



Materials for Helicopter Gears: Report (1979)

Pages
102

Size
8.5 x 10

ISBN
0309333091

Committee on Helicopter Transmission Gear Materials;
National Materials Advisory Board; Commission on
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BIBLIOGRAPHIC DATA SHEET	1. Report No. NMAB-351	2.	3. Recipient's Accession No.
4. Title and Subtitle			5. Report Date October 1979
7. Author(s) Committee on Helicopter Transmission Gear Materials			6.
9. Performing Organization Name and Address National Materials Advisory Board National Academy of Sciences 2101 Constitution Ave. Washington, D.C. 20418			8. Performing Organization Rept. No. NMAB-351
12. Sponsoring Organization Name and Address Department of Defense and the National Aeronautics and Space Administration			10. Project/Task/Work Unit No.
15. Supplementary Notes			11. Contract/Grant No. MDA 903-78-C-0038
16. Abstracts Some of the power train transmission gears in helicopter drive systems can become critical components as performance requirements are increased; accordingly, increasing attention must be paid to new alloys in order to obtain required performance, reliability, and survivability. The major limitation of the alloy presently used, SAE 9310 steel, is its tendency to score and scuff under high-temperature conditions. Candidate advanced alloys, with improved high-temperature properties, while increasing the resistance to scoring and scuffing, tend to have lower ductility and fracture toughness. In this report, an attempt is made to identify design, materials, and process problems and requirements. In addition, it is recommended that the characterization of candidate steels be accelerated; preliminary investigation indicates that new alloys may provide improved capability against surface distress. Other short- and long-term recommendations also are presented.			13. Type of Report & Period Covered Final Report
14.			
17. Key Words and Document Analysis. 17a. Descriptors Carburizing Fatigue Gears Helicopters Scuffing Spalling Transmission 17b. Identifiers/Open-Ended Terms 17c. COSATI Field/Group			
18. Availability Statement This report is for sale by the National Technical Information Service, Springfield, Virginia 22151.		19. Security Class (This Report) UNCLASSIFIED	21. No. of Pages 89
		20. Security Class (This Page) UNCLASSIFIED	22. Price

MATERIALS FOR HELICOPTER GEARS

Report of

The Committee on Helicopter Transmission Gear Materials

**NATIONAL MATERIALS ADVISORY BOARD
Commission on Sociotechnical Systems
National Research Council**

**Publication NMAB-351
National Academy of Sciences
Washington, D.C.
1979**

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This study by the National Materials Advisory Board was conducted under Contract No. MDA 903-78-C-0038 with the Department of Defense and the National Aeronautics and Space Administration.

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ABSTRACT

Some of the power train transmission gears in helicopter drive systems can become critical components as performance requirements are increased; accordingly, increasing attention must be paid to new alloys in order to obtain required performance, reliability, and survivability. The major limitation of the alloy presently used, SAE 9310 steel, is its tendency to score and scuff under high-temperature conditions. Candidate advanced alloys, with improved high-temperature properties, while increasing the resistance to scoring and scuffing, tend to have lower ductility and fracture toughness.

In this report, an attempt is made to identify design, materials, and process problems and requirements. In addition, it is recommended that the characterization of candidate steels be accelerated; preliminary investigation indicates that new alloys may provide improved capability against surface distress. Other short- and long-term recommendations also are presented.

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Chapter 1

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

A. SUMMARY

Some of the carburized steel gears used in helicopters and other aircraft can be critical components. The maximum performance gears in helicopters operate under higher speeds, loads, and surface temperatures than most other gears and are prone to scoring, surface pitting, and tooth bending fatigue. Gear tooth scoring occurs under high speed and load conditions and increases significantly when the temperature exceeds the maximum recommended for the lubricant employed.

It has been common practice to specify SAE-AISI 9310 (AMS 6265) steel for almost all aircraft gear trains but these steels soften at high operating temperatures, and thus offer poor scoring and scuffing resistance. New high-temperature gear steels that offer improved scoring and scuffing resistance and an improved chance of survivability after oil interruption and are more tolerant to ballistic impact are being recognized for use at current operating temperatures (i.e., 200 to 300°F). Such properties will be required in the new transmissions being developed for operation between 450 and 600°F. Use of these new steels, however, may require additional development for the following reasons:

1. The high hot hardness, high-temperature gear steels tend to have lower ductility and fracture toughness than the SAE 9310 steels.

2. The factors that influence the fatigue threshold of these steels are not completely understood, thus limiting the ability of designers to predict gear behavior.

3. A constant controllable case hardening process (i.e., carburizing) has been achieved for some of the high hot hardness gear steels; may need demonstration for others.

4. The load-carrying capabilities at elevated temperature (above 300°F) of high-temperature steels and the conditions resulting from oil film loss and high frictional heat when the lubrication system fails have not been defined.

5. The statistical expressions for characterizing design allowable relationships under oil film loss conditions are well understood but additional data are needed to develop the required confidence in their validity.

6. A standardized test methodology that will permit development of a handbook to assist gear designers is lacking. It is recognized that we need standards to permit meaningful comparisons.

7. Existing data are insufficient in number and scope to permit correlation and substantiation of high-temperature operational gear performance.

Despite these problems, some helicopter manufacturers and Army program managers are committed to the use of the new gear steels; therefore, the Department of Defense (DoD) and the National Aeronautics and Space Administration (NASA) asked the National Materials Advisory Board (NMAB) to examine the issues involved and to develop guidelines concerning significant mechanical properties required of helicopter gear steels. To respond to this request, the NMAB appointed the Committee on Helicopter Gear Materials to:

1. Identify test procedures that reflect helicopter gear failure modes;
2. Identify design, materials, and process problems
3. Provide information on gear requirements, both near and long-term, and identify data needs; and
4. Establish a definite and comprehensive plan for advancing gear material technology to meet future requirements.

B. CONCLUSIONS

1. All commercially available modified tool steel compositions that provide high hot hardness by secondary hardening during tempering offer low ductility in terms of fracture toughness. The relationship of this mechanical property to successful gear operation is not well understood, but the high hot hardness of tool steels results in significant resistance to scoring.

2. Threshold values relate the size of flaws to nonpropagating stress levels and are important criteria for assessing cleanliness of steel, steel selection, and method of manufacture. Where cyclic rates of loading are sufficiently high to grow fatigue cracks to critical lengths within the normal inspection interval, the associated component will fail in fatigue.

3. High-temperature steels with approximately two or more percent chromium must be preoxidized prior to carburization, or alternatively, vacuum carburized to offset the effects of high chromium, molybdenum, and vanadium content. These elements cause surface reactions, resulting in nonuniform carbon penetration during carburizing.

4. There currently is no industry-accepted standard carburizing technique for all the high-temperature steels

that will provide the processing uniformity needed in future testing programs.

5. The dependence of design on the residual stress condition is generally recognized. While carburizing typically provides a desirable residual compressive stress on the surface, this condition can be destroyed by abusive machining. To assure a desirable compressive stress, it is customary to shotpeen the component. Full understanding of the factors that control magnitude and distribution of residual stresses through the processing cycle of a complicated geometry such as a gear is beyond our present capability.

6. Data concerning the relationship between single tooth fatigue, rotating bending fatigue, and actual gear performance at temperatures especially above 300°F are lacking. An expansion in scope of metallurgical testing to permit the definition of Weibull expressions of high confidence for gear design also is needed.

