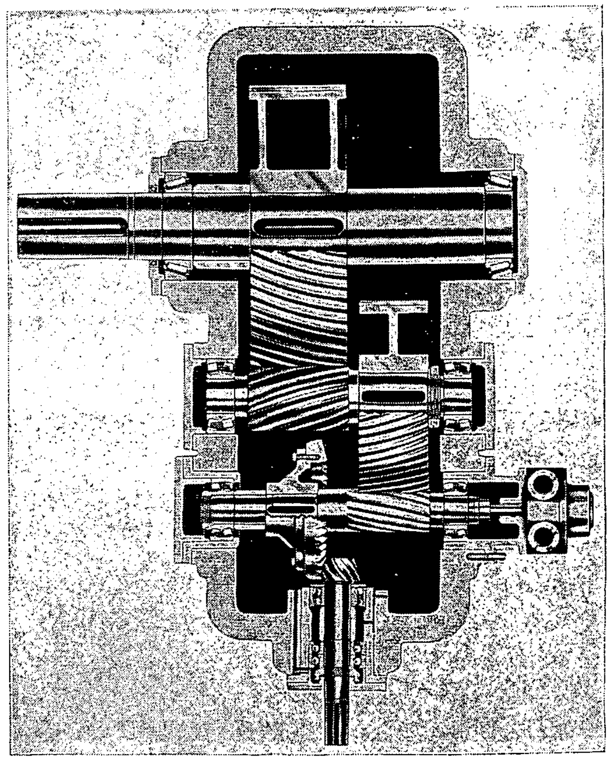


FIG. 3-10. Triple-reduction unit using a bevel first reduction. (*Courtesy of Philadelphia Gear Corp.*)

Design Problems with Parallel-axis Gears. There are several problems in parallel-axis gear design that are more or less peculiar to this arrangement. In picking an arrangement the designer should be alert to the design problems and satisfy himself that he can design the arrangement he has picked so as to obtain a practical design to manufacture and a reliable design in the field.

Some of the problems that may show up are how to

1. Obtain uniform tooth loading across the face width.
2. Compensate for excessive wind-up of long skinny pinions or long shafts.
3. Obtain equal torque through each power path in a multiple-power-path design.
4. Choose numbers of teeth, gear locations, and gear-to-shaft connections so that multiple-power-path gearing will assemble.

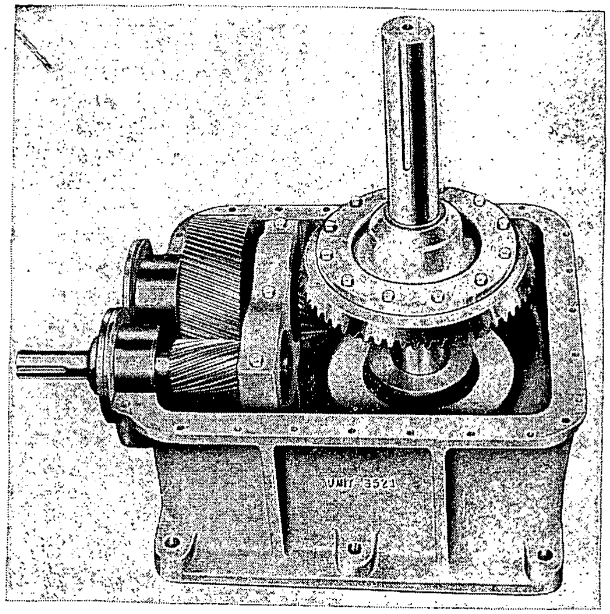


FIG. 3-11. Double-reduction unit with a bevel second reduction. (*Courtesy of Philadelphia Gear Corp.*)

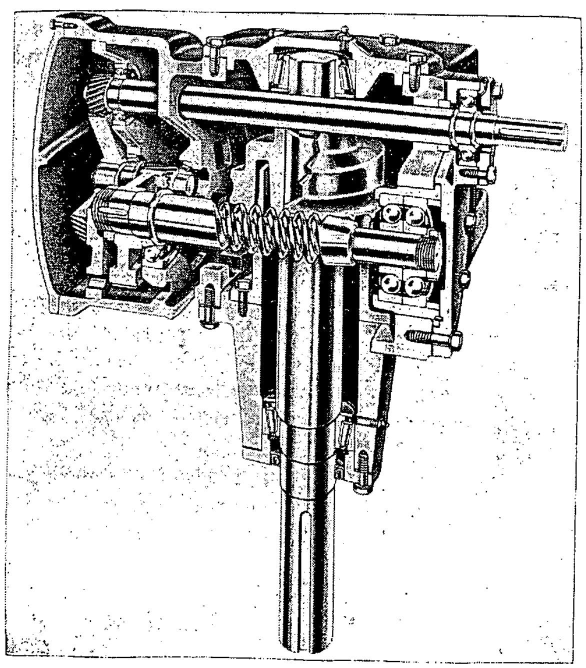


FIG. 3-12. Double-reduction unit with a worm second reduction. Typical of high-ratio units used to drive conveyors. (*Courtesy of Philadelphia Gear Corp.*)

The length-to-diameter ratio of helical and spur gears can be fairly large. In most other gear types the pinion will not have face width more than about one-third the pinion pitch diameter. In spur gearing the L/D ratio may be as great as 1 while in double-helical gearing it may approach 2. Needless to say, high accuracy is needed in boring the casings and machining the teeth when a relatively large face width is used. Low-hardness gearing of about 200 BHN will exhibit some tendency to wear in and improve contact. As much as 0.001″ may be worn off the high ends of the teeth to equalize the contact without too serious a result. Medium- to high-hardness gears have but slight tendency to wear in. If the contact across the face width is not good at full load, the teeth may break before any equalizing wear occurs. (See Chap. 13 for data on how to handle misalignment when load-rating gears.)

Large pinions with an L/D ratio of 1 or more will twist elastically under load an appreciable amount. It is necessary to provide helix modification to compensate for this on large gears where the material is fairly hard and the horsepower is high. (Helix modification was necessary on the gears shown in Figs. 3-4 and 3-7.)

Nested pinions tend to suffer from bowing and twisting of the long shaft between the two pinion halves. Helix compensation is often needed.

In multiple-power-path units, each power path must be able to have the teeth engage simultaneously when a slight torque is applied. In addition errors in tooth spacing and runout must be slight enough in the gears so that the flexibility of the connecting shafts can keep a uniform load on each branch. If the connections are too stiff or the teeth not precise enough, the load will shift rapidly from one power path to another as the gear revolves.

In multiple-power-path gearing, the gears will not assemble if there is a random orientation between the teeth of a gear and a pinion connected by the same shaft. Either the teeth and coupling means must be cut so that the orientation is exactly alike on each like assembly or adjustment must be provided so the proper tooth "timing" is achieved at assembly. Generally the teeth are cut at a random orientation. Then at assembly the parts are rotated until the pinion and gear by chance have their teeth in almost the right orientation. Then the final adjustment is made by some mechanical means like axial shimming (single-helix design), elongating coupling boltholes, redoweling, or using a stepped key. Incidentally, a shrewd choice on tooth numbers can make the timing problem work out much more easily.

3-4. Parallel-axis Epicyclic Gearing. A large family of gear arrangements goes under the general name of "epicyclic" gearing. Generally speaking the epicyclic train has a central "sun" gear, several "planets" meshing with the sun and spaced uniformly around the sun and an "annulus" or ring gear meshing with the planets. The sun and planets are externally toothed while the ring is internally toothed.

The name epicyclic comes from the fact that points on the planets trace out epicycloidal curves in space.

The name "planetary" gear is sometimes used interchangeably with epicyclic to denote the whole family of gears. In a more strict sense the *planetary* is one of the types of epicyclic gears.

There are a large number of possible epicyclic arrangements. These may be divided into three general groups:

1. Single epicyclic trains
2. Compound epicyclic trains
3. Coupled epicyclic trains

Single Epicyclic Trains. The simple epicyclic train consists only of the elements of sun, planet, and ring gear. This arrangement is often called a "simple" epicyclic gear or a "simple" planetary gear to distinguish it from the more complicated arrangements.

Figure 3-13 shows schematic arrangements of the three simple epicyclic gears. These are the *planetary*, the *star*, and the solar *arrangements*. The figure shows only three planets. The number of planets will vary quite considerably in actual practice. With some ratios it has been possible to squeeze in as many as twenty planets. As few as one is sometimes used for light load applications.

The derivation of ratios for planetary units is somewhat complicated. Several references[1,2,3,*] go into the derivation of ratios in considerable detail. Table 3-5 shows the over-all ratio for each of the three single epicyclic trains. If one of these units were driven backward, the ratio would be less than one and would simply be the reciprocal of the ratio as a speed-reducing set.

A range of ratios is shown for each kind of epicyclic gear. Note that only the solar will handle very low ratios. The ratio range of 1.7:1 to 2:1 is a range that is rather difficult to handle with simple epicyclic gearing. The star will work in this range, but it turns out to be rather awkward.

Epicyclic gears will not assemble unless the tooth numbers are properly chosen. Assuming the planets are equally spaced around the sun—which is the usual case—the equations given at the bottom of Table 3-5 must be met. The sum of the ring and sun teeth must be equally divisible by the number of planets. If there were three planets, the ring teeth plus sun teeth would have to divide evenly by three.

The tooth load at each planet is balanced between the sun-planet mesh and the planet-ring mesh. This tangential driving load may be calculated by

$$W_t = \frac{126{,}050\, P_H}{n_s d_s \text{ No. planets}} \tag{3-2}$$

where n_s = rpm of sun
d_s = pitch diameter of sun
P_H = horsepower transmitted

It should be appreciated that Eq. (3-2) determines the theoretical load only when there is perfect sharing of load between the planets. There is quite a problem in load sharing in epicyclic gears plus the possibility of appreciable dynamic overload due to tooth errors. More will be said on this subject later in this article.

Table 3-5. Simple-epicyclic-gear Data

Kind of arrangement	Fixed member	Input member	Output member	Over-all ratio m_G	Range of ratios normally used
Planetary	Ring	Sun	Cage	$\frac{N_R}{N_S} + 1$	3:1–12:1
Star	Cage	Sun	Ring	$\frac{N_R}{N_S}$	2:1–11:1
Solar	Sun	Ring	Cage	$\frac{N_S}{N_R} + 1$	1.2:1–1.7:1

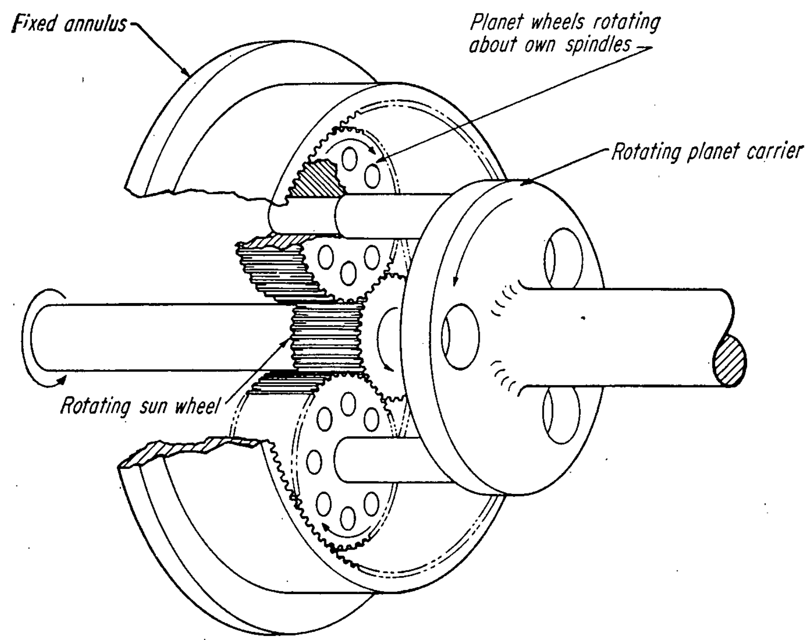
N_S = number of sun teeth
N_P = number of planet teeth
N_R = number of ring (annulus) teeth

To assemble,

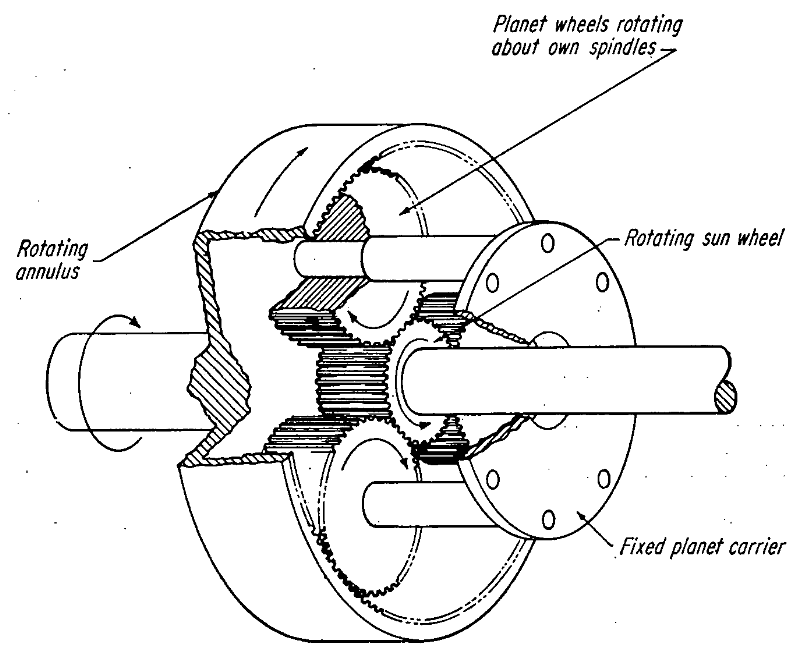
$$N_R = N_S + 2N_P \tag{3-3}$$

$$\frac{N_R + N_S}{\text{No. planets}} \text{ must equal an integer} \tag{3-4}$$

* Superscript numbers refer to references at the end of the chapter.



(a)



(b)

Fig. 3-13. Simple epicyclic gears. (a) Planetary

GE-1024.043

Compound Epicyclic Trains. The "compound" epicyclic train has two planet members attached together on a common shaft. Figure 3-15 shows the schematic arrangement of a compound star gear. Note the large planets (or stars) which mesh only with the sun and the small planets which mesh only with the ring gear.

Table 3-6. Compound-epicyclic-gear Data

Kind of arrangement	Fixed member	Input member	Output member	Over-all ratio m_G	Range of ratios normally used
Compound planetary....	Ring	Sun	Cage	$\frac{N_R N_{P_1}}{N_S N_{P_2}} + 1$	6:1–25:1
Compound star.........	Cage	Sun	Ring	$\frac{N_{P_1} N_R}{N_S N_{P_2}}$	5:1–24:1
Compound solar........	Sun	Ring	Cage	$\frac{N_S N_{P_2}}{N_R N_{P_1}} + 1$	1.05:1–2.2:1

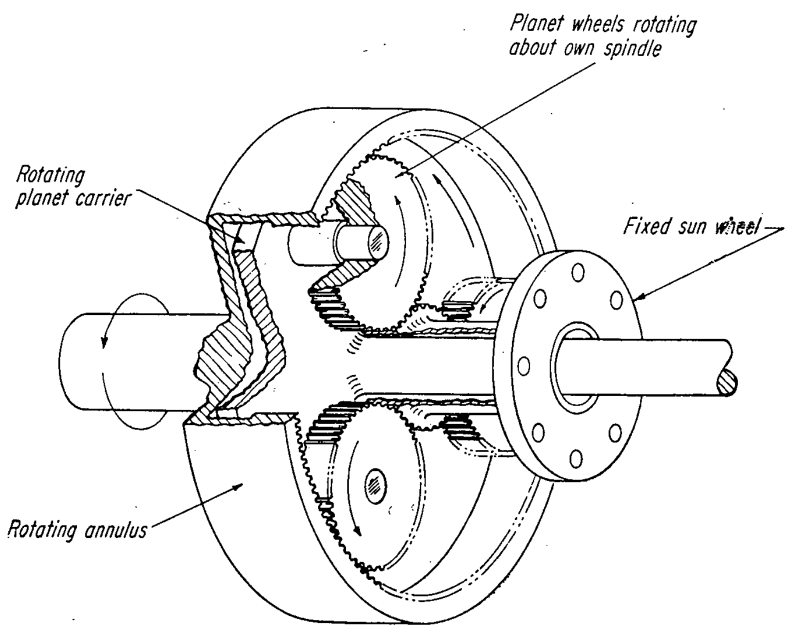
N_S = number of sun teeth
N_{P_1} = number of first-reduction planet teeth
N_{P_2} = number of second-reduction planet teeth
N_R = number of ring teeth

To assemble,

$$D_R = D_S + D_{P_1} + D_{P_2} \tag{3-5}$$

where D_R = ring pitch diameter
D_P = planet pitch diameter

$$\frac{N_{P_1}N_R + N_{P_2}N_S}{\text{No. planets}} \text{ must equal an integer} \tag{3-6}$$



(c) Solar gear

(c)

gear. (b) Star gear. (c) Solar gear.

Table 3-6 shows the basic data for compound epicyclic gears. Note that the compound planetary and the compound star will handle much higher ratios than the simple epicyclic types. The compound solar is still limited to a narrow ratio range.

Table 3-6 shows the rules for assembly of the compound epicyclic. Equal spacing of planet shaft assemblies about the sun is assumed.

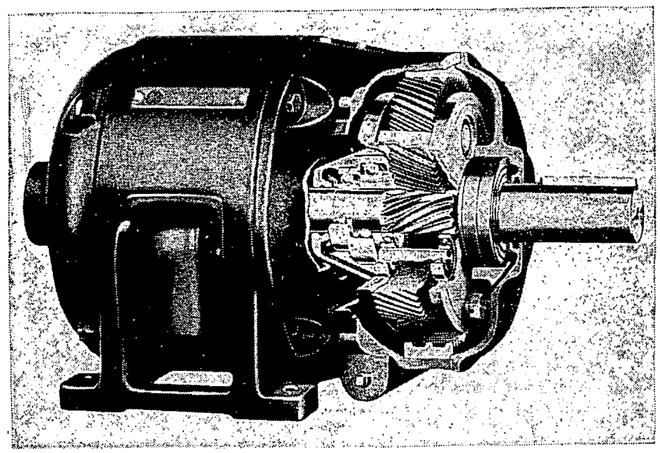


FIG. 3-14. Planetary gear as part of a geared motor. (*Courtesy of General Electric Co., Paterson, N. J.*)

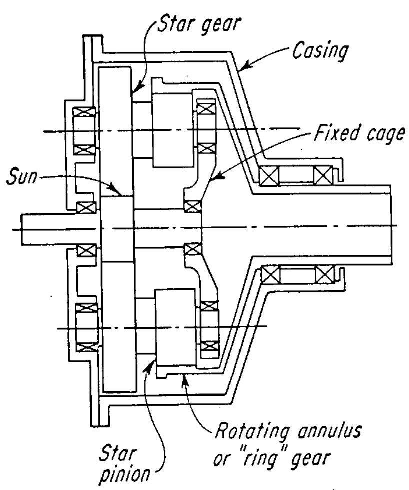


FIG. 3-15. Schematic arrangement of a compound star gear.

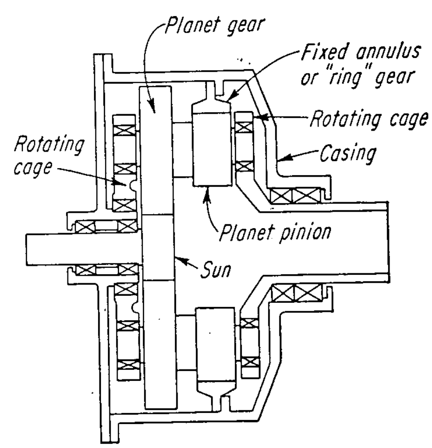


FIG. 3-16. Compound planetary.

Equation (3-2) may be used to get the theoretical driving load at the first-reduction planets. The load at the second-reduction planets is

$$W_{t_2} = \frac{W_{t_1} D_{P_1}}{D_{P_2}} \tag{3-7}$$

In the case of the solar unit Eq. (3-2) will not work since the sun is not rotating. By changing the equation to use the speed of the ring and the diameter of the ring, the driving load can still be obtained. For the compound solar, the load determined from the ring should be considered as W_{t_2}. Equation (3-7) can be worked backward to solve for W_{t_1}, the load at the sun-planet mesh.

Coupled Epicyclic Trains. Coupled epicyclic trains consist of two or more single epicyclic trains arranged so that two members in one train are common to the adjacent train. Figure 3-17 shows some schematic arrangements of coupled epicyclic trains. There are many more possible combinations.

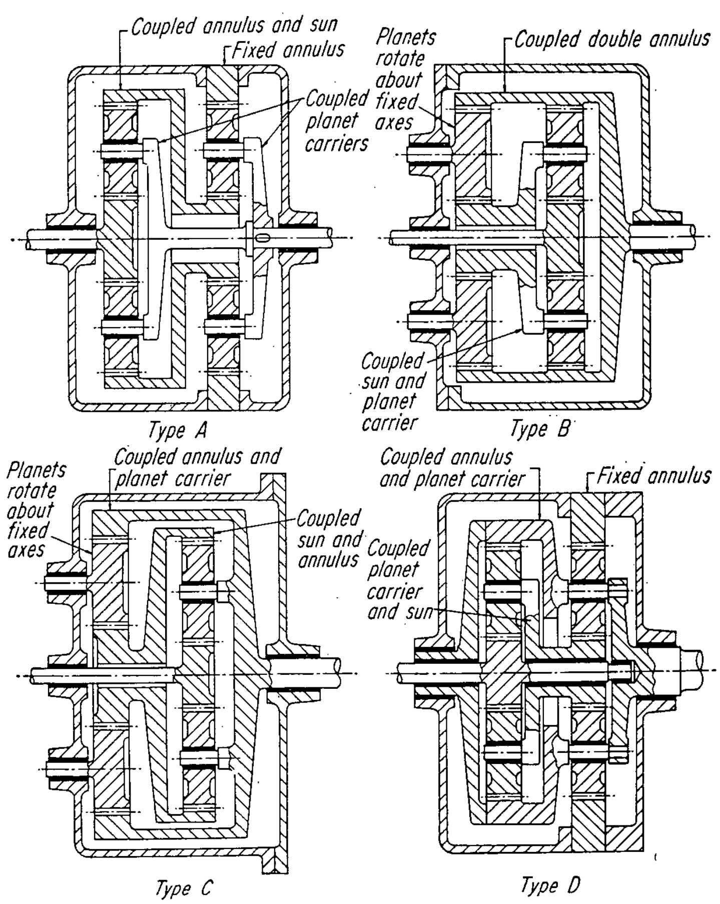


Fig. 3-17. Some schematic arrangements of coupled epicyclic gears.

The calculation of ratios, tooth loads, and efficiencies becomes quite complicated for the coupled epicyclic train. Since these gears are not widely used, detail information will not be given here. The reader is referred to refs. 2 and 4.

Application of Epicyclic Gear Trains. Epicyclic gears have been used for many centuries in the gear trade. James Watt in 1781 patented a sun and planet gear which he used with one of his early engines. There have been many patents on epicyclic gears. Even in the recent period since the Second World War, several refinements in planetary-gear design details have been patented.

The epicyclic gear is widely used in the aircraft industry. Almost all propeller-driven commercial aircraft use an epicyclic gear of some type to drive the propeller. The piston engines do not need too much ratio, and this permits them to use only a single-reduction planetary. The higher-speed gas-turbine engines have used double-reduction gearsets with the last reduction being epicyclic, or they have used compound epicyclic gears.

There has been a wide use of epicyclic gearing in geared motors and other industrial applications where the horsepower was fairly low (usually under 100 hp).

The epicyclic gear requires great precision and very clever use of flexibility to make all the planets work evenly. The best of modern aircraft planetary units probably divide up the load so uniformly that no planet ever sees more than about 125 per cent of the theoretical load. Many early aircraft planetary units (1935–1945 era) appeared to have a nonuniform split of loading that permitted as much as 200 per cent load to be applied to the most heavily loaded planet.

Epicyclic gears have not been widely used in the high-horsepower high-speed industrial or marine field. There is a trend to more use of these units in this field. If modern aircraft-quality units can be provided at industrial-gear prices, then there may be a considerable increase in the use of epicyclic gears to handle thousands of horsepower in land and marine applications.

Epicyclic gears are widely used in tank transmissions, truck transmissions, and automatic-shift transmissions for passenger cars In these cases the basic epicyclic gear is generally combined with other gears, clutches, brakes, and pumps to make an over-all transmission package. Article 3-9 gives some examples of such transmissions. The horsepower handled by these transmissions is generally under 500.

Problems of Epicyclic Gears. Several of the problems in designing and using epicyclic gears have already been mentioned. A summary of the problems is as follows:

1. Dividing load between planets
2. High bearing loads on planet pins
3. Balance and vibration of rotating cage
4. More complicated assembly
5. Hazard of jamming the whole drive if a piece of metal gets loose in the mesh

3-5. Right-angle-gear Arrangements. Right-angle[11] gears are used in a wide variety of meshing arrangements. Table 3-7 shows a general survey of the field.

In theory all the right-angle gear types could be used on either right angles or angles other than 90°. In practice, though, because of limitations in manufacturing

Table 3-7. Angular-gear Meshing Possibilities

Type of gear teeth	Pinion and gear		Arrangements		
	Right angle	Skew angle	Planetary	Differential	Reversing
Straight bevel	Yes	Yes	Yes	Yes	Yes
Zero bevel	Yes	Yes	Yes	Yes	Yes
Spiral bevel	Yes	Yes	Yes	Yes	Yes
Hypoid	Yes	...	No	Yes	Yes
Face gear	Yes	No	...	Yes	Yes
Crossed-helical	Yes	Yes	No	...	No
Cylindrical worm	Yes	...	No	Yes	No
Double-enveloping worm	Yes	...	No	Yes	No
Beveloid	Yes	Yes	Yes	Yes	Yes
Spiroid	Yes	...	No	Yes	No
Planoid	Yes	...	No	...	Yes
Helicon	Yes	...	No	Yes	No

equipment and other considerations, some of the gears are not ordinarily used on skew angles. Note the dash lines in the skew-angle column of Table 3-7.

Some types can be used in a planetary-style arrangement. When this is done the ring member becomes externally toothed. The data given in Table 3-5 can be used to calculate the ratio of a planetary made with angular gears.

Some angular gears are used in differentials. Even the worm gear may be used as the gear drive to rotate a differential package.

Some right-angle gears lend themselves to reversing drives. Two pinions may be shifted axially to make a quick reverse in a drive.

Figure 3-18 shows a typical bevel-gear speed reducer. Typical applications of a wormgear drive and a Spiroid-gear drive are shown in Figs. 3-19 and 3-20.

One of the more important skew-angle drives is the V drive for boats. This permits the engine to be horizontal and the propeller shaft to pass through the hull of the ship at an angle. Figure 3-21 shows a V-drive unit.

Figure 3-22 shows the elements of a bevel reverse unit. The bevel planets are attached to a fixed cage. Since the bevel *sun* and the bevel *ring* are really the same size parts, this *star* type of arrangement has a 1:1 ratio. It is used only to reverse direction of rotation.

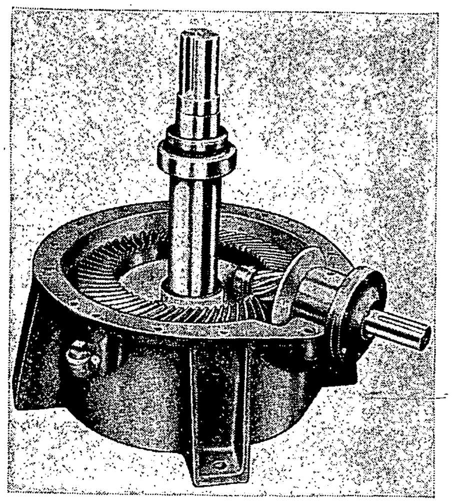


FIG. 3-18. Bevel-gear speed reducer. (*Courtesy of Philadelphia Gear Corp.*)

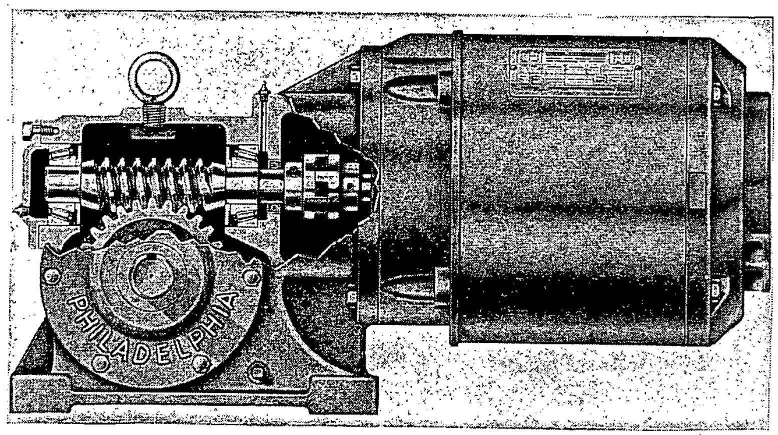


FIG. 3-19. Motor with wormgear speed reducer. (*Courtesy of Philadelphia Gear Corp.*)

Figure 3-23 shows the parts that could be used to make a 2:1 ratio bevel planetary. In this case the cage would carry the planets and drive the output shaft through one of the hollow bevel gears. One bevel gear would function as an input sun and one would function as a fixed ring.

Bevel planetaries can be made to handle a range of ratios. To do this the input sun is made smaller than the bevel functioning as the ring. The axes of the planets are not at right angles to the input and output shafts in this case.

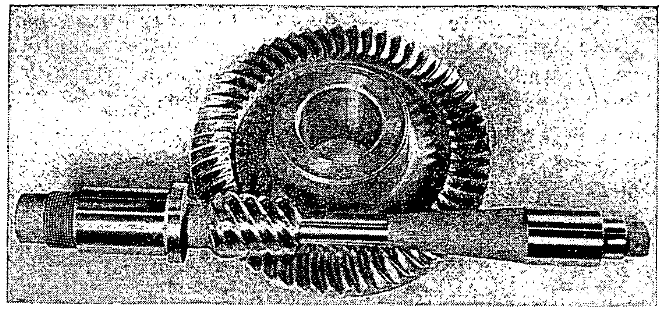


FIG. 3-20. Spiroid-gear drive for a band saw. (*Courtesy of Spiroid Division of Illinois Tool Works, Chicago, Ill.*)

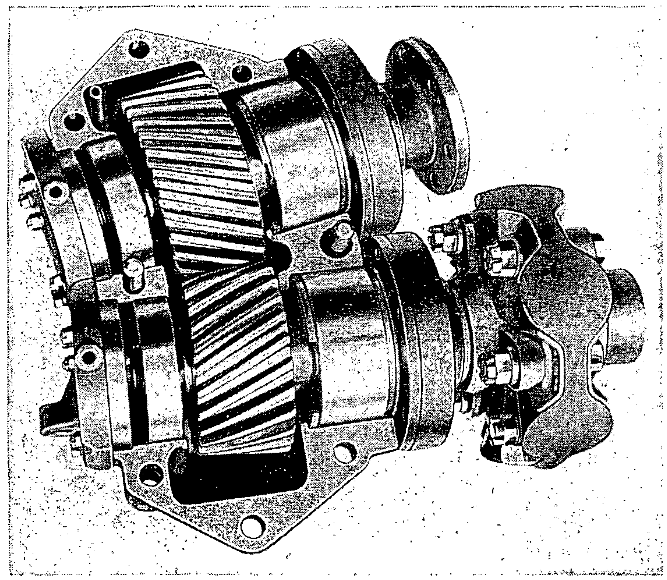


FIG. 3-21. V-drive unit. (*Courtesy of Gleason Works, Rochester, N.Y.*)

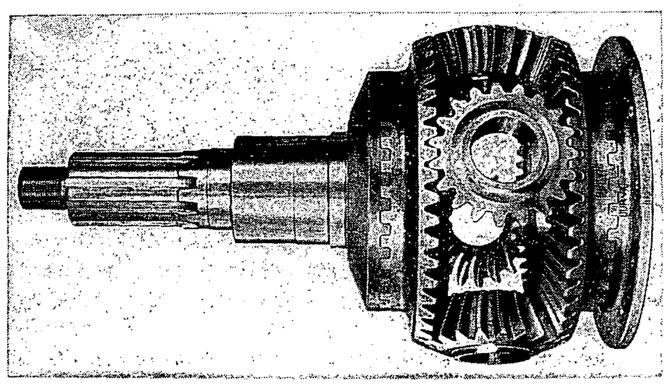


FIG. 3-22. Bevel reverse unit. (*Courtesy of Gleason Works, Rochester, N.Y.*)

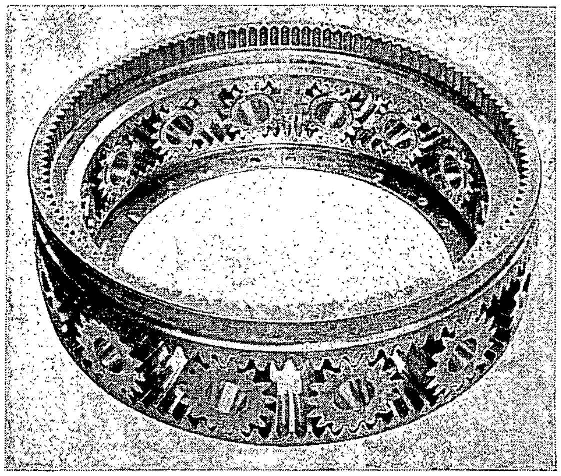


FIG. 3-23. Parts for 2:1 ratio bevel planetary. (*Courtesy of Gleason Works, Rochester, N.Y.*)

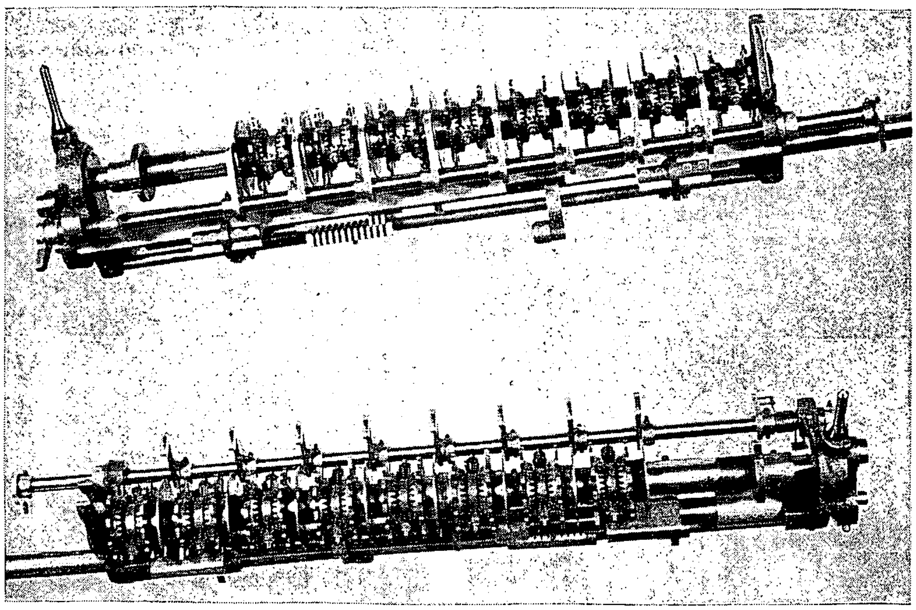


FIG. 3-24. Bevel-gear parts used in a cash-register machine. (*Courtesy of Gleason Works and National Cash Register.*)

Figure 3-24 shows an array of bevel gears used in a cash-register machine. Note how this compact arrangement provides a multiplicity of input shafts and a ready means for reversing any input.

3-6. Differential Gearing. In a general sense a "differential" gear is an arrangement where the normal ratio of the unit can be changed by driving into the unit with a second drive. Figure 3-25 shows a simple bevel-gear differential of the type often used in instruments. Input rotations may be applied to the gears at either end of the bevel planetary set. The output speed is taken off the "spider" or cage shaft. Either input speed may be stopped and the device will still transmit a ratio between the other input and the output. By varying the speed of the second input it is possible to get an infinite series of ratios.

Figure 3-26 shows the elements of a spur-gear differential. Again there are two inputs and one output, or it is possible to have one input and two outputs.

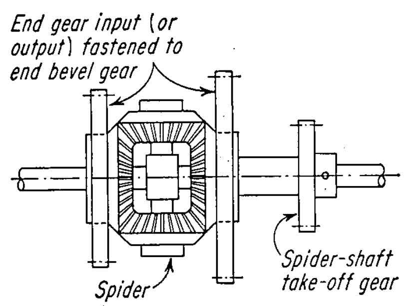


Fig. 3-25. Simple bevel-gear differential.

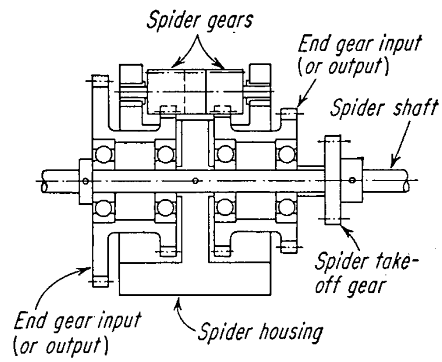


Fig. 3-26. Spur-gear differential.

Figure 3-27 shows a ground spiral bevel differential. One of the most familiar uses of a differential is in the rear end of an automobile. When an automobile goes around a corner, one wheel must make more turns than the other. A differential is needed with one input and two outputs. Figure 3-28 shows a photograph of a typical hypoid differential for an automobile. The large gears are hypoid and the small gears are bevel. The hypoid type of car rear end allows the drive shaft to be lower

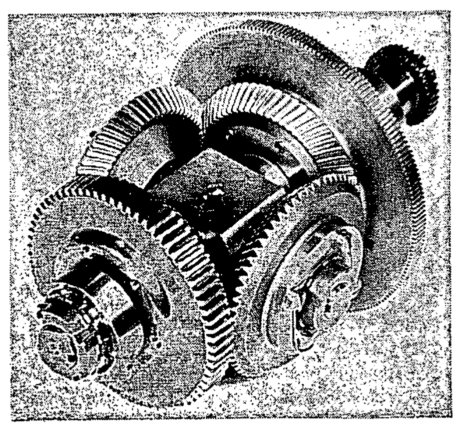


Fig. 3-27. Ground spiral bevel differential. (*Courtesy of Gleason Works, Rochester, N.Y.*)

Fig. 3-28. Hypoid differential used in rear end of automobile. (*Courtesy of Gleason Works, Rochester, N.Y.*)

than would be the case with an all-bevel differential. A low drive shaft permits the automobile to be built lower.

Fixed Differential. The differentials just discussed with two inputs and one output or two outputs and one input are used to vary ratio. There is another kind of differential that has a fixed ratio and develops an unusually large ratio. This kind is called a differential because the output speed is the *difference* between the speeds of two parts of the drive that are running at almost the same speed.

GE-1024.051

There are dozens of arrangements of the (free) differential discussed earlier. Likewise there are dozens of fixed-differential arrangements. Figure 3-29 shows four typical arrangements of fixed differentials.

Table 3-8 shows some general data for the four fixed differentials shown in Fig. 3-29. Note that there is a subtractive item in the expression for over-all ratio for each of

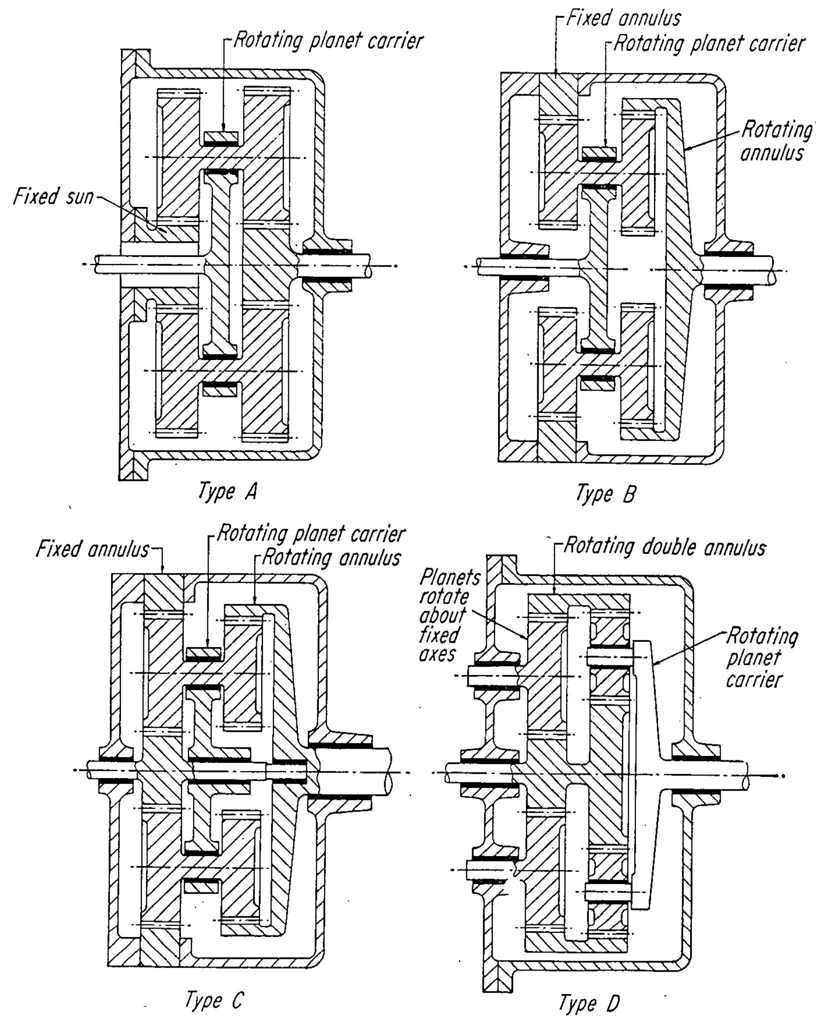


FIG. 3-29. Fixed-differential gear arrangements.

these types. If the pitch was made infinitely fine, it would be possible to get the subtractive term to almost reach zero. This would tend to make the over-all ratio infinitely large. Practical considerations on number of teeth, though, tend to limit the ratio that can be obtained. The ratio range shown represents normal design practice rather than any theoretical limit on the ratio.

