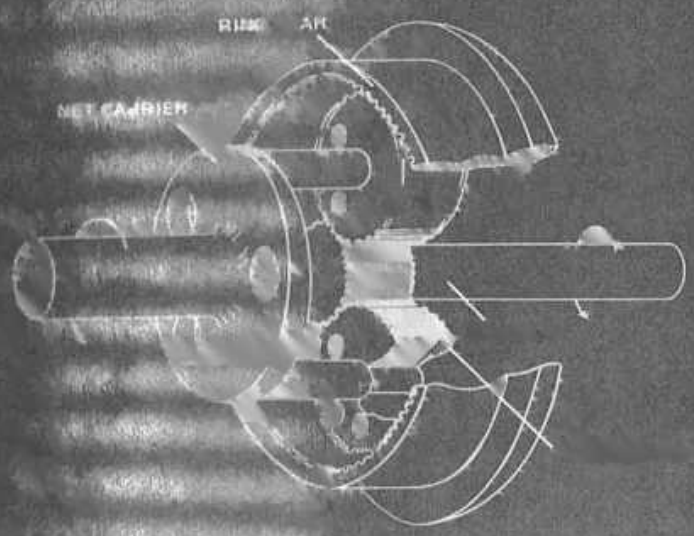
MECHANICAL ENGINEERING

GEAR DRIVE SYSTEMS

DESIGN AND AP ON



PETER LYNWAN ER

LIBRARY OF CONGRESS
00011273579

about the book . . .

This outstanding reference provides th[illegible] and theoretical information—*with over 250 detailed illu[illegible] figures, and tables*—needed to design, manufacture, and operate highly reliable and efficient, state-of-the-art gear drive systems, emphasizing parallel shaft and planetary units with spur and helical gearing.

Incorporating standards established by the American Gear Manufacturers Association, *Gear Drive Systems: Design and Application* features

- *a systems approach*—considering all components that affect gear performance in the specification, installation, operation, and maintenance of the unit
- *in-depth treatment*—spanning the full spectrum of topics, including lubrication, bearings and seals, manufacturing methods, testing and monitoring, failure analysis, gearbox economics, and many more
- *the latest advances*—including high-speed gearing, hardened and ground gear technology, and planetary gear design and application

This thorough work is the essential reference for mechanical, design, and plant engineers; installation, operation, and maintenance technicians; and purchasing agents in such diverse industries as turbo-machinery, automotive, process, refinery, marine, steel, construction, and mining, among others. In addition, this work is an excellent source for mechanical engineering courses dealing with gear design and application.

about the author . . .

PETER LYNWANDER is the Manager of Engineering and Sales at American Lohmann Corporation, Hillside, New Jersey. His experience encompasses 25 years of design, analysis, and development of mechanical systems, and includes establishing computer programs for the analysis and optimization of gear configurations. Mr. Lynwander received the B.S. and M.S. degrees (1959, 1964 respectively) from the University of Bridgeport, Connecticut, and is a registered Professional Engineer. He is Vice Chairman of the Aerospace Gearing Committee of the American Gear Manufacturers Association, and is active on several other committees of this organization. Mr. Lynwander has authored numerous publications concerning gearing, seals, and clutches.

Printed in the United States of America

ISBN: 0–8247–1896–8

marcel dekker, inc./new york · basel

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 3 of 71

GEAR DRIVE SYSTEMS

DESIGN AND APPLICATION

MECHANICAL ENGINEERING

A Series of Textbooks and Reference Books

EDITORS

L. L. FAULKNER
Department of Mechanical Engineering
The Ohio State University
Columbus, Ohio

S. B. MENKES
Department of Mechanical Engineering
The City College of the
City University of New York
New York, New York

1. Spring Designer's Handbook, *by Harold Carlson*
2. Computer-Aided Graphics and Design, *by Daniel L. Ryan*
3. Lubrication Fundamentals, *by J. George Wills*
4. Solar Engineering for Domestic Buildings, *by William A. Himmelman*
5. Applied Engineering Mechanics: Statics and Dynamics, *by G. Boothroyd and C. Poli*
6. Centrifugal Pump Clinic, *by Igor J. Karassik*
7. Computer-Aided Kinetics for Machine Design, *by Daniel L. Ryan*
8. Plastics Products Design Handbook, Part A: Materials and Components, *edited by Edward Miller*
9. Turbomachinery: Basic Theory and Applications, *by Earl Logan, Jr.*
10. Vibrations of Shells and Plates, *by Werner Soedel*
11. Flat and Corrugated Diaphragm Design Handbook, *by Mario Di Giovanni*
12. Practical Stress Analysis in Engineering Design, *by Alexander Blake*
13. An Introduction to the Design and Behavior of Bolted Joints, *by John H. Bickford*
14. Optimal Engineering Design: Principles and Applications, *by James N. Siddall*
15. Spring Manufacturing Handbook, *by Harold Carlson*

16. Industrial Noise Control: Fundamentals and Applications, *by Lewis H. Bell*
17. Gears and Their Vibration: A Basic Approach to Understanding Gear Noise, *by J. Derek Smith*
18. Chains for Power Transmission and Material Handling: Design and Applications Handbook, *by the American Chain Association*
19. Corrosion and Corrosion Protection Handbook, *edited by Philip A. Schweitzer*
20. Gear Drive Systems: Design and Application, *by Peter Lynwander*

OTHER VOLUMES IN PREPARATION

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 8 of 71

GEAR DRIVE SYSTEMS

DESIGN AND APPLICATION

Peter Lynwander
American Lohmann Corporation
Hillside, New Jersey

MARCEL DEKKER, INC. New York and Basel

TJ184
.L94
1983

JUN 28 1983

Library of Congress Cataloging in Publication Data

Lynwander, Peter, [date]
Gear drive systems.

(Mechanical engineering ; 20)
Includes index.
1. Gearing. I. Title. II. Series.
TJ184.L94 1983 621.8'33 83-5278
ISBN 0-8247-1896-8

COPYRIGHT © 1983 by MARCEL DEKKER, INC. ALL RIGHTS RESERVED

Neither this book nor any part may be reproduced or transmitted in any form or by any means, electronic or mechanical, including photocopying, microfilming, and recording, or by any information storage and retrieval system, without permission in writing from the publisher.

MARCEL DEKKER, INC.
270 Madison Avenue, New York, New York 10016

Current printing (last digit):
10 9 8 7 6 5 4 3 2 1

PRINTED IN THE UNITED STATES OF AMERICA

PREFACE

Gear drives are critical components of mechanical systems used in such diverse industries as turbo-machinery, process, refinery, steel, construction, mining, and marine. In all these fields there is a continuing trend toward higher reliability and improved technology in mechanical components. Higher reliability is desired to reduce downtime. In many applications, the cost of one day's lost production due to a gearbox malfunction far exceeds the initial cost of the unit; therefore, in critical installations there is a strong emphasis on conservative design and quality manufacture. In addition to achieving high reliability, mechanical systems must be increasingly efficient to conserve energy. Gear manufacturers are constantly refining their analytical, design, and manufacturing techniques to take advantage of new technologies and provide reliable, efficient gearboxes at minimum cost.

The purpose of this book is to present practical gearbox design and application information to individuals responsible for the specification and operation of mechanical systems incorporating gear drives. Sufficient theoretical information is included to enable the engineer interested in gear analysis and design to understand how gear units are rated and detail gear tooth geometry is defined. The major emphasis is on parallel shaft and planetary units using spur and helical gearing.

In addition to basic data on gear design and manufacture, such subjects as installation, operation, maintenance, troubleshooting, failure analysis, and economics are covered. Material on lubrication systems, bearings, couplings, and seals is presented in order to cover all aspects of gear system operation.

Several new trends in the gear industry, due in part to the emphasis on energy conservation, are discussed.

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 12 of 71

1. As mechanical equipment such as pumps, motors, compressors, turbines, etc. are designed for higher efficiencies, rotating speeds are increased and, therefore, higher speed transmissions are required. High speed gearing characteristics are featured throughout the book.
2. Also, as a result of energy consciousness, there is a tendency to package smaller mechanical systems; therefore, there is a trend developing in the United States toward the use of planetary gear units which are far more compact than parallel shaft designs. Included in the book is a section on planetary gear design and application.
3. In order to achieve the highest load carrying capability in a minimum envelope, case hardened and ground gear tooth designs are finding wide application. This technology is covered in the book.
4. The book attempts to take a systems approach to gearbox application. It has become apparent that gear units, when incorporated into a system of rotating machinery, are susceptible to a variety of problems. All characteristics of the drive system from the driver to the driven equipment, including the lubrication system and accessories, can influence gearbox operation and must be considered in the specification, installation, operation, and maintenance of the unit.

Throughout the book, standards and practices developed by the American Gear Manufacturers Association are referred to. Successful selection, rating, and operation of gearboxes can be accomplished by the use of AGMA publications and the gear designer and user should be familiar with the Standards system. The AGMA is located at 1901 North Fort Myer Drive, Arlington, Virginia 22209.

I would like to thank Mr. Alvin Meyer and Mr. Alan Swirnow for their assistance and comments. I am also indebted to American Lohmann Corporation, Hillside, New Jersey for its support.

Peter Lynwander

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 13 of 71

CONTENTS

Preface iii

1. Types of Gear Drives: Arrangements, Tooth Forms 1
Physical Arrangement Gear Ratio Torque Loading Efficiency Space and Weight Limitations Physical Environment References

2. Gear Tooth Design 21
The Involute Curve Gear Tooth Definitions Gear Tooth Generation Gear Teeth in Action Rolling and Sliding Velocities Helical Gears Internal Gears Measurement Over Balls or Wires Engineering Drawing Format Spline Design References

3. Gearbox Rating 93
Tooth Loads Strength Rating Bending Stress Rating Durability Rating Compressive Stress Rating AGMA Standards for Enclosed Drive Ratings High-Speed Gearing API Standard Sample Rating Case Scoring Flash Temperature Index Scoring Criterion Number Minimum Film Thickness Criterion Shaft Rating Shaft Stresses Keyways Spline Rating References

4. Bearings and Seals 145
Bearings Seals References

5. Lubrication Systems 213
Viscosity Viscosity Index Pour Point Gear Lubricants Rust- and Oxidation-Inhibited Oils Extreme Pressure Gear Lubricants Synthetic Lubricants Lubricant Viscosity Selection Types of Lubrication Systems Pumps Filtration Coolers Oil Reservoir Breather (Vent) Piping References

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 14 of 71

6. Materials and Heat Treatments 245
Hardening Fundamentals Hardenability and the Use of Alloy Steels Steel Quality Processing of Through-Hardened Gears Carburizing Nitriding Forging of Gear Blanks Housing Materials Hardness Testing Nondestructive Testing Magnetic Particle Inspection Liquid Penetrant Inspection Surface Temper Inspection References

7. Manufacturing Methods 267
Hobbing Shaping Lapping Shaving Grinding Honing Other Methods of Gear Tooth Machining Shot Peening Inspection Equipment Functional Inspections References

8. Gearbox Economics 287
Gear Quality Material Costs Effect of Quantity on Costs References

9. Planetary Gear Trains 293
Analysis of Planetary Gear Arrangements Stoekicht Design References

10. Gearbox Installation: Mounting, Alignment, Couplings 325
Couplings and System Alignment Alignment Procedure References

11. Gear Unit Operation: Testing, Startup, Condition Monitoring 335
Testing Special Tests Initial Field Startup Condition Monitoring References

12. Maintenance and Failure Analysis: Scheduled Maintenance Actions 383
Storage Overhaul and Spare Parts Troubleshooting References

Appendix 403

Index 407

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 15 of 71

GEAR DRIVE SYSTEMS

DESIGN AND APPLICATION

1

TYPES OF GEAR DRIVES: ARRANGEMENTS, TOOTH FORMS

The function of a gearbox is to transmit rotational motion from a driving prime mover to a driven machine. The driving and driven equipment may operate at different speeds, requiring a speed-increasing or speed-decreasing unit. The gearbox therefore allows both machines to operate at their most efficient speeds. Gearboxes are also used to change the sense of rotation or bridge an angle between driving and driven machinery.

The gearbox configuration chosen for a given application is most strongly influenced by three parameters:

Physical arrangement of the machinery
Ratio required between input and output speeds
Torque loading (combination of horsepower and speed)

Other factors that must be considered when specifying a gear drive are:

Efficiency
Space and weight limitations
Physical environment

PHYSICAL ARRANGEMENT

The location of the driving and driven equipment in the mechanical system defines the input and output shaft geometrical relationship. Shaft arrangements can be parallel offset, concentric, right angle, or skewed as shown in Figure 1.1. The material presented in this book focuses on parallel offset and concentric designs.

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 18 of 71

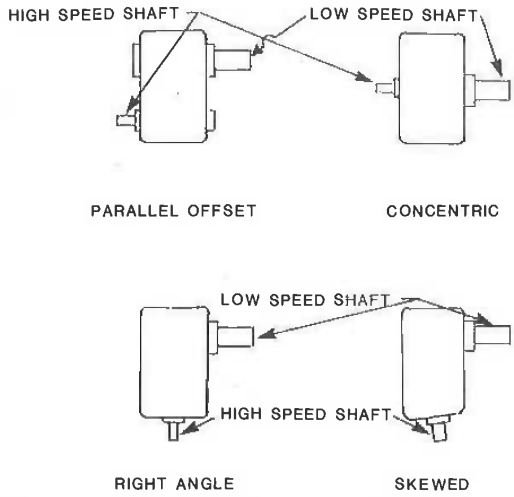


Figure 1.1 Gearbox shaft arrangements.

In the majority of parallel offset units in use, the input and output shafts are horizontally offset; however, vertical offsets are used and any orientation of input to output shaft is possible. Figure 1.2 illustrates a typical horizontally offset parallel shaft gearbox and Figure 1.3 presents a cutaway view of such a unit. In this case there is one input shaft and one output shaft located on opposite sides of the unit. There are many different options available as far as the input and output shaft extensions are concerned. Figure 1.4 shows the various possibilities and presents a system for defining the extensions desired on a gearbox. There may be two inputs driving a single output, such as dual turbines powering a large generator, or two outputs with a single input, such as an electric motor driving a two-stage compressor. Often shaft extensions are used to drive accessories such as pumps or starters.

The minimum amount of offset required is determined by gear tooth stress considerations. The offset of a gearbox incorporating a single mesh, as shown in Figure 1.3, is the sum of the pitch radii of the pinion and gear, otherwise known as the center distance. The pitch radii must be sufficiently large to transmit the system load. An offset greater than the minimum may be required to provide enough space for the machinery incorporated in the system. Figure 1.5 illustrates an accessory drive where the input and output shafts are offset through two meshes to separate the machinery located at these shafts.