7. The operation of high performance gears requires clean steels that are free from inclusions to preclude flaws of critical size in the highly stressed sections of the gear.

8. Industry finds it necessary and convenient to use SAE 9310 composition gear steel as a frame of reference for design allowables and material and processing characteristics.

9. There is a general lack of agreement about whether the fatigue strength and toughness testing mode should be in bending. There was agreement that the specimen should represent the total case-hardened structure in a section size comparable to the actual gear.

10. Reaction to high temperature exposure prior to hardness testing can be used to assess material structural instability.

11. The slow-bend Charpy precracked specimen can be used to study both case and core fracture toughness characteristics, and the test allows for the incorporation of gear tooth metallurgical characteristics into the material.

12. A cantilevered bending impact test conducted at specified elevated temperature has been reported as a measure of fatigue, strength, and toughness effects as influenced by alloy composition, carburizing, and hardening.

13. Computer-aided gear design may be used in current high performance, low temperature applications but requires revision based on various properties and failure modes of the newer high alloy steels under consideration for high temperature applications. The current computer programs can be used with present steels to predict stress distribution through the case, microstructure in relation to cooling

rate, and response to rapid cooling and transient stresses induced during processing.

14. The development and standardization of high temperature oils that will maintain a film at high contact stresses is required for successful application of gears at temperatures above 300°F.

C. RECOMMENDATIONS

1. Characterization of high temperature gear steels should be extended and intensified to provide a statistical data base for operation at 200°F, 300°F, and 450°F utilizing established standard specimen size and geometry. Testing should cover bending fatigue (stylized samples representing gear and shaft), impact bending fatigue, tooth bending fatigue, crack growth under alternating stress, fracture toughness using slow-bend precracked Charpy specimens, threshold strength in the presence of a flaw, hot hardness, and tensile properties (including 0.1 percent proportional limit).

When elevated temperatures are involved, specimens should be exposed to the test temperature for at least 1000 hours before testing to assure structural stability of the material. When bending stress modes are applied, the test specimen should be representative of the case and core heat-treat processing established for the gear and shaft. The material hardenability must be well understood in terms of Jominy bar tests and interrelated cooling rates to assure compatibility of case, core, and section thickness of the represented gear. The design relationship of this subtle but major effect of residual stresses should be measured and understood.

2. Development work to qualify a specific gear design should include tooth bending fatigue testing, testing utilizing four square or "back to back" gear rig, Ryder rig testing for score resistance at specified operating temperature, and rolling contact fatigue testing.

3. The use of threshold stress intensity range ΔK_{TH} is a promising method for evaluating resistance to flaw growth under cyclic loading. This is a relatively new concept, and it is a subject of active research in many laboratories. Research on the effects of test conditions on ΔK_{TH} in a given heat of steel should be continued and should include temperature, environment, K_{MAX} , specimen shape, cyclic frequency, loading wave form, and the specific details of how ΔK is varied in the threshold regime. The reproducibility of the ΔK_{TH} of different specimens of one heat of steel tested under identical conditions should be checked. Further research is required to understand what aspects of material microstructure, composition, and

processing history have an influence on ΔK_{TH} values. Until a more precise understanding of the material properties and flaw growth have been established, the practice of reporting ΔK_{TH} of $x \pm y K_{IC} \sqrt{\text{in.}}$ in to some stated level of statistical significance is acceptable and necessary when one wishes to use the values for flaw tolerance criteria. (ΔK is the range of variation of the stress intensity factor in cyclic loading.)

4. An industry standard for steel quality acceptance should be established to control metallurgical composition and processing in terms of melting system and chemistry (to control tramp elements), ingot size, bar and billet size (to preclude "overforging" or "underforging"), and flaw size and quantity.

5. Nondestructive testing techniques (sonic and magnetic) should be developed to monitor the cleanliness of the steel in the bar form or in the semifinished forging and thus, to minimize all exogenous inclusions. A nondestructive method to evaluate case depth and hardness gradient in carburized areas should also be pursued.

6. Rig testing of actual gears operating with induced flaws should be expanded to provide a broader statistical base concerning the relationship of material properties to fatigue threshold. The scope of this testing should be limited to parametric correlations.

7. Research and development should be conducted to examine the significance of the Charpy impact energy and the relationship of blunt notch and sharp notch testing so as to provide possible guidelines on acceptable steel quality and to generate data on threshold stress intensity for fatigue crack growth so as to assist in design of gears.

8. In the area of scoring tests, a suitable screening procedure (using stylized samples) should be developed for material and associated processes.

9. New high-temperature steels should be screened as candidates and tested for mechanical properties significant to the gear user. It is recognized that definitive and accepted elevated-temperature laboratory tests that predict gear life have not been developed, but examination for probable acceptability is possible.

10. Long-term efforts to develop an advanced helicopter gear steel should, as a minimum, aim at:

(a) Ease of manufacture, from raw steel to finished form, at least equivalent to SAE 9310, including good response to electron beam and/or friction welding as a finished form.

(b) Providing hardening characteristics that will preclude the need for die quenching

(c) Providing strength at elevated temperature at least as good as X-2M and CBS 1000

- (d) Providing toughness characteristics at least equivalent to double vacuum melted X-2M
- (e) Providing fatigue strength at 600°F at least equal to SAE 9310 steel at 200°F
- (f) Providing scoring resistance at 600°F at least equal to SAE 9310 steel at 200°F
- (g) Providing ballistic tolerance in accordance with military requirements, including tolerance for foreign objects that could induce secondary failures
- (h) Establishing standard test procedures to provide a uniform method for assessing materials and processes
- (i) Employ the best resources (manpower and facility) of both steel producers and users and, using this plan as a reference, assess the state of the art of helicopter gear steel development at regular intervals under the cognizance of an independent organization.

11. Developing a gear lubricant compatible with all gear box components for application at bulk oil temperatures up to 600°F.

The accomplishment of the tasks itemized would provide the foundation for advancing to a new plateau in gear materials. The "plan" as outlined, would necessarily require substantial time and some additional funding to accomplish; items 1, 2, and 3 should be started first.

Chapter 2

INTRODUCTION

Some of the carburized steel gears used in helicopters and other aircraft can be critical components. The maximum performance gears in helicopters operate under higher speeds, loads, and surface temperatures than most other gears and are subject to scoring, surface pitting, and tooth bending fatigue operating conditions. Gear tooth scoring occurs under high speed and load conditions and increases significantly when the temperature exceeds the maximum recommended for the lubricant employed. Helicopter transmissions that will operate under strenuous conditions that inevitably increase temperatures to between 450 and 600°F are being developed in order to reduce the size of the oil cooler or by eliminating it entirely. Such a "high temperature gearbox" will require high hot hardness gear steels coupled with high temperature parts and lubricants. In addition, the need for survivability of the aircraft operating under reduced and/or starvation oil conditions is recognized.

The SAE 9310 (AMS 6265) steel that has been used softens at high operating temperatures (i.e., over 300°F) and scoring and scuffing resistance on the active gear tooth profile in some applications is poor; therefore, it has become necessary to examine some of the new compositions that through high hot hardness and high temperature stability provide improved surface scoring resistance. (Popiano and Kula, 1978; Jatczak, 1978b). Although these compositions satisfy high contact stress conditions, some have limitations that constrain their application in such areas as the uncarburized flange and web and shaft locations subject to high-cycle fatigue.

