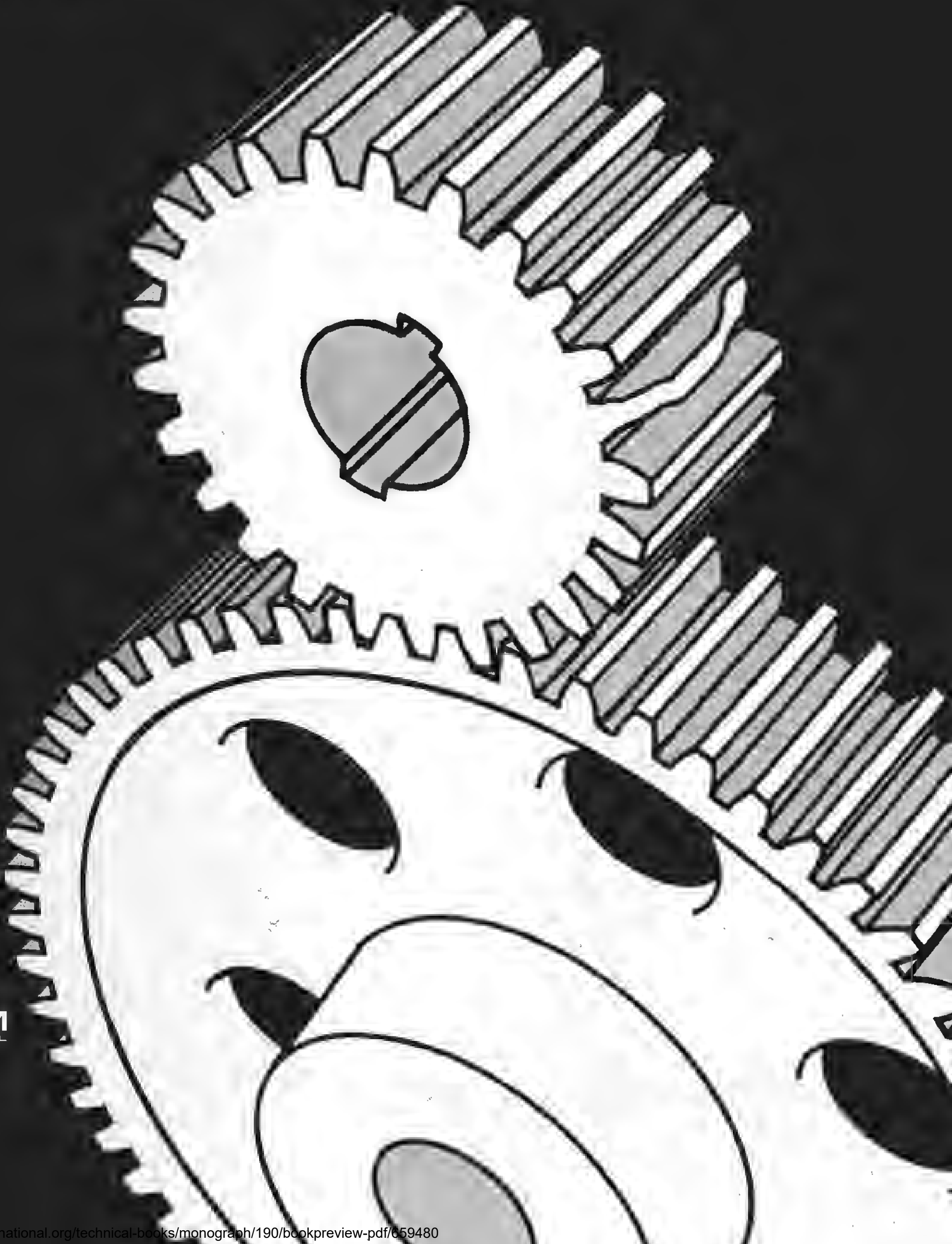
SYSTEMATIC ANALYSIS OF GEAR FAILURES

Lester E. Alban



Systematic Analysis of Gear Failures

Lester E. Alban
Metallurgical Engineer
Fairfield Manufacturing Company, Inc.
Subsidiary of Rexnord, Inc.

ASM INTERNATIONAL
ASM International®
Materials Park, Ohio 44073-0002
www.asminternational.org

First printing, June 1985
Second printing, August 1986
Third printing, November 1990
Fourth printing, July 1993
Digital printing, March 2023

Library of Congress Catalog Card Number: 85-70125
ISBN: 0-87170-200-2 ISBN-13: 978-0-87170-200-5
SAN: 204-7586

PRINTED IN THE UNITED STATES OF AMERICA

Preface

From the discovery of the failure to the final letter of transmittal, the line of communication should remain open, accurate, and sincere. There is no other way to achieve an acceptable plan for corrective action.

Not only the title of this book, but the entire contents strongly emphasize the systematic approach to an analysis of the problem. Putting the complete picture in a sequential pattern makes sense. Case histories are good, but they often do not detail the methods used to determine the final answer. All too frequently a person views a gear failure, finds a similar picture in a book, and believes it to be of the same mode and cause; this may not be so.

The purpose of this text is to train the reader in the art of discipline, to establish a logical step-by-step system of analysis: begin at the beginning and continue methodically to the end. If only we can influence field personnel, field representatives, mechanical and metallurgical analysts, and responsible engineers in management to be aware of the overall picture and to appreciate the role each plays in the final analysis, this book will be a success.

No work of this extent is accomplished by the author alone. Not only has 36 years of first-hand experience been necessary, but also the experience of many experts in our peer group. Of course, Fairfield Manufacturing Company, now a subsidiary of Rexnord, Inc., has been uppermost in supplying the environment of quality reputation necessary to maintain a consistently ethical leadership. My first employer, the late Harrold J. Bates, was meticulous in his concern for detail and accuracy. We

learned to orient all our efforts toward the needs of our customers. This training has been good, and this philosophy continues.

Since I retired a year ago, Fairfield has graciously supplied an office for my use while writing. The Metallurgical Department, under the direction of R. L. Hughes, has been exceptionally patient and extremely helpful in allowing me full use of its time, talent, past records, photographic ability, and laboratory facilities, and sometimes just by listening. Without the stenographic help of its secretary, this work would have been almost impossible. My wife, Faye, was always encouraging and really didn't believe I had retired a year ago, since I always went to work at the same time every morning. This has been a work of discipline, but possible only with the help and the faith of those mentioned above.

Preview for: *Systematic Analysis of Gear Failures*, by Lester E. Alban, ASM International, 1985
Full digital book available at https://doi.org/10.31399/asm.tb.sagf.9781627084529

ABOUT THE AUTHOR

LESTER E. ALBAN, whose career as a metallurgical engineer includes 36 years with Fairfield Manufacturing Company, Inc., in Lafayette, Indiana, is a frequent speaker and author in the area of gear failure analysis. A graduate of Purdue University, Mr. Alban also studied at the School of Mining and Metallurgy at Washington State College and served with the U.S. Navy for two years. In 1947 he joined Fairfield, now a subsidiary of Rexnord, Inc., where he was responsible for all gear failure analysis.