The fixed-differential gears tend to be low in efficiency. The differential has a large amount of tooth meshing going on at very high tooth loads. Losses are much higher than for regular gearing. A fixed differential of 100:1 ratio might have an efficiency as low as 25 per cent. With good design a fairly good efficiency can be obtained.

Table 3-8. Fixed-differential-gear Data

Kind of differential*	Fixed member	Input member	Output member	Arrangement	Over-all ratio	Range of ratios normally used
A. All external teeth	No. 1 sun	Cage	No. 2 sun	No. 1 sun/No. 1 planet No. 2 planet/No. 2 sun	$\frac{N_{S_2}N_{P_1}}{N_{S_2}N_{P_1} - N_{S_1}N_{P_2}}$	10:1–50:1
B. Internal gear	No. 1 ring	Cage	No. 2 ring	No. 1 planet/No. 1 ring No. 2 planet/No. 2 ring	$\frac{N_{R_2}N_{P_1}}{N_{R_2}N_{P_1} - N_{R_1}N_{P_2}}$	15:1–100:1
C. Planetary	No. 1 ring	Sun	No. 2 ring	No. 1 sun/No. 1 planet No. 1 planet/No. 1 ring No. 2 planet/No. 2 ring	$\left(\frac{N_{R_1}}{N_{S_1}} + 1\right)\frac{N_{R_2}N_{P_1}}{N_{R_2}N_{P_1} - N_{R_1}N_{P_2}}$	20:1–500:1
D. Star-planetary	No. 1 cage	No. 1 sun	No. 2 ring	No. 1 sun/No. 1 planet No. 1 planet/No. 1 ring No. 2 sun/No. 2 planet No. 2 planet/No. 2 ring No. 1 sun attached to No. 2 sun No. 1 ring attached to No. 2 ring	$\frac{1}{\frac{1}{\left(\frac{N_{R_2}}{N_{S_2}} + 1\right)} - \frac{1}{\frac{N_{R_1}}{N_{S_1}}\left(\frac{N_{S_2}}{N_{R_2}} + 1\right)}}$	15:1–100:1

Follow rules of assembly for compound epicyclic for applicable parts of each differential (see Table 3-6).
* See Fig. 3-29 for schematic arrangement of each kind.

The chief value of the differential is that it permits a large ratio to be obtained in a very compact unit. References 7 and 8 give detail information on how to calculate the efficiency of fixed differentials.

3-7. Contra-rotating Concentric-shaft Drives. There is sometimes a need to have concentric output shafts rotating in opposite directions. Some torpedoes have concentric propellers rotating in opposite directions. A few airplanes have had concentric propellers rotating in opposite directions. Helicopter rotors have also been used in a contra-rotating concentric-shaft arrangement.

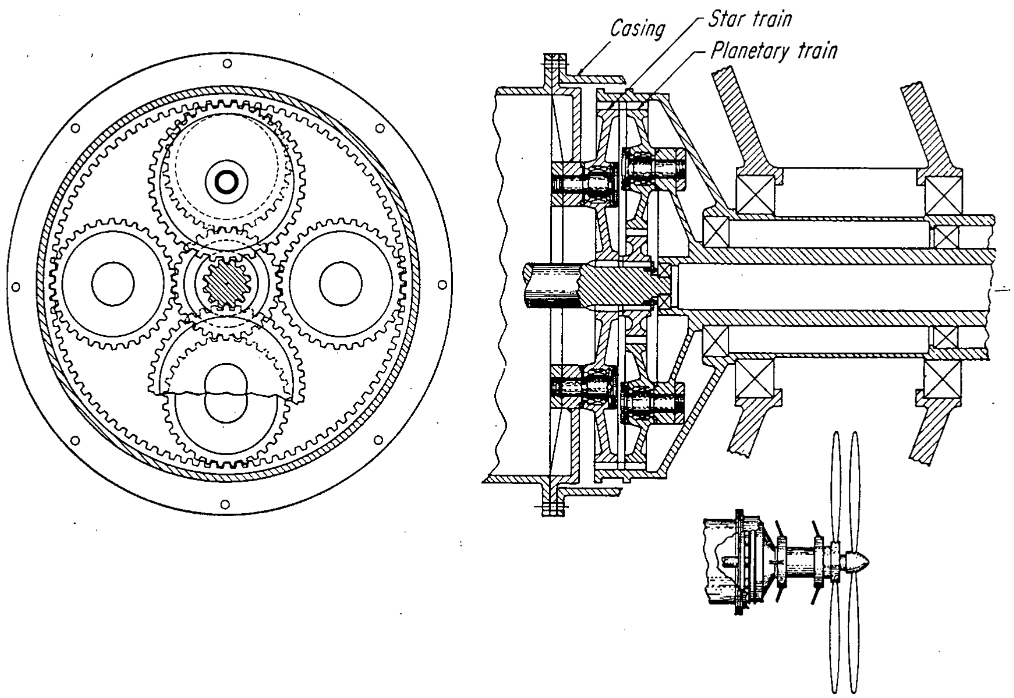


FIG. 3-30. Contra-rotating gear arrangement.

Several kinds of gear drives will accomplish this result. Three general approaches to the problem are

1. Use two final drive pinions on the same shaft. Drive an external gear with one and drive an internal gear with the other. Adjust pinion and gear sizes so that each set has the same ratio.
2. Use a bevel reverse gear after the last stage of the regular reduction gearing. Make the bevel gears hollow and connect the input bevel to the inner output shaft. Connect the output bevel to the outer shaft.
3. Use a combination star- and planetary-type arrangement. Connect one output shaft to the cage. Connect the other to the ring gear. Attach the star ring and the planetary ring together and permit this assembly to be rotatable.

The arrangement shown in Fig. 3-30 illustrates the star-planetary arrangement. In this arrangement most of the power goes through the planetary side of the drive. Only enough to balance the output torques goes through the star side of the drive.

3-8. Speed-changing Transmissions by Shifting. Gear arrangements that change speed by shifting have been widely used in automobiles, trucks, tanks, and

GE-1024.054

many other applications. The gear-shift transmission may have a variety of parallel-axis gears, right-angle-axis gears, and clutch elements.

The normal mode of operation for these transmissions is to declutch the drive, shift a gear or coupling axially, and reengage the clutch.

The general way these transmissions are made can best be understood by looking at cross-section views or cutaway views of typical designs.

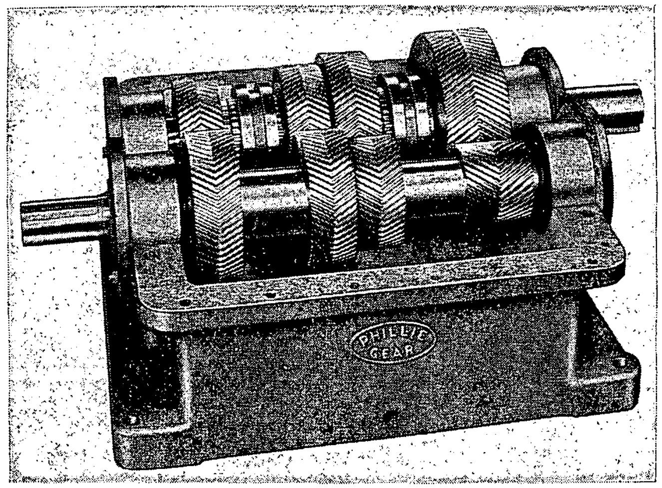


FIG. 3-31. Industrial gear-shift transmission. (*Courtesy of Philadelphia Gear Works.*)

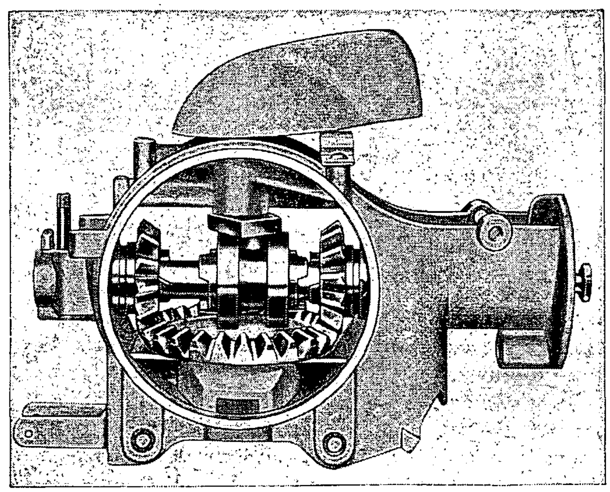


FIG. 3-32. Simple reverse gear shift. (*Courtesy of Gleason Works, Rochester, N.Y.*)

Figure 3-31 shows an industrial-type gear-shift transmission. Note the slidable coupling members and the herringbone gear teeth.

Figure 3-32 shows a simple reversing unit using bevel gears. Note the slidable dog clutch.

Figure 3-33 shows a five-speed transmission of the type used in automobiles and trucks. Note the compact arrangement. Note how the upper right-hand gear is slidable on its shaft.

GE-1024.055

Figure 3-34 shows a 12-speed transmission of the type used in a heavy-duty truck. This is a very compact arrangement with a minimum of parts for a 12-speed transmission.

Figure 3-35 shows an auxiliary transmission. When a vehicle needs more speeds than can be provided by the normal transmission, an auxiliary transmission can be installed in series with the main transmission to give several more speeds.

3-9. Speed-changing Transmissions by Clutches and Brakes. Many transmissions are made so that a gear shift can occur without disengaging the drive (or declutching). These transmissions use things like fluid couplings, brakes, or differentials to change speed while torque is being transmitted. The so-called "automatic"

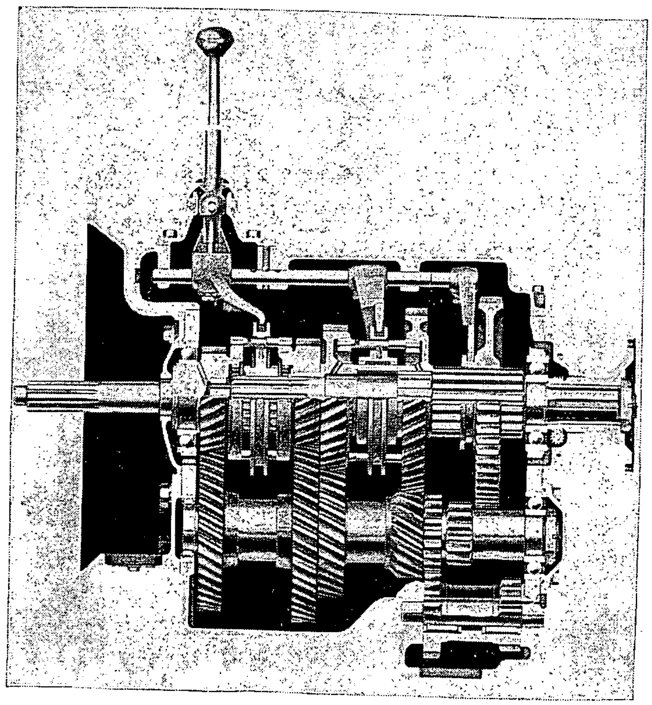


FIG. 3-33. Five-speed transmission. (*Courtesy of Dana Corporation, Toledo, Ohio.*)

transmission used on most passenger automobiles is of this type. Some tanks and other military vehicles use the automatic transmission.

Figure 3-36 shows a fully automatic transmission for on-highway truck use. Note the fluid-coupling parts. Figure 3-37 shows a transmission for off-highway vehicles like a front-end loader. Note the fluid coupling and the several sets of brake or clutch disks.

Figure 3-38 shows a cross-drive transmission for a tracked vehicle like a tank. This transmission has two speeds forward and one reverse. The transmission has the capability of controlling the speed ratio to each drive shaft. A tank is steered by driving one track faster than the other.

The one transmission shown in Fig. 3-38 has almost all the gear arrangements discussed in this chapter in one compact package. There are planetary-type gears, bevel gears, helical gears, skew-axis wormgears, differentials, simple spur gears, and multiple-power paths all in this one transmission. There are 4,007 parts in the unit, the weight is approximately 3,000 lb, and the unit will handle up to 850 gross hp.

GE-1024.056

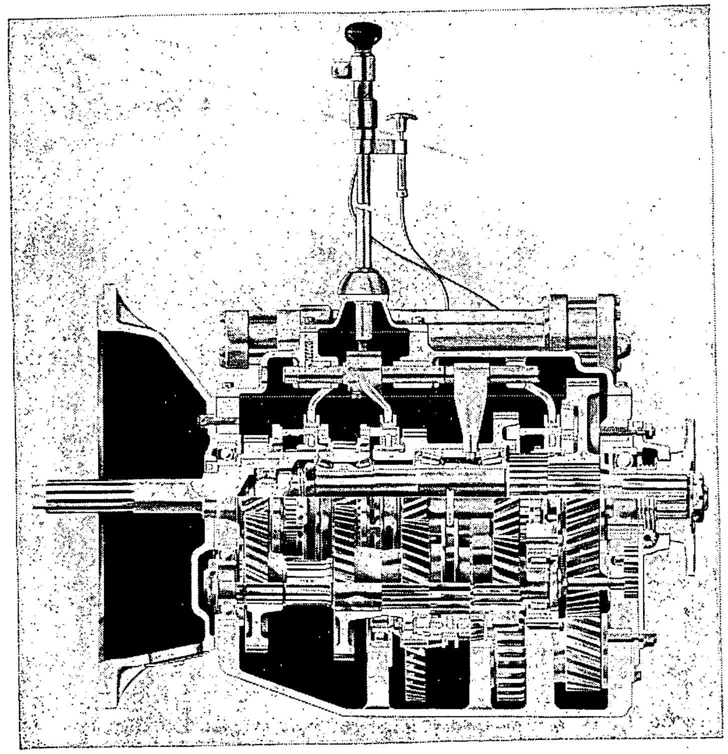


Fig. 3-34. Twelve-speed transmission. (*Courtesy of Dana Corporation, Toledo, Ohio.*)

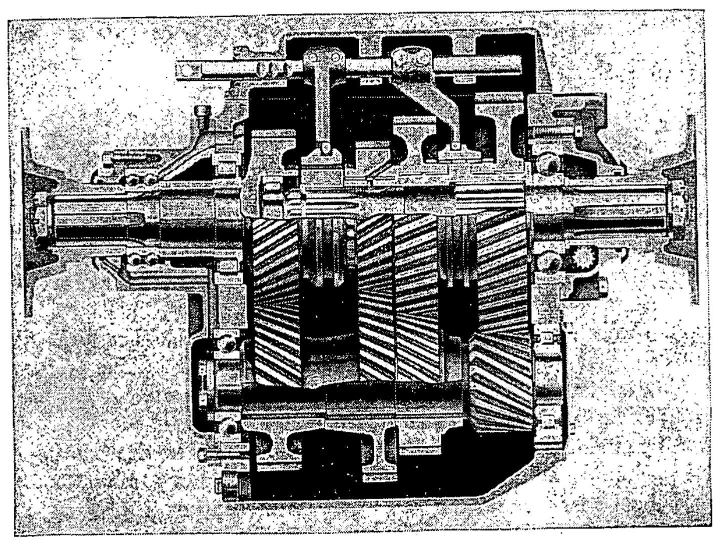


Fig. 3-35. Auxiliary transmission. (*Courtesy of Dana Corporation, Toledo, Ohio.*)

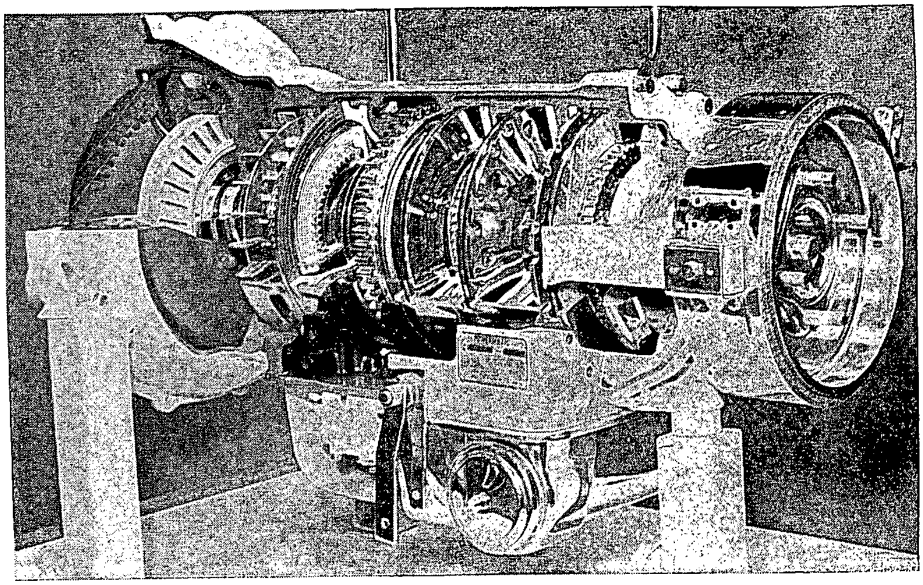


FIG. 3-36. Automatic truck transmission. (*Courtesy of Allison Division, General Motors Corp., Indianapolis, Ind.*)

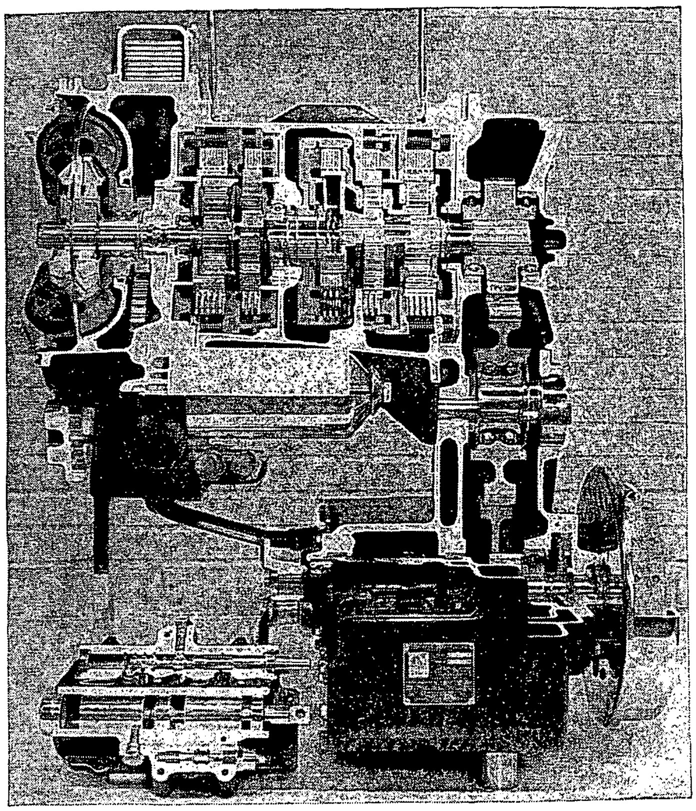


FIG. 3-37. Transmission for material-handling vehicle. (*Courtesy of Allison Division, General Motors Corp.*)

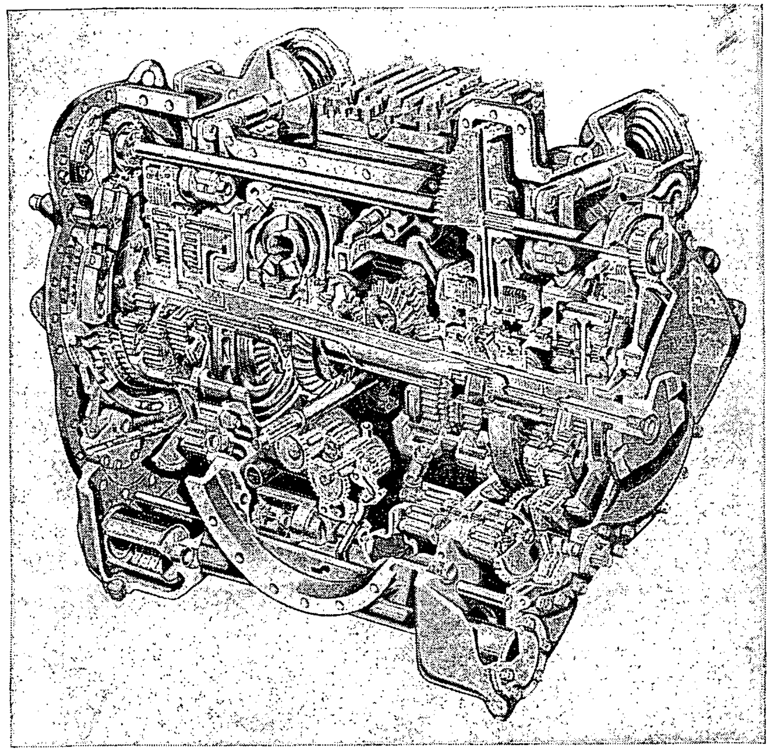


FIG. 3-38. Allison CD-850 cross-drive transmission for military track-laying vehicles. (*Courtesy of Allison Division, General Motors Corp.*)

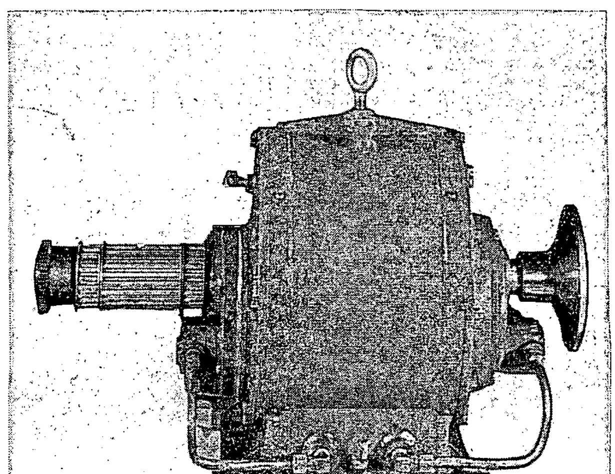


FIG. 3-39. Marine drive unit rated at 825 hp, 6,840 input speed, 3.22:1 forward speed, and 2.84:1 reverse speed. (*Courtesy of Jered Industries, Inc., Detroit, Mich.*)

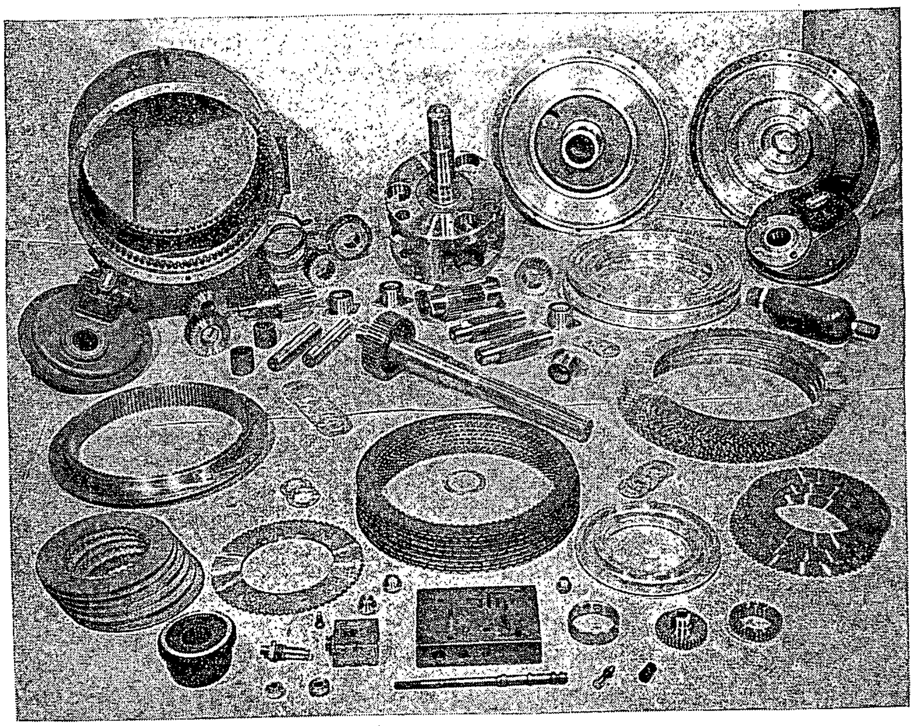


Fig. 3-40. Disassembled parts for unit shown in Fig. 3-39. (*Courtesy of Jered Industries, Inc.*)

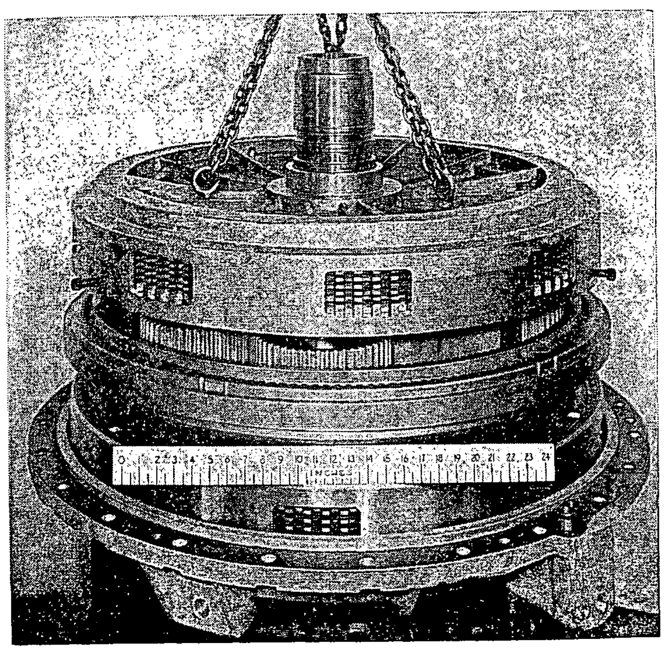


Fig. 3-41. Partially disassembled view of marine drive unit rated at 2,500 hp. (*Courtesy of Jered Industries, Inc.*)

Figure 3-39 shows a marine unit with a forward and reverse speed. This unit weighs about 520 lb and handles up to 825 hp. Input speed is 6,840 rpm. It has a 3.22:1 forward ratio and a 2.84:1 reverse ratio. The gearing and brakes in the unit make it work as either a planetary unit or a star unit. A planetary unit has its output turning in the same direction as the input. A star unit has an output that turns in the reverse direction from the input. By switching from the star to the planetary, or vice versa, it is possible to change direction of rotation.

Figure 3-41 shows a partially disassembled view of a 2,500-hp marine reverse unit. This unit handles an input speed of 1,000 rpm with a 1:1 forward ratio and a 1.2:1 reverse ratio. The unit weighs a little over 4,000 lb. Figure 3-42 shows the sun, planet, and cage parts.

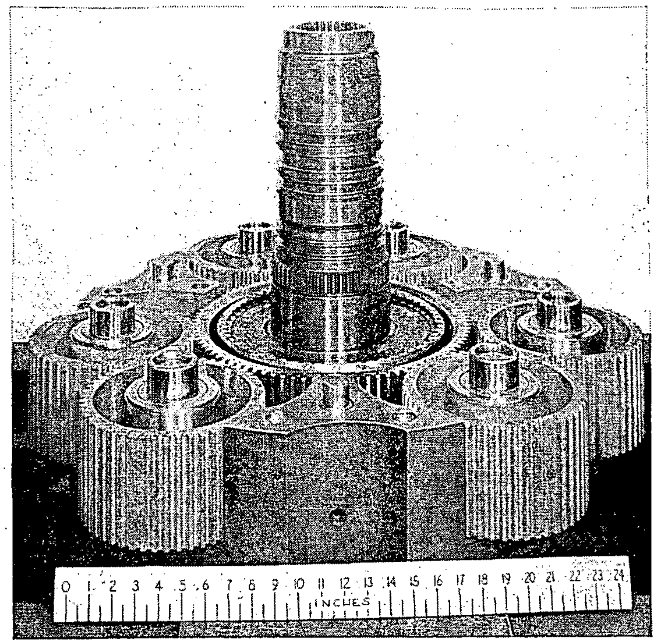


FIG. 3-42. Sun, planets, and cage structure for unit shown in Fig. 3-41. (*Courtesy of Jered Industries, Inc.*)

The unit shown in Fig. 3-41 has some novel features. In forward drive the cage is held to the sun by a clutch. The whole gear assembly rotates as a unit and there is no tooth meshing. In reverse the first clutch is opened and a second clutch holds the ring gear to the casing. The unit then functions as a planetary gear, but the fact that there are two planets meshing with each other makes the output shaft turn opposite from the input and the ratio is much lower than in a normal planetary.

3-10. High-ratio Arrangements. Besides the fixed-differential gears discussed in Art. 3-6, there are several other gear arrangements that have high-ratio capabilities. These are used under varying trade names. Theoretically they do not belong in any of the gear classifications already discussed in this chapter.

Planocentric Gear. An example of one of the high-ratio types is the kind of gearset called the "planocentric." This unit can have as few gear parts as a spur pinion and an internal gear and yet develop as much as 100:1 ratio! A ratio this great can be readily obtained with a single-thread worm and gear, but it is surprising that this much can be achieved with spur gearing or helical gearing.

Figure 3-43 shows an over-all view of a unit of this type. Figure 3-44 shows the details of the meshing elements.

The pinion is mounted on an eccentric. A bearing separates the pinion from the eccentric and its drive shaft. The pinion is thus under no constraint to rotate at the speed of the input shaft.

The pinion wobbles instead of truly rotating. It does develop a rotation, superimposed on the wobble. Strangely, the rotation of the pinion is at output speed

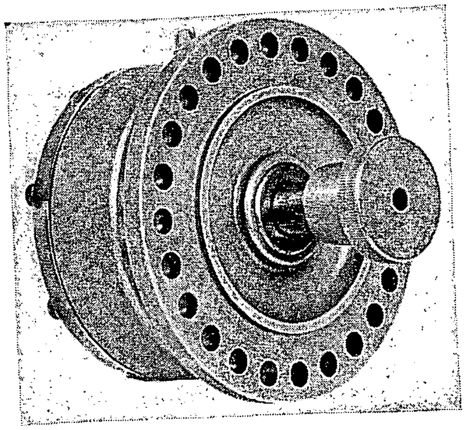


Fig. 3-43. Planocentric gear, 64:1 ratio, oblique low-speed end. (*Courtesy of General Electric Co., Lynn, Mass.*)

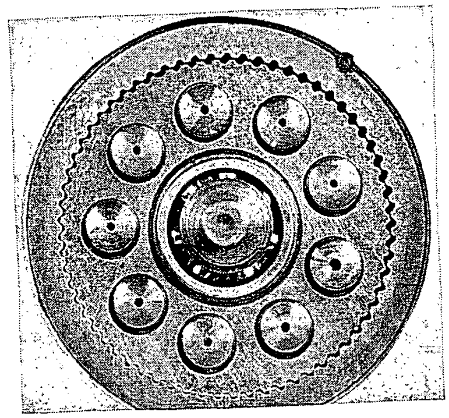


Fig. 3-44. Planocentric gear, 64:1 ratio. High-speed end showing 64-tooth pinion mating with 65-tooth gear. Pinion drives cage pins. (*Courtesy of General Electric Co., Lynn, Mass.*)

instead of input speed! The pinion transmits its rotation to the output shaft by a pin coupling (or other coupling means). The internal gear is fixed to the casing.

The equation for ratio is

$$m_G = \frac{N_P}{N_G - N_P} \qquad (3\text{-}8)$$

This type of drive has been used for high-torque high-reduction applications in aircraft, marine, and appliance fields. Under favorable conditions the efficiency may be as high as 90 per cent or more.

3-11. Harmonic Drive Arrangements. Harmonic Drive is the name given to a new family of machine systems which use the controlled elastic deflection of one or more parts for the transmission, conversion, or change of mechanical motion. The basic mechanism of Harmonic Drive has the broad multipurpose capabilities of the simple lever and has proved itself adaptable to such diverse forms of mechanical systems as rotary-to-rotary motion transmissions, rotary-to-linear motion converters, linear-to-linear transmissions, and rotary pumps and valves.

Until now, nearly all mechanical systems have been based upon the well-known laws of "rigid-body mechanics," and considerable effort has been devoted to reducing the spring flexure of machine parts to an absolute minimum. Rotating elements have been assumed to remain rigid and to rotate circularly about fixed axes. The following study of Harmonic Drive describes a radical departure from traditional mechanics and requires entry into a new and challenging realm of "nonrigid mechanics," or "elastokinesis"—the new field of elastic-body dynamics.

A close look at the functioning of Harmonic Drive elements is provided in Fig. 3-45. Here the wave generator is elliptoidal in shape and surrounded by a ball bearing.

The Flexspline has 130 external teeth and the circular spline 132 internal teeth, resulting in a 2-tooth difference equal to the number of lobes of the wave generator.

Harmonic Drive is the subject matter of U.S. Patent 2,906,143 and of other United States and foreign patents and patent applications. All proprietary rights are held by the United Shoe Machinery Corporation.

As the wave generator is turned, the Flexspline is progressively deflected to follow the rotating elliptoidal shape. Flexspline and circular spline are held in engagement

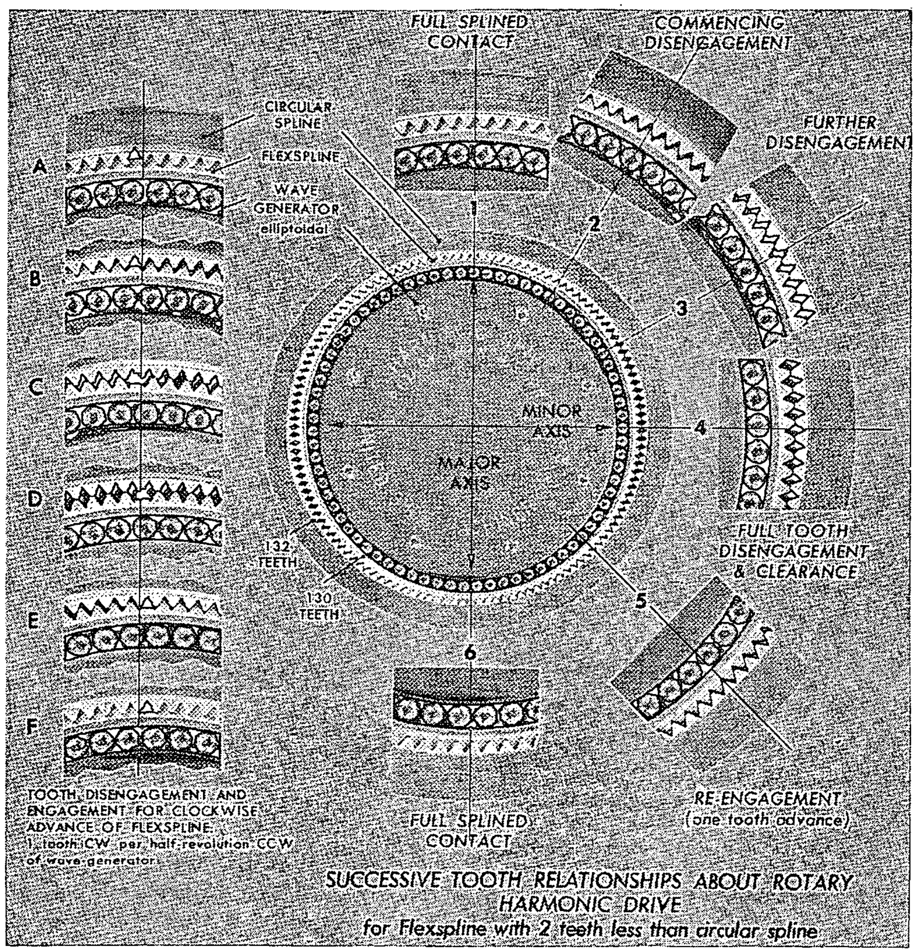


FIG. 3-45. Spline-motion relationships of rotary Harmonic Drive.

at the major axis of the wave generator and are fully disengaged and clearing at the minor axis. At the major axis the teeth are in full spline contact and are rotationally stationary. At the minor axis where the teeth are disengaged, they are in angular motion because of deflection of the Flexspline. Enlarged sections (1 through 6) show successive conditions of spline-tooth engagement and disengagement. At the left in Fig. 3-45, the spline relationship at any one region of the teeth is shown as the wave generator is rotated. It can be seen here how a particular tooth gradually disengages, advances, and reengages, as the wave generator is turned a half revolution. For a full rotation of the wave generator, the Flexspline counterrotates through an angle equivalent to 2 of its 130 teeth, resulting in a reduction ratio of 65:1 magnitude.

Special Properties of the Harmonic Drive. From the foregoing discussion of the basic rotary Harmonic Drive structure, the basis for several unique and inherent properties can be seen:

1. Tooth-motion relationship between Flexspline and circular spline makes unusually high speed reduction or speed increase possible in a single stage.
2. Under load, many spline teeth are in simultaneous engagement, resulting in high torque capacity.
3. Spline teeth come into contact with an almost pure radial motion and have essentially zero sliding velocity, even at high input speeds. Tooth friction losses and tooth wear are thus very low.
4. Because of low friction losses, high mechanical efficiencies can be obtained, which are particularly outstanding at high ratios.
5. Spline teeth in contact and under load are practically stationary. Dynamic loading, under normal operating conditions, is very low.
6. Regions of tooth engagement and application of load torque are usually diametrically opposed and result in a force couple that is symmetrical and balanced.
7. Harmonic Drive elements are concentric and tend to be self-aligning.
8. Tubular construction of splined elements results in high torsional rigidity and thus less windup under load.
9. Backlash is normally low but can be completely eliminated between Flexspline and circular spline by spring loading the two splines into mesh.
10. With Flexspline formed as an integral section of a flexible cylindrical wall, positive transmission of rotary mechanical motion through the wall can be achieved.

Harmonic Drive can also transmit rotary-to-linear motion into a hermetically sealed chamber. The wall can be part of a welded steel enclosure which is capable of maintaining complete isolation of two environments. For applications requiring the absolute containment of pressure, vacuum, liquids, or contaminants, Harmonic Drive can provide positive and reliable mechanical control and power transmission without the danger and inconvenience of pack glands or other partial seals.

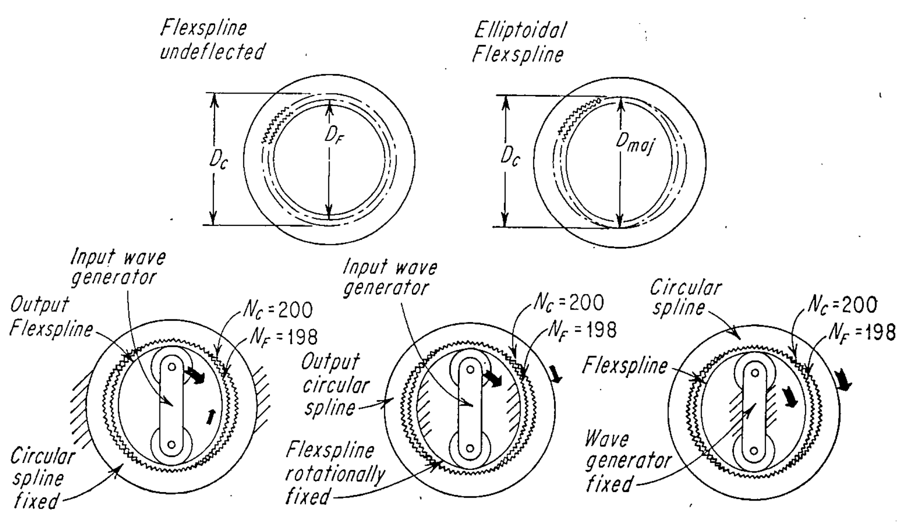


FIG. 3-46. Design relationships and ratios of single-stage rotary Harmonic Drive.