The present state of the art in gear design is lacking in precise definition of design allowables and mechanical property data concerning use at elevated temperatures; therefore, industry has no basis for identifying and quantifying future needs under the new high temperature and high load conditions. The present method of design iteration for achieving product acceptance involves designing, manufacturing, and testing the entire gear (either in rigs or in gear boxes) under simulated operational environment. For example, having established a gear tooth design through certain criteria, some of which are shown diagrammatically in Figure 1, it is common practice to test different materials and configurations to the extent

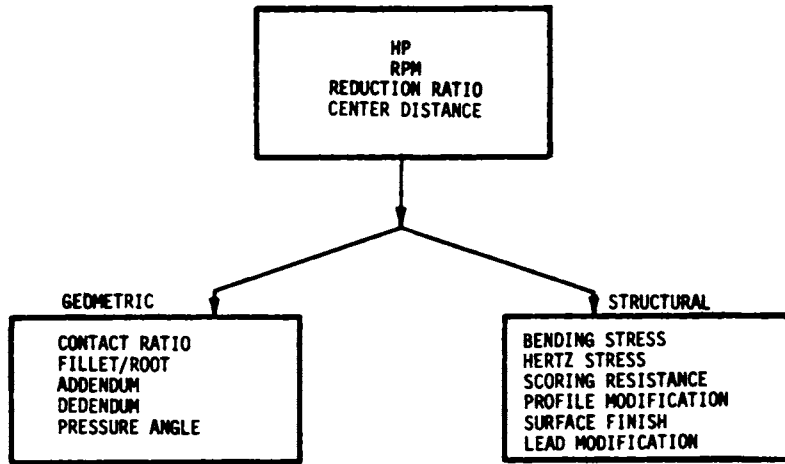


FIGURE 1 Diagrammatic illustration of special interaction relationships leading to the design of a gear tooth.

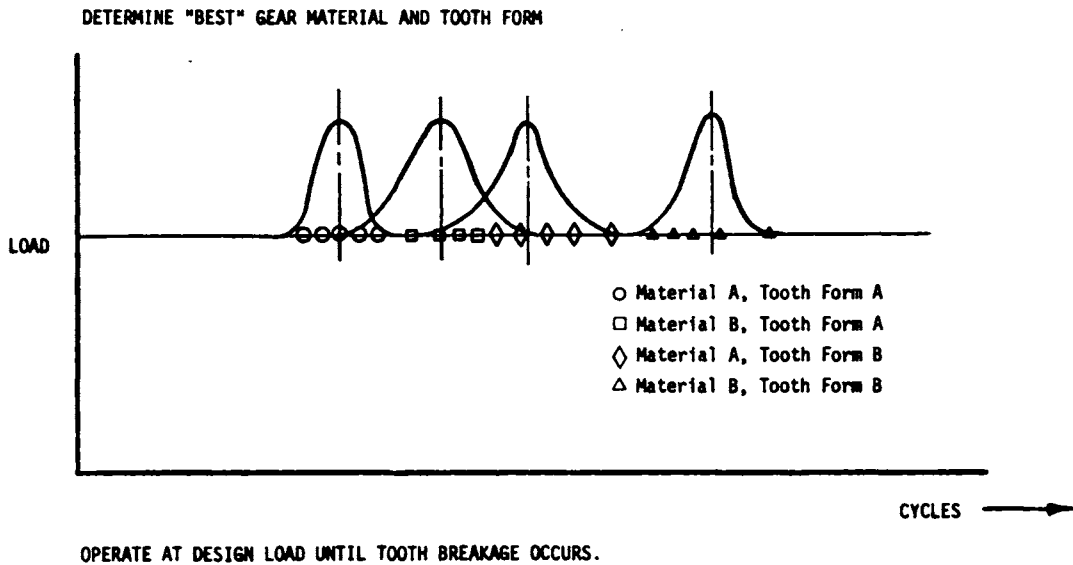


FIGURE 2 Diagrammatic illustration of preliminary test objectives to achieve a "best" gear material and tooth form.

shown in Figure 2. It also is common practice to develop a preliminary full S/N curve under the dynamic test conditions shown in Figure 3; however, full-scale qualification tests still are required to substantiate the final design of the helicopter transmission.

In actual testing during gear development, loads are selected carefully to provide early substantiation of satisfactory applied design stress and material strength distribution in the gear tooth. For example, Figure 4 shows two levels of Hertzian contact loading overlapping areas of low proportional limit compared to applied stress, thereby creating locale of potential failure due to repetitive plastic deformation. The drop in proportional limit is the result of high residual austenite (25 to 35 percent); however, because such undesirable variations in hardened carburized cases can occur, gear designs should be accommodating.

Regardless, the need to intensify the testing of the new class of high temperature steels for future applications is considered to be a precursor to following presently acceptable product test/design practice. It is important to recognize that selection of testing methods and the condition of the material (i.e., the case hardening process, morphology of final heat treat structure, the residual stress and prior high temperature exposure condition) should represent the gear's operational and service environment to the fullest extent possible.

CONDUCT TESTING ON 40 GEARS - OBTAIN 20 TEST POINTS
TESTING CONDUCTED AT FOUR DIFFERENT LOAD LEVELS
ESTABLISH MEAN S/N CURVE

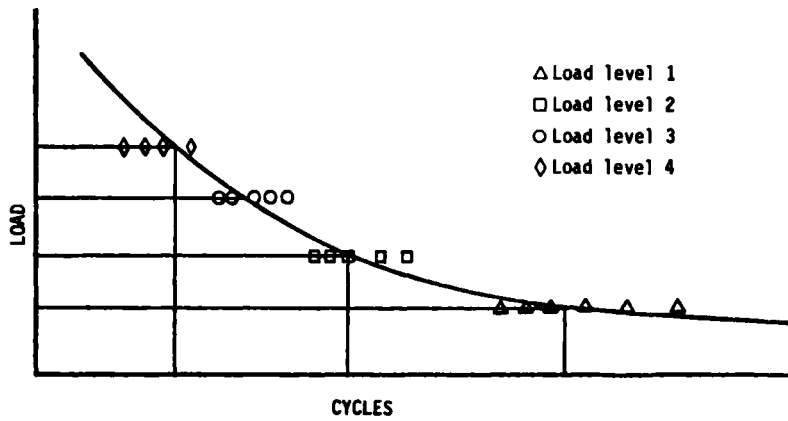


FIGURE 3 Typical required dynamic test for S/N curve relationship in the development of a new gear design.

