Systematic Analysis of Gear Failures

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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.

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°

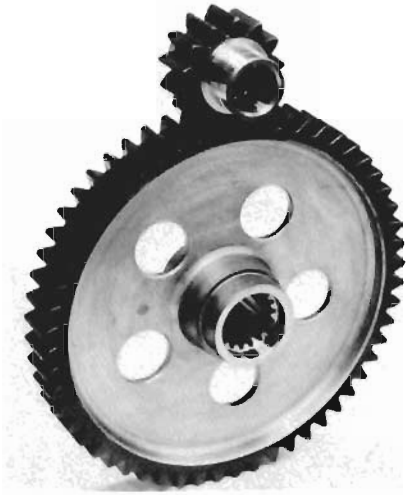


Fig. 1-1. Spur gear and pinion.

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

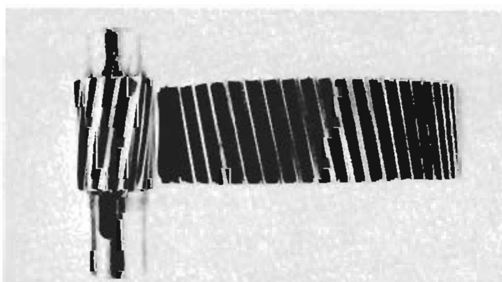


Fig. 1-2. Helical gear and pinion.

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).

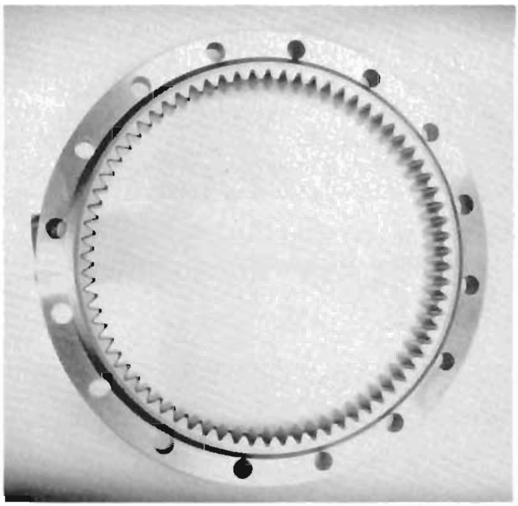


Fig. 1-3. Internal gear and pinion. This set can be either spur or helical.

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

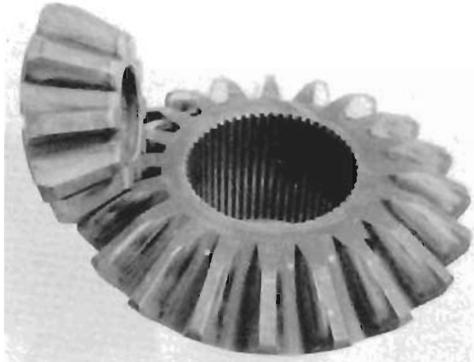


Fig. 1-4. Straight bevel gear and pinion.

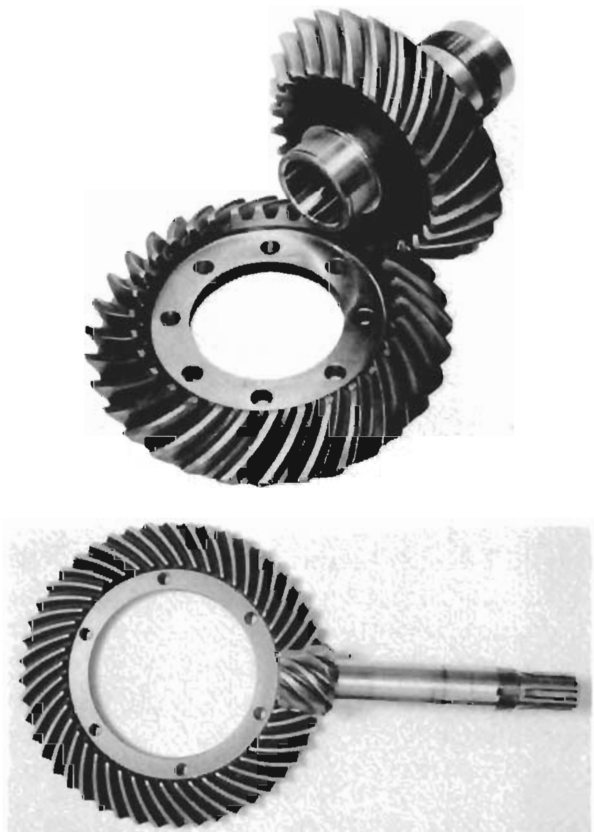


Fig. 1-5. Two types of spiral bevel sets, consisting of spiral bevel gear and pinion.

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.

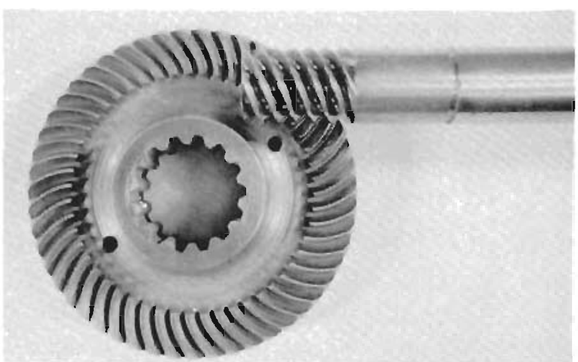


Fig. 1-7. High-ratio hypoid set.

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

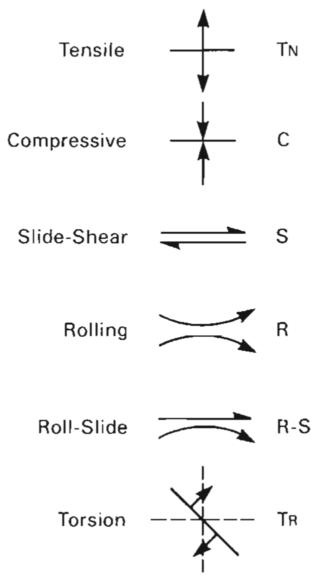


Fig. 1-8. Basic stresses that are applied to gear teeth. Often, two or three are simultaneously applied to a specific area.

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

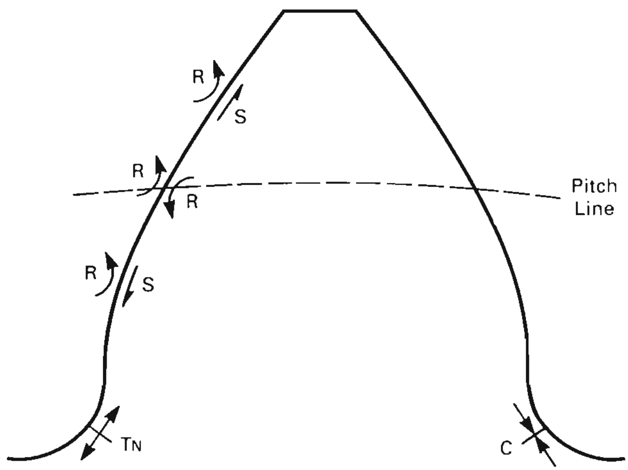


Fig. 1-9. Diagramatic stress areas on basic spur gear tooth.

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

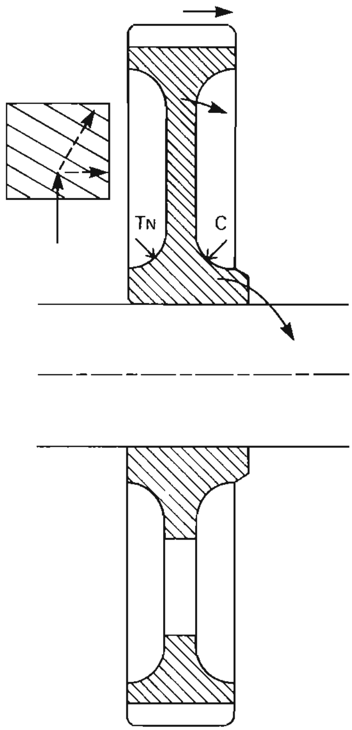


Fig. 1-10. Secondary stresses set up in associated parameters of a helical gear due to the side thrust action of the helix.

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

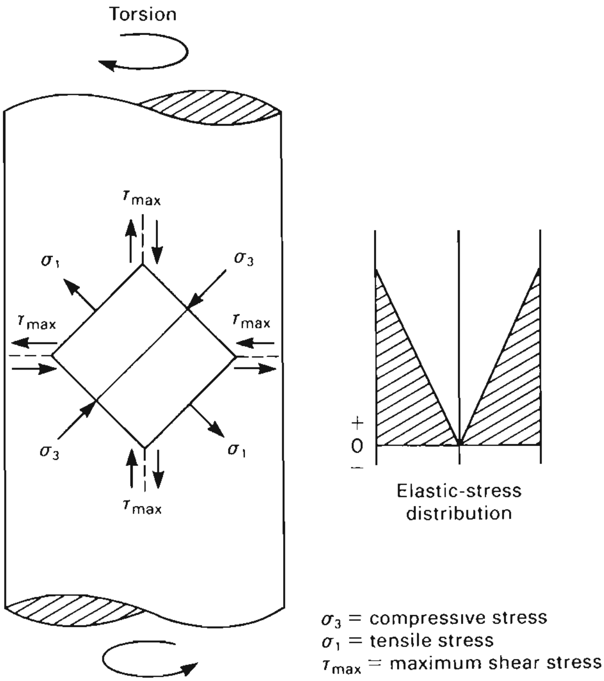


Fig. 1-11. Free-body diagram of maximum tensile and shear stress orientation on a surface element of a shaft in a torsional mode. Both maximums are at the surface. Stress is considered to be zero at the central axis.[4]

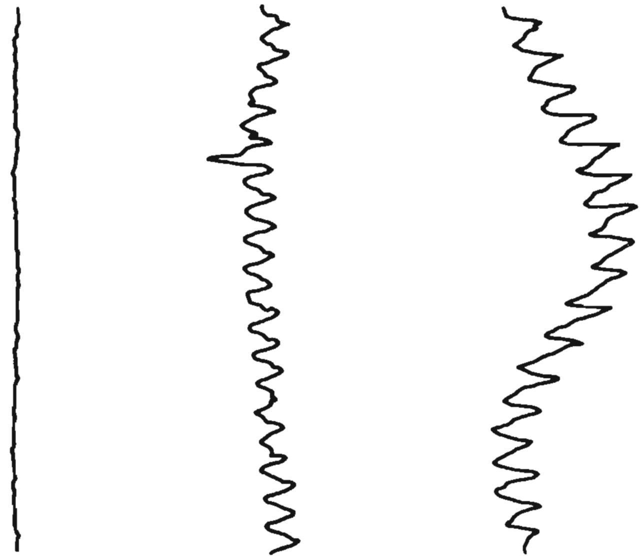
tensile failures of carburized parts are rare unless a specific type of stress raiser is present at the surface. This is not so with through hardened or non-heat treated parts since the strength is uniform throughout the part. Under this condition, torsional tensile failure is expected to originate at 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).

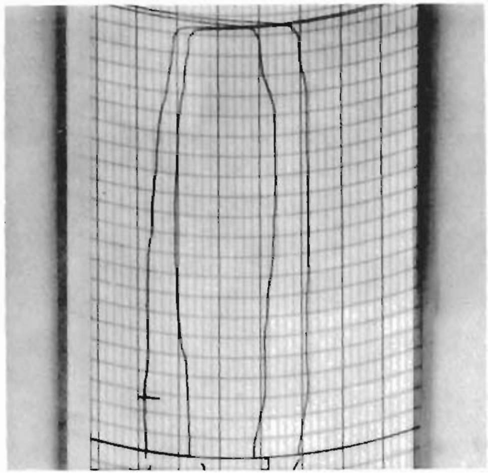


Fig. 1-13. Charted involute pattern. The straight line indicates true involute of the active tooth profile.

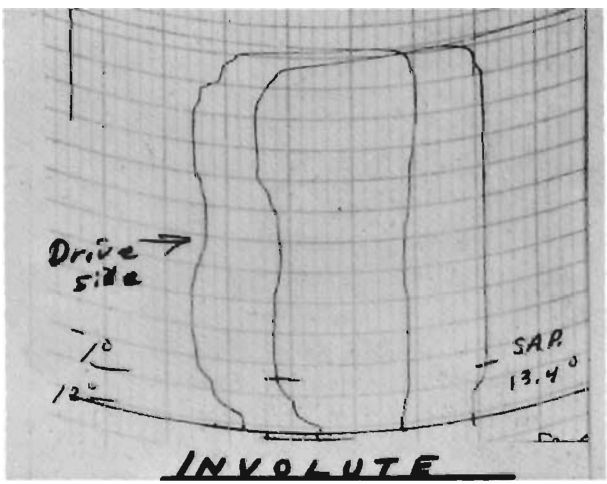
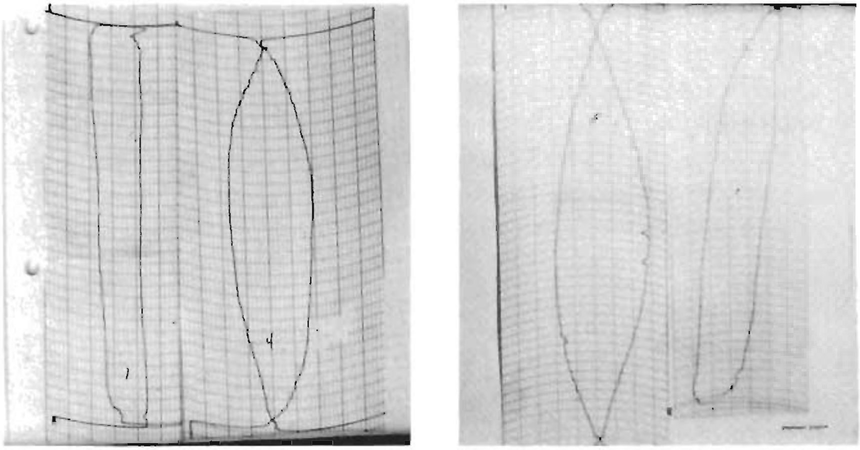


Fig. 1-14. Charted involute pattern of two teeth (180° apart) from a gear in service. Unused profile has an excellent involute; the loaded profile shows pitchline and tip wear.

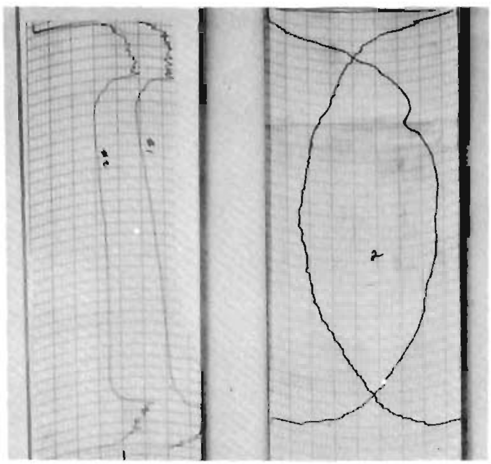
Again, this charting also can be used as an inspection tool for reading what has happened in service. As an example, Fig. 1-14 shows an involute pattern of two teeth of a gear returned from service. One side profile has not been worn; the graph is a straight line. The pattern of the opposite side shows 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) (b) (c) (d)



(e) (f)

(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

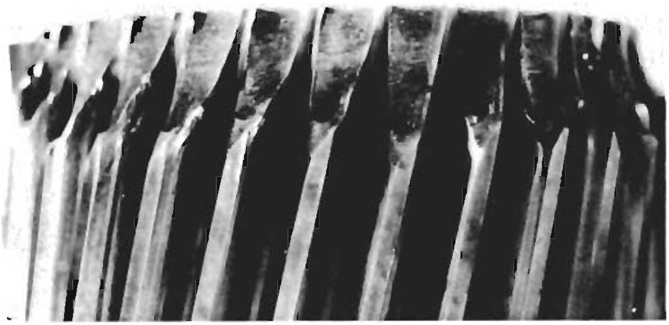


Fig. 1-16. Helical gear. All tooth characteristics were "perfect." Field failure had been matched with pinion showing lead pattern of Fig. 1-15(d).

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-

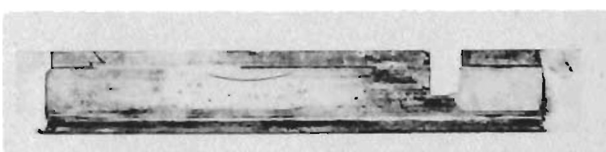


Fig. 1-17. Spur gear. Tooth crushing at midprofile contact area. Reason: mated with pinion showing lead pattern of Fig. 1-15(f).

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.

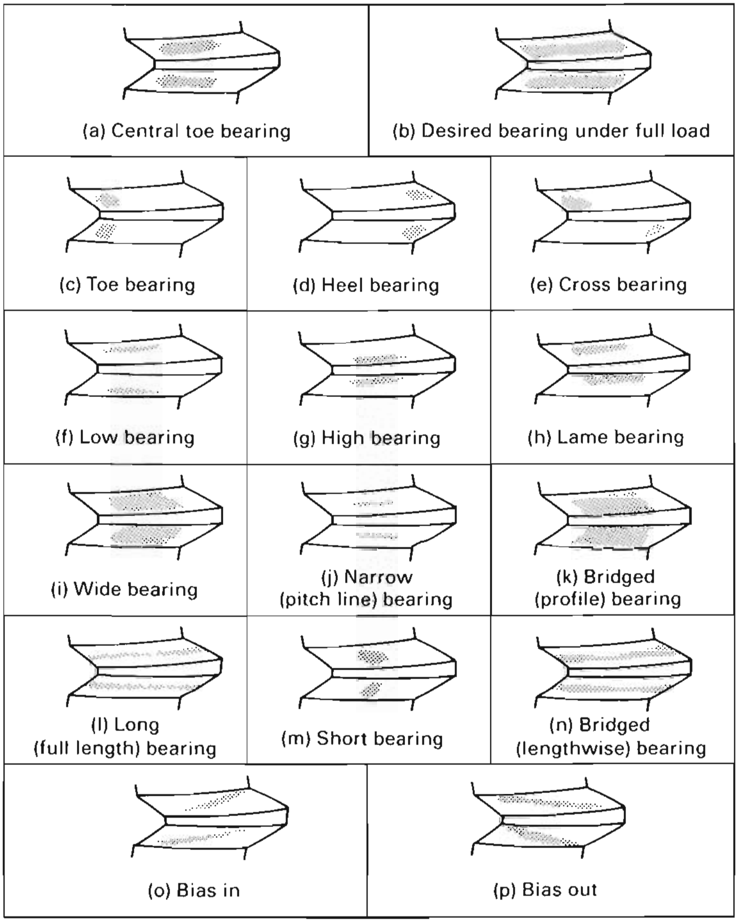
In general, the normal tooth contact pattern under no-load will be central profile toward the toe end. Ideally, as the load is increased to full-load, the deflection of the tooth causes a shift of the contact to a full-length pattern (shown in Fig. 1-18a and b).[7]

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.

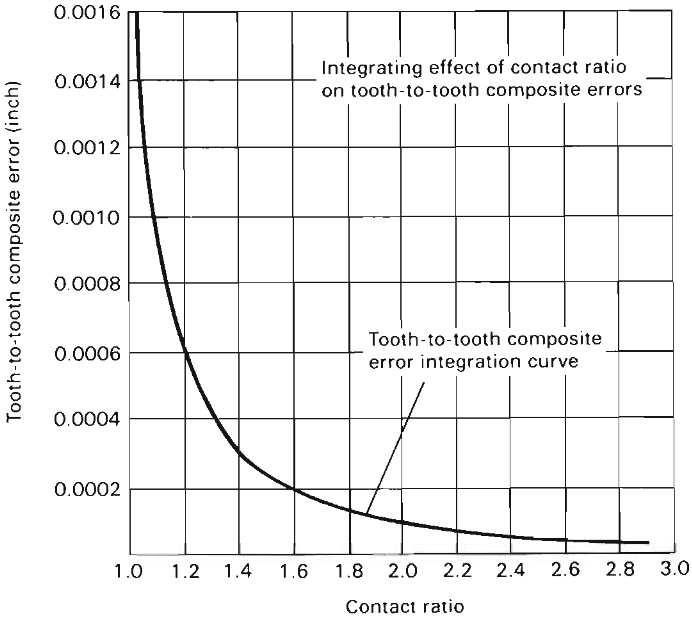
Composite Errors

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

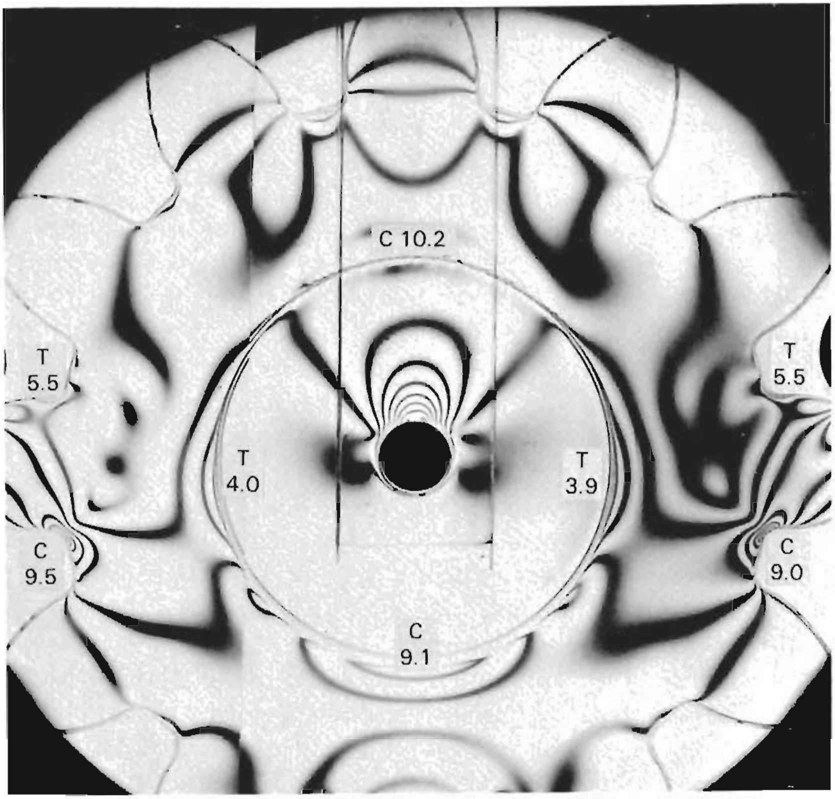


Fig. 1-20. Photoelastic pattern of idler gear between load input pinion and load output gear.

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.

References

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

CHAPTER 2

Basic Understanding of Environment

Environment is a set of conditions that exists in, around, and about an object, and has had, is having, or can have an influence on that object, either for betterment or for detriment during its lifetime or its present activity.

It is very important, when studying a gear failure, that the examiner obtain an understanding of all possible environmental factors. In order to accomplish this, he must determine the things to look for; and, more important, he must realize that it is indeed "things" he is looking for (Fig. 2-1). Yet these things are so intricately intertwined that it is very difficult to assess a quantitative value for any one of them. An attempt should be made, however; but how all-inclusive is environment when related to things? For instance, in a survey conducted by the author, a question—what is environment?—was asked, and the answers given were lubrication, temperature, and mechanical stability. It is also interesting that invariably the negative response was given; for example, lack of lubricant, or dirt in oil; parts running extremely hot; or a lot of deflections in housing, causing misalignment. Then there was a shift in the responses to the inclusion of subtler details, such as operation of the equipment, maintenance personnel and procedures, and finally, management attitude. Are all of these answers interrelated? They certainly appear to be.

Fig. 2-1. Environmental factors consist of mostly "things": lubrication, temperature, and mechanical stability.

Lubrication

Although the majority of persons—both professional and non-professional—tends to blame lubrication for most gear failures, there is no doubt in their minds that lubrication is a must. The blame appears to rest on not enough of, too much of, not the right kind of, breakdown of, or contamination of the lubricant. Looking at these causes, one immediately recognizes them to be faults not of the lubricant, but of the application of that lubricant, or of a number of external forces and conditions working against that lubricant.

If one were to consider what a specific lubricant is and what its performance is to be, it would be easy to understand the reasons for it not to function under certain conditions. Lubrication is accomplished on gear teeth by the formation of two types of oil films: the reaction film, also known as the "boundary lubricant," produced by physical absorption and/or chemical reaction to form a desired film that is soft and easily sheared but difficult to penetrate or remove from the surface; and the elastohydrodynamic film that forms hydrostatically on the gear tooth surface as a function of the surface speed. This second film is very thin, has a very high shear strength, and is not affected by compressive loads as long as constant temperature is maintained.

There are certain natural rules about a lubricant that should be remembered when gearing is designed, as well as when it is examined for failure:[1]

(*a*) Load from a gear tooth to its mating tooth is transferred through a pressurized oil film. If not, metal-to-metal contact may be detrimental.
(*b*) By increasing viscosity, a thicker oil film will be developed (keeping load, speed, and temperature constant).
(*c*) Heat generation cannot be controlled above a certain maximum viscosity (for a given oil).
(*d*) Breakdown of the oil film will occur when the gear tooth surface equilibrium temperature has reached a specific value.
(*e*) Scuffing load limit of mating tooth surfaces is speed-dependent. With increasing speed, the load required to be supported by the reaction film decreases, whereas the load required to be supported by the increasing elastohydrodynamic film increases. The result is a decreasing scuffing load limit to a certain speed as the reaction film decreases; then, as the speed picks up to the point at which the elastohydrodynamic film increases, the scuffing load limit increases, allowing an increase in the overall load-carrying capacity (assuming no change in temperature that would change viscosity).
(*f*) At constant speed, surface equilibrium temperature increases as load increases, lowering the scuffing load limit of the reaction film. NOTE: Surface equilibrium temperature is attained when the heat dissipated from the oil is equal to the heat extracted by the oil.

Damage to and failure of gears can and do occur as a direct or indirect association with the lubricant. There are several such occasions.

Incorrect lubrication means either the wrong specification was given for the application, or the right specification was not followed by the user. This means that type, quality, or viscosity was incorrect, and damage to the gears was accomplished in a short period of time. For instance, as one type of oil can be used for spur and helical gears, another type is necessary for spiral bevel or hypoid gearing. Keep in mind also that the bearings may be running in the same lubricant as the gears, and they, too, require that certain characteristics are present. A constant juggle of several characteristics of an oil is necessary in order to satisfy all moving components within a system.

The lubrication system cannot be overlooked or ignored. If a splash lubrication is working satisfactorily for a certain system, it may be completely inadequate if higher speeds are required. One very important function of proper lubrication is heat dissipation. Any lubrication system must take into account that the oil is supplied to the gear teeth at a greater rate than that necessary for lubrication, in order to remove frictional heat as quickly as possible. If not, the oil films will be broken down by continuing higher temperatures. After studying many works on the subject, T. I. Fowle[2] of Shell International Petroleum Company, Ltd., concludes: "Heat is best removed from the gear teeth by spraying the oil onto them when they are the hottest; i.e., as they come out of mesh. Only in the case of very high speed gears is there a danger that the amount of oil left on the teeth on reentry into mesh will be insufficient to form a lubricating film and, therefore, to need a supplementary supply at the ingoing side of mesh. And, only in the case of heavily loaded gears with large numbers of teeth and running at high speeds, do these authors conclude that better cooling than by spraying onto the gear teeth is needed." Some lubrication systems become very complicated; but the more complex the system, the more easily it becomes contaminated, and the more difficult it is to inspect and clean.

Lack of lubricant is perhaps the most devastating condition that can exist within a unit of moving parts (Fig. 2-2). There must be an amount of lubricant sufficient to form the two necessary oil films; but, of utmost importance, there must also be an amount sufficient to dissipate the heat away from those

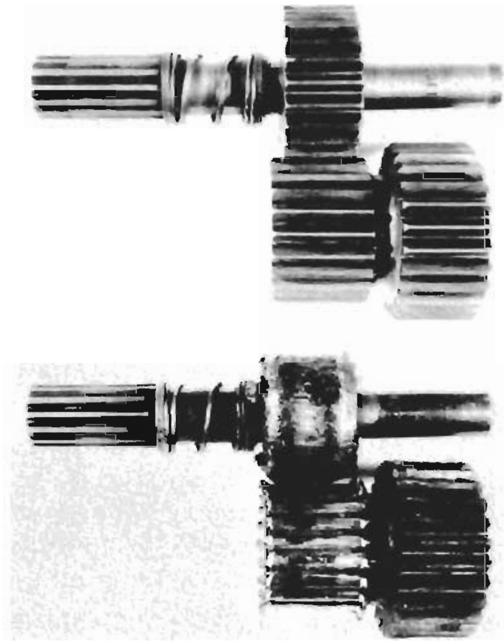


Fig. 2-2. Comparison of gear sets showing that lack of lubrication can be most devastating.

films to maintain their stability. Leakage is the most common means of oil loss, but occasionally an assembly will be placed into service without ever having the initial oil added. Oil can also be lost by vaporization or solidification under high temperatures and pressures.

Excessive lubrication presents some unusual problems in a gear train. The churning effect resulting from excessive lubrication can raise ambient temperatures, because the energy involved in cutting the shear plane of the oil is reabsorbed as heat. A continual churning of the oil, along with increasing temperatures, will tend to solidify the oil. In some instances, high vaporization pressure has broken seals, evolving a leakage problem.

Lubricant contamination is probably the most common lubricant-associated problem and is also the most complex.

Atmosphere contamination consisting of oxidizing vapors and water vapors tends to oxidize freshly worn surfaces, to break down the oil chemistry, or to set up cells of corrosion on metal surfaces. Liquid contamination may consist of water (which may also be contaminated with gases, acids, salt, or mud), anti-freeze compounds, liquid coolant compounds, or even other types of grease or oil added inadvertently. These contaminants will cause corrosion and general deterioration of both the lubricant and the metallic surfaces. Sand and dust particles introduced by atmosphere or water are common solid contaminants.

There is another very common solid contaminant that is internal to the gear system and is self-perpetuating. As gears are "running-in," an adjustment of the mating surfaces occurs with high rates of local wear, producing a fine metallic debris that is small, flaky, magnetic, and abrasive. As contacting surfaces become smooth, the rate of this debris production becomes nil. For this reason, many gear box assemblies have a running-in period at the assembly department, or the customer is instructed to completely flush the system and replace the entire supply of lubricant after 50 hours of operation. Other examples of solid contaminant are molding sand remaining in cast iron housings, welding rod splatter from repaired housings, drillings and turnings not cleaned out of machined parts, chips from nicks and bumps during assembly, carbon buildup from "burned" oil, shot and grit carryover from the gear-cleaning department, lapping compounds, grinding-wheel breakdown particles, and floor-cleaning compounds. Then, of course, whenever there is an initial stage of failure, those particles become contaminants that can easily propel further progression in astounding proportions (Fig. 2-3).

Temperature

Well over 90% of all lubrication has one specific function—to remove heat. It is difficult to separate this discussion into two distinct categories, such as causes of temperature increase and effects of temperature, because the causes and effects may be so closely related as to be simultaneous. For instance, two asperities come in direct contact with each other and one is sheared by the other. The energy of shearing is converted immediately to an extremely high temperature at the shear point that is above the flash point of the lubricant, and also above the critical temperature of steel transformation. The quenching of this instantaneous point of heat not only causes a slight temperature rise in the lubricant, but burns a particle of oil that forms a minute amount of free carbon black and transforms the steel spot to an untempered martensite. The martensite may be susceptible to cracking or to becoming an abrasive point during continued service. In other words, the most common cause of temperature rise is the conversion of mechanical energy to heat, which is measured by temperature. A few of the more obvious sources of temperature variants will be discussed in this chapter, though the discussion will not be all-inclusive.

Lubricants have a certain tenacity or resistance to shear. The higher the viscosity, the higher the shear strength. As the gears and other moving parts cut through the oil, the energy involved changes to heat and the temperature rises. It will continue to rise until the viscosity is reduced, the shear strength is reduced, and a surface equilibrium temperature is reached. A surface equilibrium temperature is attained when the heat dissipated from the oil is equal to the heat extracted by the oil. Any subsequent change in speed or load will change the surface equilibrium temperature.

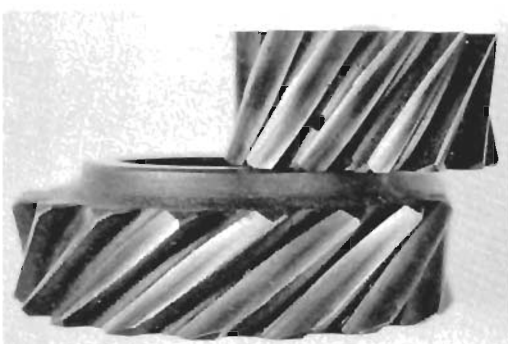


Fig. 2-3. Abrasive wear progresses rapidly and uniformly when an abrasive contaminant saturates the lubricant.

Ambient temperatures of the surrounding atmosphere will directly influence the temperature and viscosity of the lubricant. Extremely low temperatures may cause a lubricant to become so viscous it cannot function. In these instances, the oil may be heated externally and used in a reverse function, transmitting heat back into the gears. More frequent, though, is the occurrence of higher atmospheric temperatures. As temperatures increase, viscosity decreases, the oil film on gear teeth becomes thinner, and the friction load increases, causing scuffing and scoring and higher temperatures. The optimal condition exists when a surface equilibrium temperature is maintained with a sufficient oil film remaining.

Two identical units were operating in the same locality in a hot, arid climate. Both were high-speed gearing units exposed to direct sunlight. One user liked bright, shining equipment, so his unit was covered with a bright, metallic, aluminum finish. The other unit was covered with a dull, black finish. The sun heat absorbency of the black unit was so much greater than that of the metallic unit that it allowed premature scuffing of the gear teeth in the black unit.

Rising temperatures will expand the volume of both the lubricant and the entrapped vapor in a closed unit. As volume increases, pressure increases, and seals may break loose. However, the housing of the closed unit is usually made of steel, cast iron, or cast aluminum—all three with a higher thermal coefficient of expansion than the liquids and gases inside, which will tend to alleviate the internal pressures. Again, higher temperatures will increase the vaporization of the lubricant, and the vapor pressure will increase. If the vapor volume is large compared to the lubricant volume, the vapor pressures will rise slowly and may not be a factor. However, if the vapor volume is small compared to the lubricant volume, the vapor pressures may increase rapidly and become dangerous.

Gear tooth operating temperature is a significant source of heat because loads that cause friction and deformation of the surface are changed from mechanical energy to heat energy. For instance, under normal loading conditions of 3300 lb. per inch of face, and a speed of 7500 rpm, a spur gear tooth was calculated to have a rise in surface equilibrium temperature of 27 °F (15 °C) at the point of maximum compressive and sliding stress, which was just below the pitchline on the dedendum as the teeth were coming out of mesh.[3]

Components, such as gears, bearings, brakes, shifters, and seals, all tend to have rubbing contact or friction. The frictional energy is converted to heat and should be absorbed by the lubricant. As long as there remains an oil film between two metallic surfaces, friction will be at a minimum and metallic seizure or scuffing will not occur. As the surface equilibrium temperature rises, the protective film over the gear teeth decreases. At some point, the first danger point is reached; i.e., the film can no longer withstand the load, and metal-to-metal contact occurs. Rubbing, scuffing, scoring, seizing, and galling of the gear teeth ensue, which rapidly increase the amount of generated heat. The heat generation may progress faster than the lubricant will absorb. In that case, the gear itself will reabsorb the heat, its temperature will rise, and the second danger point becomes evident: the tempering of the gear.

The tempering of the gear, when certain temperatures are reached, reduces the hardness and allows a faster progression of frictional wear. For instance, a carburized gear is placed in service at a hardness of 61–62 on the Rockwell "C" scale (HRC). The normal tempering temperature during production was 325 °F. Any service-induced temperatures up to that point will have no effect upon the original hardness. However, suppose higher temperatures are suspected. What are the clues? Not only is lack of hardness a clue, but (if air is present) the surface of the parts will change color. This is called "temper color":

Temperature (°F)	Hardness expected (HRC)	Temper color
350	60–62	None
400	57–59	Light straw
450	55–57	Dark straw
500	52–55	Dark purple
600	47–52	Medium blue
700	35–47	Very light blue

The third danger point soon becomes apparent. Not only does the lubricant cease to exist in any state of usefulness, but the gear teeth melt and become extinct (Fig. 2-4).

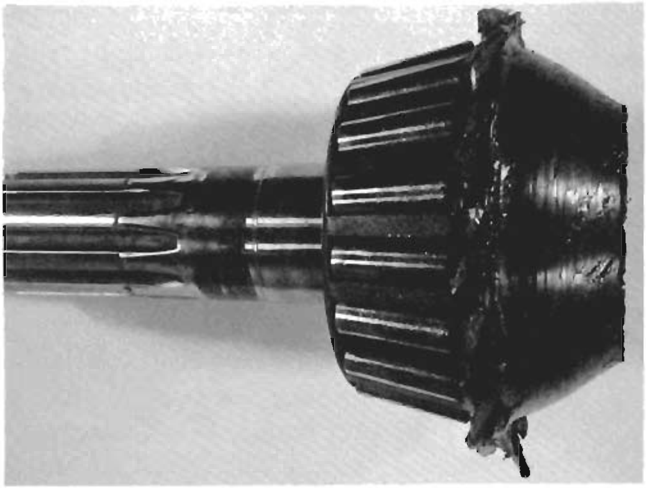


Fig. 2-4. With continuing rise of temperatures, the melting point is ultimately reached.

Mechanical Stability

All gears within a composite system depend not only on their interaction, but on the action and reaction of every other component within the system. However, before any action or reaction occurs within the unit, each component must be perfectly aligned with its functioning counterparts (Fig. 2-5). Reviewing this important step, in the first place, each part must be machined correctly with the awareness that each dimension and surface finish is crucial not only to itself, but to its functioning counterpart. Any mistake in the quality of manufacturing may be a multiplying factor in the breakdown of the entire unit.

Size change in gears that occurs due to change in temperature might have been discussed under the "temperature" heading; but size change is also mechanical. Nevertheless, the diameter of the pitch circle of a gear will increase in size in direct proportion to an increase in temperature. The linear coefficient of thermal expansion for the average gear steel is 6.36×10^{-6} in. per inch per degree Fahrenheit. Why is this important? Many gear sets running at high speed with minimal backlash will soon reach a zero or negative backlash with an unexpected rise in temperature. The ensuing pressure will break down the lubrication film, resulting in high frictional wear.

It is imperative when inspecting a failed gear to ask from what direction this failure occurred and what conditions existed to cause this directionality. These questions invariably will lead the analyst to check alignment, deflections, stability, tooth characteristics, and dimensional accuracies.

These accuracies are so very important that any deviation from the norm will tend to throw a tooth contact pattern from its normal central area to perhaps an end-face contact, or from a broad contact to a point contact. Each condition will drastically increase the applied load per unit area, which tends to increase immediately the surface equilibrium temperature and to break down the lubrication film.

In any transmission or power train equipment there are vibrations. Even the smoothest source of power will generate an output in pulses. Roller bearings vibrate; gear teeth vibrate; rotating shafts vibrate. Each part vibrates in its own peculiar amplitude and frequency. There is usually no problem unless one or more add together proportionately to form a high noise level, or to form a sequence of extremely high load peaks. Some ana-

Fig. 2-5. Perfect alignment is necessary for accurate results.

lysts claim that surface fatigue is caused by torsional vibration in approximately 85% of the cases. This figure may be high, but torsional vibrations, or chatter, will often reduce the fatigue life of a unit drastically.

Interestingly, under the heading "Life Adjustment Factors for Environmental Conditions," the Timken Company lists the following factors, relative to tapered roller bearings: load zone (as a function of end play, preload, geometry, clearance, temperatures, and housing deformations); misalignment; and lubrication.[4]

Personnel-Related Activities

This subject is as much an environmental factor as any of the above and should be recognized as such.

Assemblymen putting parts together in one unit or in placing several units into a composite machine must be scrupulous regarding cleanliness and accuracy of alignment. There is no allowance for error at this point of the process. Contamination at this point is instantly destructive. Misalignment will immediately place one or several parts under stress.

Operators of the equipment are the first line of defense. A good, conscientious operator will have his equipment outlast any other in the field. The author has said many times: "When a catastrophic failure occurs, don't look for a microscopic cause. Find the cowboy at the wheel or the one that threw the monkey wrench into the works!"

Maintenance personnel, equipment, and procedures will make or break a company with several pieces of equipment that must be kept in good working order. Not only must good replacement parts be used, but care should be taken that all used parts are cleaned and thoroughly inspected before being placed back into service. No gear should be reused unless a magnetic-particle inspection method has determined a freedom from crack propagation. No mated gear or pinion should be run unmated or with a random part. Again, cleanliness and correct alignment must be maintained during the reassembly of repaired items. There are many users of equipment who have programs of preventive maintenance, which are in various degrees of sophistication from well-thought-out, systematic checking of all components to a yearly hit-or-miss checkup of only the major operating parts. It appears that the slogan of some programs is "If it breaks down, fix it!" The utilization of these programs leads directly into the next category of personnel-related activities—management attitude.

Management attitude. This is an aspect related to personnel that is often overlooked. Those in management need to determine how their attitudes affect the work habits of those working for them. Very few assembly, maintenance, or operational men will consistently do a job better than that which is expected of them. Expect the best from your workers, and you'll get the best from your equipment. Give them good tools and good equipment along with your good attitude, and they will do a good job.

Several years ago, a routine inspection of a new planetary system was sponsored by a major user and the gear manufacturer. The inspection was done on the working sites, which were open-pit mines located over the full breadth of both Canada and the United States. In only four of 143 instances were there found negative environmental conditions:

- A truck that pulled up for inspection was making a lot of rear-end noises and had been for quite a while. It was discovered that a large idler gear had broken away from the holding-cap screws and was running freely. The gear was being held in place partially by the side of the housing, which by the time of discovery had worn a large area ¼-in. deep into the housing.
- The cover plate of a wheel unit was removed, and out dropped a blob of black grease with the consistency of lard. The reason? The bottom drain plug had been battered by a large boulder and could not be removed. Con-

sequently, the unit had never been drained or refilled. The transformation of oil to lard resulted from a constant churning for over 43,000 miles. The gears were in very good condition.

- When the cover plate of another wheel unit had been removed, it was discovered the gears were running in a bath of water with only a small amount of oil. An explanation was forthcoming from the maintenance foreman: "I guess when I told that new boy to fill up the unit with oil from a 5-gallon can which was across the room, he must have picked up the open can by the drinking fountain by mistake." The gears were still in very good condition.
- In one southwestern state, two drivers were in constant argument as to the strength of the truck each was driving. One truck had a conventional drive consisting of drive shaft, differential, reduction gears, and axle shafts. The second truck had electric motor-driven wheels that drove through a planetary gear set. The criterion for settling the argument? Nose both loaded trucks against the vertical wall of the pit and gun the engine. Whichever truck stalled the engine lost. The result? The truck with the conventional gearing stalled out first. It lost. It also lost two twisted and split axle shafts, two broken and mangled wheel-reduction assemblies, and one masticated differential. The engine of the second truck never even slowed down! It won the privilege of having two melted 350-hp, DC motors replaced, as well as all the resistance panels and wiring. Two drivers were out looking for new jobs.

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.

CHAPTER 3
Systematic Examination

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 correct answer to the problem. In fact, if there are several components to an assembly, each component must be suspect until eliminated by the examiner.

Systematic examination implies the use of a specific system or procedure to finalize a conclusive answer. It may sound laborious, and it may even be boring; but systematic means methodical in procedure or plan, and this means to begin at the beginning.

There are three types of approaches that have always been frustrating because so much information is lacking.

Approach No. 1. Several very good books and many articles have been written on failures. Invariably, each is a case history; that is, a picture is shown, and the cause and effect are explained. The assumption is that any part that looks like the picture should fail in the same way for the same reason. This is not so, for every failure is an individual case and must be treated as such. The unfortunate fact about a case history is that the author often does not explain the steps he took to reach his conclusions.

Approach No. 2. Often a failed gear or a small segment of a failed gear is set on a failure analyst's desk with the sincere question, "What kind of a failure is this, and what caused it?" Most of the time, an immediate answer is required, though often the background information supplied is grossly inadequate to enable an analyst to find a solution quickly. It is a difficult situation for an analyst to be in, but a tentative answer must be given, based solely on past experience and a whole bunch of "ifs."

Approach No. 3. This approach is illustrated by an example: A man was using his electric shaver one morning when it stopped cutting and started pulling. The change in tactics was painful. He had difficulty removing the cutters from the head because they had not been cleaned for a long time and were severely caked with whiskers and grease. When all the parts were laid out, he found a broken actuating spring. He was about to brush all the parts and the gunky mess into a white envelope and take them to the shaver supplier, but instead, he did what any good engineer would do. First, he washed all the parts under the hot water faucet to remove the grime. Then he put the clean parts in a white envelope and took them to the supplier. "Look what happened to this spring," he said. "Shouldn't I have a new shaver?"

This is not meant to advocate the return of all gears in a grimy condition, but it would certainly be helpful to the examiner for the environmental conditions to be carefully documented. The examiner should visit the field application to see first-hand the failed parts as they are being disassembled and to observe the conditions under which they had operated. Since the metallurgical examiner rarely has this opportunity, field service personnel assume this responsibility. The following instructions are specifically for them.

Field Examination

In the examination of failed systems, documentation is very important. It is not a good practice to rely on memory or oral transcription. Photographs and sketches are excellent for documentation, along with written history. Record the type of equipment, model number, serial number, date of purchase, date of failure, number of operating hours, number of miles, and type of service (continual, daily, yearly, seasonal). Note the loading cycle involved. (Is it in only one direction, or is it in both directions? Is it steady under maximum load, or is it intermittent? Is it uphill, downhill, over roads, or over rough terrain? Is it a gradual loading, or is it an abrupt impact type?) Copy the maintenance history relating to the units in question. (What is the type of lubricant and its condition? Any recorded temperatures? Had any parts been previously replaced? Had the gear assemblies been disassembled? If so, had they been submitted to magnetic-particle inspection before being reassembled?) This is all past history, but it might contain the key to present problems.

The next step is to remove the assembly from the equipment. Note the condition of the retaining bolts—tight or loose. Has the assembled unit been leaking oil or is it dry? Does the assembly remove easily from the equipment or is there obvious difficulty? Has the lubricant been drained from the assembly? What is the condition of the drained oil? Are there any visual contaminants in the oil? If so, are the contaminants magnetic or not? Note the assembly number and all other stampings or identification marks on the assembly. If helpful, take photographs.

The cover of the assembly may now be opened. Note the condition of the bolts. As the cover is being removed, note the condition of the seal, especially important if there has been oil seepage. What is the appearance of the remaining oil on the parts? Does it appear that lubrication was sufficient throughout the assembly, or are there dry areas? Is there evidence of corrosion (rust) anywhere? Always keep in mind the spacial relationship of all parts within the assembly. The top of the assembly may be dry, or it may be rusty due to condensation. The bottom of the unit will act as a sump and will collect all types of metallic fragments, as well as water. Gears and bearings low in the assembly will continually grind up metallic parts and, if there is moisture present, might corrode easily. Could photographs help at this point?

Each part is now to be removed from the assembly. The operation must be methodical. Starting at the power input, take out each part in order, record all identification marks, and note the condition of the part. Keep parts in some sort of order. Many times, a person will spot the failed part, take it out of the assembly first, and walk away with it to get the answer, assuming that its failure was both cause and effect. This is often an inaccurate assumption. Rather than removing it, take a photograph showing the relationship of all parts involved.

Now that all parts of the assembly have been reviewed, determine which fracture is primary and which are secondary. A primary fracture is one that occurs initially; other fractures occurring subsequent to the initial fracture are secondary fractures. Determine if the primary fracture is random (only one) or one of many. If there are many, is there a pattern to the failures? For instance, are the gear teeth failing in no particular sequence, or is every sixth tooth failing? If there is a pattern, determine what has influenced that type of pattern. Count the number of teeth in both gear and pinion. If the pinion has an odd number of teeth (hunting tooth), perhaps every gear tooth will be affected. Take a good look at the tooth contact patterns. Are they well centered or are they at one end? (As a reference, study Fig. 1-15.) Is there evidence of runout or taper in any of the parts? Runout may have been in the original cutting of the gear teeth, but it may be caused by a bent shaft. If the failure is surface deterioration, are the affected areas consistent, variable, or random? Try to find out why.

When the primary failure has been determined, a decision should be made regarding the mode of the failure; then it should be decided if the failed part or parts are to be sent to the gear failure analyst. If the primary failure is in one of two mating gears, by all means send both parts. If there are several gears in an assembly and there is a question of cause and effect, send all parts. In any situation, send all documented information available.

The field service man should now make some very important observations. He should, if at all possible, determine the cause of the primary failure. If the cause is in the assembled unit itself and is not externally controlled, the failure may be beyond the scope of his control. It could be a wise move at this time to send the entire subassembly back to the manufacturer, who should be in a better position to make a detailed analysis. On the other hand, if he determines that the cause is external to the assembly and that a remedy or adjustment can be made in the field to keep other machines in operation, he should make those recommendations to his management.