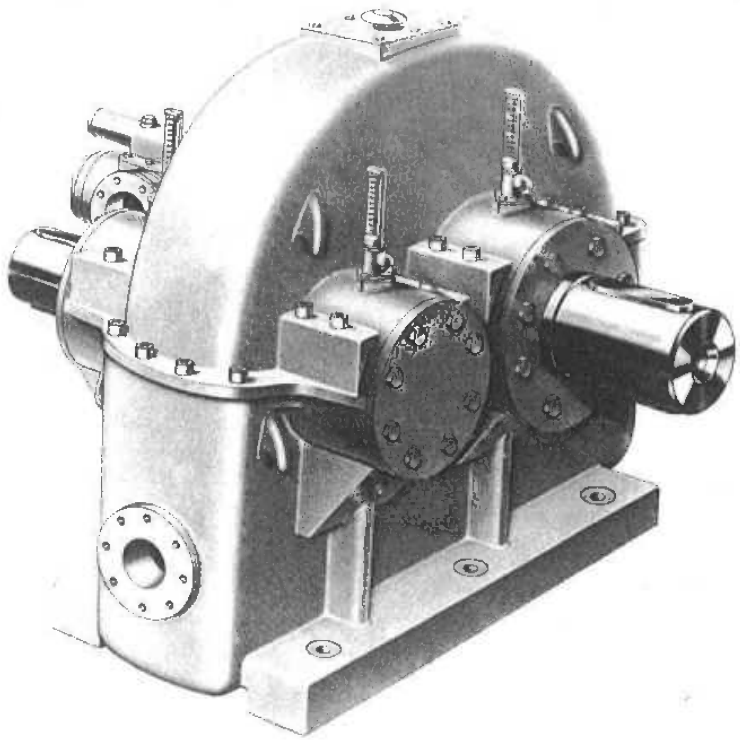


Figure 1.2 Parallel offset gearbox. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

Figure 1.6 illustrates a gearbox with concentric input and output shafts. The driving and driven machinery will therefore be in line. Planetary gearing (described in the next section) has concentric shafts and is used to achieve high ratios in minimum space. It is also possible to package parallel shaft gearing such that the input and output shafts are in line when such a configuration is desired. Figure 1.7 presents external and internal views of a right-angle gear drive.

The gearboxes in Figures 1.3, 1.6, and 1.7 are foot mounted; that is, they are meant to be bolted to a horizontal base through a flange at the bottom of

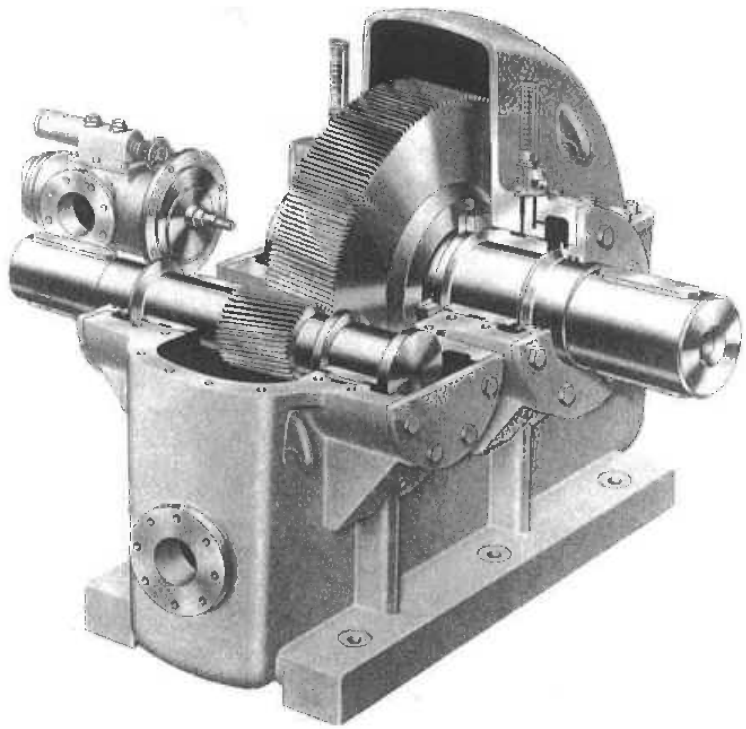


Figure 1.3 Parallel offset gearbox sectional view. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

the gear casing. Although this is the most common design, gearboxes can be mounted in many other configurations and operate in attitudes other than horizontal. Figure 1.8 illustrates a flange-mounted unit. Such a gearbox can be operated horizontally or be vertically mounted on a horizontal base. Vertically operating gearbox designs must have special lubrication provisions to provide lubricant to the upper components in the unit and seal the lower end from oil leakage. Figure 1.9 shows yet another mounting configuration. This unit is shaft mounted with a support arm that is fixed to ground to react the gearbox housing torque.

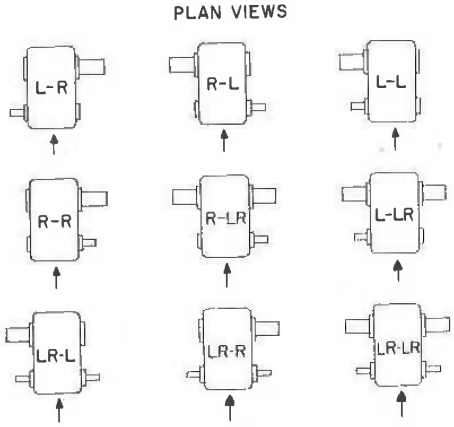


Figure 1.4 Definition of shaft extensions. Parallel shaft-helical and herringbone gear reducers; single, double, and triple reduction. Code; L = left; R = right; arrows indicate line of sight to determine direction of shaft extensions; letters preceding the hyphen refer to number and direction of highspeed shaft extensions; letters following the hyphen refer to number and direction of low speed shaft extensions. (From Ref. 1.)

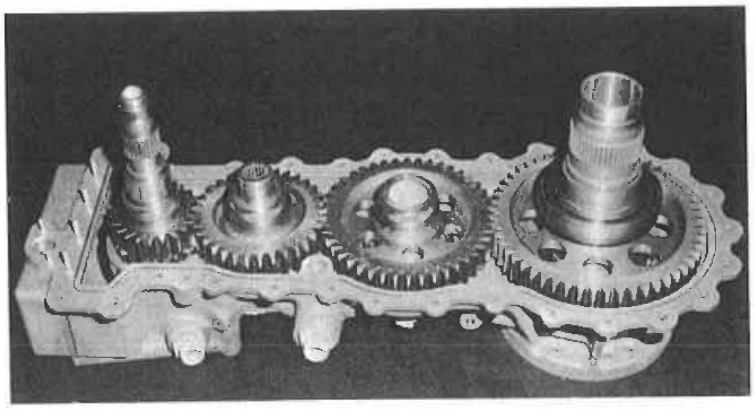


Figure 1.5 Parallel offset accessory drive gearbox. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

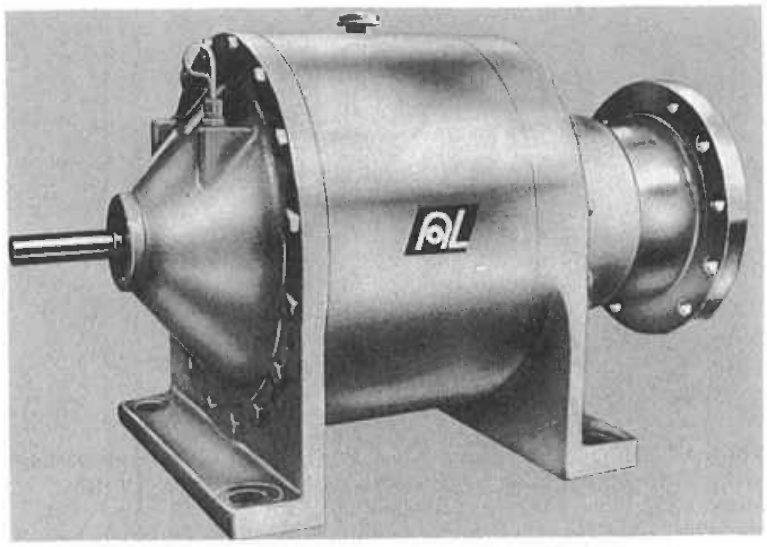


Figure 1.6 Gearbox with concentric input and output shafts. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

GEAR RATIO

There is no limit to the reduction or speed increasing ratio that can be achieved using gearing; however, for high ratios the arrangement of the components can be quite complex. In a simple gear mesh a maximum ratio in the order of 8:1 to 10:1 can be achieved. The amount of speed reduction or increase is simply the ratio of the pitch diameter of the larger gear to the smaller gear. The number of teeth in a gear pair is related to the pitch diameters, so the speed ratio can also be calculated by dividing the larger number of teeth by the smaller. The smaller gear is often called the pinion. To attain a ratio of 10:1, therefore, the gear must be 10 times larger than the pinion and there usually are stress or geometrical limitations on the pinion when this ratio is exceeded. To achieve higher ratios with parallel shaft gearing, stages of meshes are combined as shown in Figure 1.10. This unit has three stages of reduction and achieves ratios on the order of 100:1.

An efficient method of achieving high reduction ratios in minimum space is the use of planetary gearing. This design, completely described in Chapter 9, is illustrated in Figure 1.11. The high-speed sun gear meshes with a number of

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 23 of 71

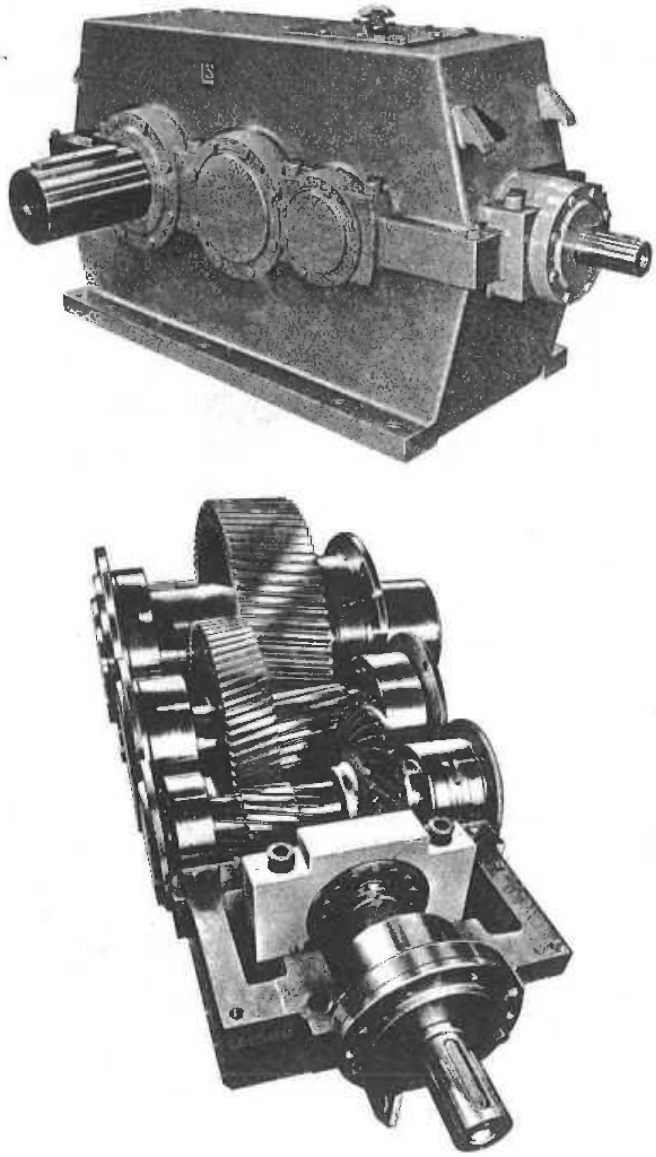


Figure 1.7 Right-angle gear drive. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 24 of 71

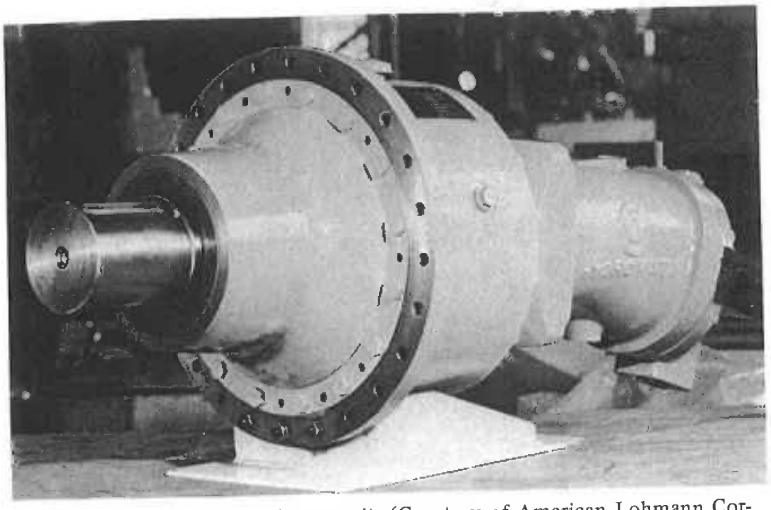


Figure 1.8 Flange-mounted gear unit. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

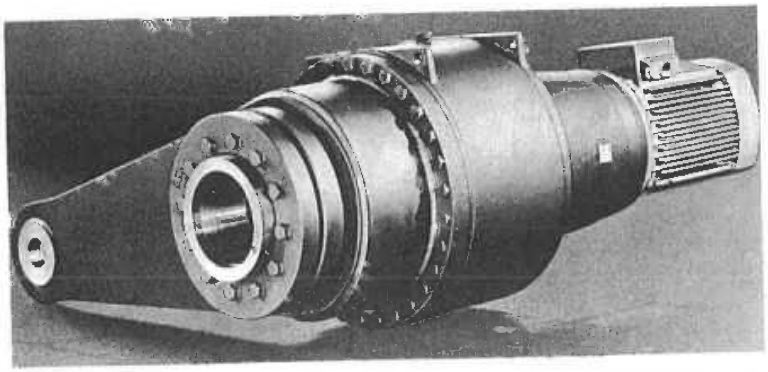


Figure 1.9 Shaft-mounted gearbox. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

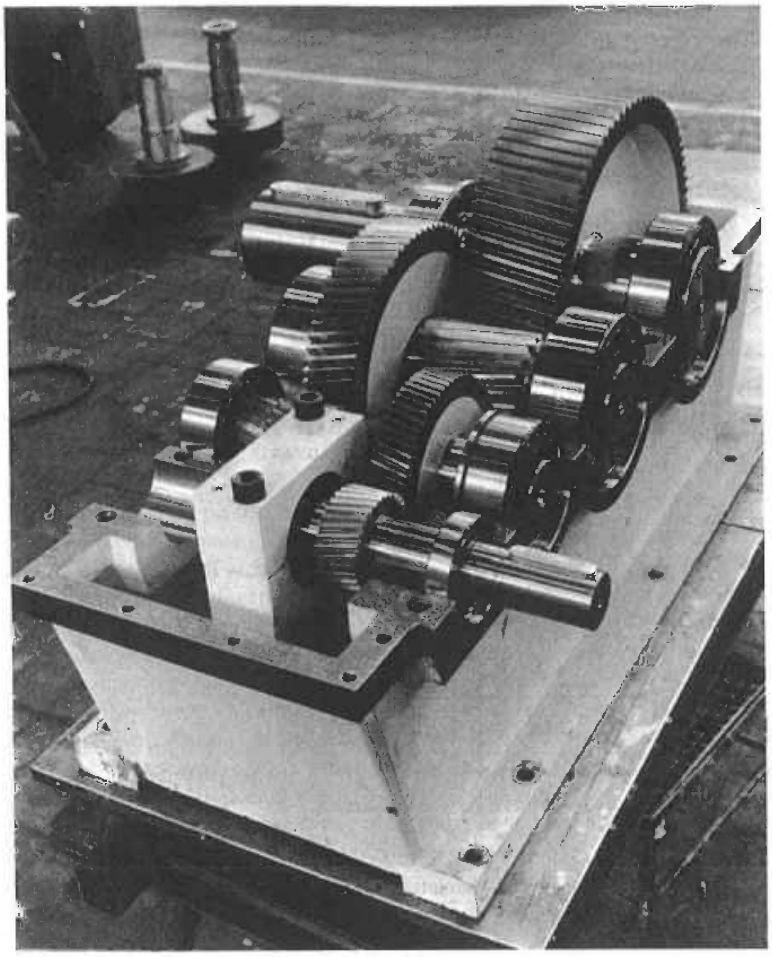


Figure 1.10 Multi-stage parallel shaft gearbox. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

planets, usually three, which in turn mesh with a ring gear. The ring gear has internal teeth. Either the ring gear or the planet carrier rotate at the low speed of the gear set. Occasionally, all three members are connected to rotating equipment. When the low-speed shaft is either the ring gear or the planet carrier, the ratios in a planetary gearset are:

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 26 of 71

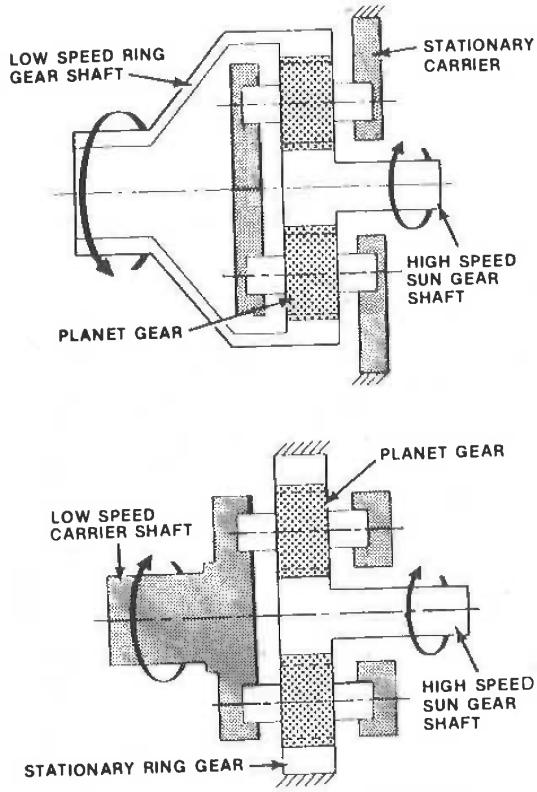


Figure 1.11 Basic planetary gear configurations.

$$\text{Ratio} = \frac{\text{ring gear pitch diameter}}{\text{sun gear pitch diameter}} \quad \text{for rotating ring gear}$$

$$\text{Ratio} = 1 + \frac{\text{ring gear pitch diameter}}{\text{sun gear pitch diameter}} \quad \text{for rotating planet carrier}$$

Because of the multiple load path of planetary gearing the horsepower transmitted is divided between several planet meshes and the gear size can be reduced significantly compared to parallel shaft designs. Planetary stages can be linked together to achieve high ratios, as shown in Figure 1.12. This is a three-stage planetary gear with a ratio of 630:1. The first-stage planet carrier drives the second-stage sun gear and the second-stage carrier drives the third-stage sun gear.

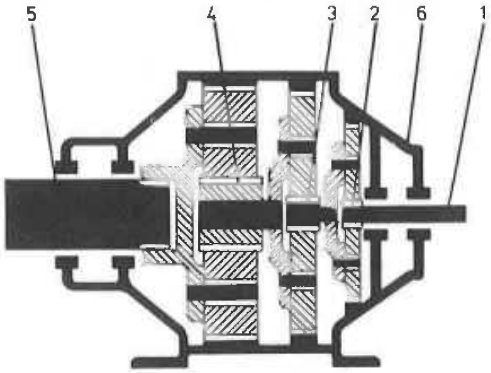


Figure 1.12 Multistage planetary gearbox. High-speed shaft, 1; first planetary stage, 2; second stage, 3; third stage, 4; low-speed shaft, 5; gear housing, 6.

In theory, any parallel shaft or planetary gearbox can be used either as a speed reducer or increaser. There may be details within a gearbox, however, that require modification if such a changeover is made. The same holds true if it is desired to reverse the direction of rotation for which the gearbox was initially designed. For instance, one side of the teeth may have been favored in the finishing process when the initial design was manufactured. If the gearbox is used in such a manner that the initially unloaded face is now loaded, poor tooth performance may result. The direction of rotation of the input shaft with respect to the output shaft depends on the gear design chosen. For a simple parallel shaft gear mesh the sense of rotation will change through the mesh. A planetary arrangement with a stationary ring gear will not change the sense of rotation between input and output, while a rotating ring gear will turn in the opposite sense compared to the sun gear's rotation.

TORQUE LOADING

The size of gearbox required for a given application is dependent primarily on how large the gear pitch diameters and face widths are. These dimensions are determined on the basis of tooth stresses which are imposed by the transmitted tooth load. The tooth load is simply the torque on a given gear divided by the gear pitch radius:

$$\text{Tooth load (lb)} = \frac{\text{torque (in.-lb)}}{\text{pitch radius (in.)}}$$

Torque is calculated from the horsepower transmitted and the speed of the rotating component in question:

$$\text{Input torque (in.-lb)} = \frac{63{,}025\ (\text{hp})}{\text{input rpm}}$$

$$\text{Output torque (in.-lb)} = \frac{63{,}025\ (\text{hp})}{\text{output rpm}}$$

When designing a gearset one cannot consider torque alone. The operating speed of the gears has a significant effect on the design definition. As an illustration of this point, consider a high-speed unit transmitting 2000 hp at 20,000 rpm input. The input torque would be the same as a low-speed unit operating at 2000 rpm input with a transmitted horsepower of 200. On a simple stress basis, if the ratio of both gearboxes were the same, the same gearbox could be used for both applications; however, the high-speed design must differ from the low-speed design in the following respects:

At high speeds, component geometry discrepancies such as tooth spacing error, shafting unbalance, and so on, generate significant dynamic loading, and these dynamic effects must be taken into account in the design process. Also, the components experience high numbers of load cycles and are more prone than low-speed units to fatigue failures. For all of these reasons, high-speed components must be of high accuracy to minimize dynamic problems.

Heat generation within the unit is proportional to speed; therefore, high-speed units usually require pressure jet lubrication systems and external cooling systems. Low-speed units often operate with integral splash lubrication, the heat being dissipated through the gear casing.

The bearing design is strongly dependent on shaft speeds. Low-speed units generally incorporate antifriction bearings, while high-speed industrial gearboxes typically use journal bearing designs.

There is no clear demarcation between low-speed and high-speed gearing. Units with several gear meshes may have some of each. An arbitrary definition sometimes used is that units with pinion speeds exceeding 3600 rpm or pitch line velocities exceeding 5000 fpm are considered high speed [1].

Pitch line velocity is a measure of the peripheral speed of a gear:

$$\text{Pitch line velocity (fpm)} = \frac{\pi\ (\text{pitch diameter, in.})\ (\text{rpm})}{12}$$

The pitch line velocity of a gear is a better index of speed than is rotational velocity, since a large gear operating at a relatively low rpm may experience the same velocity effects as a small gear operating at high rpm. Standard high-speed gear units operate at pitch line velocities up to approximately 20,000 fpm.

Applications exceeding this speed must be considered special and exceptional care must be taken in their design and manufacture. Pitch line velocities of 40,000 fpm have been attained in practice.

Parallel offset or concentric shaft gearboxes incorporate gears with spur, single helical, or double helical tooth forms. The face of a spur gear is parallel to the axis of rotation, whereas a helical gear tooth face is at an angle, as shown in Figure 1.13. The figure illustrates that helical gears have an overlap in the axial direction, which results in the following advantages:

Helical gears have more face width in contact than do spur gears of the same size; therefore, they have greater load-carrying capability.

With conventional spur gearing the load is transmitted by either one or two teeth at any instant; thus the elastic flexibility is continuously changing as load is transferred from single-tooth to double-tooth contact and back. With helical gearing the load is shared between sufficient teeth to allow a smoother transference and a more constant elastic flexibility; therefore, helical gearing generates less noise and vibration than spur gearing.

The disadvantage of helical gearing in relation to spur gearing is that axial thrust is generated in a helical gear, which necessitates the incorporation of a thrust bearing on each helical gear shaft.

To take advantage of the helical gearing benefits described above, yet not generate axial thrust loads, double helical gearing is used (Figure 1.14). The two halves generate opposite thrust loads, which cancel out. When the two helices are cut adjacent to one another with no gap between, the gearing is termed herringbone. Because helical gear thrust is proportional to the tangent of the helix angle, single helical gears tend to have lower helix angles than do double helical designs, where the thrust loads cancel. Typical single helical helix angles are 6 to 15°. Double helical gearsets have helix angles of up to 35°.

Another advantage of double helical gears is that the ratio of face width to pitch diameter in each half can be held to reasonable limits. When the face widths become longer than the pitch diameters in spur or single helical gearing it is difficult to achieve complete tooth contact since thermal distortion, load deflections, and manufacturing errors tend to load the gear teeth unevenly. A double helical gear with a face width/pitch diameter ratio of 1 will have twice the face width of a spur or single helical gear with the same L/D ratio and therefore greater load-carrying capability.

Double helical gearing has two disadvantages. Because the two halves of each gear cannot be perfectly matched, one member of the gearset must be free to float axially. This gear will be continually shifting to achieve axial force equilibrium since the thrust loads of each half will rarely cancel exactly. This shifting can lead to detrimental axial vibrations if tooth geometry errors are excessive. Another potential problem with double helical gearing is that external

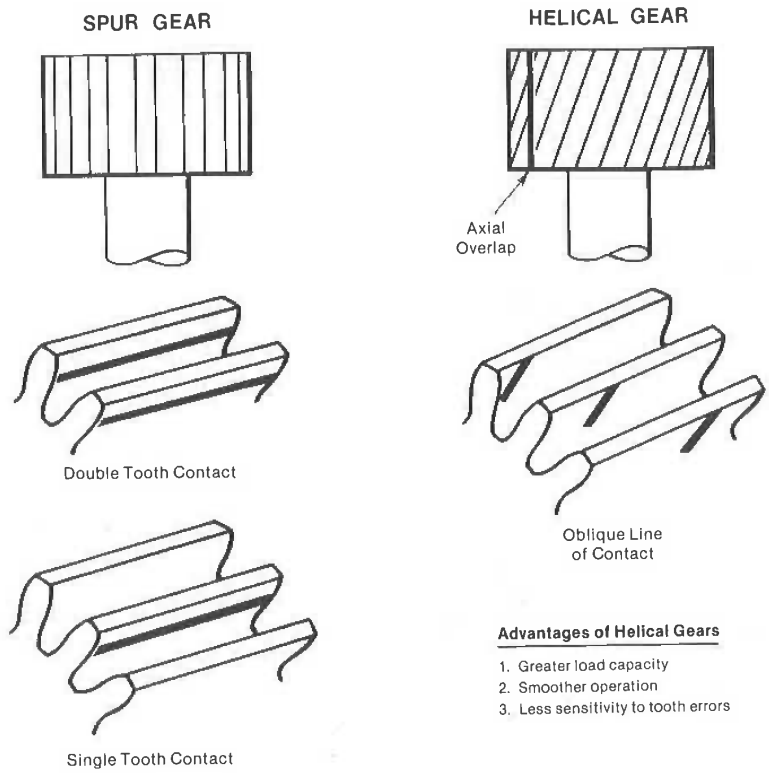


Figure 1.13 Comparison of spur and helical gear teeth.

thrust loads will tend to overload one helix. For instance, if a double helical gear is attached to a gear tooth type of coupling and the coupling locks up axially due to tooth friction, axial loads transmitted through the coupling will be reacted by the teeth of one-half of the gearset. With single helical gears an external axial load will either add to or subtract from the gear tooth load and be reacted by the thrust bearing.

Gear metallurgy, although not mentioned heretofore, is one of the most significant factors in determining gearbox size, since the strength of a gear tooth is proportional to the hardness of the steel. Most gears are in the hardness ranges of approximately Rc 30 to 38 or Rc 55 to 64. The region from Rc 30 to 38 is usually termed "through-hardened," while the range Rc 55 to 64 is almost always "surface-hardened," where the tooth has a hard surface case and a softer inner core. Through-hardened gears are cut by such processes as hobbing,

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 31 of 71

Figure 1.14 Double helical gearing.

shaping, and shaving. Surface-hardened gears are cut and then hardened. They may be used in this state, but the more accurate surface-hardened gears are ground after heat treatment. Spur, single helical, and double helical gearing may be produced by any of the methods noted above. Generally, double helical gearing is through-hardened and cut. It is possible to harden and grind double helical gearing; however, to grind a one-piece double helical gear a large central gap is required between the two helices to allow runout of the grinding wheel. Gears can be ground in halves and then assembled, but this presents serious alignment and attachment problems.

To achieve minimum envelope and maximum reliability, the latest technology utilizes single helical, hardened, and precision ground gearing. With single helical gears the thrust load axially locates the gear shaft against the thrust bearing. Bearing design has progressed to the point where thrust loads are routinely handled either by hydrodynamic tapered land or tilting pad configurations or an antifriction thrust bearings. Because case-hardened gears have maximum

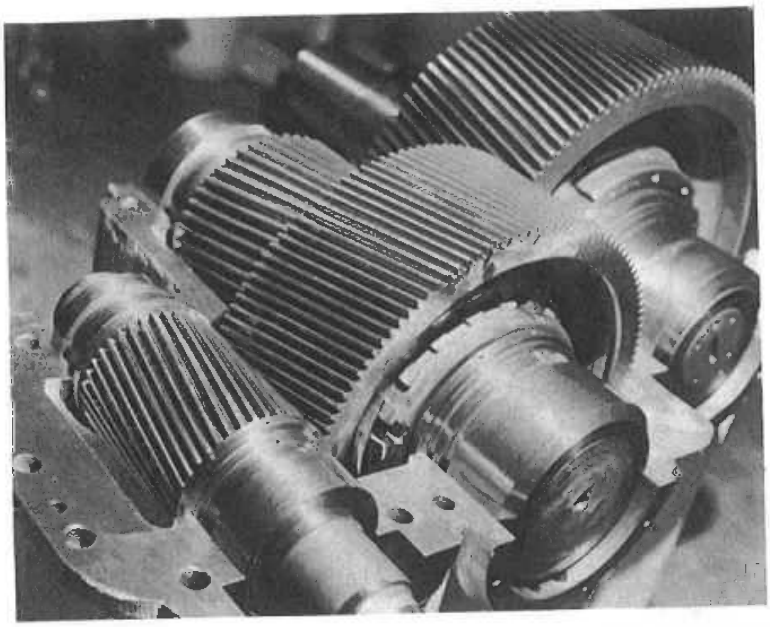


Figure 1.15 High-speed single helical hardened and ground gearset. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

load-carrying capacity, gear size can be minimized; therefore, the ratio of face width to diameter of a single helical gear can be held to reasonable limits. Pitch line velocities are minimized, reducing dynamic effects. Also, the bearing span with single helical gears is short, resulting in lower elastic deflection. Figure 1.15 illustrates a generator drive gearbox with two stages of single helical gearing. This unit transmits 4500 hp at an input speed of 14,500 rpm. The high-speed mesh pitch line velocity is 18,000 fpm. Gearbox weight is 3500 lb.