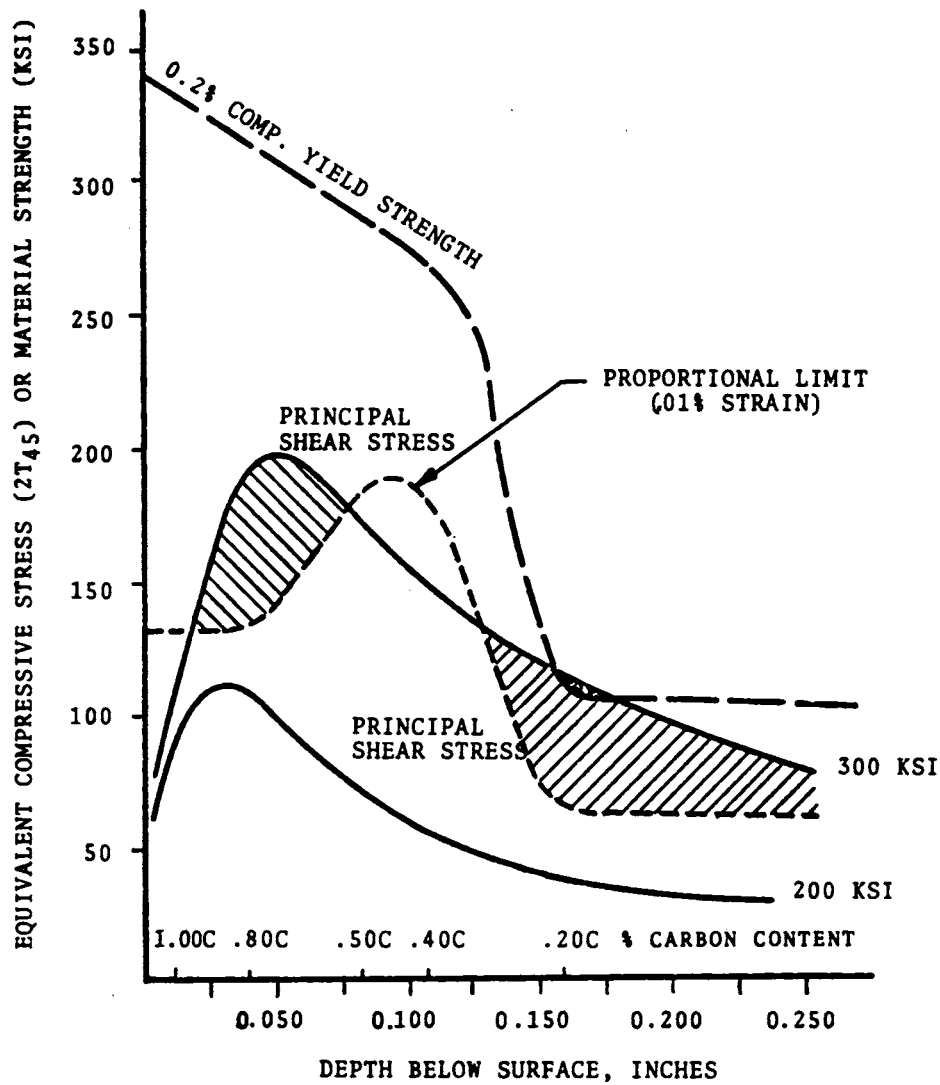


FIGURE 4 Comparison of material strength and stress profile in depth at two levels of Hertzian contact loading (200 and 300 ksi) in a large carburized gear. Shaded areas indicated probable regions for accumulation of plastic deformation with time. (From Jatzak, 1978a.)

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Chapter 3

GEAR FAILURES

A. DEFINITION

The ability to predict the behavior of gears in helicopter high performance gear trains depends primarily on the capability of engineers to analyze failures and provide correlating information useful in the design iterative process. Credible and accurate analysis already has provided the basis for design concerns that recognize:

1. The typical high torque and loads at stress levels which fluctuate from maximum to zero (Figure 5 illustrates the size and torque range of gears operating under these conditions);

2. The high speeds (7000 to 20,000 rpm) of turbine driven gears which generate a large number of fatigue bending cycles in a relatively short time; and

3. The maximum applied loads at reduced output speeds which represent the largest portion of life cycles for the final one or two stages of gearing.

Figure 6 is a presentation of data on generic modes of gear failure representing the performance in terms of total transmissions. The chart serves a useful purpose in comparing inherent gear failure modes. The absence of fretting as a failure mode in the chart is related to elimination of flange failures and including gear teeth failure only.

The comparatively high proportion of surface damage (pitted and spalled) observed in the UH-1 examination led to the conclusion that the EHD film thickness of a specific UH-1 gear fell below a critical minimum value and that this was instrumental in contributing to surface damage. The material released was in all probability a major cause for another major mode; debris damage. The authors of this report also note that many primary failure modes have been altered in later models of the transmissions studied because of product improvement changes. Thus, this data compilation should be regarded as a snapshot in time.

B. PRINCIPAL FAILURE MODES

Helicopter manufacturers agree that failure modes encountered repetitively during product development are tooth bending fatigue, pitting, scoring and scuffing.

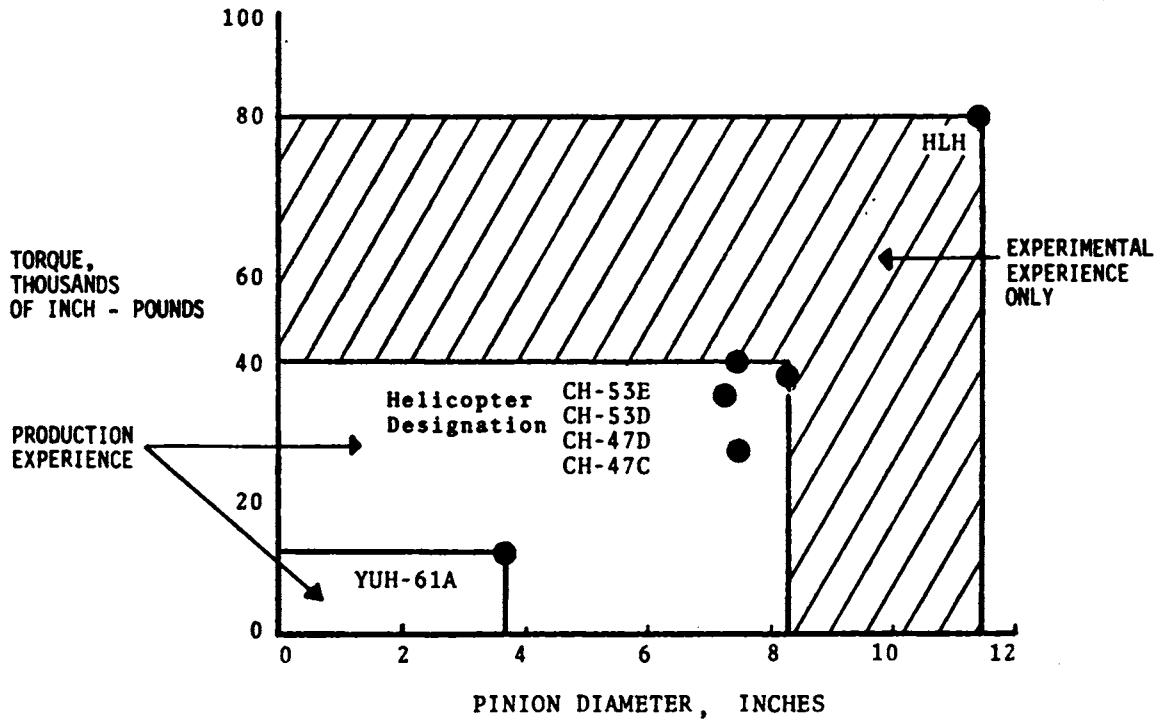


FIGURE 5 Helicopter transmissions size and power comparison.