Cooperation between field service personnel and the gear failure analyst is extremely important and cannot be overemphasized. The analyst sees usually one or two failed gears and must base his findings on the story they tell. Field service personnel usually have access to all other parts and to the background knowledge. The analyst may state that a set of gears is grossly overloaded. The field service person may then, for example, take the input shaft and find torsional fatigue cracks by the use of magnetic-particle inspection. Verification of facts by the sharing of knowledge should always be common practice.

So much is left unsaid about field examination, such an important aspect of gear failure analysis. Perhaps it will suffice to say that many bits of knowledge about a failure will point directly back to the field application and can in the future be used by those responsible for field examination.

When the failed parts and documented information are in the hands of the failure analyst, he too must be systematic in the study that is now before him. He must also "begin at the beginning" from his perspective.

Visual Examination

By far the most important phase of visual examination is that which is done by the unaided eye. In most instances, all of the field examination has been accomplished by this method; and now, the most important decisions of procedure must be made by this same method.

First, document the part that is being examined. Write down the part name, part number, and all other markings or hieroglyphics found stamped, etched, or embossed on the part. Next, describe in detail what you actually see; i.e., the physical condition of the surface, such as rubbing, scratching, and marks of abusiveness (hammer marks, cutting-torch marks, thread stripping, etc.). Then describe the tooth contact patterns from both directions and the amount of normal or abnormal wear on these patterns. Finally, describe in detail the appearance of the failure, and categorize it according to mode (see Chapter 4), specifically looking for the origin of the fracture.

Look at the failure. Study it closely. What does it reveal? The following example shows how an analyst determines the cause of a failure.

One spiral bevel drive gear was returned from a coal mining operation. Every tooth was crushed at and over the toe end, midprofile, convex (loaded) side (Fig. 3-1). The overall tooth contact area was well centered and extended almost full length. However, there had been a recent shift, since the load pattern went directly to the toe end at the area of failure. The customer field service was asked to check for pinion alignment and to submit the pinion for examination. It reported that there was "no misalignment and the pinion is okay." However, three more gears were returned with the toe end in exactly the same crushed condition. Again, the report stated that the "pinions are okay." By this time an analysis of the first gear had been made and all tooth characteristics were as specified. When three ad-

Fig. 3-1. Spiral bevel gear, 2.5 D.P. SAE 4820H, case depth 0.068 in., 58 HRC. Operation: coal mining. Every tooth crushed and subsequently broken at toe end, midprofile, convex (loaded) side.

ditional gears arrived with the same failure and with no mating pinions, the decision was made to visit the mine. The maintenance department was waiting with nearly a dozen gears on the floor, in the same damaged condition, but with no pinions. The first question asked was "Where are the mating pinions?" The reply: "They're all okay, but you'll find them over there in the corner." The pinions were then inspected visually. Every tooth of every pinion had a visible crack at the root radius, heel end, concave (drive) side that extended not only down the back face angle, but along the tooth root radius toward the toe end, to about three-quarters of the tooth length. This pinion condition caused full load to be applied from the toe end of each pinion tooth to the toe end of the gear teeth, which were crushed by the extremely high load per unit area. The pinion teeth were the primary failure; the gear teeth were the secondary failure. The entire story was told by the parts themselves, but someone had to listen.

This example points out four basic lessons of observation and logic that a failure analyst must know:

- Only the gear was submitted to the analyst for consideration. The analyst observed that every tooth was crushed at the toe end of the driven side, and that the normal tooth contact had been properly centered for the greater portion of its service. The next step was to ask two questions: What exerted the extremely high pressure contact at the toe end of every tooth? And why did the contact load shift to the toe end? Logically, the only answer to the first question is that the toe end of the pinion tooth is the only object available to impress the toe end of each gear tooth. The answer to the second question must be found by examining the pinion.
- The pinion was examined. Every tooth showed evidence of bending fatigue at the root radius of the drive side, from the heel end to three-quarters of the length toward the toe end. Each tooth should show a normal central contact pattern against the gear tooth until the bending fatigue crack originated. The progression of this fatigue crack automatically started the sequence of events as follows: the tooth deflection increases each time it is loaded;

the deflection allows a premature loading of the next adjacent tooth, which becomes suddenly overloaded until it originates a bending fatigue crack; and the deflection actually relieves the load at the point of contact and shifts the heavier contact load toward the toe end. Thus, with every pinion tooth failed at three-quarters of the heel portion, the entire load was concentrated at one-quarter of the toe end portion. This answers the questions about the gear's failure.

- The preceding discussion of observation and logic could have been handled by knowledgeable field service personnel, if only they had recognized that the fault was within the function of the pinion and had submitted both parts to the analyst. Time, always an important element, could have been saved not only in the final analysis, but also in a number of down-time hours used for replacing parts. This shows the very close association that should exist between field examination and failure examination. In this instance, the field service personnel did not read the failure correctly, nor did they submit enough evidence (the mating pinion) to the analyst. The analyst read the failure that was before him, recognized the problem, but did not have the part that provided the cause of the failure. Only when the whole story was together was something constructive accomplished.
- This example was used primarily to stress the importance of visual examination; but two conclusions stated were not definitive. The conditions of "crushed" and "tooth bending fatigue," although generally recognized by an experienced analyst, cannot be conclusively determined without microscopic examination. Also, without a doubt, even the experienced analyst would have required that the parts undergo microexamination before recommending corrective action.

Before leaving the above example there are other questions and comments that are pertinent to the present discussion. It was obvious to the analyst that the gear was a secondary failure; but the analyst could not be absolutely certain that the pinions had failed by tooth bending fatigue at the heel portion before he had ever seen the pinion. He was seeking a reason for the shift in load pattern that caused the extremely heavy toe contact. He found the specific reason. Other reasons could have been the breakdown of the pilot bearings, unstable assembly allowing for deflections under load, improper adjustment or readjustment in the alignment, or improper tooth cutting or distortion during manufacturing.

Although the primary failure was established within the pinion, it could have been possible that the crushing of the gear teeth was the primary failure. If the microexamination of the gear teeth had established a metallurgical condition that could never have withstood a normal load application; if the gear teeth had been cut or distorted enough to have allowed heavy toe contact; or if severe deflections or misalignment had occurred; the crushing of the gear teeth could have been the primary failure. Keep in mind that there had been normal tooth contact for an established length of time, and that the gear teeth had been analyzed and were found to meet the necessary metallurgical requirements for normal loading.

It is generally understood that occurrences of tooth bending fatigue are caused by overload. Is this always the case, and was it the case here? Tooth bending fatigue is a tensile overload, but specifically, of the actual design and metallurgical conditions existing. Theoretically, the pinion in question had not been overloaded. Actually, it was severely overloaded due to sharp undercutting at the root radii, shallow case depth, and insufficient surface hardness.

Assuming that all manufacturing and process characteristics for the pinion and the gear had been within acceptable limits, the same mode of failure could have happened, but probably not as extensively as to include every set in operation. The cause would necessarily have been an occasional high peak overload that would have resulted in a random tooth failure rather than a patterned failure.

Visual examination does not preclude the use of low-power microscopes up to 20× that can and should be used to study fracture surfaces. But the examiner should be careful not to overlook the lower magnification. Very often, the best resolution is 1× (eye contact) for an overall or larger view, then 3×, 10×, and 20×, in that order. Findings should always be reported under the magnification that clearly shows what has happened. Many times the story will best be shown with photography by a combination of two different magnifications: i.e., 1× or lower for position of a failed area, and 20× for details.

Since a visual pattern is the result of reflected light, much can be accomplished or hindered by the light source. When examining a fracture, it is best to use a rather small spotlight source that can be easily manipulated to give the light beam a directional pattern. Fatigue striations are more easily seen when the light beam is almost parallel with the fracture surface. This flat-angle beam will highlight the peaks and darken the valleys, allowing an excellent resolution for a photograph. Changing direction in lighting is often more important to photography than is intensity or magnification.

There are many pieces of equipment that can be utilized for visual examinations. The equipment alluded to in the above two paragraphs is described in detail by R. C. Anderson in Chapter 2 of his book *Inspection of Metals, Volume I: Visual Examination.*[1] This book is highly recommended as a supplement to any visual study of materials.

In the first paragraph of this section, it is stated that important decisions of procedure are to be made. The procedures that must now be determined come after the visual examination and the documentation of that examination. The decisions are to be made concerning which physical tests and which metallurgical tests are to be conducted. Since these procedures are to be accomplished after visual examination, there are some very important points to be remembered:

Do not jam two broken parts together to see how they fit. Delicate fracture surfaces may be altered.

Do not indiscriminately cut through a fracture surface for microscopic examination before completing visual examination. There must be a valid reason for sectioning. Finally, cut only in the area that is most likely to give a conclusive answer. It may be wise to take several photographs of the area before cutting.

Do not grind fractured areas flat to take hardness readings until after visual examination, and then only if necessary.

Do not clean fracture surfaces with the palm of your hand or with greasy rags, or remove any rust with an acid. Cleaning is best accomplished with a spray solvent and gentle air pressure.

Physical Examination

Physical examination involves any procedure of nondestructive testing, including magnetic-particle inspection, tooth characteristic studies, surface hardness testing, ultrasonic testing, nital etching, profilometer measurements, and dimensional checking.

Magnetic-particle inspection is perhaps the most useful and important procedure to use; as such, it is the first step in our systematic physical examination. There are several types of testing equipment that may be used in specific places or instances, but the wet method, using fluorescent magnetic particles, is the most common and the most revealing. In fact, photographs are accomplished more easily by this method (Fig. 3-2). The first purpose of magnetic-particle inspection is to determine the extent of damage. Perhaps only a portion of one gear

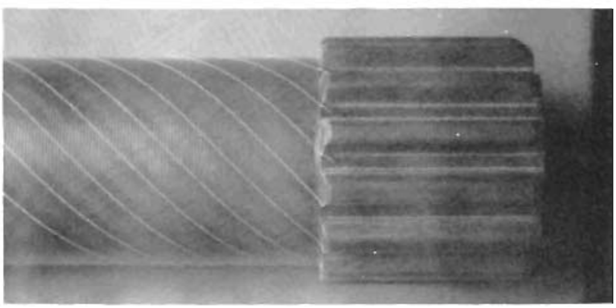


Fig. 3-2. Unidirectional torsion overload in a 1-in.-diam jack shaft, as revealed by magnetic-particle inspection by "Magna-Glo" method. Photographed using a standard "black" light and Polaroid Polacolor-2 Type 58, 4×5 Land film, ASA75, *f*/11, for one minute.

tooth appears fractured visually when, in reality, several teeth have cracked in the same area. More experience with the analysis of fractures brings recognition that the magnetic indications can clearly show the progression, or sequence of events, that leads to final failure. In fact, the indications may also reveal the point of origin on other teeth. If this is so, one should section for microexamination the area showing the beginnings of the failure mode that has not yet been destroyed or mutilated by additional damage. Other surface discontinuities discovered by magnetic-particle inspection may be surface or slightly subsurface inclusions, bar stock seams, forging laps, machine tool tears, grinding checks (Fig. 3-5), and quenching cracks.

Tooth characteristic studies give significant analysis concerning what has happened to the gear teeth. These studies are detailed in Chapter 1 and should be reviewed at this time to determine which are appropriate.

If there is any question about the tooth contact pattern on a spiral bevel gear/pinion set, the set should be placed in a spiral gear tester to determine actual field loading position. The tester should be adjusted first at the specified position to see if the parts had run as expected. Then the tester should be readjusted to the existing pattern. This will show the conditions of deflections or misalignment prevailing at the time of field service. (See "Mating-Tooth Contact Pattern" in Chapter 1.)

Surface hardness testing is always necessary but must be used with caution. Techniques of testing should be in accordance with the practices outlined by the manufacturer of the test. In failure analysis, surface hardness readings should be taken primarily in areas pertinent to the analysis. Most likely, a microhardness survey (discussed below under "Metallurgical Examination") will be advisable. Since a large percentage of gears have been carburized (or surface hardened), the case depth is significant in the interpretation of surface hardness results. For instance, a Rockwell "C" indenter and load might penetrate a case of less than 0.025-in. depth and give an erroneous reading. For lighter case depths, it would be wiser to use a hardness scale with lighter penetration loads. Table 3-1 is a

Table 3-1. Minimum work-metal hardness values for testing various thicknesses of metals with regular and superficial Rockwell hardness testers (a)[2]

Metal thickness, in.	Minimum hardness for superficial hardness testing, diamond "N" Brale indenter 15N (15 kg)	30N (30 kg)	45N (45 kg)	Minimum hardness for regular hardness testing, diamond Brale indenter A (60 kg)	D (100 kg)	C (150 kg)
0.006	92	—	—	—	—	—
0.008	90	—	—	—	—	—
0.010	88	—	—	—	—	—
0.012	83	82	77	—	—	—
0.014	76	80	74	—	—	—
0.016	68	74	72	86	—	—
0.018	(b)	66	68	84	—	—
0.020	(b)	57	63	82	77	—
0.022	(b)	47	58	78	75	69
0.024	(b)	(b)	51	76	72	67
0.026	(b)	(b)	37	71	68	65
0.028	(b)	(b)	20	67	63	62
0.030	(b)	(b)	(b)	60	58	57
0.032	(b)	(b)	(b)	(b)	51	52
0.034	(b)	(b)	(b)	(b)	43	45
0.036	(b)	(b)	(b)	(b)	(b)	37
0.038	(b)	(b)	(b)	(b)	(b)	28
0.040	(b)	(b)	(b)	(b)	(b)	20

(a) These values are approximate only and are intended primarily as a guide. (b) No minimum hardness for metal of equal or greater thickness.

"Hardness vs Minimum Thickness" chart for checking hardness of thin material.[2] It is also a fairly reliable practice to substitute "case depth" for "metal thickness" when checking the surface hardness of a carburized part.

Hardness is the only criterion normally used to interpret the strength of the part being examined and can be used interchangeably with confidence. Table 3-2 lists the approximate equivalent hardness numbers for steel as tabulated by the Society of Automotive Engineers *Handbook*.[3]

Ultrasonic testing is not universally required, but it is used to an advantage for revealing subsurface discontinuities. Ultrasonic testing can be utilized to locate subsurface casting defects such as porosity, shrinkage, or blow holes; subsurface voids or cracks at or near welded zones; large inclusions or

Table 3-2. Approximate equivalent hardness numbers for steel[1]

Rockwell C-scale hardness No.	Brinell hardness No., 10-mm standard ball, 3000-kg load	Rockwell Hardness No. A-scale, 60-kg load, Brale penetrator	Rockwell Hardness No. B-scale, 100-kg load, 1/16-in. diam ball	Rockwell superficial hardness No., superficial Brale penetrator: 15N scale, 15-kg load	Tensile strength (approx), 1000 psi
60	—	81.2	—	90.2	—
59	—	80.7	—	89.8	—
58	—	80.1	—	89.3	—
57	—	79.6	—	88.9	—
56	—	79.0	—	88.3	—
55	—	78.5	—	87.9	301
54	—	78.0	—	87.4	292
53	—	77.4	—	86.9	283
52	500	76.8	—	86.4	273
51	487	76.3	—	85.9	264
50	475	75.9	—	85.5	255
49	464	75.2	—	85.0	246
48	451	74.7	—	84.5	237
47	442	74.1	—	83.9	229
46	432	73.6	—	83.5	222
45	421	73.1	—	83.0	215
44	409	72.5	—	82.5	208
43	400	72.0	—	82.0	201
42	390	71.5	—	81.5	194
41	381	70.9	—	80.9	188
40	371	70.4	—	80.4	181
39	362	69.9	—	79.9	176
38	353	69.4	—	79.4	171
37	344	68.9	—	78.8	168
36	336	68.4	—	78.3	162
35	327	67.9	—	77.7	157
34	319	67.4	—	77.2	153
33	311	66.8	—	76.6	149
32	301	66.3	—	76.1	145
31	294	65.8	—	75.6	142
30	286	65.3	—	75.0	138
29	279	64.7	—	74.5	135
28	271	64.3	—	73.9	132
27	264	63.8	—	73.3	128
26	258	63.3	—	72.8	125
25	253	62.8	—	72.2	122
24	247	62.4	—	71.6	120
23	243	62.0	100.0	71.0	117
22	237	61.5	99.0	70.5	114
21	231	61.0	98.5	69.9	112

Table 3-2 (continued)

Rockwell C-scale hardness No.	Brinell hardness No., 10-mm standard ball, 3000-kg load	Rockwell Hardness No. A-scale, 60-kg load, Brale penetrator	Rockwell Hardness No. B-scale, 100-kg load, 1/16-in. diam ball	Rockwell superficial hardness No., superficial Brale penetrator: 15N scale, 15-kg load	Tensile strength (approx), 1000 psi
20	226	60.5	97.8	69.4	110
—	219	—	96.7	—	106
—	212	—	95.5	—	102
—	203	—	93.9	—	98
—	194	—	92.3	—	94
—	187	—	90.7	—	90
—	179	—	89.5	—	87
—	171	—	87.1	—	84
—	165	—	85.5	—	80
—	158	—	83.5	—	77
—	152	—	81.7	—	75

"pipe" in steel parts (Fig. 3-3a); internal rupture at the case and core transition zone in gear teeth (Fig. 3-3b); and extreme variations in material composition.

Nital etching is used to locate tempered areas on hardened steel surfaces. This method should not be confused with the nital etch used for microstructure examination; the solution, procedure, and purpose are not the same. The most common usage for the subject etch is to locate grinding burns—spots or areas on a ground surface overheated by the grinding wheel. Figure 3-4 shows distinct chatter burns extending the full length of the profile of helical gear teeth; the tempered spots are softer than the remaining carburized surface and more susceptible to surface breakdown. As Fig. 3-5 shows, by nital-etching a failed gear, a darkened surface reveals burning of the entire end face by severe grinding, and magnetic-particle inspection reveals that grinding checks of the same pattern as the fracture origin also occurred.

Nital etching is also used extensively to show an induction hardened pattern. The induction heated zone will show a very clear pattern on the end face of gear teeth. Figure 3-6 shows an end face pattern of a consecutive-tooth-space hardening tech-

Fig. 3-3(a). A massive flat inclusion parallel to the top face of a spur gear tooth at a distance of 0.228 in. from the surface. This inclusion gave a sharp sonic indication for a distance of 1/2 in. along the tooth length. 3% nital etch, 200×.

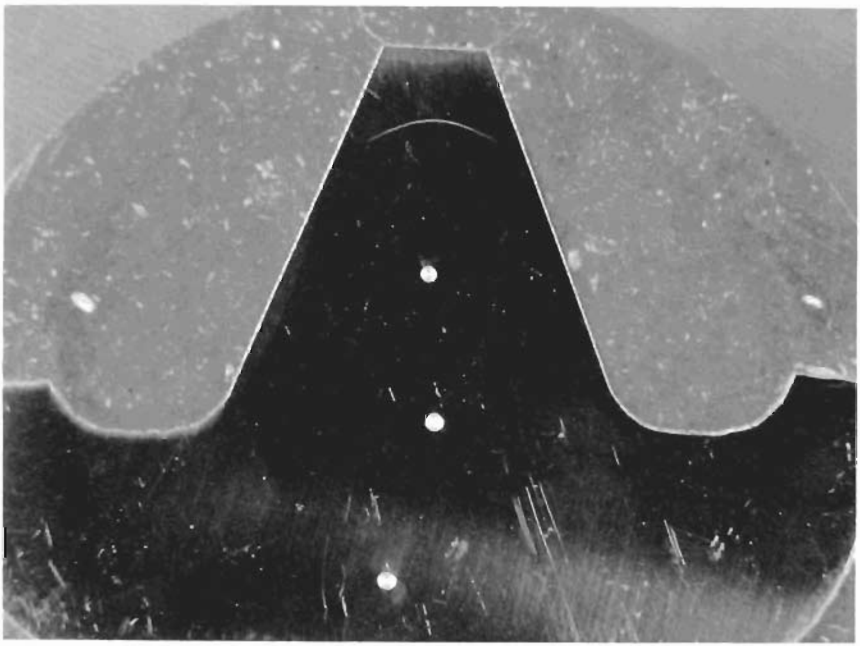


Fig. 3-3(b). An internal rupture in a gear tooth at the case/core transition zone which does not reach the surface. This condition can be discovered by ultrasonic testing.

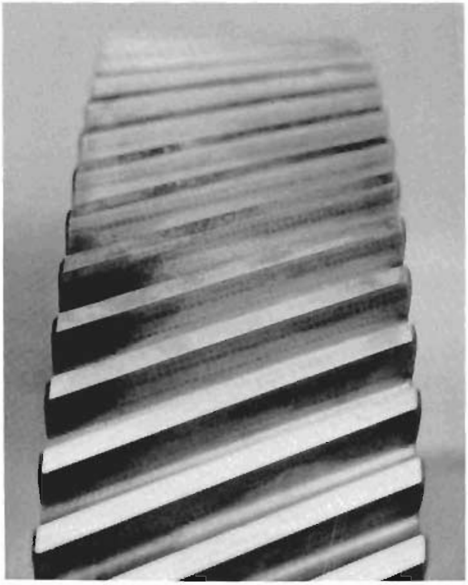


Fig. 3-4. Helical gear carburized and hardened to 60-63 HRC. A chattering tooth grinding wheel has "burned" small consecutive spots the full length of the profile on several teeth. Discovered by nital etching.

Fig. 3-5. Spur gear, end face ground. Grinding severity tempered the entire face (revealed by nital etching) and checked the surface (revealed by magnetic-particle inspection).

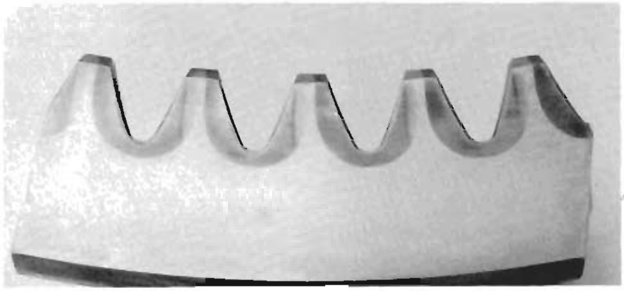


Fig. 3-6. End face nital-etched pattern of a consecutive-tooth-space induction hardening technique. Pattern is exaggerated and distorted due to the extra heat picked up by the square edges. The true pattern should be established below the surface.

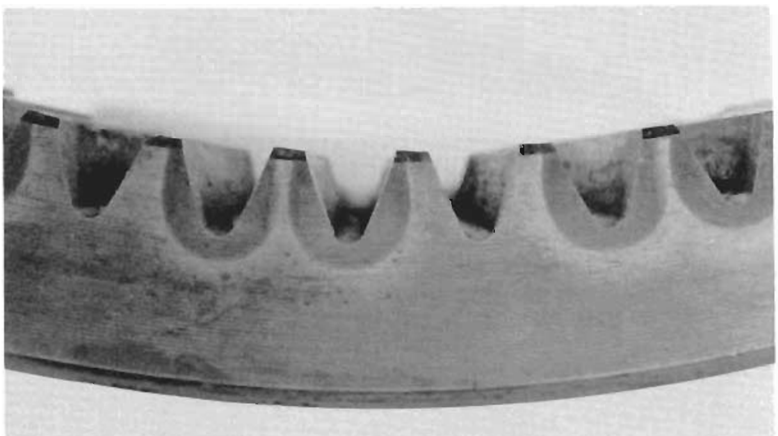


Fig. 3-7. End face nital-etched pattern of an induction hardened internal gear, showing two tooth spaces that had not been hardened.

nique used on a large internal wheel reduction gear. The pattern shown is exaggerated due to the end effect of the square edges picking up a greater amount of flux. Since tooth end surface patterns of induction hardening processes can be somewhat distorted, it is recommended that they be used only as a guide, and that the true pattern, which may be 1/4 in. from the end face, be examined. Also, as an inspection tool, nital etching

Table 3-3. Nital etch production test for ground tooth gears

Solution 1 – 5% concentrated nitric acid in denatured alcohol.
Solution 2 – 10% concentrated hydrochloric acid in denatured alcohol.
Solution 3 – 15% concentrated ammonia in denatured alcohol.
(All solutions are to be used at room temperature.)

Etching procedure

1. Clean gear of all grease and oil (trichlorethylene).
2. Place part in Solution 1 for 20 seconds.
3. Remove and rinse in clean cold water, dry with air pressure.
4. Place part in Solution 2 for 20 seconds.
5. Remove and rinse in clean cold water, dry with air pressure.
6. Immerse part in Solution 3.
7. Remove and rinse in clean cold water and then dry with air pressure.
8. Immerse the part in rust-preventive oil.
9. Tempered areas due to grinding burns will be dark against the light-gray background.

will quickly point out discrepancies such as the two nonhardened tooth spaces shown in Fig. 3-7.

There are several very similar nital etching processes being used by industry as a production check for all ground tooth gears. One procedure is outlined in Table 3-3. However, if a quick check of only a few parts is to be made either in the field or in the lab, induction hardened patterns can be etched by swabbing the areas in question with a 5% nital solution (5 ml concentrated nitric acid with 95 ml denatured alcohol) and by rinsing with warm water.

Profilometer measurements show the variations in average roughness height that occur on the surface being tested. This may be important information where surface deterioration is evident, or at radii and other points of high stress concentration.

Dimensional checking has not been mentioned in Chapter 1 as a characteristic, but its importance is self-evident. Wrong dimensions may often place a high stress raiser in an area of high stress concentration. If this condition had been in the design, proof of the results should demand a change in that design.

Metallurgical Examination

In this section will be discussed the tests that necessitate the destruction of some portion of the part for further study. The decision to cut into a part must be done with forethought and knowledge; and the examiner should ask whether it is necessary to section the part to gain additional data; what the data are that must be known; and where the cut must be made in order to make the data meaningful.

It is often the case that cutting merely for the sake of cutting turns out to be an exercise in futility. However, when it has been established that sectioning a part is necessary, one or more of the following procedures will be applicable:

Cross-sectional hardness survey
Macroscopic examination
Carbon gradient traverse
Chemical analysis
Case hardness traverse (microhardness)
Microscopic examination
Scanning electron miscroscopy

The cutting or sectioning of a part must be done very carefully to prevent the heating of any structure. Lower heating may temper a surface, and higher heating may radically alter the microstructure. Parts of lower hardness (less than 38 HRC) should be cut with a mechanical hacksaw, using a continuous stream of cutting/cooling fluid. Harder parts, including carburized parts, should be cut with a relatively soft, powered, abrasive wheel that will break down readily to ensure a nonglazed surface. A glazed surface will tend to "burn" the sample. At the same time, a heavy spray of coolant should play on the part at the cutting edge. In fact, two sprays are recommended, one at the cutter inlet and one at the cutter outlet.

Electrostatic methods are also very accurate means of cutting sections. But these methods metallurgically alter the cut surface for a depth of a few thousandths of an inch, and the surface then must be ground sufficiently to eliminate this condition.

Cross-sectional hardness survey is used for two purposes: to determine the actual strength of a failed part, and to determine if the heat treatment was optimum for the material used.

This procedure, which is generally used on pinion shanks or shafts that have failed, is done as follows: cut laterally across the part, making a cross section about 3/4 to 1 in. thick, as close to the fracture area as possible without including any altered structure. Place the sample in a surface grinder and grind (cool) both sides so they are parallel. Take surface hardness readings at four positions and record. Take readings across the sampled section at 1/16-in. intervals, starting 1/16 in. from the surface at the established four positions and continuing to the center (Fig. 3-8). Record each reading. Average the readings taken at each

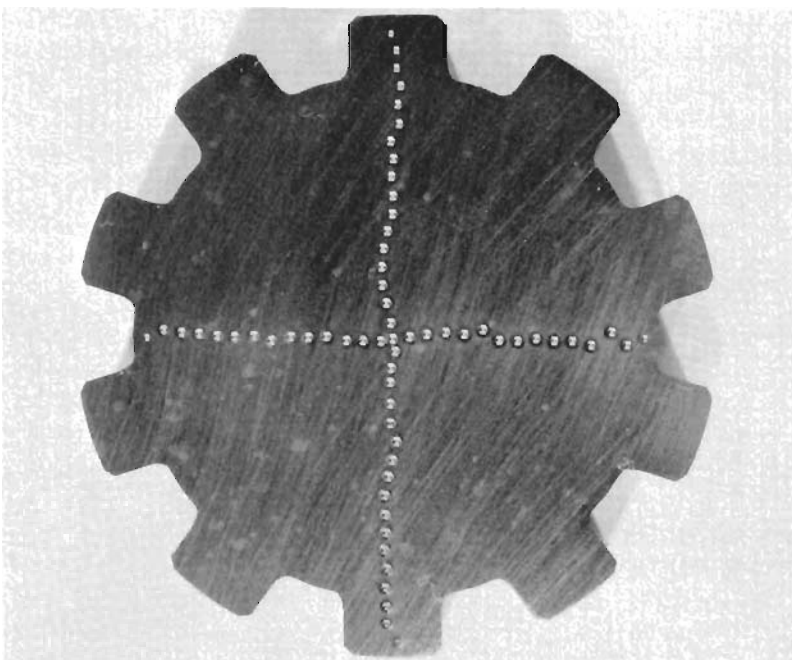


Fig. 3-8. Cross-sectional hardness survey through the splined section of a spiral bevel pinion shank. First readings at the surface (O.D. of the sample). Continuous readings at 1/16-in. intervals starting 1/16 in. from the surface.

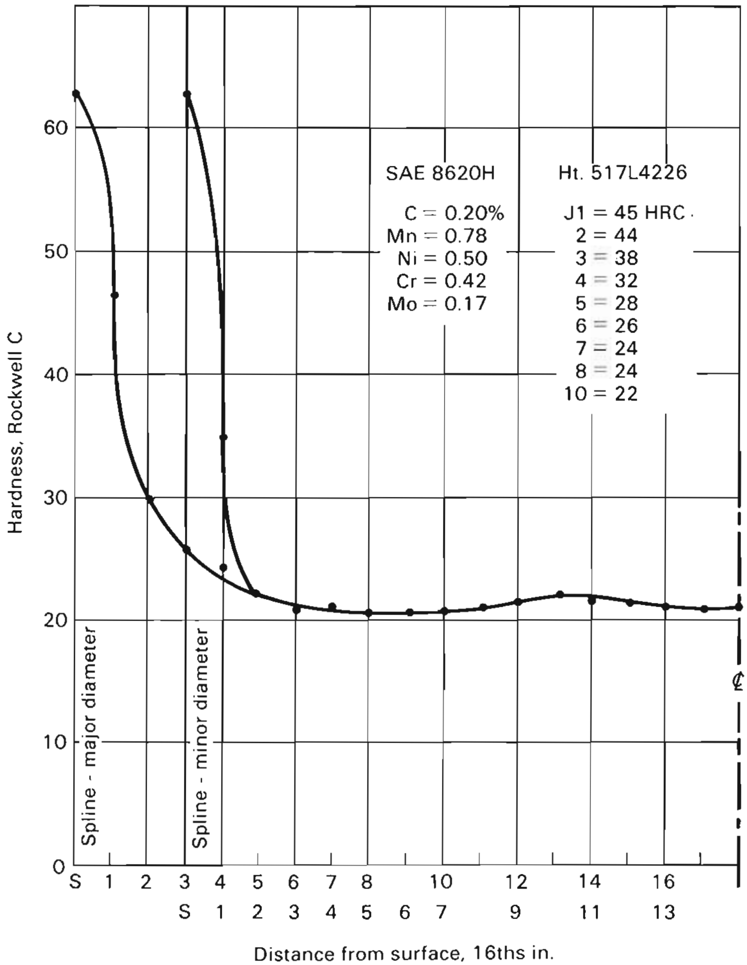


Fig. 3-9. Cross-sectional hardness survey chart of splined section shown in Fig. 3-8.

position and incorporate them onto a graph similar to the graph in Fig. 3-9.

The information on the graph can now be used to interpret the strength of the part by substituting strength for hardness, and to interpret the results of the heat treating process by utilizing quench rates for the hardenability data. The part as represented by this procedure had failed by torsional shear, so obviously, the torsional stress applied was greater than the strength of the part. If the applied stress had been abnormally high and could have been controlled, nothing more needed to be done. If the stress is to be applied regularly, the material and/or heat treating process must be changed to compensate.

Macroscopic examination is the visual inspection of any sectioned area by use of low magnification, aided by an etching process that intensifies differences within a material.

The "macroetch" procedure is accomplished by placing the desired surface in a solution of 50% hydrochloric acid and 50% water, boiling for 20 minutes, rinsing well in hot water, and drying immediately. Samples prepared specifically to study material flow lines are shown in Fig. 3-10 and 3-11. The macroetching of two sections 180° apart (Fig. 3-12) illustrates the differences in material flow that can occur, even in one forging. Nonmetallic inclusions are vividly exhibited in Fig. 3-13 and 3-14. Forging defects (laps) that sometimes show up very lightly on the surface will open up vividly on surface and subsurface when sectioned and macroetched (Fig. 3-15). Soundness of material will range from clean, to slight porosity and pinholes, to a center pipe condition as exhibited in Fig. 3-16.

A 5% nital etching solution (5 ml nitric acid to 95 ml denatured alcohol) is also used effectively to show induction hardened areas (Fig. 3-17) and differences in basic material (Fig. 3-18).

A carbon gradient traverse on a failed part is taken only when the carbon content in the carburized case is questioned. There are also severe restrictions in this procedure. There must be a smooth, round surface of sufficient length to allow for the turning of the outside diameter, in order to obtain enough steel chips from each cut to get an analysis. When the proper section has been taken from the part, it must then be tempered to a machinable hardness in an atmosphere that will not deplete carbon from the surface. A submersion in molten lead at 1200 °F is recommended. Cool and clean the surface, and prepare the ends for accurate centering in the lathe. An example of a carbon gradient traverse is shown in Fig. 3-19. To obtain the points on the graph (as shown in the example), take the first cut across the surface at a depth of 0.003 in. that measures 0.006 in. from

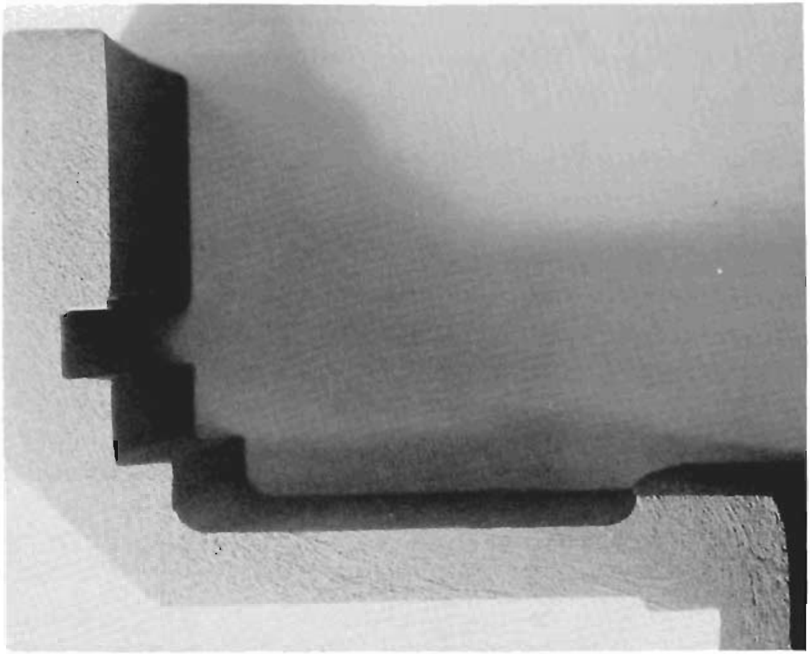


Fig. 3-10. Internal gear blank, 1/4×. Macroetched to show material flow lines after forging and machining.

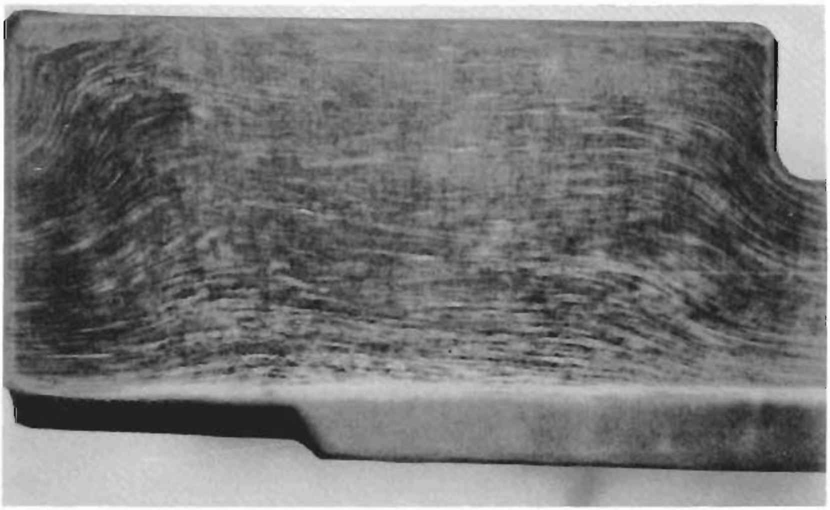


Fig. 3-11. Longitudinal section through the center of a spur gear tooth, 1/2×. Macroetched to show material flow lines after forging and machining.

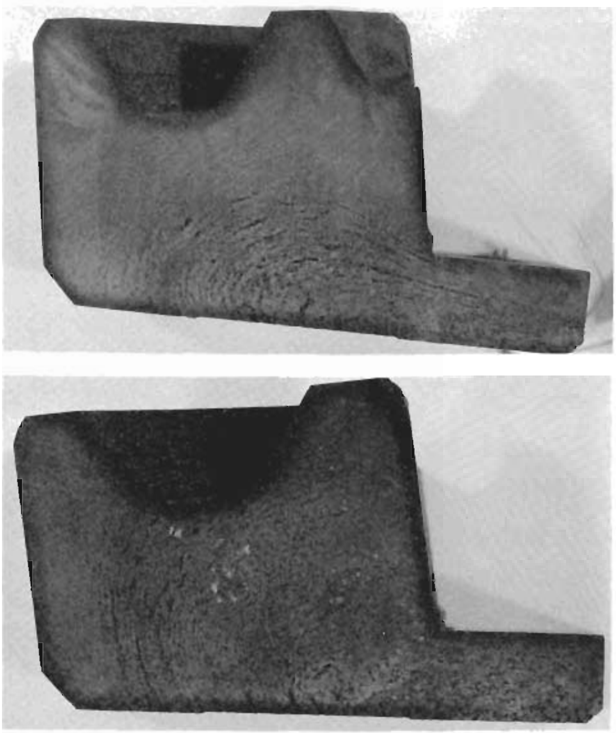


Fig. 3-12. Spiral bevel gear, 2/3×. Macroetching of two surfaces 180° apart from this gear reveals a major difference in material flow lines during the forging operation. The dark voids had contained several inclusions.

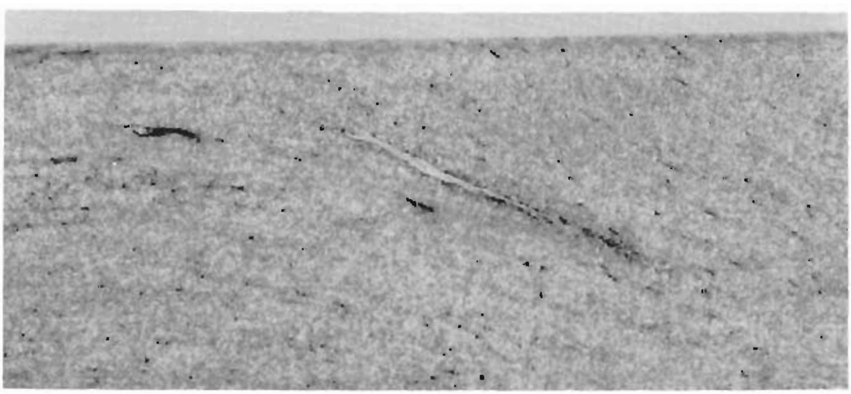


Fig. 3-13. Macroetched section near a bearing diameter showing nonmetallic inclusions following the flow lines near the surface. 10×.

Fig. 3-14. Spiral bevel gear tooth, 3/4×. Macroetched to show a large inclusion, which is aluminum silicate from the wall of the steel-producing furnace. This inclusion was hard enough to break the gear tooth cutter edge, which then scored the rest of the tooth profile.

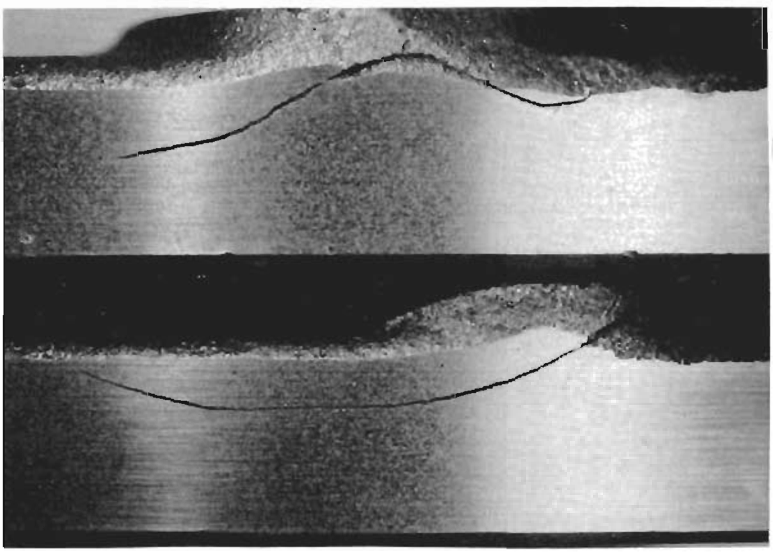


Fig. 3-15. Steel-forged gear carriers, 3/4×. Macroetched on the machined surfaces to show the extent of forging laps.

(a)
(b)
(c)

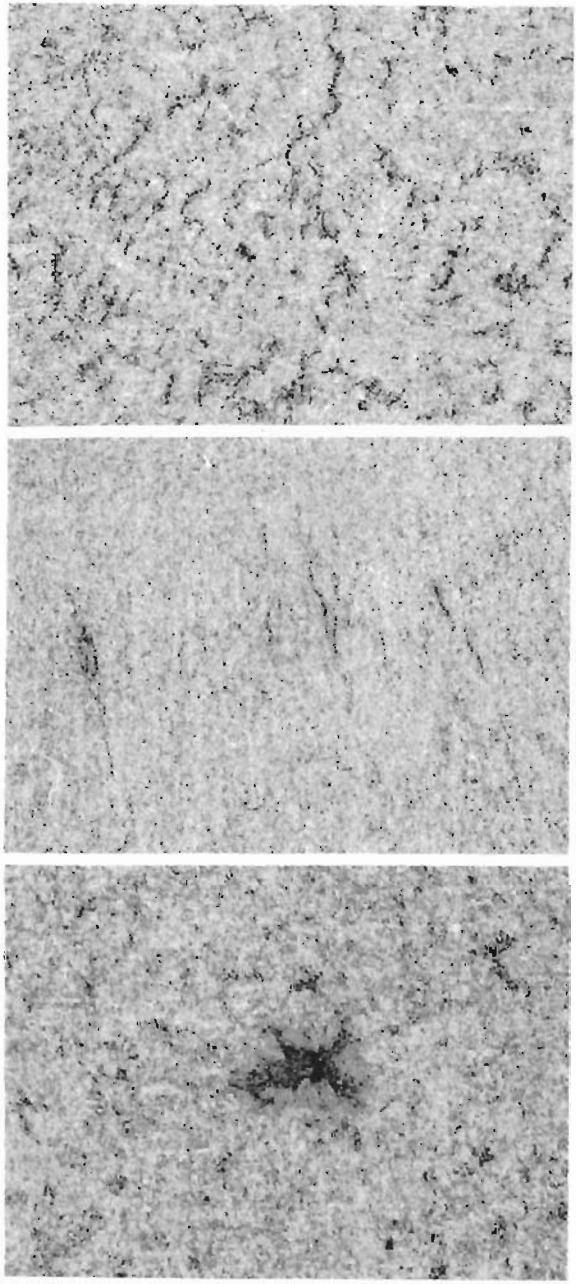
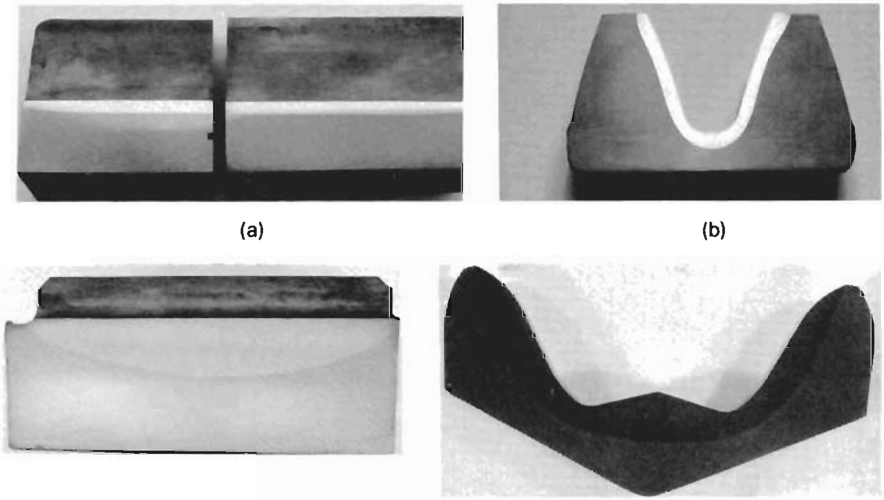


Fig. 3-16. Macrostructure, 100×. Macroetched to show center soundness of a steel part: (a) fairly sound, small pinholes; (b) slight porosity; (c) open pipe.



(a) (b) (c) (d)

(a) 1/2×. Longitudinal section through tooth root. Hardened by traversing one tooth space at a time. (b) 1/2×. Cross section of tooth space near center of same tooth as in (a). (c) 2×. Longitudinal pattern includes entire tooth and well below the roots. Entire gear hardened at once by a circular inductor. (d) 1/2×. Sprocket tooth space hardened by a stationary tooth space inductor.

Fig. 3-17. Nital (5%) etching to show induction hardened patterns.

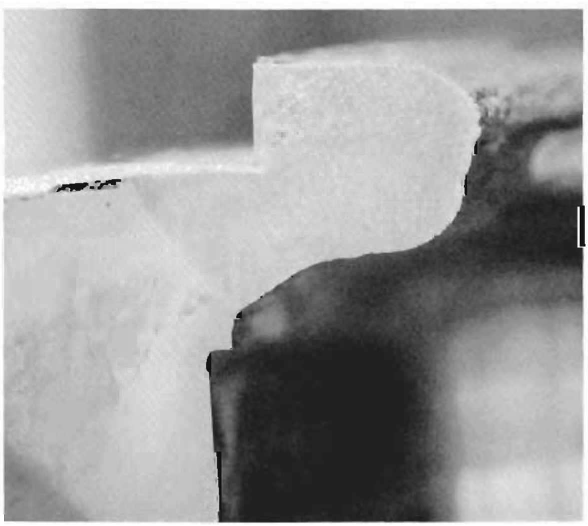


Fig. 3-18. Spindle shank, 1×. 5% nital etching reveals a welded structure.

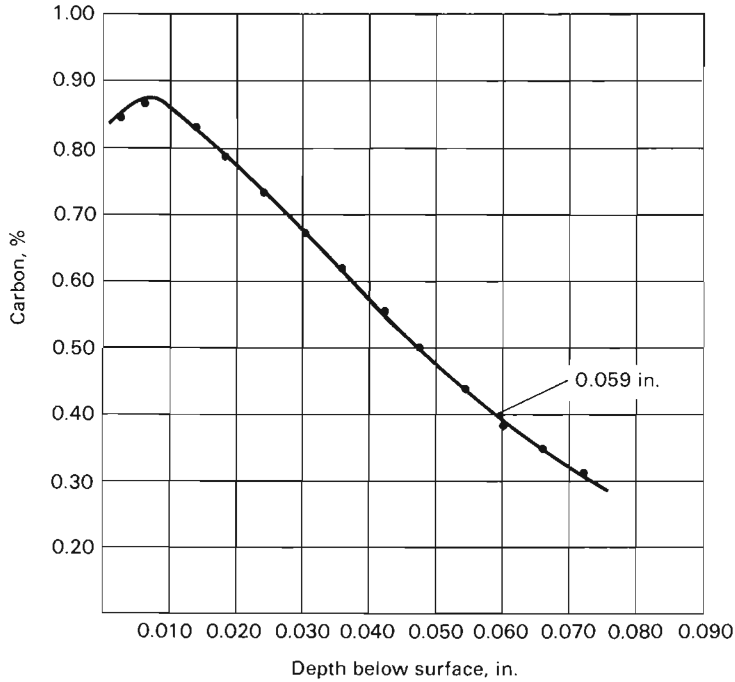


Fig. 3-19. Carbon gradient traverse taken from a shank section of a hypoid pinion. Note that the 0.40% carbon level at 0.059 in. is very close to the effective case depth.

the diameter. Take each succeeding cut at a depth of 0.006 in. or a measurement of 0.012 in. from the diameter. Analyze the turnings from each cut for percentage of carbon, and record the results on the graph. Document the necessary information. Note that the depth at 0.40% carbon is identified. This carbon level in a carburized case usually indicates closely the effective case depth, which is defined below under "Case Hardness Traverse."