A charter member of the Purdue Chapter of the American Society for Metals and of ASM's Heat Treating Council of the Technical Divisions, he is registered in Who's Who in Engineering. Mr. Alban's professional memberships also include the Society of Automotive Engineers and the International Metallographic Society. Since his retirement in 1983, he has continued his involvement in the field through teaching seminars and consulting.

Contents

CHAPTER 1

Basic Understanding of Gears

Analyzing a failed part without taking into account the entire picture of magnitude and direction of applied forces, the spacial relationship of all component parts within the assembly, and the environmental conditions existing around the unit would be like examining a torn automobile tire for defects but never examining the bent axle that was causing the detrimental side thrust. For a systematic study of a gear failure, the basic parameters of a gear must be understood.

Purpose, Design, Function

A gear is a machined component that transmits motion and force from one element in a working unit to another element in the same unit or to another working unit in either the same plane and direction or a completely different plane or direction. The force, due to this transmission, may either increase or decrease in power from one element to the next. Design and function are closely associated because a gear is designed with a specific function in mind. The question is if this gear will perform the function that was intended by the designer.

Spur gear and pinion. A spur gear and pinion (Fig. 1-1) is a parallel-axis unit with tooth forces and motion at exactly 90°

Fig. 1-1. Spur gear and pinion.

from the central axis of each part. Due to the ratio of numbers of teeth in each part, the speeds may be increased or decreased. The transmission of power is in a straight line.

Helical gear and pinion (Fig. 1-2) is also a parallel-axis gearing application with transmission of power about a straight line. However, the teeth, being helical, exert a resultant force in an angular direction at the tooth contact interface in proportion to the angle of helix. Along with any tendency to move the tooth laterally by the applied contact pressure, there is also the tendency for a surface sliding action in the same direction. A double-helical gear is one in which two sets of helical teeth are

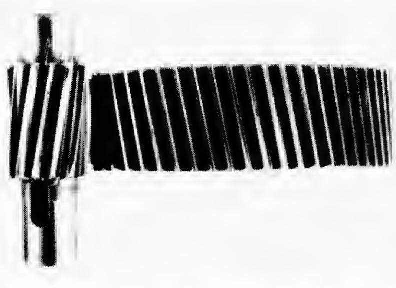


Fig. 1-2. Helical gear and pinion.

cut around the same periphery, but with an opposing angle of helix. The central area is machined out so that each portion is disconnected. If the central portion is securely connected, a herringbone gear will be obtained. The advantage to the double-helical and/or herringbone gears is that the side thrust common to the single helix has been eliminated.

Internal gear. Both the spur gear set and the helical gear set may be designed so that the gear member will have the teeth on the inside diameter of a ring. This is called an internal gear (see Fig. 1-3).

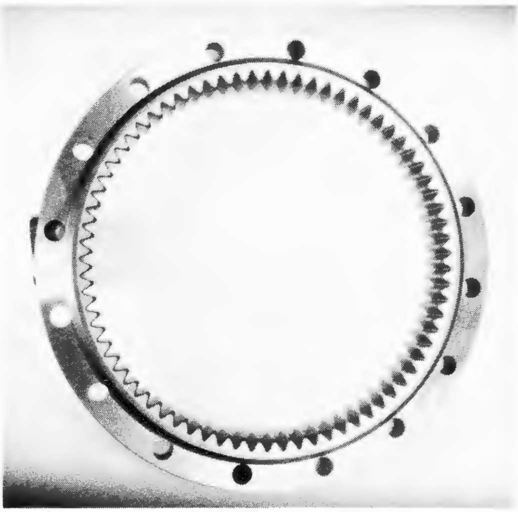


Fig. 1-3. Internal gear and pinion. This set can be either spur or helical.

The straight bevel gear and pinion (Fig. 1-4) will transmit motion in an angular direction (usually at 90°). If the speeds are to remain the same with only a 90° change of direction, the set is called a miter gear set. Any change in the number of teeth will change speed as well as direction. The contact forces tend to push the opposing teeth apart as well as to cause a lateral slide along each tooth surface.

Spiral bevel sets. As the off-angle gear teeth are given an angular displacement, they naturally assume a spiral (circular) shape to conform to the rotating motion. This gearing is called

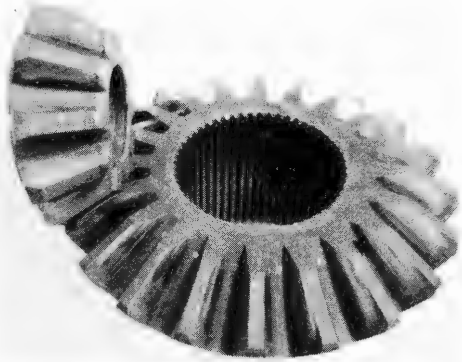


Fig. 1-4. Straight bevel gear and pinion.

Fig. 1-5. Two types of spiral bevel sets, consisting of spiral bevel gear and pinion.

spiral bevel sets. There are many types of spiral bevel configurations; two types are shown in Fig. 1-5.

Hypoid sets. Usually the axis of the spiral bevel gear and pinion will intersect at a common point in space. However, when the pinion axis is raised or lowered in relation to the gear axis, the result is a hypoid set (Fig. 1-6). When the axis of the pinion is displaced almost to the center of the gear teeth at the periphery, the number of teeth in the pinion decreases to three or less and has the action of a worm pinion. This set is called the high-ratio hypoid set (Fig. 1-7).

Fig. 1-6. Hypoid gear and pinion set.

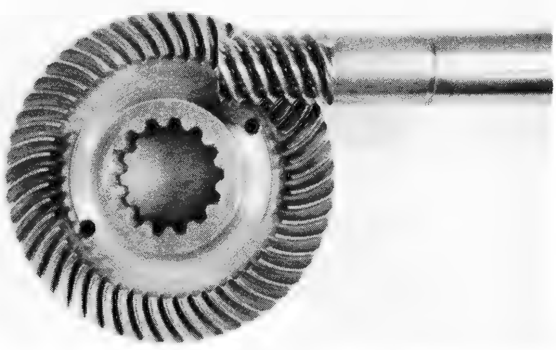


Fig. 1-7. High-ratio hypoid set.

Basic Applied Stresses

The loads applied to one tooth by the action of its mating tooth are at any moment of time a line contact at the most; or, at the least, a point contact. As the loads are increased, the line may lengthen or even broaden, or the point may expand to a rounded area.

The basic stresses applied to a gear tooth include the six types listed in Fig. 1-8; often, a combination of two or three types are applied at a time. Commonly they are tensile, com-

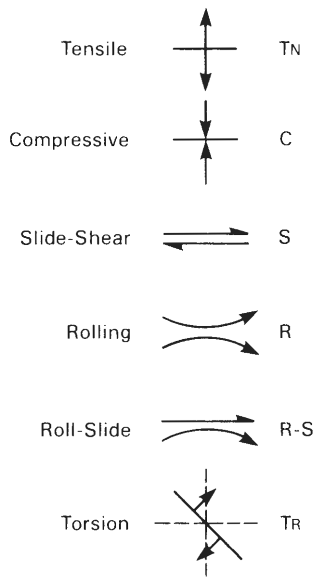


Fig. 1-8. Basic stresses that are applied to gear teeth. Often, two or three are simultaneously applied to a specific area.

pressive, shear (slide), rolling, rolling-slide, and torsion. Each type of gear tooth will have its own characteristic stress patterns.