A single helical hardened and ground gearset can reduce by up to one-half the envelope and weight of a through-hardened double helical gearbox with equivalent capacity. The inherent precision of the grinding process results in accurate tooth geometry, leading to minimum noise and vibration.

EFFICIENCY

Gearbox efficiency is a much discussed subject, but accurate values are very difficult to determine. Analytical estimates must be confirmed by testing to gain

U.S. Wells - Exhibit 2002
Liberty v. U.S. Wells - IPR2025-00066
Page 33 of 71

a degree of confidence in the procedure. With good design and manufacturing practice, efficiencies of 99% per mesh and better are possible. Often, lubrication system development is required to attain the highest efficiency potential.

Power losses in a gearbox are divided between friction losses at the gear and bearing contacts and windage losses as the rotating components churn the oil and air. In high-speed units the churning losses may exceed the friction losses; therefore, the type and amount of lubricant, and its introduction and evacuation, are critical in terms of efficiency. Journal bearings require significantly more oil flow than do antifriction bearings and generate higher power losses.

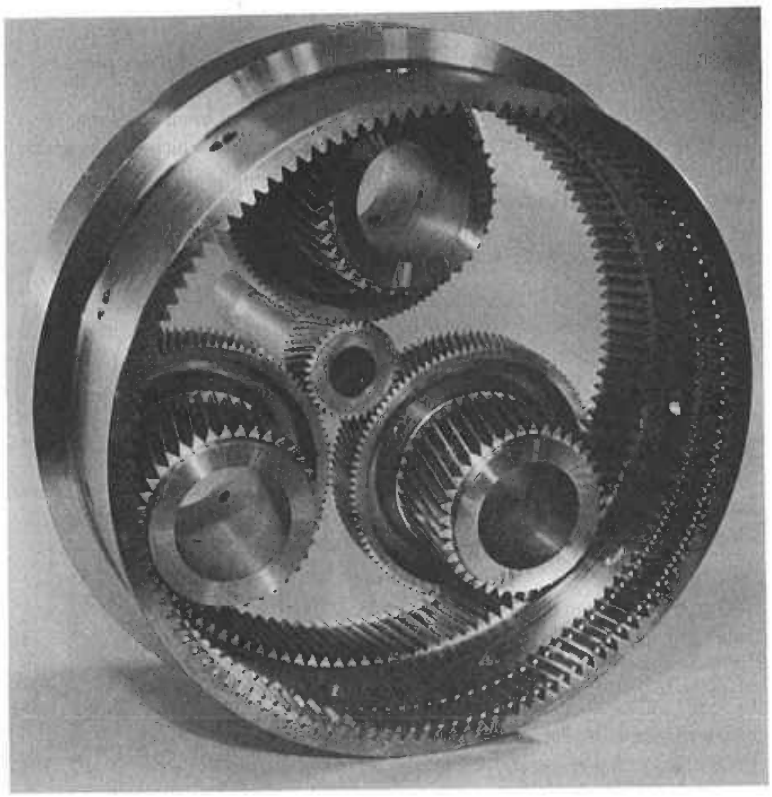


Figure 1.16 Compound planetary gearset. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

USSW - Exhibit 2002
Liberty v. USSW - IPR2025-00066
Page 34 of 71

A reasonable estimate of efficiency for industrial gear boxes is 1 to 2% power loss per mesh. A three-stage unit, therefore, might be expected to have an efficiency in the range 94 to 97%. The efficiency is quoted at the design load and speed conditions. At full speeds and lower loads the efficiency will drop off because the churning losses will remain constant.

SPACE AND WEIGHT LIMITATIONS

There are industrial applications where gearbox space and weight is limited. For instance, generator-drive gearboxes on offshore oil platforms or units used on mobile equipment must have minimum envelope. To achieve small gear units, several techniques can be used:

To minimize gear size, the highest-quality steel, case carburized and precision ground, is incorporated in the unit.

Planetary configurations are used to achieve high ratios in small envelopes. Figure 1.16 illustrates a compound planetary gearset which demonstrates a very efficient use of space, in the approximate range 9:1 to 12:1.

Lightweight design techniques such as thin-wall casings and hollow shafts are employed.

Lightweight materials such as aluminum housings are used.

Maximum application of these techniques can be found in the aerospace industry. An aircraft gearbox might handle the same design conditions as a conventional unit but at one-fiftieth the weight [2].

PHYSICAL ENVIRONMENT

When specifying a gear drive, the physical environment must be addressed in the design stage. Listed below are detrimental environments which can have an adverse effect on lubricant, bearings, gears, or seals:

Dusty atmosphere
High ambient temperature
Wide temperature variation
High humidity
Chemical-laden atmosphere

Such environments require special consideration in the design of the gearbox lubrication system and seals.

USS Wells - Exhibit 2002
Liberty v. USS Wells - IPR2025-00066
Page 35 of 71

REFERENCES

1. AGMA Standard 420.04, Practice for Enclosed Speed Reducers or Increasers Using Spur, Helical, Herringbone and Spiral Bevel Gears, American Gear Manufacturers Association, Arlington, Va., December 1975.
2. Dudley, D. W., *Gear Handbook*, McGraw-Hill, New York, 1962, pp. 3–5.

USSV Weis - Exhibit 2002
Liberty v. USSV Weis - IPR2025-00066
Page 36 of 71

4
BEARINGS AND SEALS

BEARINGS

One of the first questions to be asked when reviewing a gear box design is: "How are the shafts supported"? Each shaft must be radially and axially located by bearings and proper bearing design and application is at least as important to the operation of the unit as the gears themselves. In fact, bearing problems are more common in gear boxes than are gear failures. A basic choice when designing a unit is which type of bearing to use. There are two general classes, one being journal-type bearings (sometimes called fluid film or slider), the other being rolling-element-type bearings (sometimes called antifriction).

Journal bearings are characterized by operating with a relatively thick oil film between the rotating and stationary elements, the oil film being sheared by relative sliding. Rolling element bearings include ball, roller, and needle bearings with many design variations in each of these classes. The rolling elements are in intimate contact with one another, the oil film being relatively thin compared to journal bearings. In many applications it is not clear which configuration to use. Let us look at various parameters as they relate to journal and rolling element bearings.

1. *Load capacity*. As will be seen later in the chapter, rolling element bearing fatigue life can be calculated on a statistical basis; therefore, in critical applications where load is significant and/or speeds are high, finite lives are predicted. Journal bearings are thought of as having infinite lives provided that the loading is kept below a predetermined level. Therefore, in many applications requiring long life, journal bearings are chosen since the calculated rolling element life is considered insufficient. It is true that a journal bearing when

USS Wells - Exhibit 2002
Liberty v USS Wells - IPR2025-00066
Page 37 of 71

9
PLANETARY GEAR TRAINS

A planetary gear train is one in which the power is transmitted through two or more load paths rather than the single load path of a simple gear mesh. Figure 9.1 illustrates the components of the simplest type planetary, a sun gear, planet gears, a ring gear, and the planet carrier. When the carrier rotates about the center of the system a point on a planet gear not only rotates about the axis of the planet gear but also about the center of the system and this type of drive is called epicyclic.

In the United States, planetary gears are widely utilized in automotive automatic transmissions and in aerospace drives such as turbine engine reduction gears or helicopter transmissions but are not used extensively in industrial applications. There is a trend toward increasing utilization of planetary gears because they offer the following advantages:

1. Because they share load between several meshes, planetaries are more compact than parallel shaft drives and offer significant envelope and weight savings. An example illustrates this point:

> Let us take a 1000-hp electric motor at 1800 rpm driving a compressor at 7200 rpm. The required American Gear Manufacturers Association (AGMA) service factor is 1.6.
>
> A hardened and ground, helical, parallel shaft gear set would have the following dimensions:
>
> Pinion pitch diameter: 4.0 in.
> Gear pitch diameter: 16.0 in.
> Face width: 4.0
> Gearbox envelope: 24 × 12 × 12 in.
> Gearbox weight: 650 lb

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 38 of 71

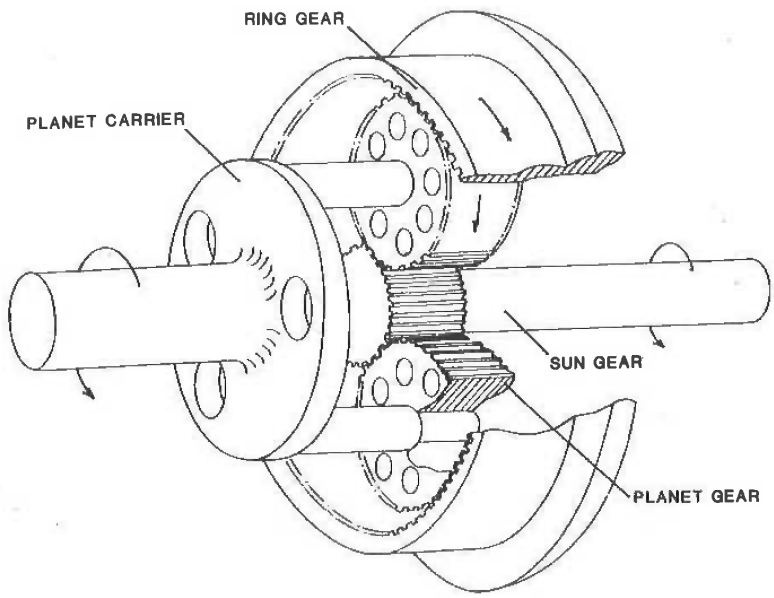


Figure 9.1 Simple planetary gearset components.

The equivalent planetary gear set with a stationary ring gear and rotating carrier would have the following dimensions:

Sun pitch diameter: 3.5 in.
Ring pitch diameter: 10.5 in.
Planet pitch diameter: 3.5 in.
Face width: 3.0 in.
Gearbox envelope: 15 in. diameter × 10 in. length
Gearbox weight: 250 lb.

Figure 9.2 illustrates the gearbox size comparison.

2. In addition to achieving minimum weight and envelope, the relatively smaller and stiffer components which result from the use of planetary gearing lead to reduced noise and vibration and increased efficiency. Noise and efficiency are strongly dependent on the speed of the components. In the example above, the parallel shaft pitch line velocity is 7540 fpm compared to the planetary gear pitch line velocity of 4948 fpm, a significant difference. The lower planetary pitch line velocity is a result of the smaller sun gear and the epicyclic

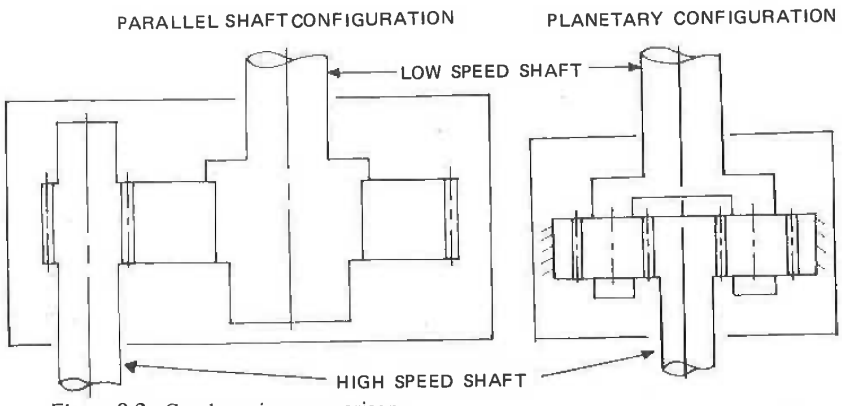


Figure 9.2 Gearbox size comparison.

action of the gear mesh. Later in this chapter the method of calculating epicyclic speeds is derived.

3. The input and output shaft axes in planetary gearing are concentric. This can lead to space savings in some installations by allowing the driving and driven equipment to be in-line. The coaxial feature is very important in automatic transmissions since it makes rapid speed changes possible without the necessity of taking gears out of mesh. The rotating components are controlled by the use of clutches and brakes to achieve speed changes.

4. The resultant radial forces on the input and output shafts of planetary gearboxes are zero since the arrangement cancels out all radial forces. In effect, the planetary gearbox transmits only torque. This simplifies bearing design and in some applications allows close coupling of the gearbox to the driving or driven equipment.

ANALYSIS OF PLANETARY GEAR ARRANGEMENTS

Planetary gears can be arranged in a multitude of configurations to achieve specific ratios and power splits. This section presents a method of calculating planetary speed ratios and power flows. The method is applied to several examples and by following the logic the reader can generate equations describing the mechanics of any planetary configuration. The following nomenclature will be used throughout.

Symbols

R = gear pitch radius, in.
W = gear angular velocity, rad/sec

W_T = tangential load, lb
n = gear rpm

Subscripts

s = sun gear
p = planet gear
r = ring gear
c = carrier
1 = primary stage (high-speed stage)
2 = secondary stage (low-speed stage)

For example, the symbol R_{s_1} would be the pitch radius of the primary-stage sun gear.

Simple Single-Stage Planetary

Figure 9.1 shows the general case where the sun gear, ring gear, and planet carrier are all free to rotate. Directions of rotation are assumed to be as shown in the figure. With all members rotating this is termed a differential system and there are potentially six different ways to connect prime movers and driven equipment:

Inputs	Outputs
Sun	Carrier, ring
Carrier	Sun, ring
Ring	Carrier, sun
Sun, ring	Carrier
Carrier, sun	Ring
Ring, carrier	Sun

The rotating elements in each of the arrangements above have a distinct speed and torque relationship. In order to define the angular velocity of all three elements in the planetary gear train, the angular velocity of two elements must be specified. Let us derive the generalized speed relationships of the gear train in Figure 9.1.

1. A point on the pitch diameter of the sun gear has a tangential velocity of W_sR_s.
2. The point on the sun gear pitch diameter is meshing with a point on the planet gear pitch diameter which has a tangential velocity made up of two components, $W_pR_p + W_cR_s$.

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 41 of 71

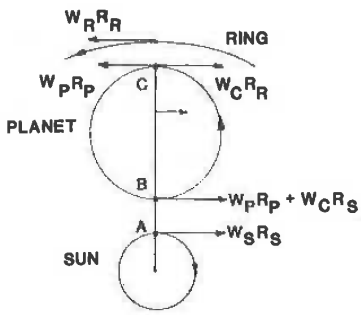


Figure 9.3 Idealized planetary system.