Chemical analysis of the failed gear is very often necessary to establish or confirm the grade of steel. In most instances, the part must be sectioned in an area where a drilling for sample chips can be taken safely away from the high surface carbon. Since carbon is very important to the interpretation of the physical properties of the gear, it must, above all the alloying elements, be accurately measured.

Case hardness traverse by the use of microhardness testing is a necessary procedure in examining any failed carburized part. Be very careful in selecting the area from which this section is taken. The sample for this procedure will, in all probability, be used also for microscopic examination. In fact, the polishing technique is the same for both procedures. Select the area that will provide the most meaningful information. It may be through the fractured area, or near the fractured area, or through another tooth in the same relative position.

If a carburized gear tooth is to be sectioned, it must be cut at an angle that is normal to the tooth face at that point. Cut slowly with much cooling. Trim the sample to fit a mounting press, but do not mar or distort the area to be examined. Mount the sample according to the mounting press instructions, using a thermosetting or a thermoplastic compound as a holding vehicle. Prepare the surface of the sample by grinding and polishing in graduated steps until the final polish is given by using a compound of five microns or less. Place the polished sample in the microhardness tester and take a series of hardness readings

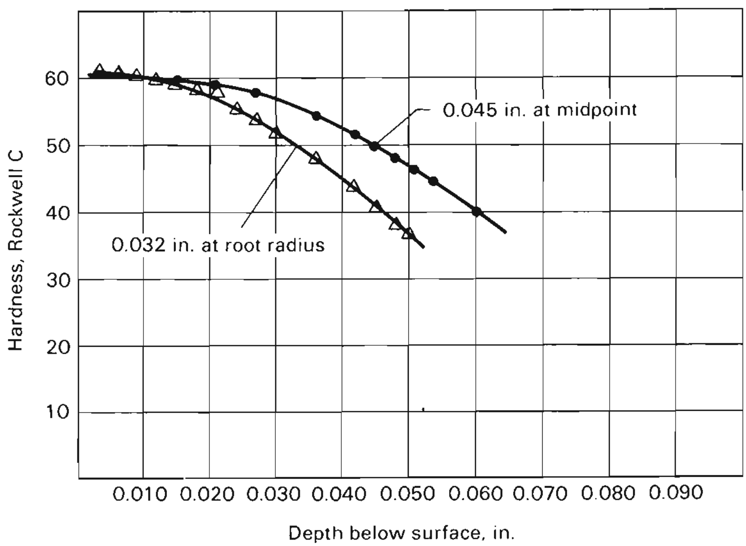


Fig. 3-20. Case hardness traverse of an 8620H steel spur gear tooth using a Tukon microhardness tester with an indenter load of 500 g. Effective case depths are indicated by arrows.

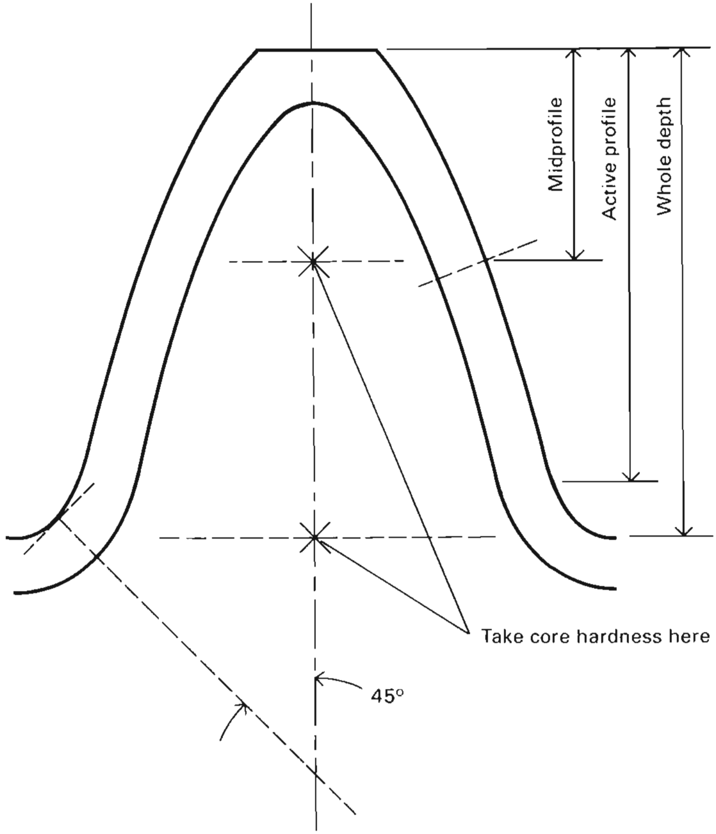


Fig. 3-21. Defined positions on a gear tooth for taking a hardness traverse at midprofile and root radius. Accepted core hardness is taken at root center and mid-height.

through the case structure. Record the results on a chart similar to that in Fig. 3-20 and document the information. The accepted positions for taking a case hardness traverse are at the outside diameter of a shaft or ground bearing of a pinion; at the center of a space (root) between two splines; or, for a gear tooth, at both the midprofile and the root radius (Fig. 3-21).

The root radius traverse generally starts at that point normal to the surface on a direct line that will intersect the radial line through the center of the tooth at 45°. If a failure occurs at a radius between two different diameters, by all means take a microhardness survey at 45° through that radius. In many instances, the case at a radius may be undercut by grinding, and a traverse on the outside diameter will not indicate the proper answer.

Case depth is generally understood to mean "effective case depth." Effective case depth is the distance from the surface where the case hardness traverse curve intersects the hardness level of 50 HRC. Although the effective case depth of the root radius is less than that of the midprofile, it is an accepted practice for it not to be less than 60% of the specified minimum effective case depth. Sufficient tests have been conducted to indicate that 50% is acceptable.

Total case depth is the perpendicular distance from the surface to the point where the hardness of the core has been reached, or where the carbon content is the same as the core.

The case hardness traverse for an induction hardened case is accomplished by the same procedure as above, but the shape of the curves is different (Fig. 3-22). Note the consistent straight-line hardness until the start of the transition zone, and the quick drop of hardness through the transition zone. This

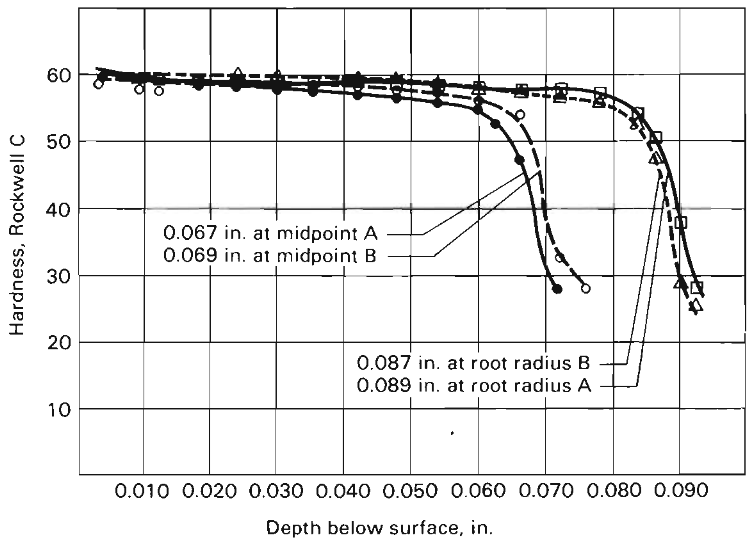


Fig. 3-22. Case hardness traverse of an induction hardened 4145H steel gear tooth using a Tukon microhardness tester with an indenter load of 500 g. Effective case depth is measured to 45 HRC as indicated by arrows.

sectioned gear tooth was very similar to the one shown in Fig. 3-17(b). The difference in case depth between profiles A and B is due to the difficulty of accurately positioning the tooth space inductor in the exact center of the space, especially since there is only a 0.007- to 0.012-in. air gap between the tooth surface and the inductor surface. Notice also that the effective case depth is measured to 45 HRC. SAE Recommended Practice J423a states: "In instances involving flame and induction hardened cases, it is desirable to use a lower hardness criterion."[4] In J423a suggested hardness levels are tabulated as follows for various nominal levels of carbon content:

Carbon content, %	Effective case depth hardness, HRC
0.28-0.32	35
0.33-0.42	40
0.43-0.52	45
0.53 and over	50

Microscopic examination is the inspection of a sectioned area using the higher magnification of 100× through 1000×, aided by an etching process that enhances the different microstructures. In most instances, the sample prepared for microscopy will be the same sample used for case hardness traverse. In fact, it is desirable to use this sample, because the hardnesses taken across the sample can now be associated with a specific microstructure (Fig. 3-23). The sample illustrated in Fig. 3-23 was taken from the midprofile of a gear tooth that had not yet been in service; therefore, the surface is unaltered. There has been no "cold working" of the first few thousandths of an inch, which would tend to transform the retained austenite into nontempered martensite.

The surface and near-surface microstructure of a gear tooth will very often tell the story of the mode of failure. In fact, several modes of surface pitting may appear to be identical when visually inspected, but they can be correctly diagnosed only by studying the microstructure. These are detailed in Chapter 4. Also, the microstructure is a factor in any mode of pitting and should be recognized as such.

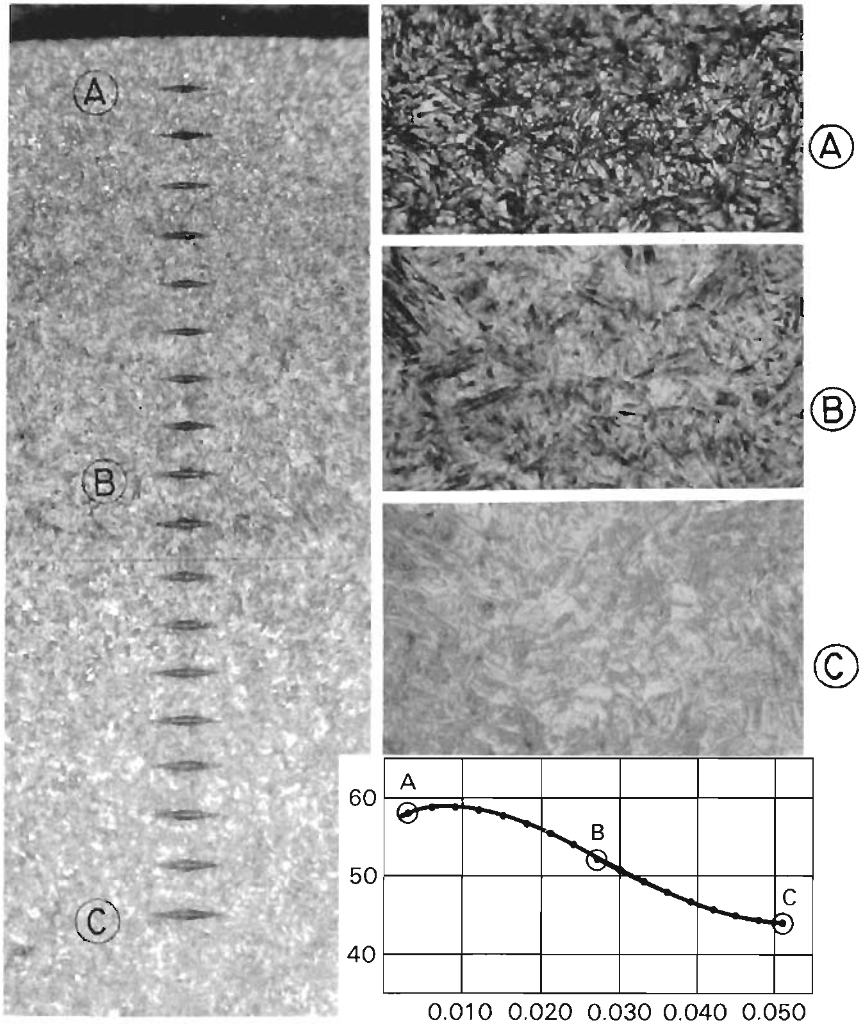


Fig. 3-23. Case hardness traverse with Tukon 500 g at 0.003-in. interval (100×) charted to show Rockwell "C" hardness at each distance from surface. Material is SAE 4320H, carburized at 1700 °F, diffused at 1550 °F, direct quenched in oil at 300 °F, and tempered at 350 °F. Microstructure, 500×. 3% nital etch. (A) Tempered acicular martensite retaining approximately 15% austenite, 58 HRC; (B) Medium-carbon (0.40% C) martensite, 52 HRC; (C) Low-carbon (0.21% C) martensite, 44 HRC.

The surface and near-surface microstructures are generally associated with heat treatment and may contain one or more of the following: martensite, austenite, carbides, decarburization, and intergranular oxidation. Martensite may be either acicular (Fig. 3-23a) or massive (Fig. 3-24). Austenite is usually not objectionable in modest amounts (Fig. 3-23) and may be welcome for many applications. However, the amount in Fig. 3-25 (45%) may be somewhat immodest. Do not be alarmed by a dispersal

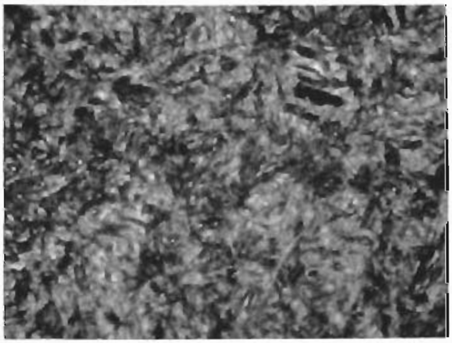


Fig. 3-24. Microstructure, 500×. 3% nital etch. Dense tempered martensite (0.75% carbon). Carburized SAE 4320H. Retained austenite, nil.

Fig. 3-25. Microstructure, 500×. 3% nital etch. Tempered acicular martensite in matrix of austenite (45%). Carburized SAE 4620H at 0.95% C.

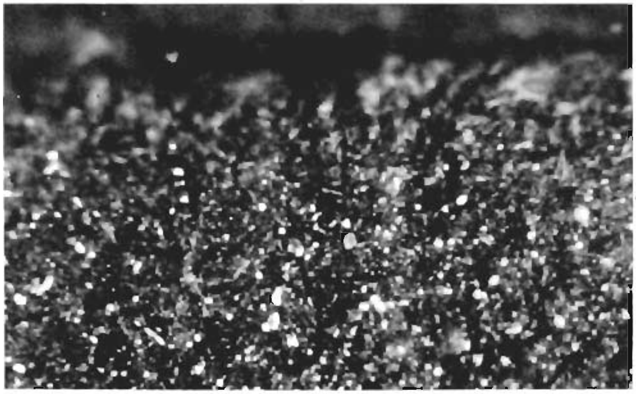


Fig. 3-26. Microstructure, 500×. 3% nital etch. Fine spheroidal carbides scattered throughout a tempered martensitic case.

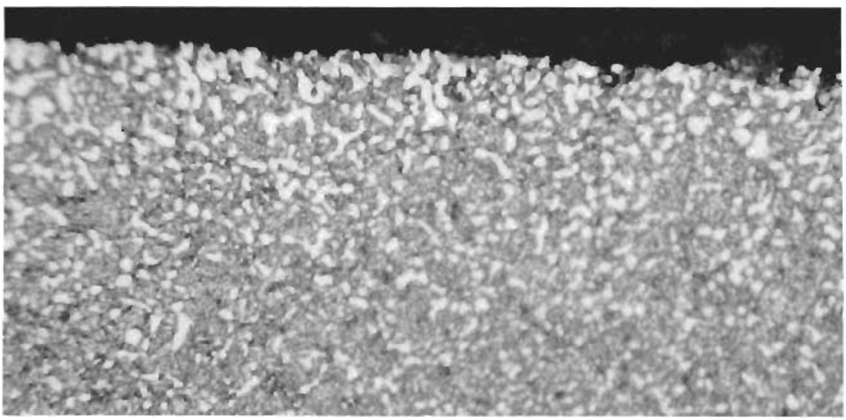


Fig. 3-27. Microstructure, 200×. 3% nital etch. Massive globular carbides in matrix of tempered martensite.

of fine spheroidal carbides throughout the case (Fig. 3-26); they are welcome. Massive carbides (Fig. 3-27) may be acceptable in some instances, but if a network appears (Fig. 3-28), it is most unwelcome.

Decarburization may be either gross or slight. Gross decarburization is an almost complete change to ferrite at the surface (Fig. 3-29) that is very soft, with no strength or resistance to

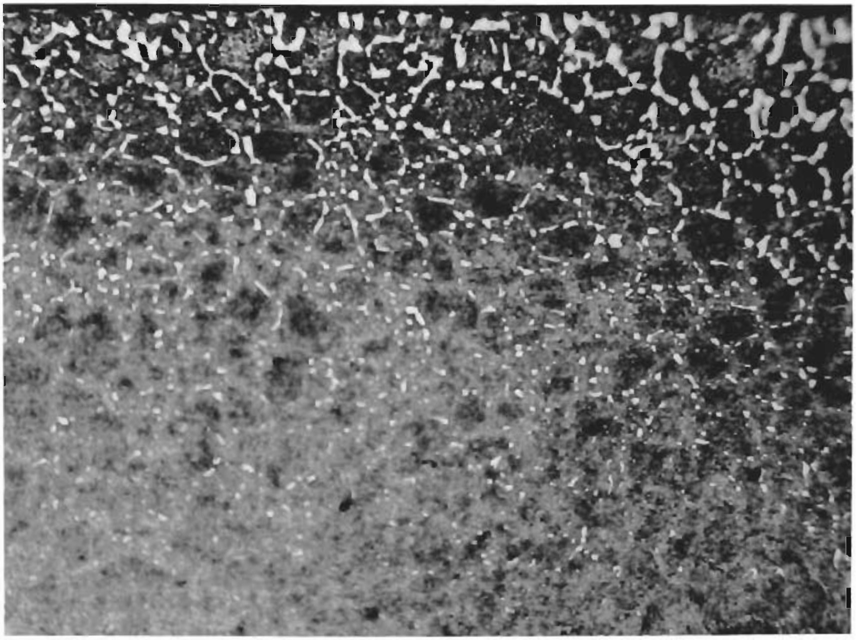


Fig. 3-28. Microstructure, 200×. 3% nital etch. Massive carbides in network around grain boundaries. Matrix of tempered martensite and bainite.

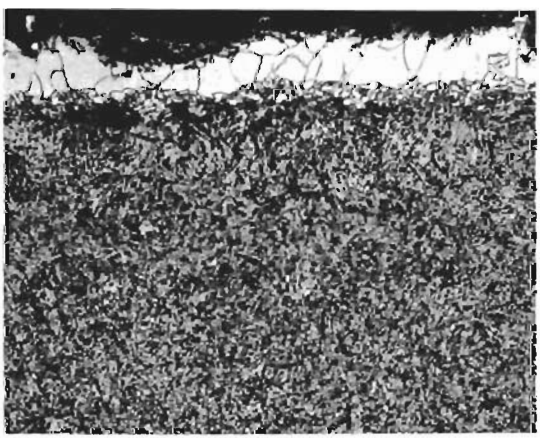


Fig. 3-29. Microstructure, 250×. 4% nital etch. Gross decarburization of carburized SAE 4118 steel above matrix of tempered martensite.

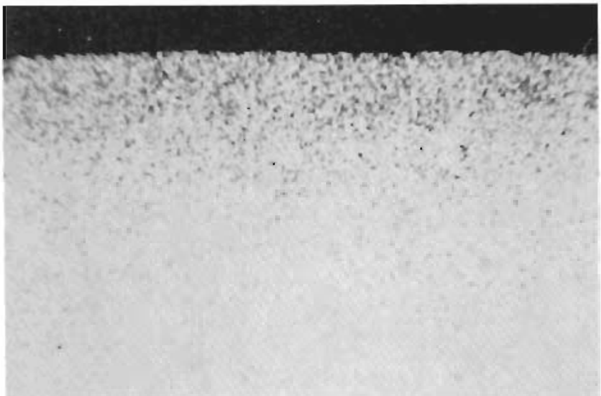


Fig. 3-30. Microstructure, 100×. Picral etch. Scattered bainite (dark spots) along surface of a tempered martensitic case.

Fig. 3-31. Microstructure, 100×. 3% nital etch. Bainite (dark structure) extending from the case/core transition through the tempered martensitic case to the surface.

adhesive wear. Slight decarburization is more subtle, replacing surface martensite with a bainitic structure (Fig. 3-30) that may be only slightly lower in hardness and strength. Depending on the amount, this may or may not be detrimental. Bainite, perhaps due to certain quenching problems, may also be found to extend from the case/core transition zone to the surface (Fig. 3-31). When this occurs at the root radius, the tooth bending strength is somewhat reduced. Intergranular oxidation (Fig. 3-32) at or near the surface is always present and usually ignored. However, this can be practically eliminated today by mechanical or chemical means, if necessary. If the mechanical means is grinding, caution should be taken to avoid grinding burns and checks. Electrochemical etching to remove intergranular oxidation may change the tooth pattern characteristics and should be used with caution.

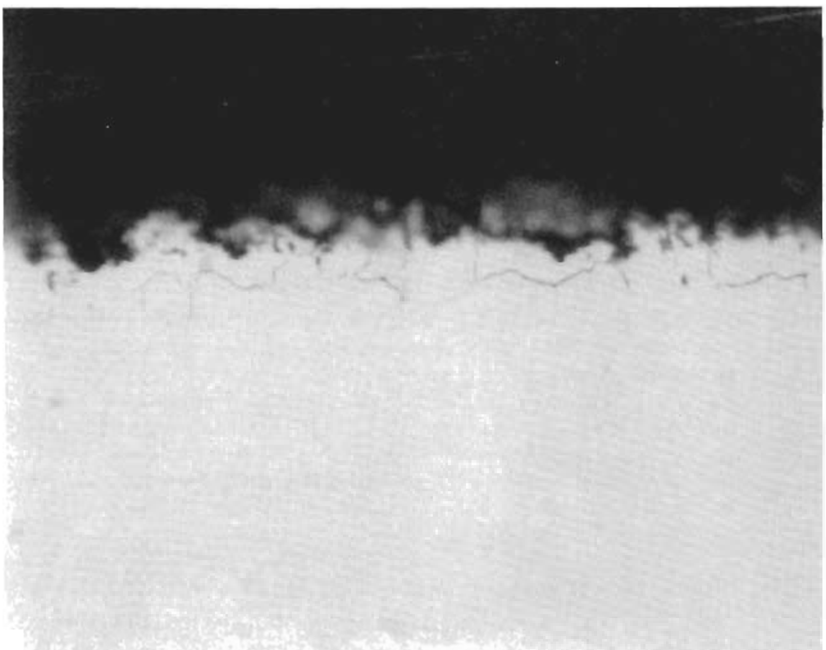


Fig. 3-32. Microstructure, 500×. As polished. Black grain oxide network extending into the case from the surface to approximately 0.0004 in.

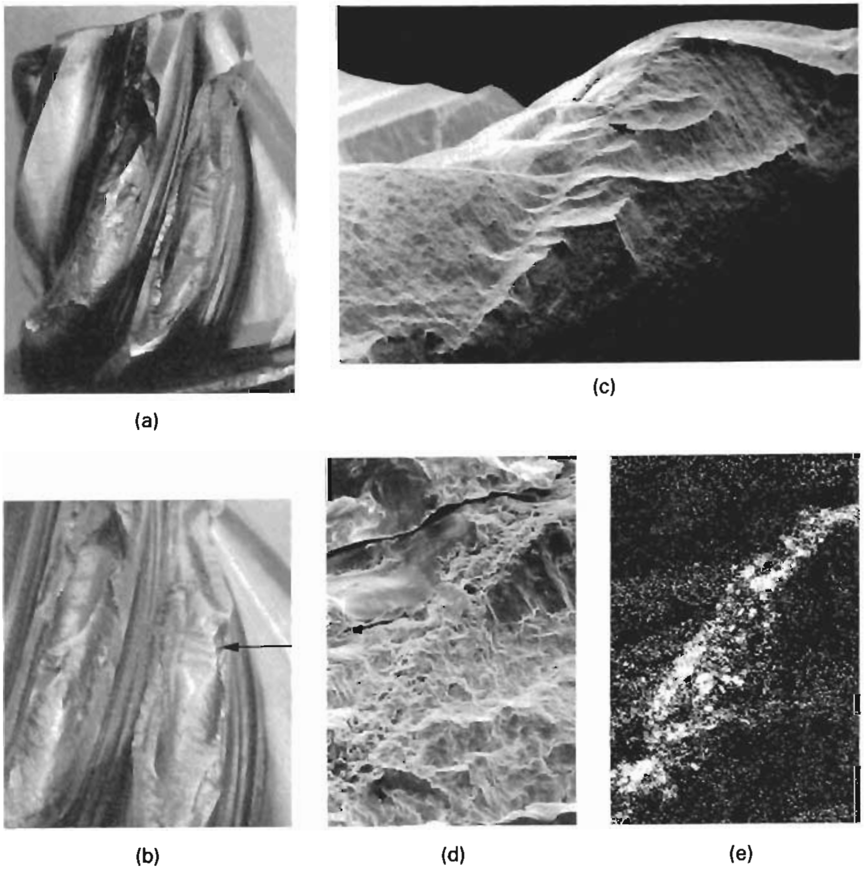


Fig. 3-33. (a) Spiral bevel pinion showing tooth spalling of two adjacent teeth; 1.7×. (b) Spalling fatigue originating subsurface, nucleated by a nonmetallic inclusion (arrow); 2.5×. (c) Scanning electron micrograph of the fatigue origin at the inclusion (arrow); 27×. (d) SEM closeup of the central area of the inclusion; 237×. (e) SEM qualitative analytical picture of the amount of aluminum within the inclusion and the surrounding area.

Scanning electron microscopy (SEM) is a recent, highly sophisticated tool in the study of fracture analysis; it is an examining method with a very great range of magnification used on the fracture surface per se. The sample must not be altered or abused prior to examination. In general, SEM is not a procedure that is required for every fractured part, but it is highly valuable when positive identification of a fracture mode is required or when confirmation is needed for documentation. Its extreme depth of field coverage, which allows for beautiful three-dimensional photographs, is an important contribution. It also has the capability of making a selective qualitative chemical analysis of a specific structure. Occasionally, an opportunity will arise when a unique structure within a fracture pattern must be identified. Figure 3-33 is an example. It should be remembered that the scanning electron microscope is a tool to be used to complement the other procedures and is not the tool for a complete analysis.

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

A gear has failed when it can no longer do efficiently the job for which it was designed.

Although this definition generally has been accepted, there are instances when a customer might claim a gear failure when the supplier believes that the gear has been merely "worn-in" properly and should not have been removed from service. Also, a gear may have been placed in service under circumstances completely different from those for which it was designed. Although it is very difficult to separate mode from cause in many instances, the distinction should be made.

A mode of gear failure is a particular type of failure that has its own descriptive identification. One mode may or may not be unique to a specific failure, since the origin may be of one mode, the progression of a second mode, and the final fracture of a third mode. Generally, the dominant mode in each instance will be discussed.

Certain failure modes occur more frequently than others. Several failure analysts in various fields of expertise have compiled their findings regarding the frequency of failure modes, and they have agreed that the most frequent is fatigue, followed by impact (tensile or shear), and wear (abrasive or adhesive). In an analysis of over 1500 studies, the three most frequent gear failure modes were tooth bending fatigue (32%), tooth bending impact (12½%), and abrasive tooth wear (10%).

Fatigue

Fatigue failure[1]—cracking under repeated stresses much lower than the ultimate tensile strength—ordinarily depends on the number of repetitions of a given stress range rather than the total time under load, and it does not occur below the stress amplitude called the "fatigue limit." The likelihood of fatigue failure is increased greatly by the presence of notches, grooves, surface discontinuities, and subsurface imperfections, all of which will decrease the stress amplitude that can be withstood for a fixed number of stress cycles. The likelihood of fatigue failure is also increased significantly by increasing the average tensile stress of the loading cycle.

There are three stages within a fatigue failure to be studied closely: the origin of the fracture, the progression under successive cycles of loading, and final rupture of the part when the spreading crack has sufficiently weakened the section. Most attention is devoted to the first, to determine why the crack started at a particular point. The second stage is observed to determine the direction of the progression. A fatigue crack will follow the path of least resistance through the metal. The final area of fracture may result from shear or tension, but, in either case, examining this area may help determine the apparent magnitude of stress that had been applied to the part.

Tooth Bending Fatigue

Tooth bending fatigue is the most common mode of fatigue failure in gearing. There are many variations of this failure mode, but the way to understand better the variations is first to understand the classic failure.

The classic tooth bending fatigue failure is one that occurs and progresses in the area designed to fail. Figure 4-1 shows an example. Five conditions indicate why this is the classic tooth bending fatigue failure:

(*a*) The origin is at the surface of the root radius of the loaded (concave) side of the tooth.
(*b*) The origin is at the midpoint between the ends of the tooth where the normal load is expected to be.

Fig. 4-1. Spiral bevel pinion showing classic tooth bending fatigue with origin at midlength of the root radius on the concave (loaded) side. 0.6×.

(c) One tooth failed first; the fracture progressed slowly toward the zero-stress point at the root, which shifted during the progression to a point under the opposite root radius and then proceeded outward to that radius.
(d) As the fracture progressed, the tooth deflected at each cycle until the load was picked up simultaneously by the top corner of the next tooth; because of the overload, a tooth bending fatigue failure started in the same area. (Note that the fracture of the second tooth appears to have an origin more recent than the first fracture.)
(e) The material and metallurgical characteristics are within specifications, and the pinion should have withstood normal operating conditions.

If any of the five conditions is changed, there will be a deviation from the classic example of tooth bending fatigue.

Following is a guide to the rudiments of classic tooth bending fatigue with a study of photoelastic models. Figure 4-2

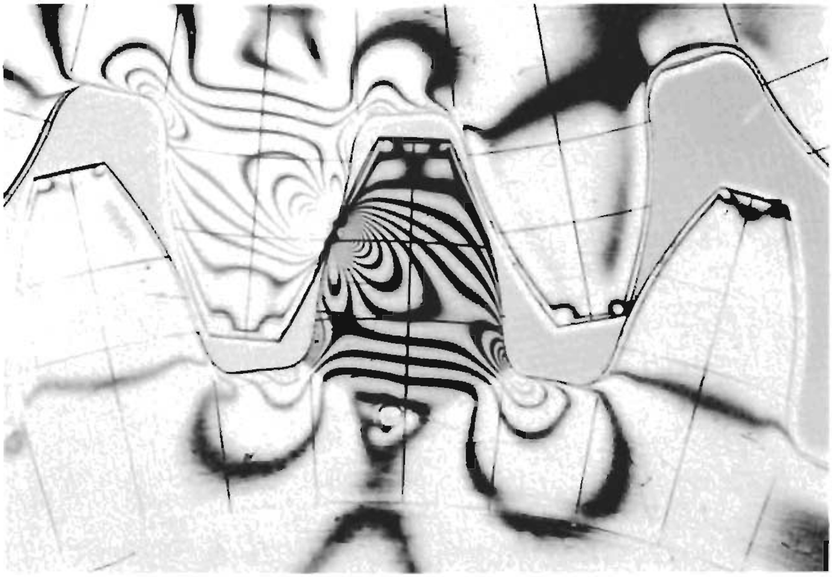


Fig. 4-2. Photoelastic study of two mating pinion teeth receiving full load. Note the high concentration of compressive stress at the point of contact, the tensile stress at the root radius, and the zero-stress point at the tooth centerline below the root circle.

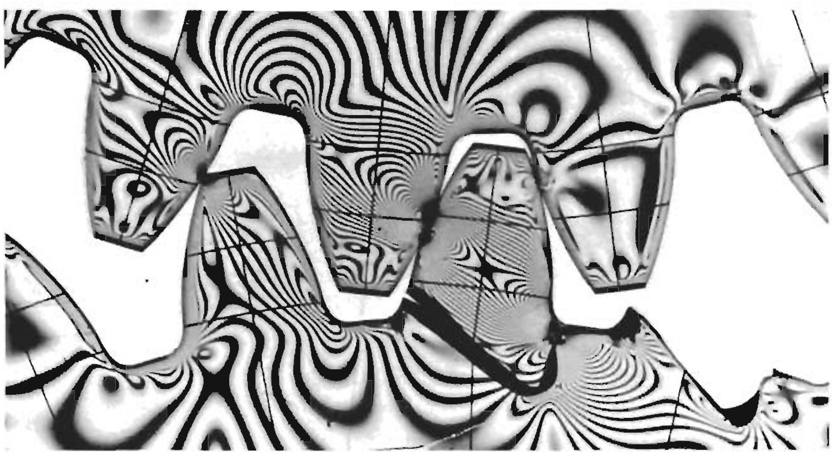


Fig. 4-3. Photoelastic study of mating teeth indicates the shift of the zero-stress point during crack propagation until final fracture reaches the opposite root radius. At the same time, deflection of the tooth allowed the adjacent tooth to pick up the load.

shows two mating teeth in contact at or near the pitchline. Note the high concentration of compressive stress at the pitchline, or point of contact; the concentration of tensile stress at the root radius of the loaded side; the concentration of compressive stress at the root radius of the opposite or nonloaded side; and the zero-stress point below the root circle at or near the tooth centerline. The progression of any fatigue crack initiated at the root radius will be in the direction of this zero-stress point, since this is the path of least resistance.

Once the crack at the root radius is initiated and is moving toward the zero-stress point (see Fig. 4-3), the point moves away from the leading edge of the crack and shifts laterally, until it reaches a position under the opposite root. At that time, the shortest remaining distance is outward to the root, where it terminates. In the meantime, as the failing tooth is deflected, the top of the adjacent tooth picks up the load. The load on the first tooth is relieved, allowing a slower progression due to lower cyclic stress. It is only a matter of time until the next tooth initiates a crack in the same position.

Variations of the classic tooth bending fatigue occur for many reasons. A few are described in the paragraphs that follow.

Tooth bending fatigue of a spur gear tooth with the origin along the surface of the root radius of the loaded side, but at a point one-third the distance from the open end (Fig. 4-4), can be considered a classic failure. The maximum crown of the teeth had been placed intentionally at the area where the failure occurred, and thus, the maximum applied loads were in the same area. This, then, is the area that was designed for the point of initiation of normal tooth bending fatigue. Note that the progression was directly to the bore—the shortest direction and the path of least resistance.

Tooth bending fatigue of a spur gear tooth, meeting all previously listed conditions for a classic failure except (*b*), is shown in Fig. 4-5. The origin was at one end of the tooth and not at the midlength. There might be several reasons for this point of initiation; but in this instance, a severe shock load had twisted the parts until a momentary overload was applied at the end of the tooth. This caused the formation of a crack that

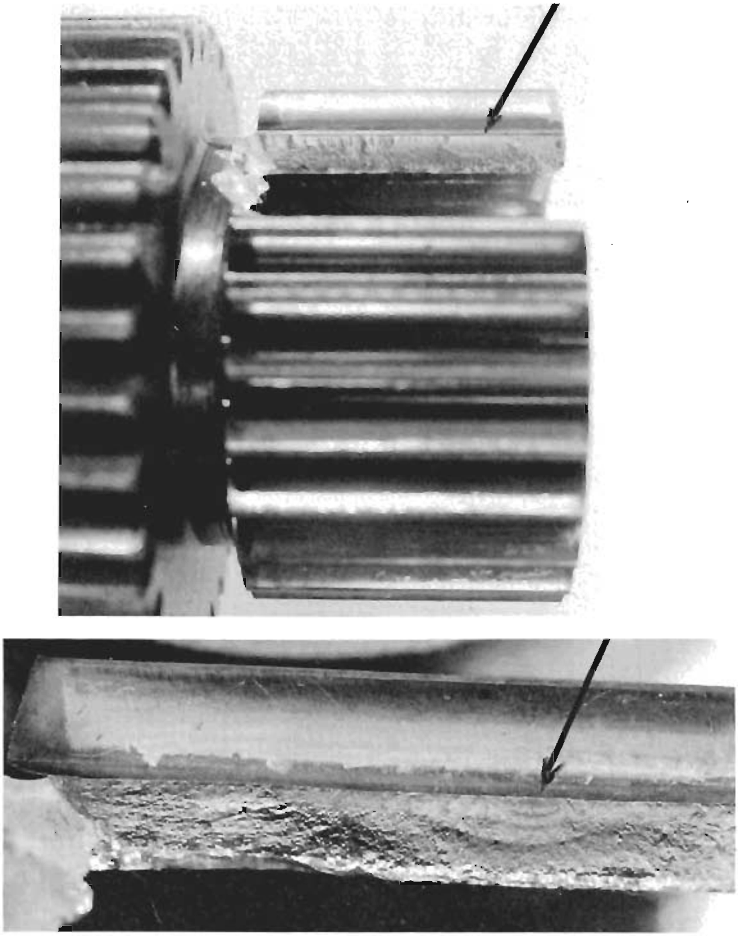


Fig. 4-4. Spur tooth pinion at 1× (top) and 2.5× (bottom). Tooth bending fatigue originating at the root radius (arrows), loaded side, one-third the distance from the open end. Progression was to the bore.

became a high stress-concentration point, from which fatigue progression continued.

Another example of an almost classic failure, though lacking conditions (*b*) (off-center) and (*c*) (continuing under tooth to opposite root radius), is shown in Fig. 4-6. Two reasons for this specific variation altered the conditions. Condition (*b*) was changed because of an off-center web under the gear teeth. The distortion of the teeth during quenching created a self-imposed

Fig. 4-5. Spur pinion, 7/8×. Tooth bending fatigue with origin at root radius of loaded side at one end of the tooth.

Fig. 4-6. Helical gear, 1.12×. Tooth bending fatigue followed by tooth bending impact. Origin is off-center of the tooth midpoint but is directly above the center of the web.

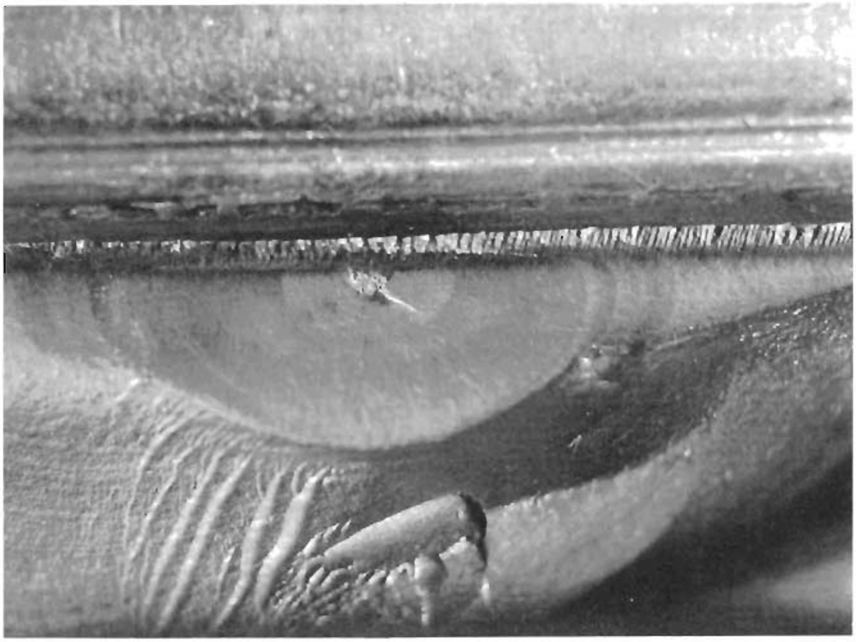


Fig. 4-7. Spur pinion, 100×. Tooth bending fatigue at midlength of tooth at root radius, but origin is at an inclusion located in the case/core transition.

crown directly above the center of the web, which became a post-designed area of initiation for normal tooth bending fatigue. And the propagation of the fatigue crack did not extend very far, until the stress applied by the normal loads exceeded the tensile strength of the remaining material. In this instance, the initial mode of failure was tooth bending fatigue, followed closely by tooth bending impact.

An example of tooth bending fatigue that did not conform to classic condition (*a*) is shown in Fig. 4-7. Although the initiation was at the root radius at the midlength it did not originate at the surface. The origin was at a nonmetallic inclusion at the case/core interface. Subsequent progression occurred through the case to the surface, as well as through the core toward the zero-stress point.

Another example of a tooth bending fatigue that did not conform to condition (*a*) is depicted in Fig. 4-8. The failure was at the root radius and at the midlength, but the origin was not at

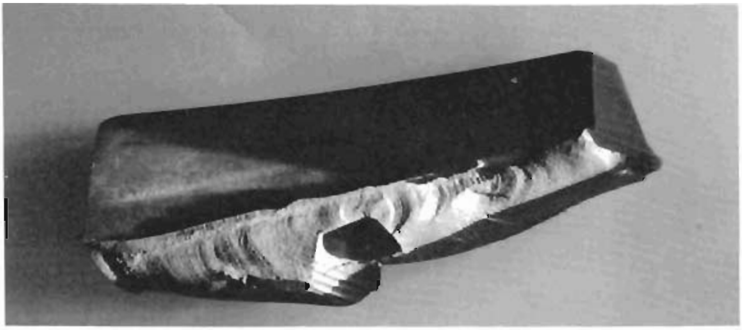


Fig. 4-8. Spiral bevel gear tooth, 0.7×. Tooth bending fatigue with origin at the apex of the drilled bolt hole, which terminated just below the root radius.

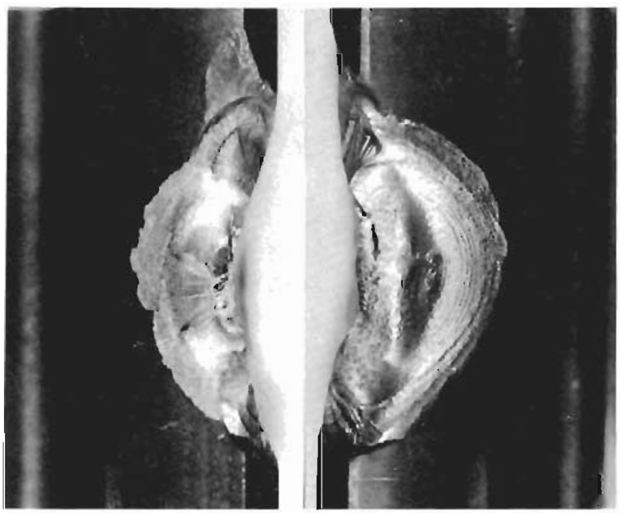


Fig. 4-9. Spur pinion tooth, 2×. Bidirectional tooth bending fatigue, with applied loads at a high spot at the top edge on both sides. Origin is at the case/core interface, near the crown of the tooth.

the surface. The origin was at the apex of a tapped bolt hole that had been drilled from the back face of the gear and had terminated less than ¼ in. from the root radius. The fatigue crack progressed to the surface at the root radius, as well as into the core toward the zero-stress point.

Figure 4-9 also shows tooth bending fatigue, but it is of a different nature. In fact, from appearance only, it might be called "spalling." However, due to the reasons of origin and

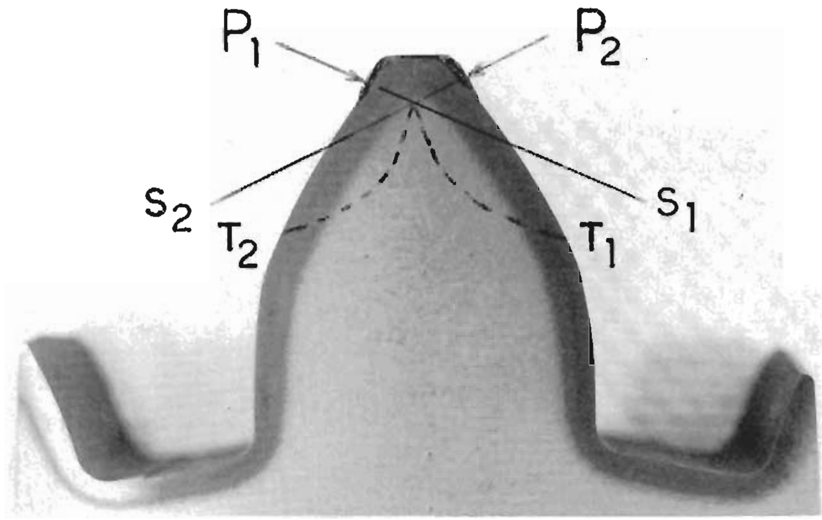


Fig. 4-10. Sketch of forces, shear and tensile stress planes, and directions present in the failure depicted in Fig. 4-9.

progression, it must be placed in the tooth bending category. Referring to the sketch in Fig. 4-10, picture a spur tooth with a high spot at the top of the active profile on both sides, which is taking the full pressure and deflection (P) as the tooth goes into and out of mesh. The maximum shear plane (S) is in line with the pressure point and passes through the case/core transition zone, which is of lower tensile strength than the case. The tensile stress (T), which is maximum along a plane 45° from the shear plane, is greater than the strength of the core material, and an internal stress rupture originates at the case/core interface where the two shear planes intersect. The fatigue progression is along the planes of maximum tensile stress, until its direction is changed by the fulcrum effect of the opposite face, causing it to follow the path of least resistance to the surface. Since this instance is bidirectional, and the two events occur simultaneously, a classic bidirectional fatigue pattern is evident.

Surface Contact Fatigue (Pitting)

Pitting, as a surface or near-surface failure, is recognized as a material-fatigue mode of failure. As elements of the structure at and near the surface are subjected to alternating compressive stresses, plastic deformation occurs in some microstructures and elastic deformation in others. The differences of plasticity between grains will encourage fatigue cracks to initiate under pulsating and alternating stresses. Also, each grain is randomly oriented to the direction of its shear plane. Under a specific compressive load, some grains will tend to break apart in tension, while others will tend to shear, and still others will be unaffected. It is these internal stress raisers that will allow a crack to form at a specific point when the resultant strain exceeds the elastic limit of that point. The cracks flow together and accumulate to form a plane of progression; then the path of least resistance is followed and a particle falls away from the surface. This is the start of pitting.

All too often, there may be present at or near the surface an embedded inclusion that will act as a nucleus for a crack progression. The point of inclusion is usually very weak and most susceptible to crack initiation (Fig. 4-11).

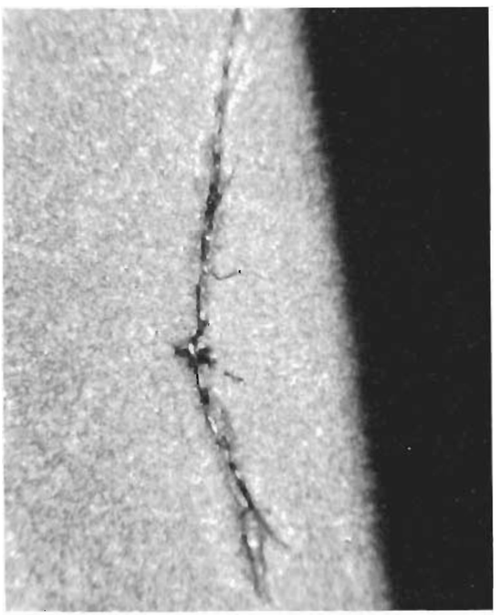


Fig. 4-11. Spur gear tooth, 100×. Nital etch. An internal crack originating at an oxide type of inclusion below the surface at the pitchline. A pit is being formed.

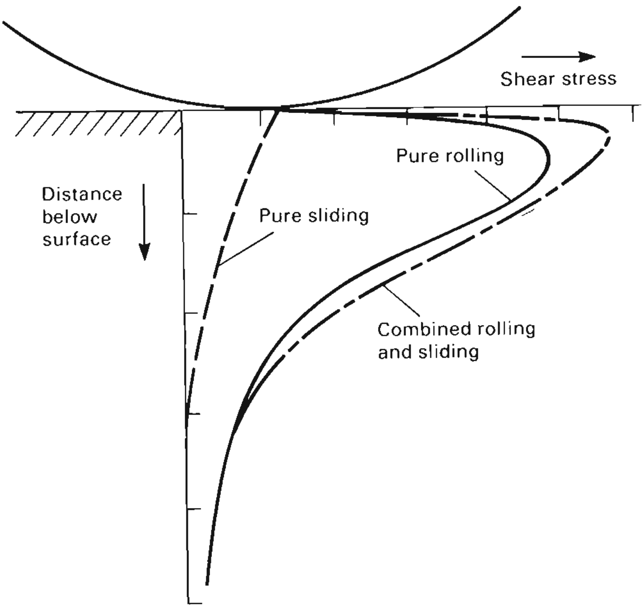


Fig. 4-12. Stress distribution in contacting surfaces due to rolling, sliding, and combined effect.

The initiation of pitting is confined mostly to three areas along the profile of a gear tooth: (*a*) the pitchline, (*b*) the area immediately above or below the pitchline, and (*c*) the lowest point of single tooth contact.