For specific gearing terminology and nomenclature of tooth elements, refer to "Geometry and Theory of Gears" by Paul M. Dean[1,2] and Standards published by American Gear Manufacturers Association.[3]

Spur Gear

As the contacting tooth moves up the profile of the loaded tooth, a sliding-rolling action takes place at the profile interface. At the pitchline, the stresses are pure rolling. Above the pitchline, the rolling-sliding action again takes over, but the sliding will be in the opposite direction. Keep in mind that the action on the profile of the contacting tooth is exactly the same as the loaded tooth except in reverse order (see Fig. 1-9). The

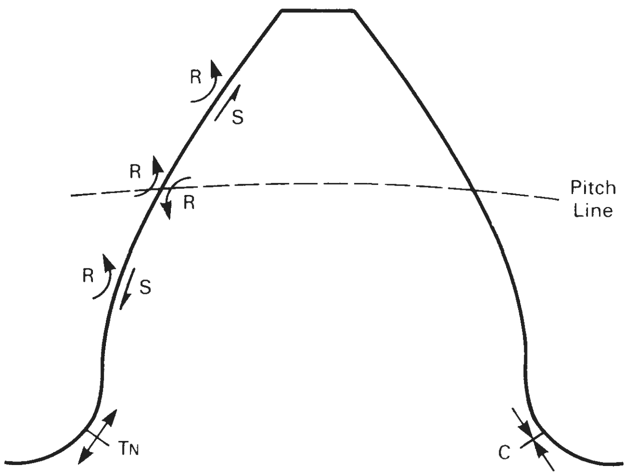


Fig. 1-9. Diagramatic stress areas on basic spur gear tooth.

sliding action of two surfaces, when lubricated properly, will have no problem. However, surface disparities, insufficient lubrication, improper surface hardness, higher temperatures, and abrasive or adhesive foreign particles will contribute to a breakdown during a sliding contact. At the same time, there is a tensile stress at the root radius of the loaded side of the tooth and a compressive stress at the root radius of the opposite side.

Helical Gear

The helical gear tooth receives the same contact action as the spur gear; i.e., a rolling-sliding action from the lowest point

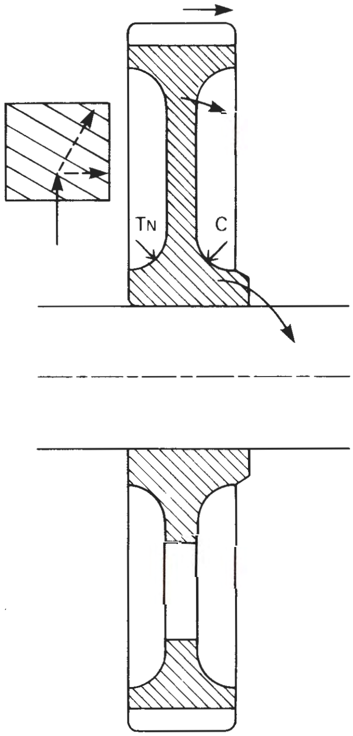


Fig. 1-10. Secondary stresses set up in associated parameters of a helical gear due to the side thrust action of the helix.

of active profile up to the pitchline, rolling over the pitchline, then sliding-rolling from the pitchline over the addendum. An additional stress is being applied to the helical tooth; a lateral sliding action is applied at all contact levels, including the pitchline. The force component at 90° to the direction of rotation increases as the helical angle increases. Resultants of this side thrust are often overlooked (see Fig. 1-10). The web between the center shaft hub and the outer gear rim is constantly undergoing a cycle of bending stress; it is not uncommon for a relatively thin web to fail in bending fatigue. If the hub of the gear faces against a thrust bearing, the bearing itself is under a constant thrust load. The shaft carrying the gear undergoes a continual rotation bending stress. It is also not uncommon to have such a shaft fail by rotational bending fatigue. The above secondary stresses are found only in a gear of a single-helix pattern. A double-helical gear or a herringbone gear will not have a side thrust component of stress; therefore, the entire stress load will be absorbed by the teeth.

One additional stress that should be discussed at this time is a stress common to all gearing because it involves rolling surfaces. It is a shear stress running parallel to the surface at a distance from 0.007 to 0.012 in. below the surface. The distance below the surface given above is the average depth for a normal loading condition. The actual depth of maximum shear could be deeper, depending on the radius of curvature of the mating surfaces and the tangential forces being applied. In one instance there has been evidence of rolling loads above the shear strength as deep as 0.034 in. The subsurface shear stress is most often the originator of initial line pitting along the pitchline of gear teeth, line pitting low on the profile due to tooth tip interference, line pitting along the tooth tip due to the same tooth tip interference, and subsurface rolling contact fatigue. The subject of rolling contact fatigue is discussed more fully in Chapter 4.

Straight Bevel Gear

The straight bevel gear undergoes the same stresses as discussed above, including a very slight helical action laterally. The larger sliding action component is parallel to the axis of the gears and tends to push the gears apart, causing a higher profile contact, and to exert a rotational bending stress in the web of the part as well as in the shaft.

Spiral Bevel Gear

Aside from all the stresses applied above, a spiral bevel gear has a resultant peculiar to itself. As the rolling-sliding stress tends to move in a straight line laterally, the progression of the points along the stress line moves in a bias across the profile of the tooth. As long as at least two teeth are in contact, the resulting load per unit area is well within reasonable limits.

However, there are circumstances (and it may be only momentary) when there is a 1-to-1 tooth contact. This very narrow line contact may be accepting an extremely high load per unit area, and a line of pitting will result early in the life of the tooth. Careful attention should be given to the design characteristics of these parts, such as spiral angle and pressure angle.

Hypoid Gear

The hypoid gearing has the same applied stresses as those discussed for the spiral bevel, but sliding becomes the more predominant factor. This predominance increases as the axis of the pinion is placed farther from the central axis of the gear, and is maximum when the set becomes a high-ratio hypoid.

Strength

The strength of any component is measured by the amount of stress that can be tolerated before permanent strain (deformation) takes place.

Strain, or deflection under load, is a constant for steel regardless of hardness or heat treatment. The amount of deflection under load of a thin gear web or the shank of a pinion cannot be changed by heat treatment or by use of a stronger material. Hooke's law is the same: A change of deflection can be accomplished only by a change of design.