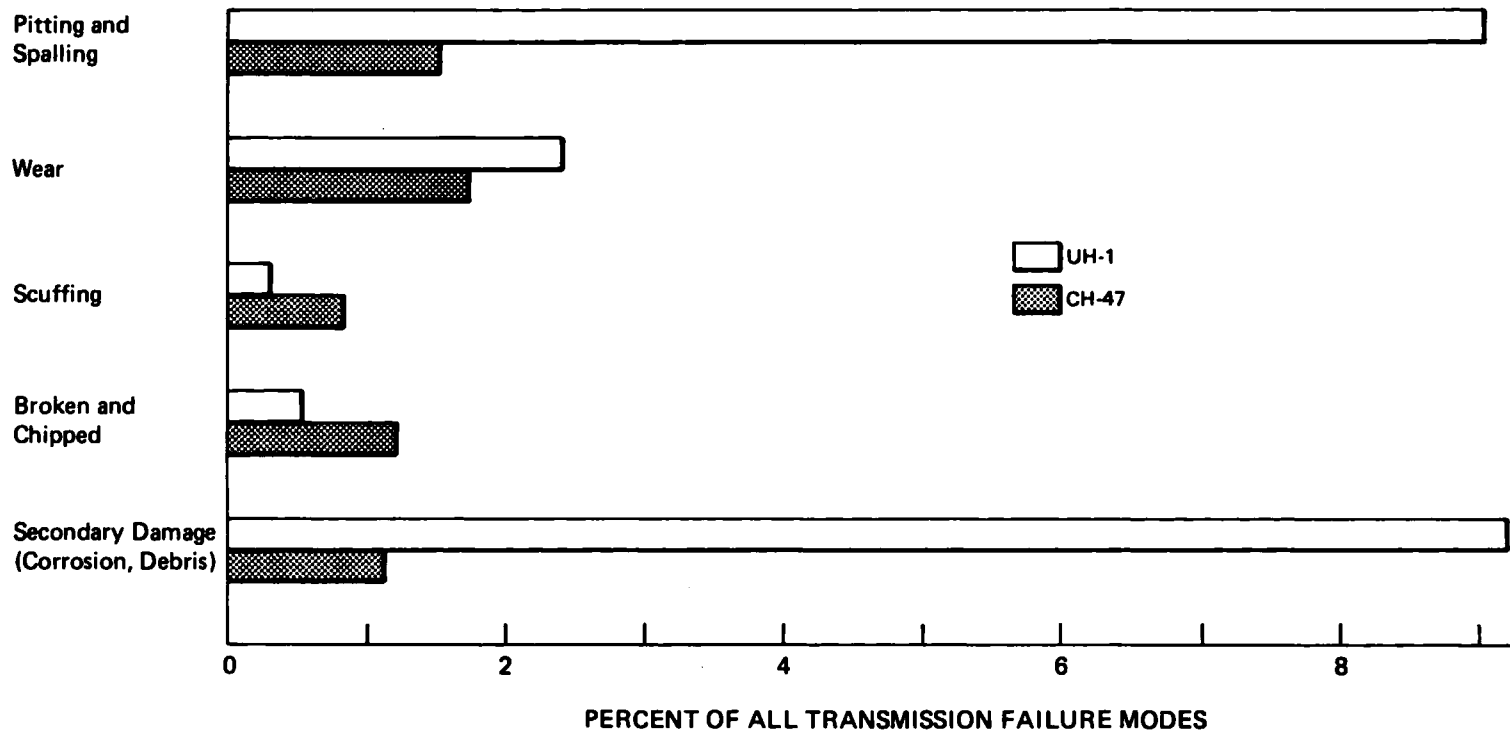


FIGURE 6 Distribution of Gear Tooth Failure Modes for Two Model Helicopters (from Overhaul data).

SOURCE: USAAMRDL Technical Report 73-58, Appendix VIII, Table XXXIX

1. Tooth Bending Fatigue

Tooth bending fatigue is well understood and is recognized as a design sensitive mode of failure. For unidirectional bending, the gear tooth rotates through a mesh from a zero stress condition, to a maximum stress condition, and back again to a zero stress condition on each revolution. The vibratory stress is one-half of the maximum and the critical area for bending is at the base of the tooth. Tooth bending stresses generally are calculated using the methods outlined by the American Gear Manufacturers Association (AGMA), and allowable bending stresses are based on this one-way bending mode. Allowable bending stresses also have been developed for case-carburized, SAE 9310 air and vacuum-melted steel gears and for non carburized steel gears heat treated to values ranging from 125,000 psi to 220,000 psi. The accepted formula for and a diagrammatic representation of the stress calculation is presented in Figure 7.

Helicopter gears usually are designed in bending for infinite life at the highest expected operating condition. A three-sigma reliability factor representative of approximately 1 failure in 800 samples generally is used. Some manufacturers produce gears with contact ratios greater than 2.0 (e.g., those found in spiral bevel gears) and maintain that a lower reliability factor can be used because the loss of one complete tooth will not cause the loss of the drive. Nevertheless, for design purposes, the material mean endurance limit curve of life vs. cycles (S/N curve) and the coefficient of variation (a measure of the scatter of fatigue test data) must be known. For the latter a value of 10 percent typically is used for gear steels.

While there is no complete agreement on the specific condition for qualification testing, load levels as high as 140 percent of the maximum expected in field service have been imposed. The probability of successfully passing the test may become unacceptably low. More commonly, recently designed transmission systems (including bearings) are required to have at least an 80 percent probability for survival under the "Overstress Test Spectrum" of loads and times. Since this statistical type of design involves finite life calculations of gear bending failure, the complete S/N curve must be available. The S/N curves for SAE 9310 case-carburized steel gears are presented in Figure 8.

2. Pitting

The tooth surface distress caused by high Hertzian contact stress on pinions normally progresses in fatigue at

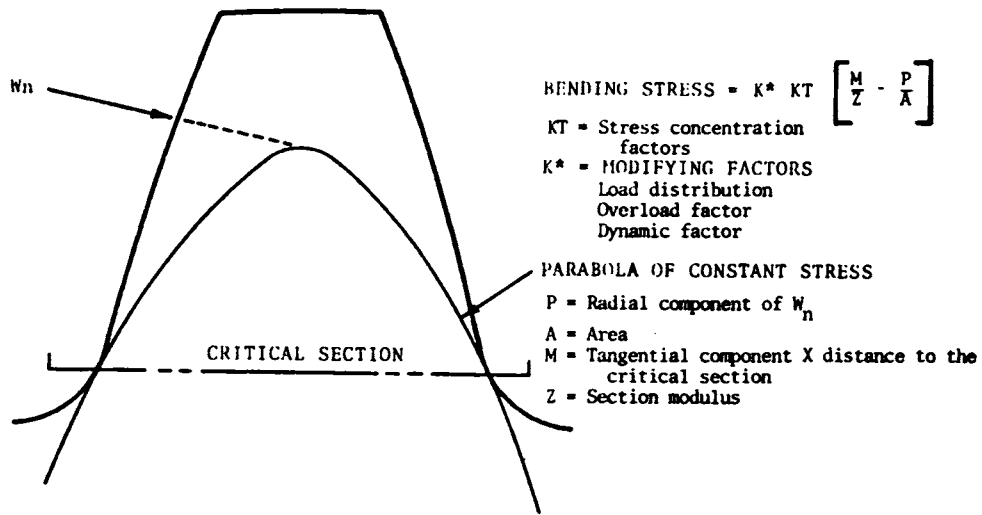


FIGURE 7 Gear Tooth bending stress.