The pitchline is the only area of line contact that receives pure rolling pressure. The pits forming directly at the pitchline are usually very small and may not progress beyond the point of origin. Some analysts claim that this type of pitting "heals" itself and is not detrimental. That may be true when the pitting is formed by the breakdown of a rough surface. It should also be noted that lubrication will not deter the origin of pitting along the pitchline. Oil is noncompressible and will not "cushion" the pressures exerted in the area of pure rolling.

The area immediately above or below the pitchline is very susceptible to pitting. Not only is the rolling pressure great at this point, but sliding is now a crucial factor. The mechanics of surface/subsurface pitting can best be understood by looking at the resultant applied stresses illustrated in Fig. 4-12.[2] The extra shear stress of the sliding component, when added to that of the

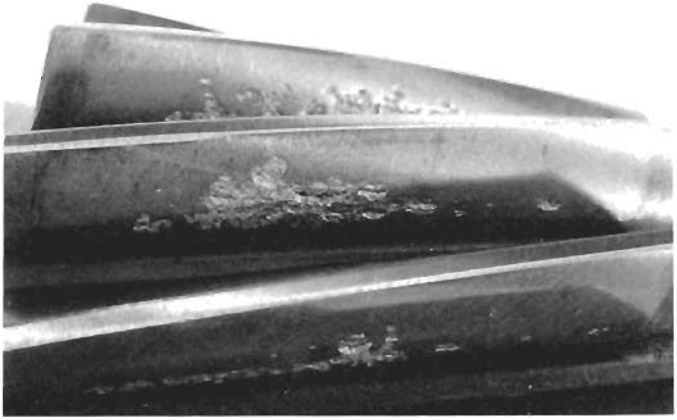


Fig. 4-13. Helical gear teeth, 2×. Pitting initiated along the pitchline and just above the pitchline. In some areas, the progression has been continuous.

Fig. 4-14. Spiral pinion tooth, 200×. Near-pitchline pitting fatigue. Origin is subsurface at plane of maximum shear.

rolling component, often results in surface fatigue at the point of maximum shear below the surface. In many instances, it is difficult to determine whether some pitting cracks actually initiate at or below the surface. Figure 4-13 shows initiation of pitting fatigue both at the pitchline of a helical gear tooth and directly above the pitchline. Progression up the addendum in some areas makes it difficult to differentiate between the two. A surface-pitted area near the pitchline is illustrated in Fig. 4-14.

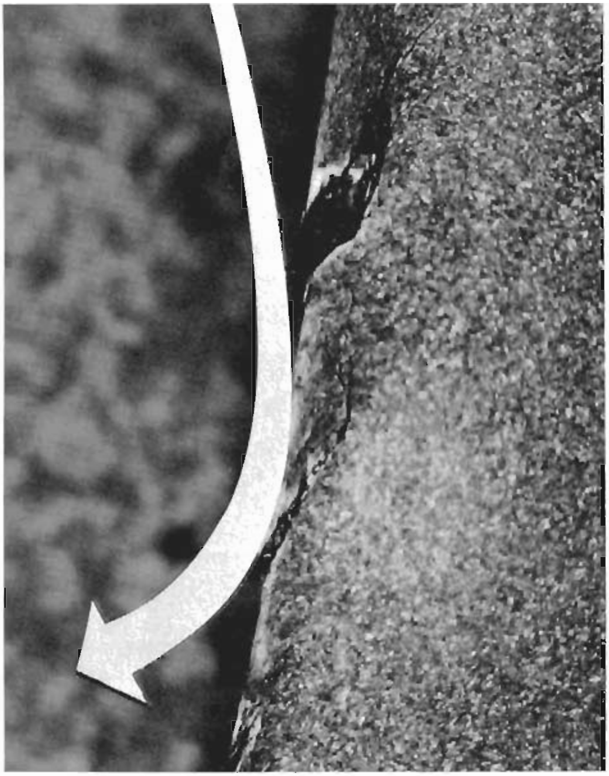


Fig. 4-15. Spiral bevel tooth, 100×. Nital etch. Pitting at lowest point of single tooth contact, illustrating contact path of the tip of the mating tooth.

Although the surface was in a rolling-sliding area, the immediate surface shows no catastrophic movement; so it is most probable that shearing fatigue took place as the initiating force below the surface. Also, the freedom from surface movement is an argument for the lubrication of the gear teeth.

The lowest point of single tooth contact is the point that receives the tip of the mating tooth as it makes first contact low on the active profile. Tip contact will produce high pressures per unit area, even if a small fraction of the actual load is transferred by the tip. Also, the maximum sliding speed, both in approach and recess, occurs at the tip contact. Two types of pits prevail at this area. One is shown in Fig. 4-15; this appears to be a swift shear and lift-out type. Perhaps it can be questioned whether this is fatigue or impact; however, if it is a shearing fatigue, it is certainly rapid. Causes are not being assessed in this chapter, but if they were, this example would certainly call for a study of tip interference and lubrication. The second type of pit is the same as that illustrated in Fig. 4-14, showing failure only by compressive loads. This is not uncommon in spur gear teeth or helical gear teeth; but when found, it is puzzling, since one does not expect to find tip interference pits other than those previously discussed. However, it must be remembered that dynamic effects are to be taken into account. Gear vibration is a fact but is seldom ever recognized as such. Studies indicate that gear vibration at high speeds, excited by static transmission error, may cause corner contact, even though the tip relief is sufficient to prevent corner contact at low speed.[3] Also, with sufficient lubrication, the pressures of tip interference will result in compressive-type pitting, such as in Fig. 4-14, rather than the adhesive type, illustrated in Fig. 4-15.

Besides the three specific and most common areas of surface fatigue pitting just discussed, there are random areas that demand recognition, such as pitting only at one end of the face, or at opposite ends of opposite faces, or along the top edge of the addendum, or at one area of one tooth only. The random incidences are without number and can be most baffling; but each one occurs because of a reason, and that reason must be found, if at all possible.

One patterned type of surface fatigue pitting is found only on spiral bevel teeth. To understand this unusual pattern of pitting, which can lead to total destruction, one must recognize the direction of the loaded patterns of a spiral bevel tooth. An excellent, detailed study reproduced in the *Source Book on Gear Design, Technology and Performance*[4] explains, in effect, that the load pattern is a line contact lying at a bias across the tooth profile, moving from one end of the contact area to the other end. Under load, this line assumes an elliptical shape as its pattern and thus distributes the stress over a larger area. Also, the purpose of spiral bevel gearing is to relieve the stress concentrations by having more than one tooth enmeshed at all times. There are instances, however, when a certain designed spiral angle will allow a one-to-one tooth ratio for a very short dura-

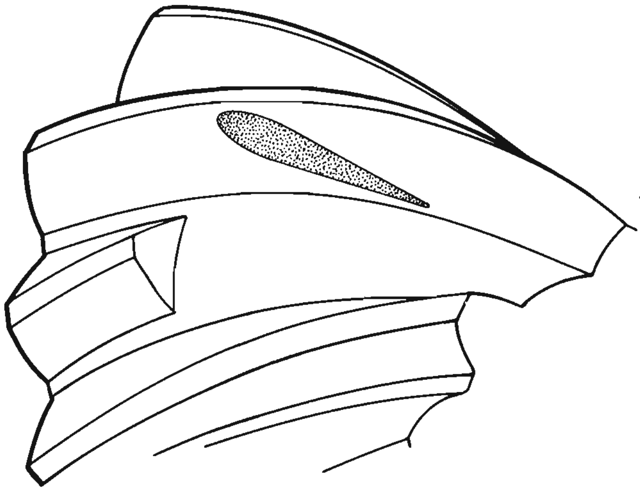


Fig. 4-16. Surface fatigue pitting initiated in a short concentrated area of a spiral bevel tooth, when that tooth was momentarily assuming full load with no help of overlap from adjacent teeth.

tion of time. During this moment, the load more than doubles on an area that is not much more than a line contact (Fig. 4-16). The resulting stress will be high enough under full-loaded conditions that pitting will quickly initiate in this area.

Rolling Contact Fatigue

Rolling contact fatigue can be described only by using the illustration in Fig. 4-12. Under any conditions of rolling, the maximum stress applied at or very near the contact area is the shear stress, parallel to the rolled surface at some point below the surface. For normally loaded gear teeth, this distance is from 0.007 to 0.012 in. below the surface, just ahead of the rolling point of contact. If sliding occurs in the same direction, the shear stress increases at the same point. If the shear plane is close to the surface, light pitting will probably occur. If the shear plane is deep due to a heavy rolling load contact, the crack propagation tends to turn inward (Fig. 4-17). The cracks continue under repeated stress until heavy pitting or spalling takes place (Fig. 4-18).

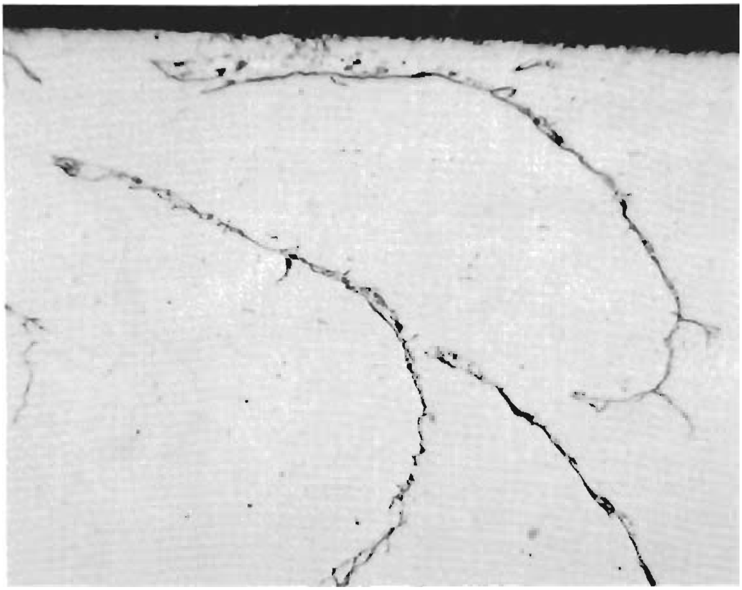


Fig. 4-17. Gear tooth section, 100×. Unetched. Rolling contact fatigue. Crack origin subsurface. Progression parallel to surface and inward away from surface.

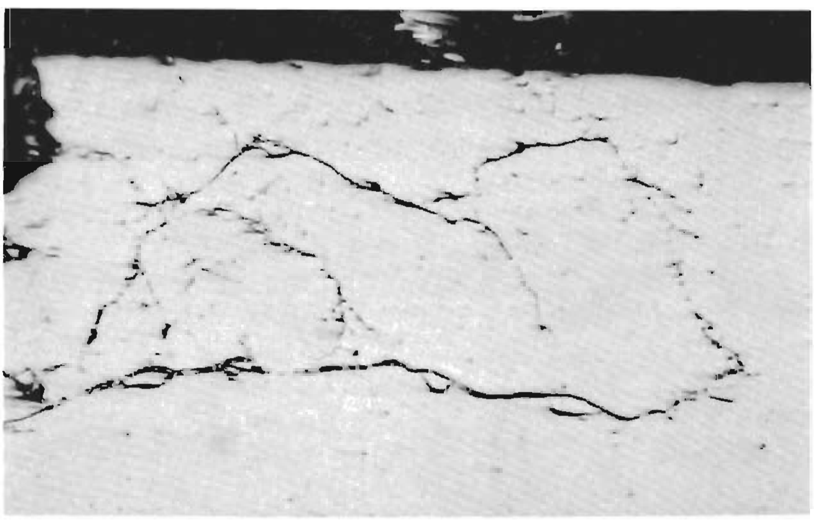


Fig. 4-18. Gear tooth section, 100×. Unetched. Rolling contact fatigue. Crack origin subsurface. Progression parallel with surface, inward, and finally to surface to form a large pit or spall.

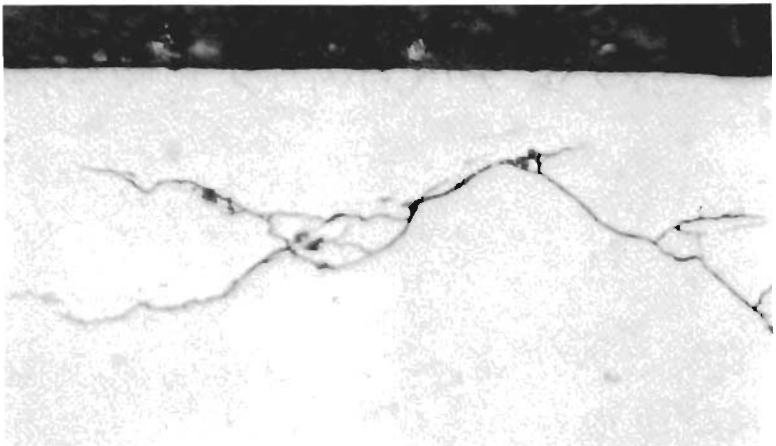


Fig. 4-19. Gear tooth section, 200×. Unetched. Rolling contact fatigue distinguished by subsurface shear parallel to surface. Note the undisturbed black grain oxides at the surface, indicating no surface material movement.

There is always one and often two characteristics of rolling contact fatigue that distinguish it from other modes of surface contact fatigue. Both characteristics can be observed only by examination of the microstructure, which is a metallurgical examination (Chapter 3). In rolling contact, the surface will not show a catastrophic movement; it will remain as the original structure. For instance, an unetched, polished sample taken near the origin of a subsurface fatigue crack (Fig. 4-19) very clearly shows undisturbed black grain oxides at the surface. The subsurface cracking could not have been caused by either abrasive or adhesive contact—it had to be caused by rolling. This is the first evidence to look for in determining the type of applied stress.

The second characteristic of rolling contact fatigue is common only in a martensitic case that contains very little or no austenite and is found at, along, or in line with the shear plane. This is a submicrostructure that has been named "butterfly wings." Take the sample of Fig. 4-19 and etch it properly with a 3% nital solution, and a classic wing shows up, as in Fig. 4-20(a). Increasing the magnification to 500× results in a bit more detail, as in Fig. 4-20(b). The original dissertation was presented by

(a)

(b)

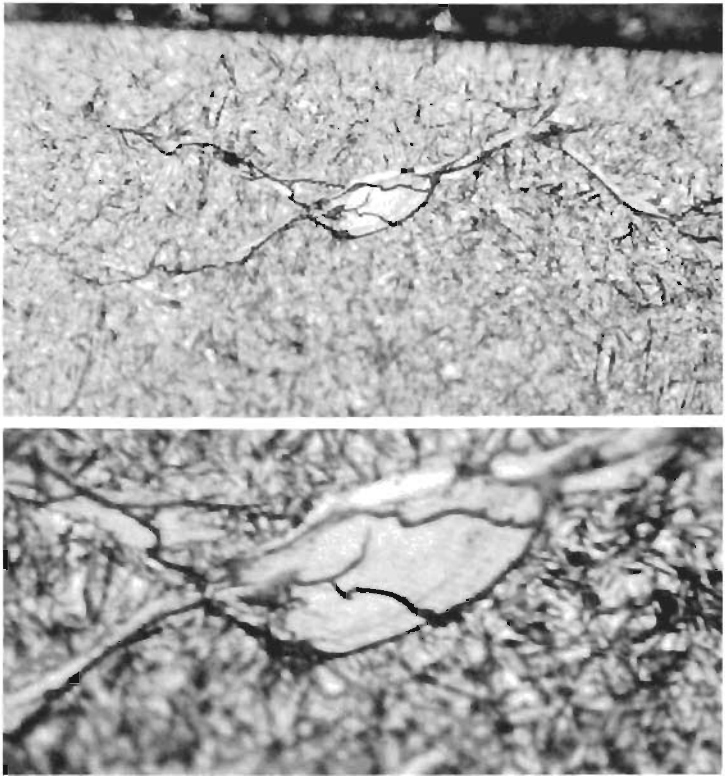


Fig. 4-20. Same sample as Fig. 4-19, 3% nital etch; at 200× (a) and 500× (b), showing details of submicrostructure called "butterfly wings."

the roller bearing manufacturers. They had no explanation for the origin of this structure, except that it "was always in conjunction with and radiated from an inclusion." It is true that many times an inclusion is present, but not always. However, it is arguable that the fracture is a plane and not a line such as in a polished sample; therefore, an inclusion could be present somewhere else along the plane where it is not observed. The next group of studies was twofold—one extensive study by Arthur D. Little Company of Boston, Massachusetts, and a very detailed study of rolling contact fatigue by Illinois Institute of Technology Research Institute (IITRI) in cooperation with Division 33 (ISTC) of the Society of Automotive Engineers. In their reports, the gray substructure was referred to as "white bands" of altered martensite.

During the time of these intensive studies, many such altered structures were observed in gear teeth where there was evidence of very heavy rolling contract. For example, observe Fig. 4-17 and 4-18 as etched, which are now Fig. 4-21 and 4-22, respectively. Note the altered martensite substructure along the sheared plane. These substructures seem to be caused when, under an extreme shearing stress, movement is needed, but is restricted and contained to such an extent that the energy absorbed institutes a metamorphic change in the microstructure ahead of the progressing crack. The substructures are never observed when significant amounts of austenite are present, because austenite will quickly absorb the energy and be converted to untempered martensite. The substructures are also in an area that has not been deformed but has definitely been transformed. Each area has distinct boundaries, and the

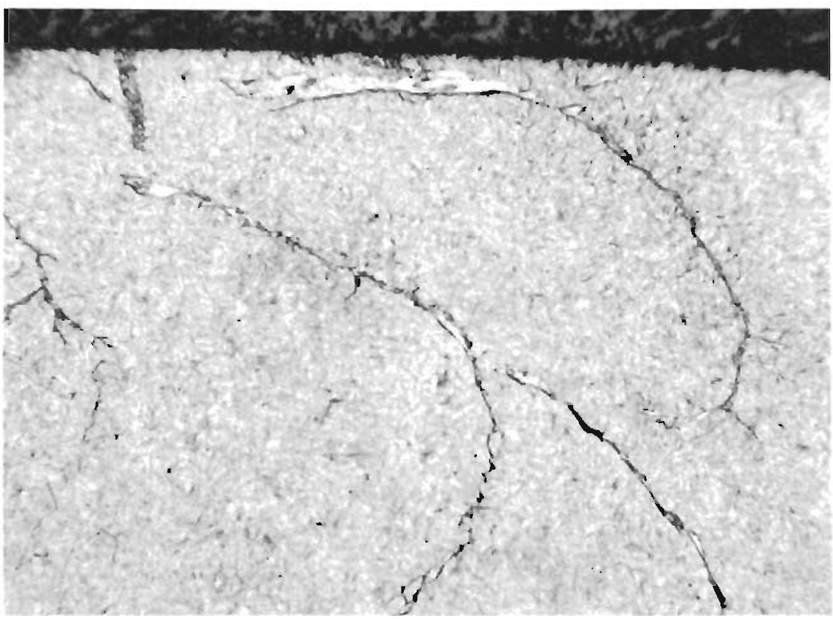


Fig. 4-21. Same sample as in Fig. 4-17. Nital etched. Note the "butterfly wings," or altered martensite, along the shear planes.

Fig. 4-22. Same sample as Fig. 4-18. Nital etched. Note the "butterfly wings," or altered martensite, along the shear planes.

oncoming cracks appear to follow those boundaries. Many academic studies refer to these same structures as transformed shear band products formed by "adiabatic shear."[5]

Contact Fatigue (Spalling)

Spalling generally is not considered an initial mode of failure, but rather a continuation or propagation of pitting and rolling contact fatigue. It is very common to refer to this failure mode as "pitting and spalling." As an example, Fig. 4-23 shows a spiral gear tooth with pitting low on the profile, which subsequently progressed until spalling occurred over the top face and back side profile. This apparently rapid and extensive progression is often referred to as the "cyclone effect."

Idler gears were removed from three planetary drive assemblies. The teeth on each gear showed a remarkable example of pitting and spalling progression. Figure 4-24(a) shows lines of pitting just below the pitchline; (b) shows the pitting progressed to light spalling up and over the addendum; and (c) shows complete spalling of all teeth.

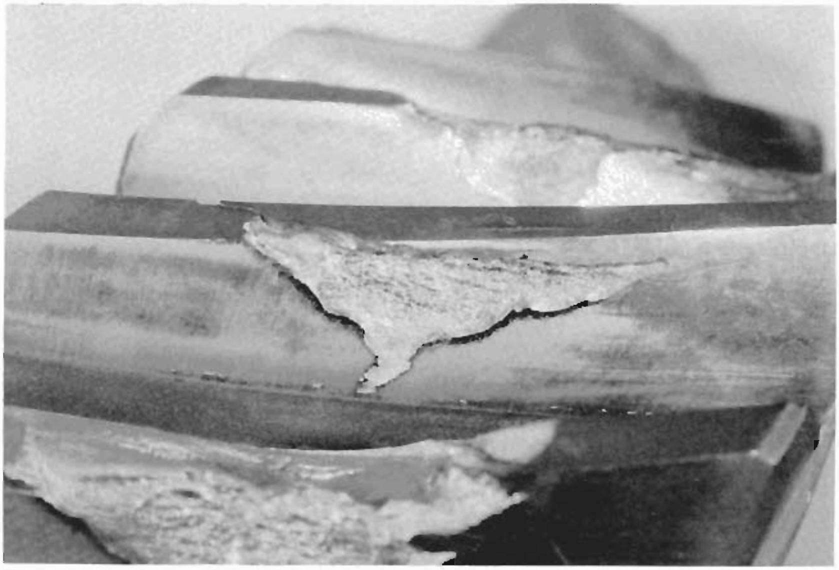


Fig. 4-23. Spiral bevel gear teeth, 1.5×. Original pitting low on the active profile gives initiation to a fast and extensive progression of spalling over the top face and down the back profile. This is often called the "cyclone effect."

Spalling, in the true sense of the word, is a distinctive mode of fatigue failure unique in origin.[6] It originates subsurface, usually at or near the case/core transition zone. As illustrated in Fig. 4-25, the origin is at the point where the sum of all applied and misapplied stresses intersects the net strength of the part. The applied stresses are substantially in shear, and the point of intercept is most likely under a carburized case. Fatigue fractures generally progress under the case as shown in Fig. 4-26, and will eventually spall away from the gear tooth.

Thermal Fatigue

Thermal fatigue in gearing is most often considered (although not necessarily correctly) to be synonymous with frictional heat. This is virtually the only kind of alternating heating and cooling that is applied in the field. Then, too, thermal fatigue is not commonly found on the active profile of the gear

(a)

(b)

(c)

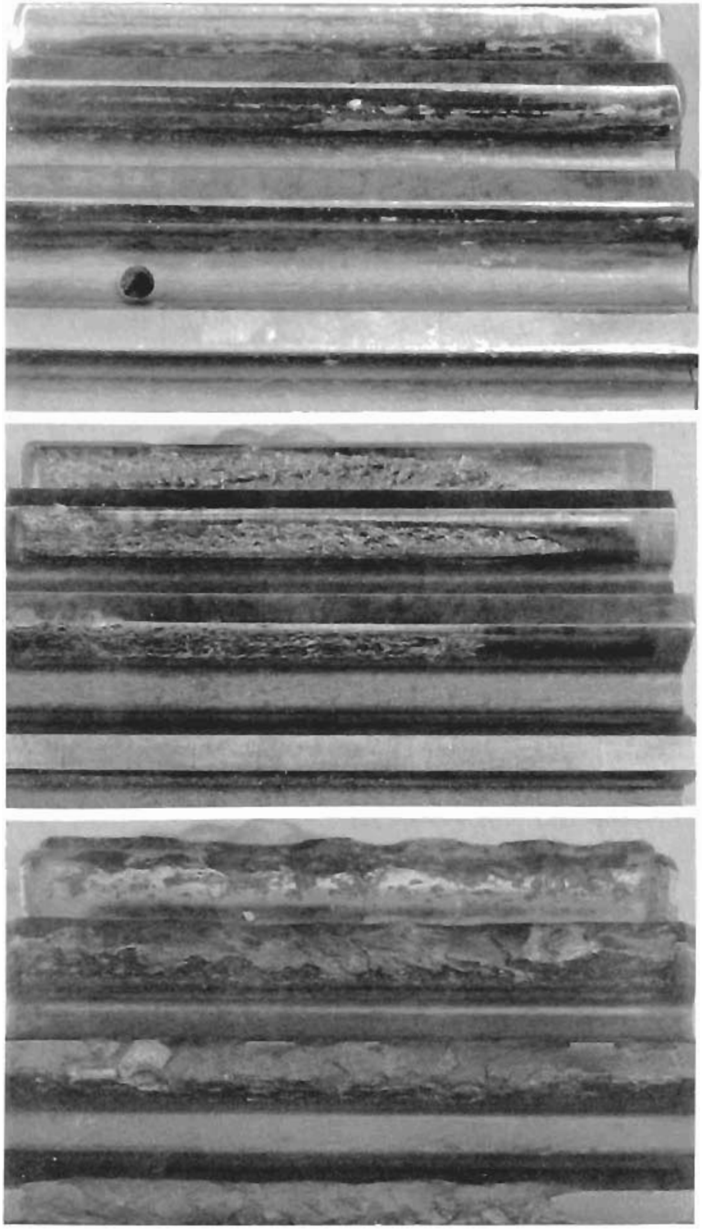


Fig. 4-24. Spur gear, 0.5×. Pitting fatigue progressing to spalling. (a) Lines of pitting just below the pitchline; (b) light spalling up and over the addendum; (c) complete spalling of all teeth.

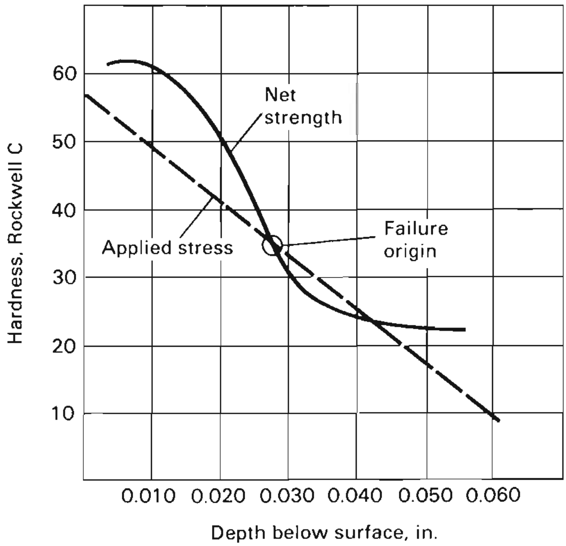


Fig. 4-25. Applied stress vs case depth (net strength).

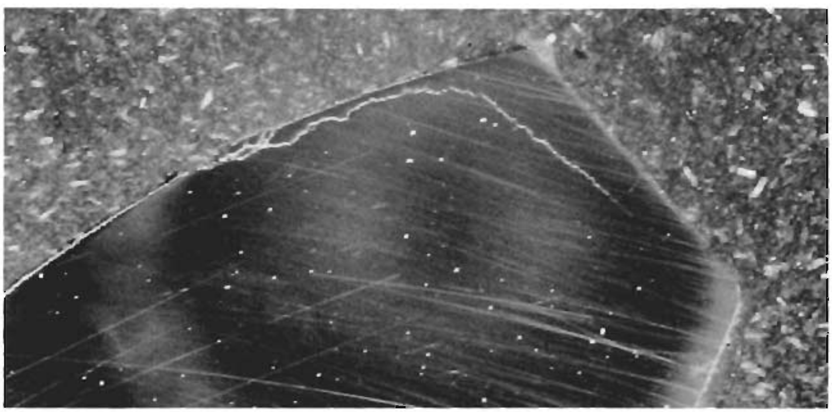
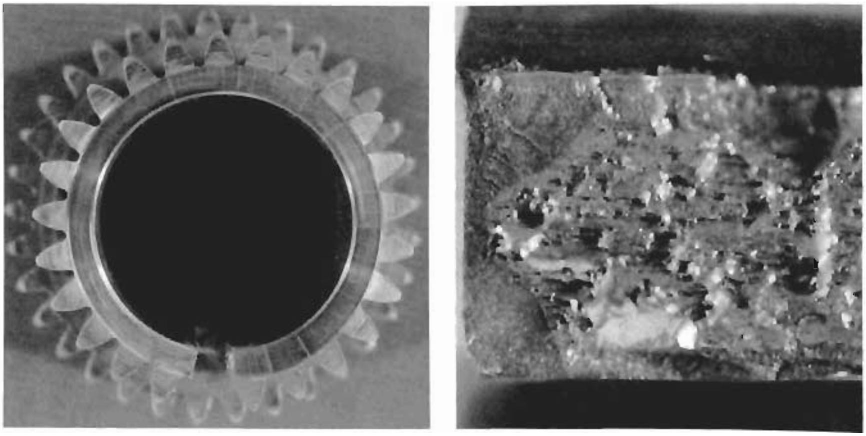


Fig. 4-26. Spalling—a subsurface fatigue failure originating at the case/core interface, subsequently progressing under the case.

teeth but on a rotating face exerting a high amount of thrust. Figure 4-27(a) is an example of high end thrust that resulted in frictional heat. The thermal expansion and contraction initiated a large number of radial cracks that progressed as a fatigue fracture (Fig. 4-27b) both into the bore of the part and along the roots of the teeth.



(a) (b)

Fig. 4-27. (a) Spur gear showing radial cracking due to frictional heat against the thrust face; 0.5×. (b) Progression of thermal fatigue produced by the frictional heat; 2×.

There is a remote chance that grinding checks and quenching cracks could be construed as very rapid thermal fatigue. Both are discussed as manufacturing causes in Chapter 5.

Fatigue of Round, Splined, and Keyed Bores

Fatigue fractures moving from the bore outward, with very few exceptions, originate at a high stress-concentration point. The subject of these applied stresses has been covered thoroughly in Chapter 1, in the section titled "Associated Parameters."

Shaft Fatigue

Torsional fatigue as a failure in the tensile plane follows a direction of 45° to the central axis (see Fig. 3-2). However, the weakest plane of a shaft or pinion shank is the shear plane in the longitudinal direction. Under continual reversal of torsional loads that exceed the shear strength of the material, the crack will initiate at the surface, if the hardness is consistent

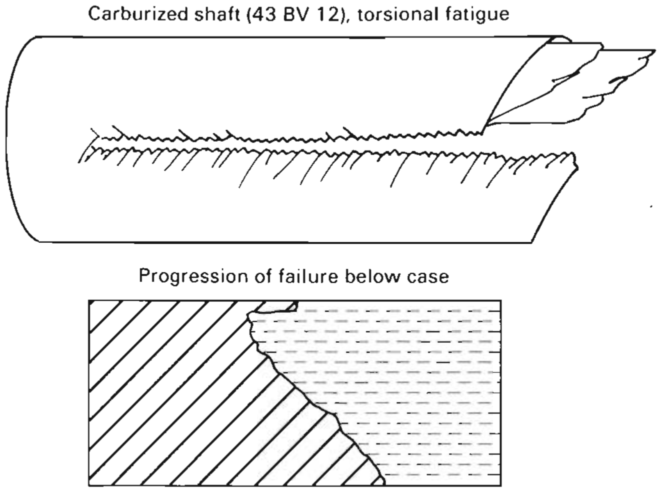


Fig. 4-28. Schematic showing subcase failure of bidirectional torsional shear fatigue followed by torsional tensile failure of the case.

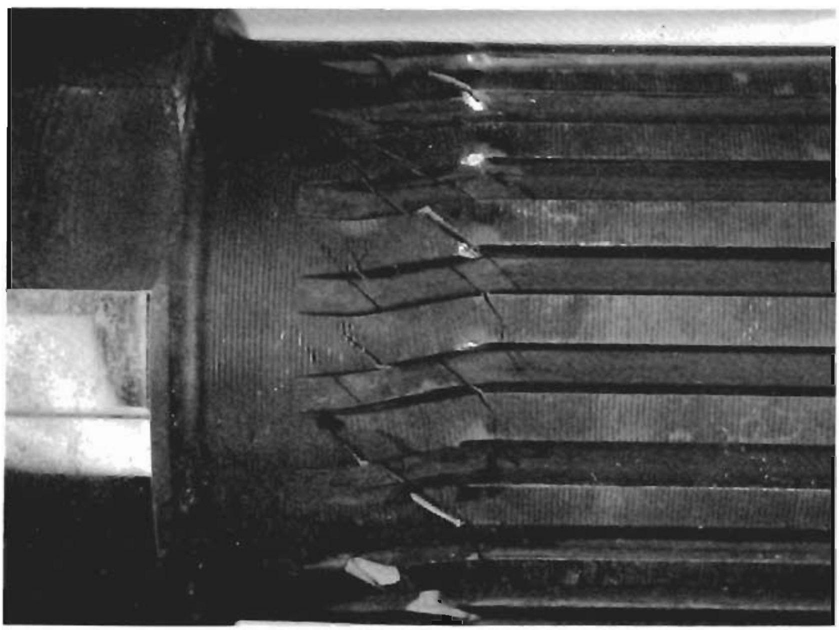


Fig. 4-29. Splined section of a pinion shank. Torsional tensile fatigue in one direction showing the 45° tensile failure lines and evidence of longitudinal shear cracking.

throughout the cross section. However, if the surface is harder, either by carburizing or by induction hardening, the crack will most likely initiate subsurface at the case/core interface. The schematic shown in Fig. 4-28 is typical of a longitudinal fatigue failure in a carburized shaft. The initial failure is shear fatigue at the case/core interface, which continues to progress through the shaft and longitudinally under the case. The second failure occurs as the case ruptures in a 45° cross-hatched pattern typical of a tension failure in torsion, since the case is weaker in tension than in shear.

When splines are present, such as in the pinion shank shown in Fig. 4-29, the area most susceptible to failure is between the juncture of the mating part and the change of section beyond that juncture. Note three fractures involving the failure in Fig. 4-29. First, the 45° cracks show torsional tensile failure in one direction only; also, the entire short area has been "stretched" in the displacement; and finally, there are longitudinal cracks along the root radii of the displaced splines. A typical three-directional fatigue failure leads ultimately to the classic "rosette" fracture (Fig. 4-30) often observed in splined shaft failures.

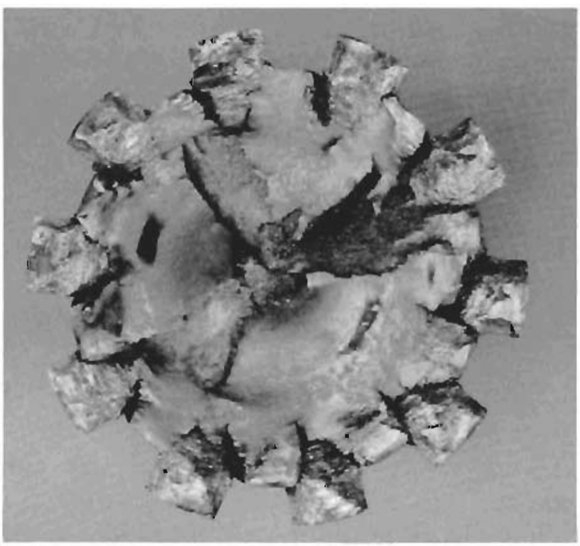
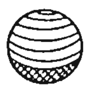
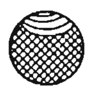
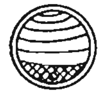
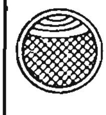
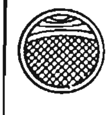
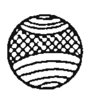
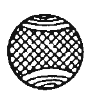
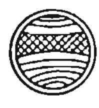
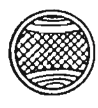
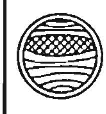
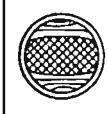
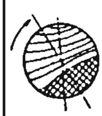
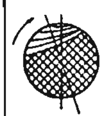
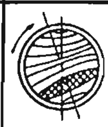
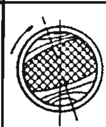
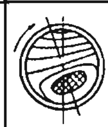
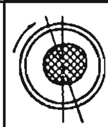


Fig. 4-30. Splined section of a shaft. The typical three-directional reversing torsional fatigue "rosette."

Table 4-1. Fracture appearances of fatigue failures in bending[7]

Case \ Stress condition	No stress concentration		Mild stress concentration		High stress concentration	
	Low overstress	High overstress	Low overstress	High overstress	Low overstress	High overstress
One-way bending load						
Two-way bending load						
Reverse bending and rotation load						

Bending fatigue in shafts and pinion shanks is seen in many different patterns depending on the direction of loading and the continued load application. Charles Lipson[7] devised a table (Table 4-1) comprising the qualitative effect of loading method, loading magnitude, and stress concentration on the appearance of the fracture. This table can be used very effectively to judge types of bending fatigue. For instance, observe the similarity between illustrations in the table and Fig. 4-31, 4-32, 4-33, and 4-34.

Impact

Tooth Bending Impact

When a tooth is removed from a gear within very few cycles (usually one or two), the resulting fracture is uniform in structure and does not show the fatigue striations common to the fatigue mode of failure. The failures are usually random, due to a sudden load, in either a forward or a reverse direction, and do not necessarily originate at the root radius. In fact, if the fracture originated at the root radius, it would follow a rather

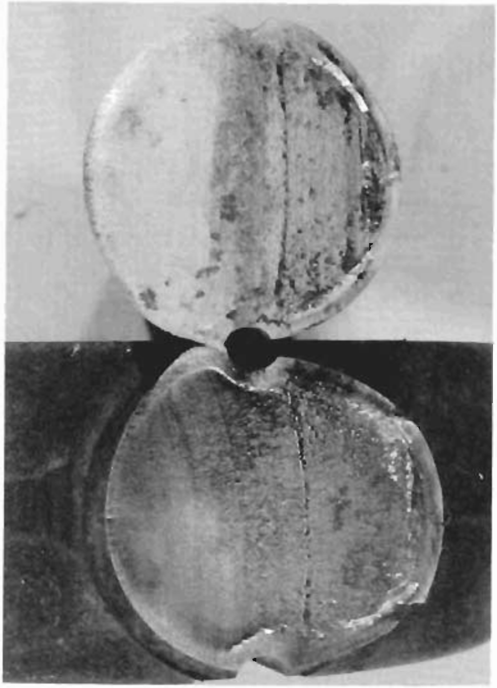


Fig. 4-31. An arm of a differential spider. Two-way bending load; mild stress concentration; very low overstress.

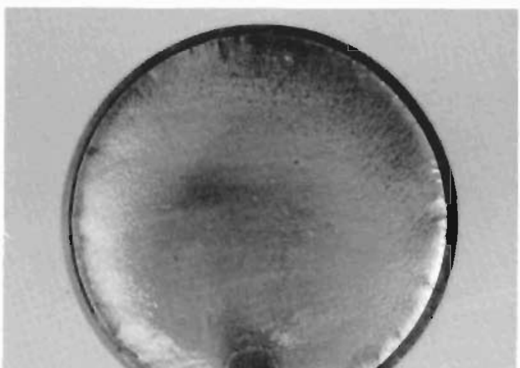


Fig. 4-32. A 1⅛-in. shaft next to thread relief. Slight reversed bending under rotational load; high stress concentration; very low overstress.

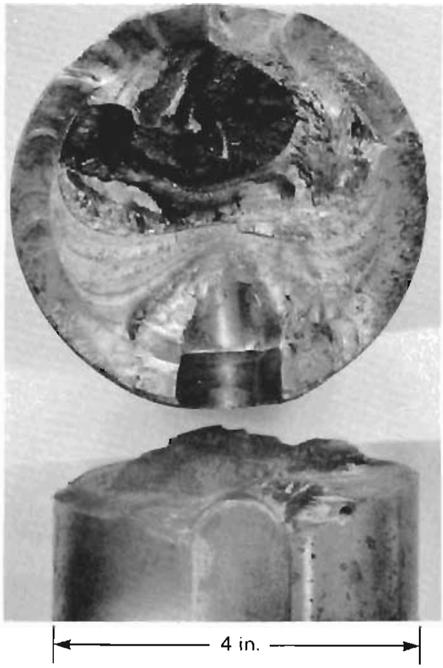


Fig. 4-33. A 4-in.-diameter keyed shaft. Reversed bending; rotational load; high stress concentration; high overstress.

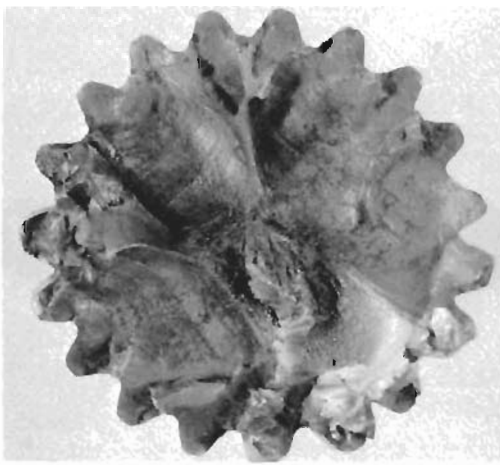


Fig. 4-34. A 5-in.-diameter taper splined shaft. Reversed bending; rotational load; very high stress concentration; high overstress.

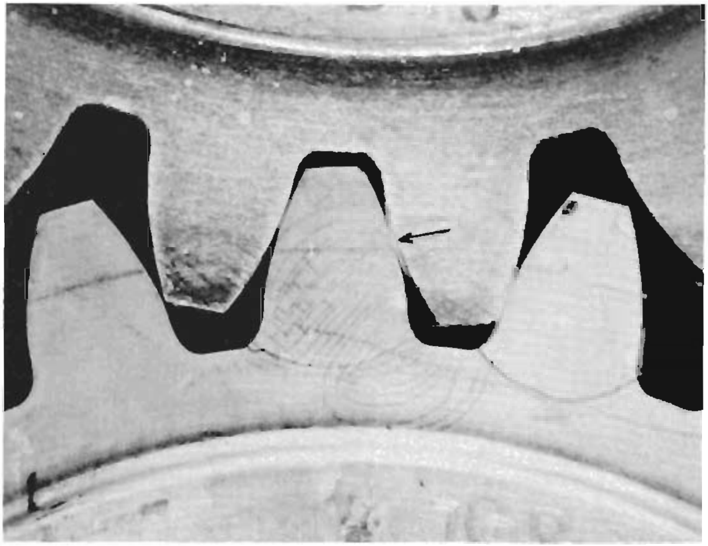


Fig. 4-35. Tooth bending impact with stress-coat overlay, showing fracture path short-circuiting the usual stress flow-lines.

flat path across to the opposite root radius, rather than travel downward toward the zero-stress point. Figure 4-35 shows a stress-coated overlay above a one-shot broken tooth. The fracture did not have time to follow the usual stress pattern. A field failure example is shown in Fig. 4-36.

Tooth Shear

When the impact load is very high and the time of contact very short, and if the ductility of the material will allow it, the resultant tooth-failure mode will be shear. The fractured area appears to be highly glazed, and the direction of the fracture is from straight across the tooth to a convex shape. For instance, a loaded gear and pinion set were operating at a high rate of speed when the pinion stopped instantaneously [Fig. 4-37]. The momentum of the gear was great enough to shear the contacting pinion teeth from the reverse direction, leaving the remaining teeth in excellent condition. The gear teeth were partially sheared and were all "scrubbed" over the top face from the reverse direction.

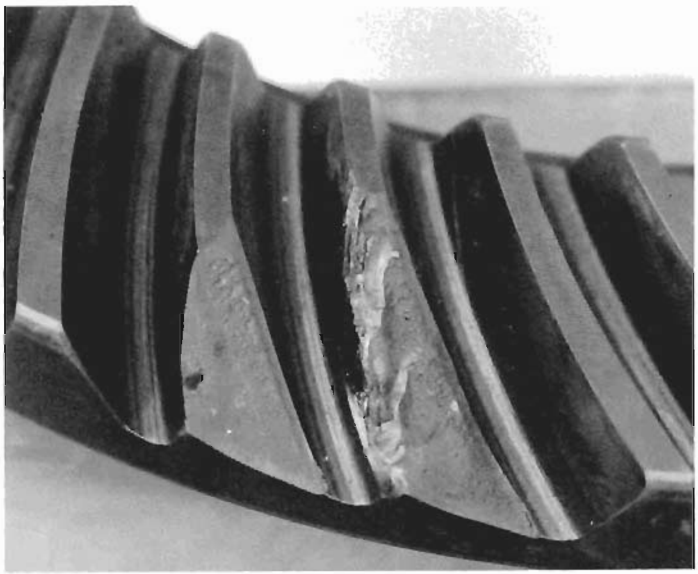
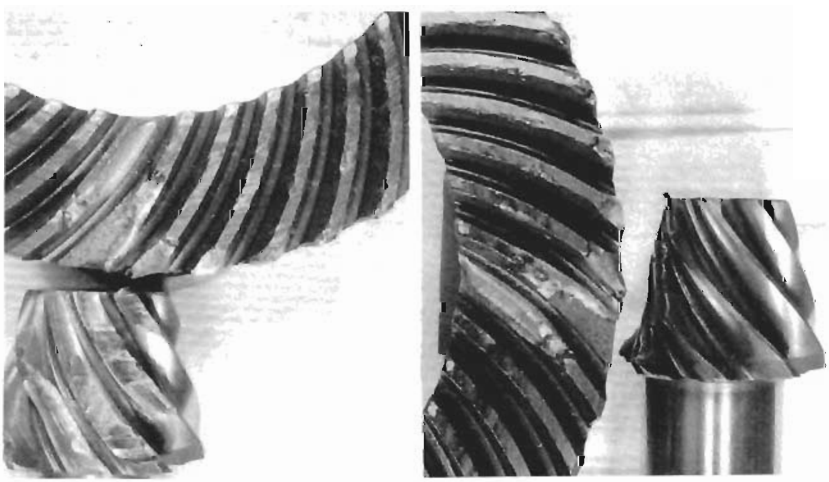


Fig. 4-36. Spiral gear teeth, 1×. Tooth bending impact with peak loads being applied high on the profile over the top corner of the heel end of the convex (loaded) side.



(a) (b)

Fig. 4-37. Spiral bevel gear and pinion set, 1.5×. Sheared in reverse direction. The pinion came to a sudden and complete stop at the instant of a primary failure of the unit, allowing the gear to shear the contacting teeth and to continue rotating over the failed area.

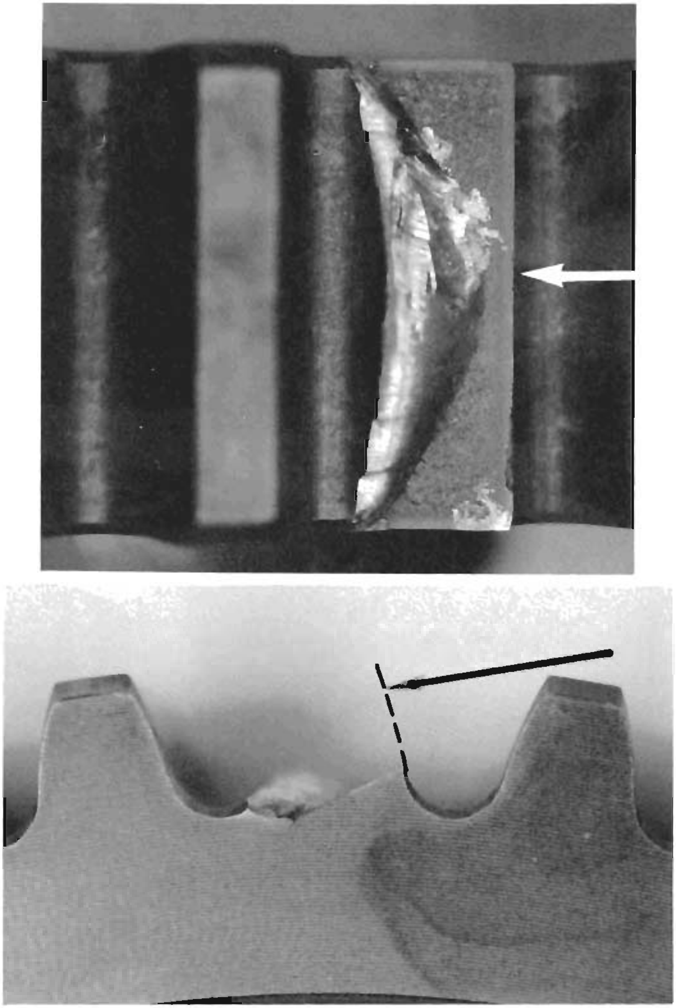


Fig. 4-38. Spur gear tooth. Combination modes: Tooth bending impact (top); tooth shear (bottom). Arrows indicate direction of applied force.

Most commonly, bending impact and shear are found in combination. Figure 4-37(a) shows this to be the case with the gear teeth fractures. Also clearly defined is a spur gear tooth shown in Fig. 4-38. Tooth bending impact fractures supersede the shearing fractures, and thus, the direction of impact can be determined.

Tooth Chipping

Tooth chipping is certainly a type of impact failure but is not generally considered to be tooth-to-tooth impact. It is usually accomplished by an external force, such as a foreign object within the unit, a loose bolt backing out into the tooth area, or even a failed tooth from another gear. Figure 4-39 shows a chipping condition that existed during the operation of the part. Most chipping, however, has occurred through mishandling either before, during, or after the finished parts have been shipped from the manufacturer, and is not a "field" failure. This is one mode that must be identified by a cause.