Bending strength of a gear tooth is the amount of load per unit area acceptable at the root radius to the point of permanent deformation. Permanent deformation of a carburized tooth is usually accompanied by a crack at the root radius, whereas with a noncarburized tooth, actual bending may occur. The root radius is mentioned as the point of deformation because it is the area of greatest stress concentration in tension. Also, stress (load per unit area) calculations assume that the load is applied at the pitchline or the midheight of the tooth. Actually, the realistic stress at the root radius varies from approximately one-half, when the load is applied low on the active profile, to double, when the load is applied near the tooth tip. Bending strength of the root radius is a function of the surface hardness and the physical condition of the surface, such as smoothness, sharpness of radius, and/or corrosive pitting.

The strength of the core material—i.e., the basic material under the carburized case—is generally to be considered as compressive strength rather than tensile strength. It measures the ability to withstand surface pressures that may crush through the case and/or brinell the surface.

Torsional strength of a pinion shank or of a shaft is a bit more complex. The maximum tensile stress is at the surface in a direction 45° from the central axis or longitudinal direction. The maximum shear stress, also at the surface, is longitudinal (parallel to the central axis) and transverse (90° across the central axis) (see Fig. 1-11). The strength at the surface is a function of surface hardness; therefore, surface-originated torsional

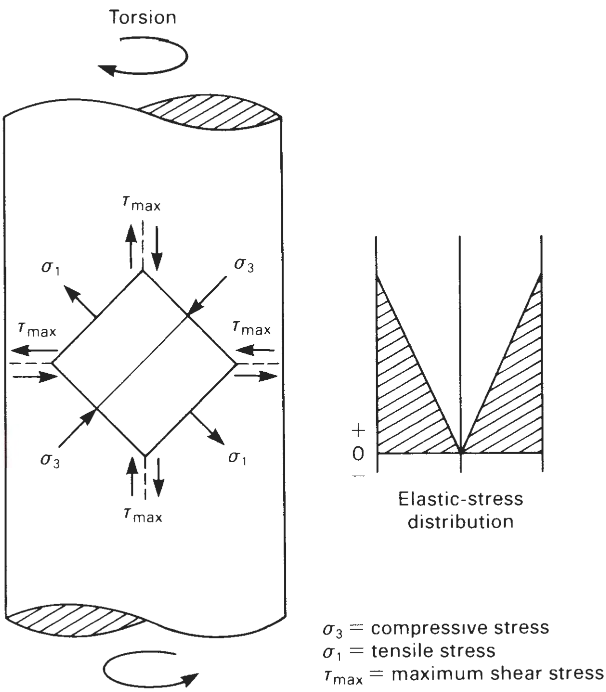


Fig. 1-11. Free-body diagram of maximum tensile and shear stress orientation on a surface element of a shaft in a torsional mode. Both maximums are at the surface. Stress is considered to be zero at the central axis.[4]

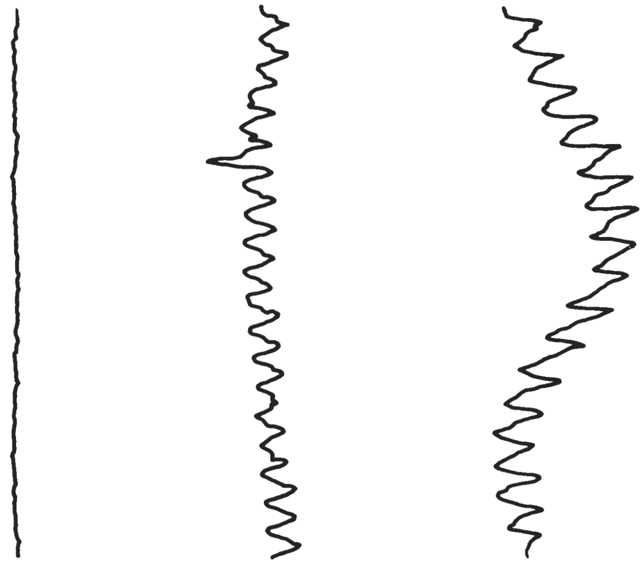
tensile failures of carburized parts are rare unless a specific type of stress raiser is present at the surface. This is not so with through hardened or non-heat treated parts since the strength is uniform throughout the part. Under this condition, torsional tensile failure is expected to originate at the surface. In most instances of torsional failure of carburized or induction hardened shafts and pinion shanks, the initial fracture is along the shear plane (i.e., longitudinal or transverse), and not at the 45° angle. This means that the shear strength of the subsurface material is the controlling factor. Shear strength is considered to be only about 60% of the tensile strength. The area most vulnerable to the origin of torsional shear failure of a shaft is the transition zone between the case and the core of either a carburized or an induction hardened part. The maximum applied stress often exceeds the shear strength of the material at this area and initiates a start of subsurface failure.

Gear Tooth Characteristics

Tooth characteristics are designed into the gear as an integral part and are usually included in the dimensional specifications. Each characteristic contributes significantly to the tooth load contact pattern, which greatly influences the unit area accepting the maximum loads.

Tooth-to-Tooth Running Pattern

Tooth-to-tooth running pattern is plotted on a chart as a closely controlled master gear runs with the production gear. The accuracy of the master gear is such that any discrepancy noted in the chart can be attributed to the lack of accuracy in the production gear. The characteristics noted in this test are gear tooth runout and tooth-to-tooth jump. The ideal, straight line pattern, shown in Fig. 1-12(a), can be attained only from a helical gear that has more than one tooth in contact and has no runout. A spur gear is expected to have some amount of "jump" as the contact goes from tooth to tooth. Of course, the more accurately the teeth are cut, the less jump. The pattern shown in Fig.



(a) Ideal straight line, no runout; very little tooth-to-tooth jump, due to helical tooth overlap. (b) Almost straight line, 0.001-in. overall runout; perceptible tooth-to-tooth jump, typical of spur gearing. Note the one protuberance due to surface nick. (c) Spur gear with 0.006-in. overall runout.

Fig. 1-12. Tooth-to-tooth running pattern.

1-12(b) is an example of a spur gear considered to be excellent, with only 0.001-in. runout. The chart shown in Fig. 1-12(c) is of a spur gear that has a fairly good tooth-to-tooth pattern, but a runout of 0.006 in. is indicated.

It becomes evident, also, that these tests are functional in ways other than those specified. For instance, the chart in Fig. 1-12(b) shows one abrupt movement beyond the normal pattern, which indicates a discrepancy that in most instances would be due to a slight external nick or bump.

Involute Pattern

As an indicator follows the tooth profile from the lowest point of the active profile to the tip of the tooth, the involute is scribed onto a chart. A true involute for a tooth would show up as a straight line on the chart (Fig. 1-13).

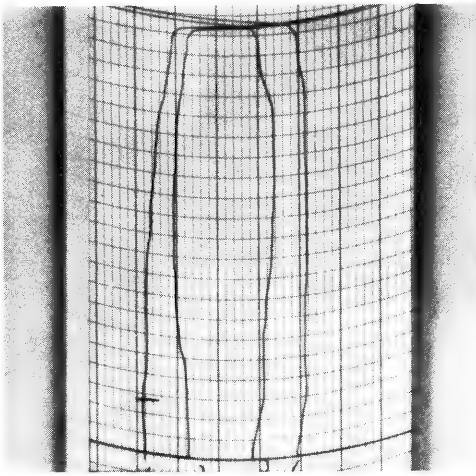


Fig. 1-13. Charted involute pattern. The straight line indicates true involute of the active tooth profile.