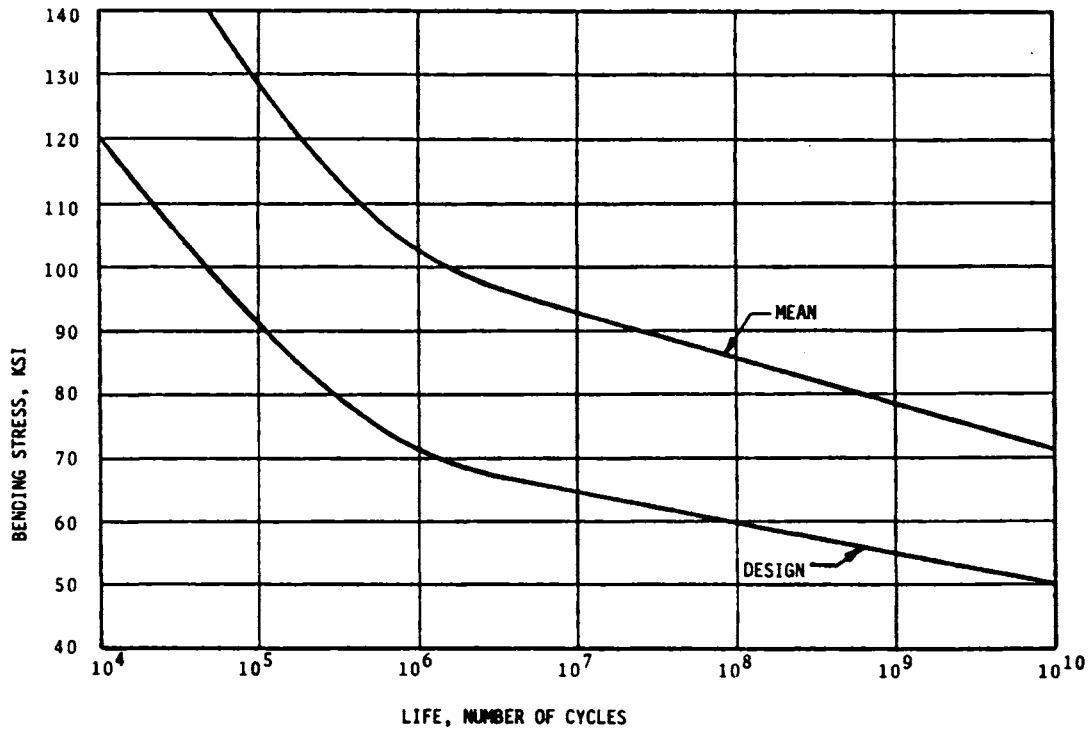


FIGURE 8 Fatigue bending S/N curve for spur or helical gears SAE 9310 melt - RC 60-64 case. (From AGMA-411.01. American Gear Manufacturers Association Design Procedure for Aircraft Engine and Power Takeoff Spur and Helical Gears. September 1966.)

the lowest point of single-tooth contact during the mesh cycle. In advanced stages, surface pitting can lead to loss of large chunks of the gear teeth and eventual loss of the drive. The AGMA analysis is accepted for design calculations of Hertzian stress, but a variety of methods are used for film thickness. Loads used for design of contact stresses usually are selected for maximum operating condition and infinite life. However, in gear life prediction, we have available contact stress vs life curves which have been developed around SAE 9310 steel permitting finite life calculations using the cumulative damage theory similar to fatigue bending.

Since contact fatigue is much more complicated than bending fatigue, the interaction of the many variables that define the operating environment of the contacting surfaces must be taken into account. These variables include temperature, contact stress, surface convergence velocity, amount of sliding, surface hardness, surface roughness, type of lubricant, lubricant viscosity, and, of course, material chemistry and microstructure. In the final analysis, these variables simply are conditions of metal-to-metal contact between two mating surfaces and are the principal determinants of pitting fatigue life. Three critical factors must be determined for pitting fatigue analysis:

1. Hertzian stress level vs cycles to failure;
2. Film thickness, roughness, ratio relationship to surface roughness; and
3. A Weibull expression of percent failures vs cycles to failure at constant contact stress and constant oil film/surface roughness ratio.

Geared roller test data may be coupled with specific analytical techniques in the same three sequential steps to provide a gear with compatible contact stress for the operating condition. These are presented in Appendix A.

For high temperature gear operation under conditions where frictional force is likely to occur because of lubrication breakdown, the existence of thermal stresses caused by high T from surface to subsurface is illustrated in Figures 9 and 10. These stresses and temperatures were developed from the following data for illustrative purposes:

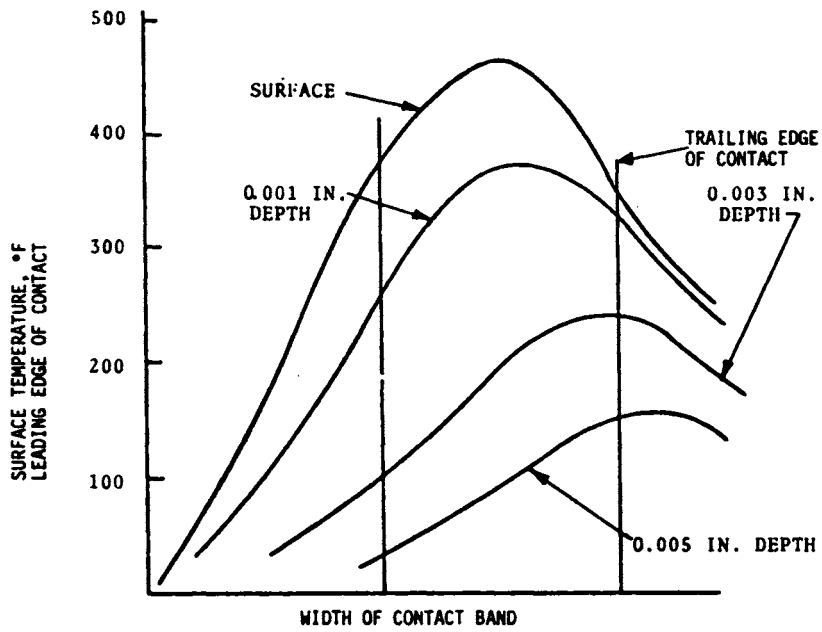


FIGURE 9 Gear tooth flash temperature on and below surface in terms of contact width. (From B. W. Kelly, Caterpillar Tractor Co.)