GE-1024.064

Wave-generator Configurations. The role of the wave generator is to produce controlled deflection of the Flexspline and to make possible the continuous rotation of the resulting elliptoidal shape. In the preceding figures the wave generators are *internal* and deflect the Flexspline outward into engagement with the circular spline. Figure 3-47 shows that wave generators can also be *external* to deflect the Flexspline inward into engagement with the circular spline. In this instance, spline engagement is at diametrically opposite regions on the *minor* axis of the elliptoid.

The most common form of wave generator is the elliptoidal cam, several configurations of which are shown in Fig. 3-47. The wave generators may form the internal

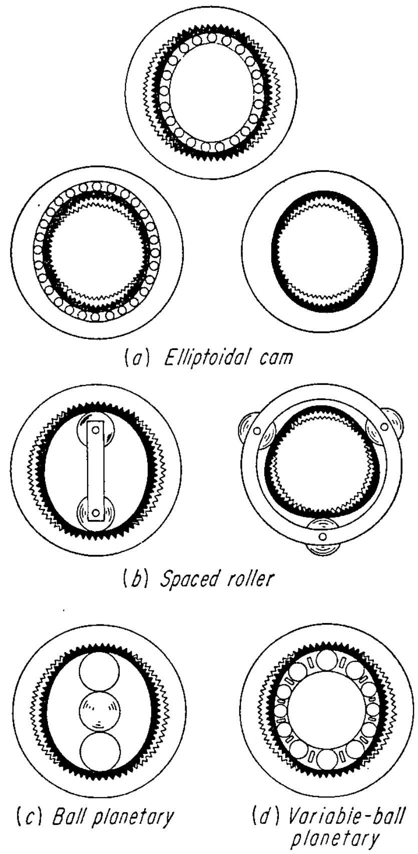


Fig. 3-47. Wave-generator configurations.

or external element of the Harmonic Drive stage and may have plain (or sleeve) bearing contact with the Flexspline or utilize a form of antifriction device. Antifriction bearings may be ball or roller and usually are the retainer type. There are other shapes as well that can be used.

Coupling to Flexspline. Motion of the wave generator or the circular spline can be joined rigidly or rotationally to external equipment by standard coupling means, but the nonrigid Flexspline requires less conventional means of position and motion coupling. Figure 3-45 shows how the Flexspline and circular spline are firmly splined together at the major axis of the elliptoid but move angularly relative to one another at all other points, with maximum relative motion occurring at the minor axis. For most applications, motion of the nonrigid Flexspline is transmitted through a type of coupling which integrates these variations in angular velocity and produces a uniform angular motion.

1. *Internal coupling.* From a performance point of view, the preferred Flexspline couplings are integral, as shown in panel 1 of Fig. 3-48. The Flexspline coupling is in the form of a cup with spline teeth in the cylindrical wall near the open end. Controlled deflection is imposed in the cup at the open end, while the bottom of the cup remains essentially circular. The length of the tubular wall acts as a deflection attenuator and provides a form of integrating coupling between

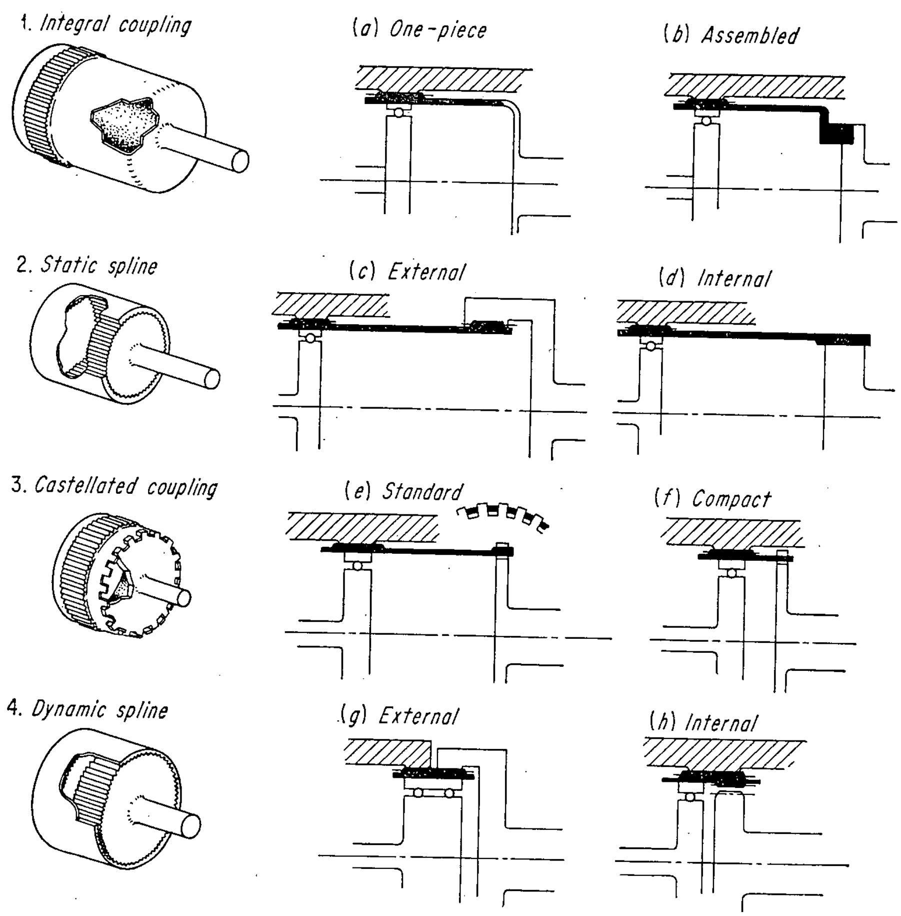


FIG. 3-48. Methods of mechanical coupling to Flexspline.

the spline engagement at the open end and a shaft joined to the cup bottom. The shaft and cup may be formed as one piece or assembled. Integral couplings have high torsional rigidity, low backlash, long life, high torque capacity, and no power loss within the coupling. Harmonic Drive efficiencies, with integral coupling, can be over 90 per cent. The assembled coupling can contain a drive motor, overload clutches, r-f pumping, etc.
2. *Static spline.* The next most efficient coupling is the static spline shown in panel 2 of Fig. 3-48. Here the flexing Harmonic Drive spline is shown at the left end of an open tube, with a standard circular spline being engaged at the right end. Coupling spline teeth can be cut either externally or internally

GE-1024.066

into the tube wall. The static spline coupling permits slight axial motion of the tube without allowing angular or radial freedom. Operating efficiency of the static spline is slightly less than that of an integral coupling. Fabrication cost is lower, but the static spline requires lubrication.

3. *Castellated coupling.* Another type of coupling is the castellated coupling shown in panel 3. The efficiency, positioning accuracy, and torque capacity of this coupling are governed by the number of lugs and the tube length between spline teeth and lugs. The shorter couplings, however, have about one-fourth the load capacity of an integral coupling since torque is not equally distributed among the lugs. Castellated couplings have lower positioning accuracy and efficiency but are the least expensive and most compact Flexspline couplings for a single-stage drive.

Composite Single-stage Harmonic Drive. With the principal operating elements of Harmonic Drive having been defined and illustrated schematically, it will be helpful to see how these elements function together to form a single Harmonic Drive stage. Figure 3-49 shows a simplified form of Harmonic Drive unit having an

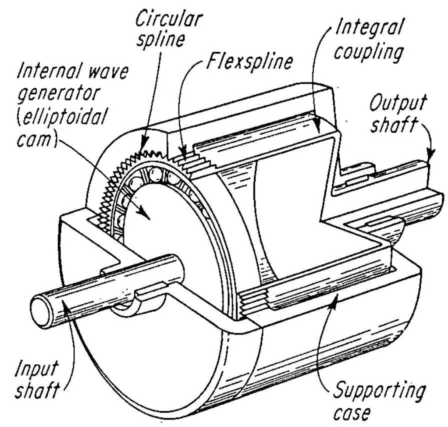


FIG. 3-49. Composite single-stage Harmonic Drive—simplified.

internal elliptoidal wave generator with ball bearing, a Flexspline with integral coupling to the output shaft, and a circular spline machined in the supporting case.

The wave generator imposes its elliptoidal shape onto the open end of the Flexspline cup, causing the Flexspline to mesh with the circular spline at two diametrical regions on the major axis. Rotation of the wave generator continuously advances the regions of spline engagement. The Flexspline rotates relative to the circular spline and transmits its motion through the integral coupling to the output shaft. While the Harmonic Drive in Fig. 3-49 is shown as a speed reducer, it can be used equally well as a speed increaser by interchanging the roles of the input and output shafts.

Harmonic Drive Performance Capabilities. Plotted Harmonic Drive performance curves are based on operational tests conducted with experimental Harmonic Drive units. One piece of test apparatus used is the special stand shown in Fig. 3-50. This was developed by United Shoe Machinery Corporation for the prolonged testing of rotary-to-rotary drives under accurately controlled load conditions. Here independent measurements are made simultaneously on two Harmonic Drive units, which can be tested interchangeably as speed reducer and speed increaser. Units have been driven for hundreds of millions of cycles, and their performance measured to better than 1 per cent accuracy.

Additional comparative data have been taken from Harmonic Drive systems developed for special military and industrial applications.

A band of values is shown in most plots, and this ranges from conditions of standard design and manufacture to conditions of custom design and more exacting methods of manufacture. These broad curves are drawn about "design center" values which

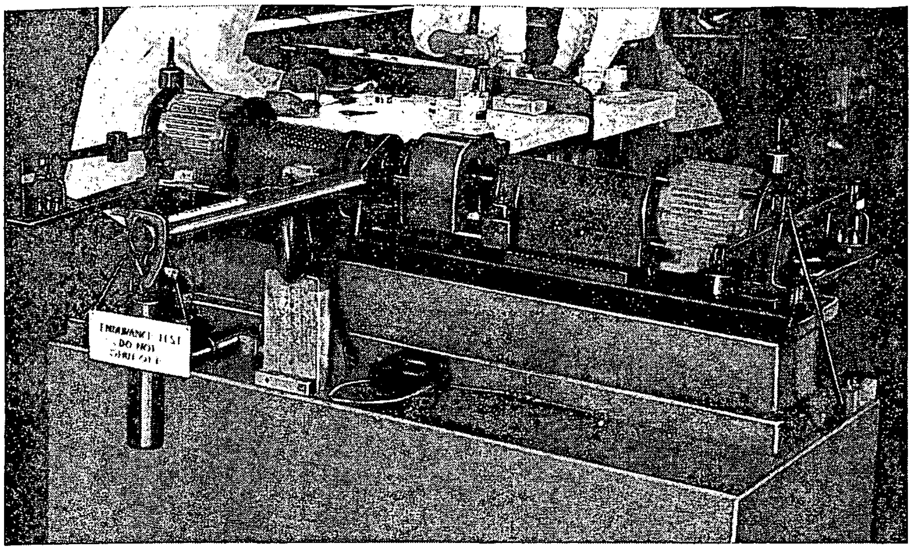


FIG. 3-50. Back-to-back endurance and efficiency test of Harmonic Drive units. *(Courtesy of United Shoe Machinery Corp., Boston, Mass.)*

are conservatively achievable today with Harmonic Drive. The design of any particular drive unit must, as usual, represent a balancing of these various parameters.

1. *Volume vs. output torque.* External volume of a separate 100:1 Harmonic Drive is plotted in Fig. 3-51, graph 1, against output torque. Harmonic Drive is presently capable of producing output torques of 25 lb-in. per cu in. for standard design, and up to 50 lb-in. per cu in. for compact design.
2. *Weight vs. output torque.* The weight of a separate 100:1 Harmonic Drive unit is plotted in graph 2 of Fig. 3-51 against output torque. Harmonic Drive torque output can range from 150 lb-in. per lb for standard design up to 500 lb-in. per lb for special lightweight design. Reducer weight varies with the type of construction, the materials used, and the manner in which the reducer is mechanically coupled with related elements.
3. *Reduction ratio vs. configuration.* In graph 3, the reduction ratios which can be achieved with four different Harmonic Drive configurations are shown. These are the general ranges achievable with steel Flexsplines in units of moderate physical size. In addition to the ratios shown, near-unity ratios and both positive and negative rotations can be achieved with Harmonic Drive. Except at the ultra-high ratios, the plotted ratios apply to speed increase as well as speed reduction. Only a few of the possible Harmonic Drive configurations are indicated here, and these types can be combined.
4. *Efficiency vs. ratio.* The range of efficiencies achievable with Harmonic Drive are shown in graph 4 as a function of a ratio. These figures apply to steel Flexsplines and include the losses of both input and output shaft bearings and couplings. The graph assumes an integral Flexspline. Where Harmonic Drive

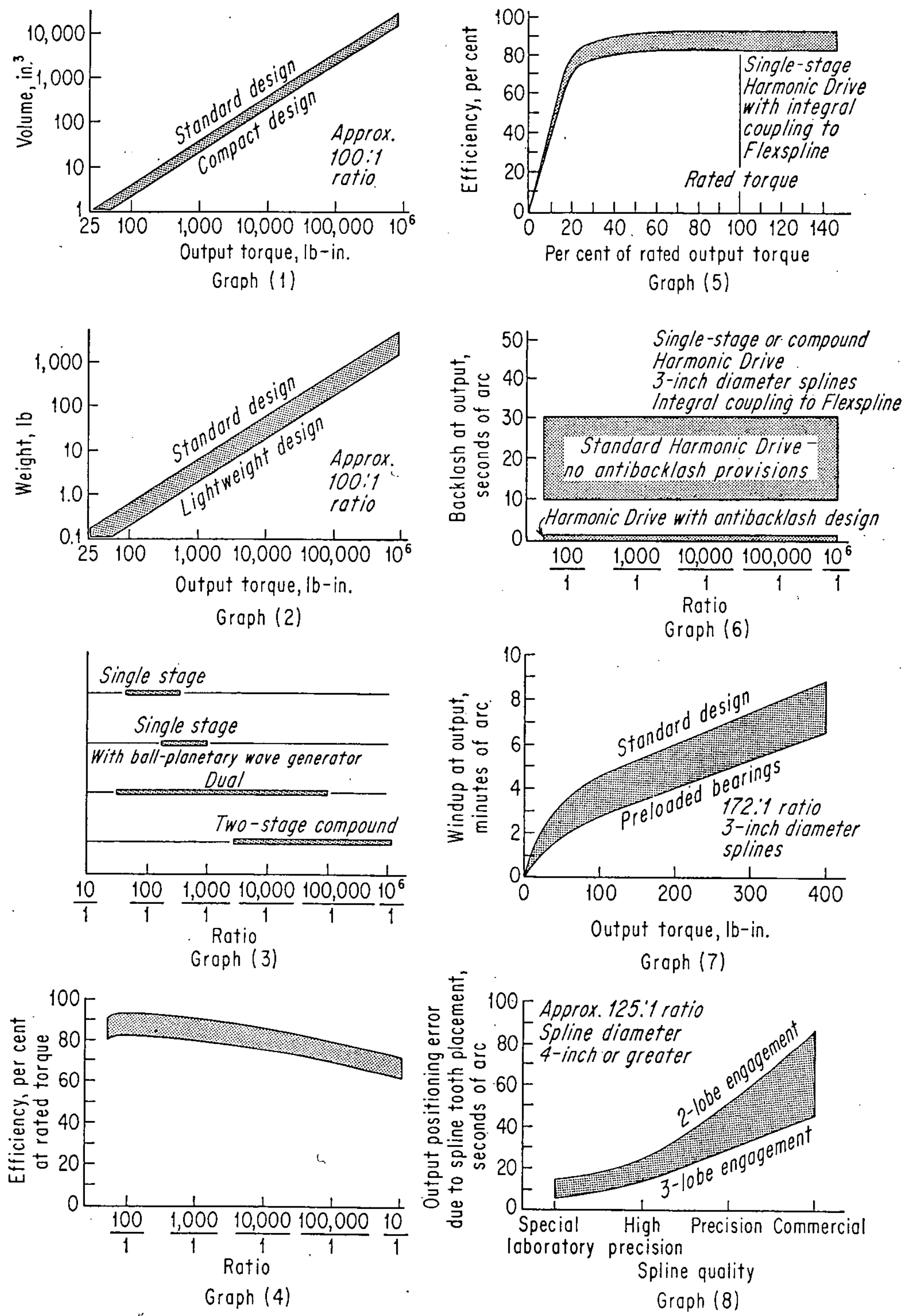


FIG 3-51. Normal capabilities of Harmonic Drive units.

has been used as both speed increaser and speed reducer, it has been found that the operating efficiencies are approximately equivalent.

5. *Efficiency vs. output torque.* Graph 5 shows that Harmonic Drive efficiency rises rapidly with increasing output torque and remains flat over a wide range. This plot pertains to a single-stage Harmonic Drive with integral coupling to the Flexspline.
6. *Backlash vs. ratio.* Backlash at the output of a Harmonic Drive stage is plotted in graph 6 against ratio. This graph is for single-stage or compound drive with approximately 3″-diameter splines and integral coupling to the Flexspline. The upper broad band relates to standard Harmonic Drive design with no special antibacklash provisions taken; the lower line shows substantially zero backlash achievable with minor design modification. An unusual characteristic of Harmonic Drive backlash is that it is essentially constant with increase in ratio; this results from the extremely high ratios possible per stage. Backlash of standard Harmonic Drives can be reduced by increasing the diameter of the splines.
7. *Windup vs. output torque.* Harmonic Drive has inherently low windup for a drive of given physical size and output torque capability. Windup characteristics of Harmonic Drive are illustrated by measurements taken on a specific unit with 172:1 reduction ratio, 380-lb in. rated output torque, and approximately 3″-diameter splines. With the input clamped, graph 7 shows the windup angle developed at the output as a function of the torque applied at the output. The plot shows the windup range between standard design and design using a preloaded output bearing. Windup can be decreased by increasing spline diameter.
8. *Positioning accuracy.* Graph 8 shows the positioning accuracy possible with Harmonic Drive, in terms of the general class of splines being used. While tooth-placement errors in standard gearing appear across the gear radius, errors in two-lobe Harmonic Drive appear across a full diameter and are both statistically and geometrically reduced. Further reduction in error of a three-lobe Harmonic Drive is shown in graph 8. This plot is for spline diameters 4″ and greater. No allowance has been made in these curves for the fact that multiple tooth engagement, unique in Harmonic Drive, tends to average and reduce errors caused by the shape or misplacement of individual spline teeth.

Harmonic Drive Applications. Industrial and military applications include speed reducers, combined motor reducers, linear and rotary actuators, servomechanisms, lifts and jacks, elevators and escalators, power take-offs, rotary-to-translatory motion converters, fine-coarse (two-speed) controllers, torque generators, instrument-panel controls, radio and radar antenna drives, and special turret-traversing mechanisms. Because of its ability to achieve positive mechanical drive through a sealed pressure wall, Harmonic Drive is uniquely suited for hermetically sealed power transmissions, sealed and pressurized valve actuators, mixer and agitator drives, control-rod drives for nuclear reactors, and special marine and medical uses.

REFERENCES

1. Benford, Robert L.: Planetary Gears, *Machine Design*, Aug. 21, 1958, pp. 129–135.
2. Wilkinson, William H.: Four Ways to Calculate Planetary Gear Trains, *Machine Design*, Jan. 7, 1960, pp. 155–159.
3. Sicklesteel, D. T.: Method for Design and Calculation of Planetary Gear Sets, *Design News*, June 15, 1952, pp. 45–49.
4. Poppinga, R.: The Efficiency of Planetary Gear Trains, *Engrs. Digest*, December, 1950, pp. 421–425.
5. Allen, H. N. G.: Epicyclic Gears, *Trans. Inst. Marine Engrs.*, Vol. 64, pp. 79–122, 1952.

6. Michalec, G. W.: Gear Differentials, *Machine Design*, October, December, 1955.
7. Tuplin, W. A.: Compound Epicyclic Gear Trains, *Machine Design*, Apr. 4, 1957, pp. 100–104.
8. Poppinga, R.: The Efficiency of Planetary Gear Trains, *Engrs. Digest*, December, 1950, pp. 421–425.
9. Eisbrenner, Ray D.: Brake on Output Shaft Permits Rapid Forward-reverse Shifting, *Design News*, Mar. 28, 1960.
10. (Editors Report): Planetary-type Reverse Gear Transmits 2500 H.P., *Design News*, Feb. 19, 1959.
11. Bryant, Richard C., and Darle W. Dudley: Which Right-angle Gear System?, *Product Engineering*, Nov. 7, 1960.

Chapter 14

LOADED GEARS IN ACTION

By

EUGENE E. SHIPLEY, *Project Engineer—Advance Development, General Electric Company, Lynn, Mass.*

Gear analysis and design are no more than an engineering guess until the gears go to work. They must be fully loaded and subjected to dynamic conditions to be sure they will perform as predicted. It is only then that the efficiency of the gear mesh, the dynamic-load effects, and the over-all performance of the gearbox can be noted and measured.

Those who design gears are constantly surprised that some gears run better and last longer than would be expected by the design formulas while others fail prematurely, even when operated well within the design limits of transmitted horsepower. New gear designs must be based on both theoretical and practical experience to assure that the very best gearsets are manufactured and installed in customer locations. The gear designer must be able to evaluate the friction of the gear mesh, bearing and windage losses, dynamic loads, and kinds and causes of gear failures in order to do the best possible job. In addition, he will be required to know the various types and kinds of gear-performance testing that can be done at the factory to simulate and supplement field conditions.

EFFICIENCY OF GEARS

One of the chief problems of any gear designer is that of obtaining reliable efficiency information on which to base his gear designs. Most gear designers, however, realize that gears are a very efficient method of transmitting power. In fact, in many applications gears are the only practical way to perform the task. Gears have the advantage not only of transmitting power, but also of transmitting this power at any speed ratio desired. Gears have efficiencies in the range of 98 per cent or more. On the surface it appears that this small loss could be neglected by gear designers and often it is. In many applications, however, this small friction loss can cause considerable concern, since it must be dissipated as heat throughout the gear system.

Gears with nonintersecting nonparallel axes, such as wormgears or Spiroids,®* have a meshing action consisting primarily of a sliding motion and consequently higher friction losses than parallel-axes gears such as spur or helical gears which have a combination of a high degree of rolling motion with a somewhat lesser degree of

* Registered trade-mark of the Spiroid Div. of Illinois Tool Works, Chicago, Ill.

sliding motion. It is quite evident that the efficiency of a given type of gear is tied in with the amount of sliding that takes place in the meshing action.

14-1. Sliding Velocity. Often the efficiency of various types of gears is expressed in terms of a sliding velocity. This being the case, it becomes quite important to understand how to calculate the sliding velocity of different gear drives.

Spur Gears. In the case of conjugated gear teeth such as spur or helical gears, the sliding velocity can be determined by the following equations:[1,*]

$$v_s = v_1 - v_2 \tag{14-1}$$

where v_1 = rolling velocity of the point in question on the pinion
v_2 = rolling velocity of the corresponding point on the gear

$$v_s = s(\omega_P + \omega_G) \tag{14-2}$$

where ω_P = angular velocity of pinion
ω_G = angular velocity of gear
s = distance along line of action to the point in question from the pitch point

Since the sliding velocity is expressed as the algebraic difference of the two rolling velocities of the point in question, it is often more convenient to calculate v_s in terms of v_1 and v_2 as indicated below.

$$v_1 = \frac{n_P \pi \rho_1}{360} \tag{14-3}$$

$$v_2 = \frac{n_G \pi \rho_2}{360} \tag{14-4}$$

where n_P = rpm of pinion
ρ_1 = radius of curvature of pinion
n_G = rpm of gear
ρ_2 = radius of curvature of gear

Sometimes sliding velocities are required at specific points such as the sliding velocity at the pinion tip or the sliding velocity at the gear tip. In this case,

$$v_{sP} = \frac{n_P}{114.59} \times Q_r \frac{m_R + 1}{m_R} \tag{14-5}$$

where v_{sP} = sliding velocity at pinion tip
Q_r = arc of recession of driving gears, radians

$$v_{sG} = \frac{n_G}{114.59} \times Q_a \frac{m_R + 1}{m_R} \tag{14-6}$$

where v_{sG} = sliding velocity of gear tip
Q_a = arc of approach, radians

Bevel Gears. As in the case of spur and helical gears, the sliding velocity of a point on a bevel gear, a distance s from the pitch point, is approximately

$$v_s = s(\omega_P{}^2 + \omega_G{}^2 + 2\omega_P\omega_G \cos \Sigma)(\sin^2 \phi + \cos^2 \psi \cos^2 \phi)^{1/2} \tag{14-7}$$

where ω_P = angular velocity of pinion radians per sec
ω_G = angular velocity of gear radians per sec
Σ = shaft angle

* Superscript numbers refer to references at the end of the chapter. See p. 141 of reference.

In the case of right-angle bevel drives, $\Sigma = 90°$ and

$$v_s = s(\omega_P \cos \gamma + \omega_G \cos \Gamma)(\sin^2 \phi + \cos^2 \psi \cos^2 \phi)^{1/2} \tag{14-8}$$

where γ = pitch-cone angle of pinion, $\gamma + \Gamma = 90°$
Γ = pitch-cone angle of gear
ψ = spiral angle

Wormgears. The sliding velocity, or rubbing velocity, as it is generally referred to for wormgears, can be calculated as follows:[2]

$$v_r = \frac{0.262 n_W d}{\cos \lambda} \tag{14-9}$$

where v_r = rubbing velocity, fpm
λ = lead angle
n_W = rpm, worm
d = pitch diameter of worm

Crossed-axes Helical. Crossed-axes helical gears can be calculated in a similar manner.[3,*]

$$v_r = \frac{0.262 nD \sin \Sigma}{\cos \psi_2} \tag{14-10}$$

where $\Sigma = \psi_1 + \psi_2$
D = diameter of driver
n = rpm of driver
ψ_2 = helix angle of driven
ψ_1 = helix angle of driver

When the shaft angle = 90° sin of Σ equals 1.00 and the above equation becomes

$$v_r = \frac{0.262 nD}{\cos \psi_2} \tag{14-11}$$

Hypoids. The sliding velocity for a hypoid gear can be found by using a type of equation similar to that for wormgears or crossed-axes helical gears. In fact, a hypoid gear is an offset-axis bevel gear and has much in common with an offset-axis helical, or "crossed-axis" helical, as it is more properly called.

$$v_s = \frac{0.262 Dn}{\sin \gamma} \tag{14-12}$$

where v_s = sliding velocity, fpm
D = diameter of base cylinder of pinion hyperboloids
n = rpm of pinion
γ = angle of generatrix with axis of base cylinder of pinion

14-2. Efficiency of Coplanar Gears (*Parallel Axes and Intersecting Axes*). In applications where large amounts of power are being transmitted, the efficiency of gears becomes very important. The most efficient gear designs are required from the point of view of conservation of energy as well as a reduction in the amount of heat rejected to the geared system. In general terms, the nonparallel- nonintersecting-axis drives have higher sliding or rubbing velocities, resulting in higher losses than parallel-axis and intersecting-axis drives. Consequently, the latter types are most efficient. For this reason, spur, helical, or bevel gears are usually desired for high-power-transmission applications unless the design conditions are such that other types of gear drives have more attractive features.

Spur Gears. The over-all efficiency of spur gears, all gears for that matter, is dependent on three separate and distinct types of losses. These three types are

* See p. 191 of reference.

commonly known as (1) windage and churning losses, (2) bearing losses, and (3) gear-mesh losses. The efficiency information presented here is based only upon the losses of the gear mesh.[4]

The per cent efficiency for spur gears is

$$E = 100 - P_l \tag{14-13}$$

where P_l = per cent power loss

The power-loss equation is

$$P_l = \frac{50f}{\cos\phi}\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right) \tag{14-14}$$

where f = average coefficient of friction, dimensionless
ϕ = pressure angle, deg
H_s = specific sliding velocity at start of approach action
H_t = specific sliding velocity at end of recess action

H_s and H_t can be determined as follows:

$$H_t = \frac{m_G + 1}{m_G}\left[\sqrt{\left(\frac{r_o}{r}\right)^2 - \cos^2\phi} - \sin\phi\right] \tag{14-15}$$

and

$$H_s = (m_G + 1)\left[\sqrt{\left(\frac{R_o}{R}\right)^2 - \cos^2\phi} - \sin\phi\right] \tag{14-16}$$

where m_G = gear ratio, a whole number
r_o = outside radius of pinion, in.
r = pitch radius of pinion, in.
R_o = outside radius of gear, in.
R = pitch radius of gear, in.

H_s and H_t can also be determined from the following relationships:

$$H_s = \frac{v_{sG}}{v} \tag{14-17}$$

where v_{sG} can be determined from Eq. (14-6) and

$$v = 0.262 \times n \times d \tag{14-18}$$

where d = pitch diameter of pinion, in.
n = rpm of pinion

$$H_t = \frac{v_{sP}}{v}$$

where V_{sP} can be determined from Eq. (14-5).

Figures 14-1 and 14-2 give values for the average coefficient of friction f vs. pitch-line velocity for various K-factor gear loadings.

A short-cut method has been developed to analyze standard spur-gear trains which is quite helpful, particularly in the early design stages. The power loss of a given set of gears is a function of the coefficient of friction of the gear mesh and the so-called mechanical advantage of the gear mesh, that is:

$$P_l = \frac{f}{M} \times 100 \tag{14-19}$$

where P_l = per cent power loss
f = coefficient of friction
M = mechanical advantage of the mesh

The mechanical advantage M, for 20 and 25° pressure angle standard spur gears, has been plotted for various combinations of number of gear and pinion teeth and can be found by referring to Figs. 14-3 and 14-4. The same coefficient of friction shown in Figs. 14-1 and 14-2 is used in the above equation. The curves shown in Fig. 14-5 give the nominal efficiency of spur gears.

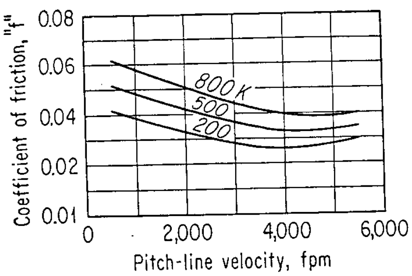


FIG. 14-1. Average coefficient of friction of lightweight petroleum oil. 120°F oil inlet hardened gears 45 SUS at 100°F.

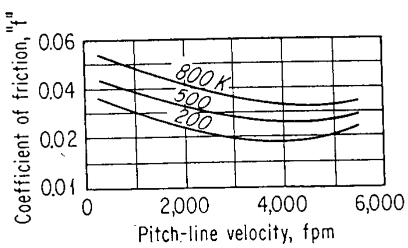


FIG. 14-2. Average coefficient of friction of medium-weight petroleum oil. 120°F oil inlet hardened gears 300 SUS at 100°F.

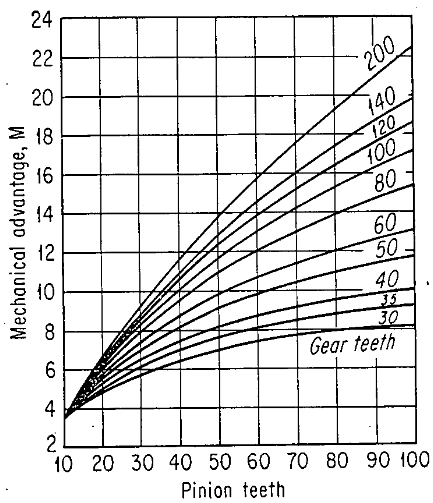


FIG. 14-3. Mechanical advantage 20° pressure angle spur gears. Pressure angle = 20°. Per cent power loss = P. Coefficient of friction = f. Mechanical advantage = M. $P = f/M \times 100$.

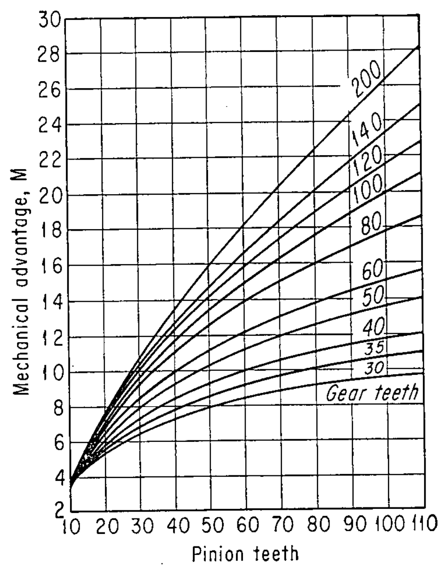


FIG. 14-4. Mechanical advantage 25° pressure angle spur gears. Pressure angle = 25°. Per cent power loss = P. Coefficient of friction = f. Mechanical advantage = M. $P = f/M \times 100$.

Internal Spur Gears. The efficiency of internal spur gears is very similar in nature to that of external spur gears and can be determined by the same general equations, namely:

$$E = 100 - P_l \tag{14-13}$$

and

$$P_l = \frac{50f}{\cos \phi} \left(\frac{H_s^2 + H_t^2}{H_s + H_t} \right) \tag{14-14}$$

except that H_s and H_t are determined in a slightly different manner, as specified by the following equations:

$$H_t = \frac{m_G - 1}{m_G}\left[\sqrt{\left(\frac{r_o}{r}\right)^2 - \cos^2\phi} - \sin\phi\right] \tag{14-20}$$

and

$$H_s = (m_G - 1)\left[\sqrt{\left(\frac{R_i}{R}\right)^2 - \cos^2\phi} - \sin\phi\right] \tag{14-21}$$

where R_i = inside radius of internal gear

It is quite obvious from these equations that the power loss of internal spur gears will be lower than that of external gears and consequently more efficient for a given coefficient of friction. Since both internal and external spur gears are very similar in nature except for the differences in specific sliding velocities, the curves of coefficient of friction vs. rpm as shown in Figs. 14-1 and 14-2 can be used for internal spur gears as well as external spur gears.

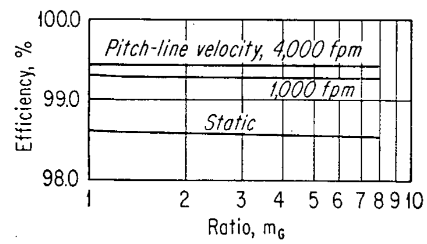


FIG. 14-5. Nominal efficiency of spur gears. Based on a good grade of petroleum oil. Hardened and ground gears 60 Rockwell C.

Single-helical Gears. The efficiency of helical gears is somewhat similar in nature to spur and internal gears. In fact, if a helical system is analyzed in terms of an equivalent spur gear, the equation for the per cent power loss can be written as follows:[5,*]

$$P_l = \frac{50f\cos^2\psi}{\cos\phi_n}\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right) \tag{14-22}$$

Where H_s and H_t are determined by solving Eqs. (14-15) and (14-16), it must be kept in mind, however, that

ϕ_n = normal pressure angle of helical gear
ψ = helix angle
f = coefficient of friction

The resultant efficiency is determined by the same general efficiency equation $E = 100 - P_l$.

It will become obvious from these equations that a helical gear is somewhat more efficient than a similar spur gear. Often this is an important consideration when designing gears for minimum power loss.

Helical gears are similar in nature to spur gears; therefore, the equations of coefficient of friction as shown in Figs. 14-1 and 14-2 can be used for calculation purposes.

Double-helical and Herringbone Gears. Power loss or the efficiency of double-helical gearing and herringbone gearing is determined in exactly the same manner as for single-helical gears. Since the power being transmitted by a set of double-helical gearing splits exactly in two, the total power loss is just double that for one side. The per cent power loss calculation need be determined for only one side. This per cent loss can then be multiplied by the total transmitted horsepower of the double-helical set to obtain total power losses.

Straight Bevel Gears. Straight bevel gears can be likened to spur gears except, of course, that they mesh at right angles. Bevel gears are often analyzed as equivalent spur gears by using Tredgold's approximation to determine the spur-gear cross

* See p. 345 of reference.

sections. Fundamentally, the number of teeth in virtual spur gears can be found as follows:[6]

$$N_{vG} = \frac{N_G}{\cos \Gamma} \tag{14-23}$$

$$n_{vP} = \frac{N_P}{\cos \gamma} \tag{14-24}$$

where N_{vG} = number of teeth in virtual spur gear
N_G = number of teeth in actual bevel gear
N_{vP} = number of teeth in virtual spur pinion
N_P = number of teeth in actual bevel pinion
Γ = pitch-cone angle of bevel gear
γ = pitch-cone angle of bevel pinion

When the above information is completely integrated into the power-loss equations for spur gears, the following equation can be derived for determining the per cent power loss for straight bevel gears:

$$P_l = 50f\left(\frac{\cos \Gamma + \cos \gamma}{\cos \phi_n}\right)\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right) \tag{14-25}$$

H_s and H_t are determined by solving Eqs. (14-15) and (14-16). It must be kept in mind that

ϕ_n = normal pressure angle of the bevel gear
r_o = outside radius of large end of bevel pinion
r = pitch radius of large end of bevel pinion
R_o = outside radius of large end of bevel gear
R = pitch radius of large end of bevel gear

The efficiency for straight bevel gears is again determined by the same general efficiency equation.

$$E = 100 - P_l \tag{14-13}$$

Since bevel gears have so much in common with spur gears,[7] it is quite proper to use the curves of Figs. 14-1 and 14-2 to determine the coefficient of friction of bevel gears. The curve shown in Fig. 14-6 gives the nominal efficiency of straight bevel gears. It is interesting to note from these curves that the straight bevel gear mesh is quite efficient, being in the range of 99 per cent or better.

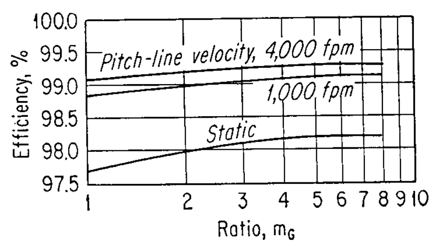


Fig. 14-6. Nominal efficiency of straight bevel gears. Based on a good grade of petroleum oil. 33 to 40 Rockwell C.

Zerol®* Bevel Gears. Zerol bevel gears are made in a different type of machine than the straight bevel gears.[8] The difference is that the straight bevels are cut on a bevel-gear planer which moves the cutting tool back and forth in straight lines, whereas the Zerol bevel gears are cut with a rotary cutter. This rotary motion produces a curved tooth section in the length of the tooth; however, the tooth has a zero-degree spiral angle and consequently is very similar to a straight bevel gear from an efficiency point of view. Actually, the efficiency of a Zerol bevel gear can be calculated using the information and equations for straight bevel gears as described in the above article.

Figures 14-1 and 14-2 can also be used to determine the mesh efficiency of Zerol bevel gears. One of the chief advantages of Zerol bevel gears is that they can be bardened and ground quite accurately. They are often made this way and con-

* Registered trade-mark of the Gleason Works, Rochester, N.Y.

sequently can be loaded higher than straight bevel gears. This being the case, then the curves in Fig. 14-6 would be slightly higher for hardened and ground Zerol bevel gears.

Spiral Bevel Gears. Spiral bevel gears have often been referred to as helical bevel gears.[9] In practice, however, the spiral bevel gear does not have a true helical spiral. They are cut on or ground on the same machine that produces Zerol bevel gears, except that the cutting tool is set at an angle to the axis of the gear, thus producing the helical-gear effect.

From an efficiency point of view, a spiral bevel gear is very good and can be compared with the results of helical gearsets. Generally, spiral bevels are used where high load and high speeds are encountered, since they can be hardened and ground and produced quite accurately.

The per cent power loss for a spiral bevel gear can be determined by solving the following equation:

$$P_l = 50f(\cos\Gamma + \cos\gamma)\frac{\cos\psi^2}{\cos\phi_n}\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right) \tag{14-26}$$

where Γ = pitch-cone angle of spiral bevel gear
γ = pitch-cone angle of spiral bevel pinion
ψ = spiral angle
ϕ_n = normal pressure angle
f = coefficient of friction

H_s and H_t must be calculated by using Eqs. (14-15) and (14-16); however, the general comment made in the discussion of straight bevel gears applies equally to spiral bevel gears. The coefficient of friction as shown in Figs. 14-1 and 14-2 can be used for calculation purposes.

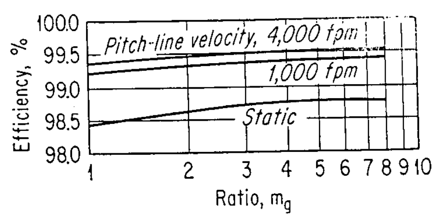


FIG. 14-7. Nominal efficiency of spiral bevel gears. Based on a good grade of petroleum oil. Hardened and ground gears. Spiral angle = 35°.

Figure 14-7 gives the *nominal* efficiency of spiral bevel gears vs. gear ratio. Note that the efficiency has been plotted for various values of pitch-line velocity. Actually, these curves have been based on full rated power being transmitted by the gearset and are only representative of what can be done. They are quite useful as a guide, but if close efficiency estimates must be made, it would be wise to calculate your design from the very beginning. By using the proper equations, the values of coefficient of friction, pressure angle, and gear geometry for your particular design can be fed into the equation and a more precise answer will be obtained.