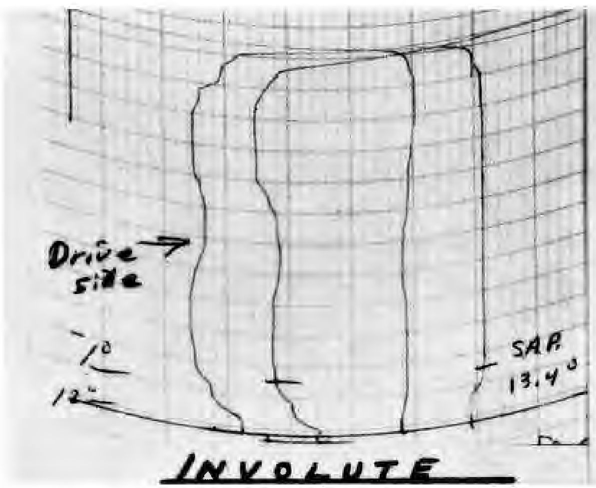
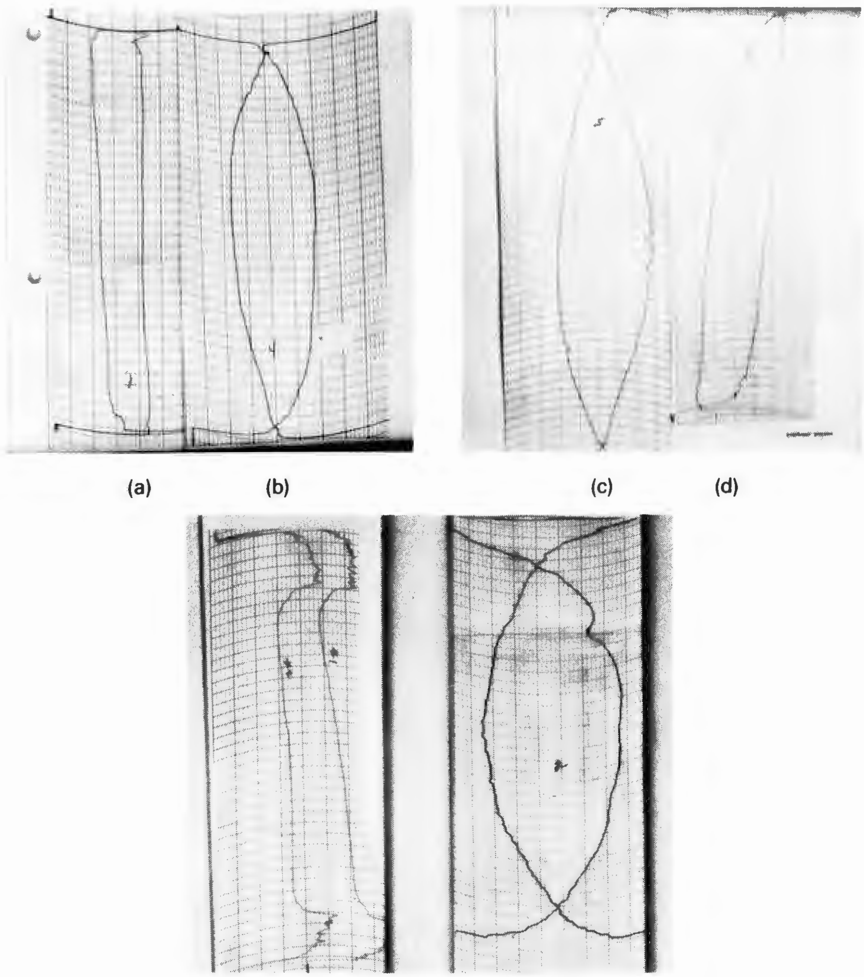


Fig. 1-14. Charted involute pattern of two teeth (180° apart) from a gear in service. Unused profile has an excellent involute; the loaded profile shows pitchline and tip wear.

Again, this charting also can be used as an inspection tool for reading what has happened in service. As an example, Fig. 1-14 shows an involute pattern of two teeth of a gear returned from service. One side profile has not been worn; the graph is a straight line. The pattern of the opposite side shows a definite wearing of the midprofile area and slight wear over the addendum. This pattern also reveals that the gear ran in one direction only. The involute charts should be read carefully to distinguish differences of thicknesses low on the profile (positive for interference), high on the profile (positive for tip interference or negative for excessive falling away), and at midprofile (a positive or negative barreling effect that would concentrate the load pattern at the midpoint or at either extremity).

Lead Pattern

The lead pattern is the graphic line inscribed on a chart when an indicator traverses from one end of the tooth to the other end along a line parallel to the central axis, most commonly along the pitchline. A true lead should be a straight line for both the spur gear tooth and the helical gear tooth. A traverse with very little error, as shown in Fig. 1-15(a), is considered a quality lead. Teeth often are crowned for the purpose of keeping contact loads off the ends of the teeth. This is accomplished by cutting a taper toward the ends of the teeth and leaving the central areas somewhat thicker. Figure 1-15(b) shows a crowned condition of approximately 0.0015 in. on a side. Note that the ends of each traverse line are at the same position on the chart, indicating a true lead. Lead charts also can be used for diagnosis of problems or conditions that may exist. Figure 1-15(c) shows a gear tooth that has a true lead but shows pitting along the pitchline where the indicator had traversed. Note that the pitting is central on one side of the tooth but is off-center on the opposite side, which would indicate that the lead of the mating tooth may have a tapered condition on one side or that it had been deflected under load. Figure 1-15(d) shows a tooth that has considerable lead error that could be due to either a tapered condition or an actual lead error in cutting. Note also that the tooth had been crowned on both sides, but because of the errors in lead, the crown has shifted toward the ends of the tooth and each side at an opposite end. Figure 1-15(e) shows the amount and position of abrasive wear on two teeth in opposite positions on a gear but on the same side of the teeth. Figure 1-15(f) shows a gear tooth that is crowned and well centered, but 0.006 in. on a side seems to be abnormally high.



(a) True lead along both profiles. (b) True lead, tooth crowned 0.0015 in. per side. (c) True lead with crown. Indicator shows pitting along the pitchline. (d) A crowned tooth with considerable lead error, shifting the crown toward opposite ends of each profile. (e) Lead pattern of loaded side of two random teeth of same gear; result due to abrasive wear. (f) True lead, tooth crowned 0.006 in. per side, which seems abnormally high.

Fig. 1-15. Charted lead patterns.

What can result from the above lead characteristics is described here. Figure 1-16 is a photograph of a "perfect" gear; i.e., the involute is correct, the lead is true, and there is no runout or taper. But every tooth has failed at one end in one direction by heavy tooth contact at that end. The reason? It had been mated

Fig. 1-16. Helical gear. All tooth characteristics were "perfect." Field failure had been matched with pinion showing lead pattern of Fig. 1-15(d).

with the pinion teeth shown in Fig. 1-15(d). Refer now to Fig. 1-17, which shows a crack at the center of the tooth contact area characteristic of tooth crushing through the case. The lead was true, the case depth was within specification, and the core hardness was normal for the material. The reason for the crack? The tooth had been matched with the pinion teeth shown in Fig. 1-15(f).