Fig. 4-39. Tooth chipping as a field failure generally shows a "pattern"; i.e., some object of impact was within the assembled unit.

Case Crushing

Case crushing will occur when an extreme overload is applied to a carburized case. Four factors combine to cause a case to be crushed. Consider Fig. 4-40: Case crushing depends on the stress applied at the point of contact, the radius of curvature of the contacting surfaces, the thickness of the case, and the hardness of the core material. The resulting failure is caused by

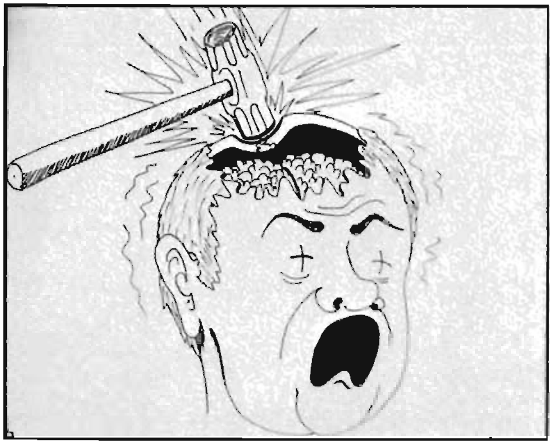


Fig. 4-40. Case crushing depends on stress applied, radius of curvature, case depth, and core hardness.

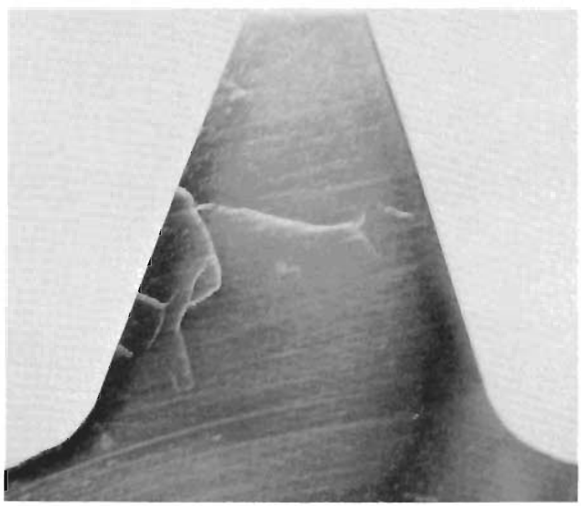


Fig. 4-41. Case crushing at midprofile of a spiral bevel gear tooth. Progression is from the subcase area into the core and outward to the surface.

compressive loads per unit area that are excessive for the existing conditions. The fracture starts at the case/core interface and continues to shear into the core and outward to the surface. Figure 4-41 shows the subsurface propagation of the crushing effect. (Refer also to Fig. 3-1 for a view of a typical surface appearance of case crushing.) For an example quite the opposite of

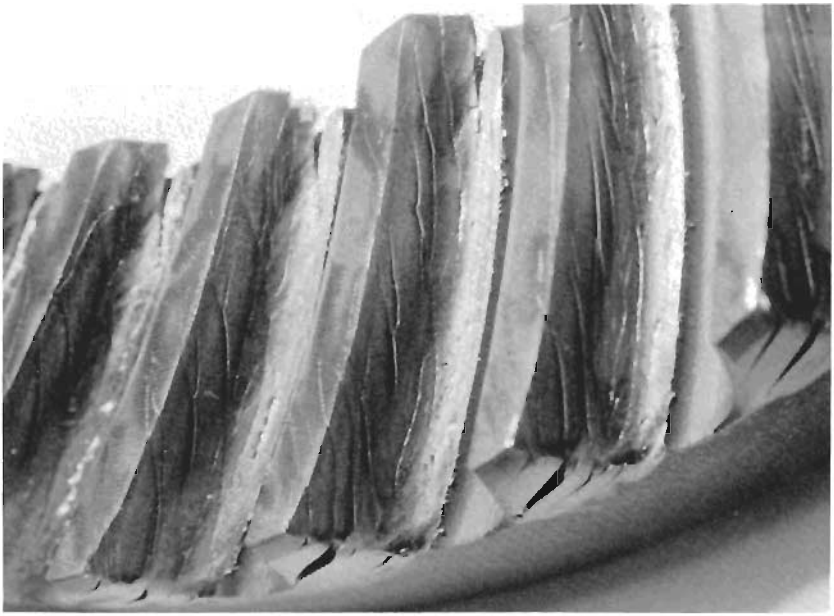


Fig. 4-42. Extensive case crushing. Heavy loads, thin case, soft core.

the one in Fig. 4-41, see Fig. 4-42: very heavy loads, light case depth, and low core hardness; it displayed excellent ductility and never lost a tooth.

Torsional Shear

Again, our attention is turned to shafts or to pinion shanks directly subject to torsional stress. In most instances, torsional shear is a secondary mode following the mode of torsional fatigue, as shown earlier in Fig. 4-30. But sudden shear is readily identified; its appearance (Fig. 4-43) is unique in straightness and texture.

Wear

Surface deterioration of the active profile of the gear teeth is called "wear." There are two distinct modes of wear that will be discussed—abrasive and adhesive.

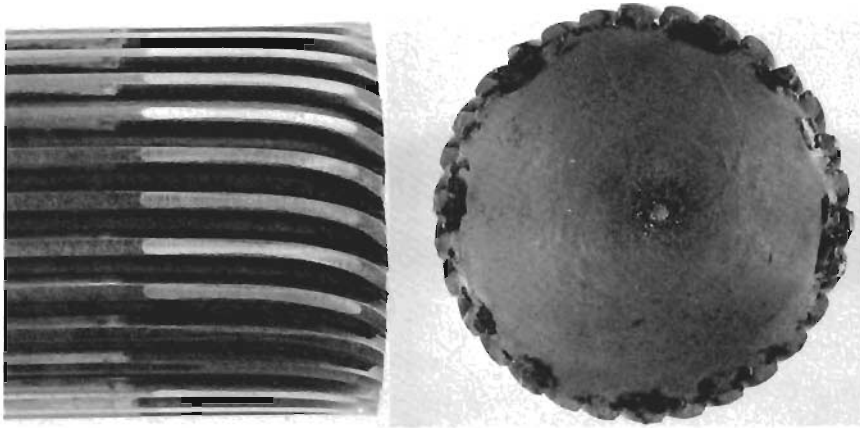


Fig. 4-43. Torsional shear. Suddenly applied extreme torsional overload.

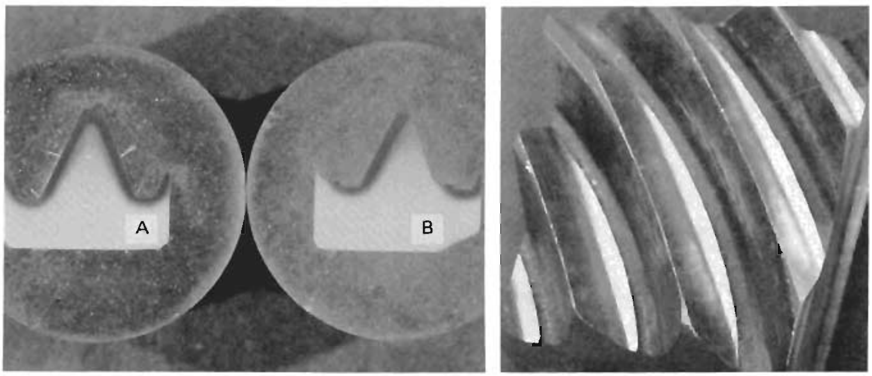


Fig. 4-44. Spiral bevel gear teeth showing contact wear. Insert A is a tooth area nonworn. Insert B shows abrasive wear clearly cutting away 1/8 in. of the surface without damage to underlying material.

Abrasive Wear

Abrasive wear occurs as the surface is being cut away by hard abrasive particles. It can happen only as two surfaces are in sliding contact. The dissociated material must continually be washed away and not be allowed to build up on the sliding surfaces, or adhesions may take place. The first evidence of abrasive wear is the appearance of light scratches on the surface, followed by scuffing. As the scuffing deepens, scoring results. Figure 4-44 is an excellent example of pure abrasive wear,

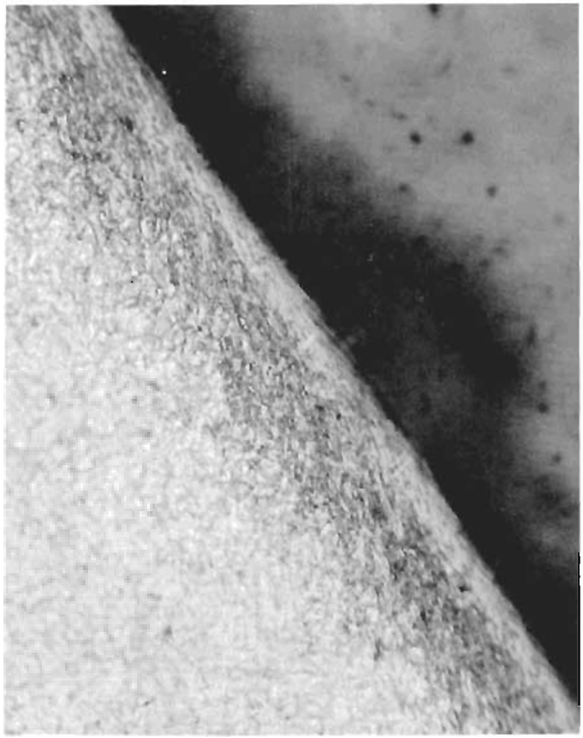


Fig. 4-45. Pinion tooth profile, 100×. Glazed surface showing the start of catastrophic movement of surface material. Frictional heat has already started to temper the surface.

showing the entire tooth profile cleanly cut away with no microstructural damage to the underlying material.

Abrasive wear cannot be deterred by lubrication, because the lubricant is often the vehicle that contains and continually supplies the abrasive material as a contaminant. When contamination occurs, all moving parts within the assembly are affected, including seals, spacers, bearings, pumps, and mating gears (see Fig. 2-3).

When abrasive wear is isolated to only one part (i.e., either gear or pinion), it is imperative to examine closely the surface of the mating part. For instance, a very heavy amount of massive carbides (refer to Fig. 3-27 and 3-28) that impinged upon the surface may easily cut into a softer mating surface.

Adhesive Wear

Adhesive wear occurs on sliding surfaces when the pressure between the contacting asperities is sufficient to cause local plastic deformation and adhesion.[8] Whenever plastic deformation occurs, energy is absorbed as heat—frictional heat. The first indication of trouble is a glazed surface, followed by galling, then seizure. A glazed surface may not undergo any dimensional changes, but examination of the microstructure reveals a catastrophic movement of surface material (Fig. 4-45). As frictional heat increases, the surface softens, and adhesive ability becomes greater; further plastic deformation occurs; the heat becomes high enough locally to change the microconstituent at the surface completely (Fig. 4-46); and now galling oc-

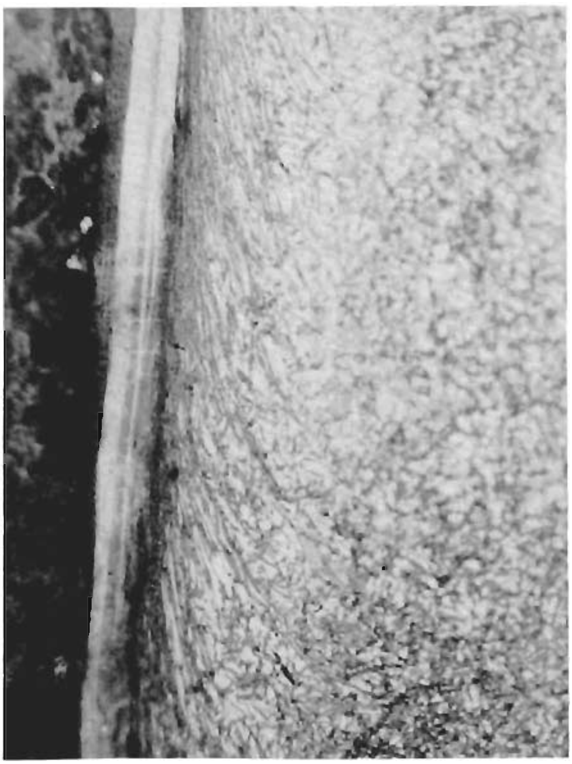


Fig. 4-46. Pinion tooth profile, 100×. Plastically deformed by frictional heat and sliding pressures. Surface layer has locally rehardened and galling is evident.

Fig. 4-47. Spiral bevel pinion, 3/4×. Glazing, galling, and adhesure over the active tooth profile.

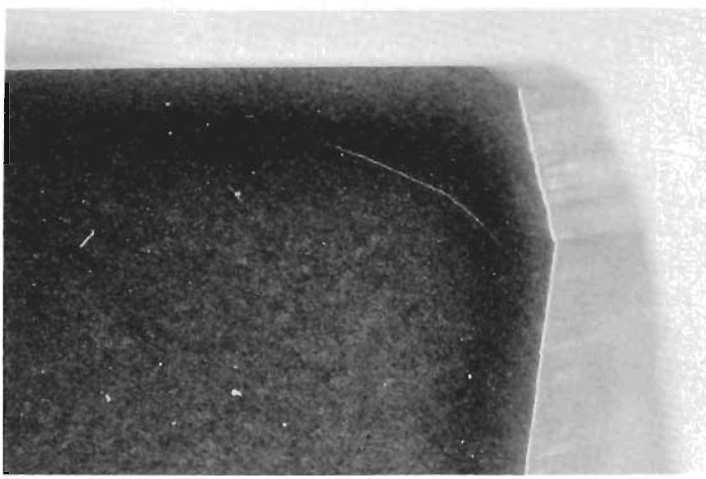


Fig. 4-48. Spiral bevel gear tooth. Internal rupture at top corner of case/core interface.

curs. As galling continues, the sliding surfaces begin welding together, and adhesures take place. Large areas may be pulled away from the surface (Fig. 4-47). Some of the welded particles are so hard, due to change of microstructure, that they become sharp points that cut into the mating parts. As adhesive wear continues with the parts still in service, there is the distinct possibility that the part will ultimately be worn beyond repair (see Fig. 2-4).

Adhesive wear on gear teeth may not always point to the set of gear teeth as the culprit. There are instances when severe adhesures are secondary; particles of material from a primary failure of another component in the assembly may impinge upon the gear tooth surfaces, starting the sequence of events that leads to seizure and destruction.

Stress Rupture

When the internal residual stresses build up to a magnitude beyond the strength of the material, the part will rupture. The rupture occurs at the point at which this critical value is exceeded, either internally or externally.

Internal rupture. The point within a gear most likely to attract a buildup of residual stresses is the case/core interface near the top face or corner of a tooth. Figure 3-3(b) shows an excellent example of case/core separation due to internal residual stresses exceeding the strength of material at the case/core transition zone. Figure 4-48 shows the same mechanism at the corner of a tooth. In severe cases, the entire top of the tooth might "pop off" (Fig. 4-49).

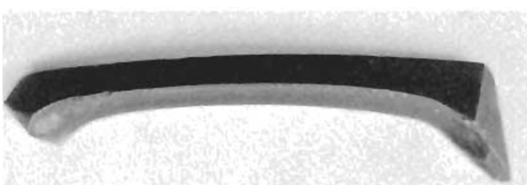


Fig. 4-49. Spiral bevel gear tooth. Internal rupture lifting the entire top of a tooth.

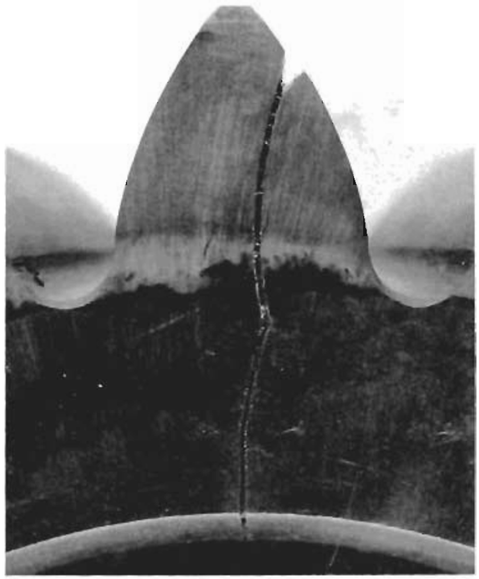


Fig. 4-50. Spur gear. External rupture (assembled, but not in service) with origin at the end face. (See Fig. 3-5 and 4-51.)

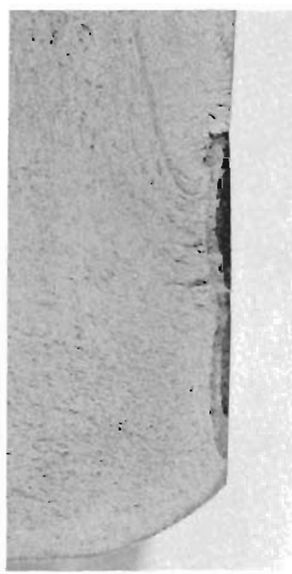


Fig. 4-51. Spur gear. External rupture originating from a grinding check. (See Fig. 3-5 and 4-50.)

External rupture is much easier to understand because it almost always originates with a prearranged stress raiser. Refer to Fig. 3-5. This part (not yet in service) ruptured starting at the end face (Fig. 4-50) from a grinding crack shown in Fig. 3-5 and confirmed by the fractured face (Fig. 4-51). The grinding checks were not the entire cause of the failure, but they supplied the "notch" in the highly stressed area.

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 and L. V. Cowell, *Handbook of Mechanical Wear*, Ann Arbor: University of Michigan Press, 1961, p 135.
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 SD4052A, May 1966, in *Source Book on Gear Design, Technology and Performance*, American Society for Metals, 1980, p 366-411.
5. Harry C. Rogers, Adiabatic Plastic Deformation, in *Annual Review of Materials Science*, Vol 9, 1979, p 283-311.
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 1), in *Source Book on Wear Control Technology*, American Society for Metals, 1978, p 1-10.

CHAPTER 5

Causes of Gear Failure

The cause of a gear failure may be blatant enough to be recognized by anyone, or it may be subtle enough to defy recognition by the most experienced analyst. At times, an examiner almost believes that the number of causes may equal the number of failures. This is close enough to being correct that each failure must be treated individually, as a complete case requiring its own attention. As stated earlier, it is very difficult in many instances to separate cause from mode. A single cause can initiate several different modes, depending on the forces applied. Conversely, a specific mode of failure can be initiated by one of several causes. Again, it must be emphasized that a fracture pattern of a failed gear should not be matched with a picture in this text and the conclusion made that the cause and effect are the same. They may be, but not necessarily so. Use the pictures of this text as a guide; then, systematically find the parts of the puzzle and piece them together.

Causes are discussed under five major headings and several subheadings. Undoubtedly there may be more, and the reader should feel free to add to this list. The major divisions are as follows:

- Basic material
- Engineering
- Manufacturing
- Heat treatment
- Service application

Basic Material

With full understanding that gears are also manufactured from nonferrous materials such as copper and aluminum alloys, and from powdered metal products, both ferrous and nonferrous, this text will discuss only steel gears.

Steel

Steel, a basic product utilized in bar stock, forgings, and weldments, has inherent characteristics and defects that carry over into its final products, and thus to gears. Those characteristics applicable are discussed.[1]

Pipe is the shrinkage cavity located in the upper central portion of an ingot. All escaping gases and slag pass through this area and some may be entrapped. As the ingot is rolled, the pipe area is cropped or sheared and discarded. Vestiges of this pipe sometimes remain in the finished bar product and appear in the gear or pinion as exceptionally large inclusions or a grouping of inclusions near the central portion of the rolled or forged product. Figure 3-16(c) shows a clearly defined pipe at the center of a 4-in. square billet to be used for forging. Beyond the area of the pipe, there may be a condition of "sponginess" consisting of slight porosity and pin holes, as shown in Fig. 3-16(a) and (b). Unless the rolled product is cropped sufficiently, the condition of the ingot center will be rolled into its final form as a machinable bar or into a billet for the forging manufacturer. The steel mills check for this condition by macroetching specific samples. Likewise, the forging companies and the users of machinable bar stock usually make routine checks for the same purpose. (Macroetching methods are explained in Chapter 3.) A pinion will certainly retain all of the central area of the bar, but a gear from a center-punched forging may have its central core removed. If not, the remaining pipe material found in a ring-type gear may be in a restricted and concentrated area of the bore.

Blowholes are gas-produced cavities, cylindrical or spherical in shape, found randomly throughout the ingot. They are usually small, and if not oxidized, weld tightly as they are rolled. However, those that are slightly oxidized do not weld together and remain as flakes. The presence of oxygen and hydrogen in the steel-making process is the main culprit in the formation of flaking; so vacuum degassing is now used extensively.

Segregation is a departure from the average chemical composition. Generally the material richest in the alloying elements will be in the zone of last cooling, the ingot center. The amount of segregation, or chemical difference, is very seldom of much consequence, except that it may influence heat treatment. As an example, in a case of extreme segregation, there may be a 0.04% difference in average carbon content from the area outside the ingot to the central area. The central area contains the higher amount of carbon. This difference in carbon content is maintained in the same proportion, even in the rolled bar stock to be used later as forgings or as machined parts. Any gear manufactured from this heat of steel has a marked difference of hardenability from the outer material to the inner material.

Ingotism is the remaining evidence of the columnar structure of the ingot as it has been rolled into usable bar stock sizes. A microstructural type of segregation, it slightly affects tooth characteristics during heat treatment in extreme instances.

Nonmetallic inclusions are contained in all steel ingots and thus, in all steel products. They consist of oxides and sulfides in various combinations and are derived chiefly from the oxidizing reactions of the refining processes, and from the deoxidizing materials added to the molten steel. Some of the large inclusions may be silicates or aluminates resulting from the erosion of the ladle or other refractories during pouring. This latter type of inclusion (illustrated in Fig. 3-3) is usually an isolated incident, random in its position, and not consistent throughout other parts of the same lot.

In the alloy steels used for heavy duty gearing, most nonmetallic inclusions are very small, well scattered, and random. They cause no problem at all, unless they occur at points of critical stress. Figure 4-11 shows a subsurface crack originating

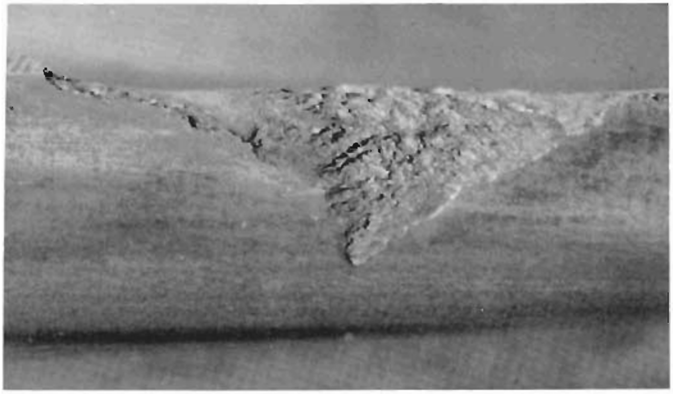


Fig. 5-1. Helical gear tooth, 1×. Pitting/spalling mode originating at a pit caused by subsurface fatigue around a small nonmetallic inclusion. Only one tooth affected. An isolated, random case.

from an inclusion at the pitchline in the area of maximum shear stress, with continued progression outward in all directions toward the surface to form a pit.

Observe Fig. 5-1 and compare with Fig. 4-23. At first glance the fractures may look the same. In fact, this is graphic evidence that two failures that look alike are not necessarily alike. Both are pitting/spalling starting low on the tooth profile and progressing over the top of the tooth; but here the similarity stops, because as the one failure (Fig. 4-23) originates from a definite line of pitting low on the profile, the other (Fig. 5-1) originates from one pit below the pitchline. Also, every tooth in the gear in Fig. 4-23 has been affected by tooth tip interference, whereas only one tooth of the gear in Fig. 5-1 has been affected by one nonmetallic inclusion at a critical stress point, causing a reaction identical to that shown in Fig. 4-11.

Nonmetallic inclusions at a crack initiation point are universally found in a fatigue mode of failure. They must be in the path of high stress concentration, and they must influence that specific area to be the point of least resistance to stress fatigue. (Refer to Fig. 4-7 and the related text discussion.) Note the subsurface circle of striations around the inclusion. This encircling crack progressed rather slowly until it broke through the case to the surface, at which time it rapidly followed the normal

Fig. 5-2. Top corner of a spur gear tooth, 10×. A "bull's-eye" fatigue fracture centering around a nonmetallic inclusion at the case/core interface. Only one tooth spalled. An isolated, random case.

tooth bending fatigue pattern. An example of the effect of a remotely placed inclusion is illustrated in Fig. 5-2. The inclusion is located near the top corner of one spur pinion tooth at the case/core interface. It would never have been instrumental in this tooth failure if the applied load had not shifted from the central area of the tooth to the high outside edge of the tooth profile. At this position, the inclusion is at the applied shear plane and becomes the nucleus for the crack propagation. The fracture starts slowly with a "bull's-eye" pattern, progresses subsurface for a period of time, and finally, drops off as a spall. The cause of failure was the inclusion since the strength of the tooth was sufficient to function under the applied loads, even though they had shifted position.

Figure 5-3 shows an example of a shaft failure in torsional fatigue along the longitudinal shear plane, with its origin centered along an elongated oxide stringer (inclusion) in the core near the case/core interface.

Note: Many fractures show one or more inclusions somewhere on their surfaces. All fractures cannot be caused by an

Fig. 5-3. Torsion fatigue in the longitudinal shear plane centering along an elongated inclusion below the case/core transition zone.

inclusion, even if it is close to the origin (see Fig. 5-29a). For an inclusion to be the cause, it must be evident that the fracture did indeed originate at the inclusion. Sometimes extensive work must be done to establish definite proof (refer to Fig. 3-33).

Flow lines occur in all rolled products; they are the elongation of the crystalline structure along the principal direction in which movement of the material has taken place[2]—a microstructural type of segregation through mechanical means. Flow lines in the bar stock are utilized by the forging industry to strengthen highly stressed areas. For instance, Fig. 5-4 illustrates how the forger has forced the natural flow lines to follow the contoured pattern of a pinion forging blank, thus increasing the bending strength of the pinion head radius.

In the low-alloy steels, the condition of flow lines is more prominent and is usually termed "banding," referring to the observed alternating layers of ferrite and pearlite. Two conditions are noted by these bands: nonmetallic inclusions generally follow within the ferrite band (Fig. 5-5 and Fig. 3-13); and the pearlite band will generally respond to martensitic transformation during heat treatment more easily than will the ferrite band. This is probably caused by a difference of hardenability within the bands, the result of which is not erased by the heat treat

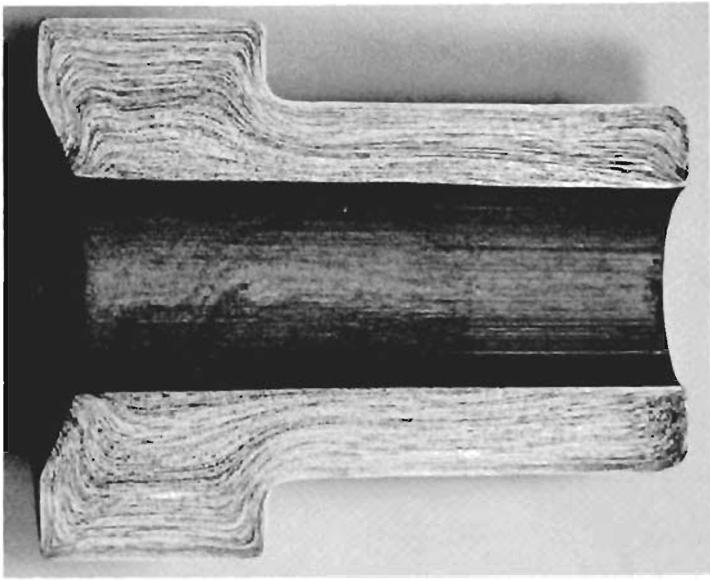


Fig. 5-4. Normal hot rolled steel bar flow lines have been pressed into a forging-die cavity to strengthen the ultimate product.

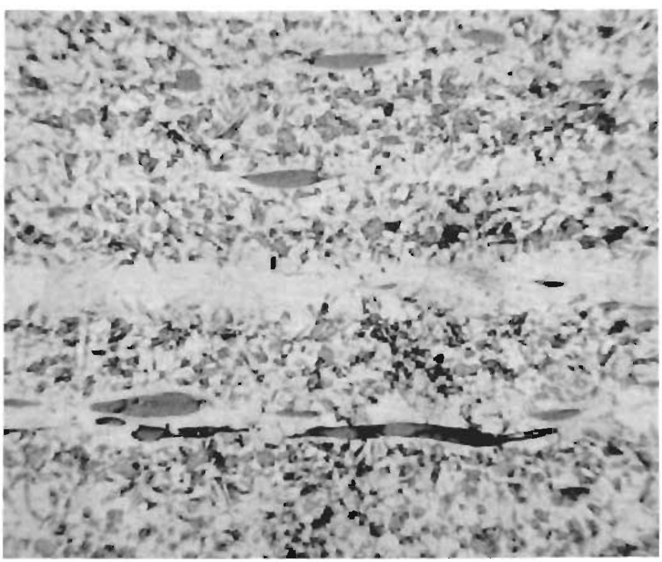


Fig. 5-5. Microstructural banding shows alternating layers of ferrite and pearlite. Nonmetallic inclusions tend to follow the ferrite bands.

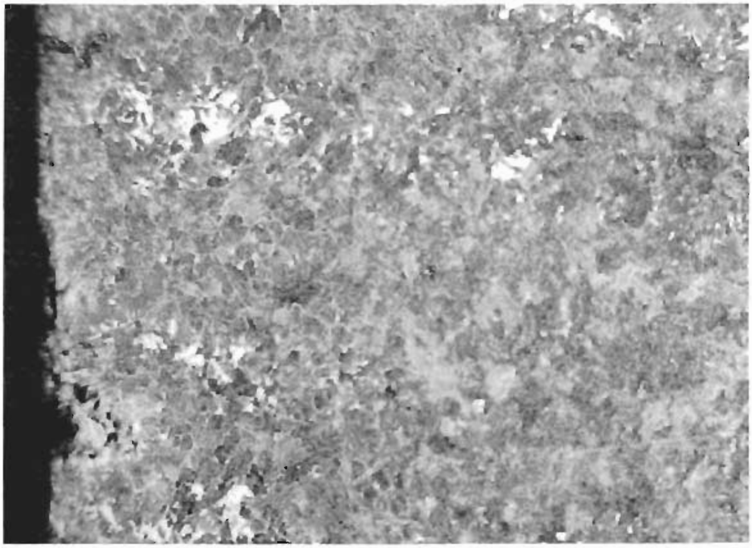


Fig. 5-6. Banding influences the diffusivity of carbon during carburizing. Note the resulting banded areas of retained austenite (white structure).

operation but may also extend into the case (Fig. 3-31). Even in high-alloy steels, the diffusion of carbon into the surface by carburization is affected by banding. Figure 5-6 illustrates the bands of austenite in a matrix of martensite surrounded by a cementite network.

Seams are formed during the rolling or drawing operation of bar stock and are a surface condition. They appear as open cracks and may not be very deep on the rolled stock, but the forging operation may open them into deep cracks. Also, the open seams are found on each surface of the forged part regardless of how many reduced steps may have been taken.

Forgings

As the defects and characteristics of basic steel products affect the forging operation, the forging defects and characteristics affect all subsequent operations. Again, there is repetition, but repetition with refinement. During hot forging operations, it is common for surface defects to occur.[3] Larger defects

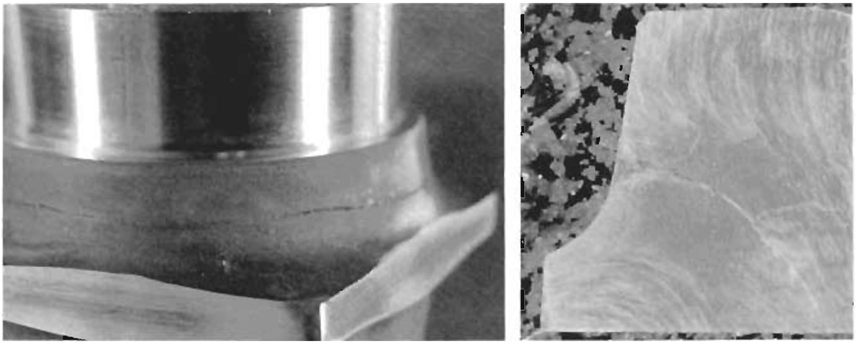


Fig. 5-7. Forging lap around a forged hub. Crack followed the normal flow-line pattern and retained oxidized scale along its surface.

are visible and can be removed at the forge source; but small imperfections may not be visible, so most of these imperfections are removed during subsequent machining operations.

Flow lines. The flow lines and banding discussed above have been reoriented to follow the direction of the material into the forging die. The forging press is designed to direct this flow evenly and consistently throughout the volume of the die (Fig. 3-10, 3-11, and 5-4). When this is not the case (Fig. 3-12) nonuniform gear tooth characteristics may occur during subsequent heat treatment.

Forging laps are formed at or near the surface of a forged part when an overlapping piece of material is pressed back into the part, yet is separated by an oxidized film or scale (Fig. 3-15). At times, a hardening quench crack can be confused with a forging lap; but a microscopic examination reveals the difference, because the crack formed by a forging lap will show a flow-line pattern and will have a scaled surface (Fig. 5-7). A special type of forging lap occurs when the cooled corners of a billet are pressed into the gear blank. It is easily recognized by the occurrence of one to four quarter-moon-shaped laps, each at a quadrant.

Internal rupture may be caused by uneven heating of the billet that allows a differential stress during deforming. Generally such ruptures are formed by an opening of residual internal cracks and flakes from the bar product. Various types are called bursts, mechanical pipe, clinks, or fire cracks.

Shear tears are evident when the material has been torn apart during the shearing operation. This occurs under only two circumstances: when a continuous bar is being sheared into short multiples, the end face of the short bars may tear if not cleanly sheared; or when a center punch is shearing the central plug from a forged blank, it may tear the inside bore-face of the blank.

Overheating or burning during the heating operation of the billet for forging will most frequently occur in a localized area due to flame impingement, poor furnace design, or poor operation. This causes large-grain growth structures in the steel and ultimately instills oxides, voids, segregation, porosity, or cracks within the grain boundaries. When burning is suspected, it is easily detected by magnetic-particle inspection; however, when it is unsuspected, the discovery can be as dramatic as that shown in Fig. 5-8.

Castings

The castings associated most with or having an influence on gears are the housings used in the gear box assembly. This may be a transmission box, a differential carrier, a wheel reduction unit, or a speed reducer assembly. The purpose of the casting is to house the gear assembly and the lubricant.

The first casting characteristic to note is the accuracy of the alignment of the locating holes in the housing. Misalignment within the assembly housing causes many problems for both the gears and the bearings. A misaligned gear tends to have a load pattern shifted toward one end of the teeth, but the pattern will be uniform and consistent on all teeth. The shaft will be in a continuous state of bending in one spatial direction, but with rotational bending around its own surface.

The second characteristic to note is the rigidity of the hous-

Fig. 5-8. Burned forging. A localized area of exceptionally large-grain structure with oxides, voids, segregation, and cracks within the grain boundaries.

ing assembly. Lack of proper stability leads to deflections that cause many unusual problems for the gears. If cyclic loads are applied, a cyclic problem may result; if a random peak load is applied, a random type of failure may result. Lack of rigidity invariably increases the possibility of active vibrations.

The third characteristic to note is the material soundness of the casting. Shrinkage cavities, blow holes, and porosity generally are very detrimental, not only to the strength, but to the ability to retain lubricant. Leakage of oil from a cast housing is not an uncommon occurrence, and lack of lubricant in the gear train is a catastrophe. (See Fig. 2-2 and 2-4.)

Before leaving the subject of basic material, it must be emphasized that relatively few failures in the field occur due to the factors listed in these three categories. The reason for their scarcity in field failures of the final product is the extensive quality inspection procedures carried out by the manufacturers of the raw products. The steel mills have excellent inspection facilities, as do the forging and casting companies. The reputable gear manufacturer also has sophisticated testing and inspection equipment and personnel capable of locating most of the basic material discrepancies. Any of these discrepancies actually causing a failure in the field is usually an isolated case.

Engineering

Engineering of gears is not only a matter of design. The entire finished gear is accomplished by the mechanical (design) engineer, with close cooperation from the materials engineer, who in turn must consult the metallurgical engineer, who must work closely with the industrial engineer. This places heavy responsibility on several persons, with the ultimate goal of a good product by the integration of design, material selection, heat treat specifications, and allowable tolerances for final grinding or finished sizes.

Design

The designer's world is unique, one that staggers the imagination with the possibilities of achievement. The following words were written many years ago by Ken Lane of Lynn, Massachusetts:

> The designer bent across his board,
> Wonderful things in his head were stored.
> And he said as he rubbed his throbbing bean,
> "How can I make this thing tough to machine?
> If this part here were only straight
> I'm sure the thing would work first rate.
> But 'twould be so easy to turn and bore
> It never would make the machinists sore.
> I better put in a right angle there
> Then watch those babies tear their hair.
> Now I'll put the holes that hold the cap
> Way down in here where they're hard to tap.
> Now this piece won't work, I'll bet a buck,
> For it can't be held in a shoe or chuck.
> It can't be drilled or it can't be ground
> In fact, the design is exceedingly sound."
> He looked again and cried—"At last—
> Success is mine, it can't even be cast!"[4]

Although the days of purely empirical design appear, for the most part, to be over, some design characteristics remain that should be noted.

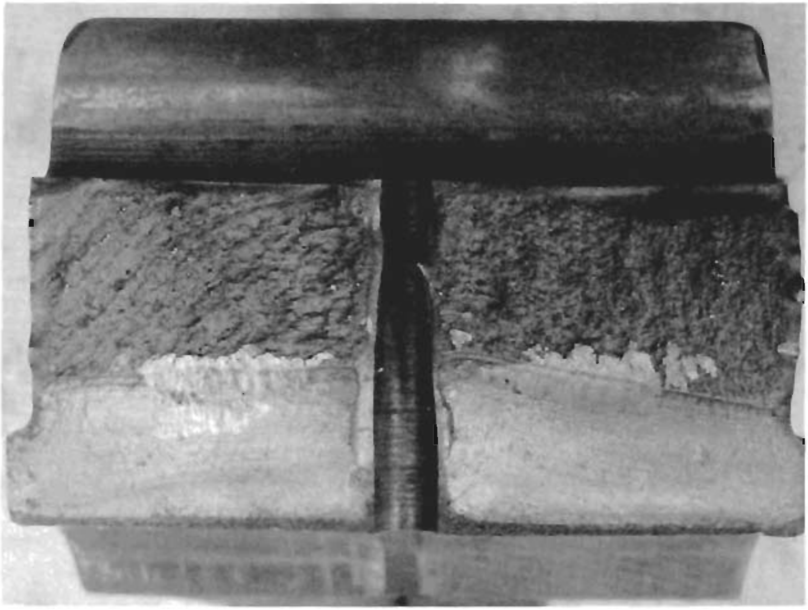


Fig. 5-9. Spur idler gear, 1×. Fatigue crack origin in the bore at each of the six oil holes.

One of the simplest gears to design is a spur gear with a round bore, rotating on a ground shaft, and operating as an idler between the power input and the power output gears. These idler gears are commonly found in planetary-type speed reducers and wheel reduction assemblies. The complex stress patterns for this type of gearing (as discussed in Chapter 1, and illustrated in Fig. 1-20) cause the bore to be highly vulnerable to fatigue. Consequently, any oil holes extending from the bore outward act as stress raisers (Fig. 5-9). This is not always the case, however, since occasionally the fatigue originates at the oil hole end that terminates at the roots of the teeth (Fig. 5-10). It has always been a question whether this type of oil hole is as beneficial as it is detrimental. It seems as though centrifugal force should always throw the oil out and away from the central bore area. The spider-arm straight bevel pinions in a differential undergo the same type of abusive stress as do the spur gears discussed above. Observation of a large number of failures points to fatigue from the bore at an oil hole as the cause in

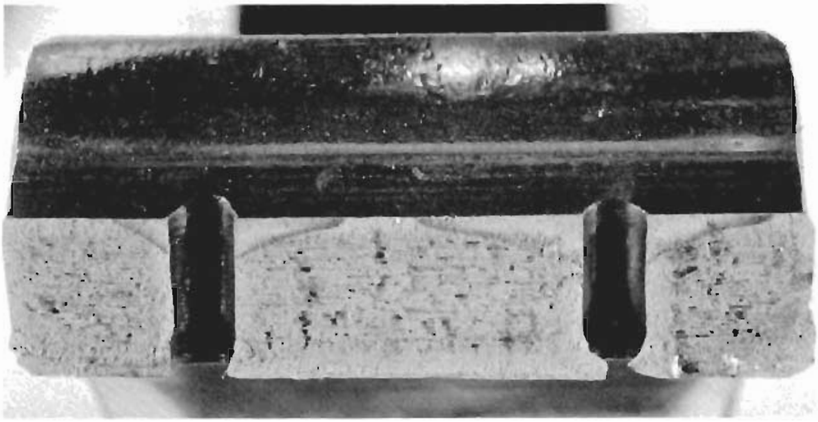


Fig. 5-10. Spur idler gear, 2.7×. Fatigue crack origin in tooth root at oil holes. Final fracture was rapid.

nearly every instance. The remaining causes would be fatigue (rapid) at the toe end of the tooth root, due to the absence of sufficient material between the root and the bore. Take a look at Fig. 5-11. Looking at a case as hopeless as this differential, the only response is to attempt to avoid similar catastrophes in future designs.

Spiral bevel drive gears have many methods of being fastened to the driven members. They may be bolted through an inner flange or fastened from the back face by a cap screw, to name two. In either case, there is the danger of a designed stress raiser. Figure 5-12 is an illustration of two weaknesses: the spiral tooth cutters were allowed to cut directly into the bolt hole, forming a sharp acute angle at a highly stressed area; and the cantilever effect of the design caused the gear to bend away from the bolts under heavy loads, adding more stress to the same bolted areas. A back-face cap screw hole may actually be designed too closely to a tooth root (Fig. 4-8), causing a fatigue failure to originate at the apex of the drilled hole.

Keyways in the bore of pinions and gears, as well as along pinion shanks and shafts, are very high-risk stress raisers (see Fig. 4-33). Any sharp corner where there is a change of dimension should be closely observed as a designed stress point.

Designers and field engineers are generally aware of the dangers of designed stress raisers in shafts, gears, bolts, and

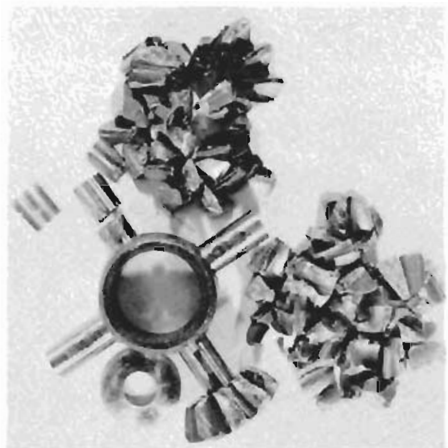


Fig. 5-11. Differential internal spider and pinions. Fatigue cracking origin at toe end of the root to bore section, which is rather thin.

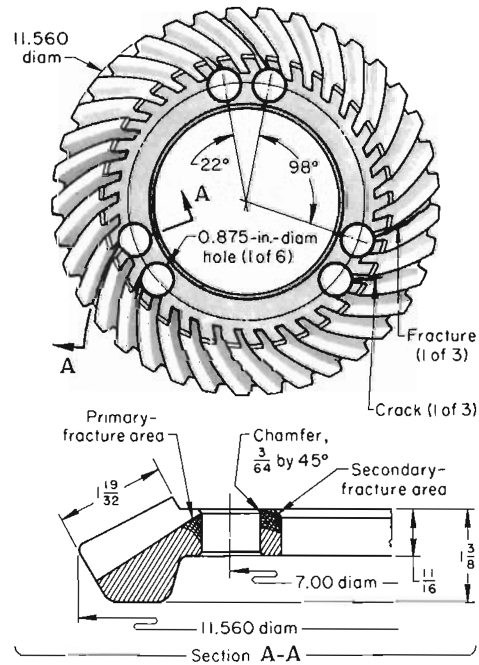


Fig. 5-12. Carburized spiral bevel gear fractured from fatigue originating at the acute angle intersection of the root fillet to the bolt hole.[5]

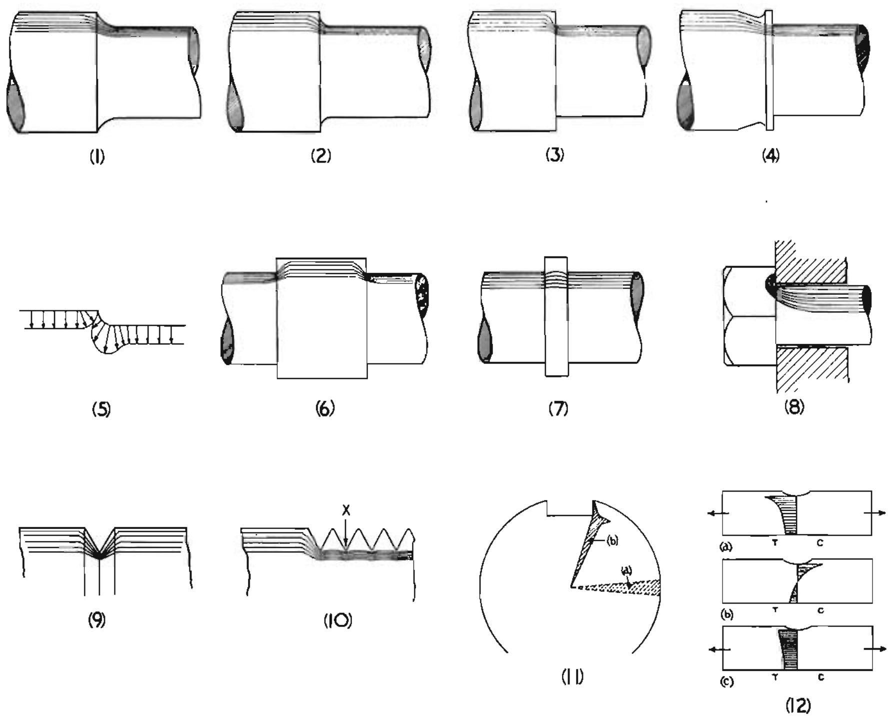


Fig. 5-13. Stress distribution at changes of section.[6]

associated parts, but a reminder of the stress distribution lines may be helpful.[6] Figure 5-13 shows many of the possibilities. Sharp corners are especially susceptible to fatigue unless modified by secondary notches or nearby corners. For instance, item 3 in Fig. 5-13 shows a high concentration of stress at the corner radius, whereas in item 4, a modified groove away from the shoulder reduced the stress concentration at the same radius. Item 7, with two close radii, shows much less concentration of stress than do the same radii farther apart, as in item 6. A notch (item 9) has a very high stress concentration, yet the notches at the base of threads (item 10) show very little concentration. There is one exception, however—the last thread maintains the highest amount of stress concentration in a line of threads. Failures originating from designed stress raisers are generally of the fatigue mode.

A very subtle design characteristic is discussed in Chapter 4, in the text illustrated in Fig. 4-16. This very narrow yet high

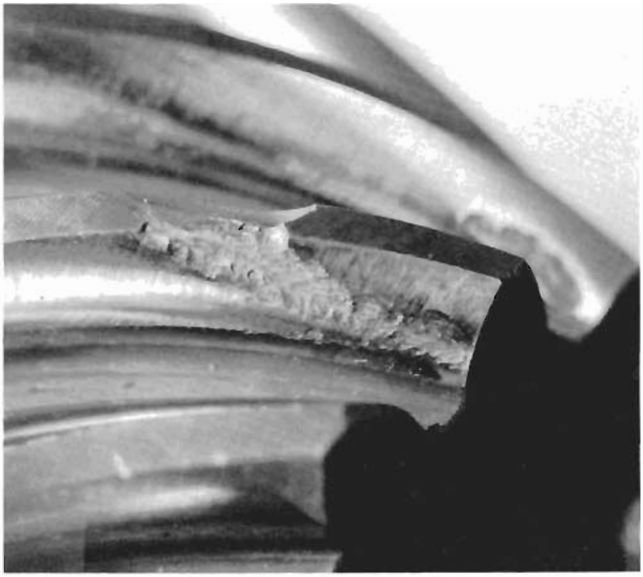


Fig. 5-14. Spiral bevel tooth, 2×. Pitting and spalling due to rolling contact fatigue in a concentrated area (see Fig. 4-16) as a designed failure.

load concentration band, which can easily be overlooked by the design engineer, is the site of rolling contact fatigue, as shown in Fig. 5-14, and is a designed failure.

Material Selection

Material selection can be accomplished only when the actual loading conditions are known. Often it occurs that loads applied to a set of gears far exceed the designed load characteristics of those gears. Therefore, the material and heat treatment (or even the design) would have been correct if the field operation were what the designers thought it would be.