Figure 9.3 shows the planetary system idealized as friction disks with a point A on the sun disk meshing with a point B on the planet disk and also shows their respective tangential velocities. Because the points are meshing they must have the same tangential velocities; therefore,

$$W_s R_s = W_p R_p + W_c R_s \tag{9.1}$$

3. At the ring gear mesh a similar equation may be written:

$$W_r R_r = W_p R_p - W_c R_r \tag{9.2}$$

Note that at point C on the planet gear (Figure 9.3) the components of the tangential velocity are in opposite directions and subtractive, whereas at point B they are in the same direction and additive. The tangential velocity of the planet carrier is zero at the center of the system and equals $W_c R$ at any radius R. Combining Eqs. (9.1) and (9.2), the general speed ratio equation is

$$W_s R_s = W_r R_r + W_c (R_s + R_r) \tag{9.3}$$

Let us apply this equation to the example shown earlier in this chapter:

$$R_s = \frac{3.5}{2}$$

$$R_p = \frac{3.5}{2}$$

$$R_r = \frac{10.5}{2}$$

In this case the ring gear was stationary; therefore,

$$W_r = 0$$

$$\text{Ratio} = \frac{W_s}{W_c} = \frac{R_s + R_r}{R_s} = 4$$

In a simple gear train the rpm of the planet gear would be rpm sun $\cdot \left(\frac{R_s}{R_p}\right)$; however, because of the epicyclic action the planet rpm is reduced and can be calculated from either Eq. (9.1) or (9.2) with

$$\text{rpm sun} = 7200$$

and

$$\text{rpm carrier} = 1800$$

the planet rpm = 5400 and the pitch line velocity of the sun planet mesh is

$$\text{PLV} = \frac{5400(3.5)\pi}{12} = 4948 \text{ fpm}$$

The rolling velocity of the sun with the planet is 5400(3.5/3.5) = 5400 rpm and its absolute velocity is 7200 rpm.

If, in the system shown in Figure 9.1, the carrier is held stationary, $W_c = 0$ and Eq. (9.3) becomes

$$\frac{W_s}{W_r} = \frac{R_r}{R_s}$$

In this case, where there is no epicyclic action, the speed ratio is simply the ratio of the sun and ring gear radii. This configuration is sometimes called a star system.

If the sun gear is held stationary, $W_s = 0$ and Eq. (9.3) becomes

$$\frac{W_r}{W_c} = -\frac{R_r + R_s}{R_r}$$

The negative sign indicates that the sense of rotation of one of the components shown in Figure 9.1 was chosen opposite to the direction in which it actually rotates. In this case, if the ring gear is driving, the carrier would rotate in the same direction as the ring gear rather than the direction shown in Figure 9.1 (viewed from the sun gear end).

In the equations above rpm can be used in place of angular velocity and number of teeth in place of pitch radius. Quite often, planetary gear equations are presented in this manner.

If all components of a planetary gear set are free to rotate, the speed ratios are dependent on the power split between the components. Figure 9.4 shows the tangential tooth loads at the sun and ring gear mesh points. The torque and horsepower of each component are as follows:

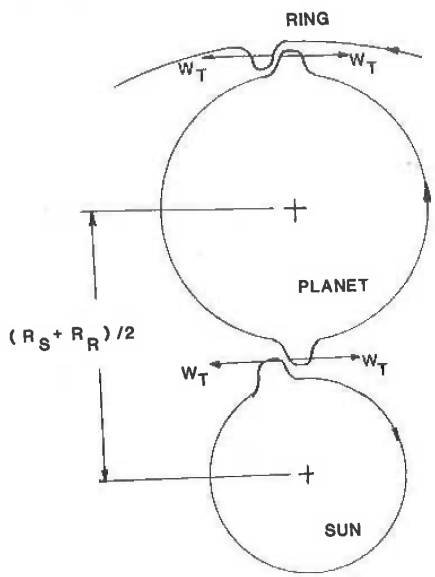


Figure 9.4 Planetary tangential tooth loads.

Component	Torque (in.-lb)	Horsepower
Sun	$W_T R_s$	$\frac{W_T R_s n_s}{63{,}025}$
Carrier	$2W_T \frac{R_s + R_r}{2}$	$\frac{W_T(R_s + R_r)n_c}{63{,}025}$
Ring	$W_T R_r$	$\frac{W_T R_r n_r}{63{,}025}$

It should be noted that the tangential load W_T used in the table is the summation of the tangential loads at each planet gear mesh.

Let us assume that in the previous example the 1800 rpm carrier with a 1000 hp input drives a compressor which is attached to the sun gear and also drives an oil pump which is attached to the ring gear. The oil pump absorbs

50 hp. The three horsepower equations in the previous table can be solved for the three unknowns W_T, N_s, and R_r. Using the carrier horsepower equation with $R_s = 1.75$, $R_r = 5.25$, and $n_c = 1800$ the tangential load $W_T = 5002$ lb. Knowing W_T, the sun and ring horsepower equations yield

$$n_s = 6840 \text{ rpm}$$
$$n_r = 120 \text{ rpm}$$

It should be noted that in this discussion no attempt was made to account for the power loss due to friction in the train.

The arrangement shown in Figure 9.1 is capable of producing gear ratios in a range of approximately 2:1 to 10:1. The speed ratio derived earlier in this section is R_r/R_s for the stationary carrier, rotating ring gear case. In this situation the input and output directions of rotation are reversed. With a stationary ring gear and rotating carrier, the speed ratio is $1 + (R_r/R_s)$ and the input and output rotation directions are the same. At low ratios the sun gear diameter approaches the ring gear diameter and therefore planet gear diameters become small and impractical. As one strives for the higher ratios in a single-stage planetary, the sun gear becomes smaller and is stress limited. Also, at ratios above approximately 10 the ring gear becomes large and more economical arrangements are available.

Compound Planetary

For a ratio of approximately 10:1 to 16:1, the compound planetary gear set (Figure 9.5) provides a practical and economic configuration. The speed reduction equations for this design follow:

$$W_s R_s = W_{p1} R_{p1} + W_c R_s$$
$$W_r R_r = W_{p2} R_{p2} - W_c R_r$$
$$W_{p1} = W_{p2}$$

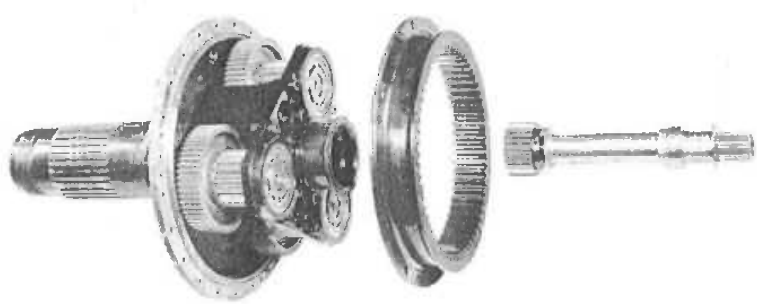


Figure 9.5 Compound planetary gear.

Combining these equations, we have

$$W_s = W_r \frac{R_r R_{p1}}{R_s R_{p2}} + W_c \left(1 + \frac{R_r R_{p1}}{R_s R_{p2}}\right)$$

In the case of a stationary ring gear and rotating carrier the input and output direction of rotation are the same and the speed ratio is

$$R = \frac{W_s}{W_c} = 1 + \frac{R_r R_{p1}}{R_s R_{p2}}$$

In the case of a stationary carrier and rotating ring gear the input and output directions of rotation are opposite and the speed ratio is

$$R = \frac{W_s}{W_r} = \frac{R_r R_{p1}}{R_s R_{p2}}$$

For low ratios in the range 3:1 a reverted type of compound planetary is used as shown in Figure 9.6. The speed ratio for this train is simply

$$R = \frac{W_{s1}}{W_{s2}} = \frac{R_{s2} R_{p1}}{R_{s1} R_{p2}}$$

In this case the input and output directions of rotation are reversed.

Multistage Planetaries

By combining stages of planetary gearing in various arrangements, large ratios can be achieved. For instance, let us take two stages of simple planetary gearing as shown in Figure 9.1 and combine them in two ways as shown in Figure 9.7A and 9.7B. Figure 9.7A presents a drive where the primary-stage carrier drives the secondary-stage sun gear. The reduction ratio of the primary stage is

$$R_1 = \frac{W_{s1}}{W_{c1}} = \frac{R_{s1} + R_{r1}}{R_{s1}}$$

The reduction ratio of the second stage with a carrier output is

$$R_2 = \frac{W_{s2}}{W_{c2}} = \frac{R_{s2} + R_{r2}}{R_{s2}}$$

and the total reduction is

$$R = R_1 R_2 = \left(1 + \frac{R_{r1}}{R_{s1}}\right)\left(1 + \frac{R_{r2}}{R_{s2}}\right)$$

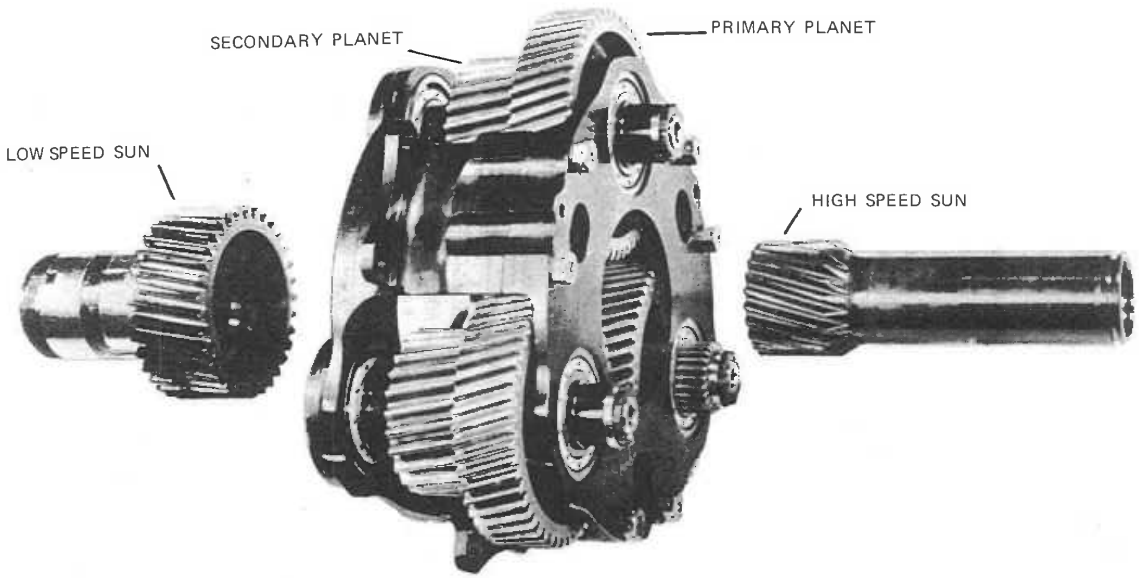


Figure 9.6 Reverted compound planetary. (Courtesy of AVCO, Lycoming Division, Stratford, Conn.)

USSV Weeks - Exhibit 2002
Liberty v. USSV Weeks - IPR2025-00066
Page 47 of 71

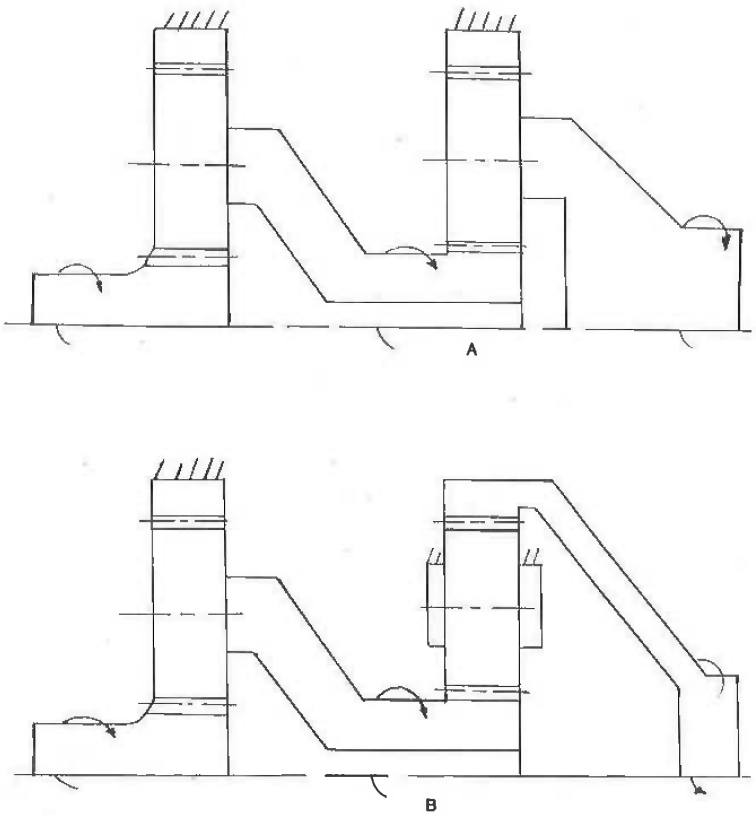


Figure 9.7 Two-stage planetary gearing.

The input and output rotation are in the same direction. Figure 9.7B shows a ring gear output from the second stage and in this case the second-stage reduction ratio is

$$R_2 = \frac{W_{s2}}{W_{r2}} = \frac{R_{r2}}{R_{s2}}$$

and the total reduction ratio is

$$R = R_1 R_2 = \left(1 + \frac{R_{r1}}{R_{s1}}\right) \frac{R_{r2}}{R_{s2}}$$

The input and output directions of rotation are opposite in sense.

USS Wells - Exhibit 2002
Liberty v. USS Wells - IPR2025-00066
Page 48 of 71

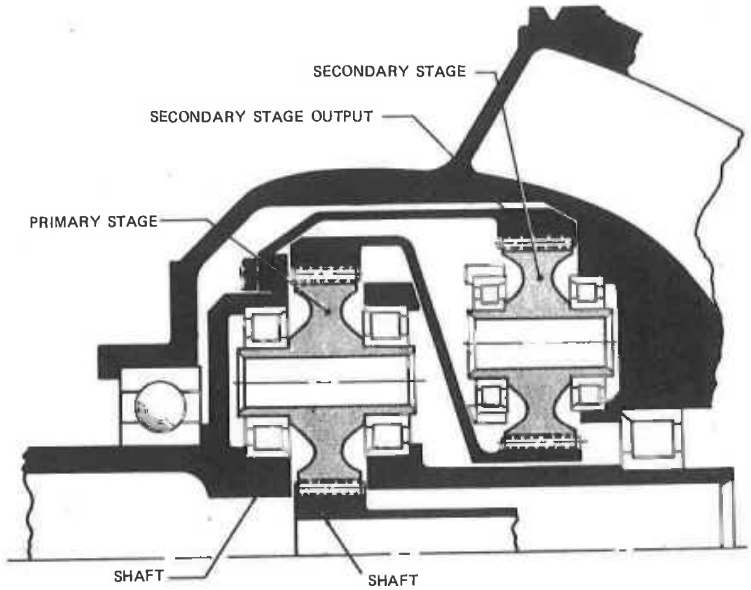


Figure 9.8 Split power transmission.