It is time to mention a very important consideration of failure analysis. The actual part that failed will show the mode of failure, but very often the cause of failure is to be found in the mating or matching part. One should always inspect both parts very closely for the solution to the problem. In fact, if there are several components to an assembly, each component must be suspect until eliminated by the examiner.

Mating-Tooth Contact Pattern

It is difficult to determine any graphic method of checking characteristics of a spiral bevel gear tooth. Therefore, a mating-

Fig. 1-17. Spur gear. Tooth crushing at midprofile contact area. Reason: mated with pinion showing lead pattern of Fig. 1-15(f).

tooth contact pattern method has been established. The supplier and customer should agree on a contact pattern acceptable for the application.

The tooth contact pattern method is as follows: A prepared substance is brushed onto the tooth profile of both matching parts; the parts are then run on a tester at a predetermined position; the parts are first run at no-load and the pattern is noted; and finally, the parts are run under a predetermined load and the pattern is noted. The pattern is the position on the profile of each tooth that runs tightly against its mating tooth, thus pushing away the applied substance. The substance used was commonly a mixture of litharge (lead-oxide) and oil. However, the use of lead products is now highly discouraged, and two proprietary compounds (one white[5] and one yellow[6]) are available.

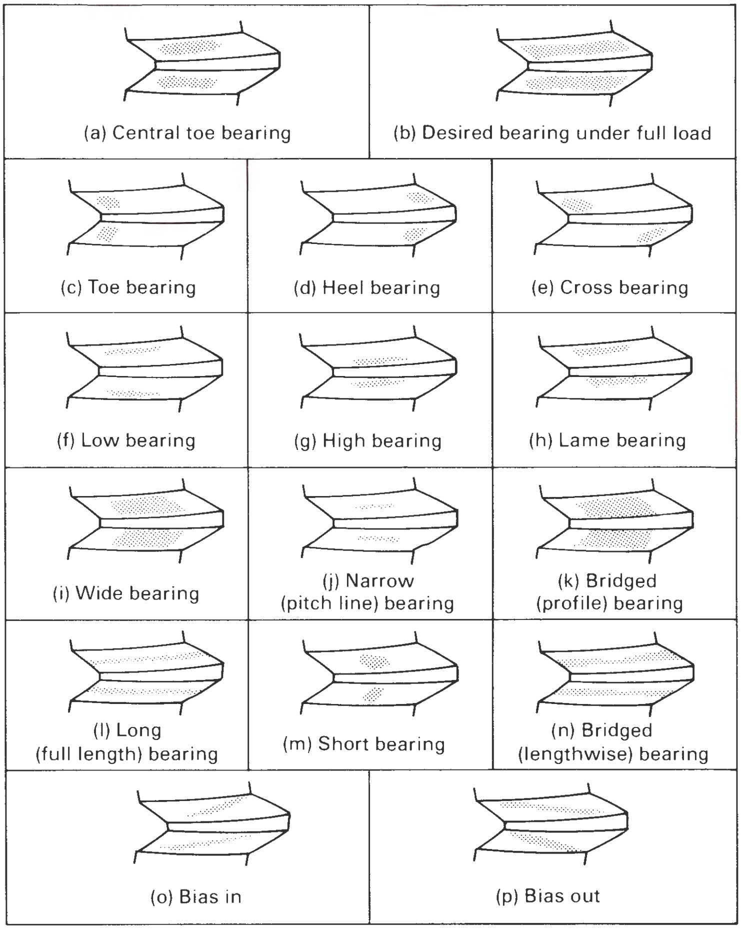
In general, the normal tooth contact pattern under no-load will be central profile toward the toe end. Ideally, as the load is increased to full-load, the deflection of the tooth causes a shift of the contact to a full-length pattern (shown in Fig. 1-18a and b).[7]

There are also a large number of usually undesired variations found in tooth-to-tooth contact patterns. They are usually found during the development stages of gear tooth cutting and can be eliminated by readjustment of cutting. These are shown in Fig. 1-18(c) through (p).

This method of checking tooth contact patterns is very valuable in determining actual field loading applications. The test machine can be set on the specified positions to see if the parts had actually run as expected. Then the tester can be readjusted to show the prevailing conditions (such as deflection or misalignment) under which the parts had been running in the field.

Backlash

Backlash is the rotational arc of clearance between mating gear teeth when the gears are set at the proper mounting distance. Ideally, when a very accurate involute is generated on both parts, they will rotate smoothly and easily with zero backlash. However, this condition precludes the use of any lubricant. An intentional backlash is engineered into the design to compensate for many factors, including lubrication, differences



Sketches illustrate tooth bearings on the pinion tooth. Although a left-hand pinion is used throughout, the bearings are representative of those on a right-hand pinion or a straight bevel pinion as well.

(a) Central toe bearing. Note that the bearing extends along approximately one-half the tooth length and that it is nearer the toe of the tooth than the heel. In addition, the bearing is relieved slightly along the face and flank of the tooth. Under light loads the tooth bearing should be in this position on the tooth.

(b) Same tooth as in (a) with a bearing as it should be under full load. It should show slight relief at the ends and along the face and flank of the teeth. There should be no load concentration at the extreme edges of the teeth.

(c) through (e) show differences in spiral angle between the gears tested. (f) through (h) show differences in pressure angle between the gears tested. (i) through (k) illustrate width of tooth bearing. (l) through (m) illustrate length of tooth bearing.

(o) and (p) illustrate bias bearings. Regardless of the hand of spiral on the pinion, "bias in" will always run from the flank at the toe to the top at the heel on the convex side and from the top at the toe to the flank at the heel on the concave side.

Fig. 1-18. Tooth bearings.

of involute profile, tooth tip interferences, tooth runout, tooth deflections under load, dimensional changes of teeth due to heat treatments, and size changes of gears due to a rise in temperature during operating conditions.

A designed backlash allows the teeth to be cut slightly thinner than the theoretical size, with a resulting involute profile that will give optimum contact under load. When an optimum backlash has been established for a gear/pinion set, it is important that this amount be maintained at the initial assembly operation. The life of a gear set can be prolonged by a slight reduction of backlash from the amount specified for initial assembly, as long as none of the deterrent factors prevails. A tendency to hit a zero backlash at any time or position will immediately become destructive. On the other hand, as backlash may be increased at the time of assembly, the expected life of the gear set is greatly reduced. This condition accelerates improper contact and brinelling.