14-3. Efficiency of Nonplanar Gears (*Nonintersecting and Nonparallel Axes*). Many gear applications require the features of this general class of gearing. Although coplanar gears are generally more efficient, this class of gearing can be designed with high-performance characteristics.

Wormgears. The per cent efficiency for wormgear drives can be determined as follows for the case where the worm drives the worm wheel:

$$E = 100\frac{\cos\phi_n - f\tan\lambda}{\cos\phi_n + f\cot\lambda} \tag{14-27}$$

where ϕ_n = normal pressure angle
f = coefficient of friction
λ = pinion lead angle

When the worm wheel drives the worm, Eq. (14-27) becomes

$$E = 100\frac{\cos\phi_n + f\tan\lambda}{\cos\phi_n - f\cot\lambda} \tag{14-28}$$

In many instances, not all the information pertaining to a given set of wormgears is known. In this case a simplified equation for wormgear efficiency can be used.[5,*]

$$E = 100\frac{\tan\lambda}{\tan(\lambda+\eta)} \tag{14-29}$$

where η = friction angle
$\tan\eta = f$ approximately

Since generally the coefficient of friction has been determined with some experimental error, Eq. (14-29) is a fair approximation.

It is quite helpful to analyze the relationship of lead angle and the efficiency of various combinations of wormgear drives by studying Tables 14-1 and 14-2. Tables

Table 14-1. Efficiency of Worm Gearset, Worm Driving

Worm lead angle, deg	f = 0.015 eff., %	f = 0.02 eff., %	f = 0.03 eff., %	f = 0.04 eff., %	f = 0.05 eff., %	f = 0.07 eff., %	f = 0.10 eff., %
3	77.5	72.0	65.0	55.0	48.0	42.0	35.0
5	86.0	81.5	75.1	70.0	63.0	55.0	48.0
10	92.0	89.5	84.5	82.0	77.0	72.0	63.0
15	94.5	92.5	89.5	87.0	83.2	78.0	72.0
25	96.2	95.1	92.6	90.5	88.5	84.0	78.0
35	96.8	95.9	93.9	92.1	90.2	86.8	81.5
45	97.1	96.2	94.2	92.4	90.5	87.2	82.0
55*	96.8	95.8	93.8	91.8	89.8	86.0	80.5
65	96.0	94.6	92.1	89.3	87.3	82.0	75.0
75	93.9	91.9	88.1	84.0	80.0	72.0	61.0
80	91.0	88.0	82.1	75.3	71.0	59.0	40.0
85	82.5	75.3	63.0	50.3	40.0	20.0	

* Generally worms are not designed for more than 45° lead angle. An angle of 55° is usually the very upper limit for a practical design.

Table 14-2. Efficiency of Worm Gearset, Wormgear Driving

Worm lead angle, deg	f = 0.015 eff., %	f = 0.02 eff., %	f = 0.03 eff., %	f = 0.04 eff., %	f = 0.05 eff., %	f = 0.07 eff., %	f = 0.10 eff., %
3	69.0	58.0	40.0	20.0	0		
5	82.5	77.0	63.0	53.0	40.0	20.0	
10	91.0	88.0	82.5	75.3	71.0	58.0	40.0
15	93.9	91.9	88.2	84.0	80.0	72.0	61.0
25	96.0	94.6	92.1	89.8	87.0	81.9	75.0
35	96.8	95.7	93.8	91.8	89.8	86.0	80.5
45	97.1	96.2	94.3	92.5	90.6	87.1	82.2
55*	96.8	96.0	93.9	92.0	90.2	86.8	81.5
65	96.3	95.1	92.6	90.7	88.5	84.0	78.0
75	94.3	92.8	89.5	87.0	83.3	78.0	72.0
80	92.0	89.5	85.3	82.0	78.0	72.0	64.0
85	86.0	81.9	81.3	70.0	63.0	56.0	47.0

* Generally worms are not designed for more than 45° lead angle. An angle of 55° is usually the very upper limit for a practical design.

* See p. 347 of reference.

14-1 and 14-2 were determined by assuming various values for the coefficient of friction and solving the efficiency equations. In order to use Tables 14-1 and 14-2 profitably, values of the coefficient of friction for your given conditions must be known.

Figure 14-8 gives some experimental values of f based on using a good grade of mineral oil after the gearset has been well broken in.

Figure 14-9 shows the nominal efficiency of cylindrical wormgears as a function of lead angle and rubbing velocity. It is understood that double-enveloping wormgears also follow such a pattern as outlined in Fig. 14-9, although it has been said that double-enveloping wormgears are more efficient under starting conditions than cylindrical wormgears.

Crossed-axes Helical. Crossed-axes helical gears are actually nonenveloping wormgears and as such can be calculated with efficiency equations very similar in nature to equations for cylindrical wormgears.

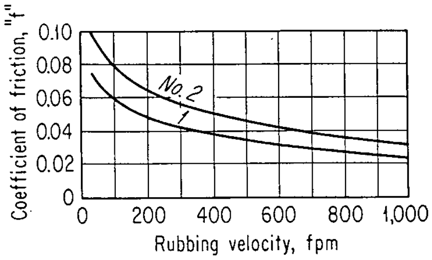


FIG. 14-8. Coefficient of friction for wormgears. Based on a good grade of petroleum oil. No. 1 = case-carburized and ground worm and phosphor-bronze gear. No. 2 = cast-iron worm and wormgear.

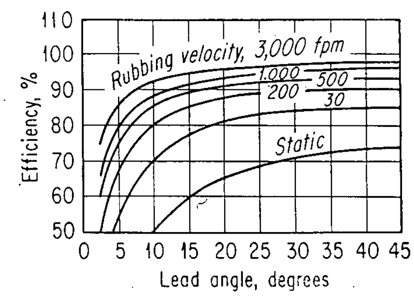


FIG. 14-9. Nominal efficiency of wormgears. Based on case-carburized and ground worm and phosphor-bronze gear.

Since crossed-axes helical gears can be of the same hand or opposite hands and the spiral angle or helix angle can be different for either member, the equations for the efficiency are a little tricky and must be used with caution. For the case where the gears are of the same hand, then,[5,*]

$$E = 100\,\frac{1 - f\tan\psi_f}{1 + f\tan\psi_d} \qquad (14\text{-}30)$$

where f = coefficient of friction
ψ_d = helix angle of driver
ψ_f = helix angle of follower

When the gears are of opposite hand and ψ_d is less than ψ_f,

$$E = 100\,\frac{1 - f\tan\psi_f}{1 - f\tan\psi_d} \qquad (14\text{-}31)$$

When the gears are of opposite hand and ψ_d is greater than ψ_f,

$$E = 100\,\frac{1 + f\tan\psi_f}{1 + f\tan\psi_d} \qquad (14\text{-}32)$$

The coefficient of friction f for spiral gears is practically the same as that for wormgears; therefore, the same curves of f vs. rubbing velocity can be used (see Fig. 14-8).

Spiroid Gears. Spiroid gears are comparable in performance and efficiency with wormgears and in general can be used in many applications where heretofore worm-

* See p. 355 of reference.

gear drives were specified.[10] The efficiency of Spiroid gears can be determined from an equation which has much in common with the basic wormgear-efficiency equation. Efficiency in per cent for Spiroid gears can be expressed as follows:

$$E = 100 \frac{\cos \phi_n + f \cot \alpha_g}{\cos \phi_n + f \cot \lambda} \tag{14-33}$$

where ϕ_n = normal pressure angle
f = coefficient of friction
α_g = gear spiral angle (with respect to the plane of rotation)
NOTE: This is 90° minus the spiral angle as used for spiral bevel gears.
λ = pinion spiral angle

$$\tan \lambda = \frac{L}{2\pi r_m} \tag{14-34}$$

and L = conical lead
r_m = mean cone pinion radius at a point half way across the face

In order to evaluate Eq. (14-33) fully, however, additional information is required, such as

$$\alpha_g = 90° - (\sigma - \lambda) \tag{14-35}$$

where σ = normally standard at 40°
λ = lead angle as indicated above

The normal pressure angle is taken as 10° except when the gearset is run with the contact on the opposite side of the pinion teeth. Then it is driving on the high-pressure-angle side, which usually has a pressure angle of 30°. In order to understand better the geometry relationship of the above equations, refer to Fig. 14-10, which shows a contacting segment of a Spiroid gear and a Spiroid pinion.

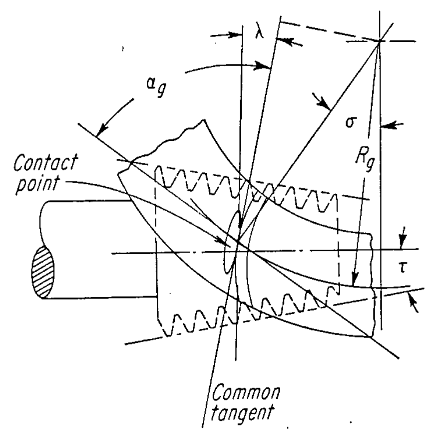


FIG. 14-10. Contacting segment of a Spiroid gear and pinion.

The relationship of the coefficient of friction vs. sliding velocity can be seen in Fig. 14-11. Curves have been plotted for the efficiency of Spiroid gears vs. pinion Spiroid angle (see Fig. 14-12). Note from Fig. 14-12 that the gears used were hardened and ground. Little difference has been observed when bronze gears have been used instead of hardened gears. At very low speeds, though, a bronze gear and a hardened pinion combination does appear to be somewhat more efficient.

Helicon®* Gears. Helicon gears are very similar in nature to Spiroid gears in that they are skew-axis gears. They differ by virtue of the fact that the pinions have a zero degree taper angle, whereas standard Spiroid pinions have a 5° taper angle.[11] Helicon gears also are flat-faced instead of conical-faced like Spiroid gears. Other than this, the gear types are very much alike and can be calculated using exactly the same equations as outlined above for Spiroid gears. The standard σ "sigma" angle is 40°. The 10° pressure angle applies to the Helicon gear as well as the Spiroid gear. Figure 14-11 can be used to determine the coefficient of friction for Helicon gears. Figure 14-12 also depicts typical Helicon gear efficiency vs. rubbing velocity. It must be kept in mind, however, that Helicon gears by design have less face width in contact and will carry less load than their brothers the Spiroid gears. Consequently, the

* Registered trade-mark of Spiroid Div., Illinois Tool Works, Chicago, Ill.

GE-1024.082

over-all efficiency of Helicon drives will be less efficient as compared with the over-all efficiency of Spiroid gear drives.

Hypoid Gears. Hypoid gears have much in common with plain or spiral bevel gears.[12] In fact, hypoid gears are actually offset-axis bevel gears, and as such, they are subjected to a high degree of sliding as are most of the other types of offset or "nonplanar" gears. In comparison with a wormgear, however, the rubbing speed of

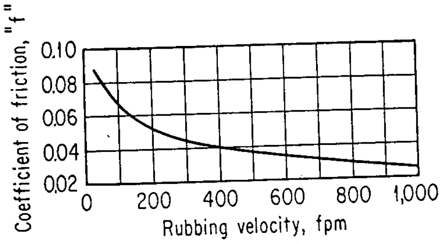


FIG. 14-11. Coefficient of friction vs. sliding velocity of Spiroids. Based on carburized pinion and gear, 60 Rockwell C, and a good grade of petroleum oil.

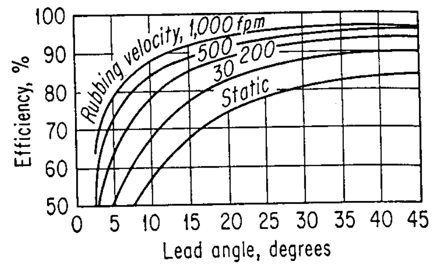


FIG. 14-12. Nominal efficiency of Spiroid gears. Based on case-carburized pinion and gear, 60 Rockwell C, and a good grade of petroleum oil.

a hypoid relative to its own pitch-line velocity is not so high. The per cent efficiency for hypoid gears can be determined as follows:

$$E = 100 \frac{\cos \phi_n + f \tan \psi_g}{\cos \phi_n + f \tan \psi_p} \tag{14-36}$$

where ϕ_n = normal pressure angle
f = coefficient of friction
ψ_g = gear spiral angle
ψ_p = pinion spiral angle

The nominal efficiency for hypoid gears can be found in Fig. 14-13. These curves were plotted for the conditions where the sum of the gear and pinion spiral angle equals 75° and the normal pressure angle is 22°30′. It is quite interesting to note that at low ratios the hypoid gears are quite efficient. Generally speaking, the coefficient of friction for hypoid gears is similar to friction values of wormgears, and consequently the curves shown in Fig. 14-8 can be used for hypoid-gear calculations.

Planoid®* Gears. Planoid gears perform a function similar to that of hypoid gears. They are right-angle gears with offset axes. The tooth surfaces of the gears are planes and the pinions are slightly tapered with helical teeth.[13] The recommended ratio range for Planoid gears is from 15:1 to 10:1. The efficiency of Planoid gears compares quite favorably with that of hypoid gears.

The efficiency of Planoid gears can be determined from the following equation:

$$E = 100 \frac{\cos \phi_n + f \cot \alpha_g}{\cos \phi_n + f \cot \lambda} \tag{14-37}$$

where ϕ_n = normal pressure angle
α_g = gear spiral angle
λ = pinion spiral angle
f = coefficient of friction

* Registered trade-mark of Spiroid Div., Illinois Tool Works, Chicago, Ill.

GE-1024.083

The efficiency of Planoid gears compares favorably with that of hypoid gears. Since Planoids are usually designed for low ratio, they generally are quite efficient, being in the range of 96 per cent or better. In reference to Fig. 14-13, Planoid gears fit nicely in the range of industrial and automotive gears from an efficiency point of view. Like hypoid gears, the efficiency of Planoid gears can be calculated using the coefficient-of-friction data of wormgears as outlined in Fig. 14-8.

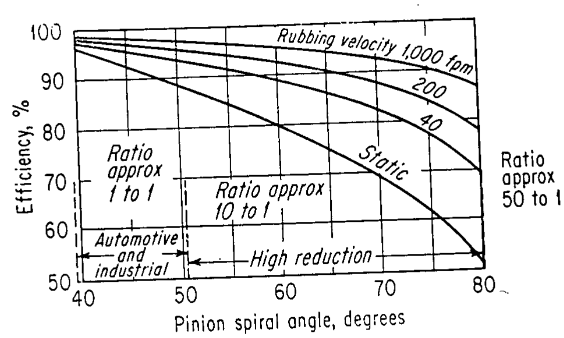


FIG. 14-13. Nominal efficiency of hypoid gears. Based on the sum of gear and pinion spiral angles equal to 75°, and normal pressure angle of 22°30′.

14-4. Gear Trains. In order to analyze a complete gear train, information pertaining to all the losses of a gearbox must be considered, not just the gear mesh itself. As stated previously, windage and churning losses and bearing losses are very important factors and at times contribute considerably to the over-all efficiency of a complete gear train. Even the gear arrangement itself, in some cases, must be carefully analyzed to be sure that a complete loss picture is obtained. For example, a simple planetary is inherently more efficient than a simple star arrangement, even though they are generally composed of very similar gear elements. It is the load transmitted and the meshing velocity, among other things, that in the final analysis determine the amount of power being dissipated in the mesh. Sometimes both the load and the meshing velocities are difficult to determine in complex gear arrangements.

Bearing Losses. In order for gear elements to transmit power, they must be supported in the gear casing by some sort of bearing which will permit free and accurate rotation with a minimum amount of heat generation. Generally, of course, the bearings are lubricated with an ample supply of oil both to minimize the coefficient of friction in the bearing and to act as a cooling medium to carry away the heat that has been generated.

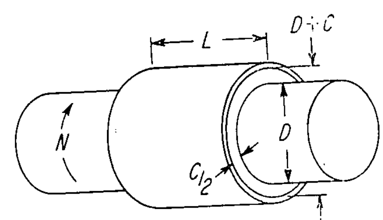


FIG. 14-14. Elements of a sleeve bearing.

These bearings usually fall into two main categories, namely, rolling contact bearings such as ball and roller bearings, or hydrodynamic bearings commonly known as "sleeve"-type bearings. Usually, a given gearbox design is equipped with either one type of bearing or the other. In a few applications, however, a mixture of the two types has been used.

The horsepower loss of the hydrodynamic-type sleeve bearing can be calculated from the laws of viscosity since the power consumption of this type of bearing results from the shearing of the oil film.

If the shaft is assumed to be in the center of the bearing, the losses can be calculated using the well-known Petroff equation[14,*] with reference to Fig. 14-14.

$$P_b = 3.79 \times 10^{-13} \frac{Z_2 n^2 D^3 L}{C} \qquad (14\text{-}38)$$

where P_b = horsepower loss in bearing
Z_2 = outlet oil viscosity (absolute), cp
n = rpm
D = diameter, in.
L = length of bearing, in.
C = clearance diametrical, in.

This equation serves as a very good approximation for lightly loaded bearing since the shaft on lightly loaded bearings tends to ride in the center.

In practice, however, and particularly with bearings that carry an appreciable load, the shaft usually runs eccentric in the bearing. In this case, the above calculated horsepower loss (P_b) must be modified by a power-loss coefficient which is a numerical representation of the bearing characteristics.

$$P_b' = JP_b \qquad (14\text{-}39)$$

where J = power-loss coefficient

The power-loss coefficient can be calculated[14,†] by solving the following rather complex equation with reference to Fig. 14-15:

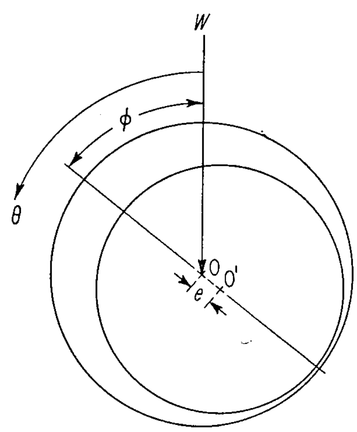


Fig. 14-15. Definition of a loaded sleeve bearing.

$$J = \frac{pCe \sin \varphi \times 10^7}{2.90\pi ZnD^2} \frac{1}{2\pi} \int_0^{2\pi} \frac{BC}{2Lh}\, d\Theta \qquad (14\text{-}40)$$

where C, Z, D, L, and n are as previously defined
p = load per unit projected area, psi
e = eccentricities, in.
Θ = angle to point on bearing surface measured from load line
B = width of reduced bearing oil film in expanding region of bearing, in.
h = film thickness, in.
φ = attitude angle between line of center and load line

The above equation for J is somewhat difficult to solve; however, J has been determined for many applications and plotted in Fig. 14-16. You will note from Fig. 14-16 that J has been plotted for various L/D ratios vs. the Summerfield number S.

$$S = 2.42 \times 10^{-9} \times \left(\frac{D}{C}\right)^2 \times \frac{Zn}{p} \qquad (14\text{-}41)$$

In order to determine J, it is a simple matter to calculate the Summerfield number and the L/D ratio, then go to Fig. 14-16 and "pick off" the proper value for J.

If the particular bearing under consideration is an elliptical-type sleeve bearing, then the J values found in Fig. 14-17 should be used. The P_b in Eqs. (14-38) and (14-39) can be used to calculate the basic horsepower loss. For solving these equations for an elliptical bearing design, the average diameter is used for D.

* See p. 192 of reference.
† See p. 220 of reference.

Fig. 14-18 can be used for determining the proper value of J for four axial-groove sleeve bearings. Again, Eqs. (14-38) and (14-39) can be used to calculate the horsepower loss.

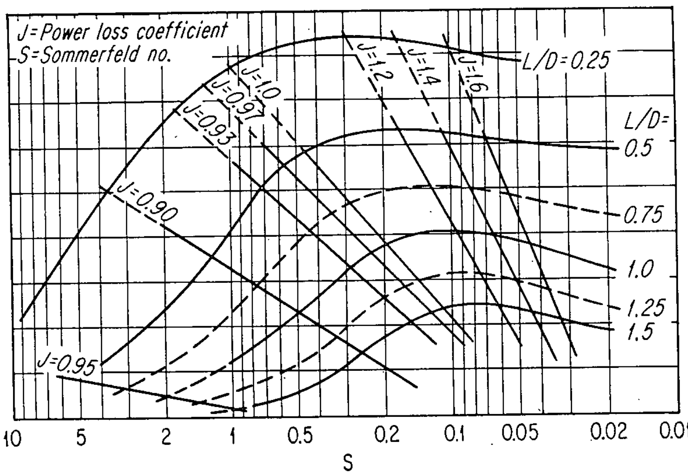


FIG. 14-16. Plot of J cylindrical bearings.

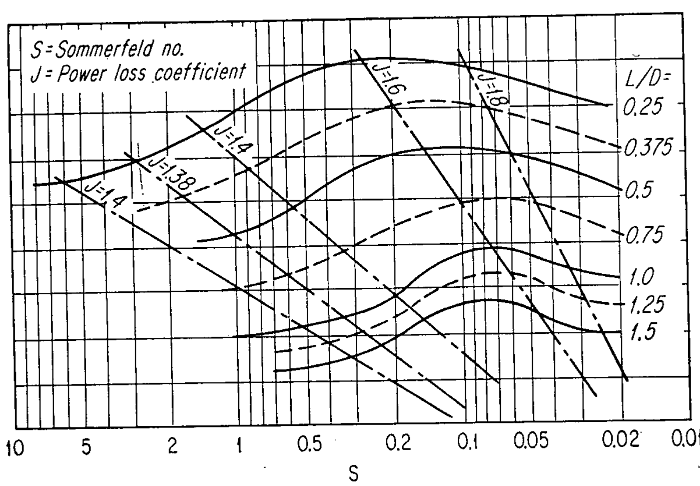


FIG. 14-17. Plot of J elliptical bearings.

The power loss of simple thrust bearings can also be determined in similar fashion by using the laws of viscous flow.

$$P_t = 3.79 \times 10^{-13} \times \frac{Z_2 n^2}{t} (R_2^4 - R_1^4) \qquad (14\text{-}42)$$

where P_t = horsepower loss of a simple thrust bearing
Z_2 = outlet viscosity, cp
t = oil-film thickness
R_1 = inside radius of thrust bearing
R_2 = outside radius of thrust plate

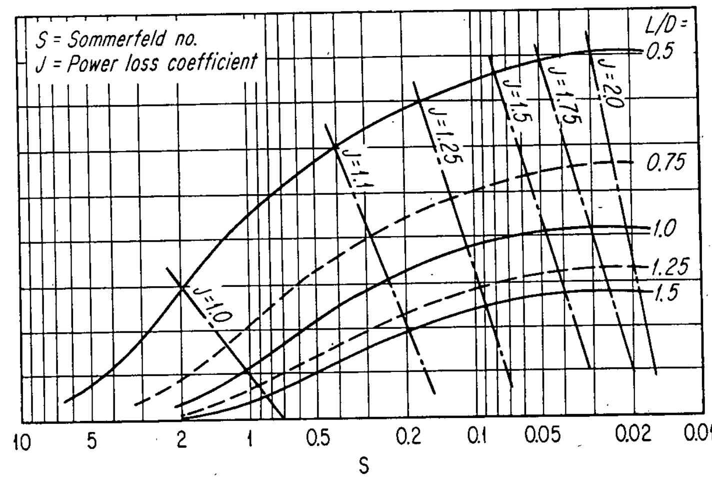


FIG. 14-18. Plot of J four-groove bearings.

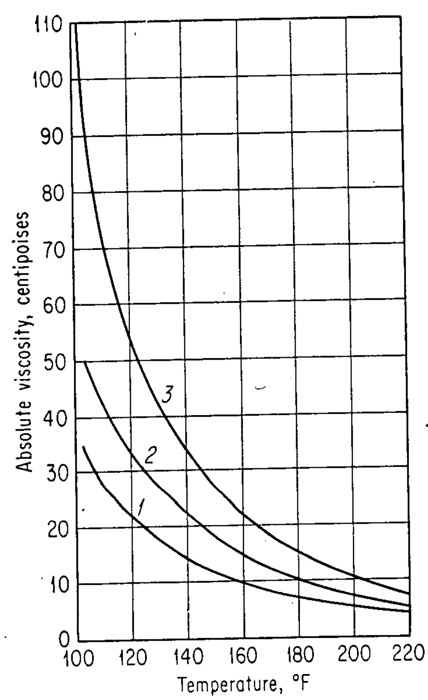


FIG. 14-19. Viscosity of petroleum oil. (1) AGMA 1—200 SSU at 100°F. (2) AGMA 2—300 SSU at 100°F. (3) AGMA 3—600 SSU at 100°F.

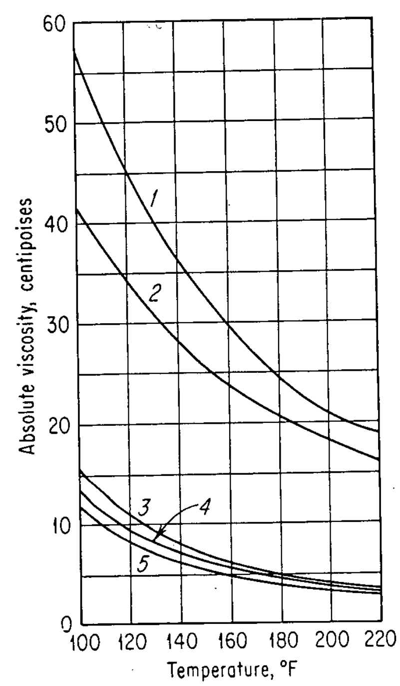


FIG. 14-20. Viscosity of synthetic oils. (1) Dow Corning XF-258 (Silicone). (2) GE Versalube F-50 (Silicone). (3) MIL-L-9236B (Celluthem 2505A). (4) MIL-7808D. (5) MIL-L-25336 (Sinclair L-743).

Figure 14-19 gives the values of absolute viscosity in centipoises for various grades of petroleum oil vs. temperature of the oil. This chart is quite useful when calculating bearing losses. Figure 14-20 gives similar information on other types and grades of oils, some of which are synthetics.

It is extremely difficult to calculate bearing losses and have them coincide exactly with measured bearing losses. There are many sound reasons why variations between actual and predicted bearing losses exist. Some of these reasons are:

1. Bearing misalignment at assembly
2. Variation in viscosity-temperature behavior of the lubricating oil from that used in computation
3. Presence of phenomena such as oil turbulence or recirculation
4. Variation in operating variables such as shaft speed and bearing loads from those used in computations

The losses of rolling-contact bearings depend to a large degree on the type and quantity of lubricant used in a given application. Exact predictions of these losses are extremely difficult. However, a considerable amount of experimental work has been done. Empirical equations were determined from these investigations. Like most experimental work, the results must be used with extreme care and understanding.

One of the simplest approaches to the bearing-loss problem is to determine the torque loss in the bearing and then calculate the horsepower loss as follows:[14,*]

$$P_b = \frac{Tn}{63{,}000} \tag{14-43}$$

where P_b = horsepower loss in bearing
T = torque loss per bearing, lb-in.
n = rpm of bearing

The torque loss T can be found simply by the use of a constant coefficient of friction for each type of bearing.

$$T = f\frac{D_1}{2}W \tag{14-44}$$

where T = loss torque lb-in.
D_1 = bore diameter of bearing, in.
W = load on bearing, lb

This approach offers a fast and reasonable approximation based upon normal operating loads and speeds with favorable conditions.

The above equation can be used for a variety of bearing types as long as the proper coefficients of friction are used[15,†] (see Table 14-3). It must be kept in mind, however, that, under starting conditions or if higher-viscosity lubricants are used, higher values of coefficient of friction will be obtained.

Table 14-3

Type of bearing	*Coefficient of friction f*
Radial ball bearing (single-row deep groove)	0.0015
Self-aligning ball bearing	0.0010
Angular-contact bearing	0.0013
Thrust ball bearing	0.0013
Cylindrical roller bearing	0.0011
Spherical roller bearing	0.0018
Tapered roller bearing	0.0018

* See p. 83 of reference.
† See p. 21 of reference.

Figure 14-21 can be used as a guide to determine the exact type of bearing used in your particular design.

Some gear designers prefer to determine the approximate torque loss of rolling-contact bearings by using the pitch diameter of the rolling elements for the appropriate diameter in the torque equation[16] or

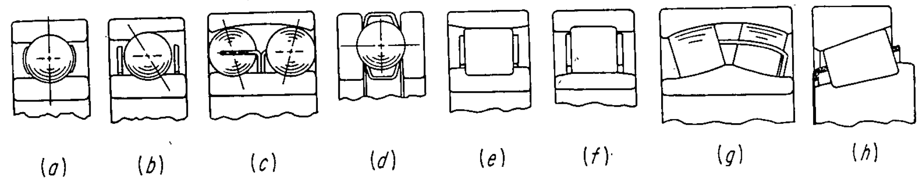


FIG. 14-21. Types of bearings. (a) Radial ball bearing. (b) Angular-contact bearing. (c) Self-aligning ball bearing. (d) Thrust ball bearing. (e) Cylindrical roller bearing (outside race free). (f) Cylindrical roller bearing (inside race free). (g) Spherical roller bearing. (h) Tapered roller bearing.

$$T = f\frac{D_p}{2}W \tag{14-45}$$

where D_p = pitch diameter of rolling elements

This equation is perhaps theoretically more correct in that the rolling-contact area is where the actual loss takes place, not in the bore of the bearing. This equation also permits some consideration for the differences between bearings of the same bore diameter but with larger balls or rollers. Of course, the larger ball or roller operates a larger pitch diameter. It is somewhat more difficult to obtain the operating pitch diameter of a rolling-contact bearing; however, it can be approximated by the following equation:

$$D_p = \frac{D_1 + D_2}{2} \tag{14-46}$$

where D_p = pitch diameter of elements, approximately, in.
D_1 = bore diameter
D_2 = outside diameter

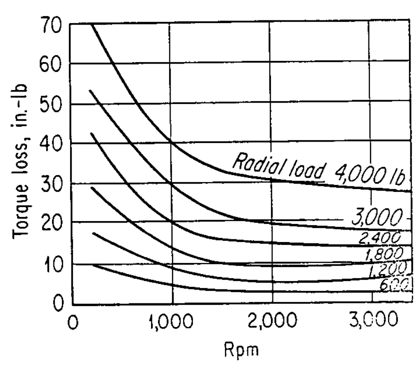


FIG. 14-22. Losses of 22,308 spherical roller bearings. Bearing torque loss vs. rpm and load 40-mm bore × 33-mm width. Double-row spherical roller bearing. Light aircraft oil at 120°F inlet.

As an example of how the actual bearing loss varies for a spherical roller bearing, some 22,308 bearings were tested by the author. Figure 14-22 shows these results. Note that the loss torque at slow speeds is quite high compared with that at the faster speeds. It is also quite interesting to note that, as the speed is increased the losses in the bearing tend to increase, undoubtedly because of the churning effect of the oil.

A very complete work on friction torque of ball bearings was carried out by Manlio Muzzoli.[14,*] The equation developed for the friction torque of deep-groove ball bearings under radial load is as follows:

$$T = 0.0868KV\left[n^{0.5}(2.6d^{0.2} - 3.33)n^{0.375d-0.2} + \frac{(0.454W)^m}{72n^{0.73} + 0.150N_b(d - 6)^{0.055}}\right] \tag{14-47}$$

* See p. 84 of reference.

where d = ball diameter, mm
N_b = no. of balls
n = rpm
W = load, lb
K = constant, lubricant
V = constant, conformity
m = determined by Eq. (14-48)

$$m = 1.65 + 0.004n^{0.54} + 0.005dN_b \tag{14-48}$$

The constant V can be evaluated by referring to Fig. 14-23. Note that V is plotted against conformity, which is the race curvature divided by ball diameter. Bearings are generally manufactured to a conformity of 52 per cent, which results in a value of V equal 1.1.

The constant K, which takes into consideration the type of lubricant, can be found in Table 14-4.

Table 14-4

Fluid grease	$K = 1.0$
Dense grease	$K = 1.2$
Very dense grease	$K = 2.0$
Very light oil	$K = 0.4$
Light oils	$K = 0.6$

Dr. Booser and Dr. Wilcock[14,*] state they have found that modern good ball-bearing greases will give a K value of 0.10 to 0.15, which is somewhat better than those given in Table 14-4. They also indicate from the results of many of their tests that the Muzzoli equation (14-47) is fundamentally correct.

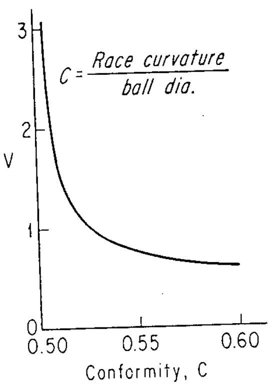


FIG. 14-23. Plot of V vs. conformity.

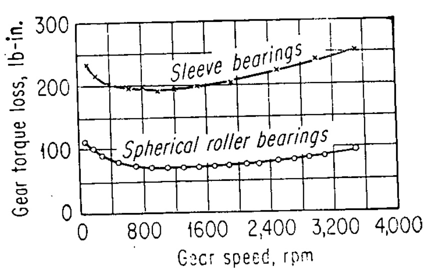


FIG. 14-24. Comparison of gearbox torque losses (sleeve bearings vs. rolling-contact bearings).

The curves as indicated in Fig. 14-24 give a good graphical comparison of the losses of a given gearbox when mounted on first sleeve-type bearings and then rolling-contact bearings. In both cases, the gearbox was loaded with exactly the same amount of torque. In many respects the curves are just what would be expected in that the sleeve-bearing design has higher breakaway torque. The losses of the sleeve bearing at speed are still higher in that a considerable amount of oil is required to keep the

* See p. 85 of reference.

bearing functioning properly and this generally results in a high viscosity-loss effect. However, at really high speeds and loads, these two curves tend to cross, thus making the sleeve-bearing performance more comparable with the rolling-contact bearing.

Windage Losses. The windage losses of a gearbox are very difficult to measure or calculate. Investigations have been carried out on this subject, but very little good information has been published. In general terms, windage-loss determination is still pretty much based on individual experience of the gear designer and experimental measurements on specific gearboxes in question.

Actually the windage losses for a given gear design depend on many things such as:

1. The diameter of the rotating elements
2. The length of the rotating elements
3. The speed of rotation
4. The web or gear-blank design
5. The over-all casing design
6. The type of oil-feed system
7. The operating temperature and viscosity of the oil
8. The pressurization of the casing

Some equations have been written that give a fair approximation of the windage losses of small-diameter gears, say up to approximately 20" in diameter with an L/D ratio of approximately 0.5.

$$P_w = \frac{n^3 D^5 L^{0.7}}{100 \times 10^{15}} \tag{14-49}$$

where P_w = horsepower loss due to windage
n = rpm
D = diameter of rotating element
L = length of rotating element

Another interesting equation that sometimes is quite useful is the equation that was empirically derived for a rotating smooth body in air. The rotating element was not enclosed by a gear casing, nor was it exposed to any lubricant.

$$P_w = \frac{15}{0.746}\left(\frac{n}{1{,}000}\right)^3\left(\frac{D}{100}\right)^4\left(\frac{5L}{100} + \frac{D}{100}\right) \tag{14-50}$$

The answer obtained by using this formula will undoubtedly give the minimum value of windage obtainable.

Sometimes it is quite helpful to realize just when you should start to be concerned about windage losses, even if the methods of evaluating it leave something to be desired.

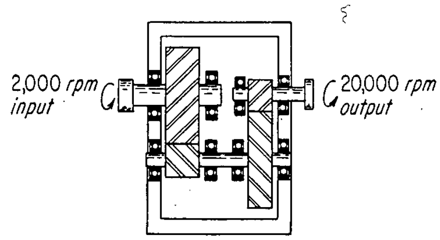


FIG. 14-25. High-pitch-line-velocity gearing.

Often windage losses are combined with other extraneous losses in the system. Stated in another manner, when the mesh losses and bearing losses are subtracted out of the total losses, windage losses are what remain.[17] A gear designer should concern himself about windage losses when his design starts to suffer from any of the troubles indicated schematically in the following sketches, namely, Figs. 14-25, 14-27, 14-28, 14-29, 14-31, and 14-33.

Figure 14-25 indicates a high-speed gearbox with pitch-line velocities higher than 10,000 fpm. Generally, when the meshing velocity reaches 10,000 fpm, the windage losses begin to increase to the point where a sophisticated design should take them into consideration. There is not much that can be done about high pitch velocities, except to keep the diameters of the rotors as small as possible.

As an indication of the effect windage losses may have on a high-speed gear train, refer to Fig. 14-26. This information was obtained experimentally from efficiency tests on a single-offset aircraft gearbox. It is quite interesting to note that in a well-designed gearbox little, if any, windage loss is present at the lower speeds.

Figure 14-27 indicates a tight-gear-casing arrangement whereby the gear casing fits tightly around the gearing. This is done because of weight and space consideration; however, it usually leads to windage troubles. Relief can often be found by shrouding the gears with oil shields. The oil-discharge port, in particular, must be protected from the "hurricane" within the casing in order to assure a free flow of oil from the casing.

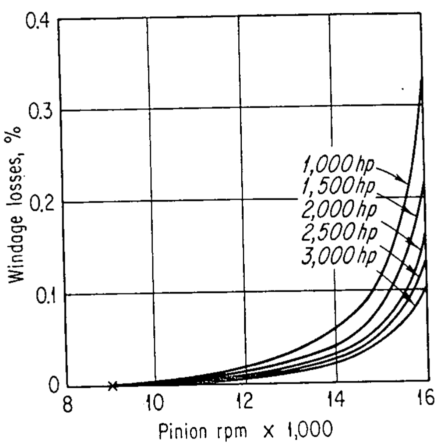


FIG. 14-26. Windage losses—aircraft gear. Single offset. MIL-L 7808D oil. 250°F inlet oil.

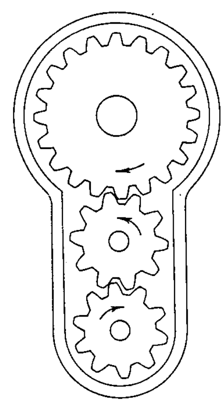


FIG. 14-27. Tight-gear-casing arrangement.

Not only does this type of arrangement lead to excessive heat being generated in the oil because of churning, but sometimes more drastic troubles are encountered, such as pumping of the oil within the casing, which generally results in a hydraulic braking action. If this trouble is encountered, the gear unit must be shut down immediately or the unit may suffer a catastrophic failure. In addition to shrouding, a unit of this nature should be scavenged with a pump that has from 2½ to 3 scavenge ratio.

Figure 14-28 illustrates a compact-gear-train arrangement. Usually a gear train of this sort is an accessory-drive package, whereby the gears themselves do not have excessively large face widths, but they generally are quite high speed. Some of the gearing is not too highly loaded in that often the center distance required dictates the size of the gearing, not the power being transmitted. Any free oil in a gearbox of this nature gets "kicked" and battered about before it finally leaves the box to be recirculated again. In a gear design of this nature, it is the best policy to keep the oil flow at an absolute minimum to the gear mesh, as well as to the bearings. Generally, by carefully observing the early test results of a given design, oil can be redirected and redistributed to lubricate the complex gear train adequately without having to resort to shrouding. In some instances, however, a well-placed oil deflector can be quite beneficial. Again, removing the oil from the gear casing and preventing any build-up of surplus oil is a must; thus usually a scavenge pump is required, but a 2:1 scavenge ratio generally is high enough.

Figure 14-29 illustrates a poor or inadequate oil-scavenge arrangement. This type of trouble sometimes accompanies a tight-gear-casing arrangement or a compact-gear-train arrangement. Often enough, it occurs in more simple types of gearboxes such as the one illustrated in Fig. 14-29, primarily because of engineering oversight. When inadequate drainage and scavenge capacity are present, the gearbox will fill up with oil, causing extreme foaming and hydraulic braking. Shrouding the gears in areas where the oil has a tendency to pile up helps. The discharge ports must be

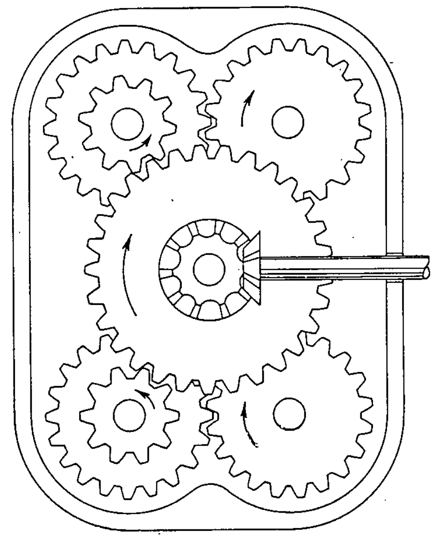


FIG. 14-28. Compact-gear-train arrangement.

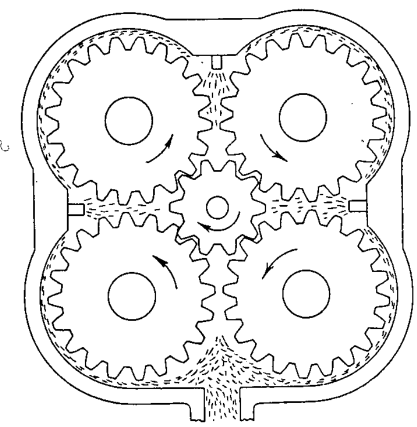


FIG. 14–29. Inadequate oil scavenge.