In case of an overload failure, check first the specifications to determine if they have been met; but exercise caution: the specifications were arrived at arbitrarily, based on assumed knowledge. The specifications may not have been acceptable. Check to determine if the loads actually applied are normal for the operation in question or if they have been abnormally out of line. If the part has been underdesigned, wrong information may have been given to the engineers by the customer, or wrong assumptions made by the engineers. Needless to say, the reverse is also true. A part may be grossly overdesigned and a high-priced alloy steel used unnecessarily. This generally implies that a gear may have been failing due to an obscure primary cause. If the primary cause had been found and corrected, the gear would not have failed. It is the same as the old story of losing a horse because of a broken horseshoe nail. A stronger horse is not the answer. Material selection is important, but the material need not be any costlier than what is necessary to operate satisfactorily.

Mixed steel is the most common cause of material selection failure. Steel mills ship the wrong grades, forging companies pick up and use material from the wrong stack of billets, and gear manufacturers use forgings from the wrong lot—all unintentional, but careless and costly.

Probably the most discouraging instance, and yet very common practice, is for a user to repair his own broken gear. New teeth are built up by welding and then hand-ground to shape. Usually a piece of steel—any piece of steel—is picked up and given to a gear shop to cut teeth. As an example (Fig. 5-15) a C1019 annealed stock was used to replace an SAE 8620 carburized alloy steel gear. The teeth did not break out of the re-

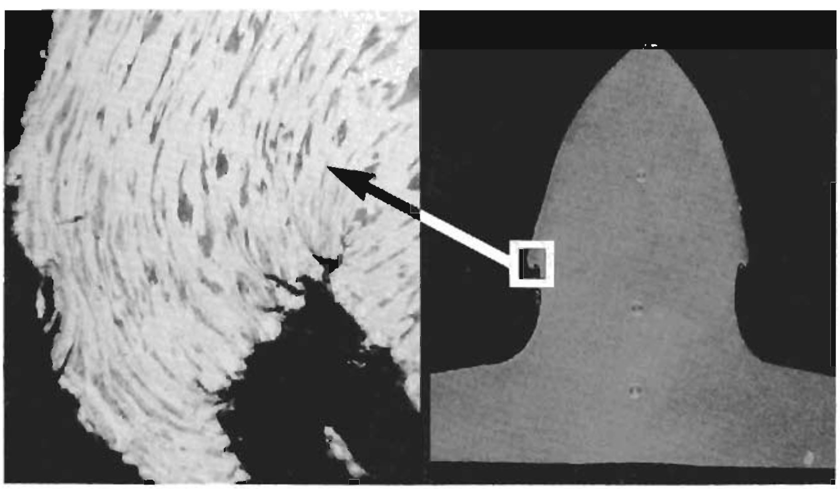


Fig. 5-15. A gear made from C1019 carbon steel (annealed) used to replace an SAE 8620 alloy steel (carburized).

placement gear (as did the teeth of the alloy steel gear); they simply wore out in a very short time.

Heat Treatment Specifications

The heat treating operations are generally the sole responsibility of the gear manufacturer, as long as the end result meets the metallurgical requirements agreed to by the customer and the manufacturer. The heat treat specifications (i.e., the specifications to the heat treat department) must be established to meet those requirements. The established specifications are based on the material used and the capability of the heat treating equipment. If a specification to the heat treat department is incorrect, it is due often to poorly relayed information or to poor judgment on the part of those making the specification. Not following the specification is not the fault of the specification.

Grinding Tolerances

The general means of obtaining a final dimension of a part, after carburizing and hardening, is through grinding. Grinding removes the surface of a carburized area, which reduces the depth of a hardened case. The judgment of the industrial engineer concerning the allowable amount of grinding stock to provide, and the judgment of the metallurgist to compensate with an additional amount of carburized case, are a very critical matter. This is true especially at a shoulder radius (as in Fig. 5-16) where bending fatigue may originate due to insufficient case.

Manufacturing

After the design stage, when all the specifications have been established, comes the manufacture of the parts. The quality of each part depends on the skill of each person participating in its manufacture and the capabilities of the equipment used to produce the part.

If the reader observes that the inspection department has been left out, he is right. Quality is never inspected into a part. Inspection gets the "klunkers" out. There is an excellent pro-

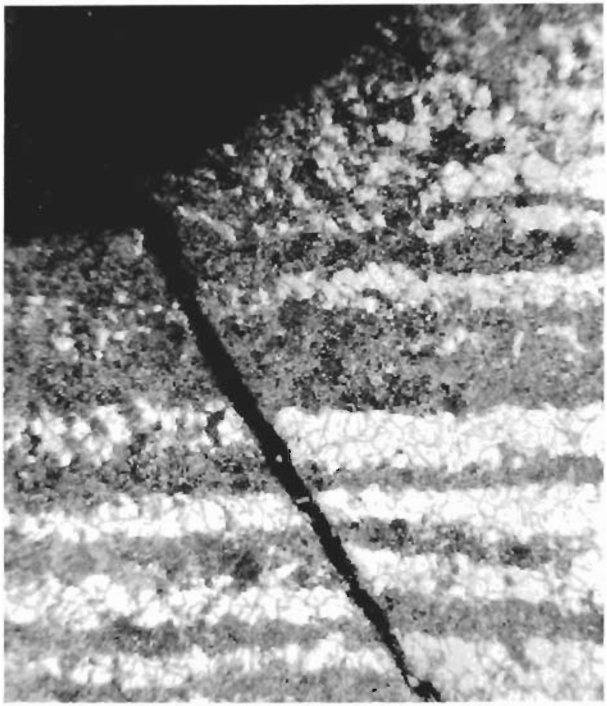


Fig. 5-16. Shoulder radius next to a bearing diameter. Carburized case had been removed by grinding, allowing a fatigue failure in bending due to low strength of surface material.

Fig. 5-17. Rapid fatigue at the base thread of a differential holding bolt (grade 8).

gram, however, under the jurisdiction of the inspection (quality assurance) department that should be utilized to its fullest potential. Called Statistical Process Control, it is a plan to develop a statistical means of continually monitoring results of a manufacturing unit, to recognize when the results are getting out of control, and to correct the unit before it starts to turn out the klunkers. This would be a positive approach for inspectors.

People and machines manufacture innumerable high-quality gears capable of operating for many years. Yet, concerning field failures, there are several characteristics that point back to a manufacturing procedure.

Tool Undercutting, Sharp Notches

Although a shoulder radius at a change of section is a designed function (see Fig. 5-13) many fatigue failures at a radius cannot be attributed to the design. All too often a cutting tool had not been properly formed to cut the designed radius and as a result, a sharp corner, a smaller radius, or an undercut was formed. This is a manufactured stress raiser that may readily lead to bending fatigue.

Figure 5-17 shows rapid fatigue originating around the base thread of a differential holding bolt. Figure 4-32 illustrates bending fatigue that started at a sharp shoulder radius, and Fig. 4-33 shows what can happen at a sharp corner of a keyway.

As a graphic example of machining incapabilities, Fig. 5-18 should be observed. Each spline had been individually cut with a small vertical milling cutter that was moved from the open end to the shoulder. In order to "square" the rounded corners, a hammer and chisel were used as a swage. This resulted in the squared corners being cracked, and field fatigue failure took place soon after.

From a visual observation, it would seem that all tooling cutter marks and all tooling undercuts are detrimental. While this is not necessarily so, generally it is good manufacturing practice to produce a part free from cutter marks, with all radii surfaces smooth and continuous.

Perhaps the one gear tooth characteristic that has been held responsible for many gear failures is that of roughing cutter

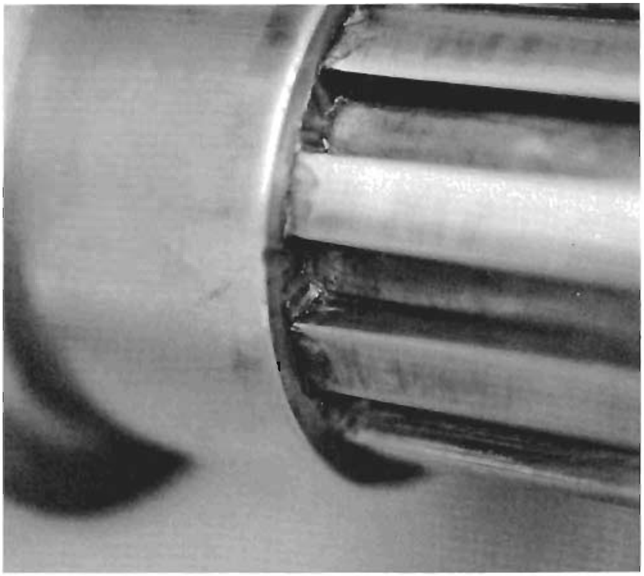


Fig. 5-18. Each individual spline had been formed with a vertical milling cutter. The rounded area at the shoulder was swaged into square corners with a hammer and chisel. Each corner was precracked, which resulted in a fatigue failure.

marks in the center of the root fillet. Oddly enough, tooth bending fatigue failures do not occur at the roughing cutter marks but at the radius just below the active profile. A photoelastic study was made to determine the actual effect of the cutter mark upon the tooth bending strength. Figure 5-19 shows the photoelastic stress pattern on a tooth with a 0.022-in.-deep cutter mark in the root center being loaded by a tooth with no cutter mark. The radii of both teeth are the same and were loaded to a capacity equivalent to 6.5 stress fringes. At the same time, the small radius of the cutter mark had a stress equivalent of 6.0 fringes. The fringe pattern was the same when the mark cut to a depth of 0.017 and 0.010 in. In each instance, the cutter mark radius was retaining a stress concentration amounting to 92% of what was retained by the active root fillet. However, whenever a deep cutter mark coincides with the radius, a notch effect is expected.

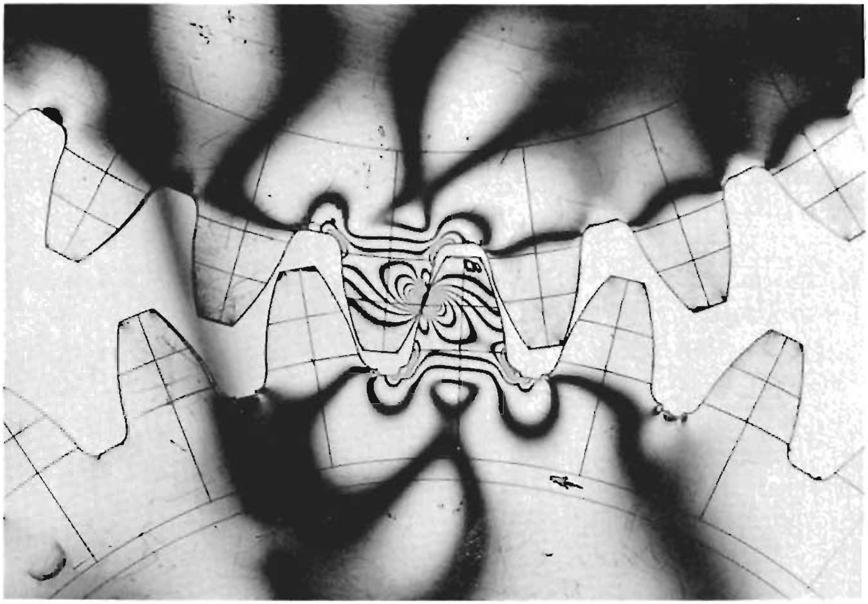


Fig. 5-19. Photoelastic study of the effect of a roughing cutter mark on tooth bending strength. The rounded fillets were loaded to an equivalent of 6.5 stress fringes. The cutter mark (0.022-in. deep) measured 6.0 stress fringes at its radius. A stress concentration equivalent to 92% of the applied load.

Tooth Characteristics

Tooth characteristics (as discussed in Chapter 1) have a marked influence on the concentration of load per unit area, which may be accentuated if a characteristic changes from the norm. Note the following examples, using illustrations from previous chapters:

(*a*) Figure 1-12(c) illustrates a part with severe runout. Transpose that severe runout with outboard taper to an actual part, and gross frictional heat may occur, as in Fig. 5-20.

(*b*) Refer to Fig. 1-16. This was a gear that had good lead and no taper. However, it was mated with the pinion that had a severe taper, as shown in Fig. 1-15(d).

Fig. 5-20. Double helical gear. Severe frictional heat developed due to runout and taper in the gear. The heated section had less than zero backlash.

(c) Refer to Fig. 1-17. This internal gear shows indications of case-crushing, because the radius of curvature of the impinging tooth caused a concentrated centerload, as shown in Fig. 1-15(f).

(d) Looking especially for surface wear patterns on spiral bevel or straight bevel teeth, observe very closely the position and magnitude of these patterns. Use Fig. 1-18 extensively as a guide to what may be occurring.

(e) When the involute pattern shows a swelling of either the lower profile edge or the top corner, tooth tip interference may be causing small in-line pitting low on the active profile. This may be an indication of something catastrophic (as demonstrated by the cyclone effect in Fig. 4-23 and Fig. 5-21).

Grinding Checks, Burns

Grinding checks characteristically are aligned perpendicular to the grinding direction and are sometimes joined to form networks. They are generally in groupings, are close together, and are not very deep. Severe grinding may also highly temper or even completely anneal a portion of the surface. (The technique of examining for burns is discussed in Chapter 3 in the section titled "Nital Etching"; the technique for examining checks is discussed in the section titled "Magnetic-Particle Inspection" in the same chapter.) Grinding generally leaves a surface in a state of residual tensile stress. If the magnitude of this residual stress is high, an additional applied stress may cause external rupture (as in Fig. 4-50 and 4-51). Profile grinding of gear teeth is a very critical operation. It has generally been accepted by the gearing industry that gear tooth surfaces should be in a state of compressive residual stresses. Grinding will either reduce the compressive stresses toward zero or reverse them to a tensile stress. Grinding may also leave soft spots on the teeth (Fig. 3-4).

Heat Treatment Changes

Dimensional change that occurs due to the heat treating process can be expected, and yet can be compensated for in

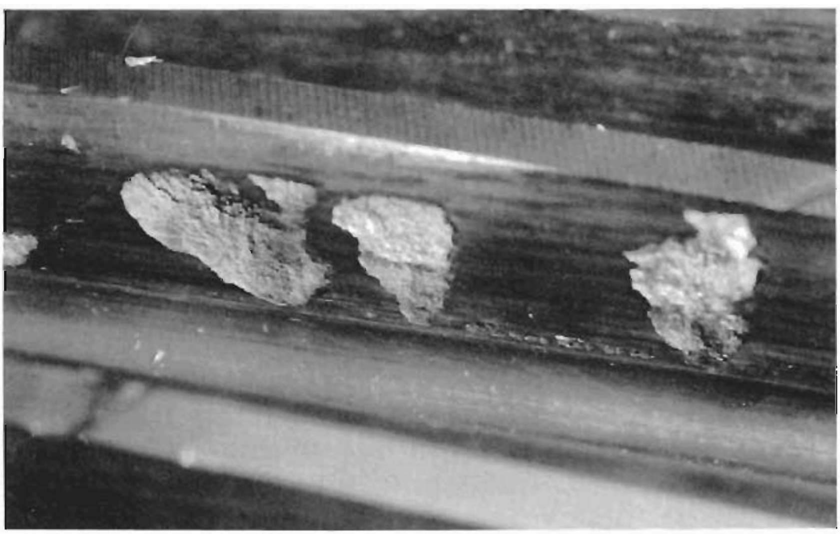


Fig. 5-21. Tooth tip interference causing in-line pitting low on the active profile. A spalling action has resulted.

prior machining practice or in the heat treating process. Four factors contribute heavily to the dimensional changes: the heat treating process, the design of the part, the mass of the part, and the material. The first three factors are consistent from lot to lot, but the material usually changes with every lot. Each grade reacts differently, and each heat within a grade may react differently. Therefore, changes can be expected with each new heat of material being processed; and if necessary, sample parts should precede the production lot.

The most harmful change that occurs is the change of tooth characteristics. A self-imposed crown is discussed in Chapter 4, in the text illustrated in Fig. 4-6. Especially susceptible are spiral bevel teeth that may lead to any of the patterns shown by Fig. 1-18. These problematic changes of tooth characteristics are not insurmountable, since a few parts, as a sample lot through the heat treating processes, can be used to determine the final cutting of the teeth as compensation.

Heat Treatment

Although heat treatment is a manufacturing process, it drastically changes the chemical and metallurgical characteristics to the extent that a separate discussion is necessary.

Case Properties

The carburized case of a gear is specified to effective depth, surface hardness, and (sometimes) percent of carbon at the surface. Unless otherwise stated, this specification is applicable to the midprofile of the teeth midway from the ends. (See Fig. 3-21).

The properties of the case are important both at the midprofile and at the root radius of the gear tooth, but for different reasons. John Halgren of International Harvester Company made an in-depth study of subsurface stresses and the hardness required to prevent subsurface failures. This work was published in the SAE Journal of March 1954, and is reproduced in Fig. 5-22. The curves are designed for a 7 D.P. spur gear, but the

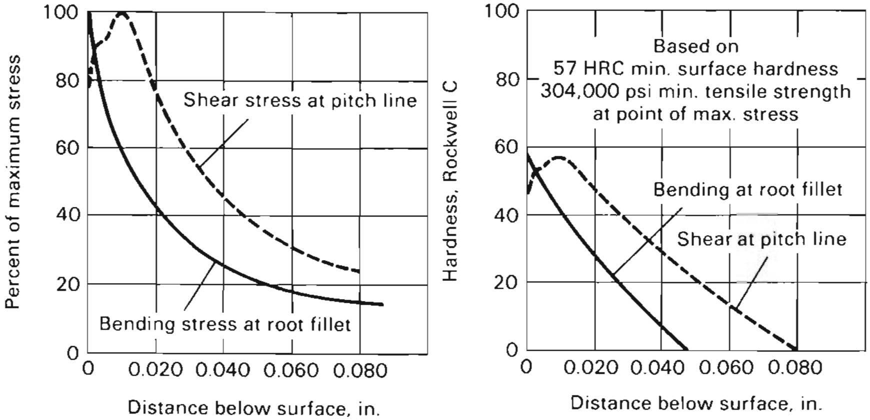


Fig. 5-22. Subsurface stresses (left) and the hardness gradients necessary to prevent subsurface failure (right); shown for a 7-pitch spur gear.

application can be made to any other tooth type. Note step by step the interpretation of the illustration:

(*a*) The shear stress is dominant at the pitchline, is 100% (or maximum) at 0.010 in. from the surface, and is about 80% at the surface.
(*b*) The bending stress is dominant at the root radius, is tensile, is 100% (maximum) at the surface, and falls off rather rapidly away from the surface.
(*c*) Based on 304,000 psi tensile strength required to prevent failure, the hardness conversion is 57 HRC.
(*d*) Using 57 HRC as a measurement of the maximum stress point, the hardness gradient requirement of the pitchline starts at about 48 HRC at the surface, reaches 57 HRC at 0.010 in., and passes through the effective depth of 50 HRC at 0.020 in. from the surface.
(*e*) At the root radius, 57 HRC must begin at the surface, but drops off abruptly to 50 HRC at about 0.005 in.

With this information, case properties can be specified as follows:

(*a*) Surface hardness: The hardness must be 58 HRC or higher.
(*b*) Case depth: The effective depth (to 50 HRC) must be at

least 0.020 in. from the surface. It must also be deep enough to ensure that 58 HRC is continued beyond the 0.010-in. maximum stress point. A range of 0.010 in. effective case depth is allowable for this size of a tooth; so a case depth of 0.030-0.040 in. is justified.

(c) The depth specified above is for the pitchline or mid-profile. An examiner should determine if this will be sufficient for the root radius. In most cases, gear manufacturers expect the root radius case depth to be 60% of the pitchline case depth or more. Based upon the curves of Fig. 5-22, one customer will accept a root radius case depth of 50% of the minimum pitchline requirement. If this is the case, would a specified root radius depth of 0.015-0.040 in. be acceptable? According to the illustrated curve, the minimum effective depth required for strength is only 0.005 in., and therefore, it is acceptable.

The above dissertation was for a 7 D.P. spur gear operating under a required strength of 304,000 psi maximum. Table 5-1 is a listing of general case depth standards for each diametral pitch. Any published standard for case depth is arbitrary; that is, based upon common knowledge of normal loading applications. Modifications of this standard and of any other standard must be made after considering the design of the part, the material used, the load applications, the areas receiving maximum stress, and the types of stresses applied. In examining a field

Table 5-1. Recommended effective case depth standards (a)

Diametral pitch	Case depth standards, in. Spur-helical	Bevels	Diametral pitch	Case depth standards, in. Spur-helical	Bevels
20	....0.015–0.020	(b)	7	....0.035–0.045	0.025–0.035
18	....0.015–0.020	(b)	6	....0.040–0.050	0.030–0.040
16	....0.015–0.022	(b)	5	....0.045–0.055	0.035–0.045
14	....0.018–0.025	(b)	4	....0.050–0.065	0.045–0.060
12	....0.020–0.028	(b)	3	....0.060–0.075	0.050–0.065
10	....0.025–0.033	0.015–0.020	2	....0.075–0.095	0.060–0.080
9	....0.027–0.035	0.015–0.020	1	....0.090–0.110	0.070–0.090
8	....0.030–0.040	0.020–0.030			

(a) This is only a guide, subject to modification depending on design and application.
(b) Consult metallurgical department.

failure, an analyst should be as specific as possible regarding the relationship between the cause and the case depth specification. He must also be aware of additional factors that may be involved. For example, a shallow case might not have failed if it had been within the range of the specification. Possibilities such as this must not be overlooked.

Properties of the case also include other than those already specified, such as the microstructure. A fully martensitic case is the goal for any heat treat. The tensile properties and wearing ability are very good. An amount of retained austenite that does not reduce the surface hardness is not detrimental; in fact, it may be helpful, because it is more ductile than martensite and will absorb some energy of movement. Enough austenite to reduce the surface hardness may cause weakness under a bending stress, as well as reduce rolling-contact resistance. However, austenite has the ability to transform to martensite under exerted pressures called cold working. This is a favorable characteristic since it generates a harder wear surface while in use; and, since martensite occupies a larger volume than did the original austenite, the surface stresses become more compressive. A fine line of differentiation exists at this point because surface compressive stress is necessary to resist tensile bending or tensile surface fatigue; but a high compressive stress can also set up conditions favorable to compressive surface fatigue and pitting.

Cementite (or iron carbides) close to the surface of the case can range from very fine spheroids widely scattered, to massive globules, to a heavy network. The scattered spheroids are usually welcome because they enhance the wearing resistance of the surface. However, the massive and network types may cause problems under certain conditions. For instance, Fig. 5-23 shows a carbide network near the surface that caused an intergranular weakness, which failed under load by crumbling. A bainitic structure at or near the surface may not cause undue wear or surface fatigue problems, but it does lower the bending strength of the root radius. Decarburization of the surface is not favorable at any time. Black grain oxides along the surface are present at all times in carburized steel parts and are, for all practical purposes, ignored.

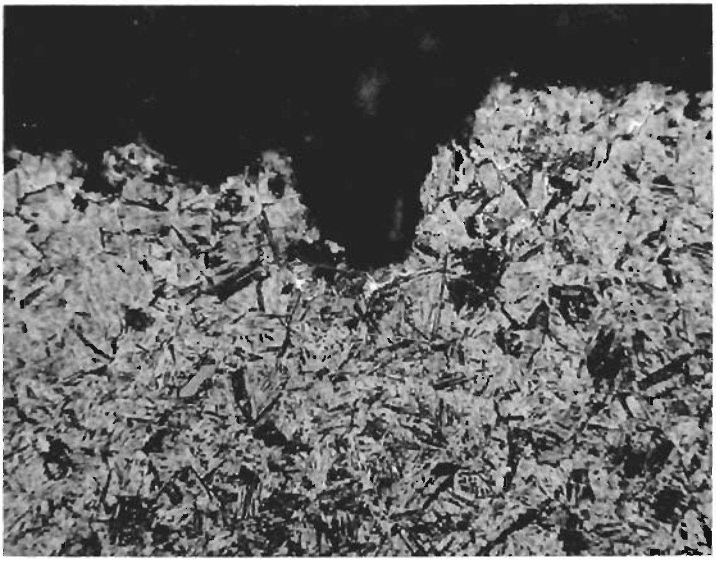


Fig. 5-23. Carburized surface that crumbled under load along the carbide network boundaries, 200×.

It is difficult to point to any of the microconstituents of the carburized case as the primary cause of a failure. Each undoubtedly contributes to the timing of a fracture or to the mode of failure; but when a catastrophic failure takes place, an examiner does not usually look for a microscopic cause.

Core Properties

The core of a gear tooth is the basic material under the carburized case that maintains the initial percentage of carbon. It may have been quenched and tempered, or it may remain as annealed, depending on the heat treating processes used.

The required properties of the core are few. The core must be ductile enough to absorb shock loads without fracturing, and hard enough to be resilient under impact and to withstand the compressive applied loads without permanent deformation. The structure may be ferrite and pearlite, as rolled, or annealed; fine pearlite as heated and slow cooled; or low-carbon martensite as quenched and tempered. The ultimate strength of any core is measured by its hardness (see Table 2 in Chapter 3).

Case/Core Combination

Failures in the core of a gear tooth are not common; therefore, the case and core should be considered together in any analysis. Two major modes of failure in gear teeth point directly to the case/core combination.

The case crushing illustrated in Fig. 4-40 vividly depicts the depth of the carburized case and the strength of the core as two contributing factors. Typical results are shown in Fig. 4-41 and 4-42.

Internal rupture of a carburized tooth is a direct result of the accumulated residual tensile stresses exceeding the strength and ductility of the material at the case/core interface, near the top of the tooth or near a top corner at one end. (See Fig. 4-48 and 5-24.) This condition is aggravated by four factors. Each factor by itself may be beneficial, but, when these factors exist together, they can be detrimental:

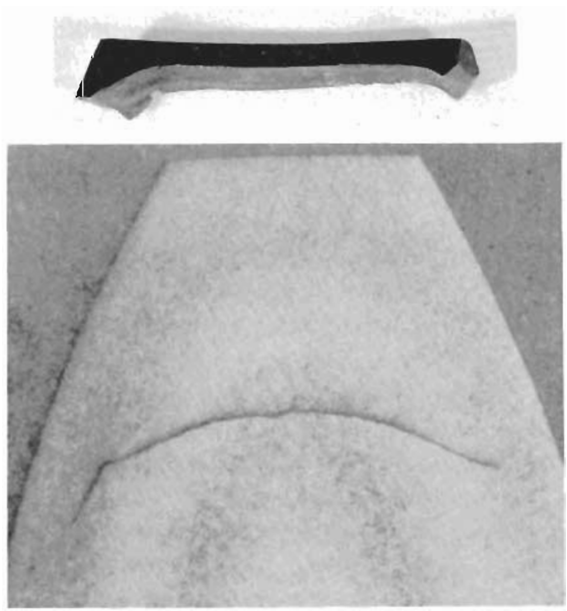


Fig. 5-24. Internal stress rupture. Origin at case/core interface near top of tooth. In extreme cases, the entire tooth top may pop off.

Deep case depth. Since a martensitic case assumes a larger volume than the precarburized material, the case gets larger than the original surface and tends to pull away from the core.

High core hardness. If the core hardenability is low, the core may be ductile enough to "stretch" with the case and thus absorb the energy of the expanding case. However, a high-hardenability core with high strength does not move. Therefore, a large amount of tensile stress is added to the first amount at the case/core interface.

Subzero treatment. If austenite were retained within the case, it would be apt to absorb much of the residual stress because of its high amount of ductility. However, subzero treatment will transform all of the austenite within the case to martensite. More volume is required by this transformation, and only one place remains for the added residual energy to go: on top of the other stress at the case/core interface.

Shot peening is an operation used specifically to increase compressive stress at the surface (usually of the root radius), which enhances resistance to bending fatigue; but, at the same time, shot peening hits all other surfaces. The tooth surfaces already high in compression are hammered into greater compression, which is counteracted by increased tension at the case/core interface.

Thus with all four factors applied, the tops of the teeth have been known to explode like popcorn while merely sitting on a shelf. It is quite likely, however, that the heat treating management, recognizing the possibilities suggested above, will certainly specify a tempering operation; the operation, performed as quickly as possible after the hardening and the subzero treatments, may relieve these residual stresses.

Another mode of failure often associated with a case/core combination is the longitudinal torsional shear of a shaft. In general the shear strength of the core is lower than the tensile strength of the case. As a result, the shear-fatigue crack will originate at or near the case/core transition zone and may progress quite far before the case fails in tension (see Fig. 4-28).

Hardening

Hardening (as opposed to carburizing and hardening) assumes the use of a medium- or high-carbon plain or alloy steel that will harden consistently throughout its cross section, and the strength generated at the surface will be maintained throughout. Concerning gears, the application may be limited. Tooth hardness may range from 40 HRC to 60 HRC, but teeth at lower hardnesses may be subject to wear and those at higher hardnesses may be somewhat brittle. Perhaps the dominant through-hardening cause of failure is quench cracking. Usually quench cracking occurs in such magnitude that it is noted immediately, but, if it is suspected in a finished part, a metallurgical examination of a cross section through the crack may be expedient. A quench crack is distinguishable from a forging lap because the fracture surface of a quench crack is homogenous and free from scale or oxides.

Selective hardening (i.e., producing a case by induction hardening) is a very successful way of utilizing the hardenability of a medium-carbon steel to produce core strength, as well as producing a fairly hard case to withstand wear and maintain bending strength. The procedure is to uniformly heat, quench, and temper a forging or a gear blank to a machinable hardness of 30-35 HRC; finish-machine and cut the teeth; and induction harden the tooth surface to 48-55 HRC. Two basic methods of induction hardening gear teeth are accomplished by placing a circular coil around the major diameter of the gear and heating all teeth simultaneously, and by traversing one tooth space at a time. The first method hardens the entire tooth and should penetrate below the root. The second method hardens only the tooth and root profile and has the effect of a hardened case (see Fig. 3-17).

For the induction hardened case to be a cause of failure, quality must be deficient. For instance, a spur pinion had a tooth bending fatigue failure originating at the root fillet at the end face (Fig. 5-25). Etching of the end face revealed the origin to be at the terminus of the induction pattern. The teeth had been through hardened but had not been hardened beyond the tooth root. As a second example, a large spur gear was failing by

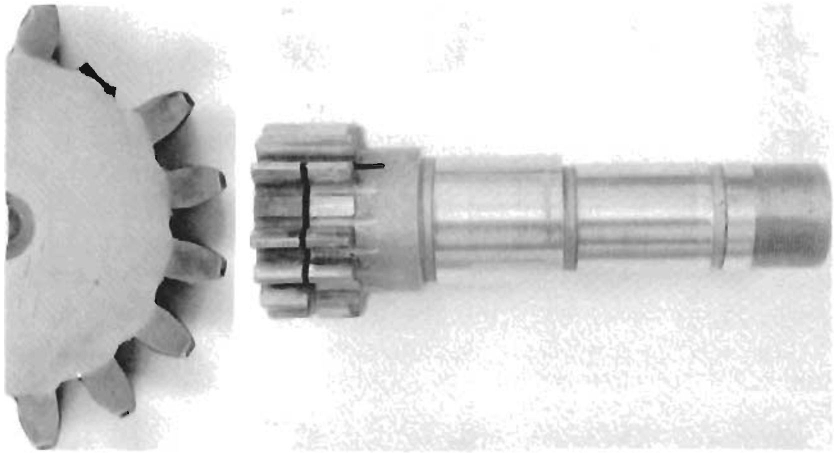


Fig. 5-25. Spur pinion induction hardened by using a single round coil. Hardened area terminus at the root fillet end face set up a stress notch that invited failure by tooth bending fatigue.

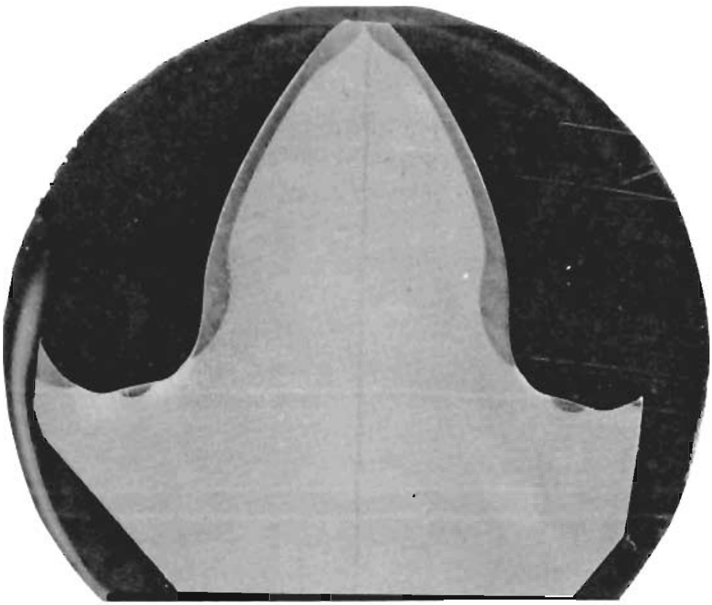


Fig. 5-26. Spur tooth induction hardened by scanning one tooth space at a time. Inductor scanning speed did not allow time for heat to penetrate the root radius. A stress notch resulted which invited failure by tooth bending fatigue.

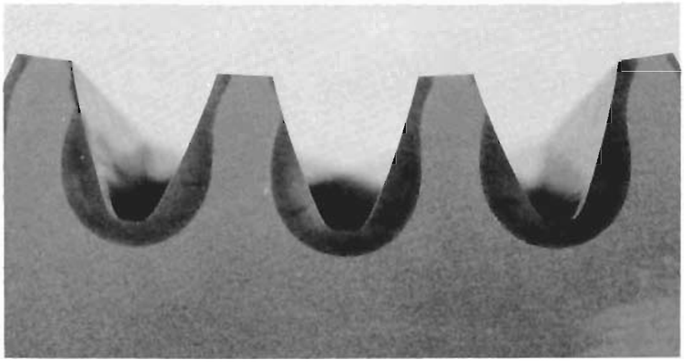


Fig. 5-27. Internal spur gear induction hardened by scanning one tooth space at a time. The inductor was misaligned, allowing for a nonhardened addendum on one tooth profile. This soft area was easily worn.

tooth bending fatigue at the root radius. Etching showed a surface hardness pattern along the entire profile, but it had terminated at the root fillet (Fig. 5-26).

Some analysts claim that induction hardening is for tooth wear only and that hardening of the root is not necessary. Other analysts differ very strongly. Observations verify that most failures, not only of bending fatigue, but of tooth surface fatigue, are associated with the hardened zone terminus. With this observation in mind, a university study was conducted to determine the residual stress pattern at the induction hardened case/core transition zone. The result was the establishment of the fact that a high peak of residual tensile stress occurs at the hardened case terminus. This condition becomes a self-induced metallurgical notch, potentially a point of fatigue origin.

Improper induction hardening may be the cause for other problems, such as excessive wear along a nonhardened addendum (Fig. 5-27), or tooth bending fatigue from a quench crack at the open end of a tooth root radius or from inductor burns at the tooth root (Fig. 5-28).

Selective hardening also includes flame hardening. The problems involved with this technique are more flagrant than with induction hardening, since flame hardening of teeth is more difficult to control. Nonuniform hardened areas and overheated corners may be the most common types of problems to occur when flame hardening is utilized.

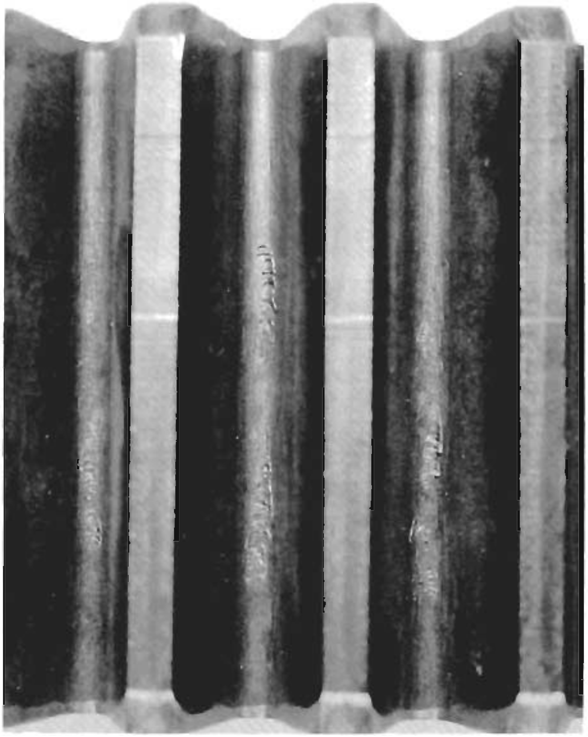


Fig. 5-28. A tooth space inductor set too close to the root caused severe burning and melted material at the root centers of several teeth.

Tempering

Tempering is the process of subjecting hardened gears to a low-temperature reheat, designed to reduce the surface hardness slightly and to reduce the residual stresses caused by quenching. Tempering of gears, after hardening or subzero treatment, places the causes mostly in one of two categories, too much or too little. If a part remains too hard (too little tempering), it may be subject to brittle fractures. If a part is too soft (too much tempering), it may be subject to surface wearing problems or tooth bending fatigue.

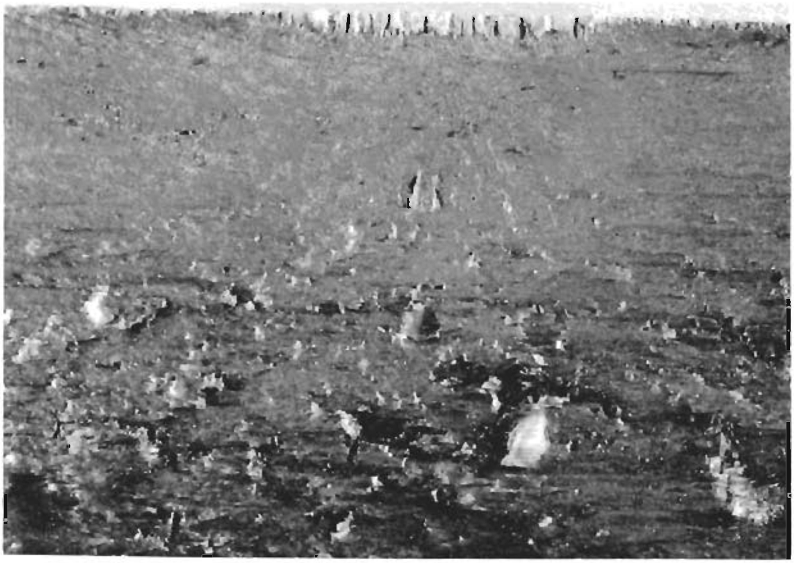
When certain alloy steels are heated to a temperature between 500 °F and 700 °F, there is an increase in strength, but a very marked decrease in ductility and impact strength. This embrittlement is called "blue brittleness" because it occurs in the tempering range that produces a blue color at the surface. (There is an excellent discussion of embrittlement in the new *Metals Handbook*.[7]) Not all alloy steels are susceptible to this phenomenon, but those that are should not be subjected to this temperature range when impact loads are expected. For information on types of steels that are susceptible, consult a good materials handbook on impact strength vs tempering temperature.[8]

Two practical aspects of the occurrence of temper brittleness must be considered as the causes of a failure. The first aspect is field oriented and has nothing to do with the heat treatment process. It will occur when a part or a portion of a part is heated to approximately 600 °F either by frictional heat or by close proximity to an area being welded. Whenever an area, regardless of how small, is heated to a very high temperature, a transition zone between the heated spot and the cold material actually reaches a tempering temperature of 600 °F. In some steels, this zone is susceptible to impact fracture. The second aspect definitely relates to the heat treat department and may be caused by faulty heat treat specifications or faulty tempering. This embrittlement is found only when a portion of a gear or pinion (namely, the threaded end of a shank) is tempered at about 1250 °F for the purpose of reducing the hardness of the threaded area. Next to the threaded area, there is usually a shoulder leading to a larger diameter and a splined section. If the tempering medium (perhaps molten lead) is not applied beyond this shoulder, the transition temperature of approximately 600 °F might possibly temper the shoulder radius. For some alloy steels, this shoulder radius area would now exhibit low impact strength. This could cause the threaded end to break off with a brittle type fracture if the thrust or bending load were great enough.

Miscellaneous Operations

Copperplating is used extensively in the industry as a stop-off for areas not to be carburized. If a spot of copperplate were

(a)



(b)

(a) 1.6×. The proximity of a long inclusion that was not a factor in this failure. (b) 3.5×. The columnar structure is the result of ram pressure from a straightening press. The crushing effect of the case into the core formed a weakened case/core interface which became the origin for sheer fatigue.

Fig. 5-29. Torsional fatigue of a shaft in longitudinal shear. Origin was at the case/core interface beneath the concentrated columnar structure of the case.

inadvertently allowed to remain in a critical stressed area (such as the root fillet of a gear tooth), a soft spot exhibiting low fatigue strength would result.

Copper stripping solutions used to remove copper plating after the heat treat processes can be corrosive if not maintained properly, or if the parts are left in too long.

Threaded plugs placed in bolt holes before carburizing may be placed improperly, allowing some type of stress in the area around the bolt hole.

Straightening of pinion shanks and shafts has several unique hazards. Surface cracking may be the common attributing cause; however, Fig. 5-29 illustrates another type of stress raiser. The fractured area of the shaft shows a longitudinal shear with its origin below an odd-looking columnar case structure and very near to a long inclusion. As expected, the customer claimed the inclusion was the cause. However, the real source of the problem was the columnar case structure caused by crushing and vertical shearing from the ram of a straightening press, which determined an excellent starting place at the weakened case/core interface for torsional shear. Often a bending fatigue failure of a shank or shaft occurs, with no apparent reason for its peculiar point of origin. To accomplish a straightening operation, the part must be plastically bent. (See Fig. 5-30a.) Also, upon the release of the bending load, a pattern of residual tensile stresses is measurable at the top and residual compressive stresses measurable at the bottom, as illustrated in Fig. 5-30(b).[9] Therefore, if bending stresses are applied in the field operation, the top of the straightened area will receive the maximum allowable stress before any other spot on the shaft, and bending fatigue will probably start at that area.

Service Application

Dr. Charles Lipson was quoted as saying: "The fact is that the vast majority of service failures cannot be traced to the metal-

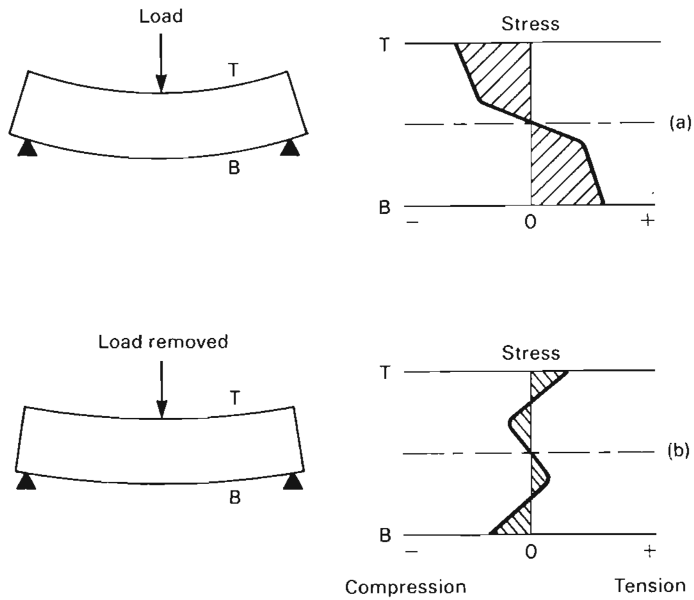


Fig. 5-30. Residual stresses in a shaft due to plastic bending during a straightening operation. (a) Stress distribution during plastic bending. (b) Residual stress measured after load removal.[9]

lurgist. Instead, they are the responsibilities of the designer, the installation man, and the service operator."[10] Richard Gaydos of Republic Steel, in his paper "Failures in Heavy Machinery," made this comment: "A historical review of many failures in a single plant over a span of many years illustrates the following reasons: operational, 36% (overload, 28%); faulty design, 34% (sharp radii, 11%); shop malpractice, 17%; and metallurgical, 13%."

Now, why, since service applications appear to be the greatest cause for gear failures, has so much energy been spent explaining all of the other reasons first? Perhaps Robert P. Haviland has the answer: "A failure point in a perfect object occurs when the energy stored by a given mechanism exceeds some critical value." He then explains, "This means the break is going to occur at the weakest link."[11] The discussion up to this point has been setting the stage for a discussion about weak links. Any service application failure is going to occur at the weakest link. It is necessary first to have an understanding of what caused those areas of weakness. Also, any failure at the weakest point within the gear train is not caused by that weakest point, if the load applied had been greater than the designed load-carrying capacity of the assembly or of the failed part. Conversely, if failure at a specific area continues as a pattern and, if the applied loads are normal for that application, the area of failure must be strengthened.

The causes of failure relating to service application will be discussed, not in order of importance or magnitude, but in order of operational sequence.

Set Matching

Spiral bevel and straight bevel gear and pinion sets are matched by tooth contact pattern and serialized as sets. Although the tooth contact pattern is generally arbitrated by the customer and vendor, there is a customarily acceptable no-load pattern (see Fig. 1-18a). Any deviation from this pattern may be caused by misassembly, misalignment, or deflections. The deviated patterns (illustrated by Fig. 1-18 c through p) can be recognized in the field by the wear pattern of the gear teeth on both sides. Another very common cause of a deviated pattern is a mismatching of sets. Very often a set is returned with different serial numbers on the pinion and gear. Mismated parts should never be in operation together.

Assembly, Alignment, Deflection, and Vibration

In most instances, the characteristics in an assembly that cause misalignment and deflections are relative to dimension and strength. However, the engineering of the entire assembly may have some shortcomings. For example, a spiral bevel drive set was placed in a differential assembly engineered for a right-hand drive. The manufacturer had tested and matched the set for a right-hand load but had never paid attention to the loading pattern on the reverse side, which is normal procedure for standard shelf items purchased from parts suppliers. The customer did not know the importance of this fact and placed two units in operation on a special double-drive machine he had designed.

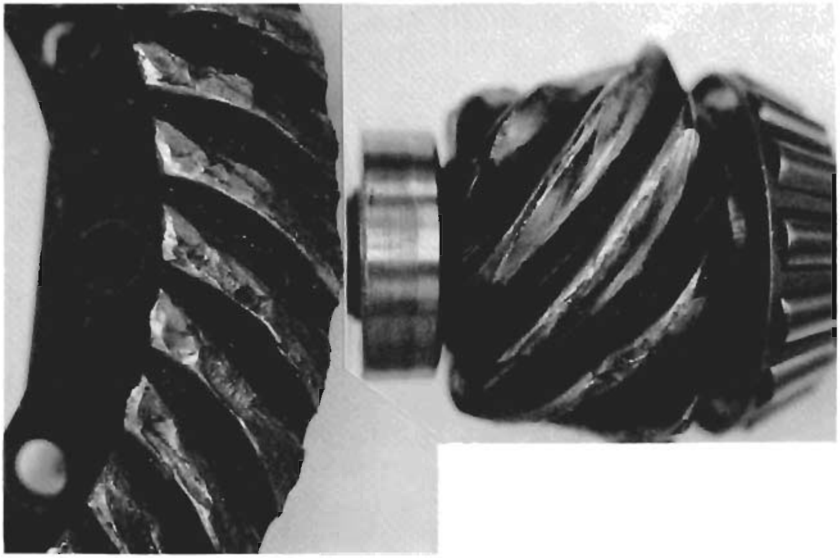


Fig. 5-31. Spiral bevel pinion (0.75×) and gear (0.70×) from a stock item differential manufactured properly for and adjusted to a right-hand drive. It was installed in a left-hand drive situation. The reverse-side pattern had not been corrected for this application.

As it happened, the left-hand unit, driving in the reverse direction, had a pinion in its assembly that had never been adjusted to this type of operation. The result is apparent in Fig. 5-31.

A fast-growing industry of renovating and reconditioning exists in the present economy. In fact, it is possible to purchase a reconditioned gear box for nearly every type of original box that has been manufactured. In order to compete with this new industry, many manufacturers of original equipment have been forced to renovate and to resell used assemblies. This is a scary practice and could have disastrous results unless every internal part has been closely inspected dimensionally, observed by magnetic-particle examination, and kept meticulously clean. The correct alignment of parts is very important. A misalignment can cause tooth loading to shift from the central profile contact to an end contact. Whenever shifting of load pattern occurs through misalignment, it is recognizable because the patterns are consistent, continuous, and uniform. But this is not the case with shifting contact patterns caused by deflections.

These patterns constantly shift back and forth as loads are increased, decreased, or reversed. In one example (shown in Fig. 4-5), a sudden overload caused by an extreme windup of all internal parts placed an immediate end contact load on one tooth of a pinion, which cracked at the root radius.

A matched gear and pinion set has inscribed on a surface of each part a setting or mounting distance, in a language agreed upon by the vendor and the customer. If this setting has not been adjusted properly in the assembly, the normal load contact area will be displaced. Figure 5-32 illustrates a tooth bending fatigue originating nearer the heel end. By placing this pinion on a tester and observing the pattern, it was determined that the part had been preset 0.020 in. out from the gear and 0.020 in. back toward the heel, away from the established setting information.