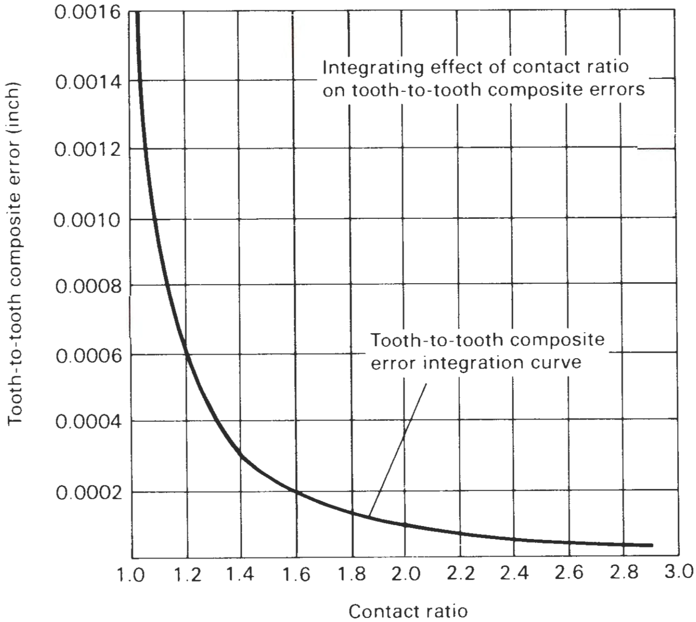
Composite Errors

When all errors of dimensional characteristics and distortion are added together, the result is the overall, or composite, error. Many times they are additive and many times they are subtractive; thus, the resulting composite error may vary from zero to a major magnitude from gear to gear, even within the same manufactured and processed lot.

L. D. Martin[8] did an extensive study of the effect of composite error and found that its influence on gearing was affected most by the contact ratio of the gears. Composite errors were not affected by a 1.0 contact ratio but tended to integrate rapidly toward zero effect as the contact ratio increased—graphically represented in Fig. 1-19.

Associated Parameters

The importance of the function of the mating part has been emphasized. There are also components in the design and structure of each gear and/or gear train that must be considered in conjunction with the teeth.



As contact ratio increases, increasingly large tooth-to-tooth composite errors are smoothed out. This curve is based on empirical data obtained in a series of tests. It is predicated on reasonably well-cut gears and is independent of pitch and pressure angle.

Fig. 1-19. Integrating effect (smoothing) of contact ratio.

Round Bores

A round bore, with close tolerances and ground, may rotate freely around a ground shaft diameter, may fit tightly against a ground diameter by having a press fit, or may be the outer race of a needle bearing that gives freedom of rotation. Each application has its own unique problems. A ground bore is always subject to tempering, burning, and checking during the grinding operation. A freely rotating bore requires a good lubricating film, or seizing and galling may result. The bore used as a bearing outer race must be as hard as any standard roller bearing surface and is subject to all conditions of rolling contact, including fatigue, pitting, spalling, and galling. The bore that is press fit onto a shaft is subject to a definite amount of initial tensile stress. Also, any tendency for the bore to slip under the applied rotational forces will set up a unique corrosive action between the two surfaces, which leads to a condition recognizable as either stress corrosion or fretting corrosion—the two having subtle differences, even though they are very similar in nature.

A spur or helical round bore gear acting as an idler (or reversal) gear between the input member and the output member of a gear train has an extremely complicated pattern of stresses to contend with. The most common application is the planet pinion group in wheel reduction assemblies or in planetary-type speed reducers. The photoelastic study shown in Fig. 1-20 reveals certain facts:

(*a*) The tensile and compressive stresses in the bore are caused by bending stresses of the gear being loaded as a ring.
(*b*) The maximum tensile and compressive stresses in the bore increase as the load on the teeth increases.
(*c*) The maximum tensile and compressive stresses in the bore increase as the clearance between the bore and the shaft is increased.
(*d*) The maximum tensile and compressive stresses in the bore increase as the ratio of the size of the bore to the root diameter of the teeth is increased.
(*e*) During one revolution of the gear under load, the teeth go through one cycle of complete reversal of stresses, whereas each element of the bore experiences two cycles of reversals.

Three modifications of the round bore alter the stress patterns considerably:

(*a*) Oil holes that extend into the bore are intended to lubricate the rotating surfaces. Each hole may be a stress raiser that could be the source of a fatigue crack.
(*b*) Tapered bores are usually "shrunk" fit onto a shaft. This sets up a very high concentration of stresses not only along the ends of the bore but also at the juncture on the shaft.
(*c*) A keyway in the bore also initiates a high stress-concentration area. A keyway is also required to withstand a very high, continuous load. In fact, the applied load to the side of the keyway is directly proportional

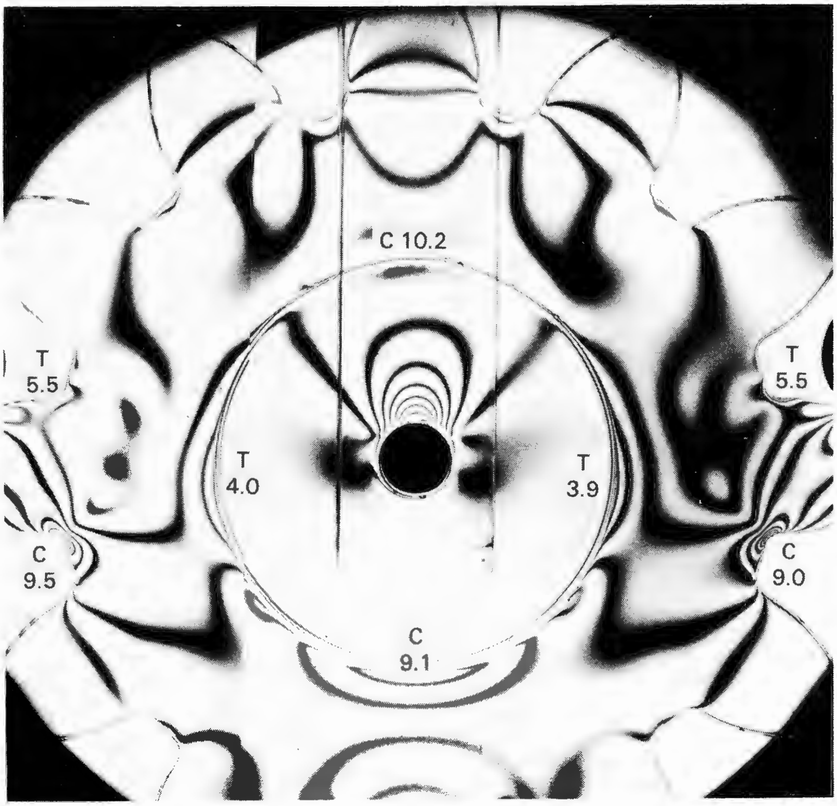


Fig. 1-20. Photoelastic pattern of idler gear between load input pinion and load output gear.

to the ratio of tooth pitchline radius to the radius of the keyway position. Fatigue failure at this point is common.

Splined Bores

The loads applied to the splined bore area are also directly proportional to the ratio of pitchline radius of the teeth to the pitchline radius of the splines. However, the load is distributed equally onto each spline, so the stress per spline is not usually excessive. It is possible for an out-of-round condition to exist that would concentrate the loads on slightly more than two splines. Also, a tapered condition would place all loadings at one end of the splines, which would be detrimental both to the gear splines and to the shaft. Heat treating of the splined area must also be watched closely for quench cracks. Grinding of the face against the end of the splines can also cause grinding checks to radiate from the corners of the root fillets.