Split Power Transmissions

An interesting planetary arrangement which results in a very compact design is the split power transmission. Figure 9.8 illustrates one version which is most useful in a ratio range of approximately 15:1. As shown in the figure the primary planetary stage is driven by the sun gear and has both a rotating carrier and ring gear. The primary ring gear drives the sun gear of the second stage which through the stationary second-stage planets drives the second-stage ring gear. Both the second-stage ring gear and the primary stage carrier are connected to the output shaft, thus enabling the power split. Part of the input power is transmitted directly to the output shaft through the primary stage carrier, thereby bypassing the second stage of gearing. Following are the speed and power equations describing the split power transmission:

$$W_{s_1} R_{s_1} = W_{p_1} R_{p_1} + W_{c_1} R_{s_1}$$

$$W_{r_1} R_{r_1} = W_{p_1} R_{r_1} - W_{c_1} R_{r_1}$$

$$W_{r_1} = W_{s_2}$$

USSV Weis - Exhibit 2002
Liberty v. USSV Weis - IPR2025-00066
Page 49 of 71

$$W_{s2}R_{s2} = W_{p2}R_{p2} = W_{r2}R_{r2}$$
$$W_{r2} = W_{c1}$$

where the subscripts $_{s1}$, $_{c1}$, $_{p1}$, and $_{r1}$ refer to the primary-stage sun, carrier, planet, and ring and the subscripts $_{s2}$, $_{p2}$, and $_{r2}$ refer to the second-stage sun, planet, and ring gears.

Combining the equations above we arrive at the expression for the speed ratio:

$$R = \frac{W_{s2}}{W_{c1}} = \frac{W_{s1}}{W_{r2}} = 1 + \left(\frac{R_{r1}}{R_{s1}}\right) + \left(\frac{R_{r2}}{R_{s2}} \quad \frac{R_{r1}}{R_{s1}}\right)$$

To determine the horsepower split between the primary and secondary stage, the following expressions are derived:

$$HP_{in} = \frac{W_T R_{s1} n_{s1}}{63{,}025}$$

where n_{s1} is the primary sun gear rpm.

$$\text{HP primary carrier} = 2W_T \frac{(R_{s1} + R_{r1})n_c}{2(63{,}025)}$$

where n_c is the carrier or output rpm. The percentage of power going out the carrier is

$$\frac{\text{HP primary carrier}}{HP_{in}} = \frac{(R_{s1} + R_{r1})n_c}{R_{s1}N_{s1}} = \frac{1}{R}\left(1 + \frac{R_{r1}}{R_{s1}}\right)$$

The exact percentage of total power transmitted directly by the primary carrier is dependent on the primary- and secondary-stage ratio split but is approximately one-third of the total. The remaining two-thirds of the power is transmitted through the secondary stage. The power split-up serves to reduce secondary-stage loading and increases gear efficiency. The disadvantage of the split power system is that it is mechanically complex.

Figure 9.9 shows a different split power variation which is used to achieve ratios in the range 24:1 to 55:1. The application here is a wheelmotor which combines a hydraulic motor and the planetary gearing as a single unit in the wheel hub. Wheelmotors are used in mobile construction and off-the-road equipment where a centrally located pump drives the hydraulic motors at each wheel.

Figure 9.10 illustrates a wheelmotor casing. In the arrangement of Figure 9.9 the hydraulic motor drives through a sun gear. The output members are the primary- and secondary-stage ring gears which are connected to the wheel rim. The power split up is as follows: Part of the input goes directly from the primary

USSV Wees - Exhibit 2002
Liberty v. USSV Wees - IPR2025-00066
Page 50 of 71

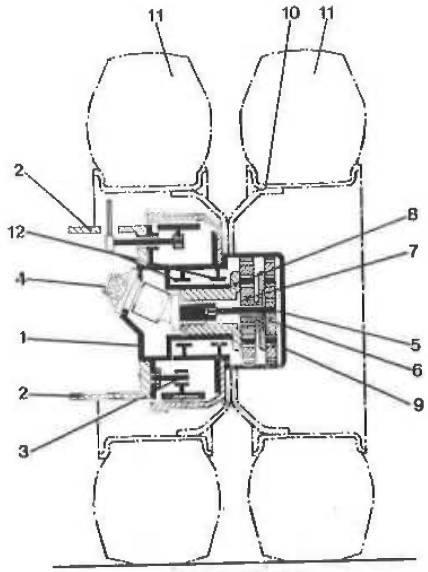


Figure 9.9 Wheelmotor transmission. Key: 1, casing; 2, vehicle chasis; 3, road brake; 4, hydraulic motor; 5, drive shaft; 6, first planetary stage; 7, second planetary stage; 8, axle journal; 9, hub; 10, rim; 11, twin tires; 12, roller bearing. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

sun to the primary ring gear. The remainder goes directly from the primary carrier to the secondary sun gear and then to the secondary ring gear.

Power Feedback Systems

When working with complicated gear systems it is possible to arrange the gears in such a manner that the power transmitted by some components is greater than the input power. Although these arrangements generally achieve unusually large ratios, the component size must be sufficient to handle the recirculating power and this disadvantage may offset the large ratio obtained.

If it is not recognized that a system has recirculating power, failures may occur. Figure 9.11 illustrates a feedback system. The input is to the primary-stage sun gear, which is connected to the secondary-stage sun gear. The output is the secondary-stage carrier. Following are the speed equations for this system:

USSV Weis - Exhibit 2002
Liberty v. USSV Weis - IPR2025-00066
Page 51 of 71

Figure 9.10 Wheelmotor casing. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

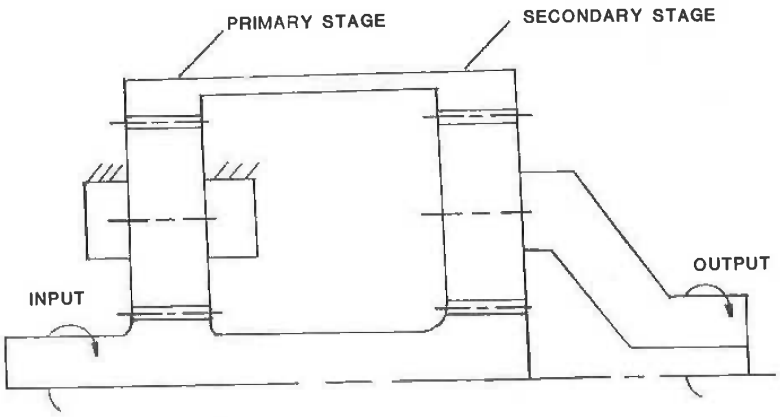


Figure 9.11 Feedback system.

$$W_{s1}R_{s1} = W_{r1}R_{r1} = W_{p1}R_{p1}$$
$$W_{r2}R_{r2} = W_{p2}R_{p2} - W_{c2}R_{r2}$$
$$W_{s2}R_{s2} = W_{p2}R_{p2} + W_{c2}R_{s2}$$
$$W_{s2} = W_{s1}$$
$$W_{r2} = W_{r1}$$

Combining the equations above the speed ratio is

$$R = \frac{W_{s1}}{W_{c2}} = \frac{1 + R_{r2}/R_{s2}}{1 - \dfrac{R_{s1}R_{r2}}{R_{r1}R_{s2}}}$$

Let us assume the following dimensions:

R_{s1} = 1 in.

R_{s2} = 1.05 in.

R_{r1} = 8.3 in.

R_{r2} = 8.25 in.

The reduction ratio R = 166:1. Let us determine the horsepower transmitted:

$$\text{hp out} = 2W_{TS2}\,\frac{R_{s2} + R_{r2}}{2}\,\frac{n_{c2}}{63{,}025}$$

where

W_{TS2} = secondary-stage sun tangential load, lb

n_{c2} = carrier rpm

$$\text{hp secondary sun} = \frac{W_{TS2}R_{s2}N_{s2}}{63{,}025}$$

where N_{s2} is the secondary sun in rpm.

$$\frac{\text{hp secondary sun}}{\text{hp out}} = \left(\frac{R_{s2}}{R_{s2} + R_{r2}}\right) R = 18.74$$

The power circulating in the gear train is 18.74 times the output power.

Planetary Gear Design Considerations

Although the basic gear tooth design of planetary gear configurations is no different from parallel shaft gearing, there are several points that must be

considered when rating planetary gears and defining the detail geometry. This section discusses load sharing, assembly, and choice of numbers of teeth and numbers of planets.

Load Sharing

The main advantage of planetary gearing, of course, is transmittal of load through two or more parallel paths. Ideally, the load should be shared equally by each planet gear, but due to manufacturing errors this is never the case. Some of the factors affecting load sharing are:

Inaccurate carrier bore location
Inaccurate gear tooth spacing
Variation in planet tooth thickness
Eccentricities between sun, ring, and carrier

Figure 9.12A shows the load distribution of a three-planet system. The situation is such that in order for the sun gear to be in static equilibrium the planets must share load equally. If the sun gear is free to move and one planet is overloaded, the sun gear will shift to equalize the planet loads. Figure 9.12B shows static equilibrium; however, the load is not equally shared between planets.

In general, there are three methods of attacking the problem of load sharing between planets:

1. A completely rigid system which relies on precise component tolerances
2. A system with flexibility built in by floating one or more of the members
3. Systems that rely on mechanical means to adjust the planets to provide load equalization

The most straightforward approach is a combination of 1 and 2, where tolerances are closely held and the sun or ring or both are flexible to adjust to load maldistributions. This technique is used in high-speed planetaries such as those in gas turbine engines. Strain gage studies have shown that load sharing within 10% can be practically achieved.

AGMA Standard 420.04, Practice for Enclosed Speed Reducers or Increasers Using Spur, Helical, Herringbone and Spiral Bevel Gears, states the following:

> To compensate for unequal loading of multiple planet pinions, the total power capacity of all pinions should be the calculated capacity of one pinion plus a maximum of 0.9 times the calculated capacity for each additional pinion. For two planet pinions multiply the calculated capacity of one pinion by 1.9; for three pinions multiply by 2.8. If a load balancing device is used which insures equal loading of all planets, the capacity modifying factors need not be used.

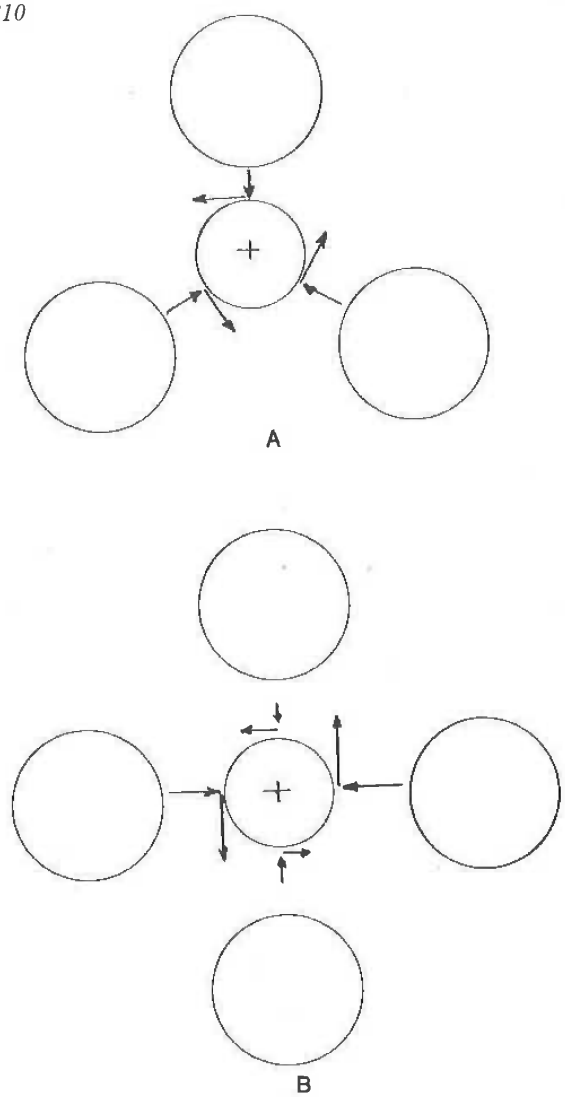


Figure 9.12 Planetary load distribution. A. Three planet system—load must be equally shared if sun is free to float. B. Four planet system in static equilibrium with unequal load distribution.

For any specific application the gear designer must determine what degree of load sharing is anticipated and rate the gearset accordingly. This estimate may be based on analysis, test, or experience, but it is important to understand that perfectly equal load sharing will not be achieved.

STOECKICHT DESIGN

A planetary configuration, extensively used in Europe for many years, is commonly called the Stoeckicht design, named after the inventor. Figure 9.13 illustrates this double helical configuration, which features flexibility not only in the sun gear but also in the ring gears, which are connected to the housing by a series of splines. The purpose of the splines is to achieve load equalization not only between the planets but also between each of the two helical gears. This system has been very successful in the past since the many degrees of freedom tend to compensate for tooth errors. There are disadvantages to the Stoeckicht design:

1. Use of through-hardened double helical gearing does not achieve as small an envelope as hardened and ground single helical gearing.
2. The design is mechanically complex.
3. The design does not lend itself to compounding to achieve higher ratios.

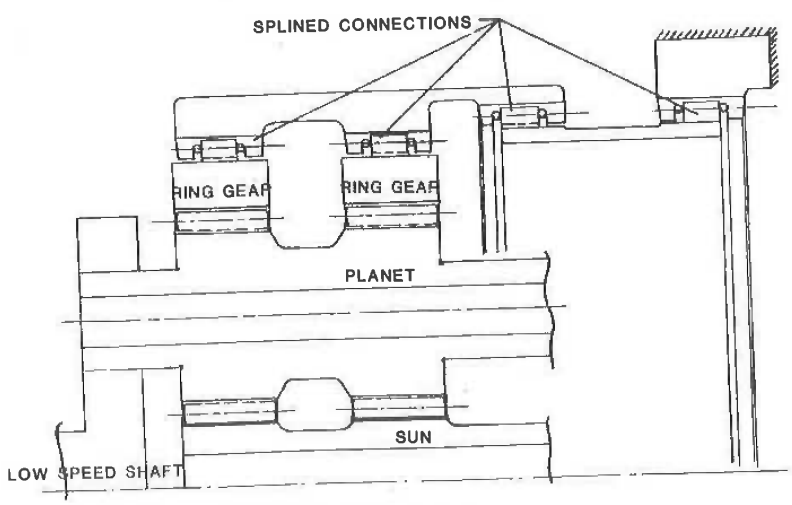


Figure 9.13 Stoeckicht planetary system design.

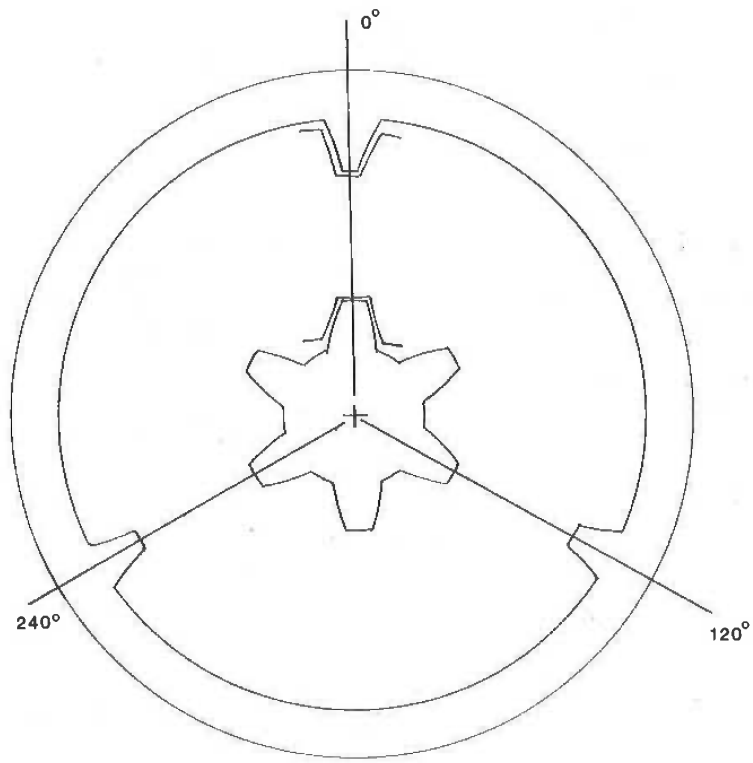


Figure 9.14 Simple planetary gear assembly.