Any transmission or power train equipment has vibrations. Even the smoothest source of power will pulsate. Roller bear-

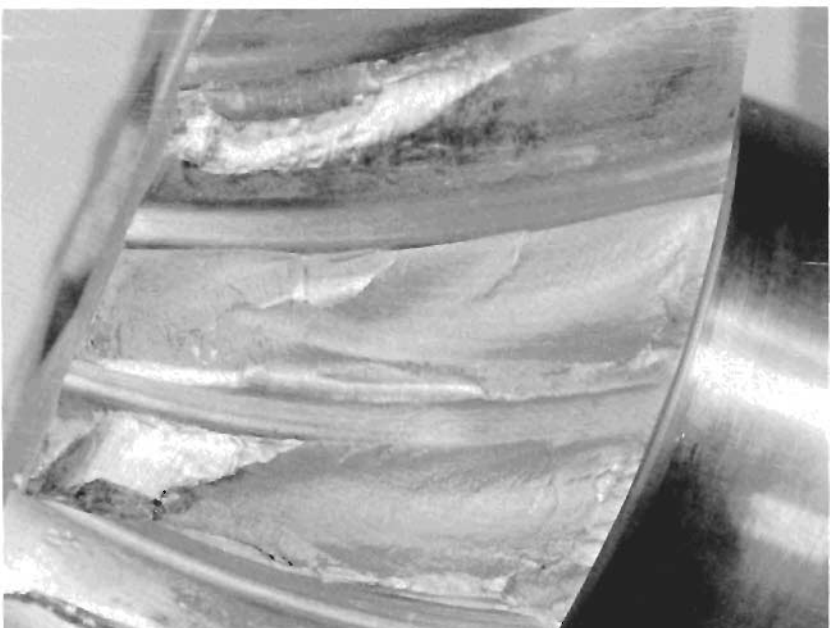


Fig. 5-32. Spiral bevel pinion, 1.0×. Tooth bending fatigue toward the heel end caused by misalignment in the assembly 0.020 in. out and 0.020 in. back from the gear.

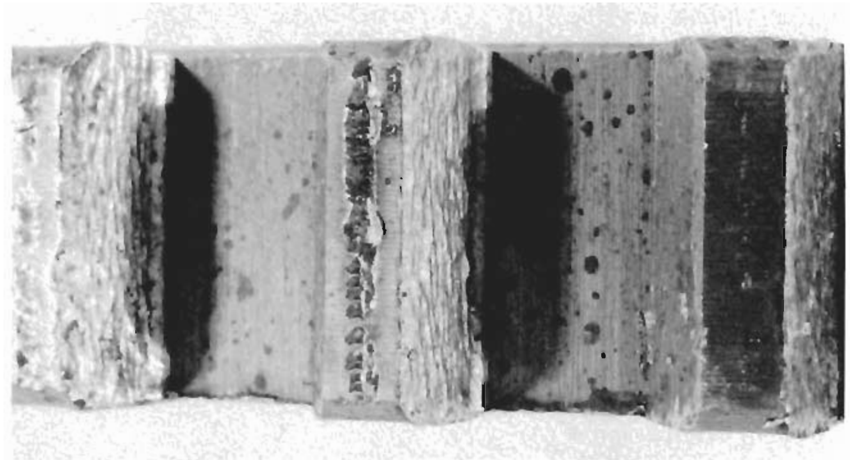


Fig. 5-33. A hub spline that permanently fitted into a wheel gear. Vibrations advancing to chatter, along with foreign particles, set up a destructive wear condition.

ings vibrate; gear teeth vibrate; rotating surfaces vibrate. Each part vibrates with its own peculiar amplitude and frequency, but this usually creates no problem, unless two or more parts synchronize to form a high noise level or to form a sequence of extremely high load peaks. Perhaps 85% of all surface fatigue is caused by torsional vibrations. This may be a high estimate, but torsional vibrations (chatter) reduce the fatigue life of a unit drastically. Figure 5-33 shows an external splined hub, fitting permanently into an internal splined wheel gear, disintegrating rather rapidly due to chatter and foreign material.

Mechanical Damage

Mechanical damage is damage to the gear inflicted by a force of another object during the manufacture, packing and shipping, unpacking, or assembly. This may be the same type of damage as that inflicted by a foreign object, but from a different context of environment and time. For instance, a cutting tool had broken and gouged a tooth surface (Fig. 5-34), not causing the trouble it might have; a bump on the top face of a gear tooth subsequently caused bidirectional tooth bending fatigue (see Fig. 4-9 and 4-10); one finished gear, dropping on the ends of the teeth of a second gear (Fig. 5-35), caused impact failure or cracking for subsequent fatigue. Also, many instances of nicks and bumps—too numerous not to be noticeable—have set up stress raisers for several modes of failure.

Lubrication

The causes blamed on lubrication (discussed in Chapter 2) concerning gear failures are usually due not to the lubricant itself, but to the application of the lubricant or to the external forces working against the lubricant. The gear failures that can be attributed directly or indirectly to the lubricant are the result of incorrect lubricant, a faulty lubrication system, insufficient or excessive lubricant, or lubricant contamination. A correct lubricant not only applies the proper surface film characteristics under the applied loads, but readily absorbs and dissipates heat. An incorrect lubricant may break down rapidly and retain very little lubricity. A breakdown of lubricity causes a reaction similar to that caused by lack of lubrication; namely, a metal-

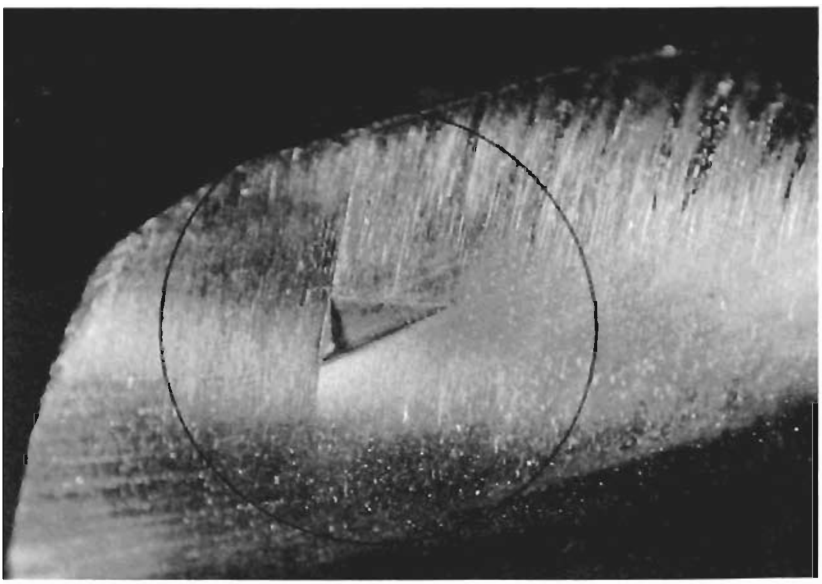


Fig. 5-34. Sprial bevel gear tooth, 3×, showing surface gouge due to the breaking of a cutter blade.

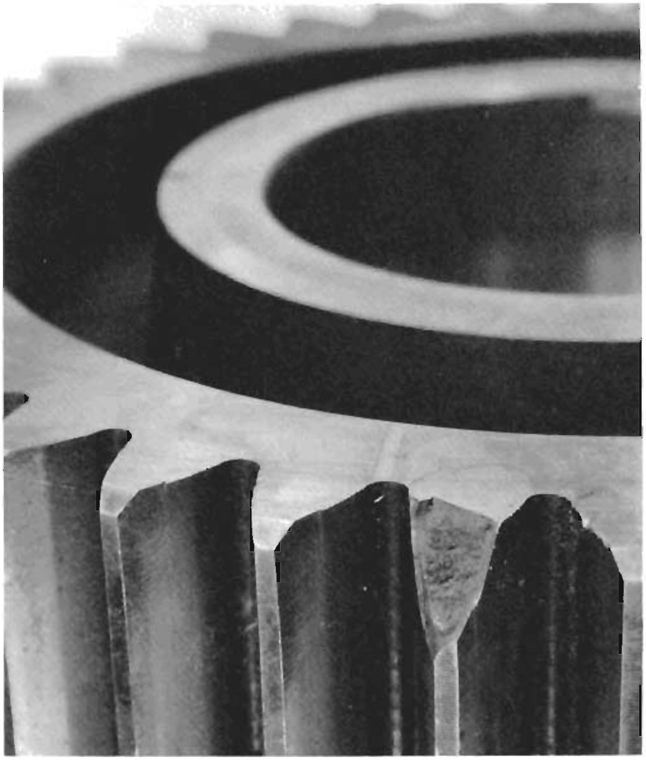


Fig. 5-35. Spur gear damaged by the impact of a dropped gear.

to-metal contact that quickly allows frictional heat to build up and adhesive wear to result. If a failure appears (as in Fig. 4-15 and 4-27), the type or amount of lubricant should be questioned.

The lubrication system may be inadequate in promoting rapid heat absorption and dissipation, resulting in high temperatures and adhesive wear on the moving parts.

Excessive lubricant is not common, but it tends to build up internal pressures and temperatures detrimental to the housing's carrying capability.

Lubricant contamination is a very complex problem. The subtle contaminants are the gas and liquids absorbed by or dissolved into the lubricant. They may change the chemistry enough to cause a breakdown of the lubricant, or they may set up chemical cells corrosive to the gear surface. The less subtle contaminant is the solid sand and dust type that is an abrasive material using the lubricant as a vehicle to transport the abradant and as a storage for additional abraded material. The results are clearly illustrated in Fig. 5-36.

Foreign Material

Although the contaminants in the lubricant are essentially foreign materials, they are considered contaminants only as long as they are small enough to be transported by the lubricant. When particles become large or heavy enough to settle out of the lubricant and become enmeshed within the gear teeth or any other moving parts of the assembly, they are classified as foreign material. The failure resulting from foreign material is usually a secondary failure, since the foreign object involved is generally a broken particle or object resulting from the primary failure. Several examples follow. A bolt became loose and

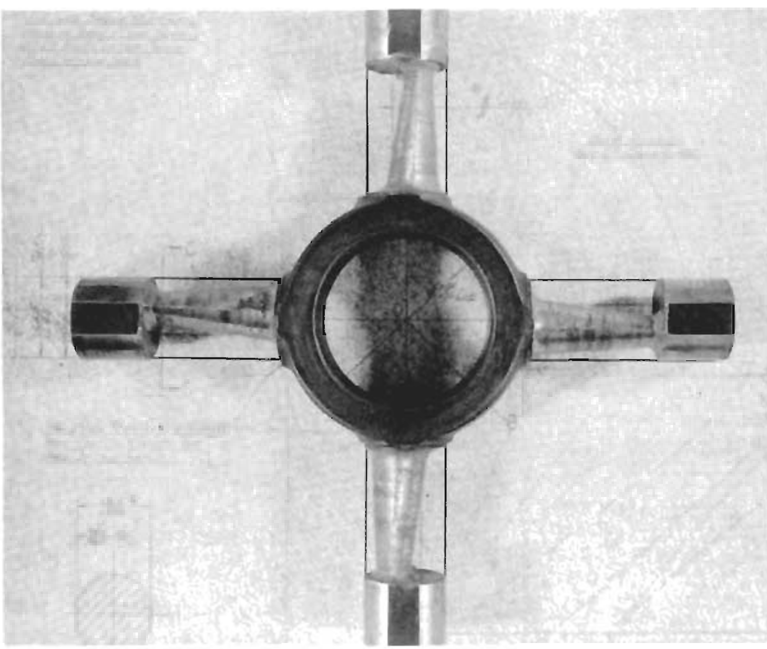


Fig. 5-36. Differential spider arms worn deeply by abrasive material as a contaminant in the lubricant.

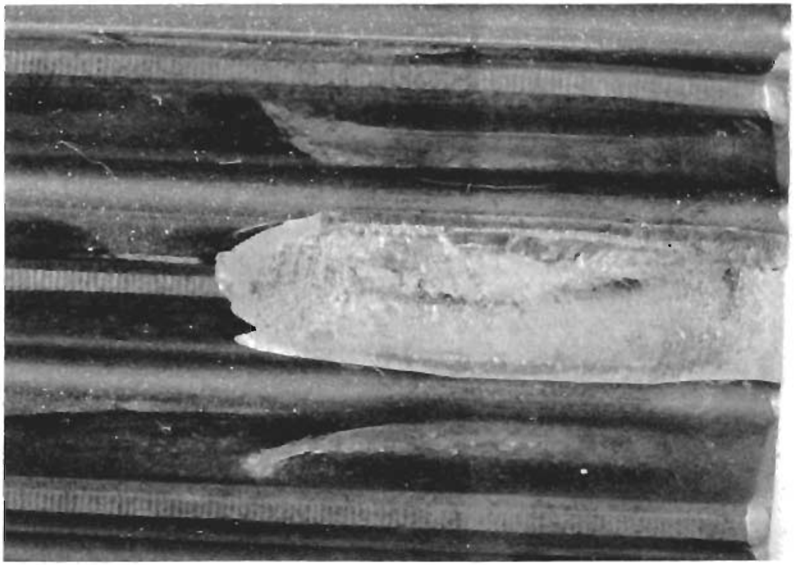


Fig. 5-37. Spur gear, 3×. Tooth bending fatigue with origin from a crack at both root radii caused by the enmeshing of a portion of a needle bearing roller.

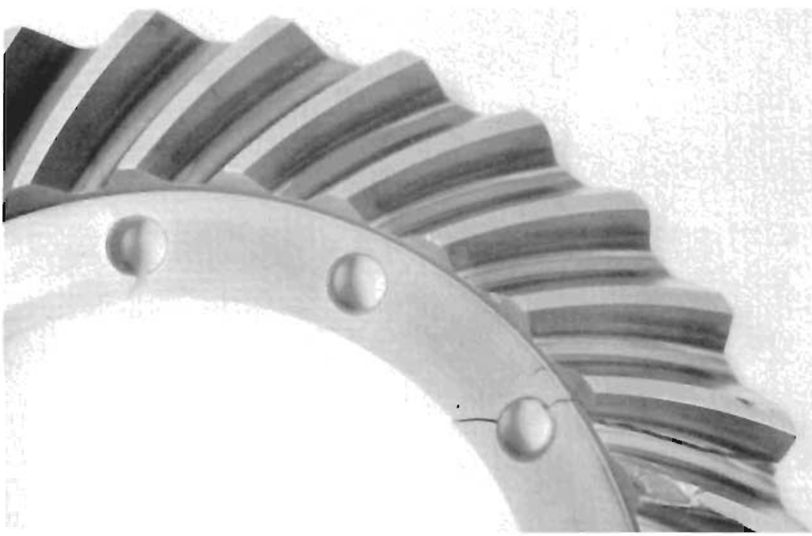


Fig. 5-38. A foreign object (top of adjacent tooth) became enmeshed in spiral gear (0.5×) teeth causing a full section break. The top of the adjacent tooth also broke by impact from some object.

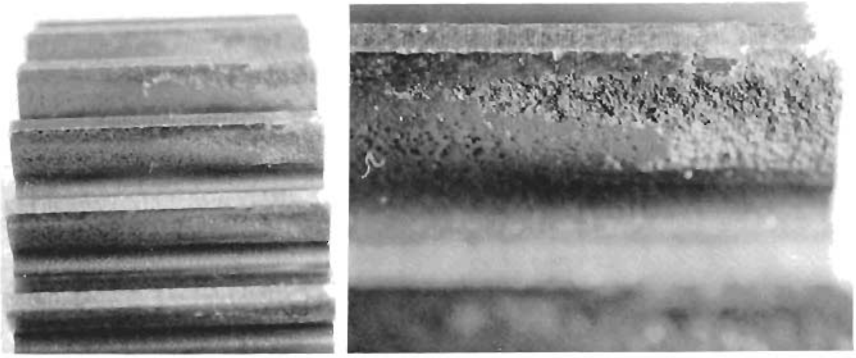


Fig. 5-39. Spur gear used in a seasonal operation was at the top of an assembly, out of the oil, and subject to corrosion by cold weather condensation during the idle season.

backed into the teeth of a spiral pinion (Fig. 4-39); a portion of a needle bearing roller became enmeshed between the gear teeth (Fig. 5-37), causing a crack at the root radius that continued as tooth bending fatigue; and an entire gear section was broken by pressures from an enmeshed tip of an adjacent tooth (Fig. 5-38), which also had been broken by impact from some object.

Corrosion

Corrosion is chemical deterioration of a surface (as shown in Fig. 5-39). The corroded surface may not be considered a failure in a power transmission gear, for example, since it can still function for its intended purpose, though it may not look good and may be slightly noisy. However, if it were a pump gear for liquid transportation, it would certainly be a failure, since it could not function efficiently for its purpose. The most dangerous aspect of a corroded surface is its ability to become a stress raiser and enhance pitting, spalling, tooth bending fatigue, and brittle fracturing, all of which can happen without notice and cause tremendous overall damage.

Continual Overloading

The result of consistency is uniformity and predictability. Consistent, continual overloading of a gear will result in a uni-

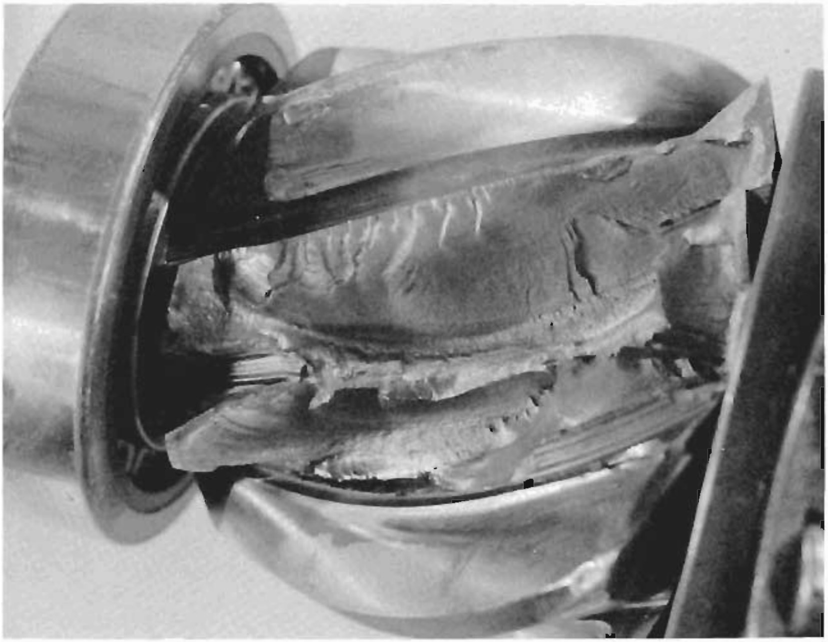


Fig. 5-40. Spiral bevel pinion, 0.6×. Two teeth out by tooth bending fatigue, but all teeth are fractured almost completely through. Continual overload.

form load pattern and the failure is predictable. Overload does not necessarily mean "beyond the designed characteristics as engineered for the product." An applied load may be well within the designed characteristics but may be a severe overload for the existing conditions of the gear.

Spiral bevel gears are interesting to study. With no load, the contact pattern is near the toe end of the profile; under full load, the contact pattern is well centered; and under continual overload, the contact pattern may shift by deflection toward the heel and higher on the profile. But this is not necessarily the case, since some spiral designs tend to keep the overload pattern at the central area and tend to go lower on the profile. Figure 5-40 illustrates an overloaded spiral pinion in which every tooth is cracked and ready to fall out. Two teeth fell out first, but the load pattern stayed well centered and deep, perhaps because the mating gear teeth also deflected enough to compensate for the load shift. The gear teeth shown in Fig. 3-1 were overloaded for the condition that existed and not for its designed strength. No gear is designed for full load to be applied at the ends of the teeth. Figure 4-42 illustrates gear teeth that were not functioning, although only normal loads were being applied. The failure was consistent and uniform. The condition of shallow case depth and low core hardness was being overloaded.

The shaft failure illustrated in Fig. 3-2 was caused by continual overloading. The failure pattern was uniform and predictable. The part was designed to have withstood normal loads. Figure 5-41 shows a spur gear that had been continually overloaded. The adhesive wear pattern is uniform on both sides (not the case with gears running in only one direction) and on all teeth. The loads were heavy enough to break through the lubricating film and allow severe surface friction. Overloading does

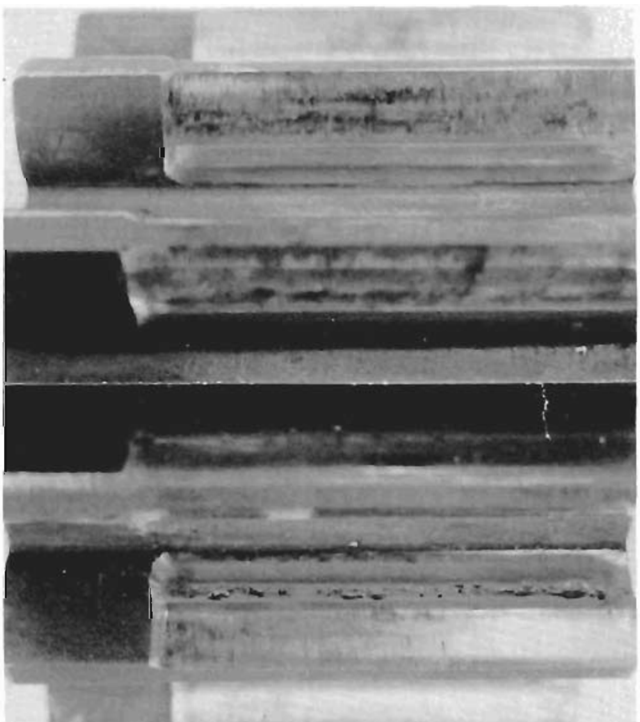


Fig. 5-41. Spur gear, 0.9×. High continual overloads broke down the lubricant barrier, causing deep adhesive wear.

not cause only breakage; it can also cause wear, pitting, and spalling. A continual overload due to improper design is illustrated in Fig. 5-14.

In general, if the mode of failure is rolling contact fatigue distinguished by "butterfly wings," continual overloading is almost certainly the cause.

Fatigue failures originating at sharp corners and undercuts are caused by continual overloading for the condition existing, but may not be overloaded for the designed application. An analyst has to be careful about making an incriminating statement about a condition. For instance, a shaft was designed with a keyway and the engineers took that into account and said that the designed load for continuous operation was X-amount. However, the part operated continually at a load of 1.25X. It failed in fatigue as expected, at the keyway, but the failure was the fault of overload, not of the keyway. Without the keyway, the shaft could have easily withstood loads of 4X; but that evidence had nothing to do with this application. However, had the designing engineer ignored the keyway and stated that the designed load was 4X, the keyway failure at an applied load of 1.25X would have been the fault of improper engineering design. From another angle, the designing engineer calculated that a certain keyway, if carburized and hardened at all radii the same as all other surfaces of the shaft, would have an operating strength of 3X. However, manufacturing decided that dimensions were the most important characteristic; so the shaft was routed to carburize, slow cool, cut keyway, and reharden. The closer dimensions were held; but the low hardness of the keyway radii allowed the shaft to fail in fatigue at 1.25X of load. The cause of failure was manufacturing misjudgment.

The obviousness of a particular mode of failure does not in any way change the analyst's method. It should be deliberate and thorough, and the analyst should be certain of the accuracy of his results.

Impact Overloading

A term equivalent to impact overloading is "sudden peak overload." Both imply that a gear may be operating at a consistently loaded condition when, for a brief moment of time, the load peaks out considerably over the designed maximum. The term "shock loading" has also been used in some reports. What happens to the gears? Sometimes nothing. Sometimes a small crack at a single root radius may start. And sometimes the entire tooth breaks away.

Some examples of the effects of impact overloading are illustrated in Chapter 4:

(*a*) Tooth bending fatigue with origin at a root radius due to sudden overload is characterized by appearing to be a random tooth and not consistently the same on all teeth. Figure 4-1 may have been such a case and Figure 4-5 was definitely so.

(*b*) A sudden overload may quickly shift a contact pattern of a loaded spiral gear toward the top of the heel end, causing an impact fracture of two adjacent teeth (Figure 4-36).

(*c*) Teeth that may have started by fatigue due to other causes may suddenly be broken out by impact (Figure 4-6).

(*d*) Impact failure due to intrusion of a foreign object may cause a consistent damage (Figure 4-39).

(*e*) A sudden stop may cause a brittle fracture, followed by shear (Figure 4-38); or, if the stop is violent enough, several teeth may be sheared simultaneously (Figure 4-37).

(*f*) In some instances, tooth crushing can be caused by impact overload (Figure 4-41).

(*g*) Torsional tensile failure may be evident (Figure 4-29) as sudden overload.

(*h*) Torsional transverse shear (Figure 4-43) is a definite case of shock loading.

Bearing Failure

Just as misalignment and deflections can shift load contact areas on gear teeth, so can bearing failures. In fact, a failed bearing actually misaligns the entire associated gear train. The modes and causes of bearing failures are just as many and varied as for gear failures; but all that will be said here is that a failing bearing will affect the life of the gears. Conversely, the bearing analyst will admit that a failing gear will affect the life of the bearings. It is in the best interest of all concerned that both analysts recognize the interrelationship, not only of gears and bearings, but of all moving and nonmoving components of an assembly.

Maintenance

Maintenance personnel, equipment, and procedures can make or break a company. Not only must good replacement parts be used, but care must be taken that all used parts are cleaned and thoroughly inspected before being placed back into service. No gear should be reused unless a magnetic-particle inspection method has determined a freedom from crack initiation. No mated gear or pinion should be run mated with a random part. Again, cleanliness and correct alignment must be maintained during the reassembly of repaired items. Many users of equipment have programs of "preventive maintenance," in various degrees of sophistication, from well-thought-out systematic checks of all components, to perhaps yearly hit-or-miss checkups of only the major operating parts. Other equipment users wait for parts to break down before fixing them. Probably the most common and most abusive evidence of lack of proper maintenance is insufficient or improper lubrication.

Operator Error

The operators of equipment are blamed for more than their share of failures; but it is often the result of operational error that failures occur. In one situation, the operator of an airport snow-removal vehicle was pushing the large bucket at 25 miles per hour through the snow and slamming into the large bank of snow at the end of the field without slowing down. Each time, the rear end of the vehicle lifted up about six feet and slammed back down on the concrete. This procedure was not at all necessary, nor was the equipment (especially the gears) designed to take this abuse time after time. (The pinion of Fig. 5-40 and the account in Chapter 6—the study of a spiral drive set—were actually taken from this model front-end loader.) Reasons such as this prompt the repetition of a previous statement: When a catastrophic failure occurs, don't look for a microscopic cause. Find the cowboy at the wheel or the one that threw the monkey wrench into the works.

Field Application

It is possible that field application is the most important factor or, as some believe, the cause of the greatest number of gear failures. It is certainly true that the failures occur during service operations; but unless it can be proven that the application is definitely overstressing the gears, the application may not be the primary cause. Whatever the mode of failure, the strength at the origin must be determined, to verify whether the field application was beyond the scope of the designed capabilities. If the applied load that originated the failure was random and an isolated case, there is not much that should be done. If, on the other hand, the field application is to be consistent with the loads causing failure, by all means, the application for the assembly being used should be changed, or the assembly redesigned to accommodate realistic loads.

One more important subject must be discussed; i.e., a change of application in the field. Again, an illustration: The author and his father bought a 1½-ton flatbed truck for use to haul wood, hay, livestock, and what-have-you. They made different racks and sides to accommodate the type of loads. The father was also a mining man; so they designed a dumping bed for hauling ore, put on two sets of overload springs, and hauled between 3 and 4 tons per load. They changed the field application but did not change the differential or the two drive shafts. Anyone can guess the results, which were rather costly to repair. The other aspect of change of application comes about when the manufacturer of the original equipment decides that greater horsepower is required to meet the challenge of competition. It is surprising how many times increased horsepower, as well as overload springs, are the only items that seem to make a difference in a job application.

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, *A 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 6

The Final Analysis

No failure examination has been completed until an evaluation of the results is made; and it is only when the failure mechanism is understood that effective corrective measures can be devised.

The preceding chapters have attempted to give the reader an understanding of failure mechanisms in a logical sequence. The purpose of this chapter is to explain how to take the known facts of a specific failure, place them in a systematic context to arrive at a logical conclusion, and point to a corrective measure. Because, understandably, the hundreds of available examinations cannot be detailed here, a chosen few will be presented for the purpose of showing how a systematic approach to failure examination actually works.

Broken Shaft

One end of a broken axle shaft was returned to the manufacturer and subsequently submitted for analysis.

Background Information

Field application: The axle shaft was used in a heavy-duty open-pit mining truck hauling overburden. The shaft had failed after operating for 27,000 hours. Previous failures had resulted from longitudinal shear, but this had not; the customer thought the material was defective.

Visual Examination

The returned shaft section did not have the identification markings (they were on the other end). The part number was listed in the accompanying report. Photographs were taken of the fracture surface, centering on the point of origin (Fig. 6-1).

The mode of failure was torsional fatigue in the tensile plane, with fatigue striations progressing over a large number of cycles before final fracture: "torsion in the tensile plane" because the fracture was at 45° across the shaft surface (see Fig. 1-11); "fatigue" because of the radiating striations, or beach marks, over the fracture surface; and "large number of cycles" because the beach marks in general were very close together. Each beach mark may represent many cycles at low stress, since only one cycle at higher stress will form an additional mark.

The origin of the fatigue fracture was at the surface along a sharp, gouged indentation at the juncture of the radius. To find the origin, follow the area of striations back to the center of the half-circle or focal point. The focal point is at the surface. Look at the surface to find the deep gouge mark.

At this point, the cause may seem obvious, and it may seem unnecessary to do any further examination. But although the part number had been submitted, it is as yet necessary to establish if the material and heat treatment verify this part as being correct or part of the problem.

Metallurgical Examination

Chemical analysis verified that the part was SAE 4340 and thus met material specification. The part was heat treated by through hardening, then quenched and tempered to 42-47 HRC. The surface hardness was 46 HRC, and the core hardness (½ radius) 46 HRC.

In conclusion, the shaft met the metallurgical requirements and should have withstood normal operating conditions. The mode of failure was torsional tensile fatigue; and the cause of failure was mechanical damage to the surface of an area of high stress concentration.

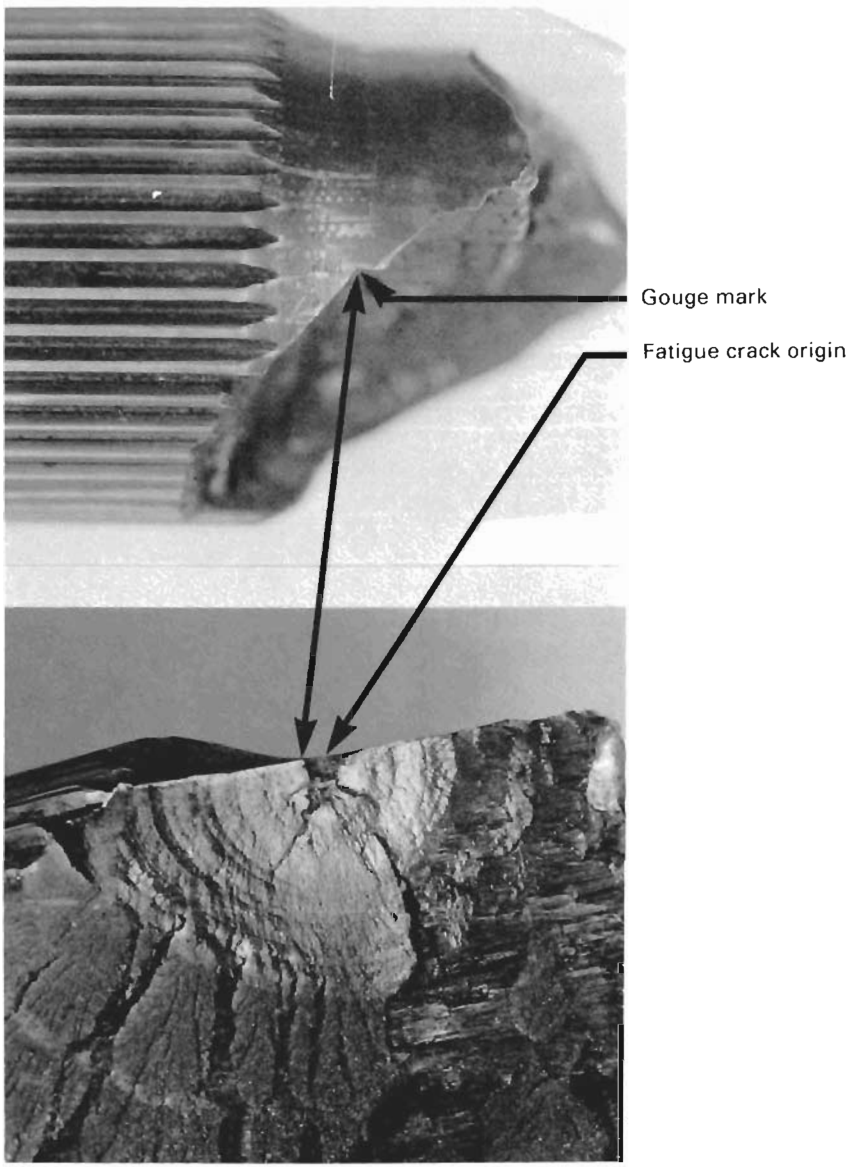


Fig. 6-1. Failure of this axle shaft resulted from torsional fatigue in the tensile plane, originating from one of several gouge marks observed around the shaft at the splined radius. The fatigue crack progressed for a large number of cycles before final fracture.

This is not the final analysis, because two questions have yet to be answered: Where did the gouges come from, and how could they have been prevented? The spacing of the gouge marks was observed to be coincident with the spacing of the splines. The spacing was then traced to the operation where the shaft had been assembled into the splines of the mating wheel gear. This operation had not been accomplished very smoothly.

Broken Spiral Bevel Gear

A spiral gear and pinion set were submitted for analysis. The pinion was intact, but the gear had broken into two sections that resulted when two fractured areas went through the body of the gear.

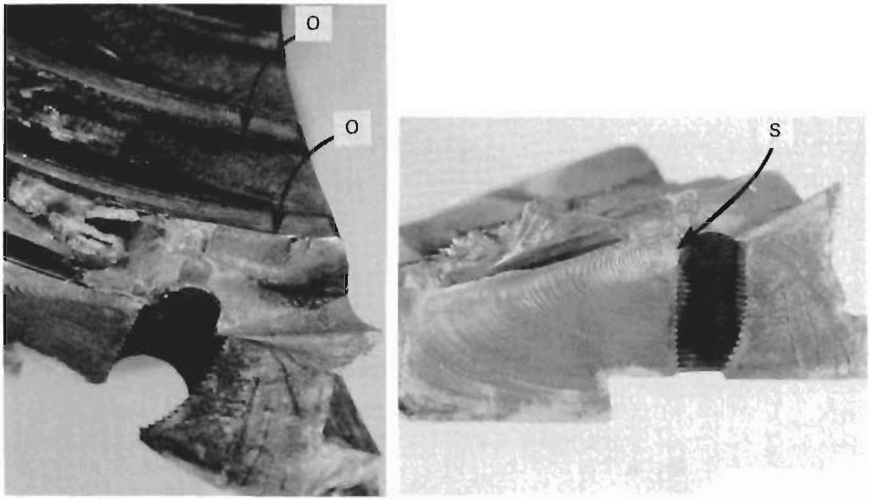
Background Information and Customer's Evaluation

Field application: The set was a high-speed electric traction motor gear unit driving a rapid transit train. The gears drive about equally in both directions; i.e., forward going north and reverse going south. Although acceleration and deceleration are generally smooth, they are rapid.

The assembly was installed in mid-1975. The wheel mileage of the assembly was 34,000 miles at the time of failure. The axial movement of the pinion and bearing stackup was zero.

The customer observed that two complete fractures were closely associated with two major sections of gear tooth fatigue: fracture area "A" (Fig. 6-2a) developed from a bolt-hole opening closely related to the root of a severely fatigued tooth, progressed along the bearing surface of the adjacent tooth and through the O.D.; fracture area "B" (Fig. 6-2b) developed from the origin of a bolt-hole opening and progressed in a parallel alignment to the root of the tooth, in a very similar pattern to "A."

The customer thought the failure resulted from deep root stress fatigue, developing from the final depth of the two mounting-bolt holes.



Progression is normally downward, toward the neutral point below the center of the tooth, and then upward to the opposite root. In this instance, a bolt hole was intercepted near its terminal end. A secondary fatigue (S) started at this intercept, and slowly continued both across and through the entire cross section of the gear until final separation occurred.

Fig. 6-2(a). Fracture "A" in a spiral gear. The origin (O) of tooth bending fatigue is near the toe end of the concave (reverse) root radius. See also Fig. 6-2(b) and 6-2(c).

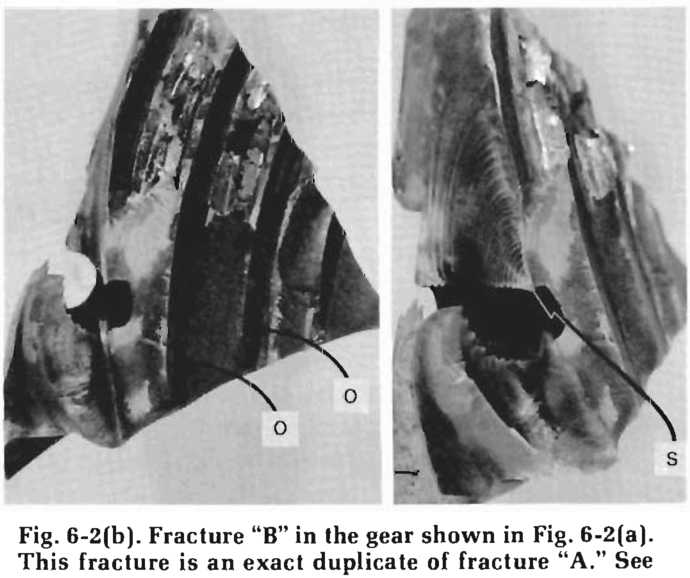


Fig. 6-2(b). Fracture "B" in the gear shown in Fig. 6-2(a). This fracture is an exact duplicate of fracture "A." See also Fig. 6-2(c).

Visual Examination

Identifying markings are to be interpreted first. (No part numbers are listed or shown in photographs here, since part numbers are the property of a customer and in some instances might identify that customer.)

Markings on pinion: PART F-3 G3-26 DLU B

PART	Part number (not disclosed).
F-3	Manufactured in June 1973.
G3-26	Mated with gear as set No. 26 in July 1973.
DLU	The material code assigned to Republic Steel Heat No. 3113989, grade SAE 4320H.
B	The second lot of this part going through the heat treating process.

Markings on gear: PART D-3 G3-26 UWJ A 57-22

PART	Part number (not disclosed).
D-3	Manufactured in April 1973.
G3-26	Mated with pinion as set No. 26 in July 1973.
UWJ	The material code assigned to Republic Steel Heat No. 6078524, grade SAE 4320H.
A	The first lot of this part going through the heat treating process.
57-22	Set ratio; gear 57 teeth, pinion 22 teeth.

Physical appearance of pinion. Two adjacent teeth were crushed and spalled severely, and all teeth were rounded over the top face; this physical damage is of a secondary nature. There had been heavy tooth contact loading high near the heel end of the concave side of all teeth. Also, magnetic-particle inspection revealed a fatigue crack originating at the root radius of each tooth, at the heel end of the concave (forward) side. The convex (reverse) sides of all teeth were deeply worn and pitted low on the profile toward the toe end.

Physical appearance of gear. (Study closely Fig. 6-2a, 6-2b, and 6-2c). Fifteen teeth broke out at the toe end with a fatigue fracture whose origin was at the root radius, approximately 1 in. from the toe end on the concave (reverse) side. The concave

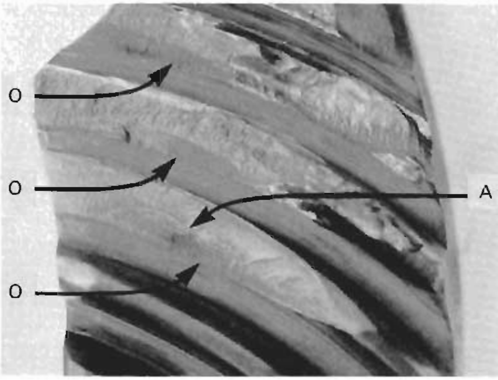


Fig. 6-2(c). Of the 15 teeth that failed by tooth bending fatigue, two fractures intersected a bolt hole (see Fig. 6-2a and 6-2b). Four other fractures touched the apex (A) of a bolt hole. Nine teeth failed by tooth bending fatigue completely away from a bolt hole. All 15 have an origin (O) in the same re ative position.

sides of the remaining 42 teeth showed extremely heavy contact at the toe end and over the top corner and face, causing a rounded condition of the corner. Magnetic-particle inspection indicated that several of the remaining teeth had cracks originating in the same area as the fractures. The tooth contact area on the convex (forward) sides was very heavy, low on the profile near the heel end.

Fractures "A" (Fig. 6-2a) and "B" (Fig. 6-2b) show the origin (O) of tooth bending fatigue 1 in. from the toe end along the concave root radius. Progression is normally downward toward the neutral point below the center of the tooth, and then upward to the opposite root. In these two instances, the progression intercepted a bolt hole near its terminal point. A secondary fatigue fracture then started at this intercept (point S in Fig. 6-2b) and slowly continued both across and through the entire cross section of the part until final separation occurred.

Of the 15 teeth that failed by tooth bending fatigue, two fractures intersected a bolt hole as discussed above; nine teeth failed in a normal pattern completely away from a bolt hole; and four teeth failed normally but touched the apex of a bolt hole (shown in Fig. 6-2c as point A).

Physical Examination

Magnetic-particle inspection was used to determine any extended damage due to the primary failure of tooth bending fatigue of the gear. It indicated several other gear teeth progessing in the same failure pattern as the 15 failed teeth. It also indicated that every pinion tooth had started to fail by tooth bending fatigue in the forward direction and at the heel end. Both discoveries substantiated the crossed tooth pattern discussed earlier.

There was also a second reason for magnetic-particle inspection. The teeth of both parts had been ground by a spiral bevel tooth grinder, so it was necessary to inspect for grinding checks that might have been a factor. None were found.

Nital etching was used to determine if there were any grinding burns that could cause tempered spots in critical areas. Again, none were found.

Metallurgical Examination

A decision must now be made as to which, if any, of the metallurgical tests are to be made. Although the cause of failure may be apparent at this time, some doubt would remain unless all remaining possibilities were checked.

The primary mode was tooth bending fatigue, both of the gear and of the pinion. Was the carburized case hard enough and deep enough at the root radius to have withstood normal loads? A case hardness traverse must be taken to determine case properties. Since the failure was at the root radius toward the toe, select a tooth from each part that remains but is cracked in the same area, and section that tooth at the point of the crack origin. Prepare the sample for the microhardness tester and take the hardness readings as closely as reasonable to the crack.

The teeth of both parts had been ground. Could this make a difference in case characteristics from one side to the other? Indeed it could, and often does. Therefore, a case hardness traverse at both radii is necessary.

Is it necessary to run a case hardness traverse at the mid-profile of both sides? When determining factors for tooth bending fatigue, it is not necessary. However, after completing all the preparation up to this point, an analyst would be amiss not to. In fact, this is the only way to establish if the case depth and hardness specification had been maintained.

Is it necessary to take a chemical analysis? Not at this time. The material and heat treatment can usually be recognized by the microstructure. If the microstructure is not consistent with expectations, an analysis will be necessary.

Microscopic examination is always accomplished when a sample has been prepared for case hardness traverse. This procedure should be automatic, since it is the only visual means of tying all other physical and metallurgical tests together.

After it has been determined which procedures are to be followed and why, the results are observed:

	Pinion	Gear
Surface hardness	61 HRC	58/59 HRC
Effective case depth, root radius	0.040 in.	0.044 in.
	0.042 in.	0.048 in.
Effective case depth, midprofile	0.061 in.	0.077 in.
	0.061 in.	0.077 in.
Retained austenite	Less than 5%	Nil
Core hardness (midprofile–root)	32–25 HRC	41–32 HRC

Final Analysis

All physical and metallurgical characteristics were well within specified standards, and both parts should have withstood normal loading conditions.

The primary mode of failure was tooth bending fatigue of the gear from the reverse direction near the toe end. Occurring perhaps simultaneously was tooth bending fatigue of the pinion teeth from the forward direction near the heel end. The bolt holes in the back face of the gear happened to be in the path of fracture progression; but they did provide a path of least resistance from the point of intercept to the back face, which allowed for the full body break.

The cause of failure was a crossed-over tooth bearing condition that placed loads at the heel end when going forward and at the toe end when going in reverse. The condition was too consistent to be a deflection under load; therefore, it most likely was permanent misalignment within the assembly.

Spiral Bevel Drive Set

A spiral bevel set that failed in service was submitted for an evaluation.

Background Information

Field application: This was the main driving set in the differential housing of a large front-end loader moving coal in a storage area. The machine had operated for approximately 1500 hours.

Visual Examination

Identifying markings:

Markings on pinion: PART F0-129 4817 41-8

PART	Part number (not disclosed).
F0-129	Mated with gear as set No. 129 in June 1970.
4817	Grade of steel used (SAE).
41-8	Set ratio; gear 41 teeth, pinion 8 teeth.

Markings on gear: PART F-0 F0-129 4820 41-8

PART	Part number (not disclosed).
F-0	Manufactured in June 1970.
F0-129	Mated with pinion as set No. 129 in June 1970.
4820	Grade of steel used (SAE).
41-8	Set ratio; gear 41 teeth, pinion 8 teeth.

Physical appearance of pinion (Fig. 6-3). Two adjacent teeth had broken out. The next adjacent tooth had broken loose in the same manner and was hanging. All fractures indicate a tooth bending fatigue failure originating at the root radius of the concave (drive) side at exactly midway between the ends of the teeth. The remaining five teeth are highly polished on both sides and the contact pattern is well centered.

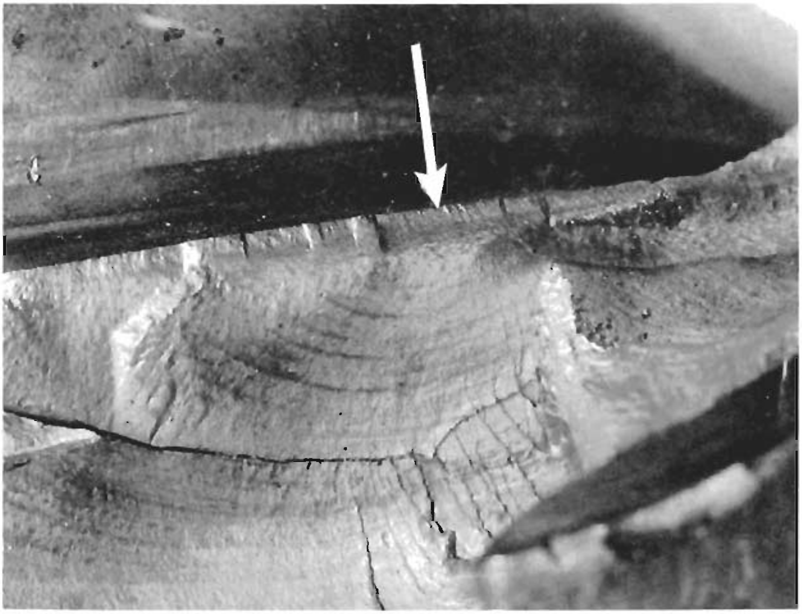


Fig. 6-3. Spiral bevel pinion tooth, 2×. Tooth bending fatigue with origin (arrow) at the root radius, exactly midway between the ends of the teeth.

Physical appearance of gear. Most of the teeth had been chipped at the back angle, but evidently this was subsequential damage to the pinion failure. The tooth contact pattern of all teeth was well centered and uniformly polished on both sides. No wear was evident.

Physical Examination

Magnetic-particle inspection was performed on each part. The pinion showed no indication of additional cracking, nor did the gear.

Metallurgical Examination

From the preceding examination, the analyst arrived at several conclusions. The primary failure involved the pinion teeth only; the gear need not be examined. Tooth bending fatigue demands a case hardness survey of the root radius and, of course, the midprofile. Also, a microscopic examination is necessary.

There appears to be no difference in the wearing of any tooth; therefore, select any one of the remaining teeth, section it midway from both ends, and prepare a sample for a case hardness traverse and microscopic examination.

From the case hardness survey and the microscopic examination, the following results were observed:

Core structure: An equal admixture of lamellar pearlite and low-carbon martensite. This indicates a well quenched and tempered structure.

Case structure: Acicular martensite retaining approximately 10% austenite near the surface. Excellent structure for material and heat treatment.

Case depth: (specified 0.075-0.090 in.)
0.085 in. effective depth at midprofile
0.054 in. effective depth at root radius

Case hardness: (58-63 HRC specified)
60 HRC at midprofile (by Tukon)
59 HRC at root radius (by Tukon)

Core hardness: (no specification)
36 HRC, tooth centerline at midprofile
28 HRC, tooth centerline at root

The material and heat treatment were as specified.