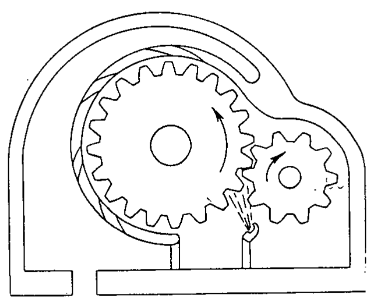


FIG. 14-30. Oil strippers.

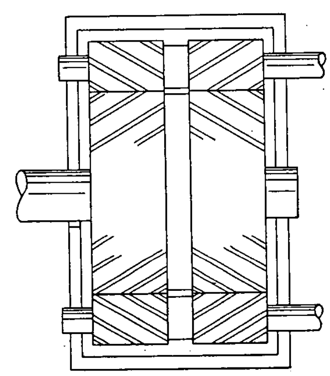


Fig. 14-31. Wide-face-width gearing.

shielded in order to permit a free flow of oil from the casing. Sometimes, if possible, oil strippers can be used to advantage (see Fig. 14-30). Of course, in any gear casing where excessive oil becomes a problem, it is always wise to cut down to the bare minimum the amount of oil that is initially put into the gear casing.

Figure 14-31 shows a wide-face-width gear design. In this sketch, the L/D ratio of the gearing is quite high. Because there is considerable face width, the windage loss tends to build up in relation to the length of the rotating bodies. If the gear design was such that the face width was only ½″ or so, the windage loss might be quite small, but since the gearing could be up to several feet in length, the losses due to windage increase considerably. Another factor that contributes to additional losses is that the oil sometimes becomes trapped in the mesh, since it has a much greater distance to go before it can be expelled from the ends. Actually, there is very little that can be done to help a design of this nature, except to allow for plenty of room within the casing. If the gearing is double-helical, as most likely it would be with large-face-width gearing, the helix angle could be so arranged that the helix angle would tend to force the oil toward the ends of the gear teeth. This arrangement would prevent the oil from being stacked up at the gap between helices.

From a windage-loss point of view, the inlet-oil temperature should be kept as high as practical within good over-all operational characteristics of both the oil and the equipment. As an example of temperature effects on the windage losses of a given gearbox, refer to Fig. 14-32. This figure shows the no-load losses of a given gearbox with respect to inlet-oil temperature. Except for the churning losses of the bearings at no load and the churning losses of the gear mesh itself, the losses shown are basically windage losses and depict to a great extent the effect of oil temperature on over-all windage losses.

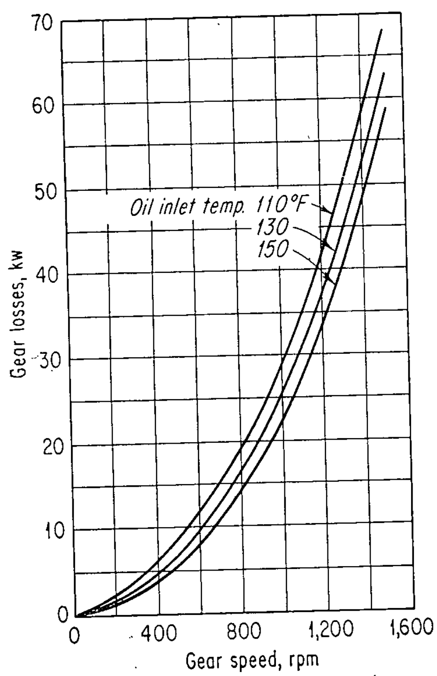


FIG. 14-32. Oil temperature effects on windage losses. Oil-inlet pressure 8 psi, Navy 2190 T oil.

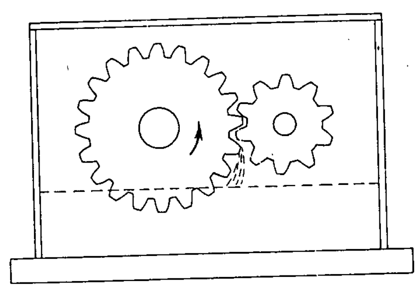


FIG. 14-33. Dip- or splash-feed lubrication.

Figure 14-33 shows a simple gearbox that is lubricated by the dip or splash feed system. Any time that a design of this nature is considered, the designer must be alert to the possibility of high losses due to churning of the oil. Not only must the designer be aware of this problem, but he also must be aware of the tendency of the lubricant to channel, thus rendering the lubrication system ineffective. This arrangement is generally utilized for low-speed drives; in fact, many wormgear applications use this system.

Single-offset Gears. A single-reduction two-element gearbox is the simplest type of gear train, and consequently the determination of the over-all efficiency of this arrangement is quite straightforward. First, the mesh loss is determined by solving Eq. (14-13) or (14-14) with the help of Fig. 14-1. Second, the bearing losses are determined by solving Eq. (14-38) for sliding-contact bearings. This horsepower loss should then be converted to the per cent loss basis, so that it may be added to the mesh loss. If rolling-contact bearings are present, then Eq. (14-43) should be used,

GE-1024.094

and the loss torque determined should also be converted to a per cent loss basis, and again added to the mesh loss. Third, the windage losses for the gearbox should be determined by solving Eq. (14-49), first for the pinion and then for the gear. The results should be converted to a per cent loss, then added to the total loss already accumulated. This new total represents the over-all loss of the complete gear train. Caution should be exercised to assure that there are no bad seal rubs or oil trapping (see Figs. 14-27, 14-28, and 14-33). Factors of this nature cause additional losses and

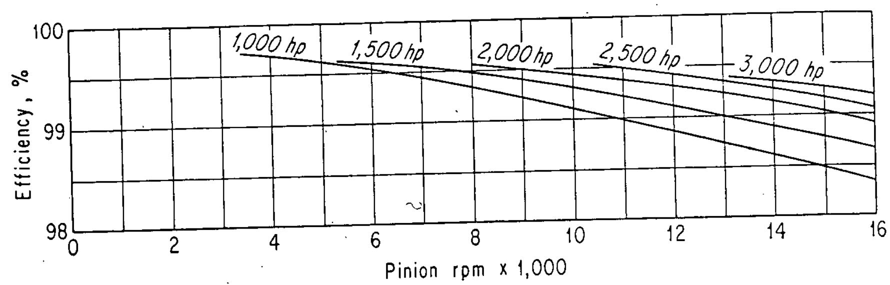


FIG. 14-34. Over-all efficiency—single-offset gear. Constant horsepower. Inlet-oil temperature 250°F; oil—MIL-L-7808C.

are not indicative of good design practice. Gear trains of this type are very efficient, particularly if high gear-tooth loading is present. Figure 14-34 shows the results of recent tests on an aircraft single-offset gearbox with rolling-contact bearings. Note from Fig. 14-34, that the gearbox is of a relatively high-speed and high-load design.

Double-reduction Gears. A double-reduction gear design is actually two single-offset gear trains in series. This being the case, the over-all gearbox efficiency is determined exactly like the single-offset gear train. Each mesh, each bearing, and each rotor is calculated for the appropriate loss. The individual losses are then totaled up and the over-all gearbox efficiency is determined. Triple-reduction gear trains, etc., are also calculated in a similar manner, as well as any combination of spur, helical, bevel, or internal gears.

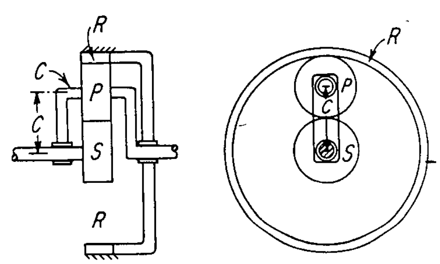


FIG. 14-35. Simple planetary drive.

Simple Planetary Drive. A simple planetary-gear system consists of three major parts; a fixed member, a driving member, and a driven member.[18,*] A common arrangement of this combination is shown in Fig. 14-35. In this train of gears, R is the fixed member, S is the driving member, and C is the driven member. In order for the cage C to function, an idler P is required in the system. The efficiency of this drive depends on the tooth loads and the velocity of the tooth arrangements. The product of tooth load and engagement velocity gives a measure of the equivalent power being transmitted by the mesh. Generally, the equivalent power being transmitted is less than the actual power being transmitted.

Actual power transmitted can be determined as follows:

$$P_a = W_t \times v \tag{14-51}$$

where W_t = actual load being transmitted

v = pitch-line velocity of actual transmitted load

* See p. 129 of reference.

$$W_t = \frac{2T}{d_S} \quad (14\text{-}52)$$

$$v = 0.2618 \times d_S \times n_S \quad (14\text{-}53)$$

where T = torque transmitted
d_S = pitch diameter of sun pinion
n_S = rpm of sun pinion

The equivalent power being transmitted through the sun gear mesh is

$$P_s = W_t \times v_S \quad (14\text{-}54)$$

where v_S = pitch-line velocity of tooth engagement of sun pinion

$$v_S = v + \frac{d_R}{d_R + d_S} \quad (14\text{-}55)$$

where v = from Eq. (14-53)
d_R = pitch diameter of ring gear
d_S = pitch diameter of sun pinion

The equivalent power being transmitted through the ring gear mesh is

$$P_R = W_t \times v_R \quad (14\text{-}56)$$

However, $v_S = v_R$
Therefore, $P_R = P_S$

The efficiency of the drive can be determined by calculating the per cent loss for each mesh. The power loss for each mesh is the product of the per cent loss and the equivalent power being transmitted. The over-all efficiency then is found by dividing output power by the input power or

$$E = \frac{W_t \times v - (\text{per cent loss sun} \times W_t \times v_S + \text{per cent loss ring} \times W_t \times v_R)\tfrac{1}{100}}{W_t \times v} \quad (14\text{-}57)$$

Note the per cent loss of the sun mesh and the per cent loss of the ring mesh can be determined by solving Eqs. (14-13) and (14-14) modified for internal gears.

To better understand the above efficiency calculations, and with reference to Fig. 14-35, a sample problem will illustrate its usage.

Given:

10 hp
Input speed, 1,000 rpm
10 diametral pitch
Sun pinion driving
$d_S = 2'' d_R = 8'' d_P = 3'' C = 3.5''$

Solution:

$$\text{Reduction ratio} = 1 + \frac{d_R}{d_S} = 1 + \frac{8}{2} = 5$$

Rpm of driven member = 1,000/5 = 200
Pitch-line velocity of transmitted load

$$v = 0.2618 \times d_S \times n_S$$
$$= 0.2618 \times 2 \times 1{,}000 = 523.6 \text{ fpm}$$

$$W_t = \frac{2 \times T}{d_S}$$

$$W_t = \frac{2 \times 630}{2} = 630 \text{ lb}$$

Actual power transmitted

$$P_a = W_t \times v$$
$$P_a = 630 \times 523.6 = 330{,}000 \text{ ft-lb per min}$$

Equivalent power transmitted by sun gear mesh

$$P_S = W_t \times v_S$$
$$v_S = v \times \frac{8}{8+2} = \frac{d_R}{d_R + d_S}$$
$$v_S = 523.6 \times \frac{8}{(8+2)} = 418 \text{ fpm}$$
$$P_S = 630 \times 418 = 263{,}000 \text{ ft-lb per min}$$

Equivalent power transmitted by ring gear mesh

$$P_R = W_t \times v_R$$
$$P_R = W_t \times v_S = P_S$$
$$P_R = 263{,}000 \text{ ft-lb per min}$$

From Eq. (14-14) and (14-14) modified, the per cent power loss for the sun gear mesh and the ring gear mesh has been determined to be 1.05 and 0.324, respectively. Then the efficiency of the over-all gearbox is

$$\sum = \frac{330{,}000 - (1.05 \times 263{,}000 + 0.324 \times 263{,}000)\frac{1}{100}}{330{,}000}$$
$$\sum = 98.9 \text{ per cent}$$

It is quite evident, however, from studying the problem, that

$$\frac{P_R}{P_a} = \frac{P_S}{P_a} = \frac{(m_R - 1)}{m_R} \tag{14-58}$$

where m_R = over-all ratio

Then Eq. (14-57) can be reduced to the following:

$$E = 100 - (\text{per cent loss sun} + \text{per cent loss ring})\frac{m_R - 1}{m_R} \tag{14-59}$$

Simple Star Gears. A star-gear arrangement consists of the three basic parts of the simple planetary, except that the cage C is stationary and the ring gear is

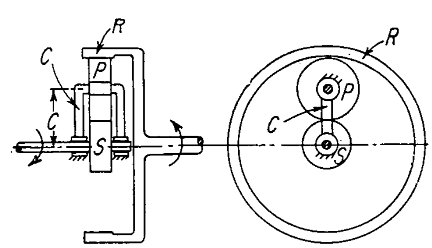


FIG. 14-36. Simple star gear.

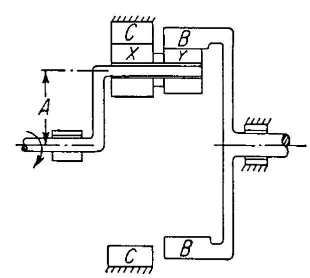


FIG. 14-37. A simple four-gear differential.

allowed to rotate (see Fig. 14-36). From this diagram it is obvious that the driven ring gear R will rotate in the opposite direction from the driver S. Actually, the star gear train is simply an internal gear train with an idler P, which changes the direction of rotation of the driven ring gear R. This being the case, the over-all efficiency can be determined by first calculating the losses of mesh S to P and then the losses of mesh P to R. It must be remembered, however, that the bearing and windage losses of the idler P should be added into the total losses only once.

Differential Gears. A simple four-gear differential gear train is shown in Fig. 14-37. A is the arm through which the input torque must act, and gear B is the output gear. Generally, in practice, the fixed gear C and the rotating gear B are nearly equal in diameter, and this, of course, also applies to both the pinions that must mesh with gears C and B. A differential gear train permits a very high speed reduction with only a few gears and in a minimum amount of space.[18,*] The amount of power that can be transmitted is somewhat limited by virtue of the high gear-tooth loads and high engagement velocities of the mesh. In order to determine the efficiency of a differential drive, the gear-tooth loads on both the fixed and rotating ring gear must be determined as well as the meshing velocities of both members.

The power being transmitted can be calculated as follows:

$$P_a = W_t \times v \tag{14-60}$$

where $W_t = \dfrac{T}{A}$ = load at center pinion, lb
$v = 0.2618 \times 2A \times n$ = velocity of center of pinion, fpm
T = torque input, lb-in.
A = torque arm, in.
n = rpm of input torque arm A

The equivalent power on the fixed ring gear C can be determined as follows:

$$P_C = v_C \times W_C \tag{14-61}$$

where v_C = meshing velocity of ring gear C, fpm
W_C = load applied to ring gear C, lb

$$v_C = V \frac{d_C}{d_C - d_X} \tag{14-62}$$

$$W_C = W_t \times Y \frac{d_C - d_X}{d_C d_Y - d_B d_X} \tag{14-63}$$

where d_X, d_B, d_C and d_Y are pitch diameters of respective parts.

The equivalent power on the rotating ring gear B can be determined as follows:

$$P_B = v_B \times W_B \tag{14-64}$$

where v_B = meshing velocity of ring gear B, fpm
W_B = load applied to ring gear B, lb

$$v_B = v \frac{d_C d_Y}{d_X(d_B - d_Y)} \quad \text{fpm} \tag{14-65}$$

$$W_B = W_t d_X \frac{d_B d_Y}{d_C d_Y - d_B d_X} \quad \text{lb} \tag{14-66}$$

The efficiency of the drive can be determined by calculating the per cent loss for each mesh. The power loss for each mesh is the product of the per cent loss and the equivalent power being transmitted. The over-all efficiency is found by dividing output power by the input power or

$$E = \frac{W_t \times v - (\text{per cent loss}_C \times W_C \times v_C + \text{per cent loss}_B\ W_B v_B)\tfrac{1}{100}}{W \times v} \tag{14-67}$$

Note the per cent loss of mesh C and the per cent loss of mesh B can be calculated by using Eqs. (14-13) and (14-14).

* See p. 128 of reference.

GE-1024.098

A sample problem will illustrate the usage of the above equations for a given four-element differential. In reference to Fig. 14-37, the following information is given:

Output torque = 840,000 lb-in.
Pitch diameter of fixed ring d_C = 31.2″
Pitch diameter of pinion d_X = 8.8
Pitch diameter of rotating ring d_B = 29.2″
Pitch diameter of pinion d_Y = 7.6″
Diametral pitch = 2½
Output rpm = 1 rpm = n
Diameter of the torque arm A = 11.2

Solution: The over-all ratio is found by solving the following:

$$m_R = \frac{1}{1 - (d_C d_Y / d_B d_X)} \tag{14-68}$$

Since the output speed is equal to 1, this is also equal to the input speed of torque arm A.

$$m_R = \frac{1}{1 - (31.2 \times 7.6/29.2 \times 8.8)} = 12.9515 = n$$

Velocity of the center of the pinion, fpm

$$v = 0.2618 \times 2A \times n \tag{14-69}$$
$$v = 0.2618 \times 2 \times 11.2 \times 12.9515 = 75.937 \text{ fpm}$$

Load at the center of pinions

$$W_t = \frac{T}{A} \tag{14-70}$$
$$W_t = \frac{840{,}000/12.9515}{11.2} = 5{,}791 \text{ lb}$$

Input power—or power being transmitted

$$P_a = W_t \times v \tag{14-71}$$
$$P_a = 5{,}791 \times 75.937 = 439{,}753 \text{ ft-lb per min}$$

Meshing velocity of fixed ring gear C

$$v_C = v \frac{d_C}{d_C - d_X} \tag{14-72}$$
$$v_C = 75.937 \frac{31.2}{31.2 - 8.8} = 105.769 \text{ fpm}$$

Load applied to ring gear C

$$W_C = W_t d_Y \frac{d_C - d_X}{D_c d_Y - d_B d_X} \tag{14-73}$$
$$W_C = 5{,}791 \times 7.6 \frac{31.2 - 8.8}{31.2 \times 7.6 - 29.2 \times 8.8} = 49{,}691 \text{ lb}$$

The equivalent power on the fixed ring gear C

$$P_a = W_C v_C \tag{14-74}$$
$$P_a = 105.769 \times 49{,}691 = 5{,}256{,}000 \text{ ft-lb per min}$$

GE-1024.099

Meshing velocity of rotating ring gear B

$$v_B = v \frac{d_C d_Y}{d_X(d_B - d_Y)} \tag{14-75}$$

$$v_B = 75.937 \frac{31.2 \times 7.6}{8.8(29.2 - 7.6)} = 94.729 \text{ fpm}$$

Load applied to rotating ring gear B

$$W_B = W_t \times d_X \frac{d_B - d_Y}{d_C d_Y - d_B d_X} \tag{14-76}$$

$$W_B = 5{,}791 \times 8.8 \frac{29.2 - 7.6}{31.2 \times 7.6 - 29.2 \times 8.8} = 55{,}481 \text{ lb}$$

The equivalent power on rotating ring gear B

$$P_B = W_B \times v_B \tag{14-64}$$

$$P_B = 94.729 \times 55{,}481 = 5{,}256{,}000$$

From Eq. (14-14) modified, the per cent power loss for ring gear mesh C and for ring gear mesh B has been calculated to be 0.503 and 0.601 per cent, respectively. The efficiency for the over-all gear train is

$$E = \frac{439{,}753 - (0.503 \times 5{,}256{,}000 + 0.601 \times 5{,}256{,}000)^1\!/_{100}}{439{,}753}$$

$$E = 86.8 \text{ per cent}$$

It is quite evident, however, from studying the problem that

$$\frac{P_C}{P_a} = \frac{P_B}{P_a} = \frac{m_R - 1}{1} \tag{14-77}$$

where R = over-all differential ratio

Equation (14-67) can be reduced to the following:

$$E = 100 - (\text{per cent loss}_C + \text{per cent loss}_B)(m_R - 1) \tag{14-78}$$

An equation of this nature is quite helpful in determining the over-all efficiency of a given design. It can also be used as a check on the more detailed calculations that must be carried on when a device such as this is designed for a particular application.

Often on a design problem of this nature, a quick approximation of the efficiency which could be determined quite quickly on a slide rule is good enough. An approximate equation for the efficiency of a four-element differential is as shown:[19]

$$E = \frac{1}{1 + \frac{1}{5}[(1/N_X) + (1/N_Y) - (1/N_C) - (1/N_B)](m_R - 1)} \tag{14-79}$$

where R = over-all ratio of train

N_X = number of teeth on gear X

N_Y = number teeth on gear Y

N_C = number teeth on gear C

N_B = number teeth on gear B

The efficiency calculated in this manner for the above-mentioned sample problem would give an approximate efficiency of

$$E = \frac{1}{1 + \frac{1}{5}(\frac{1}{22} + \frac{1}{19} - \frac{1}{78} - \frac{1}{73})(12.9515 - 1)}$$

$$= 85.4 \text{ per cent approximately}$$

This approximation when compared with the efficiency obtained from Eq. (14-67) or (14-78) shows good correlations.

DYNAMIC LOADS ON GEAR TEETH

When gears are run on a test stand or in the field, they are subject to all the forces that can be applied to rotating equipment. In addition to these forces, gears are unique in that they are used in pairs or trains to transmit power and motion, and geometric errors in the gears themselves can cause considerable overloading while the gears are in operation.

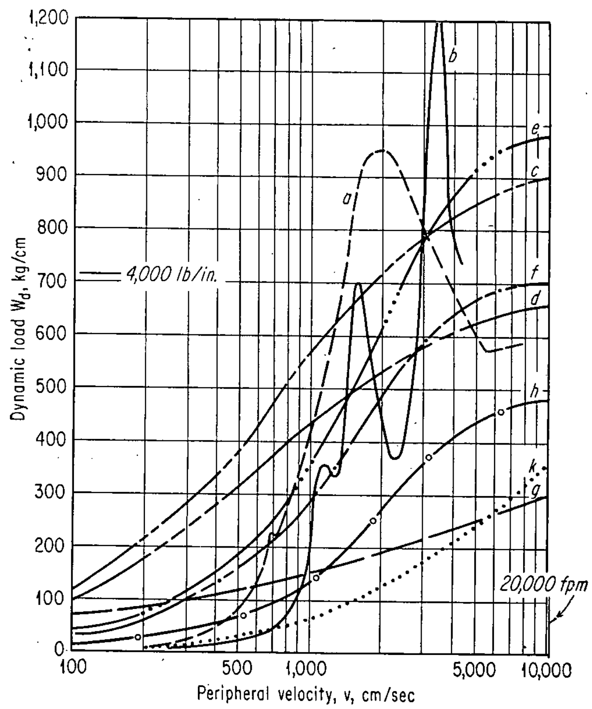
14-5. Dynamic Loads on Spur Gears. When gears are operating under dynamic conditions, there are many factors that affect their over-all operation and performance.[20] These many factors affect the loading conditions on a given gear mesh by combining and producing a critical error in action which in the final analysis may produce gear-tooth overloads in excess of safe operational limits. The dynamic load is affected by such items as:

Elasticity of the gear materials as well as the gear mounting
Torsional deflection and beam bending of the gear shafting
Gear-tooth loading and deflection
Gear-tooth spacing and profile errors
Speed and pitch-line runout of the rotating bodies
Mounting arrangement and alignment
The dynamic balance of rotating elements
Temperature differentials throughout the gearbox
The mass of the gears and shafting
The mass of the connected bodies

The critical error in action produced and maintained by a combination of the above factors is similar in nature to the total error noted on a "red-liner" chart of a gear meshing with a "master" gear. The "red-liner" chart depicts the error in action of a production gear meshing with a very accurate "master" gear with little or no amount of error. Generally, the error shown on a "red-liner" chart is considered to be the result of various manufacturing errors of the production gear. The critical error in action along with the natural vibrations of the geared system cause sudden accelerations of the gears, followed by impacts when the mating gear teeth come back into mesh. This intermittent gear-tooth action sets up forced vibrations in the gear and pinion and does not permit smooth continuous action as would generally be expected from a constant applied load.

Considerable work has been done by various investigations on dynamic loads of gear teeth such as Buckingham, 1931,[20] Buckingham, 1949,[3] Tuplin, 1950,[21] Tuplin, 1953,[22] Strauch, 1953,[23] Reswick, 1954,[24] Attia, 1956,[25] Niemann and Rettig, 1958,[26] Zeman, 1957,[27] Harris, 1957,[28] and Kohler, 1959.[29] In addition, there has been unpublished work done by Caterpillar Tractor Co., the General Electric Company, and undoubtedly other concerns. These many investigators, to say the least, are not in full agreement on how to evaluate the dynamic-load effects. Their work points to the fact that no one best method for determining dynamic loads has been found, and the gear designer is left pretty much on his own. Actually, the existence of dynamic tooth loadings has been known and taken into consideration for some time, except that it has been on a less formal basis. Designers have had to rely on past experience, reduced allowable-stress levels, service safety factors, experimental values applicable to narrow specific fields, etc., in order to put their designs into successful operation. Upon occasion, even the more experienced gear designer finds mself in trouble because he has not taken into consideration the dynamic-loading fect of gears in action.

:-6. Brief Summary of Dynamic-load Investigation. In order for one to take lvantage of what has been accomplished to date in the field of dynamic loads, a iick look at the results of this melting pot is beneficial. Zeman[27] * has done a fair b of summing this up from a practical point of view. Figure 14-38 shows a curve ken from Zeman's paper which has the dynamic load plotted against peripheral



G. 14-38. Dynamic-load comparisons. (*a*) Zeman's formula—single disturbance. (*b*) man's formula—continuous error. (*c*) Buckingham's old formulas, 420 kg. (*d*) Buck-ham's old formulas, 120 kg. (*e*) Buckingham's new formulas, 420 kg. (*f*) Bucking-m's new formulas, 120 kg. (*g*) Niemann's new formula. (*h*) Niemann's old formula. Tuplin's data.

ed for a given set of spur gears. He has calculated the dynamic load using his n method as well as the methods put forth by other investigators for the gear data en in Table 14-5.

Table 14-5

Ratio	1:1
No. of teeth	30
Pitch, diametral	2.54
Center distance	11.8"
Error, magnitude (peak to peak)	0.002"
Material	Steel
Load	420 and 120 kg (921 and 263 lb)

* See pp. 244–254 of reference.

GE-1024.102

One thing is quite obvious. All the curves follow the general trend of increasing dynamic load as the peripheral speed is increased. Using the word loosely, there seems to be some correlation in the results of the various investigators. Most of the results also indicate that a maximum value of dynamic load is reached and then the curves start to level out. Zeman's results[27] seem to peak faster than the rest; however, his maximum values are somewhat in line with Buckingham's. Both Niemann's and Tuplin's results are somewhat lower than Buckingham's. Data from Harris's investigation[28] are not presented in Fig. 14-38. However, Harris states, based on test results, that perhaps the maximum load the gear teeth will see is twice the load that gives zero error in velocity ratio. The load that gives zero error in velocity ratio is the design load with the gear teeth correctly modified. Kohler states[29] in the conclusion of his thesis that his test results revealed, irrespective of error, that the dynamic load never exceeded three times the design load. Again, in reference to Fig. 14-38, the top curve of Buckingham's results shows that the dynamic load is 2.38 times the design load. To further back up this general trend of Buckingham's work, Table 14-6 gives the results of many investigations using the Buckingham method of calculation.

Table 14-6

Design load W	Dynamic load Wd	Ratio Wd/W
4,160	9,770	2.35
5,270	12,195	2.31
625	1,567	2.52
725	1,465	2.02
2,080	4,580	2.20
2,939	8,326	2.84

From a practical point of view, it makes little difference which theory is the proper one, as long as the results obtained are correct, and if they are not correct, they should be at least on the conservative side.

Strictly from a design point of view, the author feels that the Buckingham theory of dynamic loads is the proper approach to use, considering the present state of the art. In reference to Fig. 14-38, it is somewhat on the conservative side. In comparison with Harris and Kohler, loaded gear trains calculated by the Buckingham method seem to take a middle-of-the-road trend. Professor Buckingham was consulted personally, and the results of several lengthy discussions indicated that he felt his method would produce dynamic loads somewhat on the high or conservative side. Although undoubtedly new theories and more physical investigations may reveal that Buckingham's approach is outdated, it seems at the present the most logical and most direct method of assuring the customer that the gear trains being designed, tested, and shipped will work as specified in service.

At this point it must be pointed out that, in special cases, when the design load is quite small compared with the mass effect of the gears and pinions, the connected shafts, as well as the connected bodies, the ratio of calculated dynamic load to design load is much higher[30]—in the range of 8.

The author has some practical experience that will verify this high ratio of dynamic load to design load. For example, in applications where very small amounts of power are being transmitted from a high-speed high-horsepower source (Fig. 14-41, a governor drive), gears often fail from dynamic-load effect. Generally, from a torque-transmitted basis, the gears are very lightly loaded; in fact, often the face widths are sized more from a practical manufacturing point of view instead of the torque being transmitted. When gears of this nature fail, the design engineer is hard pressed for technical reasons to explain the failure. When the problem is analyzed from a dynamic-load-effect approach, usually ratios of 6 to 10 are apparent. This readily explains the premature gear failure.

14-7. When Dynamic Loads Should Be Considered. Actually the dynamic load should be determined for each and every new gear application. This way the design engineer will have a backlog of information with which to compare a given dynamic-load calculation. If a diligent procedure is set up whereby the apparatus is calculated in the design stage and observed in service in the field, and maintenance and

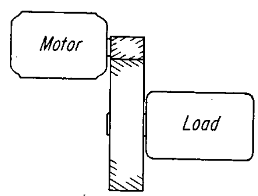


FIG. 14-39. Close-coupled stiff shafting.

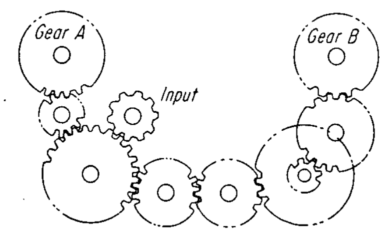


FIG. 14-40. Accessory gearbox.

failure records are carefully analyzed, the dynamic-load effect will soon have as much meaning for the average gear designer as does the transmitted load.

There are, of course, many well-established applications where it is not necessary that dynamic-load calculations be made, since past experience has already established the amount of dynamic load present. There are, however, several types of gear arrangements where it is quite important that very careful consideration be given to the dynamic-load effects during the design phase. In these arrangements, the omission of a dynamic-load evaluation would be termed "poor design practice."

Figure 14-39 shows a very close coupled gear train. In this arrangement the shafts that join the mass of the rotor to the pinion and the mass of the load to the gear are very heavy and stiff; consequently, a considerable amount of this connected mass will influence the gear mesh. A dynamic-load calculation will determine the magnitude of this mass effect as well as perhaps point out ways in which this effect could be reduced.

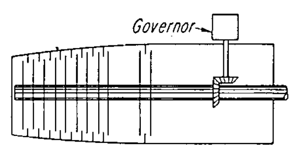


FIG. 14-41. Power take-off, light loads.

Figure 14-40 shows a gear train of an aircraft accessory gearbox. Because of the arrangement, the mass effect of each and every gear has an additive effect on the last gear in the train. If the design of this box is not carefully analyzed for dynamic-load effects, the whole arrangement could fail on test, particularly the two end gears A and B.

When very light loads are driven from a high-speed transmission as outlined in Fig. 14-41, the gearing is subjected to some of the worst dynamic-load conditions. Generally the high-speed power source has its own vibration characteristics because of its speed and relative high mass. The take-off gears are quite small because of the very light transmitted load. Usually, when this arrangement is calculated, the dynamic load is considerably larger than the rather small transmitted load, perhaps something in the order of 8 to 10.

Often it is desirable to have a multi-input drive whereby the prime movers feed into a gear train which may drive several loads but in the final analysis are crosslinked to permit operation or sharing of the load while one prime mover is down for repairs. Such an arrangement is shown in Fig. 14-42. From a dynamic-load point of view, this arrangement is quite risky and should be given every consideration.

Another very tough and difficult problem is that of a high-speed quill drive where, for example (see Fig. 14-43), a high-pressure lube pump is driven through a quill from a relatively high-speed high-power gear train. The quill is intended to compensate for misalignment due to temperature effects and the pinion shifting due to applied load. In many applications, the quill will lock up on one end and the other

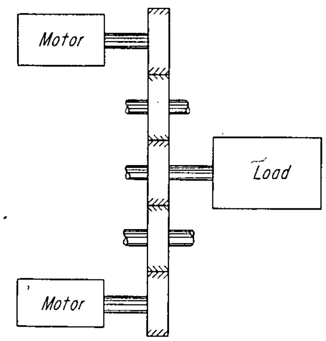


FIG. 14-42. Multiple power inputs.

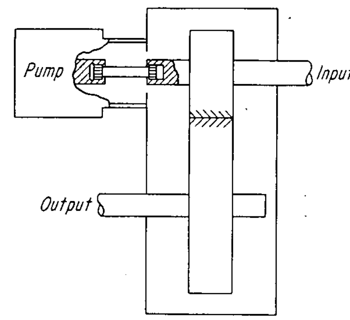


FIG. 14-43. High-speed quill drives.

end will take a beating because of a combination of mass effects of the main drive gears and wear due to misalignment.

14-8. Buckingham's[3,*] Solution to the Dynamic-load Effect on Gear Teeth. The load that causes the driven gear to move forward or ahead to make up for a distance lost due to error in a preceding mesh is known as the "acceleration load." The resultant impact when the teeth come together again in the mesh is known as the "dynamic load." Actually, there are two load surges for every tooth engagement; the "acceleration load," which is set up by the first phase of engagement, followed by the impact or "dynamic load," which is really the reaction. The impact load or "dynamic load" is always the maximum load value of the tooth engagement.

The acceleration load is a function of f_1, the force needed to accelerate rigid bodies through the critical error, and f_2, the force required to do the work of deforming the tooth by elastic deformation equal to the amount of the critical mesh error. The acceleration load f_a then is the resultant of the two where each does its part during the meshing cycle:

$$\frac{1}{f_a} = \frac{1}{f_1} + \frac{1}{f_2} \tag{14-80}$$

The dynamic load is then expressed as a function of the acceleration load f_a and f_2:

$$W_d = W_t + \sqrt{f_a(2f_2 - f_a)} \tag{14-81}$$

where W_d = dynamic load, lb
W_t = transmitted load, lb
f_a = acceleration load, lb
f_2 = force required to do the work of elastic deformation, lb

* See pp. 426–452 of reference.

In order to make a dynamic-load calculation using the Buckingham method for a given gearset, some quantities must be known and some preliminary calculation must be made:

M_p = pinion mass effect at pitch radius of pinion, lb-sec² per ft
M_{a1} = mass effect of bodies connected to pinion at pitch radius of pinion, lb-sec² per ft
M_G = gear mass effect at radius of gear, lb-sec² per ft
M_{a2} = mass effect of bodies connected to gear at pitch radius of gear, lb-sec² per ft
Z_1 = elasticity of pinion shafting, lb per in.
Z_2 = elasticity of gear shafting, lb per in.
e = effective error assumed or known, in.
E_1 = modulus of elasticity of pinion
E_2 = modulus of elasticity of gear

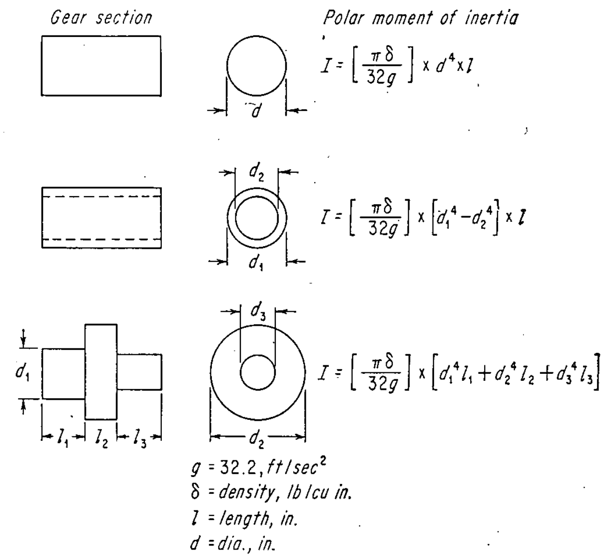


FIG. 14-44. Method used to calculate polar moment of inertia.

The above items can be determined in the following manner. To simplify the determination of moments of inertia for various shapes of gears, refer to Fig. 14-44.

$$M_p = \frac{I}{R_1^2} \tag{14-82}$$

where I = polar moment of inertia of pinion, lb-in.² sec² per ft
R_1 = pitch radius of pinion, in.

$$M_{a1} = \frac{I}{R_1^2} \tag{14-83}$$

where I = polar moment of inertia of connected bodies to pinion shaft, lb-in.² sec² per ft
R_1 = pitch radius of pinion, in.

$$M_G = \frac{I}{R_2^2} \tag{14-84}$$

where I = polar moment of inertia of gear, lb-in.^{2} sec^2 per ft
R_2 = pitch radius of gear

$$M_{a2} = \frac{I}{R_2^2} \tag{14-85}$$

where I = polar moment of inertia of connected bodies to gear shaft, lb-in.^{2} sec^2 per ft
R_2 = pitch radius of gear, in.

$$Z_1 = \frac{K_1}{R_1^2} \tag{14-86}$$

where K_1 = stiffness of shaft—pinion, lb-in.
R_1 = pitch radius of pinion, in.

$$Z_2 = \frac{K_2}{R_2^2} \tag{14-87}$$

where K_2 = stiffness of shaft—gear, lb-in.
R_2 = pitch radius of gear, in.

$$K_1 \text{ or } K_2 = \frac{\pi G}{32} \times \frac{1}{\Sigma l/(d_1^4 - d_2^4)} \tag{14-88}$$

where G = shear modulus = 11.8×10^6 for steel
l = any length, in.
d_1 = outside diameter, any shaft
d_2 = inside diameter, any shaft

In high-speed apparatus, there is a limited time in which the meshing error is permitted to act. When this critical speed is reached, the dynamic load will maximize and then decrease from this point with increasing speeds. Often it is convenient to check this critical speed for your particular application to see if your design permits time for the full error in action to take place.

$$n_c = \frac{30}{N_P}\sqrt{\frac{6W_t}{eM}} \tag{14-89}$$

where n_c = rpm of pinion where dynamic load is maximum
N_P = number of teeth in pinion
W_t = transmittal load, lb
e = error in action, in.
M = mass effect at pitch line of gears

If N_c is smaller than the design speed of the apparatus under question, then the dynamic load should be determined with a new "effective error" that is less than the original assumed error. The effective error can be determined as follows:

$$e^1 = \frac{6Wt^2}{M} \tag{14-90}$$

where e^1 = effective error
W = transmittal load
M = mass effect at pitch line of gears
t = time, sec, = $\dfrac{30}{nN_1}$
N_P = number of teeth in pinion
n = rpm of pinion

Table 14-7. Calculation of Dynamic Load by Dudley Method

1.	Design hp	865 given
2.	No. of pinion teeth N_P	37 given
3.	No. of gear teeth N_G	309 given
4.	Pitch dia. of pinion d_P	3.700 given
5.	Pitch dia. of gear D_G	30.900 given
6.	Face width F	$8\frac{1}{8}$ given
7.	Pinion rpm n_P	10,022 given
8.	Pitch-line velocity, $0.2618 \times (4) \times (7)$	9,707.9
9.	$(8) \times (8)$	94.244×10^6
10.	Pinion torque $= 63{,}000 \times (1) \div (7)$	5,437.5
11.	tan driving load $W_t = 2.00 \times (10) \div (4)$	2,939.2
12.	Helix angle ψ	23° given
13.	$\cos (12) \times \cos (12)$	0.84732
14.	Constant C_1 (read Fig. 14-45)	9.094
15.	Constant C_2 (read Fig. 14-45)	0.00113
16.	Mass effect of pinion M_P	0.550 [Eq. (14-82)] given
17.	Mass effect of body connected to pinion M_{a1}	117.3 [Eq. (14-83)] given
18.	Mass effect of gear M_g	11.000 [Eq. (14-84)] given
19.	Mass effect of body connected to gear M_{a2}	8.9042 [Eq. (14-85)] given
20.	Elasticity of pinion shaft z_1	0.20×10^6 [Eq. (14-86)] given
21.	Elasticity of gear shaft z_2	9.3×10^4 [Eq. (14-87)] given
22.	Effective error e	0.00025 (assume value)
23.	Modulus of elasticity of pinion E_1	30×10^6
24.	Modulus of elasticity of gear E_2	30×10^6
25.	$1.0 \div (4) + 1.0 \div (5)$	0.302632
26.	$2.0 \times (15) \times (25)$	0.000684
27.	$(26) \times (17) \times (9) \times (13)$	6,406,992
28.	$[(16) + (M_2)]\,(27)$	74,000,758 [assume $M_2 = (18)$]
29.	$M_2 \times (22) \times (20)$	550.0 [assume $M_2 = (18)$]
30.	$(28) + (29)$	74,001,308
31.	$M_2 \times (22) \times (17) \times (20)$	64,515 [assume $M_2 = (18)$]
32.	$\sqrt{(30)^2 + 4.0(27) \times (31)}$	74,012,479
33.	$(32) - (30)$	11,171
34.	$(33) \div 2.0(27)$	0.000871
35.	$(16) + (34)$	0.550871
36.	$(26) \times (19) \times (9) \times (13)$	486,316
37.	$(18) + (35) \times (36)$	5,617,373
38.	$(35) \times (22) \times (21)$	12.8077
39.	$(37) + (38)$	5,617,386
40.	$(22) \times (19) \times (35) \times (21)$	114.0180
41.	$\sqrt{(39)^2 + 4.0(36) \times (40)}$	5,617,406
42.	$(41) - (39)$	20
43.	$(42) \div 2.0 \times (36)$	0.000020
44.	$M_2 = (18) + (43)$	11.000020

Check how close $M_2 = M_g$; (44) = (18). If necessary substitute calculated M_2 back into (28), (29), and (31) and recalculate until assumed and calculated values of M_2 check to first five places.