Shafts

The shafts within a gear train (as well as the shank of a pinion, which constitutes a shaft in function) are of importance to load-carrying capacity and load distribution. They are continually exposed to torsional loads, both unidirectional and reversing. The less obvious stressed condition that is equally important is bending. Bending stresses can be identified as unidirectional, bidirectional, or rotational. When the type of stress is identified, the causes for such a stress can be explored.

A number of stresses applied to a shaft can be imposed by parts riding on it. For instance, helical gears will transpose a bending stress, as will straight and spiral bevel gears; round bores may be tight enough to cause scoring and galling; a splined bore may cause high stress concentration at its end face; runout in gears may cause repeated deflections in bending of the shaft; and loose bearings may cause excessive end play and more bending.

Preview for: *Systematic Analysis of Gear Failures*, by Lester E. Alban, ASM International, 1985
Full digital book available at https://doi.org/10.31399/asm.tb.sagf.9781627084529

Chapter 1: Basic Understanding of Gears

References

1. M. Paul Dean Jr., Geometry and Theory of Gears, in *Gear Manufacture and Performance*, American Society for Metals, 1974, p 1-24.

2. Paul Dean Jr., Geometry and Theory of Gears, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 1-24.

3. Standards published by the American Gear Manufacturers Association, Suite 1000, 1900 North Fort Myer Drive, Arlington, Virginia, 22209.AGMA 112.04 “Gear Nomenclature,” August 1965.AGMA 115.01 “Basic Gear Geometry,” July 1959.AGMA 116.01 “Glossary, Terms Used in Gearing,” October 1972.

4. J. Donald Wulpi, Torsional Fracture of Shafts, *Metal Progress*, Vol 120 (No. 3), August 1981, p 26.

5. #35W Gear Marking Compound, Wayne Chemical Products Company, 9470 Copland Street, Detroit, Michigan 48209.

6. #10 Yellow Gear Marking Compound, Prescott & Company, Ltd., 2625 Rue Paré Street, Montreal, Quebec H4P-1S1.

7. Gleason Works Publication SD3025B, 1955, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 341.

8. L. D. Martin, Large Contact Ratios Minimize Effects of Gear Errors, *Machinery*, Vol 73 (No. 6), February 1967, p 97.

Preview for: *Systematic Analysis of Gear Failures*, by Lester E. Alban, ASM International, 1985
Full digital book available at https://doi.org/10.31399/asm.tb.sagf.9781627084529

Chapter 2: Basic Understanding of Environment

References

1. W. J. Bartz, The Influence of Lubricants on Failures of Bearings and Gears, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 172-183.

2. T. I. Fowle, Gear Lubrication: Relating Theory to Practice, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 205-219.

3. Alan Hichcox, Computer-Aided Gear Lubrication, in *Power Transmission Design*, Penton/IPC, May 1983, Vol. 25 (No. 5), p 19-20.

4. The Timken Company, *Bearing Selection Handbook*, 1983, p 20-26.

Preview for: *Systematic Analysis of Gear Failures*, by Lester E. Alban, ASM International, 1985
Full digital book available at https://doi.org/10.31399/asm.tb.sagf.9781627084529

Chapter 3: Systematic Examination

References

1. Robert Clark Anderson, *Inspection of Metals. Volume 1; Visual Examination*, American Society for Metals, 1983.

2. *Metals Handbook Desk Edition*, American Society for Metals, 1985, p 34-7.

3. Society of Automotive Engineers, *Iron & Steel Handbook Supplement HS-30*, 1981, Table 4, p 4·07.

4. Society of Automotive Engineers, *Iron and Steel Handbook Supplement HS-30*, 1981, p 4·43-4·44.

Chapter 4: Modes of Gear Failure

References

1. H. J. Grover S. A. Gordon L. R. Jackson, “Fatigue of Metals and Structures,” Battelle Memorial Institute, prepared for Department of Navy, Bureau of Aeronautics, NAVAER 00-25-534, 1954, p 15-19.

2. Charles Lipson L. V. Cowell, *Handbook of Mechanical Wear*, Ann Arbor: University of Michigan Press, 1961, p 135. doi: 10.3998/mpub.9690186

3. D. L. Seager, ASLE, Separation of Gear Teeth, in Approach and Recess, and the Likelihood of Corner Contact, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 25-29.

4. Appendix IX of Gleason Works Publication SD4O52A, May 1966, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 366-411.

5. C. Harry Rogers, Adiabatic Plastic Deformation, in *Annual Review of Materials Science*, Vol 9, 1979, p 283-311. doi: 10.1146/annurev.ms.09.080179.001435

6. D. H. Breen, Fundamental Aspects of Gear Strength Requirements, in *Source Book on Gear Design. Technology and Performance*, American Society for Metals, 1980, p 63-65.

7. Charles Lipson, *Why Machine Parts Fail*, The Penton Publishing Company, Cleveland, Ohio, 1951, p 25.

8. T. S. Eyne, An Introduction to Wear (Section l), in *Source Book on Wear Control Technology*, American Society for Metals, 1978, p 1-10.

Chapter 5: Causes of Gear Failure

References

1. Harold E. McGannon, *The Making, Shaping, and Treating of Steel*, 9th ed., United States Steel, 1971, p 588-589.

2. A. D. Merriman, *Λ Concise Encyclopedia of Metallurgy*, New York American Elsevier Publishing Co., Inc., 1965, p 284.

3. *Manual of Open Die Forgings*, Open Die Forging Industry, New York, New York, 1949, p 13.

4. Reprinted with permission from American Society for Metals.

5. *Metals Handbook* 8th ed., Vol 10, Failure Analysis and Prevention, American Society for Metals, 1975, p 520.

6. Source Book in Failure Analysis, American Society for Metals, 1974, p 129.

7. “Embrittlement of Steels,” *Metals Handbook Ninth Edition*, Vol 1, Properties and Selection: Irons and Steels, American Society for Metals, 1978, p 683-688.

8. *Metals Properties*, Handbook of The American Society of Mechanical Engineers, 1st ed., New York: McGraw-Hill, 1954.

9. T. J. Dolan, Residual Stress, Strain Hardening and Fatigue, in *Internal Stresses and Fatigue in Metals, Symposium* on Internal Stresses and Fatigue in Metals (held in Detroit and Warren, Michigan, 1958), Elsevier Publishing Co., New York, 1959, p 304-305.

10. Charles Lipson, *Why Machine Parts Fail*, The Penton Publishing Company, Cleveland, Ohio, 1951, p 5.

11. Robert P. Haviland, Valley Forge Space Technology Center, and General Electric Company, Engineering *Reliability and Long Life Design*, Princeton: D. Van Nostrand Co., Inc., 1964, p 6-13.

Chapter 7: Writing the Report

References

1. Harry E. Chandler, *Technical Writer's Handbook*, American Society for Metals, 1983, p 111-133.