Planetary Gear Assembly

When choosing the numbers of teeth in planetary gears, it must be understood that not every combination of gear teeth can be assembled. For instance, Figure 9.14 shows a three-planet set with a six-tooth sun gear and an 18-tooth ring gear. In this instance it is obvious that planet gears can be placed into mesh at the 0°, 120°, and 240° positions and there is no assembly problem. However, if the ring gear had 19 teeth the gearset would not be assembleable. For a simple planet set (sun, planet, ring) to be capable of being assembled, the following equation must be satisfied:

$$H = \frac{N_s + N_r}{K}$$

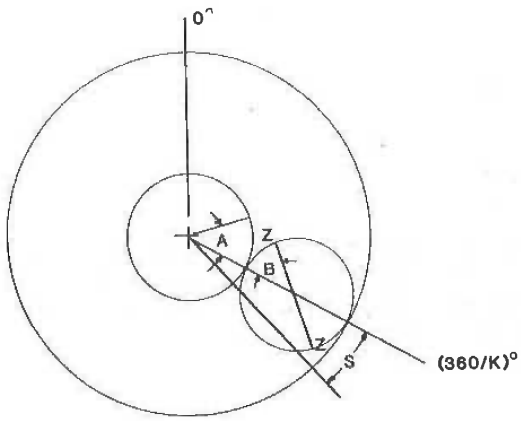


Figure 9.15 Simple planetary gear assembly analysis.

where

H = a whole number (integer)
N_s = number of sun gear teeth
N_r = number of ring gear teeth
K = number of planets

To understand this equation, refer to Figure 9.15. Assume a three-planet system with a planet gear located at the 0° position as shown. In order to assemble a planet at the 120° position, the centerline of a tooth or tooth space of the sun gear must arrive at the 120° point at the same time as the centerline of a tooth or tooth space of the ring gear. Let A be the angle between the center of that sun gear tooth or space and the 120° line and S be the angle between that ring gear tooth or space and the 120° mark. The following equation can be written for A:

$$A = \frac{360}{K} - \frac{360H_s}{N_s}$$

where

K = number of planets
H_s = whole number of sun gear teeth
N_s = total number of sun gear teeth

Note that the equation is general and can be applied to systems with any number of planets.

A similar equation can be written for the ring gear angle S:

$$S = \left(\frac{360}{N_r} \quad H_r\right) - \frac{360}{K}$$

where

H_r = whole number of ring gear teeth
N_r = total number of ring gear teeth

Line ZZ on the planet (Figure 9.15) passes through either two tooth centers or tooth spaces or one tooth center and one tooth space, depending on whether the planet has an even or odd number of teeth. The planet rotates through an angle B at the same time the sun rotates through angle A and the ring gear through angle S. The relationship between these angles is

$$B = \frac{AN_s}{N_p} = \frac{SN_r}{N_p}$$

where N_p is the total number of planet teeth. Substituting for A and S yields

$$\left(\frac{360}{K} - \frac{360H_s}{N_s}\right) \frac{N_s}{N_p} = \left(\frac{360H_r}{N_r} - \frac{360}{K}\right) \frac{N_r}{N_p}$$

which simplifies to the assembly equation

$$H = \frac{N_s + N_r}{K}$$

The assembly equation for a compound planet system is more complicated and can be understood by considering Figure 9.16. Assume that points M and N are registry teeth on the compound planet. By this is meant that the centerline of a tooth on the primary planet is aligned with the centerline of a secondary planet tooth. The purpose of the registry marks is to enable all compound planets to be manufactured identically.

As was shown in the analysis of the simple planet system, the center of a sun gear space rotated an angle A from the 120° mark must arrive at the 120° mark at the same time as the center of a ring gear space, which is shown rotated an angle S from the 120° tooth mark.

$$A = \frac{360}{K} - \frac{360H_s}{N_s}$$

$$S = \frac{360H_r}{N_r} - \frac{360}{K}$$

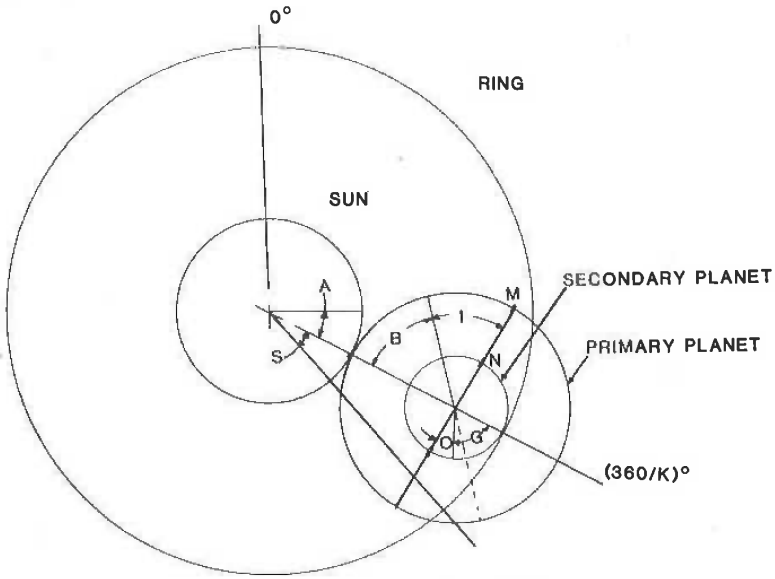


Figure 9.16 Compound planetary gear assembly analysis.

The primary planet rotates through an angle B at the same time the sun rotates through A:

$$B = \frac{AN_s}{N_{p1}}$$

where N_{p1} is the total number of primary planet teeth. The secondary planet rotates through an angle G at the same time the ring gear rotates through S.

$$S = \frac{GN_r}{N_{p2}}$$

where N_{p2} is the total number of secondary planet teeth.

The relationships between the registry teeth and the meshing teeth on the primary and secondary planets are:

$$I = \frac{360H_{p1}}{N_{p1}}$$

$$O = \frac{360N_{p2}}{N_{p2}}$$

where

H_{p1} = whole number of primary planet teeth
H_{p2} = whole number of secondary planet teeth

From Figure 9.16,

$$I = G + O - B$$

Substituting for I, G, O, and B, we have

$$\frac{360H_{p1}}{N_{p1}} = \frac{N_r}{N_{p2}}\left(\frac{360H_r}{N_r} - \frac{360}{K}\right) + \frac{360H_{p2}}{N_{p2}} - \frac{N_s}{N_{p1}}\left(\frac{360}{K} - \frac{360H_s}{N_s}\right)$$

and simplifying yields

$$\frac{N_r + N_sN_{p2}/N_{p1}}{K} + (H_{p1} - H_s)\frac{N_{p2}}{N_{p1}} = H_r + H_{p2}$$

In order to solve this equation the term on the left,

$$\frac{N_r + N_sN_{p2}/N_{p1}}{K}$$

is calculated and whole numbers are substituted for $(H_{p1} - H_s)$. If the left side of the equation works out to be a whole number, the gearset can be assembled.

In a compound planetary each of the planets must be identical not only for assembly purposes but also for proper load sharing. The registry teeth are used to satisfy this requirement and a typical allowable tolerance for the variation of the centerline of the primary planet tooth with the centerline of the secondary tooth is ±0.001 in. Figure 9.17 is a photograph of a compound planetary gear. Compound planets can be either of two-piece or one-piece construction. The processing must be very precise to achieve the registry dimension. In addition to the registry, the tooth thickness and stock removal must be closely controlled. The sequence of operations for a two-piece hardened and ground compound planet are as follows:

1. Cut primary and secondary planet teeth
2. Heat treat
3. Finish grind secondary planet
4. Press primary onto secondary using a fixture to align registry teeth
5. Grind primary planet to final tooth size and registry dimension

The reverted compound planetary gearset (Figure 9.6) with an input and output sun gear has the following assembly equation:

$$\frac{N_0 - N_{s1}N_{p2}/N_{p1}}{K} + (H_s + H_{p1})\frac{N_{p2}}{N_{p1}} = H_{p2} + H_0$$

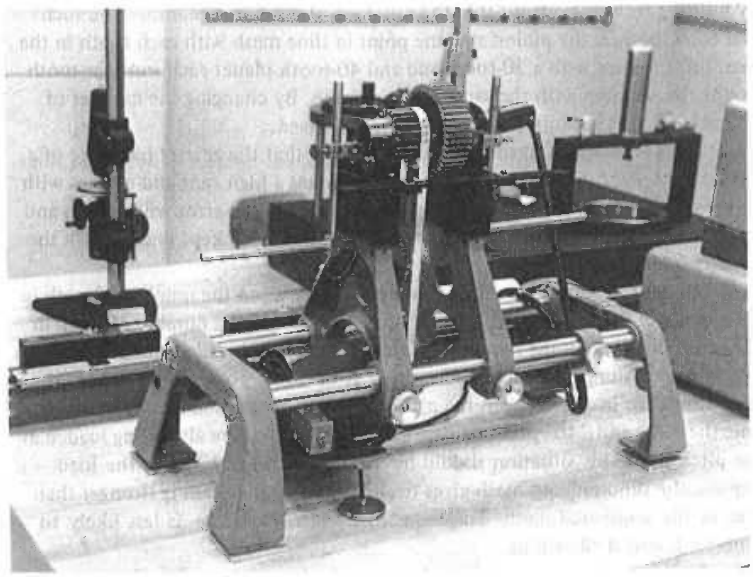


Figure 9.17 Compound planet being balanced.

where

N_0 = total number of low-speed sun gear teeth
H_0 = whole number of low-speed sun gear teeth

It should be noted that it may be physically possible to assemble a gearset that does not satisfy the assembly equations or has mistimed registry markings. If there is sufficient backlash it may be possible to force the components into mesh; however, the gears at some point in the rotation will tend to bind and will have poor load sharing.

Choice of Tooth Numbers

There are three considerations in choosing the numbers of teeth of a planetary in addition to the basic problem of achieving the correct ratio:

1. Assembly considerations as discussed in the preceding section
2. Choosing hunting teeth
3. Achieving sequential mesh rather than simultaneous mesh

By hunting teeth it is meant that the numbers of teeth in a gear mesh be such that each tooth in the pinion at some point in time mesh with each tooth in the gear. For instance with a 20-tooth sun and 40-tooth planet each sun gear tooth would always mesh with the same 2 planet teeth. By changing the number of planet teeth to 41, a hunting tooth mesh is achieved.

The advantage of having a hunting tooth is that the gearset has more of a tendency to wear in. For instance, if one tooth has a high spot and meshes with all the mating gear teeth, there is a better chance that the error will wear in and not cause permanent damage than if the discrepant tooth kept mating with the same two meshing gear teeth.

Sequential mesh occurs in a sun-planet mesh when the number of teeth in the sun is not divisible by the number of planets. When the number of teeth in the sun is divisible by the numbers of planets the mesh is called simultaneous. In this case each sun-planet mesh is occurring at the same point on the gear at the same time. For instance, when the planet at the 0° point is in mesh at its pitch line, the 120° and 240° planets in a three-planet system are also being loaded at the pitch line. This situation should be avoided, if possible, since the load impulse the simultaneous mesh gives to the mechanical system is stronger than that of the sequential mesh. The sequential mesh, therefore, is less likely to generate harmful vibrations.

The mesh frequency of a simple gearset is the number of teeth times the speed. If a planetary gear has sequential meshing, the mesh frequency will be the number of planets times the number of teeth times the speed. The higher sequential mesh frequency is less likely to excite any natural frequencies in the operating equipment.

Although having hunting teeth and sequential mesh is desirable, it is not always achieved and many gearsets operate successfully without fulfilling these conditions. The most obvious way of determining the number of planet teeth is to subtract the number of sun teeth from the number of ring teeth and divide by 2. Let us take an example:

N_s = number of sun gear teeth = 20

N_r = number of ring gear teeth = 100

CD = center distance, inches = 10

The planet gear geometry is

$$N_p = \frac{N_r - N_s}{2} = 40$$

$$DP = \frac{N_p + N_s}{2(CD)} = \frac{N_r - N_p}{2(CD)} = 3.0$$

where DP is the diametral pitch.

$$PD_p = \frac{N_p}{DP} = 13.3333 \text{ in.}$$

What if the number of ring gear teeth minus the number of sun gear teeth is not even? Let us assume the following case:

$$N_s = 20$$

$$N_r = 103$$

$$CD = 10$$

$$N_p = \frac{N_r - N_s}{2} = 41.5$$

If we make the number of planet teeth 41 there is no problem in achieving a practical gear set, but the operating conditions are different at the sun-planet mesh from the planet-ring mesh. At the sun-planet mesh:

$$DP_{s\text{-}p} = \frac{N_s + N_p}{2(CD)} = 3.05$$

The planet pitch diameter is

$$PD_{p(s\text{-}p)} = \frac{N_p}{DP_{s\text{-}p}} = 13.4426$$

At the planet-ring mesh:

$$DP_{p\text{-}r} = \frac{N_r - N_p}{2(CD)} = 3.10$$

$$PD_{p(p\text{-}r)} = \frac{N}{DP_{p\text{-}r}} = 13.2258$$

If the pressure angle ϕ is chosen at 22.5° at the sun-planet mesh, the pressure angle at the planet-ring mesh is calculated as follows:

$$\cos \phi_{s\text{-}p} \cdot PD_{p(s\text{-}p)} = \cos \phi_{p\text{-}r} \cdot PD_{p(p\text{-}r)}$$

and $\phi_{p\text{-}r} = 20.1117°$. The lead of the planet gear is

$$L = \frac{\pi \cdot PD}{\tan \psi}$$

If the helix angle ψ is chosen as 10° at the sun-planet mesh, the helix angle at the planet-ring mesh is as follows:

$$\tan \psi_{(p\text{-}r)} = \tan \psi_{(s\text{-}p)} \cdot \frac{PD_{p(p\text{-}r)}}{PD_{p(p\text{-}s)}}$$

and $\psi_{p\text{-}r} = 9.8419°$. The fact that the operating conditions at the sun-planet mesh are different from those at the planet-ring mesh is advantageous. The sun-planet mesh is critical in terms of compressive stress and flash temperature rise; therefore, it is desirable to have a large pressure angle which reduces both these parameters. Because the ring-planet mesh is internal, compressive stress and flash temperature rise are not usually a problem and the lower pressure angle can be tolerated. In fact, a lower pressure angle results in lower separating forces on the ring gear and this is desirable to reduce ring distortion and stress under load. Even in the case where the number of ring gear teeth minus the number of sun gear teeth is an even number, it can be advantageous to drop one or two teeth from the planet in order to achieve a larger pressure angle at the sun-planet mesh pitch diameter and a smaller pressure angle at the planet-ring mesh pitch diameter.