45.	$(35) \times (44)$	6.059592
46.	$(35) + (44)$	11.550891
47.	$(45) \div (46)$	0.524599
48.	$(47) \times (26) \times (9) \times (13)$	28,668
49.	$1.0 \div (23) + 1.0 \div (24)$	666.7×10^{-10}
50.	$(49) \times (14)$	0.606292×10^{-6}
51.	$(50) \times (11) \div (6)$	0.000219
52.	$(51) \div (13)$	0.000258
53.	$(22) \div (52) + 1.0$	1.968992
54.	$(11) \times (53)$	5,787.2613
55.	$(48) \times (54)$	165.90921×10^6
56.	$(48) + (54)$	34,455.26
57.	$(55) \div (56)$	4,815.9057
58.	$2.0 \times (54) - (57)$	6,758.6169
59.	$(57) \times (58)$	32.54886×10^6

Table 14-7. Calculation of Dynamic Load by Dudley Method *(Continued)*

60. $\sqrt{(59)}$	5,705.161
61. $W_d = (11) + (60)$	8,644

Item 61 is the dynamic load which will occur unless the gear is running so fast that the full reaction cannot occur.

62. $5{,}400 \times (11)$	15.8717×10^6
63. $(7) \times (7) \times (2) \times (2) \times (47)$	7.1818×10^{10}
64. $e = (62) \div (63)$	0.000221
65. $(64) \div (52) + 1.0$	1.856589
66. $(65) \times (11)$	5,456.886
67. $(48) \times (66)$	156.4380×10^6
68. $(48) + (66)$	34,124.886
69. $(67) \div (68)$	4,584.2791
70. $2.0 \times (66) - (69)$	6,329.4928
71. $(69) + (70)$	29.01616×10^6
72. $\sqrt{(71)}$	5,386.67
73. $W_d = (11) + (72)$	8,326

Item 73 is the maximum dynamic load that can occur in the time while a tooth goes through mesh. Item 64 is the maximum error that can be effective in creating dynamic load. Use whichever is smaller, item 61 or item 73, as the calculated dynamic load.

Once this information has been collected, the actual calculation of the dynamic load can get under way. Space will not permit the repeat of the many and various steps in making a calculation as outlined by Buckingham;[3] however, Dudley[1,*] in "Practical Gear Design" has a short-cut procedure for making the final dynamic-load calculation for a single pinion-and-gear combination, either spur or helical. Table 14-7 gives this method step by step. In order to use the Dudley[1] short-cut method, the deflection constants C_1 and the acceleration constants C_2 found in Fig. 14-45 are required.

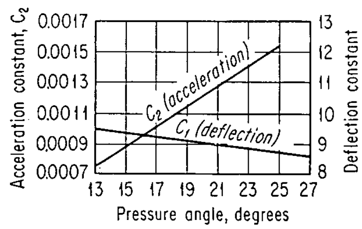


FIG. 14-45. Constant curves.

14-9. Dynamic-load Effect on Gear Trains. The analysis of trains of gears is very similar to that of a single pair of gears, except that, when a given mesh is under consideration, the mass effect of the other gears in the train must be taken into account.

Simple Train. To better understand how a simple train of gears is handled, refer to the diagram of Fig. 14-46.

Let I_1 = polar moment of inertia of driving member—pinion
I_2 = polar moment of inertia of idler
I_3 = polar moment of inertia of driven gear

For mesh—pinion to idler,

$$M_p = \frac{I_1}{R_1^2} \tag{14-91}$$

$$M_G = \frac{I_2}{R_2^2} + \frac{I_3}{R_3^2} \tag{14-92}$$

For mesh—idler to gear,

$$M_p = \frac{I_1}{R_1^2} + \frac{I_2}{R_2^2} \tag{14-93}$$

$$M_G = \frac{I_3}{R_3^2} \tag{14-94}$$

* See pp. 329–331 of reference.

The dynamic load would then be determined for each mesh by substituting the appropriate values of M_G and M_P into the calculation sheet of Table 14-7.

Compound Trains. An example of a compound train[30] is shown in Fig. 14-47.

Let I_1 = polar moment of inertia of driving pinion
$I_{2,3}$ = polar moment of inertia of cluster gear
I_4 = polar moment of inertia of driven gear

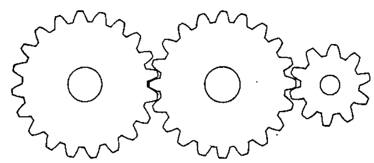


FIG. 14-46. Simple train.

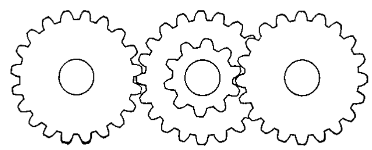


FIG. 14-47. Compound train.

For mesh—pinion to cluster gear,

$$M_p = \frac{I_1}{R_1^2} \tag{14-95}$$

$$M_G = \frac{I_{2,3}}{R_2^2} + \frac{R_3^2}{R_2^2} \times \frac{I_4}{R_4^2} \tag{14-96}$$

For mesh—cluster gear to driven gear,

$$M_p = \frac{I_{2,3}}{R_3^2} + \frac{R_2^2}{R_3^2} \times \frac{I_1}{R_1^2} \tag{14-97}$$

$$M_G = \frac{I_4}{R_4^2} \tag{14-98}$$

This same procedure can be used for more complex gear trains. In the final analysis, however, each mesh must be calculated using the appropriate values of M_G and M_P in the calculation sheet of Table 14-7.

GEAR PERFORMANCE

Gear trains or gear drives to be considered as satisfactory must meet a standard of performance as interpreted by the ultimate customer. Actually the desired performance of a given gear drive depends to a large measure on the end use of the apparatus. A good-performance gear means different things to different customers. For example, a good-performance gear to a marine engineer means extreme reliability. He envisions the difficulty of cutting through the decks and compartments of ships to replace faulty gear equipment. To a submarine captain, a good-performance gear should be ultraquiet. This would give a measure of protection against detection by the enemy. A fighter pilot wants a high horsepower-to-weight ratio to give him a speed advantage, yet also he wants a degree of reliability to prevent catastrophic failures. A controls engineer wants extreme accuracy in transmitted motion, since a few thousandths in backlash may cause him to miss the target area.

14-10. Factors That Affect the Performance of Gear Units. A good-performance gear should be described as one that operates as well as or even better than the prediction of the design engineer. From an engineering point of view, however, there are several basic fundamental criteria for judging equipment of this nature such as the efficiency of the geared drive, the reliability as compared with the design life, or the size and weight of the drive vs. the horsepower transmitted, and the over-all job that the gear must do. Table 14-8 shows a very interesting and revealing study of various types of gear drives. It must be remembered, however, that this table must be used only as a rough guide for ball-park comparisons.

Table 14-8. Relative Performance of Gear Drives

Type of gear transmission	Relative wt. per total job done	Hp per lb	Input speed	Over-all ratio
Marine	105	0.075	6,500	64
Navy	36	0.5	8,000	35
Industrial	160	1.0	8,000	4
Industrial (lightweight)	32	2.0	19,000	16
Railway	74	0.875	2,100	4.1
Aircraft (power)	16	12–18	17,000	13
Rockets and missile	12	25–50	30,000	5

From a classical point of view, perhaps the ultimate in gear performance should be synonymous with the ultimate in gear capacity. The best-performing gear drive would then be the drive that could carry the most horsepower with the least weight. This being the case, in reference to Table 14-8, the aircraft and rocket geared systems would be way out front. There are several factors other than the transmitted load and the speed of operation that affect the over-all performance of gear applications such as:

Power Loss. The amount of heat being dissipated within the gearbox is a good indication of how the over-all gearbox is operating. This known heat loss can then be compared with the calculated power loss of the unit under consideration.

The calculated power loss should be greater than the measured power loss. If this is not the case, then the gear unit must be carefully investigated. Often when there is an excessive heat loss within the gearbox, the unit is not scavenging properly and the design may be of the questionable type, as indicated by Figs. 14-27, 14-28, and 14-29. Oil* that is not immediately removed from high-speed gear units will store up and cause considerable heat loss because of churning of the oil. In many cases, improper scavenging will cause a hydraulic-braking effect, which sometimes will overload the gear unit to the point of failure. At times, hot bearings will cause an increase in the temperature differential across the gear casing, and often this is an indication of incipient failure. When a gear unit is first started up, however, it is difficult to tell if the bearings are running hot or not, unless the unit is fully instrumented. If the unit has been previously run, then it is quite obvious when the temperature differential increases.

Vibrations. In order for a gearbox to operate satisfactorily, it must run within safe vibration limits. Normally, if the vibrations of a given gearbox run more than 5 mils under steady-state conditions, you should suspect that something is wrong. Several things can cause undesirable vibrations in a gearbox. Dynamic balance of the rotating elements should be first on the list of suspects. All rotating elements should be checked balanced as soon as trouble is encountered. Next, consideration should be given to connecting bodies, such as couplings and shafting. All shafting should be rechecked for critical-speed indications; this includes the gearing within the gearbox as well as the outside shafting. Often a flexible coupling will stick in an unbalanced position and then cause considerable vibration when the unit is up to speed. Runout on all bearing surfaces and the pitch line of the gears should be checked to see if they are within drawing limits. Sometimes the face of the coupling

* See Chap. 15 for detail recommendations on gear-lubrication systems.

flanges will be out of square with respect to the rotating axis, causing bolted-on equipment to run somewhat out of balance. Profile errors of the gear-tooth elements often can cause the mesh to run rough, which in turn can set up bad vibrations. Sometimes a bad rolling-contact bearing which has been damaged at assembly can cause a gear unit to vibrate and, of course, this usually leads to premature failures. Another possible cause of rough running is a bent or damaged gear tooth. This sometimes happens at assembly and it is usually quite difficult to detect unless the gears are pin-checked.

Alignment and Mounting.* Proper or improper alignment can affect the operation of a gearbox in several ways.[31] For example, if the external alignment of the gearbox to the connected load is not made carefully, the external coupling may fail because of excessive heat generation in the coupling engagement. If this happens, usually the coupling becomes quite "sticky" and then transmits bending moments back into the gearbox. If the unit continues to run, the shaft extension to which the coupling is attached often fails back near the first firm support area, which is usually the bearing just within the gearbox. Shafts have been known to fail in this manner where the shaft-extension loading normally would be considered quite conservative. Internal alignment is also very important. The mounting arrangement plays an important part in determining the degree of alignment which will ultimately exist within the gearbox. Sometimes at manufacturing, the casing bores are not machined true with respect to each other or parallel with respect to the set of bores in which the mating gear must run. This obvious mismatch of bores causes alignment problems in the mesh which are often compounded by additional misalignment due to internal clearance in the bearings. The alignment of the mesh is also affected by shaft deflections and gear blank deflections, as well as casing distortions. If the pinion is wide-faced with a high length-to-diameter ratio, torsional windup of the pinion can also contribute to the alignment condition of the mesh. It makes little difference what causes the misalignment in the mesh. The net result is a load concentration on one end or the other of the mesh, which usually results in premature failures of the gear or pinion teeth by pitting or scoring or breakage, or a combination of all three. Generally, misalignment conditions within a gearbox must be eliminated if at all possible. If elimination is not possible, it must be corrected for by such means as crowning, dubbing off the ends of the teeth, or helix-angle correction. If the mismatch is such that the ends of the teeth are being forced into tight mesh, perhaps additional backlash is required as well as some generous profile modification.

Lubrication. In order for a gearbox to operate at all, it must be supplied with an adequate lubricant. The lubricant permits the bearings, as well as the gear mesh, to operate with a minimum of power loss.[32] In addition to the job of lubricating these rubbing surfaces to cut down on friction, the lubricant usually acts as a cooling medium to keep the rubbing parts within satisfactory temperature limits.[33] Often in high-load high-speed applications, the lubricant also acts as a carrying medium for antiwear and antiscoring agents. On wide-face-width gearing, it is very important that the distribution of the oil across the mesh is uniform to eliminate any uneven heating effects. The lubricant also helps in eliminating temperature differential throughout the gear-casing assembly. It tends to wash out high-temperature areas and heat up low-temperature areas.

Normally, oil is thought of as a necessary and beneficial fluid; however, at times it can be very troublesome and detrimental. If oil is allowed to fill up within the gearbox, it will cause excessive power loss and ultimate failure due to the braking effect. Because oil is a fluid, it often causes undue concern with respect to seal leakage and proper ventilation of the gearbox. If the complete lubrication system has not been

* See Chap. 12 for detail recommendations on gear-mounting practice.

carefully designed with due regard to oil turnover and proper settling-out time, considerable trouble can be encountered with respect to oil foaming. Some antifoaming agents can be utilized in the oil; however, a healthy solution to the problem is to allow plenty of oil for the over-all gear box with adequate scavenge capacity and allow the foam to settle out in the storage tank. Oil foaming can also be a problem at reduced pressures.[35] This trouble is encountered particularly at high altitudes, and it makes the use of a circulation oil system very difficult (see Fig. 14-48).[35] At high-altitude operation where a recirculation system is required, design engineers have had to pressurize the complete lube system. Because the gearbox depends on the lube oil for its very existence, the supply of oil must not be interrupted. Often a catastrophic gear failure can be traced back to a lube-pump failure or a lube-line break or even to plain carelessness on the part of an operator in failing to turn on the lube system in the proper manner.

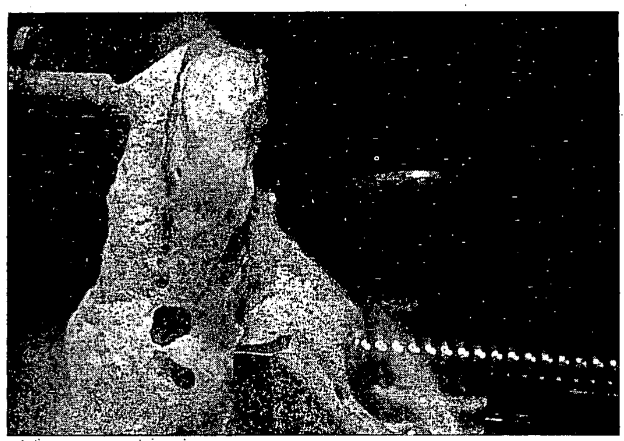


FIG. 14-48. Oil foaming from a nozzle. (*From Hartman, M. A., Gears for Outer Space, Machinery, August,* 1959, *pp.* 115–121.)

Environmental Conditions. Gears must operate in many different environments and under various adverse conditions. Gears have been designed, to name a few, to operate under water, water lubricated, poorly lubricated, and not lubricated, in high-temperature fields,[34] in low-pressure atmospheres,[35] in contaminated atmospheres of all kinds, and under a variety of high speed and high load conditions. The environments in which gears must operate are not only important from the point of view of the harm that the environment does to the gearbox, but also from the viewpoint of what the gear may do to the specific areas or atmosphere in which they are working. For example, a gear must be extremely quiet if it is located in an area where people must rest or sleep. A gear that may contaminate food will be required to operate without lubricants. Radio waves emitted by gears may interfere with various control equipment. Oil vapors escaping from geared equipment may be objectionable to personnel who must work in the immediate vicinity.

It is not the purpose of the author to go into the details of how to cope with these many environmental conditions; however, the design engineer must be aware of these conditions since, in most cases, they will affect the basic concept of his whole gear design.

14-11. Gear Failures. Gears fail in service from such causes as pitting of gear teeth, breakage of the gear teeth, long-range wear, plastic deformation, scoring, and other types of destructive wear not so common. The AGMA standard 110.03,[36] "Nomenclature of Gear-tooth Wear and Failure," breaks down all types of gear failures into four general classes:

1. Wear
2. Plastic flow
3. Surface fatigue
4. Breakage

Each class is then divided into several forms and types with definitions. It is highly recommended that design engineers obtain copies of this standard for handy reference.

A good gear designer needs to know how to recognize and analyze gear failures. In order for past experience to be of any benefit, failures must be properly investigated and appropriate corrective action taken.

Wear. Wear is a very important phenomenon, particularly in high-speed gearing that must be in operation for an unlimited length of time. Wear has been defined from a gear engineer's point of view as that kind of tooth damage whereby layers of metal have been more or less uniformly removed from the surface.[1,*] The most common causes of gear-tooth wear are metal-to-metal contact due to an inadequate oil film, abrasive particles in the oil supply, failure of the oil film in the contact area causing rapid wear or scoring, and chemical wear due to the composition of the oil and its additives.[33]

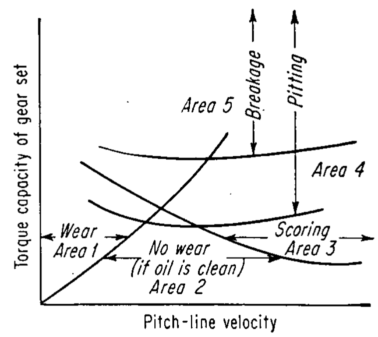


Fig. 14-49. Regions of gear failures.

Figure 14-49 shows a general plot of pitch-line velocity vs. torque capacity of a gearset.[1,†] The various areas indicate the type and regions where gear failures are most likely to occur. Areas 1, 2, and 3 are the failure regions associated with the wear phenomenon. In area 1, the gear is not running fast enough to develop an oil film. In area 2, the speed is fast enough to develop a film and the gear will run an indefinite period provided the lubricant is free of foreign material and is noncorrosive. In area 3, rapid wear or scoring will take place since the load and speed are high enough to break down the existing oil film. Ideally, of course, gear designs should be such that they operate wholly within area 2.

It is a general practice in the gear trade, and a good one, to break in gearboxes and gradually build up to full load and full speed over a period of time. This procedure permits the contacting surfaces to polish in. Normal wear or polishing in is actually a very slow wear process. However, the amount of metal removed should be well within the design life of the equipment. Figure 14-50 shows an excellent example of a well-polished spiral bevel gear. This gear has seen many cycles of operation, and although some metal has been removed, the gear is in excellent condition.

Figure 14-51 shows a helical gear with a moderate amount of wear. The pitch line is clearly visible, indicating some high metal. Unless the load is reduced or the lubrication improved this gear will ultimately fail from pitch-line pitting.

Abrasive wear can be of considerable concern to the successful operation of a gear train. It is only logical that abrasive particles must be kept from the lube system.

* See p. 294 of reference.
† See p. 295 of reference.

However, at times, this is a very difficult task to perform. Abrasive wear is self-perpetuating in that a few particles can remove metal from the tooth surfaces, which in turn act as a lapping compound which tends to increase the rate of metal removal.

Figure 14-52 shows what can happen if abrasive wear is permitted to continue. Note from Fig. 14-52 that approximately one-third of the tooth thickness has been removed and yet the tooth surface still shows deep radial grooves in the direction of sliding.

Scoring is very rapid wear of the tooth surfaces initiated by virtue of the fact that the oil film has failed, permitting metal-to-metal contact.[33] This intimate contact

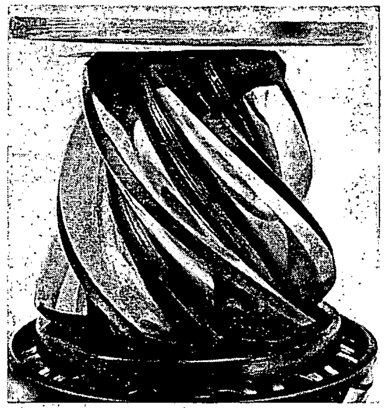


FIG. 14-50. Polishing in spiral bevels.

FIG. 14-51. Moderate wear.

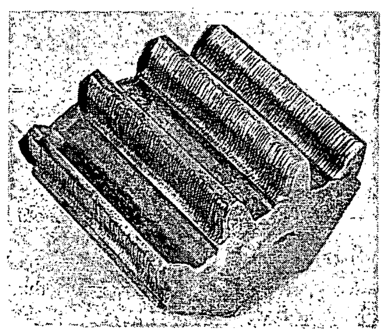


FIG. 14-52. Abrasive wear.

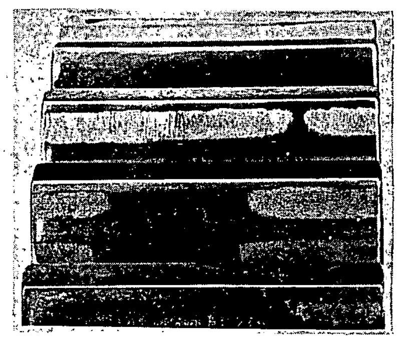


FIG. 14-53. Moderate scoring.

produces a welding and tearing action which removes metal rapidly from the tooth surfaces. Although scoring is triggered off by a lubrication failure, there are many factors other than the oil that influence the final scoring resistance of the gear mesh such as tooth-surface pressure, properties of the material, surface finish, surface treatments, and surface rubbing velocities. Although scoring failures are quite easy to recognize, they are quite hard to analyze. Scoring can be broken down into several categories, which often helps to ferret out the real cause of the failure. The several types of scorings are as follows: *initial* or *slight scoring, moderate scoring, severe scoring, misalignment scoring, scoring due to load concentrations.* Initial or slight scoring usually tends to heal itself unless the load or speed of the apparatus is increased.

Often scoring of this nature occurs if the gearbox has not been properly broken in. Figure 14-53 shows a typical case of moderate scoring. Note that there are many areas on the tooth surfaces that have not been detrimentally affected. Gear of this nature could be put back into service if something is done to improve the scoring conditions. For example, reduce the transmitted load slightly, increase the viscosity of the lubricant, decrease the speed of operation, or add antiscoring additives to the

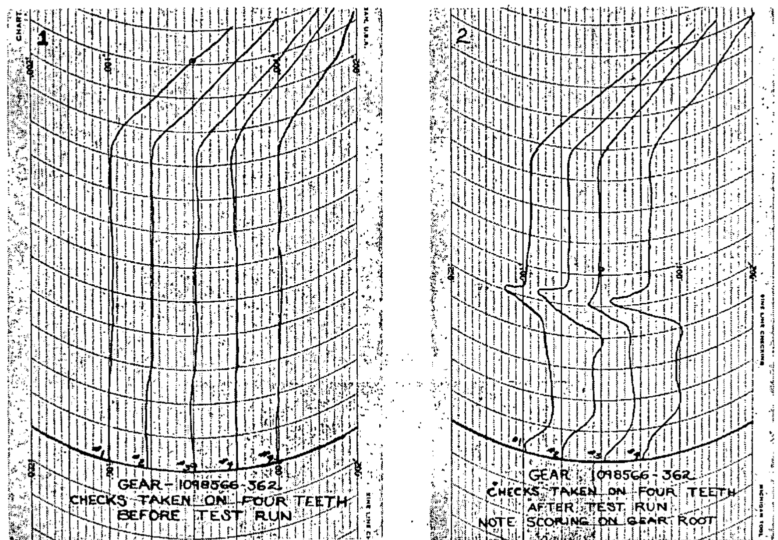


FIG. 14-54. Scoring failure—involute charts.

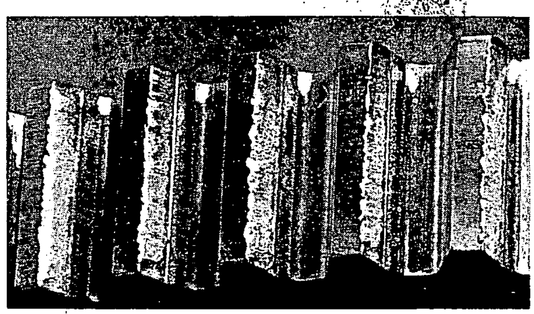


FIG. 14-55. Severe scoring.

lubricant. Figure 14-54 shows a chart of the involute profile of a gear before and after a scoring failure. The amount of profile destruction is quite evident. It is also quite interesting to note that material has been scooped out of the dedendum and built up on the pitch line of the gear. This scoring failure appeared slightly worse than Fig. 14-53 but not anywhere near so severe as that of Fig. 14-55, which depicts a really bad case of severe scoring. Observe from Fig. 14-55 the welding and tearing action that has been in progress, particularly the metal that has been dragged radially over the tips of the gear. Often the transmitted load is not high enough to cause scoring in the mesh. However, if the gears become misaligned because of deflections or manufacturing errors, the load may concentrate on the ends of the gear teeth. This concentrated load may now be high enough to break down the oil film and permit end scoring or misalignment scoring. Figure 14-56 is a good example of this type of failure. A failure of this nature at times is a blessing in that usually scoring shows up early, often on the test stand, indicating a misaligned condition. The engineer,

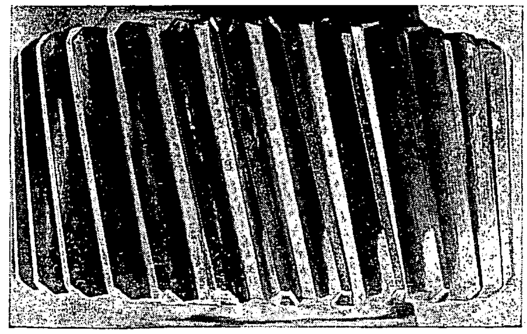


Fig. 14-56. Misalignment scoring.

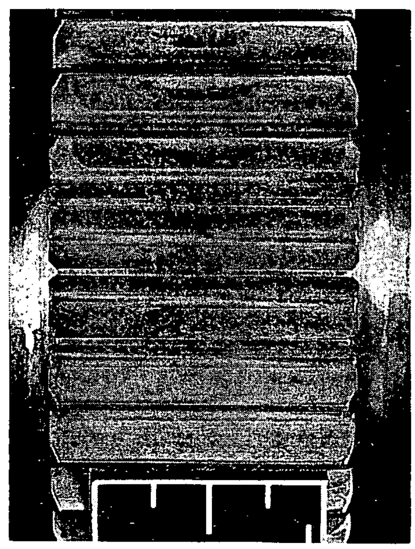


Fig. 14-57. Load-concentration failure.

realizing this condition exists, can do something about it before the unit is put into service. If a misaligned condition goes undetected, trouble usually shows up many hours later in the form of broken ends of the teeth or at best heavy pitting on the ends of the teeth.

Generally, misaligned conditions are corrected by helix-angle corrections or by relieving the ends of the teeth from contact or, if conditions warrant, putting a slight crown on the tooth surface in the axial direction. Crowning must be done with extreme care and careful analysis of the over-all situation since crowning in effect concentrates the load in the center of the gear teeth and can cause a load-concentration failure, as indicated in Fig. 14-57.

Plastic Flow. Gears fail from plastic flow when the surface yields and deforms under heavy loads. Usually this occurs with soft and medium-hard materials, but it can happen on harder gear-tooth surfaces. Generally this plastic flow is characterized by finned material overhanging the tips of gear teeth as indicated by Fig. 14-55. Actually Fig. 14-55 started out as a scoring failure, but because of the destruction of the profile, the load increase due to dynamic effects caused, in addition to the severe scoring, considerable plastic flow of the rolling and peening type.

Rippling and *ridging* are also failures associated with plastic flow. Generally rippling shows up as a wavy surface which may or may not lead to ultimate failure. It generally can be improved by extreme-pressure additives in the oil and better oil distribution. Decreasing the transmitted load of course helps, but this cannot always

FIG. 14-58. Ridging. (*Courtesy of Socony Mobil Oil Co., New York.*)

be accomplished. Ridging is quite similar in nature to rippling; however, instead of just a surface imperfection rather deep ridges appear and by their very nature are more detrimental to the gear surface than rippling. Ridging usually shows up on worm-gears and hypoid gears (see Fig. 14-58). It is the author's belief that ridging failures are quite closely associated with scoring failures, but because of the high rubbing velocity, a considerable amount of polishing takes place in the mesh, and the net result is ridging.

Surface Fatigue. Surface fatigue, or pitting as it is normally called, is a surface failure due to exceeding the endurance limit of the surface material. Gears under load produce repeated surface and subsurface stresses, and if the loads are high enough and the stress cycle repeated often enough, chunks of metal will fatigue from the surface producing pits. Pitting can be characterized by three different types such as *initial pitting* or *corrective pitting*, *destructive pitting*, and *spalling*. Generally, pits of the corrective-pitting type are quite small in diameter in the range of $\frac{1}{64}$ to $\frac{1}{32}''$ in diameter. They occur in localized overstressed areas and tend to distribute the load by progressively removing high contact spots. When the load is more evenly distributed, the pitting action is reduced and eventually polishes over. Destructive pitting is more severe and the pits are almost always much larger in size. Figure 14-59 shows a typical pinion with destructive pitting. Pitting of this nature is always fatal in that the profile is destroyed beyond any hope of recovery and, if the transmitted load remains the same, the pitting will get progressively worse until the gear unit runs so rough it must be taken out of service or the gear teeth break off. Spalling is practically the same as destructive pitting except the term is more commonly used when the pits are quite wide in diameter and cover a considerable area. The spiral bevel pinion shown in Fig. 14-60 is an excellent example of spalling. Sometimes

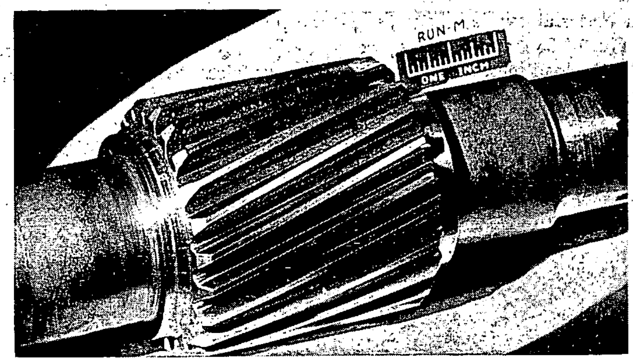


FIG. 14-59. Destructive pitting.

spalling occurs when the pits originate in a casehardened pinion at or near the transition line between the hard case and the softer core material. Spalling also can be formed by destructive pittings breaking into each other, leaving wide voids.

When gears fail from destructive pitting or spalling, the gear designer must do a considerable amount of research. Unless the mesh shows evidence of heavy misalignment, which usually can be corrected, pitting is evidence that the design may not have the capacity* to carry the transmitted load. If this is the case, then something must be done to improve the load-carrying capacity. Medium-hardened gears can be fully hardened. This will increase the endurance limit of the surface material perhaps high enough to prevent pitting. A substitute gear material can be used such as a high-capacity material nitrided or maybe a material case-carburized.† Either of these materials is excellent for increased pitting capacity. Often changes in materials and heat-treatments are not enough and a redesign of the gear must be made. Increasing the face width of the gear will reduce the load per inch of face and increase pitting resistance. The center distance of the gearbox could also be increased. This would reduce the load per inch of face, with the net result of improving the surface pitting resistance. There are undoubtedly other factors that will increase the effective endurance limit of a given gear design. However, they require a more complete understanding of the finer points of the art of gear design, and perhaps a good text should be consulted such as "Practical Gear Design."[1]

FIG. 14-60. Spalling. (*Courtesy of Lubrizol Corp., Cleveland, Ohio.*)

* See Chap. 13 for methods to calculate the load-carrying capacity of a gearset.
† See Chap. 10 for detail information on hardening gears.

Gear-tooth Breakage. Breakage is a failure by breaking a whole tooth or a substantial portion of a tooth by either overload, shock, or the more common fatigue phenomena of repeated stressing above the endurance limit of the gear teeth in beam bending. Figure 14-61 depicts a typical fatigue failure. Note from this failure that the focal point, or characteristic "eye" of the fatigue break, is clearly visible.

Failures of this nature are caused by such things as overload due to design oversights, hob tears or notches in the root fillets, inclusions, heat-treatment cracks, and misalignments.[37] A *fatigue* failure actually takes many cycles of operation to develop. This is why the "eye" is quite often evident.[1] The break initially starts with a slight crack followed by oil seeping into the initial crack. The oil and slight motion develop "fretting corrosion," which is clearly visible in the fatigue "eye" and is an indication of the time necessary for a failure. Generally the break is quite smooth, which also indicates a considerable amount of working of the cracked surfaces before actual failure.

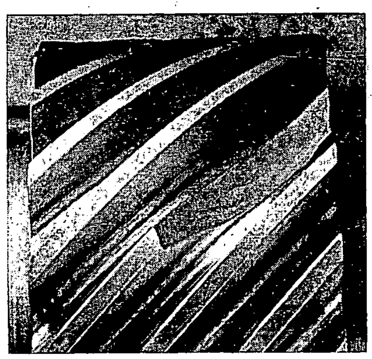


FIG. 14-61. Typical fatigue failure.

FIG. 14-62. Helical pinion failed from overload breakage.

Overload breakage causes a stringy fibrous-appearing break. In the harder materials this break has a more silky appearance, but the break still shows evidence of being wrenched apart. Overload failures are caused by sudden misalignment, bearing failure which tends to freeze up the gear transmission, or large foreign particles passing through the mesh. Figure 14-62 shows a casehardened helical pinion that has failed from overload breakage. It is quite interesting to note that several teeth have chunks broken out corresponding to the number of teeth in the contact zone. When gear teeth break or when foreign material passes through the tooth engagement, often quite catastrophic failure results. Figure 14-63 shows what can happen when something wedges the mesh apart causing a complete mismatch of tooth engagement.

Often gear-tooth breakage is a secondary effect from excessive wear or pitting failures. Figure 14-64 shows a spur gear with destructive pitting and spalling. This photograph clearly indicates fatigue cracks originating from stress risers set up by excessive pitting. If this gear is left in service, a typical pitch-line fatigue break will occur.

When gear teeth break, the investigator can often piece together the wreckage and quite accurately determine the tooth that failed first. If an end breaks off a given gear tooth, there may be some special reason. Misalignment is quite often to blame, and a careful analysis of the contact marking will point this out. Sometimes gears are damaged at assembly or during manufacture. If, perchance, the ends or the gear teeth are upset above the tooth surface, severe overloads will occur, and this section of the gear tooth fails from fatigue.

Other Types and Causes of Gear Failure. There are many cases of gear failure other than the four classical examples previously outlined. A few of the widely different kind will be considered.

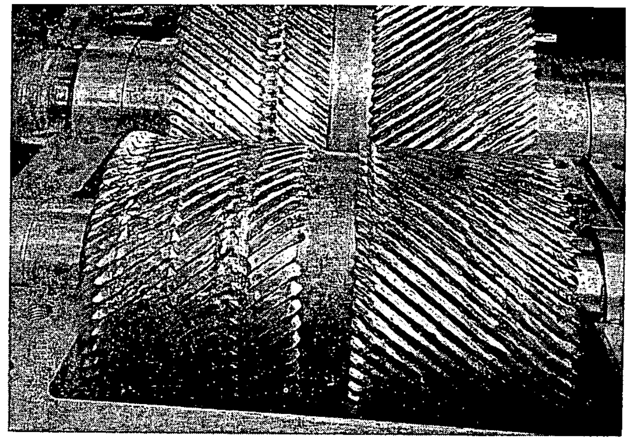


FIG. 14-63. Failed gearbox.

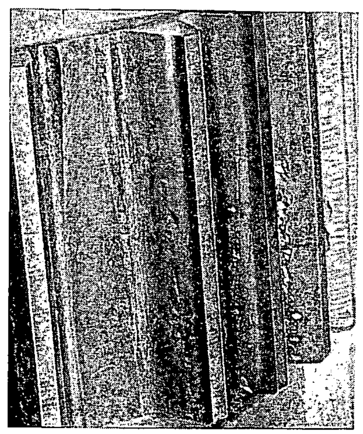


FIG. 14-64. Spur gear with destructive pitting and spalling.

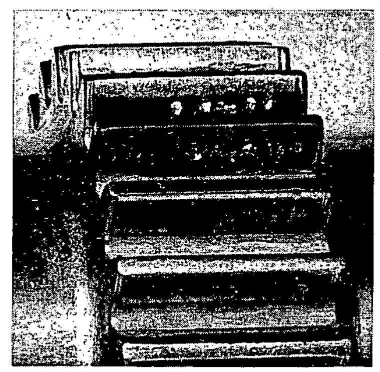


FIG. 14-65. Lubrication failure.

Quite often, a simple lubrication failure will cause a catastrophic failure of an entire gearbox. If, for example, a lubrication line ruptures or an operator fails to turn on the lubrication system, the complete gearbox will be starved for oil. The net result is extreme overheating, failed bearings, and badly scored and pitted gear teeth. Figure 14-65 is an example of a gear failure of this nature. The failure will show evidence of excessive heating, a deep blue color, the hardness of the material will be drawn back, and premature pitting due to lack of hardness will be under way. The lubricant can cause other troubles that are not quite so readily detectable as a starved gearbox such as: the viscosity of the lubricant may not be high enough to produce a good oil film; the "oiliness" of the lubricant may be below a safe level to prevent high mesh friction; the lubricant may not have adequate "wetting" properties; the lubricant may collect contaminants or break down and form harmful chemical by-products; the lubricant may oxidize and sludge up the filter causing either a restriction in flow or, by forcing the filter to by-pass,[38] permit unfiltered oil access to the bearings and gear mesh.

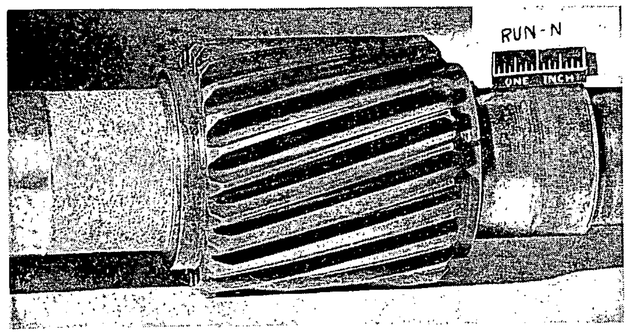


FIG. 14-66. "Fretting corrosion," ball bearing.

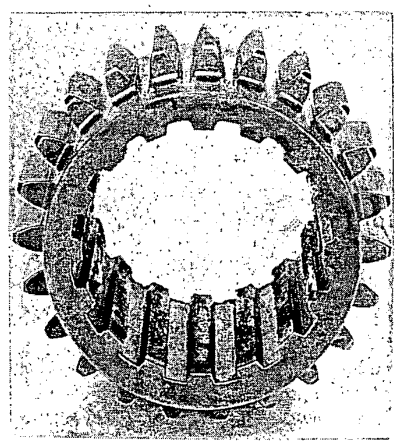


FIG. 14-67. "Fretting corrosion," spline.

"Fretting corrosion" can often lead to a failure of the complete gearbox. "Fretting corrosion" is most often found in areas where there are high loads, high vibrations, and little if any lubrication, such as ball-bearing fits, spline couplings, bushings, and grease-packed applications where there is no fluid present to wash away wear particles. Figure 14-66 shows what can happen to a gear journal when the ball-bearing fit starts to fail from "fretting corrosion." Many mils of material have been "fretted" away causing a high misalignment of the gear mesh. The by-products of "fretting corrosion" are clearly visible in Fig. 14-67. This build-up of sludge and wear particles indicates that considerable metal has been removed from the spline fit. Continuing to operate a gear in this condition would ultimately lead to gear failures.

At times the location and design of oil nozzles can influence the operation of a gearbox. It is very important to deliver oil to the bearing and the gear meshes in the proper quantities to lubricate and to carry the heat away. If the nozzles are not doing a proper job of wetting down the gear and bearing surfaces, adjustments must be made. Perhaps more oil nozzles are required or a relocation of existing nozzles must be made. Often, because of the windage of the mesh, oil nozzles do not have the ability to break down this barrier. A higher back pressure must be used to create the proper jet velocity to penetrate the windage of the gear mesh.

Another cause of gear failures can be traced to the various types of flexible couplings used to connect a given gearbox either to the prime mover or to the driven load.

Fig. 14-68. Dental-tooth coupling.

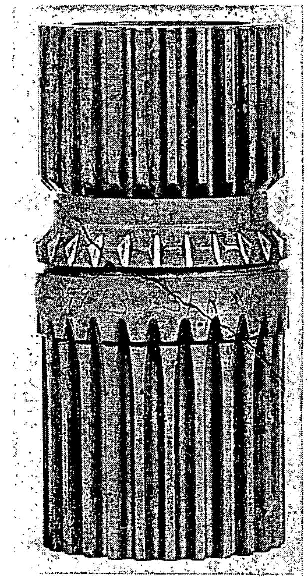


Fig. 14-69. Quill drive—failed.