Final Analysis

The mode of failure was tooth bending fatigue with the origin at the designed position: root radius at midsection of tooth. The load was well centered, and progression occurred for a long period of time. The cause of failure was suddenly applied peak overload, which initiated a crack at the root radius. Progression continued by relatively low overstress from the crack, which was now a high stress-concentration point. This was a classic tooth bending fatigue failure (as explained in Chapter 4).

Spur Pinion

One end of an axle shaft containing the integral spur pinion was submitted for examination, along with the report of a tooth pitting failure (Fig. 6-4).

Background Information

The spur pinion, integral to the axle shaft, operated in a medium-size, off-highway truck at an open-pit mine. The pinion was the input sun pinion of a gear-reduction unit in the rear

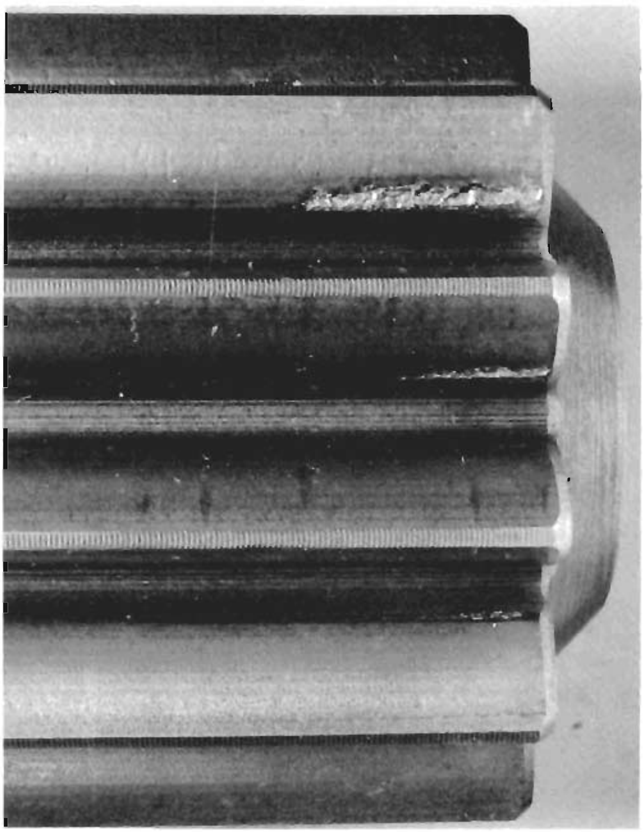


Fig. 6-4. Spur pinion, 1×, showing low-profile pitting along heavy line contact that is characteristic of tip interference from mating part.

wheel assembly, where it mated with four spur tooth idler gears. The only recorded length of service was the customer's statement—"for a relatively short time."

Visual Examination

Identifying markings. Only the pinion head had been returned. The shaft portion had been torch-cut away. There were no markings on the portion returned. The accompanying report did mention the part number, which is all that can be used at the start.

Physical appearance. All the teeth were intact and showed very little wear. One side showed a line of fine pits, mostly full-length, very low on the profile. They increased in intensity toward the open end of the teeth, as indicated by heavy pitting and spalling. The opposite sides of the teeth also showed full-length contact very low on the profile but did not show the same extensive pitting damage.

Physical Examination

Magnetic-particle inspection revealed no indication of crack initiation.

Tooth characteristic study of the involute was taken to determine if there was a positive error deviation from true involute at the lowest point of the active profile. The result was that the involute chart showed an almost straight line overall, with a slight negative deviation at the start of the active profile.

Surface hardness checks were taken along the maximum diameter of the teeth and along the root center. The resulting hardness was 57 HRC, which is within the specification of 57–63 HRC.

Metallurgical Examination

The pitting occurred along the lowest point of the active profile. A case hardness survey and a microscopic examination should be made of this area. Also, the tooth section should be taken near the open end of the pinion tooth through a medium amount of pitting, in order to study the mechanism of the pitting. Since there has been no positive identification of material, a chemical analysis should be performed.

Case hardness traverse

Surface hardness (by Tukon, 500g): 58 HRC
Effective case depth: 0.060 in.
Core hardness: 35 HRC at midpoint, and at root, 33 HRC

Microstructure

Core structure: Low-carbon martensite with fine black boundaries, and an occasional crystal of titanium nitride (TiN). The structure is normal for good heat treatment and the fine black boundaries and TiN crystals are common in boron-treated alloy steels.

Case structure: Rather coarse acicular tempered martensite retaining approximately 20% austenite near the surface.

Surface structure: At the pitted area low on the active profile, evidence of catastrophic movement of material away from the pitchline and toward the root. (An identical example is shown in Fig. 4-15.)

Chemical analysis. Carbon, 0.12%; manganese, 0.82%; nickel, 1.86%; chromium, 0.56%, and molybdenum, 0.28%. This analysis, along with the microstructure, confirms the specified material to be SAE 43BV12.

Final Analysis

The mode of failure was surface contact fatigue through the shear plane subsurface, at the lowest point of single tooth contact. The cause of failure was tooth tip interference from the mating gear teeth. Since the mating parts within the assembly had not been returned or examined, there are some unanswered questions facing the customer:

- Which of the idler gears is showing heavy load contact over the top corner?

- Do the teeth of the mating part or parts have a positive involute error at the top of the active profile?
- Are all four idler gears from the same lot and mated; or is one from another lot that may have different tooth characteristics?
- What is happening within the assembly or with the four idler gears that is causing a consistent and uniform load at the very end of the sun pinion?

Spur Gear

Three spur gears that had formed a straight-line train in a speed reducer were brought in for examination and evaluation, with no report accompanying them.

Background Information

No background information was submitted. The analyst was told only that the gears were to be returned intact. Therefore, he could not section them.

Visual Examination

Identifying markings:

PART, *8622*, *D-2*, on each gear

PART	Part number (not disclosed).
8622	SAE grade of material.
D-2	Manufactured in April 1972.

(It was a well-mated set of three.)

Physical appearance. No evidence of wear appeared on any tooth of any gear, and the contact pattern was well centered and normal. The pitting, as shown in Fig. 6-5, was on only two teeth of the input gear. Notice the heavy line of pits low on the profile of one tooth and high on the profile of the adjacent tooth. The next gear in line (idler gear) had two teeth with exactly the same pitting pattern but in the reverse direction, matching perfectly the first gear. On the opposite side of the second gear (180°) were three adjacent teeth with a line of pits—one high, one medium, and one low. The third gear in line (output gear) had three adjacent teeth with line pits exactly as in the second gear, but in the reverse direction, matching perfectly the second gear.

Physical and Metallurgical Examinations

Neither a physical examination nor a metallurgical examination was performed.

Final Analysis

A telephone call was then made to the user. A list of questions and responses from this conversation follows:

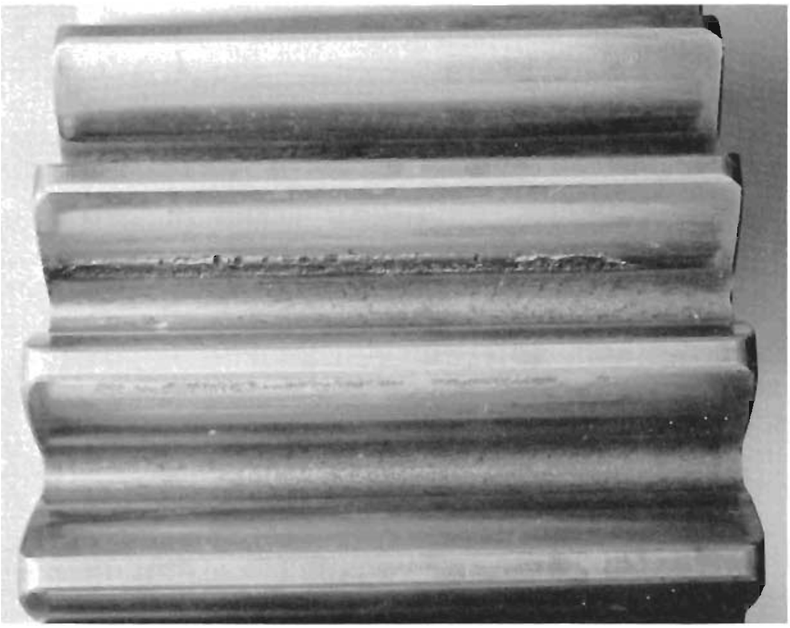


Fig. 6-5. Spur gear, 0.9×. Only two teeth pitted, one low on profile and the adjacent tooth high on profile. Mating gear had two teeth as mirror image. This could only occur with the gears in a static position under a reverberating type of load.

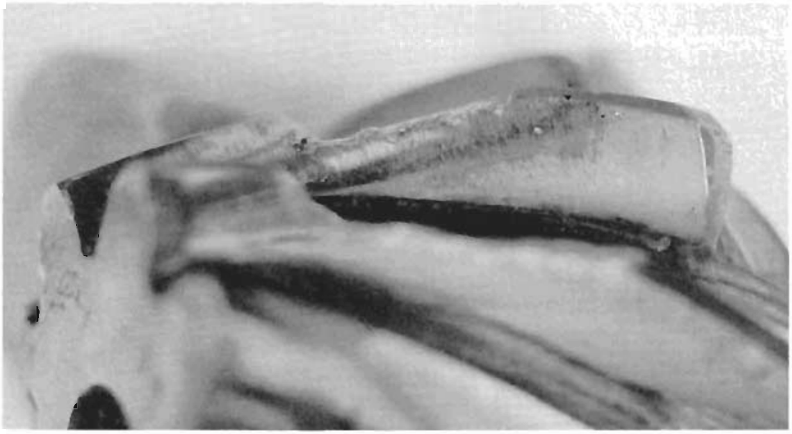


Fig. 6-6(a). Spiral bevel pinion, 0.9×. Seven of nine teeth failed by heavy rolling contact fatigue with the origin at a bias across the profile in a confined area.

Q. What type of equipment did this speed reducer operate, and under what conditions?

A. It was the power drive unit for an overslung lumber carrier running back and forth on rails.

Q. What type of power input was used—electric motor, gasoline engine, or hydraulic motor?

A. Hydraulic motor driven by a gasoline engine.

(The analyst knew from past experience that other components driven by hydraulic motors would shudder violently when held against an end stop. His next question reflected this knowledge.)

Q. Is it possible for this traveling crane to be locked in the same position time after time with the gasoline engine running?

A. Yes; in fact, every morning at one end of the track, the operator, who is 20 feet above the ground, starts the gasoline engine and lets it run until warmed up. In the meantime, the hydraulic motor and the speed reducer both vibrate constantly until the brake is released.

It was quite evident that this mode of failure could only occur with the gears in a static position under a reverberating type of load.

Spiral Bevel Set

A spiral bevel gear and pinion set was submitted for examination and evaluation because it had "worn out."

Background Information

This was a spiral bevel drive set with the gear attached to a differential. The assembled unit was driving a new, large, experimental farm tractor in the normal plowing and tilling operations.

Visual Examination

Identifying markings:

Markings on pinion: PART D-4 E4-1425

PART	Part number (not disclosed).
D-4	Manufactured in April 1974.
E4-1425	Mated with the gear as set No. 1425 in May 1974.

Markings on gear: PART E4-1425

PART	Part number (not disclosed)
E4-1425	Mated with pinion as set No. 1425 in May 1974.

Physical appearance of pinion (See Fig. 6-6a). Seven of the nine teeth failed from heavy pitchline contact on the concave (drive) side, with a progressive fatigue fracture through the case and core to the opposite side and down, parallel to the profile, to the tooth root on the convex side. The two intact teeth show the start of line crushing at a bias across the concave profile. Mating tooth contact was heaviest at center profile. Of three contiguous teeth that had been removed by continual crushing and wear, the center tooth still had a portion of the toe end remaining, which was placing an extremely heavy point pressure on the toe end of every third gear tooth.

Physical appearance of gear. 48 teeth. Every third tooth was broken at the toe end only; otherwise, all teeth were intact.

All teeth showed uniform heavy contact, well centered on the profile of the convex (driven) side.

Physical Examination

Magnetic-particle inspection showed a fatigue crack progressing through one of the two intact pinion teeth along the same path as that observed in the failed teeth. No indications were observed on the gear teeth.

Hardness checks on the surface of the pinion teeth rated the hardness at 61 HRC.

Metallurgical Examination

From the examination, a number of conclusions were drawn. The primary failure is definitely associated with the pinion; the gear need not be examined. The mode of failure must be determined through the concentrated load area. To accomplish this, a case hardness traverse must be made through a representative area and the microstructure studied. Therefore, section the one remaining pinion tooth through the midsection. Mount and prepare the sample for a case hardness traverse and a microscopic examination.

From the date of manufacture, the documented information on hand showed the pinion to be one of 10 forgings made from Republic Heat 6070929, grade 4820H. A nickel spot check confirmed the grade. No further chemical analysis was necessary.

Results of examination

Case depth: (specified 0.055-0.070 in. effective)
0.070 in. effective depth at midprofile
0.058 in. effective depth at root radius
Case hardness: (specified 59-63 HRC)
59 HRC surface at midprofile (by Tukon)
58 HRC surface at root radius (by Tukon)
Core hardness:
45 HRC tooth centerline at midheight
44 HRC tooth centerline at root circle

Core structure:

Rather blocky structure of low-carbon martensite with a small amount of pearlite

Case structure:

Tempered acicular martensite retaining approximately 10% austenite near the surface; the first transition product appears below 0.020 in. from the surface. The fracture area (originating high on the profile) indicates rolling contact fatigue, with subsurface shearing progressing toward the top of the tooth. In the high compressive area there are areas of "butterfly wings" extending from 0.010 to 0.035 in. below the surface, as shown in Fig. 6-6(b). The surface had not been smeared by any sliding action.

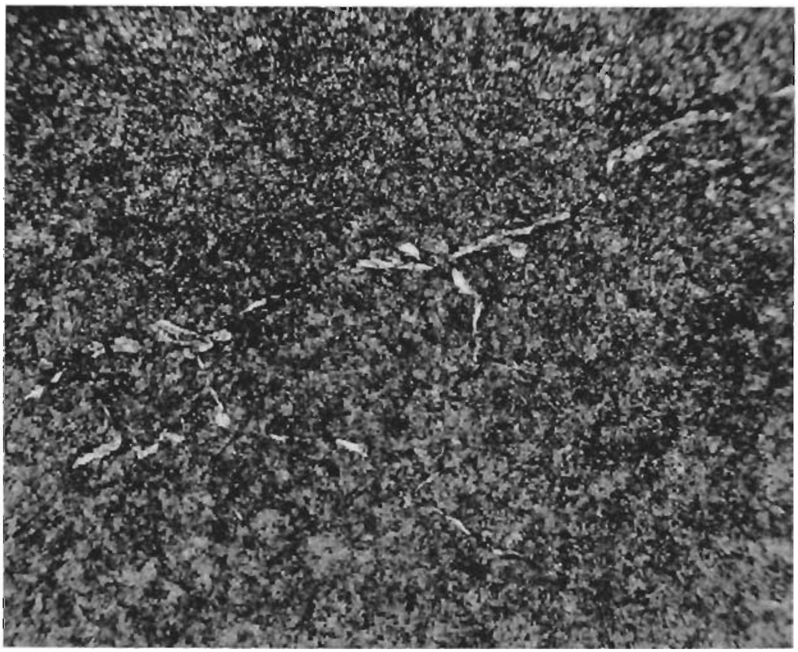


Fig. 6-6(b). Case microstructure (100×) of a high-compression area of one of the two intact teeth in the spiral bevel pinion shown in Fig. 6(a). Note the altered martensite in the subsurface shear plane. These "butterfly wings" extended from 0.010 to 0.035 in. below the surface.

Final Analysis

The mode of failure is rolling contact fatigue, and the cause of failure improper engineering design. The pattern of continual overload has been restricted to a specific concentrated area situated diagonally across the profile of the loaded side, which is consistent on every tooth. (The classic example of this failure mode is discussed in Chapter 4 and illustrated in Fig. 4-16. The cause of this mode of failure is discussed in Chapter 5 and illustrated in Fig. 5-14.) Corrective action could be to reduce the applied stress (not practical), or to change the design to allow a greater overlap of tooth contact.

Emphasis must be made that all failures displaying this pattern do not necessarily indicate a design problem. Keep in mind that the normal loading contact line of a spiral bevel tooth travels along the profile in a biased direction. Many failures look like this, even though there is greater than a one-to-one ratio. Before concluding that the condition shown in Fig. 4-16 is applicable, a complete engineering analysis must be made. The obvious may not always be obvious.

Damage to the gear teeth could have been reduced by changing the ratio slightly. The set ratio is 48 to 9. By changing to either 49 to 9, or 47 to 9, there would always be a hunting tooth situation and all teeth would receive the same loading characteristics.

The set ratio is important when noting the pattern of a failure. For instance, two gears with a 1-to-1 ratio (i.e., the same number of teeth in both parts) will always have the same tooth in contact. With a set ratio of 2 to 1 (gear 20 teeth, pinion 10 teeth), each pinion tooth will contact two gear teeth only, and those two teeth will be 180° apart. The same logic is used for a ratio of 3 to 1 or 4 to 1. The 48-to-9 ratio discussed earlier has a common denominator of three; therefore, a single tooth of the pinion will, during three rotations of the gear, contact every third tooth of the gear. The result was that every third tooth of the gear had been damaged by only one pinion tooth.

Spiral Bevel Set

A spiral bevel gear and pinion set that showed "excessive wear on the pinion teeth" was submitted for analysis.

Background Information

This spiral bevel set was the primary drive unit for the differential and axle shafts of an exceptionally large front-end loader in the experimental stages of development.

Visual Examination

Identifying markings

Pinion markings: PART L-9 L9-211 #1 6-38

PART	Part number (not disclosed).
L-9	Manufactured in December 1969.
L9-211	Mated with gear as set no. 211 in December 1969.
#1	The first set mated and tested for the order being processed.
6-38	Set ratio; pinion 6 teeth, gear 38 teeth.

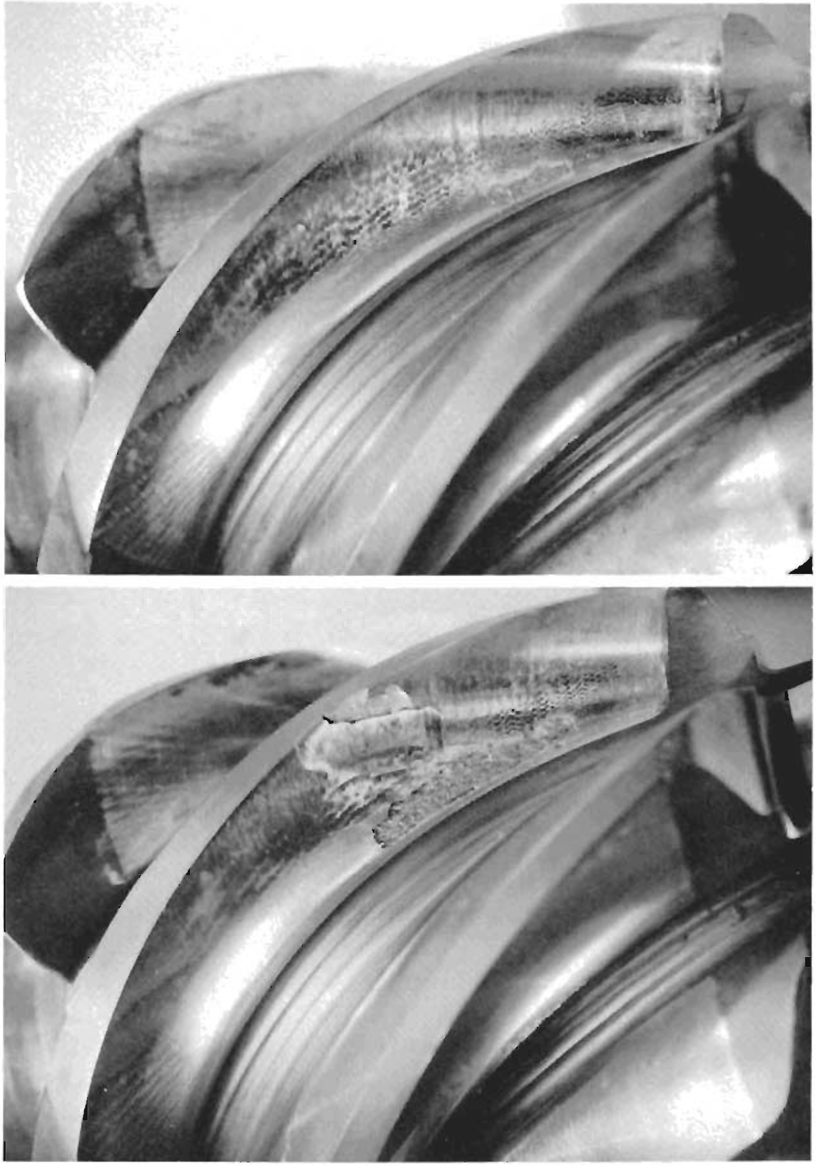
Gear markings: PART L-9 L9-211 #1 38-6

The same information as for the pinion above.

Physical appearance of pinion. The tooth contact area appears to have been heavy at the toe end and low on the profile of the concave (drive) side. The entire active profile shows surface rippling for over half the length of the tooth from the toe end, as shown in Fig. 6-7(a) and (b). Pitting has originated at the lower edge of the profile, 2 in. from the toe end of the concave side (Fig. 6-7a); it has subsequently progressed in both directions along the lower edge of the active profile and upwards toward the top of the tooth, changing the mode from pitting to spalling (Fig. 6-7b). Both photographs show evidence of a double contact: the original setting, and a secondary pattern with the pinion moved out and away from the center of the gear. The convex (reverse) sides are slightly worn and scuffed high over the profile.

(a)

(b)



(a) A rippled surface for ¾ length of the tooth from the toe end. Pitting has originated low on the active profile, 2 in. from the toe end. (b) Pitting area has extended in both directions and has broadened. The central profile has an area of spalling that appears to be contiguous with the pitting. The microstructure indicates the two modes are occurring independently.

Fig. 6-7. Spiral bevel pinion of 4820H steel, 0.6×.

Physical appearance of gear. All the teeth are intact and highly polished along the top of the profile toward the toe end of the convex (forward) side. There is some scuffing low on the profile of the concave sides.

Physical Examination

Magnetic-particle inspection. There is no evidence of tooth bending fatigue on either part. The several cracks indicated are associated with the spalling surfaces on the concave sides of the pinion teeth. The gear teeth show no indication of fatigue.

Tooth characteristics. The original setting test charts taken on this set in December 1969 are on file and document a very good and normal tooth contact pattern. The secondary tooth pattern that prevails at this time could not be duplicated on the gear tester. The pinion appears to have moved away from the center of the gear, thus activating a very heavy low-profile contact toward the toe end.

Surface hardness tests. The surface hardness of both parts is 58–59 HRC, which is within the specification of 58–63 HRC.

Metallurgical Examination

The rippling (Fig. 6-7a and b) is extensive and covers the entire area of the initial contact pattern. Rippling reflects movement or a tendency to move. The movement can be superficial adjustment of a surface heavily rolled while sliding; or it can be an adjustment of the surface during absorption of energy by retained austenite. Rippling is not always associated with a failure, nor is it detrimental in itself. It does usually indicate, however, a rolling/sliding condition that is highly compressive.

The pitting is originating in a localized area, and the spalling appears to be a contiguous event, although this type of spalling could easily have been found as an independent mode without the association of surface pitting. It is therefore necessary to section the pinion tooth normal to the active profile, near the spalled area shown in Fig. 6-7(b), and to prepare this section for a microscopic examination and a case hardness traverse.

Note: the surface shows no catastrophic movement; the "butterfly wings" are generally parallel to the surface but extend 0.027 in. below the surface; and microstructure is very fine acicular martensite retaining less than 5% austenite.

Fig. 6-7(c). Section normal to surface of tooth profile taken very near the spalled area shown in Fig. 6-7(b), 200×.

Microscopic examination. The core structure contains an equal admixture of low-carbon martensite and lamellar pearlite. The case structure displays a very fine acicular martensite retaining less than 5% austenite. The pitted area low on the active profile shows evidence of a heavy metal-to-metal sliding action tending to be adhesive. The central profile area, subjected to spalling, has a surface that is relatively unaltered (Fig. 6-7c). In the same area, there is a subsurface display of "butterfly wings" reaching to a depth of 0.027 in. from the surface.

Case hardness traverse. Figure 6-7(d) is self-explanatory as to the process specifications and the results obtained.

Final Analysis

The material and process specifications have been met satisfactorily. The primary mode of failure was rolling contact fa-

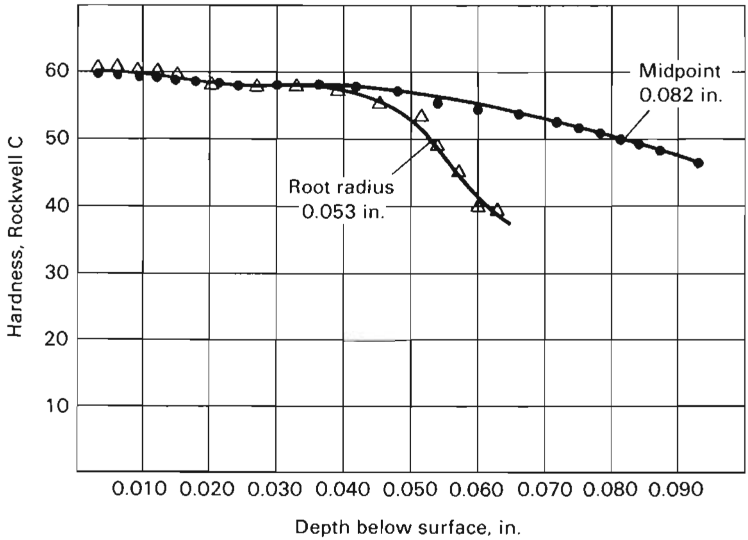


Fig. 6-7(d). Case hardness traverse of section used for Fig. 6-7(c) taken from tooth pictured in Fig. 6-7(b). Results are self-explanatory.

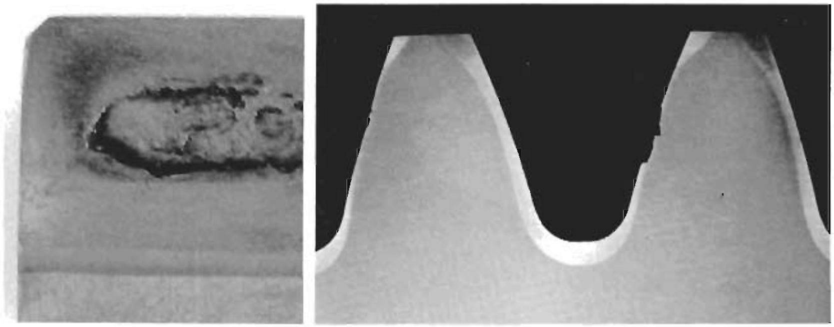
tigue of the concave (drive) active tooth profile. The spalled area was a consequence of this action. The pitting low on the profile appears to have been originating after the shift of the pinion tooth away from the gear center. The shift of the pinion is most often due to a bearing displacement or malfunction. The cause of this failure appears to be continuous high overload that may also have contributed to the bearing displacement.

Spur Gear

A portion of two large spur tooth bull gears that had spalling teeth was submitted for evaluation (Fig. 6-8a).

Background Information

The gears were taken from a final drive wheel reduction unit of a very large open-pit mining truck. The driving pinions were not submitted; the usual assurance was given that they were in good condition and were still in operation. Apparently,



(a) (b)

Fig. 6-8. Spur gear tooth, SAE 4147H, quenched and tempered to 311 HB, machined completely, induction hardened with a tooth space inductor by traversing one tooth space at a time. (a) Surface spalling along one tooth flank. (b) No hardened case on active profile of one side of the teeth.

these two gears had been operating for relatively few hours. The loads were considered to be maximum for the operation.

These parts were to have been SAE 4147H steel, quenched and tempered to 262–311 HB, finish machined, induction hardened (both the spline and the spur teeth), and tempered at 350 °F to a specified hardness of 55 HRC minimum on the hardened surface.

Visual Examination

Identification markings. None on either part; the markings were on the portion not returned. The part number had been submitted on the shipping papers.

Physical appearance. The two parts were identical. Four contiguous teeth had a spalled area on one side only at midprofile but for varying lengths (Fig. 6-8a). They were not consistent. The nonspalled teeth showed a very good surface on the same side, and all of the teeth were in good condition on the opposite side. There was no evidence of adhesive or abrasive wear.

Physical Examination

Magnetic-particle inspection. There were no indications other than those associated with the spalled areas.

Surface hardness testing. The ends of the teeth were at 311 HB (specified, 262–311 HB). The surface at the root was 58 HRC (specified, 55 HRC min).

Nital etching. The end faces of the gear teeth were polished and nital etched for a view of the hardened area. (This viewing area is the same area examined by the induction hardening operator on each production part to maintain consistency.) The end face of all submitted teeth showed a consistent hardened pattern along the full profile and around the root fillets. From this vantage point, they would have been acceptable.

Metallurgical Examination

Up to this point, no incriminating evidence has been found; but there is a need to find out what has happened in the spalled areas. The consistency in the spalled pattern is that every spalled tooth has a spalled area 1½ in. from one open end. Therefore, both eight-tooth sections should be cut parallel to and at a plane 1½ in. from the end face. The freshly cut plane will then be nital etched for a macroscopic examination.

Macroscopic examination (Fig. 6-8b) of the nital-etched surface revealed that the spalled areas retained very little, if any, induction hardened case. The compressive loads originated the failures at the case/core interface and at the terminating junction of the surface hardened zone. Also, along the profile of the teeth that had not failed, the hardened case was as shallow as 0.020 in.

Chemical analysis confirmed the material to be SAE 4147H, as specified.

Final Analysis

The parts had met the material and initial heat treat hardening specifications. The mode of failure was tooth profile spalling. By definition, spalling originates at a case/core interface or at the juncture of a hardened/nonhardened area.

The cause of this failure was either insufficient or no induction hardened case along the active profile. The cause was activated by a nonfunctioning induction hardening coil that did not or was not allowed to harden the midprofile of several teeth. The teeth had been induction hardened by the single-tooth method, wherein the adjacent profiles and included root, defined as a tooth space, are heated by a tooth space inductor traversing the length of the tooth. Continuous cooling and quench flow must be maintained very closely; any interruption will result in an improperly hardened pattern. Since the hardening pattern was uniform on most of the teeth, the problem appears to have been related to the flow of cooling water. When noticed by the operator, the condition was corrected; but the induction machine operator's error was failing to have the gears checked closely for a lack of a hardening pattern on those few teeth. Next in line was the nital etch procedure set up by the inspection department, which also failed to discover the condition. Though the problematic condition was overlooked during two other stages, the cause ultimately goes back to the heat treatment process. This failure was determined to be an isolated case and did not warrant a recall of all the gears in operation.

Hypoid Pinion

The hypoid pinion shown in Fig. 6-9(a) was placed on the failure analyst's desk with the usual plea for assistance. To the analyst's question of where the mating part was, the response was that there was nothing wrong with it, so it was still in service. Please note: Those who disassemble units must recognize the importance of observing every drive gear set to determine if it had been mated properly. Too many times, new parts are

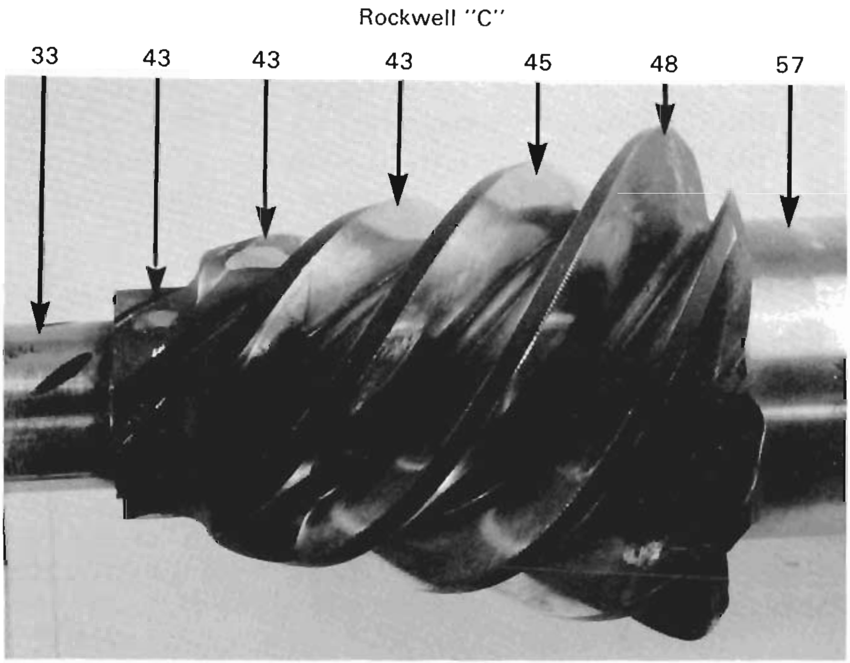


Fig. 6-9(a). Hypoid pinion, ¾×. Mode of failure was a deep wear pattern toward the toe end. Note the surface hardness pattern that was the result of an improper quench.

placed in operation with apparently "good" used parts; but the tooth characteristics of one may be absolutely wrong for the other. Subsequent failure of the second set may be exceptionally rapid. If mismating occurs, it is very difficult to determine responsibility; but no gear manufacturer should be responsible to warrant any serialized part that has been mismated.

Background Information

The hypoid pinion was the driving member of a power unit operating a rapid transit car. The pinion had been removed from service at the end of the initial test period because it showed undue wear. (Keep in mind that transit gearing runs under full load in both directions.)

Visual Examination

Identifying markings:

PART 4820 6-43 J-7 K7-97

PART	Part number (not disclosed).
4820	SAE grade of material.
6-43	Set ratio; pinion 6 teeth, gear 43 teeth.
J-7	Manufactured in October 1967.
K7-97	Mated with a hypoid gear as set No. 97 in November 1967.

Physical appearance. The contact pattern on the concave (forward) sides shows no wear at the heel end, but the wear increases uniformly to a deeply worn area near the toe end. It appears to be due to heavy friction. The convex (reverse) sides of the teeth show a more uniform contact area that is scuffed the full length of the pattern. The entire pinion head is black due to a heavy coating of iron-manganese-phosphate.

Physical Examination

Magnetic-particle inspection revealed no indication of cracking.

Surface hardness testing with a test file revealed a variation in hardness from "file hard" on the head bearing to "very soft" on the pilot bearing. The various HRC checks are indicated in Fig. 6-9(a).

Metallurgical Examination

This examination is used to determine what type of wear has occurred, and what has caused the existing hardness pattern. A section of a tooth at midlength is selected and prepared for a case hardness traverse and microscopic examination.

A case hardness traverse (Fig. 6-9b) revealed a definite tempering of the surface, along with a decrease in hardness throughout the case, which influenced the effective depth. Also, the midprofile had been affected more severely than the root radius. And the core hardness, 38 HRC, is typical of a good quenching and tempering process for the tooth root but is about three hardness points low for what would have been expected at the midpoint.

Microscopic examination of the tooth section revealed that the core structure (at the midpoint) contained very fine tempered low-carbon martensite and pearlite. The case structure displayed very fine tempered martensite throughout, and no retained austenite. And the surface structure showed no evidence of frictional distress; it had been cleanly cut by abrasion.

Final Analysis

The mode of failure was severe abrasive wear. The cause of failure was insufficient surface hardness. A question arises concerning what caused the low hardness.

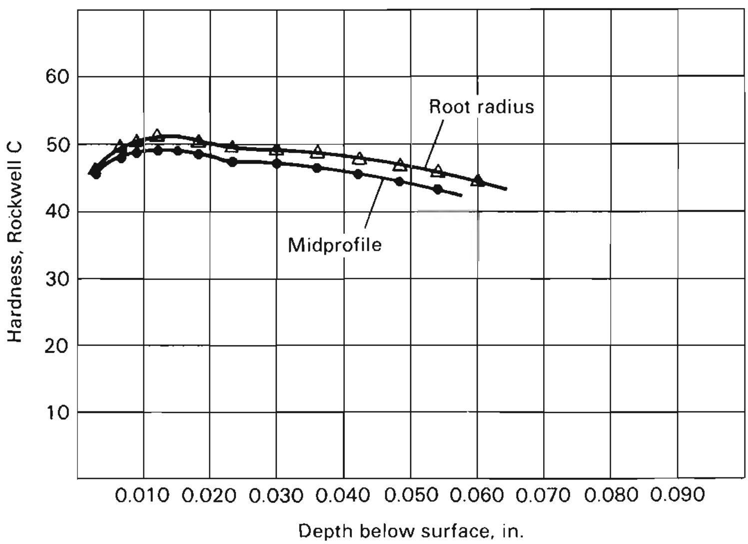


Fig. 6-9(b). Case hardness traverse used to diagnose the problem of the hypoid pinion shown in Fig. 6-9(a).

The basic quenching was acceptable, because the core hardness and microstructure at the root are indicative of a well quenched and tempered product. Could grinding of the teeth have caused this drop in surface hardness? It might have on the surface of the active profile, but not on the surface of the root radius that had not been ground. Also, grinding would not have influenced the entire depth of case or the core hardness at mid-point. There appears to be only one logical answer: A delayed quench would have allowed the surface and smaller sections to cool somewhat below the critical quenching temperature yet still have been quenched soon enough to have allowed the core at the tooth root and the more massive portions to be above the critical quenching temperature.

Further investigation revealed the possibility that a tray loaded with pinions had been stuck midway into the quench for a time until the mechanism had been manually dislodged. Thus, the primary cause of failure was improper heat treatment.

The documented records showed the total number of pinions placed on each furnace tray. A service recall for the remaining pinions was immediately initiated. The instructions were simple: A common file could be used to quickly identify each soft pinion without removing or disassembling any unit. The recall was successful and all the pinions were accounted for.

CHAPTER 7

Writing the Report

No single format for a report can serve as a working model for all writers or all readers. With certain organizations, one may be required to follow a specific format when writing or submitting a report, though generally this is not the case. It is usually through give-and-take that a format is conceived that is acceptable to both writer and reader.

A report reflects the personality of the writer as well as his objective. A report may be factual and bare, written by someone who hates to write. It may be detailed to the "nth" degree, flowery with nonessentials, written by someone who enjoys his own writing a bit too much. Or it can be concise and complete, making its points with clarity, written by someone who has a genuine ability to communicate.

To communicate properly is an art, but one that can be learned. For instruction in learning to write a report, the guidelines discussed by Harry E. Chandler in Chapter 10 of the *Technical Writer's Handbook*[1] are highly recommended.

Since a systematic approach to analysis has been continually advocated in the preceding chapters, it is no less important that report writing be kept in the same perspective. From the onset of a failure in the field operation, there are five occasions that require written accounts. Each is very important to the finalization of the incident and each has its unique purpose. In order of their usual occurrence, they are: field report, transmittal report with failed unit, laboratory notes of the analyst, failure analysis report, and letter of transmittal of the report to the customer.

Field Report

Before proceeding with the field report, review very closely the instructions given in Chapter 3 concerning field examination. The field report should document all facets of the examination and answer all questions raised in that section.

Following a particular format in the field report is not as important as making a record of all information pertinent to the field failure. Of course, it would be most helpful to have a well-written, well-documented field report, and this should be the goal of the field representative. This leads directly into the second document.

Transmittal Report With Failed Unit

The transmittal report is an opportunity for the field representative of either the user of the original equipment or the supplier of the equipment (who also might be the original equipment manufacturer) to interpret the field report, to place it in coherent order, and to add detailed information that may not have been available to the field service personnel. Such information might be the model and serial numbers of the equipment, the purchase date, the hours or mileage of operation, the maintenance history of the failed unit, and other related facts not documented in the field report. If necessary, the field representative should consult with the field service personnel (or the maintenance personnel) who submitted the field report, in order to establish all facts pertaining to the specific incident and to submit a complete report. This report will also inform the vendor as to the reference numbers to the incident, what the failure appears to be, what (if any) is the probable cause, what is expected in return (i.e., a report, an evaluation, a warranty claim, credit, etc.), and the name of the person to receive the correspondence pertaining to the incident.

Laboratory Notes of the Analyst

The laboratory notes are the accumulation point for all previous information and for all forthcoming observations that may have any bearing on the failure incident. The first recommendation is to use a permanent notebook; loose leaves of paper are of no value if misplaced or lost.

Start the entry with the date of receipt and quickly add the examination reference number, the customer's name, and the part numbers involved. Continue with the background information and cross-reference numbers. From then on, record all observations made while following closely the procedures outlined in Chapter 3, under "Visual Examination," "Physical Examination," and "Metallurgical Examination."

Do not hesitate to note any condition or characteristic that is observed. At the time of the observation, such an item may not seem pertinent, but before the final analysis it may have a profound relationship to the failure. And even if an observation proves to be incidental, no harm has been done by including it in the notes.

The examiner may be a laboratory technician or the analyst himself. The examiner's notes may be profuse or abbreviated, but they must be an accurate record of what was observed. Admittedly, two examiners may interpret an observation differently, but the physical observation should be recorded in a reasonably recognizable manner. The notebook should be compiled and written such that another analyst can determine from the data what was done and how the information was obtained. It should be a clear and complete record for use by any experienced analyst.

Although the analyst's actions of procedure should be similar to those given in Chapter 3, he must not lose sight of two very important mental inputs. The field report and the transmittal report may have initiated a thought of a possible cause and effect. He should not dismiss this thought immediately but cautiously proceed to establish facts which may, in themselves, either strengthen or eliminate the preconceived input. In addition, as the analyst proceeds with the routine of visual, physical, and metallurgical examination, he may think of possibilities for cause and effect. Each one of these thoughts should be a motivation for continued observation until a final conclusion can be made.

There is an importance attached to the examiner's laboratory notes that has not often been recognized; namely, their legality. Laboratory notes are often used as court evidence; but to be admissible, they must be documented, and to be correctly documented, each page must be dated and initialed by the examiner at the time the page has been completed.

Failure Analysis Report

The failure analysis report is the crucial report. The analyst must place all observed data in context and present the data in a scholarly manner that will be understandable, helpful to the customer, and conclusive about mode and cause of the failure. Although there are many formats used for these reports, the failure analysis report must incorporate certain basic elements. Therefore, the following format is suggested: a heading, followed by general background; examination procedures and results; and a closing.

The heading should be concise, giving only the information necessary for quick and efficient reference; namely, report number, date, customer's name, and part or parts involved.

General background should briefly state only the information known to have had a bearing on the incident. This includes the customer's reference numbers, the operating data from the transmittal report, the customer's reasons for returning the failure, and the feedback required.

Examination procedures and results include the requirements established in Chapter 3 under the headings of visual, physical, and metallurgical examination. The visual examination coverage may be the most detailed due to the nature of the observation; whereas, only the results of the physical and metallurgical examination may be reported. Since this is the main body of the report, it is essential that enough information be given to allow the reader to follow a logical path through the procedures to arrive, with the analyst, at the same result. The entire body of this report should be in a step-by-step, logical sequence of data presentation, written concisely, clearly, and accurately. It should require a minimal amount of writing and a minimal amount of reading.

The closing of the report is not a routine appendage, but it most definitely fulfills a requirement. The closing can take the form of a summary, discussion, conclusion, recommendation, or evaluation. It may be only one of these or a combination of two or more. The author must keep in mind the objective of the report and use the closing that fulfills the need of that objective. He should also know which of the closing forms will be covered by the letter of transmittal.

A summary is a shortened version of the important thoughts from the body of the report, such as the objective, the most significant findings, and the main ideas presented. A summary used alone as a closing should appear when no conclusion can be drawn from the results, or when the purpose is only to present data. A summary is generally used to precede one of the other four closings, to set the stage in a logical order for a proper finale.

The purpose of the discussion is to give the author an opportunity to analyze the significance of the results. Keeping in mind the objective of the report and the interest of the reader, the author may expand on the facts of the incident, the observations made during the procedures, and the variations within the data. A comparison with other incidents may be made at this time, as well as a directive for possible future observations or tests.

Conclusions are the reasoned judgments made on the basis of the results and findings of the overall examination. A conclusion must always be supported by the data in the report. It must be written clearly and concisely, and if there are several conclusions, they must be given in a logical order. The analyst should keep in mind that a negative conclusion is still a valid conclusion.

Recommendations should be based on the conclusions. Again, recommendations should be short and to the point. They should be clearly understood by the reader and contain answers to his questions. In other words, the objective of the report is answered by the recommendation. It is also under this heading that corrective measures are suggested.

An evaluation is a type of discussion that specifically determines the significance or net worth of the results obtained during the examination. It discusses the effectiveness of answering the "why" questions brought up by the customer in relation to the incident; and it discusses the relative merit of various corrective measures that could be recommended. In general, the evaluation is not the closing portion of the report, but it is the main context for the letter of transmittal of the report to the customer.

Letter of Transmittal

The shortest letter of transmittal will be, for example: "Accompanying this letter is our report No. 750, which is self-explanatory." The assumption of this letter is that report No. 750 is complete regarding conclusions, recommendations, and/or evaluation. However, if the report is not complete in these respects, the letter of transmittal should be expanded to include the necessary closing. The author of the report and the author of the letter of transmittal should be in communication before the report is concluded, to determine who is to do what about the closing.

It is essential to be realistic, first concerning the responsibility of the analyst who is also the author of the report. If the analyst also authors the letter of transmittal, the report should close in a logical manner and the letter used to expand or emphasize. If the analyst is not the head of the department, but the letter of transmittal is to be sent by the department head, the analyst may close his report to answer the objective and allow the letter to range from one simple sentence to a lengthy evaluation, with recommendations as necessary. If the letter of transmittal is to be sent by a member of the sales department, in most instances the report must be complete in every detail.

Also, the author of the letter of transmittal must read the report carefully. Have the objectives been answered? Check for conclusions and recommendations. Are they clear and concise? Are they in logical order? Are they clearly supported by the results in the body of the report? Are there any other conclusions or recommendations that come to mind as you study the report? If there need to be additional emphases or expanded recommendations, the letter of transmittal is the only means available to bring the incident to a final conclusion.

The letter of transmittal serves a very important function unique among the five documents. The four previous accounts consider only the facts of the incident and are not usually cognizant of the individual reader. Not so with the letter of transmittal, which is written to an individual and is, in most instances, a personal letter. It is therefore necessary for the author of the letter of transmittal to be aware of the needs of the recipient and very carefully supply all information necessary to satisfy that individual. For this reason, the letter should be authored by the person (metallurgist, engineer, or salesperson) most closely associated with the recipient of the report.

Ethical Overview

An observer may often be influenced by previous observations or by a certain desire to see things that simply are not there. For this reason, it seems expedient to discuss some points of ethics (or just good etiquette) not as criticism, but to emphasize character. In the act of studying a failed part and in passing information on to others, it is of utmost importance that three characteristics are kept in plain view: accuracy, clarity, and lack of bias.

Accuracy is in the eyes of the beholder. Whatever is seen should be explained in clear and simple detail, understandable to the reader. A catastrophic failure in the field of operation requires a first-hand report from the operator. The field personnel examining a failed unit needs to accurately portray what he sees. The field representative must be exact in submitting details to the analyst. The analyst must make a sincere effort to piece the story together and run enough tests to determine a clear-cut answer.

No action is more embarrassing than giving a customer a first-sight opinion, allowing him to accept it as fact, and later finding that the "obvious" was all wrong. The examiner should be certain that enough information is readily available to be convincing both to himself and to his customer.

Clarity is the quality of being clear, free from confusion, and simply understandable. The aim of the author of any of the printed documents discussed earlier should be clarity, not only for the author's benefit, but for the benefit of those persons yet to read the account. In field reports, the language might be gross and the grammar atrocious, but there must be no doubt about the clarity of information. A failure analysis report may by necessity contain metallurgical or engineering terms not wholly understood by the layman; but the discussion section of the report should make them understood. It is detrimental to the clarity of a report for its author to attempt to "blind people with science."

As is mentioned in Chapter 5, when a gear set failed by tooth bending fatigue due to sudden overload, the analyst admitted to having thoughts somewhat biased since he had recently watched this same model front-end loader being used abusively. Another type of bias occurs when an analyst wants to find a cause in one direction or, perhaps, does not want to admit the direction of an apparent cause. Any biased opinion is an immediate barrier to true evaluation and corrective action.

An analyst must be aware of the incident before him and careful not to introduce irrelevant information that will confuse the issue at hand. A dimensional or process characteristic that may be slightly out of specification, having nothing to do with the incident, should not be stressed beyond its immediate importance. A specification is established by people using their best intentions to comply with the existing requirements.

A cause of failure should never be that it does not meet the specifications; but rather, the cause should be discussed in light of these questions: Would the failure also have occurred if the specification had been met? Should the specification be changed to accommodate the existing situation, or should the situation be changed?

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.

Reference

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

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NOTE. The parenthetical letters (F) and (T) indicate that information is presented in a figure or a table.