The question may be asked: How many planets should one use in a particular planetary set? There is a physical limit to the number of planets that can fit into any specific application. Figure 9.18 plots the maximum number of planet gears which can be assembled into a planet system versus the ratio of the sun gear diameter divided by the planet gear diameter. As the reduction ratio becomes smaller, the sun gear becomes larger and more planets can be fit in. As shown earlier, in a three-planet system with a floating sun gear, the load between the planets tends to be equalized. As the number of planets is increased, load sharing may suffer and the full benefit of the additional planets may not be realized. The vast majority of planetary gears use three planets; however, systems with 4, 5, 7, and more planets have been successfully developed.

A point that must be considered when planet gears are idlers meshing with both the sun and ring gear is that the planet teeth experience complete reversal of stress. Allowable bending stresses must be multiplied by 0.7 to compensate for this loading condition.

Planetary Bearing Loads

Quite often the critical components from a life point of view in a planetary system are the bearings. Because the planet gears are loaded both at the sun and ring gear meshing points (Figure 9.4), the planet bearings must react twice the tooth loads.

In an epicyclic system where the planet carrier rotates about the center of the system, the centrifugal force on the planet gears must be reacted by the planet bearings. The centrifugal force on a planet gear is calculated as follows:

$$F_c = mrW^2$$

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 65 of 71

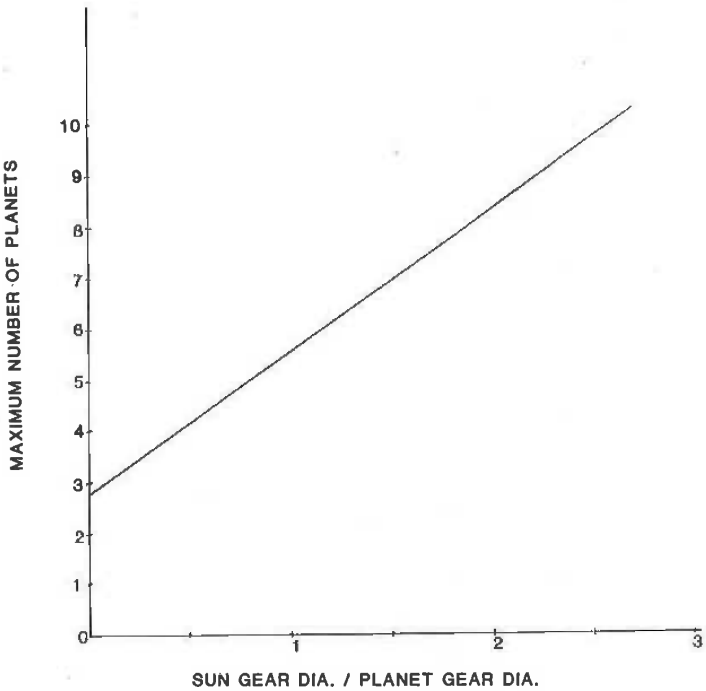


Figure 9.18 Maximum number of planets that can be assembled.

where

F_c = centrifugal force, lb
m = planet mass, lb-sec^2/in.
r = radius to planet center, in.
W = carrier angular velocity, rad/sec

Let us work through an example of a helical epicyclic planetary gear set and calculate the planet bearing loads.

PD_s = sun gear pitch diameter, in. = 3.5

PD_p = planet gear pitch diameter, in. = 3.5

HP = transmitted horsepower = 1000

n_s = sun rpm = 7200

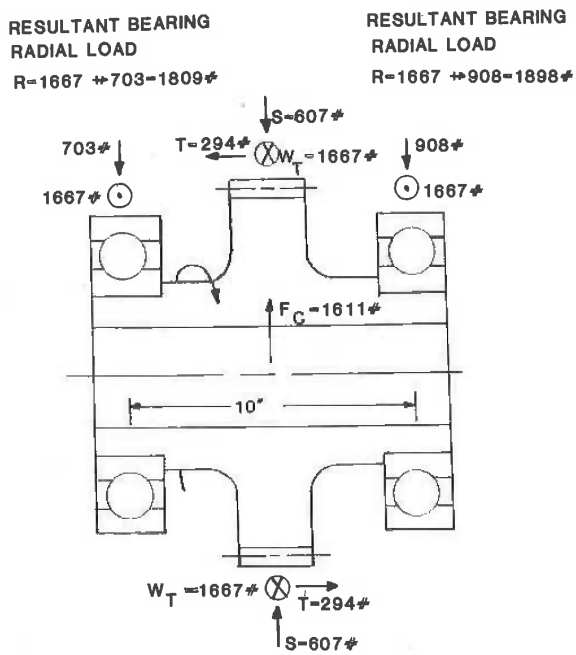


Figure 9.19 Planet bearing loading.

n_c	= carrier rpm	= 1800
ψ	= helix angle, deg	= 10
ϕ	= pressure angle, deg	= 20

Figure 9.19 illustrates the planet configuration and loading. Assume that the planet weighs 5 lb. The centrifugal force is

$$F_c = \frac{5}{386}\ (3.5)\left(\frac{1800 \cdot 2\pi}{60}\right)^2 = 1611 \text{ lb}$$

The tangential load per planet with a three-planet system is

$$W_T = \frac{63{,}025(1000)}{7{,}200}\left(\frac{2}{3.5}\right)\frac{1}{3} = 1667 \text{ lb}$$

The separating load at each mesh is

$$S = W \tan\phi = 607 \text{ lb}$$

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 67 of 71

The thrust load at each mesh is

$$T = W_T \tan \psi = 294 \text{ lb}$$

For the planet to be in static equilibrium the bearing loads are as shown in Figure 9.19. It can be seen that the centrifugal load is significant. The limiting factor in an epicyclic system quite often is the carrier speed, which, if excessive, results in overloading of the planet bearings due to centrifugal force.

Planetary Gear Economics

At first glance it would appear that a planetary gearset must be more expensive than a parallel shaft configuration. There is the added complication of a carrier and two or more additional planet gears plus more bearings. Ring gears are usually expensive items.

In fact, the reduced size of planetary components offsets the cost of additional parts and the determination of which design is less costly is not so obvious. An important factor is the quantity to be manufactured and as the quantity of units increases, the savings in material and advantages of handling and machining smaller components begin to outweigh the disadvantages of mechanical complexity.

Also, when high ratios are required (over 15:1) planetaries tend to be more economical, even in relatively small quantities.

In very high horsepower units, the components in parallel shaft boxes tend to become so large that there may be an economic advantage to planetaries even in the ratio range 2:1 to 15:1.

In any case, the technical advantages of planetary gearing are sufficient such that this type of gearing should be considered, especially since the economic penalty may be small or nonexistent.

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 68 of 71

em is made up of the sound
e components of a generator
ıibit the following noise levels:

:omponent noise levels to

'el of this system, the major

and manufacture of the gear-
ıh would be to transmit power
shaft. In such a perfect gear
ions of the gear shafts, which
ion. Of course, there are no
ırofile, or runout will always
ıtions of the gear shafts and

s. For instance, increasing
ill reduce noise. In these cases
om tooth to tooth as they go
ise, however, might adversely
ontact ratio by going to a finer
design changes that tend to

It must be realized that detail gear tooth geometry changes can only account for changes in sound level of up to approximately 4 dBA. Also, no matter how optimum the design, the quality of the gearing will determine the sound level. Table 11.3 lists the common sources of sound in a gear drive system.

Sound can be transmitted through the air or be structure borne. Structure-borne noise may travel through the support structures and radiate at some point other than its source. Because of this the noise measurements near a gearbox in an operating system may be quite different from those measured during a bench test. Also, when connected to the prime mover and load, the torsional response of the system may be different from the bench test and result in different noise measurements.

In many cases when it is determined that the noise generation of a system is excessive yet not an indication of some malfunction, the most practical method of noise reduction is the use of an acoustical enclosure. The effectiveness of such an enclosure is very dependent on eliminating all openings. Such an enclosure will also affect the ambient temperature and housing heat dissipation ability of the gearbox and means must be provided to cool the unit properly.

Efficiency Testing

Gearboxes are an extremely efficient means of transmitting power. Depending on the operating conditions, design, and manufacturing techniques, the loss per mesh of spur, helical, or bevel gears will vary between approximately ½ to 3%. Analytical means are available to calculate gearbox efficiency, but in order to be confident that an accurate measure of a unit's efficiency is known, the value must be arrived at by testing.

Obviously, the efficiency of the gearbox affects the total system efficiency and the rating of the prime mover required. The power lost in the gearbox, however, also has a large impact on system design since it must be dissipated as heat.

Many low-speed gearboxes have a self-contained splash lubrication system and the power rating of the unit may be limited by the oil temperature rise rather than a mechanical limit. This is the reason gearbox catalogs list a thermal rating in addition to a mechanical rating. When the heat generated in a unit is greater than can be dissipated through the casing an external lubrication system including a cooler is required. The heat generation or efficiency of the gearbox will determine the size of cooler required.

The heat generation can be calculated as follows:

$$Q = MC_p \Delta T$$

where

Q = heat generated, Btu/min
C_p = specific heat, Btu/lb-°F

U.S. Well - Exhibit 2002
Liberty v. U.S. Well - IPR2025-00066
Page 69 of 71

It will tend to change in amplitude with load or operating temperatures, when speed is held constant.

6. *Critical speeds*: These occur through any given speed range and are points at which a rotating system likes to vibrate torsionally or laterally at a particular frequency. Rotors characteristically show violent increase in amplitude at particular critical speeds but are fairly stable above and below these speeds. A critical speed may change frequency with load and temperatures.

tems, such as the vane-pass frequency of a pump or compressor (rotational speed times the number of vanes), and the beating of reciprocating engines which cause frequencies at one-half and one-quarter rotational speed at various amplitudes.

11. *Windage*: Couplings and other rotating parts generally create broadband noise, but can be at a bolt pass frequency or fan blade pass frequency.

[a]All of these types of vibrations and noise frequencies can be generated in a gear drive. Major frequencies can interact and cause frequency modulation and phase shifts. Any combination, sum difference, and multiple (harmonics) of prime frequencies can occur if the forcing magnitude and system freedoms are such that they will cause and allow the generated vibration to become predominant. Generally, only the prime frequencies will present themselves as problem modes. However, sometimes very elusive frequencies appear, such as periodic cutting machine error appearing on one of the gears.
Source: Ref. 3.

M = oil flow, lb/min
ΔT = oil temperature rise across the gearbox, °F

Note:

$$\frac{\text{Btu/min}}{42.44} = \text{horsepower}$$

C_p of oil $\cong$ 0.5 Btu/lb-°F

1 gpm $\cong$ 7.5 lb/min of oil

By accurately measuring the temperature of the oil entering and leaving the gearbox and the oil flow, the heat rejected to the oil can be calculated as shown above.

For example, a gearbox with oil in temperature of 130°F, oil-out temperature of 160°F and oil flow of 20 gpm will reject 848 Btu/min (20 hp) to the oil. If the gearbox is transmitting 1000 hp, the efficiency can be calculated as follows:

$$E = \frac{P_t - P_l}{P_t} = \frac{1000 - 20}{1000} = 98\%$$

where

P_t = total hp transmitted
P_l = power loss

It must be realized that in addition to the heat rejected to the oil, heat is dissipated to the atmosphere by radiation and convection through the casing. To arrive at an estimate of the heat dissipated through the casing, the following heat transfer equation may be used:

$$H = CA(Tc - Ta)$$

where

H = heat dissipated by the gear casing, Btu/hr
C = combined coefficient of radiation and convection, Btu/hr/ft/°F
Tc = casing temperature, °F
Ta = ambient temperature, °F

The value of C depends on the material, roughness, and color of the housing and the velocity of air around the housing and can vary from approximately 0.5 to 3.0. To eliminate the variable of casing radiation and convection, the gear housing can be insulated for an efficiency test. In this way all the heat generated in the unit will be carried away by the oil.

U.S. Wells - Exhibit 2002
Liberty v. U.S. Wells - IPR2025-00066
Page 70 of 71

The heat balance is the most widely used method of measuring gear unit efficiency. A more direct way is to measure input and output torque, but this method requires exact and expensive instrumentation. In the case of a regenerative or back-to-back test, only input torque need be measured and this will monitor the torque required to drive the two locked-up gear trains. Since the prime mover supplies only enough power to make up the losses in the system, the torque lost in each gearbox is one-half the input torque.

Power losses in the gearbox can be divided in two categories, friction losses in the gear mesh and at the bearing interfaces and windage or churning losses. In high-speed boxes, windage and churning losses can be considerable amounting to up to half the total power loss. An easy way to determine these losses is to operate the gearbox at full speed, no load, and then perform a heat balance.

A high-speed box is considered to be one with pitch line velocity ranging from 5000 to 20,000 fpm. Above 20,000 fpm windage and churning can be a limiting factor to the success of the design. For instance, some wide-face-width helical gearboxes operating at pitch line velocities above 20,000 fpm have experienced severe heat generation problems due to pumping of oil and air along the face width. In high-speed double helical gears the hands of helix should always be selected so as to pump oil away from the apex.

When it is determined that a gearbox has excessive windage and churning losses, the situation can be relieved by strategically placing baffles or screens in the unit. In low-speed units using splash lubrication it is beneficial to shroud the gears dipping into the sump oil. With high-speed units the sump oil level should be below the gears and the sump should be separated from the gear meshes by a baffle plate or screen. Minimizing windage and churning losses is more an art than a science and can require trial-and-error solutions verified only by testing.

Lubrication system parameters can have a profound effect on gearbox efficiency. For instance, efficiency will increase with reduced oil flow; obviously, this is due to reduced churning. The limit to this technique is the increase in temperature rise across the box and in the individual components as oil flow is reduced. Increasing oil in temperature will increase efficiency since the oil viscosity will decrease and therefore reduce churning. Changing to a lower-viscosity oil will have the same effect. Proper scavenging of the unit is extremely important in minimizing churning; therefore, great care must be taken in the location and size of drain holes. Also, back pressure must be minimized.

INITIAL FIELD STARTUP

Prior to starting the equipment, the following preliminary checks should be performed:

1. Check oil level and ensur
2. Tighten all pipe connecti
3. Check all electrical conne
4. Tighten all mounting and
5. Check mounting of all ga
6. Check all couplings for p
7. Check inspection cover in

The following instructions pe

1. The unit should be preoil startup.
2. The gearbox should be s Observe that the rotation pressure.
3. After starting, when the o and sufficient oil added t amount.
4. As the unit is brought u monitored for excessive r occur, shut down immedi action. Also check for oil
5. If possible, operate at hal of the gear tooth surfaces.
6. After the initial 50 hr of c and the case flushed with additives. Drain the flushi proper level.
7. After the initial 50 hr of retorque all bolts. Check a

If starts are made in a cold en heating the lubricant. Load sh operating temperature.

CONDITION MONITORING

On an operational gearbox the condition of the unit and whe replace internal components. I traditionally called the "batht unit at any particular point i "infant mortality" period, w

USSW - Exhibit 2002
Liberty v USSW - IPR2025-00066
Page 71 of 71