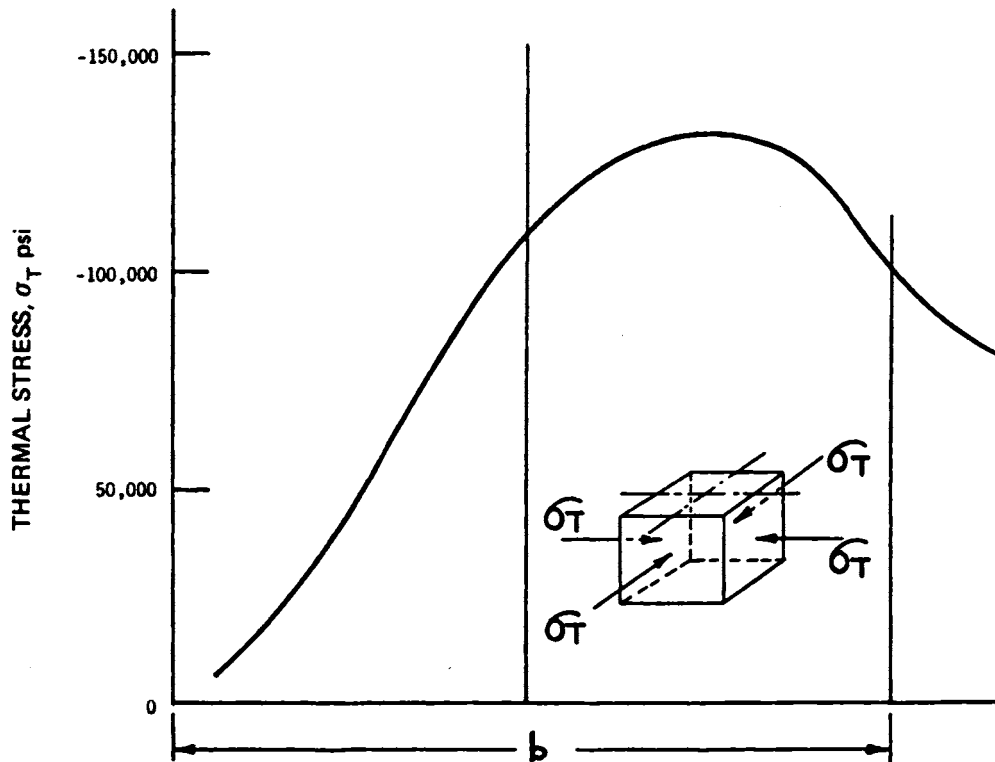


FIGURE 10 Gear tooth thermal stresses in the surface in terms of contact width. (From B. W. Kelly, Caterpillar Tractor Co.)

Roller radii	2.0 and 4.00 in.
V_1	100 in./sec
V_2	150 in./sec
W_n (load)	23000 lb-in.
f	0.1

The terms are explained under the formula below.

The 350°F ΔT in the outer 0.005-in. shell will cause this material to expand against the restraining subsurface material producing surface compressive stresses approximately proportional to ΔT . Since they are compressive in all directions and parallel to the surface, the further effect of superimposing these thermal stresses on the normal stress can create a stress shift. The trailing characteristic then significantly increases the maximum stresses on the negative sliding member; there is a lesser increase in stress on the positive sliding member. Since the tensile stress is retained on the negative sliding members, the example cited can produce additional compression surface stress on the negative sliding member of at least 20 percent. This is illustrated in Appendix B which carries the discussion further in dealing with the complex stress relationships on fixed planes and which can then relate the three differential compressive stress vectors causing pitting fatigue failures. While it is recognized that the stress levels used in the examples in this appendix are excessive, the discussion on "interacting forces" is a good explanation of pitting failure design/operating relationships. The committee agreed that possible effect of microstructural instability at high temperatures due to breakdown of tempered martensite and carbon resolution under high shear stress conditions should be carefully assessed.

3. Scoring Scuffing

Surface damage is one of the most difficult forms of failure to examine and understand because so many variables affect surface condition. The greatest number of these variables are handled by refinements to H. Blok's (1955) critical temperature model:

$$T_f = \frac{K_f W_n (V_1 - V_2)}{(C_1 \sqrt{V_1} + C_2 \sqrt{V_2}) \sqrt{b/2}}$$

where K = constant,
 f = coefficient of friction (assumed constant),
 W_n = normal load per unit length,
 V_1 and V_2 = surface velocities,
 C_1 and C_2 = material constants thermal conductivity,
specific heat and density
 b = width of band contact, and
 T_f = maximum mean surface temperature
("flash temperature" due to friction).

If $V = (V_1 - V_2)$, it has been found that the \sqrt{V} can be used to verify the temperature criteria for scoring on gears.

Variations in surface finish will affect the frictional response; therefore, an empirical value S for surface finish is introduced into the relationship to further differentiate the total surface temperature T_T and the bulk stable temperature T_B :

$$T_T = T_B + K \frac{f W_n (\sqrt{V_1} - \sqrt{V_2})}{(1 - S/50) \sqrt{b/2}},$$

where T_T = total surface temperature

T_B = bulk stable temperature of the part

S = surface finish r.m.s. micro inches.

The paper reproduced as Appendix C references the above critical temperature models by H. Blok and covers a number of principles dealing with thermal contact of two moving parts.

C. Gear Lubricating Oils

The complexity of failure analysis in gears and the overlap in modes of failure is presented in Figure 11. Recognizing that the lubricant continues as a major factor in the total spectrum of surface stress and material behavior, continuing and successful research dealing with

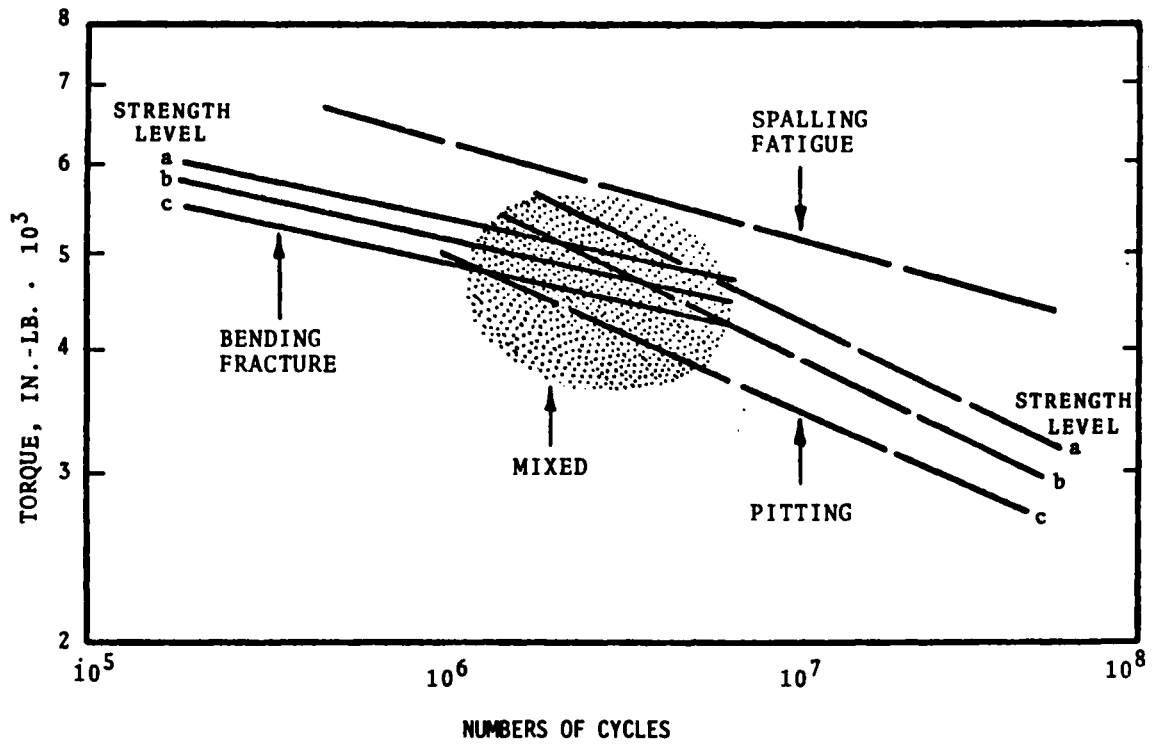


FIGURE 11 Torque versus cycles tests on a 4-square rig with tests taken to failure by breakage, pitting, or spalling (case crushing) for a 6-pitch test pinion. (Data from Breen, 1974.)

experimental oils is considered to be critical to temperature increases for gear application. Figure 12 shows load carrying capacity of different military specification and experimental oils. In military aircraft applications, gear lubricating oils are gas turbine oils since one oil is preferred for servicing all engines, transmissions, and gearboxes to eliminate possible mixing and logistical problems. The development of this single lubricating oil generally is directed by military specifications that define those fluid characteristics that must be met to provide satisfactory operational performance. The specifications essentially cover: (1) physical property limits based on intended temperature use (e.g., viscosity, flash point), (2) effect on elastomers, (3) oxidation/corrosion stability, (4) deposition characteristics/limits; (5) level of load carrying capacity, and (6) long-term storage stability. These specification requirements are presented in Table 1 for lubricants in current use and for advanced lubricants with improved performance characteristics. All of these synthetic lubricants are a general class of ester-based fluids in which the technology has already been demonstrated for operation in the temperature range of -40 to +425°F and increased load carrying capacity (up to 50 percent) under the Navy XAS-2354 specification.