Quite often, flexible couplings of the dental-tooth type are used such as illustrated in Fig. 14-68. This coupling has failed from the so-called "worm-tracking" effect as indicated by the metal that has been removed in the center of the contact zone.[39] The coupling also shows sign of heavy abrasive wear and "pickup." Generally, when couplings fail in this manner, they become sticky and seize. When this happens, bending moments are transferred into the gear casing, causing considerable trouble to the bearings and contributing to misalignment of the mesh. Couplings used for slightly different purposes can also cause considerable concern. Figure 14-69 shows a failed quill drive used to operate a scavenge pump. This quill failed from normal fatigue set up by stress concentration caused by hob nicks and abrupt section changes. Once a lube or scavenge pump fails, the failure of the complete gear unit is imminent.

14-12. Performance Testing of Gears. To assure the customer that the production gearbox will meet all the requirements as outlined by the specifications, the gear unit should be tested in the factory. Its performance should be observed and adjustments made, if necessary.

Full-power Testing. In order to measure gear performance, the unit must be operated under full-load and full-speed conditions. Ideally, the unit should be tested in exactly the same manner in which it is to be used in the field. It should be driven by its own prime mover and it should drive the required load. In practice at times it is impractical to duplicate exact field conditions in the shop. The practice of transmitting full horsepower through a given gearbox into the actual field equipment or some other load device is known as *power-absorption testing.* When the speed and horsepower rating of a given gear train is low, full-power-absorption testing is practical. Either the actual driven load can be used to absorb the power or a water brake or an electric dynamometer can absorb the power.

When the horsepower rating of the gear unit is large, power-absorption testing becomes more difficult. High-horsepower driving equipment is required as well as high-horsepower absorption equipment. Not only does the equipment become expensive, but when the tests are under way, a large amount of power must be dissipated as losses in the water brake or dynamometer. To simplify and to reduce the cost of full-load-testing high-horsepower gearboxes, a technique known as *torque*

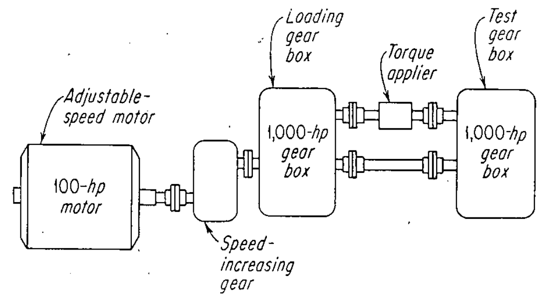


FIG. 14-70. Torque-testing setup.

testing[40] is widely used in the gear trade. Torque testing is based on the "four-square" principle whereby two gearsets are joined back to back and locked together after the full twisting couple is applied to one of the connecting shafts. Now that the gearsets are torqued together, the driving motor needs only to supply the static and dynamic friction losses of the bearing and gears as both units are brought up to speed. When the gearsets are up to speed, they are operating at an equivalent horsepower, equal to that horsepower required for a full-power-absorption test.[41] A schematic diagram of a torque-testing setup can be found in Fig. 14-70. Although torque testing does not fully duplicate exact field conditions, it does permit the factory to full-load-test even the highest-horsepower gear units manufactured.

Accessory Gearbox Testing. Accessory gears are usually designed with a single input or driver with multiple output or driven members. Accessories consist of such things as alternators, lube pumps, hydraulic pumps, power take-offs, and governors. It becomes almost impossible to make an accessory-box torque test; therefore, most accessory gear drives are tested using the power-absorption method. Since the accessories themselves are so many and varied, water brakes are usually used for the power-absorption device. Generally, accessory boxes are rated in the vicinity of 200 hp as a maximum. This being the case, the amount of power being absorbed and converted into losses is not prohibitive from a cost point of view. Figure 14-71 indicates an aircraft accessory gear being loaded with three water brakes to simulate various accessories. It is usually the practice to load accessory gears higher than the design load in order to wring out the gear-tooth loadings on the test stand. Water brakes permit extra power absorption that might not be available if the actual accessories were used instead of the brakes. Accessory-gearbox testing can also be programmed with certain prescribed load and speed cycles if water brakes are used simulating as nearly as possible certain expected service conditions.

Component Testing. Although in the final analysis the production gearbox should be run under field conditions, it has been beneficial in many respects to run components of gearboxes on various types of laboratory test equipment. Component testing can be accomplished at a lower cost and, perhaps, under more controlled testing conditions. Theoretical investigations can be undertaken such as a comprehensive program on the scoring resistance of various material-lubricant combinations or a study on the wear characteristics of certain types of gear steels.[42]

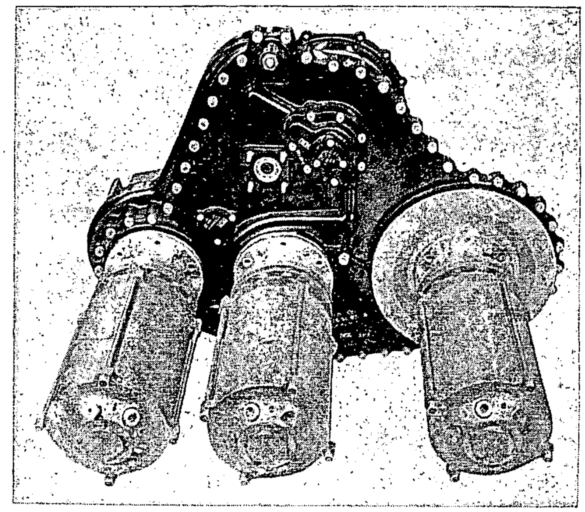


FIG. 14-71. Accessory-box testing. (*Courtesy of General Electric Co., Lynn, Mass.*)

Bearings and spline couplings can also be investigated on various component-test equipment. In fact, component testing has been an excellent source of information for design purposes. It is quite easy to try out new things on a component-test stand and observe how they operate and function. Based on this information, new equipment can be designed with a high degree of assurance for successful operation on the first production unit.

The component-test equipment shown in Fig. 14-72 is typical of the many gear-testing rigs that are in operation throughout industry. Equipment of this nature can be used to investigate pitting, wear, scoring, tooth breakage, lubrication, dynamic loads, and many other gear characteristics.

Efficiency Testing. In many gear applications, the efficiency of the gear train must be known within very close limits. Some customers, for example, request that the efficiency of a given production gearbox be measured at the factory and demand that this measured efficiency fall within the guaranteed limit. Of course, efficiencies can be calculated with a fair degree of precision;[4] however, when extreme accuracy is required, there is no substitute for efficiency testing.

If the gear train under consideration is of a design that lends itself to torque testing, then the over-all efficiency of a unit may be determined on a loss-torque basis. Two identical units are set up back to back with the prescribed amount of locked-in torque. A very accurate torque meter must be installed in the drive motor shaft next to the "four-square" torque loop. This meter will then measure the amount of torque that is required to drive the two locked-up gear trains. Since the motor supplies only the

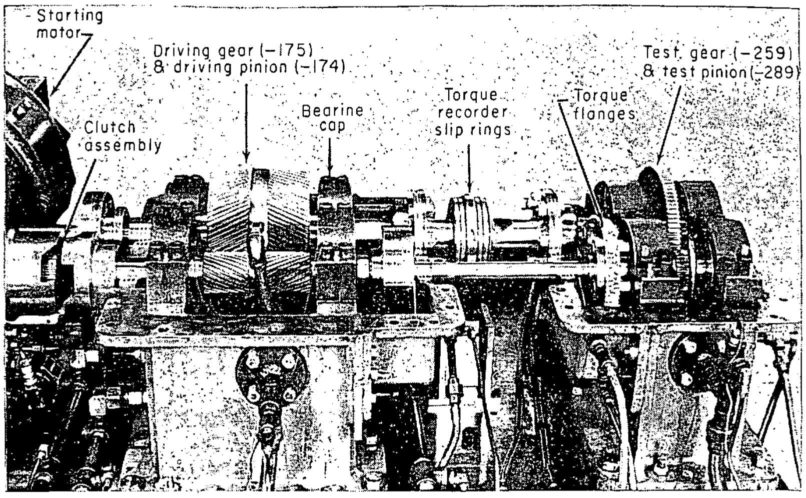


FIG. 14-72. Component-gear tester. *(Courtesy of General Electric Co., Lynn, Mass.)*

losses to the system, in effect, the inefficiency of two identical gearboxes has been measured. The efficiency can then be calculated on a torque basis.

Namely, efficiency in per cent equals

$$E = \frac{T_t - T_{L/2}}{T_t} \times 100 \qquad (14\text{-}99)$$

where E = efficiency, per cent
T_t = transmitted torque
T_L = loss torque

Often a given gearbox does not lend itself to a simple back-to-back torque test. Therefore, other means of efficiency testing must be devised. A simple heat balance on a loaded gearbox transmitting full horsepower will give the over-all efficiency of the geared transmission. In effect, all the losses of a gearset are reduced to heat, which is dissipated to the atmosphere and carried away by the lubrication system. If this heat loss can be measured accurately, then the basic efficiency of the drive has been measured.

The heat carried away by the lubrication system can be measured very accurately by carefully determining the oil flow through the gear casing and observing the temperature rise of the oil as it goes into and comes out of the gearbox. The heat loss by radiation and convection is very hard to determine; however, this problem can be by-passed by "lagging" the gearbox to keep heat loss of this nature to an absolute minimum. Of course, the heat that normally would escape will now show up as an increase in oil temperature across the gearbox. The "lagging" job on the gear casing

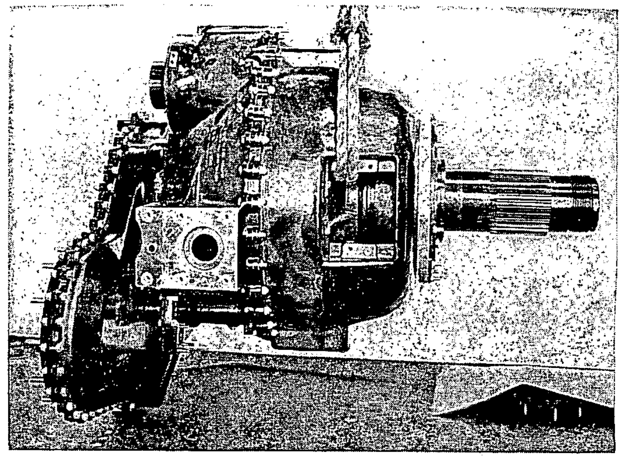


FIG. 14-73. Aircraft unit before lagging. (*Courtesy of General Electric Co., Lynn, Mass.*)

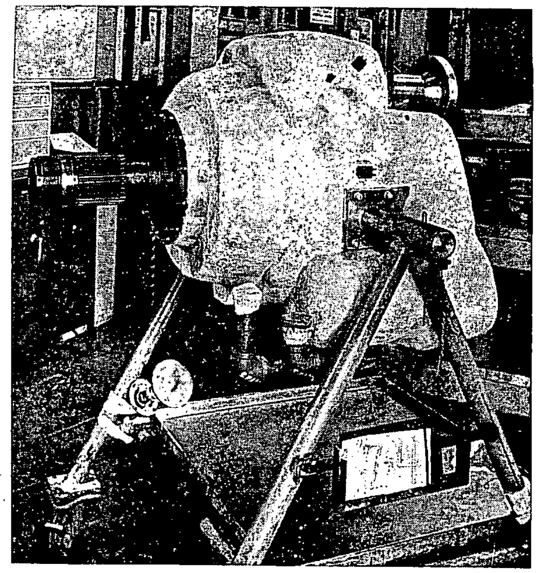


FIG. 14-74. Aircraft unit after lagging. (*Courtesy of General Electric Co., Lynn, Mass.*)

must be of high quality; in fact it should be done by an expert using the very best insulating materials. The aircraft gear casing in Fig. 14-73 and 14-74 shows a before and after "lagging" job. Note from these photographs that an ample amount of lagging has been used and that a professional job of "lagging" has been accomplished. The efficiency of a gearbox from a heat-balance test can be determined by the following equations:

$$Q = MC_p \, \Delta T \qquad (14\text{-}100)$$

where Q = heat loss, Btu per min
M = lb of oil per min
C_p = specific heat of oil
ΔT = temperature rise, °F (inlet to outlet)

$$P_L = \frac{Q}{42.4} \tag{14-101}$$

where P_L = horsepower loss
42.4 = Btu per hp

$$E = \frac{P_T - P_L}{P_T} \times 100 \tag{14-102}$$

where E = efficiency, per cent
P_T = transmitted horsepower
P_L = horsepower loss

Lubrication Testing. In order to get the best performance from a gearbox, the design engineer must not only have a good design, but he must use the best lubricant for his application. The complete lubrication system, in fact, must meet all the

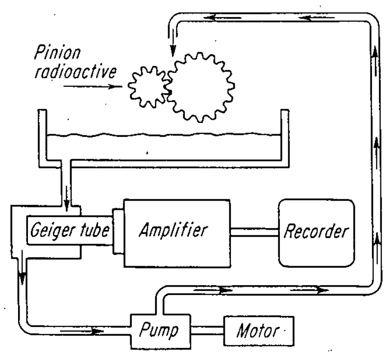


Fig. 14-75. Radioactive wear test.

performance requirements of the gearbox or else the gear itself will never make it. We know from experience that the two major failures relating to the lubricant are scoring and normal or long-range wear.[33] To evaluate a new gear oil is rather a time-consuming and costly project. The procedure may vary somewhat, depending upon the ultimate use of the oil, but in general, the following steps should be taken: All available information pertaining to the oil should be obtained from the manufacturer and the various laboratories that have been working with the oil. Laboratory checks should be made on an oil sample, such as viscosity, pour point, flash point, and oxidation stability. Comparison tests should be run on bench tests such as the Shell four-ball wear tester and the Timken wear tester. Lubricated compression-roll tests should be run under simulated service conditions and the results compared with previous tests. Scoring tests using test gears on a four-square gear-testing apparatus must be run to determine the "critical scoring temperature" of the oil. Wear tests should be run on a radioactive gear tester to determine the wear characteristics of the new oil.

Since wear as such is very hard to detect and measure, the radioactive technique of gear testing has been adapted for studying the wear phenomenon.[33] Minute quantities of wear can be detected in short periods of time and several oils may be tested with the same sets of gears.

The schematic test setup of Fig. 14-75 gives a good flow picture of how the radioactive technique is used for testing gears and gear lubricants. The pinion to be tested is sent to the nearest atomic pile, where it is irradiated with neutrons. This makes the pinion radioactive. The pinion and gear are then mounted in a test stand. The gears are loaded and the test machine put into motion. The lubricant is circulated through the gearbox, past the counter tube, and back into the sump, as indicated by Fig. 14-75. When the active gears commence to wear, the lubricating oil transports the worn iron particles past the counter where they are detected. It is a rather simple laboratory technique to convert the counting rates as recorded to milligrams of wear per unit of time.

In most cases environmental conditions affect the lubricant more than any other characteristic of a gear setup.[43] Because of this, most environmental testing has been centered around the lubricant as one of the main variables. A large per cent of environmental testing has been done under close laboratory supervision on component-gear-test equipment.[38]

At the present time, work is under way on both high-temperature and low-temperature gear operation for the rocket and missile field. High-altitude atmospheres are under investigation.[35] Scoring and wear tests are in progress under all kinds of adverse conditions. Gear-lubrication problems of all sorts are under investigation, such as grease lubrication, solid-film lubrication,[49] exotic-liquids lubrication, lubricant additives, and many other new and different environmental conditions.

Field Testing. Field operation is the best proving ground an engineer can find for evaluation of gear equipment. The various gear designs must operate successfully in the field, and much can be learned from obtaining and observing operational records of the gear arrangements that are in service.

In many instances, service operating conditions are much different from the original design conditions, and it is not uncommon to find a customer using the equipment far in excess of what has been recommended. The reverse is quite often true, however, and a design engineer can easily be fooled by the seemingly successful operation of gearsets in the field. The apparent successful operation may be the result of not operating the gear unit up to design conditions.

By firmly following the policy of keeping close tabs on field operations, the gear engineer not only obtains good information upon which to base his new designs but also creates a good working relationship with the customer.

REFERENCES

1. Dudley, Darle W.: "Practical Gear Design," McGraw-Hill Book Company, Inc., New York, 1954.
2. Philadelphia Gear Works: "Philadelphia Gears," Philadelphia Gear Works, Philadelphia, Pa., 1958.
3. Buckingham, Earle: "Analytical Mechanics of Gears," McGraw-Hill Book Company, Inc., New York, 1949.
4. Shipley, Eugene E.: How to Predict Efficiency of Gear Trains, *Prod. Eng.*, August 1958, pp. 44–45.
5. Merritt, H. E.: "Gears," 3d ed., Pitman Publishing Corporation, New York, 1955.
6. Candee, Allan H.: Bevel Gears in Aircraft, *Trans. ASME*, May, 1943, pp. 267–277.
7. Gleason Works: "20° Straight Bevel Gear System," Gleason Works, Rochester, N.Y., 1949.
8. Gleason Works: "Zerol Bevel Gear System," Gleason Works, Rochester, N.Y., 1950.
9. Gleason Works: "Spiral Bevel Gear System," Gleason Works, Rochester, N.Y., 1951.
10. Illinois Tool Works: "Spiroid Gears," Illinois Tool Works, Chicago, Ill., 1959.
11. Illinois Tool Works: "Helicon Gears," Illinois Tool Works, Chicago, Ill., 1959.
12. Gleason Works: "Bevel and Hypoid Gear Design," Gleason Works, Rochester, N.Y., 1947.
13. Illinois Tool Works: "Planoid Gears," Illinois Tool Works, Chicago, Ill., 1958.

14. Wilcock, Donald F., and E. R. Booser: "Bearing Design and Application," McGraw-Hill Book Company, Inc., New York, 1957.
15. Palmgren, Arvid: "Ball and Roller Bearing Engineering," S. J. Burbank & Co., Inc., Philadelphia, Pa.
16. Barish, Thomas: "Fundamentals of Ball Bearing Design," unpublished paper, March, 1957.
17. Watt, Hamilton R.: Gear Tooth Efficiencies, I, II, III, *Power Transmission*, March, April, June, 1951.
18. Buckingham, Earle: Manual of Gear Design, Sec. 2, *Machinery*, 1935.
19. Tuplin, W. A.: Compound Epicyclic Gear Trains, *Machine Design*, April, 1957, pp. 100–104.
20. Buckingham, Earle: Dynamic Loads on Gear Teeth (Report of the ASME Special Research Committee on the Strength of Gear Teeth), American Society of Mechanical Engineers, New York, 1931.
21. Tuplin, W. A.: Gear Tooth Stresses at High Speed, *Proc. Inst. Mech. Engrs. (London)*, Vol. 163, pp. 162–175, 1950.
22. Tuplin, W. A.: Dynamic Loads on Gear Teeth, *Machine Design*, October, 1953, pp. 203–211.
23. Strauch, H.: Zahnradschwingungen, *VDI Zeitschrift*, Vol. 95, No. 6, pp. 159–163, 1953.
24. Reswick, J. B.: Dynamic Loads on Spur and Helical Gear Teeth, *Trans. ASME*, Vol. 77, pp. 645–654, 1954.
25. Attia, A. Y.: "Dynamic Loading of Spur Gear Teeth," Doctoral Thesis, University of Sheffield, 1956.
26. Niemann, G., and H. Rettig: "Errors Induced by Dynamic Gear Tooth Loads," International Conference on Gearing, London, September, 1958.
27. Zeman, J.: Dynamic Forces in Gear Drives, *VDI Zeitschrift*, Vol. 99, No. 6, pp. 244–254, February, 1957.
28. Harris, Stephen L.: "Dynamic Loads on the Teeth of Spur Gears," The Institution of Mechanical Engineers, December, 1957.
29. Kohler, H. K.: "The Mechanism and Measurement of Dynamic Loading in Spur Gears," Ph.D. Thesis Submitted to the University of Sheffield, May, 1959.
30. Buckingham, Earle: "Gear Tooth Loads," Massachusetts Gear & Tool Company, 1953.
31. Cleare, G. V.: "Gear Tooth Failures in Highly Stressed Transmission Gears," International Conference on Gearing, The Institute of Mechanical Engineers, September, 1958.
32. Tuplin, W. A.: Gear Tooth Lubrication, *Machine Design*, August, 1954, pp. 125–131.
33. Shipley, Eugene E.: Design and Testing Considerations of Lubricants for Gear Applications, *Lubrication Eng.*, April, 1958, pp. 148–152.
34. Shipley, Eugene E.: Investigation of Factors Affecting High Temperature Gear Operation, *Lubrication Eng.*, March, 1959, pp. 98–104.
35. Hartman, M. A.: Gears for Outer Space, *Machinery*, August, 1959, pp. 115–121.
36. American Gear Manufacturers Assoc.: "Standard Nomenclature of Gear-tooth Wear and Failure," AGMA 110.03, 1960.
37. Frederick, S. H., and A. D. Newman: "Gear Failures," International Conference on Gearing, The Institution of Mechanical Engineers, September, 1958.
38. Shipley, Eugene E.: Will Gears Operate at 600°F?, *Am. Machinist*, February, 1958, pp. 101–103.
39. Batter, J. F., and F. E. Reed: "Wear in Heavily Loaded Lubricated Surfaces Subjected to Oscillating Motion," American Society of Lubrication Engineers, Paper 59GS-12, January, 1959.
40. Shipley, Eugene E.: 12 Ways to Load Test Gears, *Prod. Eng.*, January, 1958, pp. 77–82.
41. Newman, A. D.: "Load-carrying Tests of Admiralty Gearing," International Conference on Gearing, The Institution of Mechanical Engineers, London, September, 1958.
42. Crook, A. W.: The Lubrication of Rollers, *Phil. Trans. Roy. Soc. London*, A, Vol. 250, pp. 387–409, January, 1958.
43. Research Report, Inert Gas Boosts Oil's Ability, *Chem. Eng. News*, October, 1958, pp. 34–35.
44. Hirst, W., and J. K. Lancaster: The Influence of Oxide and Lubricant Films on the Friction and Surface Damage of Metals, *Proc. Roy. Soc. London*, A, Vol. 223, pp. 324–338, 1954.
45. Buckingham, Earle: "Spur Gears," McGraw-Hill Book Company, Inc., New York, 1928.
46. Radzimovsky, Eugene I.: Planetary Gear Drives, *Machine Design*, June, 1959, pp. 144–153.

47. Dudley, Darle W.: Modification of Gear Tooth Profiles, *Prod. Eng.*, September, 1949.
48. Barish, Thomas: Idler Gears, *Machine Design*, November, 1957, pp. 121–125.
49. Moore, Wayne L.: "Reliability of Gear Applications for Aero-space Vehicles," American Gear Manufacturers Association, March, 1960.
50. Editor: Full Load Efficiency Testing of Turbine Gear Units, *Engineering*, March, 1959, pp. 306–307.
51. Shipley, Eugene E.: "Efficiency of Involute Spur Gears," American Society of Lubrication Engineers, January, 1959.
52. Barish, Thomas: "On Hertz Contact with Sliding," American Gear Manufacturers Association, March, 1960.
53. Crook, A. W.: Simulated Gear-tooth Contacts. Some Experiments upon Their Lubrication and Subsurface Deformations, *Proc. Inst. Mech. Engs. London*, Vol. 171, No. 5, 1957.
54. Buckingham, Earle: Dynamic Loads on Gear Teeth, *Machinery*, October, 1958, pp. 117–121.
55. Attia, A. Y.: "Dynamic Loading of Spur Gear Teeth," ASME Paper 58-SA32, 1958.
56. Gatcombe, Ernest K., and Roy W. Prowell: "Rocket Motor—Gear Tooth Stress Analysis . . . Hertzian Contact Stresses and Times," ASME Paper 59-A-256, 1959.
57. Michigan Tool Company: "Cone-drive Gears," Michigan Tool Company, Detroit, Mich. 1950.

Chapter 15

LUBRICATION OF GEARS

By

FOLKE RICHARDZ, *Consulting Specialist, Gearing Products, Westinghouse Electric Corporation, Buffalo, N.Y.*

All gears, regardless of type and material, will gain in life expectancy by being properly lubricated. The lubrication of gears should therefore be as much a part of the design requirements for successful operation as is the calculation of gear-tooth stresses or surface durability. The art of proper lubrication is not so well established as the methods of gear-tooth calculations, and it is the purpose of this chapter to explore and evaluate not only the generally accepted methods of lubrication and types and viscosity of oil for various operating conditions, but also methods for retaining the lubricant within prescribed boundaries and means for keeping the viscosity of the lubricant at temperatures which will assure maximum protection to gears and bearings.

15-1. Systems and Methods of Gear Lubrication.* The methods by which lubricant is applied to gear teeth vary with type of gear, speed in terms of pitch-line velocity, surface finish, hardness of the material, and combination of materials. In open gearing or gearing mounted on pedestal supports and not enclosed, a method of applying lubricant by a paddle or brush applies almost exclusively to slow-speed coarse-pitch unprotected gears. Since this method is tedious and expensive, it is fast disappearing and is called to the designer's attention only as a method to be avoided whenever possible.

Figure 15-1 illustrates splash lubrication commonly used in speed reducers. The illustration depicts double-reduction gearing, and it is rather obvious that the second-reduction or slow-speed gearset should lubricate satisfactorily as the gear is dipping into the lubricant reservoir.

The high-speed gearset, since it does not dip into the lubricant, may be questionable in regard to satisfactory lubrication. Because of lack of experimental data, adequate lubrication is assured by the method indicated in Fig. 15-2, where a splash pan is introduced and the oil level is increased to permit the high-speed gear to dip. To assure adequate and satisfactory lubrication at low speed, a check should be made by calculating the critical speed.

To attain the desirable end result, the speed must be such that the centrifugal force exceeds gravity.[1,†] Viscosity will of course enter the problem to some extent but

* See Chap. 14 and particularly Art. 14-4 for further information on gear performance and lubrication.

† Superscript numbers refer to references at the end of the chapter.

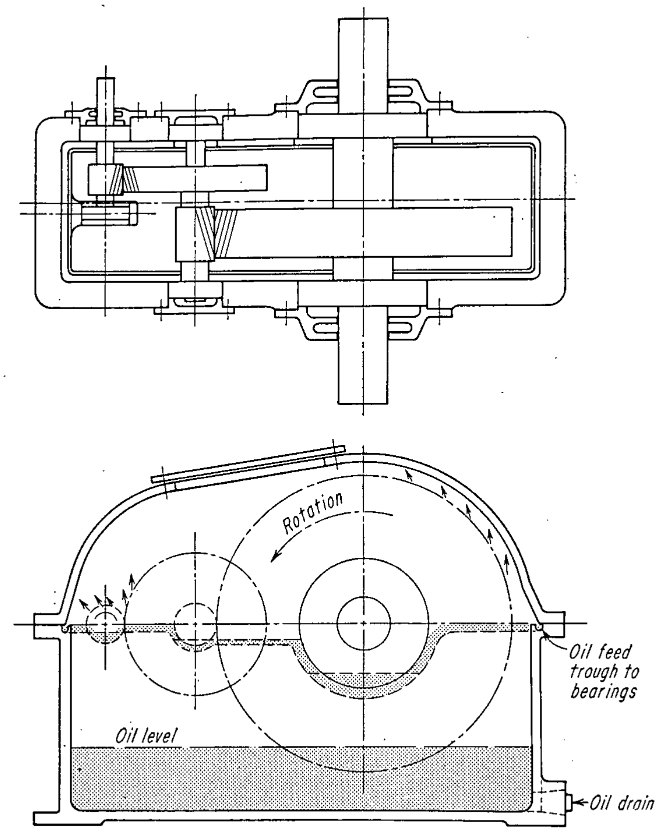


FIG. 15-1. Lubrication of gears.

apparently only to a point of varying the amount of oil thrown off the periphery rather than affecting the critical speed. Experiments have definitely proved that lubricants of viscosity of 250 SSU or less satisfy the equation $F = WV^2/gR$ in regard to critical speed when the centrifugal force is set equal to the weight of the object.

F = centrifugal force, lb
W = weight of the revolving body, lb
R = radius of the gear, ft
V = velocity, fps
N = rpm
g = acceleration due to gravity = 32.16. Since we have assumed $F = W$ our equation becomes $1 = V^2/gR = 0.000341RN^2$ or $N^2 = 2{,}935/R$.

As either of the two terms may be variable, we can solve for one with the other fixed. As an example, consider the radius of the gear as 2 ft, and it is desired to know how slowly it may run with safety for good splash lubrication. We substitute R in the equation and solve for N,

$$N^2 = \frac{2{,}935}{2} = 1{,}467.5 \qquad N = 38 + \text{rpm}$$

For unusually slow speeds, where the splash from the gears does not reach the oil troughs feeding oil to the bearings, scrapers as illustrated in Fig. 15-3 are effective. The arm of the scraper extending from the case to the side of the gear should be made adjustable so that the scraper may be brought within approximately a $\frac{1}{32}''$ from the side of the gear. The tip of the scraper may be made from either metal or a material such as synthetic rubber.

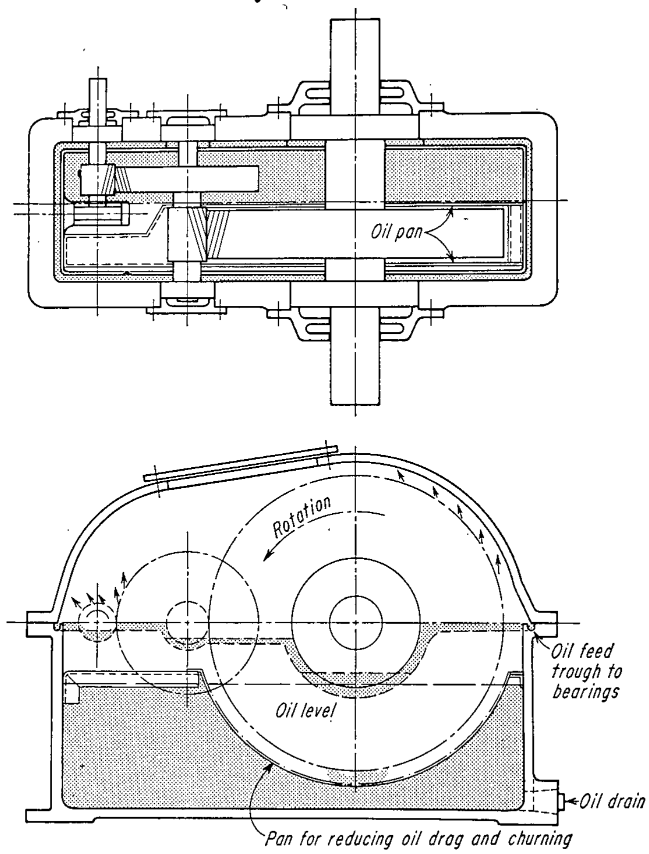


Fig. 15-2. Restricted splash lubrication.

Figure 15-4 illustrates typical splash lubrication as employed in a double-reduction gear motor. Figure 15-5 illustrates a forced-feed lubrication system frequently used either from a central system or an integral pump driven from one of the shafts in the gear drive or from a separate motor-driven pump. When the forced-feed method of gear lubrication is employed in gear reducers having antifriction bearings and no provision for oil-feed troughs to the bearings, a fine stream from the force-feed system should be directed onto the bearings. When sleeve-type bearings are employed, adequate feed lines to the bearings must always be provided.

Figure 15-6 shows a typical forced-feed system for high-speed industrial gear units employing sleeve bearings. The amount of oil required to lubricate sleeve bearings satisfactorily is dependent on the specified bearing clearance and the rpm involved. A satisfactory basis for selecting an oil pump to provide an adequate amount of lubricant to gears and bearings is to allow 0.03 gal of oil per square inch of the projected total bearing area involved and 0.5 gal* for each inch of gear face width.

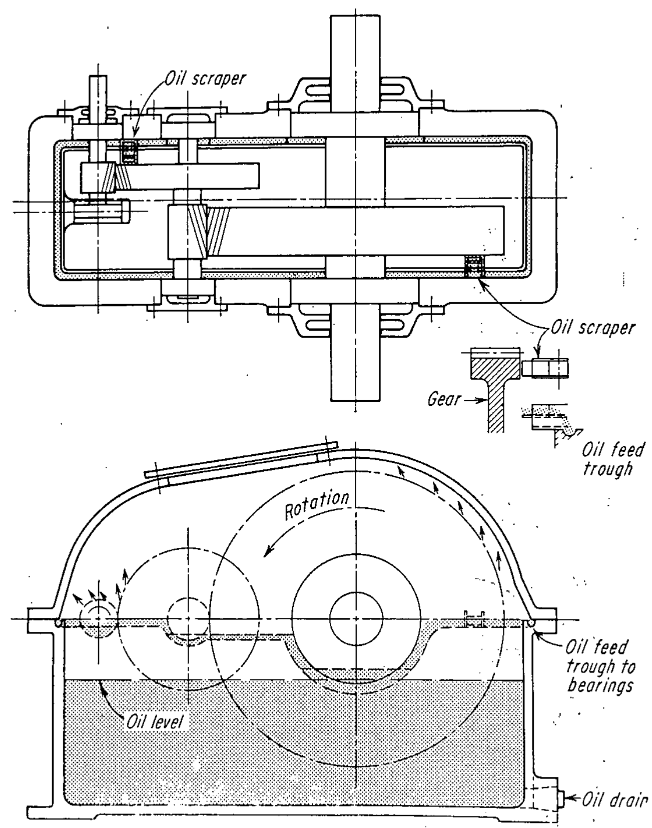


FIG. 15-3. Method of feeding oil to bearings at very slow speed without forced feed.

In general, high-speed gear drives are unidirectional; however, at times dual direction of rotation is required. In addition, on applications such as pipelines, where diesel or gas engines are used as prime movers, a back pressure in the line may develop causing the gear unit to run in the direction for which it was not designed. If a unidirectional pump is used without safeguards for such an event, the oil piping may burst. Since the bearings would be starved for lubrication, an expensive repair would result.

Figure 15-7 shows a simple check-valve arrangement acting as a safeguard for occurrences of this type and also satisfying the requirement for dual direction of rotation.

* See Art. 15-6. In some cases this is not enough oil.

GE-1024.135

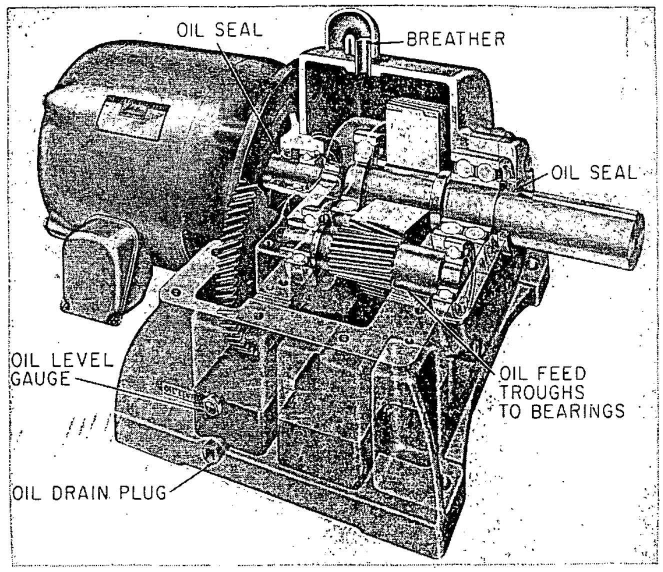


FIG. 15-4. Gear motor with splash-lubricated double-reduction gearing. (*Courtesy of Westinghouse Electric Corporation, Pittsburgh, Pa.*)

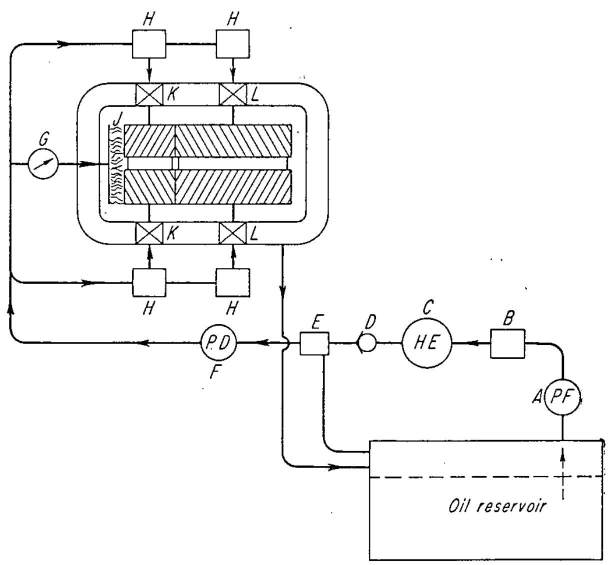


FIG. 15-5. Schematic diagram of a forced-feed gear-lubrication system. A = lube oil pump. B = lube oil strainer. C = heat exchanger. D = check valve. E = pressure-relief valve. F = protective device. G = pressure gage. H = lube oil adj. valves. J = oil spray to gearing. K = pinion bearings. L = gear bearings.

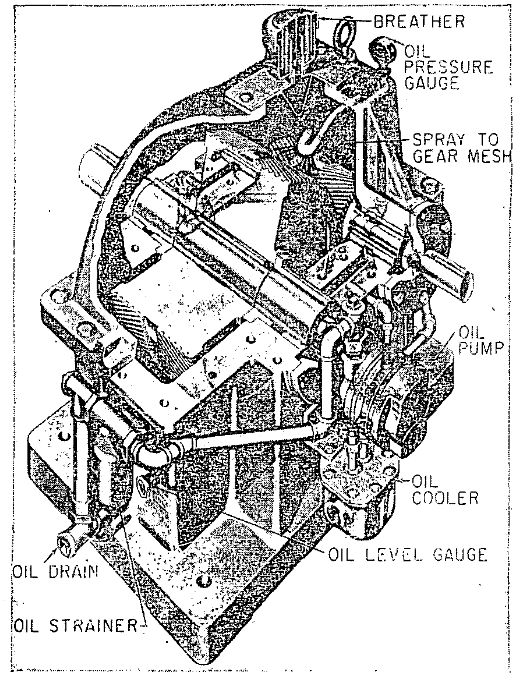


FIG. 15-6. High-speed industrial gearset with forced-feed lubrication system. (*Courtesy of Westinghouse Electric Corporation, Pittsburgh, Pa.*)

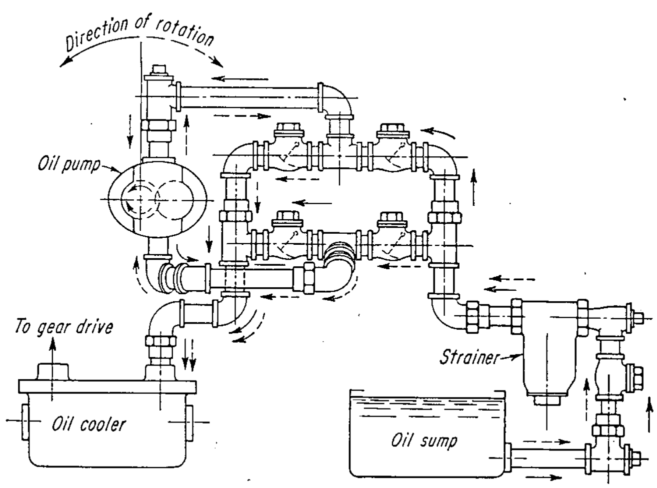


FIG. 15-7. Check-valve arrangement for dual direction of rotation.

On some aircraft and special applications of high-speed gearing where the weight is critical, the rim of the gear as well as the central web is made exceptionally thin and therefore presents a different lubrication problem. The lubricant in this case is generally directed to the underside of the rim and the sides of the web primarily to keep the metal cool and the gear mesh* lubricated essentially by the mist created by the spray on rim and web (Fig. 15-8).

Spiral bevel and hypoid gears at peripheral velocity up to approximately 2,000 fpm are generally lubricated by dipping into the oil supply. At higher velocities forced-feed lubrication is generally employed. The direction of spray should be such that it covers the entire width of the gear face.

Vertical gear drives are most frequently forced-feed lubricated and particularly where more than one gear reduction is employed. Figure 15-9 shows a typical vertical double-reduction gear drive forced-feed lubricated.

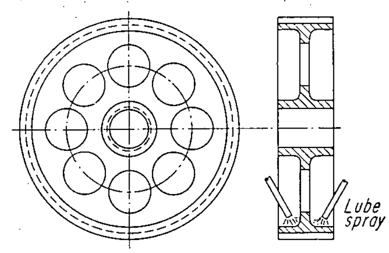


Fig. 15-8. Lubrication of aviation gearing.

In certain applications where it is not convenient or practical to immerse the pinion in the oil reservoir or employ forced feed, an oil flinger is mounted on the pinion shaft of sufficient diameter to dip into the oil reservoir as illustrated in Fig. 15-10. The oil flinger is designed in such a manner that it throws a sufficient amount of lubricant into the gear mesh and bearings.

In summary, systems and methods of gear lubrication are

1. Paddle, brush, or drip cup, used primarily for slow-speed, open, or semiprotected gearing
2. Dip or splash, used for medium pitch-line velocity up to approximately 2,500 fpm
3. Restricted dip, using oil splash pan. Employed for pitch-line velocities up to approximately 3,500 fpm
4. Pressure lubrication by
   a. Attached lube oil pump
   b. Separate motor-driven lube oil pump
   c. Central lube oil pressure system; pressure lubrication may be employed for low as well as high pitch-line velocities, provided proper control of volume and pressure is provided
5. Mist; this method is restricted to the more unusual conditions such as high rpm's or applications where build-up of lubricant is not tolerated

15-2. Types and Viscosity of Gear Lubricants. The most common types of gear lubricants used to hold gear-tooth wear to a minimum and minimize rubbing and sliding friction between metallic gear teeth are generally a product of vegetable, animal, or mineral origin and, recently, synthetic lubricants such as silicon compounds. The mineral oils, be they of the paraffin series (generally referred to as Pennsylvania oil) or

* See Art. 15-6 for other methods of lubricating aircraft gears.

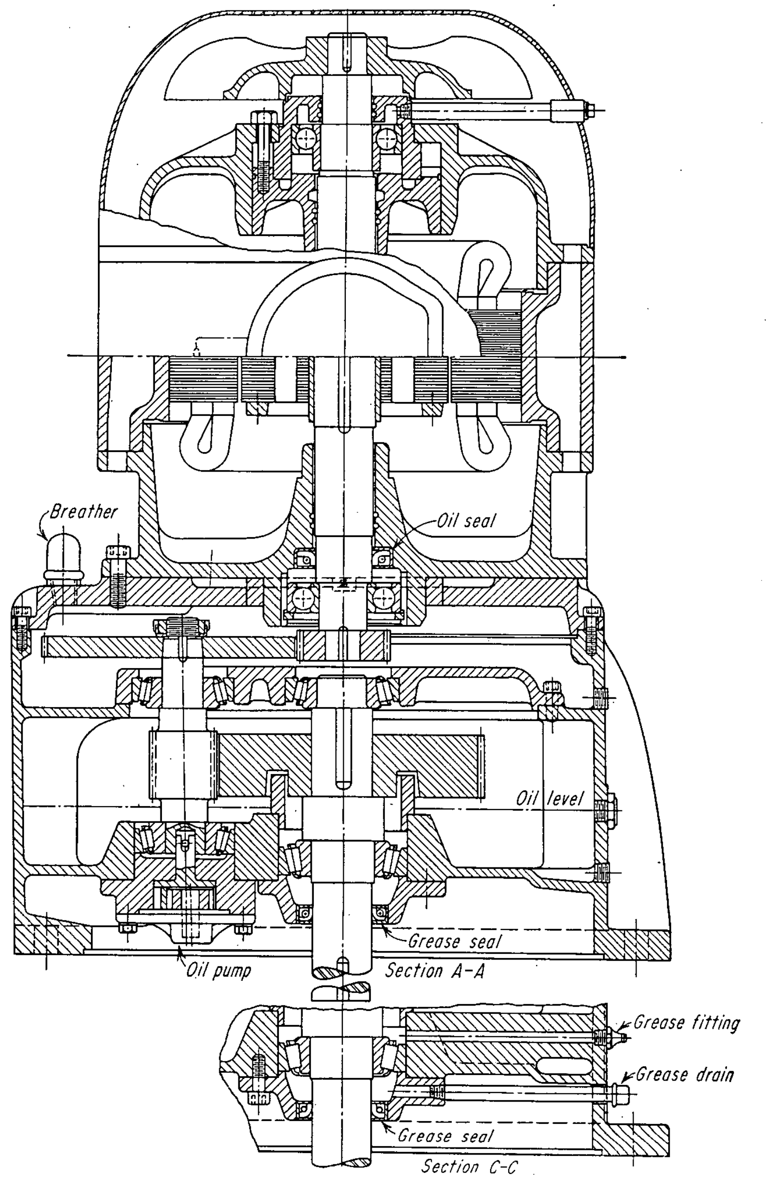
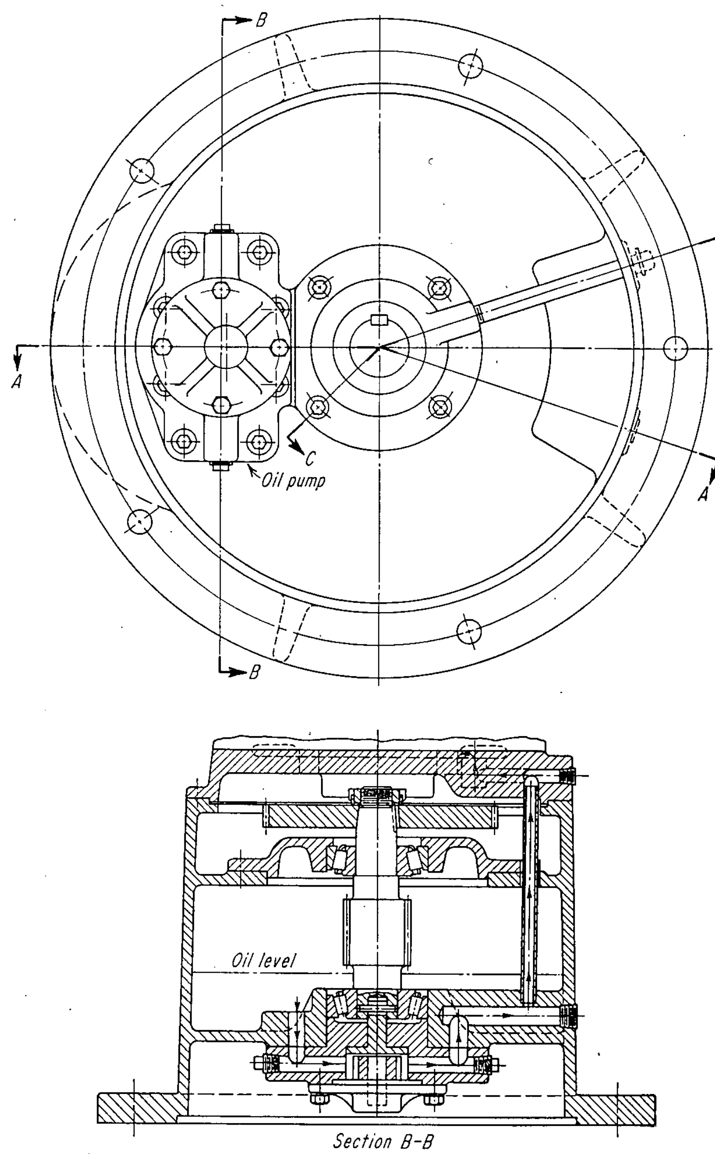


FIG. 15-9. Vertical double-reduction



gear with forced-feed lubrication.

the asphaltic series, occupy at least for the present the number-one place in gear lubrication. The mineral oils in many of today's highly loaded gears are in many instances not capable of satisfactorily carrying the load imposed and must therefore be fortified with other elements such as sulfur, chlorine, and lead naphthanates and combinations of these elements and as such they are generally referred to as non-corrosive extreme-pressure lubricants. In gear-lubrication research and testing engineers continue to explore the reasons for gear-tooth wear, pitting, and failures of various types. The American Gear Manufacturers Association has issued standard nomenclature[2] identifying the various and most common types of gear-tooth surface deterioration. Gear lubricants of high viscosity at light load show higher friction loss, but as the load increases, the heavy-viscosity lubricants show less frictional losses, indicating that viscosity is an important factor and the gear-tooth loading should therefore be given due consideration by the designer in the selection of lubricants. Based on present gear-tooth design limitations by AGMA standards for strength and

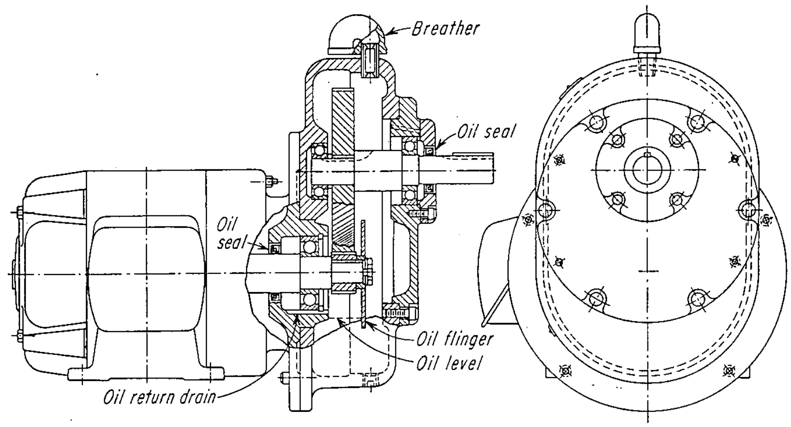


FIG. 15-10. Oil flinger used as a lubrication means.

durability, industrial gear-lubrication standards have been developed by the gear manufacturers.[3,4,5]

Oil-film-strength measurements and evaluations are to some extent questionable, but in general the Timken Roller Bearing Company's extreme-pressure testing machine has provided a comparatively reliable evaluation in regard to load rating and comparison of lubricants. An extreme-pressure oil should not have a value of less than 35 lb on the Timken extreme-pressure test machine. Presently extreme-pressure gear oils will measure 35 to 70 lb for viscosities of 44 to 900 sec at 210°F. Straight mineral oils are always less than 35. In addition to the Timken extreme-pressure testing machine there are other test equipments fairly widely used such as the Almen tester and the Shell four-ball extreme-pressure tester. The use of extreme-pressure lubricants for the initial "running-in" period of gearing has by actual experience been found to have definite advantages, as they will to a great extent prevent incipient or initial scoring, galling, and seizure. This feature is attributed to the greater film strength and antiweld properties. Application of extreme-pressure lubricants to gearing where heavy shock and vibration loads are encountered in many instances has improved the performance and minimized maintenance.

Under heavy load conditions, a heavy mineral lubricant would ordinarily be required. The operating temperatures may exceed good practice and cause rapid oxidation and general deterioration of the lubricant. By changing to a lighter-viscosity extreme-pressure lubricant, equal or better results will be found in regard to load-carrying capacity and a decided drop in operating temperatures, tending to minimize the frequency of lubricant change due to less oxidation deterioration.

The gear lubricant, be it straight mineral oil, extreme-pressure lubricant, etc., will be of additional value to the user if it contains protective additives.[6] The most commonly used and important protective additives are rust inhibitors, defoamants, and oxidation inhibitors. The rust inhibitor is one of the most important when the equipment is subjected to condensation due to fluctuating temperature and where the gearing may be idle for extended periods of time and operating outdoors with little or no protection. The oxidation inhibitor is of great value when the gearing is operating in the higher-temperature range. The defoamant is of particular value when the lubricant is subject to agitation due to splash lubrication at high pitch-line velocities or forced-feed lubrication where the oil pump capacity is in excess of the oil reservoir capacity.

Viscosity and viscosity index are important factors in the selection of gear lubricants. Gear-teeth contact areas are relatively small; therefore, the contact unit pressures are comparatively high. It is therefore important to provide a film of lubricant of sufficient strength to withstand the localized pressure during the period of tooth contact. The pitch-line velocity of the gearset governs the length of time the oil film must withstand the pressures. When speeds are high, the gear-tooth contact time is short and the loads usually not so heavy, and as a result, a lubricant of a relatively low viscosity may be used. At the lower speeds the loads are generally high and the tooth contact time longer, and consequently a lubricant of a higher viscosity should be used. Where high-speed and low-speed gearing are contained in the same gearbox and lubricated from the same source of lubricant, the lubricant viscosity required for the lower-speed gearing should be selected.

The viscosity index was established by Dean and Davis in 1929.[7,3] Essentially it is an arbitrary system evaluating the relationship between viscosity and temperature. The standard of comparison is Pennsylvania oil, rated 100, and coastal naphthinic, rated 0 when refined by the same sulfuric acid process.

Present modern refining methods make it possible to obtain higher values for both Pennsylvania and coastal oils. The significance of viscosity index (V.I.) in gear lubrication is that, when a gear drive is subjected to extreme variations in temperatures, it is possible that the oil of a low V.I. may become so viscous that it presents a problem in starting by the prime mover. For example, an oil of zero V.I. having a viscosity of 100 SSU at 210°F would have approximately 2,640 SSU at 100°F; an oil of 100 V.I. with the same viscosity at 210°F, namely, 100 SSU, would have only approximately 1,200 SSU at 100°F, or a ratio in excess of 2:1 in favor of the high V.I. oil. Generally, a V.I. of 60 and above should be selected where temperatures vary over a wide range. When gear drives are subjected to low temperatures where starting becomes a problem because of too viscous an oil, it may be advantageous to insert an electric-immersion-type heater in the oil reservoir of sufficient capacity to bring up the temperature of the volume of oil to provide a viscosity that will permit easy starting. The general requirement and range of gear lubricant for enclosed gearing according to the American Gear Manufacturers may be covered by eight AGMA lubricant numbers of viscosities as indicated in Table 15-1.[3]

There is no infallible rule as to the exact viscosity of lubricant to be used for the various sizes or ratings of gear units, but if the gearing is designed in line with AGMA standards, it is generally adequate to use the lubricant recommendations shown in Table 15-2.

Lubrication of Wormgearing. Since wormgearing is subjected to all sliding friction as against sliding and rolling in involute spur, helical, and herringbone gear-

Table 15-1. Viscosity Range for Various AGMA Lubricants

AGMA lubricant No.	Viscosity range, SSU	
	At 100°F	At 210°F
1	180–240	
2	280–360	
3	490–700	
4	700–1,000	
5		80–105
6		105–125
7 comp.*		125–150
8 comp.*		150–190
8A comp.*		190–250

* Oils marked "comp." are those compounded with 3 to 10 per cent of acidless tallow or other suitable and accepted animal fats. The oils marked "comp." are essentially intended for wormgearing but may also be used for any gearing where the viscosity is considered suitable for the application.

Table 15-2

Type and size of unit, main gear low-speed centers	Ambient temp., °F	
	15–60°F, use AGMA No.	50–125°F, use AGMA No.
Parallel shaft (single reduction):		
Up to 8″	2	3
Over 8 up to 20″	2	4
Over 20″	3	4
Parallel shaft (double reduction):		
Up to 8″	2	3
Over 8 and up to 20″	3	4
Over 20″	3	4
Parallel shaft (triple reduction):		
Up to 8″	2	3
Over 8 and up to 20″	3	4
Over 20″	4	5
Planetary-gear units:		
OD housing up to 16″ dia.	2	3
OD housing over 16″ dia.	3	4
Spiral or straight bevel gear units:		
Cone distance up to 12″	2	4
Cone distance over 12″	3	5
Gear motors	2	4
High-speed units*	1	2

* For speeds over 3,600 rpm or pitch-line velocities over 4,000 fpm. It may be desirable to use a lubricant one grade higher or lower than those specified, depending on operating conditions.

ing, the lubrication offers a different problem. Also, since the worm is generally made from steel and the worm wheel from bronze, an extreme-pressure lubricant may not prove to have the same beneficial effect since welding action between steel and bronze is not apparent. In general the best result in wormgear lubrication is obtained with a steam cylinder oil or an oil containing 3 to 10 per cent acidless tallow. The American Gear Manufacturers Association makes recommendations as indicated in Table 15-3.

Table 15-3. Cylindrical- and Double-enveloping Wormgear Units

Worm centers	Worm speed up to rpm	Ambient temp., °F		Worm speed above rpm*	Ambient temp., °F	
		15–60	50–125		15–60	50–125
Up to 6″ inclusive:						
Cylindrical worms	700	7 comp.	8 comp.	700	7 comp.	8 comp.
Double-enveloping worms	700	7 comp.	8A comp.	700	8 comp.	8 comp.
Over 6″ centers up to 12″:						
Cylindrical worms	450	7 comp.	8 comp.	450	7 comp.	7 comp.
Double-enveloping worms	450	8 comp.	8A comp.	450	8 comp.	8 comp.
Over 12″ centers up to 18″:						
Cylindrical worms	300	7 comp.	8 comp.	300	7 comp.	7 comp.
Double-enveloping worms	300	8 comp.	8A comp.	300	8 comp.	8 comp.
Over 18′ centers up to 24″:						
Cylindrical worms	250	7 comp.	8 comp.	250	7 comp.	7 comp.
Double-enveloping worms	250	8 comp.	8A comp.	250	8 comp.	8 comp.
Over 24″ centers:						
Cylindrical worms	200	7 comp.	8 comp.	200	7 comp.	7 comp.
Double-enveloping worms	200	8 comp.	8A comp.	200	8 comp.	8 comp.

* For wormgears of either type operating at speeds above 2,400 rpm or 2,000 fpm, rubbing speed may require forced-feed lubrication. In general, a lubricant of lower viscosity than shown in Table 15-3 may be used with a forced-feed system.

The pour point of any lubricant selected should be below the lowest operating temperatures expected.

Where heavy loads are encountered in all types of gears except wormgearing and hypoid gears, the so-called mild extreme-pressure lubricant should be employed as an additional insurance against scoring and incipient or initial pitting. The viscosity of the extreme-pressure lubricant should generally be equal to that recommended for straight mineral oils. AGMA standard 252.01 deals with mild extreme-pressure lubricants for industrial enclosed gearing.

Hypoid gears, Spiroid®* gears, Planoid®* gears, and Helicon®* gears all have a greater sliding action than any other gearing except wormgearing, and in addition both members are generally made from steel; therefore, an extreme-pressure lubricant should be used for best results at least during the "break-in" or "run-in" period, after which a regular mild extreme-pressure lubricant will often be satisfactory.

15-3. Gear Wear and Failures Caused by Inadequate Lubrication. All gears are to some extent dependent on lubrication. Metallic gears are particularly dependent on lubrication, and the life of nonmetallic gears is increased considerable by the use of the proper type and volume of lubricant.

Metallic gears operating dry will wear out quickly. The life of nylon gears or gears of comparable material is governed more by breaking strength than by wear; however, the life of the gear teeth prior to breaking is considerably reduced when no lubricant is provided.

* Registered trade-marks of the Spiroid Div., Illinois Tool Works, Chicago, Ill.

Wear is a relative term and must be used in our terminology with a great deal of discretion. The American Gear Manufacturers Association[3] defines *gear-tooth wear* as "loss of material from the contacting tooth surfaces resulting from the friction of metal-to-metal contact or the abrasive action of foreign matter passing thru the gear mesh." It has been said: "A gear wears out until it wears in after which it never wears out." After a gear has worn in, it will have an indefinite life, provided it is properly lubricated with a clean lubricant of the proper viscosity and film strength, and operating within its rated capacity.

Destructive wear is a deterioration of the tooth surface or a change in the tooth shape that may appreciably shorten the life or impair the smoothness of transition from one tooth to the other. The types of wear or gear-tooth failures for which lubrication or lack of lubrication is generally considered as the main cause of trouble are:

1. *Abrasive wear*, a surface injury caused by too thin or low-viscosity lubricant; a lubricant of a satisfactory viscosity at normal operating temperatures but containing abrasives in suspension from scale and sand not properly removed from the castings or gear blanks, etc.; impurities of the oil from the surrounding atmosphere or burrs detached from tooth surfaces or other machined surfaces.

2. *Scratching*, an aggravated form of abrasive wear. It shows up as scratch lines or tears of the metal in the direction of sliding. It may be caused by foreign particles brought up into the mesh by the lubricant or material embedded in the tooth faces or weld splatter, etc., loosened by the lubricant and vibration and carried into the mesh by the lubricant.

3. *Scoring*, scuffing, seizing, galling denoting essentially the same type of failure and generally caused by tearing out of small particles welded together as a result of metal-to-metal contact, and may also be caused by loading beyond the design limit for the type of lubricant used. A mild extreme-pressure lubricant will often prevent this type of failure because of its ability to prevent welding.

4. *Corrosive wear*, a surface deterioration caused by moisture or other contaminants such as excessive acid in the lubricant. Rusting due to contamination with water from condensation in localities where the temperature fluctuates widely. Some extreme-pressure lubricants are corrosive in action and may cause corrosive wear.

Where gear-tooth loads are high, the best insurance against abrasive wear is to select a heavy-bodied lubricant. Tests have proved that the use of a 600 SSU lubricant as against a 400 SSU lubricant has increased the spalling resistance by 100 per cent and reduced the efficiency by only 0.4 per cent.

Under load conditions requiring lubricants to have powerful antiwear additives, the character of these additives can influence metallurgical failures. Under moderate loads, and with the use of mild antiwear oils, surface deterioration and distortion due to friction are relatively insignificant. Laboratory and field experience have proved that improper selection of the type of additive as well as quantity of the additive employed for specific operating conditions can shorten fatigue life and cause premature surface failure such as spalling.

Certain types of tooth design make lubrication more of a problem. Enlarged-addendum or all-addendum pinions are subject to higher sliding velocities for the same rpm when compared with equal-addendum gearing, and since greater sliding is involved more attention must be given to proper lubrication.

Under condition of heavy load, employing high-hardness gear teeth of, for example, the carburized type in combination with high sliding velocity such as experienced in hypoid gears, lack of effective lubrication can result in high burnish and eventually may develop into a large number of short parallel ridges across the tooth surface in the direction of the sliding and in a relatively short time will make the gears inoperative.

15-4. Methods of Retaining Lubricants in Gearboxes. The design engineer should consider the method of retaining the lubricant inside the gear enclosure just as much a part of the lubrication as the lubrication itself, since, if adequate and effective means are not provided for retaining the lubricant, the gear drive will not function to the satisfaction of the ultimate user.

There are many types of oil seals and oil-seal arrangements; therefore, the designer should select the type most suitable for the application.

Radial Seal. Figure 15-11 shows two types of widely used and effective mechanical-type oil seals. The sealing element is generally made of a pliable material such as impregnated tanned leather, silicones, synthetic rubber such as neoprene, Buna N, etc. The sealing element requires some means of "hugging" the shaft to prevent oil leakage due to wear of this element, etc., generally accomplished by the use of a carefully selected garter-type spring designed to provide the right tension to seal effectively and yet not create high enough friction to cause the shaft and the sealing element to overheat and become brittle. An important point for the designer to consider when employing this type of seal is that the shaft portion on which the seal rides must be smooth. To specify a ground surface finish is not enough. It has been found that for best results the shaft surface at the seal should have a surface finish of 16 microinches or less. The seals must be handled with care. It is important that the sealing

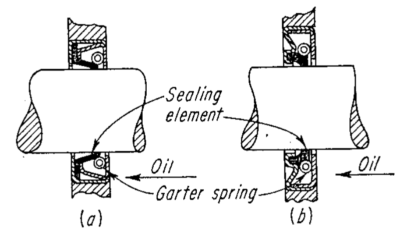


FIG. 15-11. Radial-type oil seal.

element be free of nicks and surface imperfections at the sealing surface, as otherwise it cannot prevent oil leakage. Normal care should be exercised when assembling these seals on their respective shaft. Where they are to be pushed "down" over keyways or sharp edges, it is important to protect the element by the use of a smooth sleeve or cover the sharp edges of the keyway by a material such as "scotch tape" or its equivalent. The sealing lip contacting the shaft should for best result be coated with a good grade of grease.

Face-contact General-purpose Seal. Numerous variations are available in regard to spring arrangements, means of driving the rotary part of the seal, as well as methods of attaching the seal to the shaft. The one main feature in common among all face-type seals is the two lapped sealing surfaces, as they must, in order to seal effectively, be held to an unusually high degree of flatness. Seals of this type will seal effectively up to 200 psi (see Fig. 15-12).

Concentric Radial Oil-groove Seal. This seal is extensively used in both geared speed reducers and increasers and other apparatus where design conditions justify. It is of simple, yet effective, design. It is almost invariably made as an integral part of an end cap. The radial clearance between shaft and seal should be relatively small and in the order of 0.006 to 0.012″, depending on the shaft diameter. The seal may be made from practically any material, but in order to prevent possible scoring of the shaft, in case of bearing failure, a material such as aluminum or bronze may be preferred. The radial grooves should be of sufficient depth and width to carry off any oil leakage along the shaft and provided with oil-drain back holes of sufficient size and number to provide free flow back into the gear case or oil reservoir (see Fig. 15-13).

15-5. Methods of Cooling Gear Lubricant. As indicated in Art. 15-2, the lubricant viscosity is to a considerable extent influenced by temperature; therefore, the designer must give proper consideration to the design of gearboxes in regard to thermal capacity and methods by which the lubricant may be kept at a suitable temperature in order to lubricate gears and bearings effectively and economically. In speed-reduction gear drives where the pitch-line velocity is relatively low, the design surface

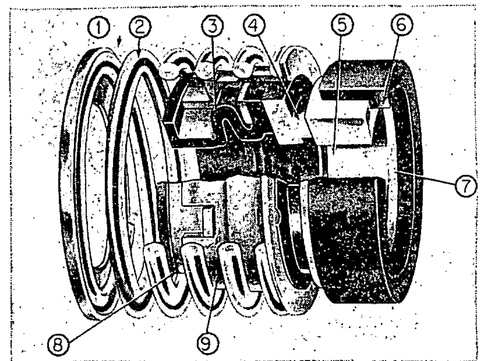


FIG. 15-12. Face-contact general-purpose seal. (1) Spring retainer. (2) Spring. (3) Synthetic rubber bellows. (4) Sealing washer. (5) Lapped sealing faces. (6) Seat ring. (7) Floating seat. (8) Driving band. (9) Driving-band pin.

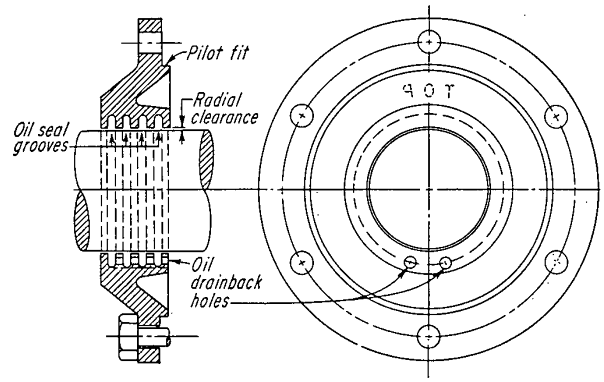


FIG. 15-13. Combination end cap and radial oil-groove seal.

area may be entirely adequate to dissipate the heat generated by the churning of the oil, the friction losses in the bearings, together with the sliding and rolling losses at the gear mesh. Chapter 14 deals with gear efficiency and may be consulted in determining the required heat removal where really close values are required. Chapter 13 discusses *thermal ratings* of gears.

Oil coolers or heat exchangers should in general be selected with a liberal safety margin. It is recommended that they be based on the maximum expected ambient temperature as well as the highest anticipated load condition and in addition a factor to compensate for deposit of impurities in the tubing and/or plate-type cooling grids. For the purpose of selecting the heat exchanger or cooler it is generally sufficient to assume an efficiency loss of 2 per cent for each gear reduction when spur, helicals, herringbone, bevels, and spiral bevels are employed. For wormgearing the efficiency loss for the selection of a suitable cooler may be assumed to be 1 − [100 − (gear ratio/2)]; where closer determinations are required refer to AGMA standards.[9,10] The American Gear Manufacturers Association[11] gives formulas and experience curves for determining the natural thermal capacity of gear drives.

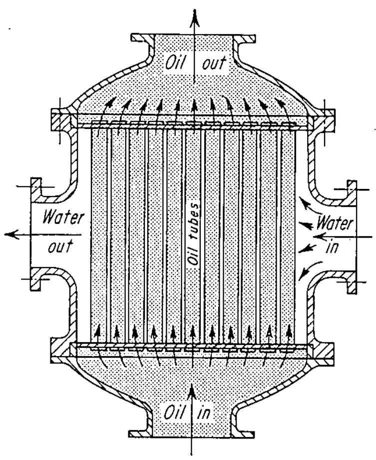


FIG. 15-14. Tube-type oil cooler.

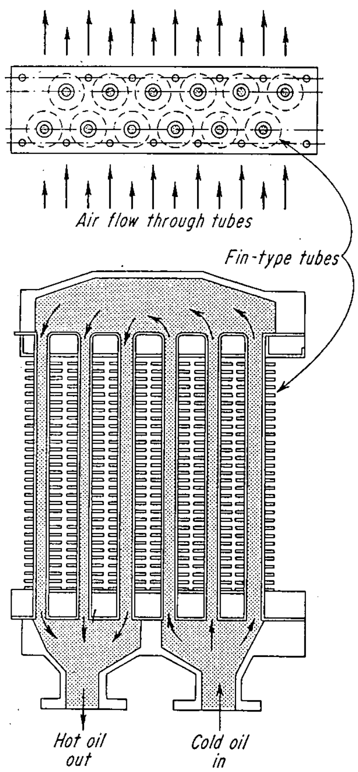


FIG. 15-15. Air-to-oil cooler.

There are many makes of oil coolers or heat exchangers. The basic principle of all types of cooling devices is the same. Their purpose is to act as a self-contained unit through which two liquids may flow or one liquid and an air stream separated from one another by metal tubing or walls. One liquid is hot and the air or the other liquid cool. While the hot liquid is going through the exchanger the cool liquid or air surrounds the hot liquid, tending to create an equilibrium of temperatures; in other words the hot liquids become cooler and the cool liquid or air becomes warmer.

The cooling medium in gear-unit cooling systems is generally water; however, where water is not generally available or is scarce, air is circulated by fans over and around the cooling tubes and/or plates. Air coolers in general take on larger proportions when compared with "water-to-oil" coolers.

Figure 15-14 illustrates schematically the principle of a water-to-oil cooler.

A typical schematic general arrangement of a cooling system for gear drives is illustrated in Art. 15-1 as Fig. 15-5.

Figure 15-15 illustrates a typical general-purpose type of air-to-oil cooler.

This chapter has dealt with gear-unit lubrication in general, and it is intended to be of aid to gear-unit designers as well as users.

Research on gears and gear lubrication is continually being pursued by gear manufacturers and major lubricant suppliers; therefore, for full information as to new developments, it is recommended that the literature of the American Gear Manufacturers Association, One Thomas Circle, Washington 5, D.C., and the American Society of Lubrication Engineers, 343 South Dearborn Street, Chicago 4, Ill., be referred to.

15-6. Aircraft-gear Lubrication. There is a steady increase in the amount of gearing used in aircraft and space-vehicle designs. Extreme lubrication problems exist in this field, and in view of this, some general design criteria are included to cover some of the peculiarities in this field.

Amount of Oil. In a small aircraft gear unit, the amount of oil is the least amount that can be pumped through an orifice in the gearbox and maintain a reasonable pressure. Because of the hazard of clogging with lint or small specks of scale or other foreign materials, a nozzle of 0.030″ diameter is generally considered a minimum. At the maximum predetermined operating temperature, a minimum pump pressure of 25 psi is generally considered acceptable. These two limits can be used to obtain the minimum pump capacity and oil quantity required.

In larger gearboxes, the amount of oil is a function of the power transmitted. For spur, helical, and bevel gears (all coplanar axes) Table 15-4 may be used as an approximate guide for the amount required provided the amount is not below the minimum discussed above.

Table 15-4. Amount of Oil per Gear Reduction for Aircraft Gears

*Ratio of hp to gal per min**	*Comment*
400 hp per gpm	A generous supply. Temp. rise should be under 50°F
800 hp per gpm	Probably OK if gears and bearings have good efficiency. Temp. rise should be under 75°F
1,200 hp per gpm	A lean supply. Use only for a high-efficiency design. Temp. rise may be under 100°F
1,600 hp per gpm	A starvation supply. Must be a very high efficiency unit. Temp. rise may exceed 100°F

* In *double*-reduction gearing, it is assumed that losses double; therefore, double the transmitted horsepower before dividing by the total oil flow.

Although Table 15-4 is intended primarily for aircraft, it indicates a good principle to use in checking high-horsepower marine and industrial units. The general rule of thumb of 0.5 gal per in. of face for industrial gearing may be too lean for many high-capacity gear units.

When wormgears, hypoids, Spiroids, and other offset-axis gears are used in aircraft, the efficiency is quite variable (depending on ratio and geometry of drive). In these cases, it is necessary to make a heat-loss calculation for the gears and bearings and then evaluate or calculate the oil flow to keep the temperature within allowable limits.

Application of Oil. The smaller aircraft gearboxes (100 hp or less) may get by with a splash system and air cooling when employing a "finned" box even with pitch-line speeds up to as high as 5,000 fpm and rotational velocities up to 8,000 rpm.

Somewhat larger gear units (up to about 300 hp) may operate satisfactorily using oil nozzles directed at the gear webs or the gear mesh, or both. An oil mist is violently circulated in the gearbox and several gears and bearings having no direct nozzle will still get adequate lubrication and cooling.

The more powerful gear drives (500 to 5,000 hp) tend to require a nozzle to each mesh and each bearing. The teeth will often heat up to as much as 500°F even with an incoming oil temperature of 200°F and with a generous oil flow directed to the web, hub, and/or gear mesh.

The subject of where to place the oil nozzle on a critical gear mesh has been debated for years by experts in the field. In the opinion of Darle W. Dudley,[12] based on experience gained from numerous experiments and tests in aircraft work and from discussion with engineers from various aircraft concerns throughout the country as well as abroad:

1. In most cases, a nozzle can be developed so that its spray pattern and trajectory are satisfactory for location on either the *incoming* or *outgoing* sides of the mesh.
2. *Cooling* is often somewhat more effective on the outgoing side. *Lubrication* is somewhat more effective on the incoming side.
3. Closely controlled scoring tests with lean oil flows tend to show the highest load capacity with oil on the incoming side.
4. Spur gears with 1" or more of face width and 10,000 fpm or more pitch-line velocity are sensitive to oil trapping when the nozzle is on the incoming side and may fail because of undue vibration caused by oil "wedging" and rough running.

Table 15-5. Aircraft Gear Oils

Type	Military Spec.	Viscosity, cs		Top limit temp., °F	Pour point (max.), °F	Application
		100°F	210°F			
Grade 1005..	Mil-0-6081B	5.0	1.7	175	−75	Jet-engine accessory gears. Moderate power
Grade 1010..	Mil-C-6081B	10.0	2.5	175	−70	Jet-engine accessory gears. Moderate power
Grade 1065..	Mil-0-6082B	105	12	175	0	Aircraft reciprocating engines. Accessory and power gears
Grade 1100..	Mil-0-6082B	260	21	175	+10	Same as above
Synthetic....	Mil-L-7808D	14.2	3.6	300	−75	Accessory gears, main power gears
Synthetic....	(Turbo 35) none	38	7.9	300	−65	Heavier diester than 7808 used in turbo-prop engine and accessory and power gears
Synthetic....	(Target) Mil-L-9236A	300 max.	10 min.*	400	−75	Would be used as a 400°F jet engine and gear lubricant
Synthetic....	Mil-L-25336	12.7	3.3	300	−75	Same as 7808. Used where more load-carrying ability is required
Gear oil.....	Mil-0-6086 Grade L	31	5	175	−40	Gearboxes where extreme-pressure properties are required. Helicopter gearboxes, rocket gears, radar gears
	Grade M	70	8	175	−20	

* 3.0 min at 400°F.

GE-1024.150

Kinds of Oil. Well-engineered aircraft gears continue to amaze gear experts by their ability to run on thin oils having relatively poor load-carrying capacity. Thin oils are needed in aircraft work to permit starting at temperatures as low as −70°F and to keep oil churning losses and oil pumping losses to a minimum.

The ability of well-engineered aircraft gearing to tolerate marginal lubricant stems from several things:

1. The tooth surface finish is exceptionally smooth and often as fine as 20 microinches (rms) or better. Surface hardness is frequently 60 Rockwell C or higher.
2. Accuracy of spacing, lead, and profile is held very close. (Tooth-to-tooth spacing is usually within 0.0002″.)
3. Profile and helix modifications are used wherever appreciable deflections occur.
4. Gearboxes and oil systems are kept extremely clean and free from foreign material. (Filters as fine as 10 microns or finer are often used.)

The kinds of oil most frequently used in aircraft gear work are shown in Table 15-5 These oils are generally made to meet a military specification. The specification covers properties of the fluid and certain performance tests that the fluid must meet. The exact molecular composition is not specified. This makes it possible for different oil companies to supply fluids of somewhat different chemical composition to meet a given specification. Users are sometimes puzzled that a drum of oil bought to a given military specification from one oil company will not behave quite the same as a drum of oil from another company.

15-7. Grease Lubrication of Aircraft Gears. Most of the gears used in aircraft actuators operate intermittently and can generally get by using grease lubrication. Small instrument and radar gears also use grease lubrication. Amounts of power are low. The gears may or may not run intermittently.

When very low speed gears are lubricated, the grease will stick to the ends and back sides of the teeth. As the grease is rubbed off the working side of the tooth, a tiny amount of oil will bleed out of the grease mixture and seep around to keep the working side of the tooth wet. In high-speed jobs that tend to fling the grease clear of the teeth, it is necessary to have the gear casing so constructed that the grease that is flung off eventually shakes back into the path of some gear and is then tracked back into the meshes so as to keep some oil on the teeth.

Technically a *grease* is a more or less solid mixture of an oil in a soap. The lubrication job is done by the small amounts of oil that bleed out of the soap and wet the rubbing surface.

Table 15-6. Aircraft Greases

Military Spec.	Oil type	Thickener	Temp. range, °F	Applications
Mil-G-3278A......	Diester	Lithium soap	−65 to 250	General-purpose lightly loaded gear drives
Mil-G-7118.......	Diester	Lithium soap	−65 to 250	Aircraft gears and actuator screws (more load-carrying ability than Mil-G-3278A)
Mil-G-7187.......	Petroleum or diester	Lithium soap plus graphite	−40 to 200	Starter gear and selector-valve gears when frequent lubrication not possible
Mil-L-3545........	Petroleum	Sodium soap	−20 to 300	High-speed high-temp. application
Mil-G-25013A	Silicone	Nonsoap	−65 to 400	High-temp. aircraft actuator, gearboxes, couplings

Grease lubrication of gears works when the rubbing surfaces can manage to keep wet with oil and when there is no particular cooling problem.

Table 15-6 shows some typical aircraft greases. The ones shown have been widely used on aircraft gears.

REFERENCES

1. Himes, W. H.: "Splash Lubrication Fails at Subcritical Speeds," *Machine Design*, December, 1932, pp. 27, 28, 53.
2. American Gear Manufacturers Assoc.: "Standard Nomenclature—Gear Tooth Wear and Failure," AGMA 110.02 (ASA B6.12—1954).
3. American Gear Manufacturers Assoc.: "Lubrication of Industrial Enclosed Gearing," AGMA 250.02.
4. American Gear Manufacturers Assoc.: "Data on Extreme Pressure Lubrication," AGMA 254.04.
5. Richardz, F.: Lubrication of Industrial Gear Drives, *Lubrication Eng.*, Vol. 4, pp. 25–27, February, 1948.
6. Shipley, Eugene E.: Design and Testing Consideration of Lubricants for Gear Application, *Lubrication Eng.* April, 1958, pp. 148–151, 191.
7. *Chem. & Met. Eng.*, Vol. 36, p. 618, 1929.
8. The Texas Company: Viscosity and Viscosity Index Publication, *Lubrication*, Vol. 31, No. 4, April, 1945.
9. American Gear Manufacturers Assoc.: "Practice for Single- and Double-reduction Cylindrical-worm and Helical-worm Reducers," AGMA 440.03.
10. American Gear Manufacturers Assoc.: "Practice for Double-enveloping Worm Gear Speed Reducers," AGMA 441.02.
11. American Gear Manufacturers Assoc.: "Thermal Capacity for Helical and Herringbone Gear Speed Reducers," AGMA 423.01.
12. Dudley, Darle W.: "Practical Gear Design," pp. 305–306, McGraw-Hill Book Company, Inc., New York, 1954.

INDEX

GE-1024.153

DARLE W. DUDLEY . . .

. . . is currently Chief of Gear Technology, SOLAR, Division of International Harvester Company, and was Manager, Mechanical Transmissions, at Mechanical Technology, Inc., Latham, New York. Previously he was Manager, Advance Unit of the Gear Engineering Subsection, Medium Steam Turbine, Generator and Gear Department, General Electric Company, Lynn, Massachusetts. From 1953 to 1957 he was Section Head of the latter company's Development Section of the Gear Engineering Division where he was responsible for gear development and testing, and for the design and development of special tools and machinery for the manufacture of gears. In this capacity, he was also responsible for the design of special product gearing which included gearing for aircraft gas turbine engines, gun control, and other special military applications.

In 1958, Mr. Dudley received the Edward P. Connell award in recognition of his outstanding accomplishment in gear work. He has since been regarded as one of the outstanding gear experts in the United States and abroad.

The author of many technical articles, Darle Dudley is a Registered Professional Engineer in the state of Massachusetts, and is a member of the American Society of Mechanical Engineers and the American Gear Manufacturers' Association.

McGRAW-HILL BOOK COMPANY

1221 Avenue of the Americas New York, N. Y. 10020

0-07-017902-6