This specification for an advanced ester-based fluid is under development to meet the high load carrying capacity requirements projected for high speed gas turbine engine gearing and heavily loaded helicopter transmissions and gearboxes. It is, of course, imperative that other physical properties and characteristics (i.e., deposition/cleanliness level, low temperature capability, elastomer compatibility, etc.) be maintained, and the fully-formulated lubricant must be a balance of base fluid and additives or additive systems that will perform satisfactorily in both high temperature environments and heavily loaded lubricated contacts.

The fluids for potential long-term development and use in specific temperature regimes with conventional lubrication methods currently include:

1. Ester-Based Fluids to 500°F--Experience indicates that user costs will be relatively low; however, there are low temperature viscosity and flammability problems and little growth potential can be expected.
2. Polyphenyl Ethers to 600°F--Some experience has been gained in military service and flammability characteristics are good. Projected growth potential is above 600°F; however, low temperature viscosity and load carrying capacity offer serious disadvantages to their application.
3. Fluorinated Compounds to 700°F--These fluids are nonflammable, form virtually no deposits, and have an

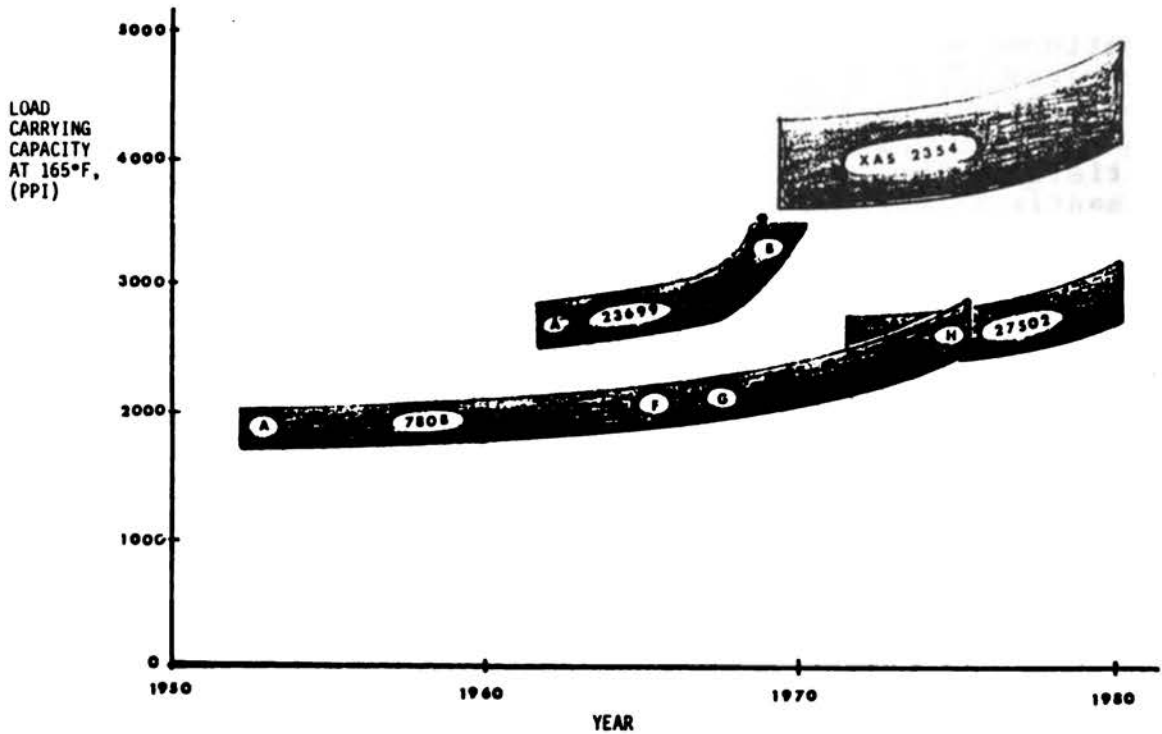


FIGURE 12 Summary status of gas turbine engine lubricant development of load carrying capacity. Letters shown in the chart relate to changes in the designated specification. The relative high temperature stability of the oils are not reported in this chart but the known characteristics of the oils for high temperature have influenced the designation of application limits. (Data provided by the Naval Air Propulsion Test Center.)

expected growth potential above 700°F. Low temperature viscosity, volatility, weight and cost are disadvantages but the most important deterrent is the formation of highly corrosive hydrofluoric acid. At present, there is no experience with these compounds and development risks would be high.

The transmissions and gearboxes in service helicopters presently operate at maximum lubricant supply temperatures of 230 to 250°F, which are satisfactorily managed by MIL-L-23699 and MIL-L-7808 oils. Although some ester-based oils provide acceptable levels of deposition/cleanliness in 425°F temperature environments, the failure mode most sensitive to elevated operating temperatures is the scuffing/scoring of gear tooth surfaces. One reason for this sensitivity is the reduction in elastohydrodynamic (EHD) film thickness that allows metal-to-metal contact and subsequent surface damage. Thus, the ability of the current service oils to operate at temperatures in the 425°F range will depend on the lubricants' ability to provide sufficient EHD film or additives systems surface protection (reactive process) without gear tooth scuffing/scoring damage (lubricant load carrying capacity). Research and development programs under Army and Navy sponsorship are scheduled to evaluate the load carrying capacity at 425°F of the specification oils listed in Table 1 in combination with standard AISI 9310 gears and the proposed high hot hardness steels. Initial results on some of the high hot hardness steels reveal a lubricant/gear material interaction effect that must be considered and factored into future high temperature lubricating oil development.

TABLE 1 Aircraft Gas Turbine Engine Oils

	Service Oils		Advanced Oils	
	MIL-L-7808	MIL-L-23699	MIL-L-27502	XAS-2354
Intended Use				
Temperature, F	-65/350	-40/400	-40/428	-40/425
Properties				
Viscosity, cs, 210 F	3 min	5.0-5.5	Report	5.0-5.5
Viscosity, cs, 100 F	Report	25 min	Report	-
Flash Point, F	400 min	475 min	475 min	475 min
Elastomers, F/hr				
NBR-H (Buna N)	158/168	158/72	158/168	158/72
F-A (Viton)	347/72	400/72	347/72	400/72
FS (Fluoro Silicon)	302/72	-	302/72	-
QVI(Silicon)	302/72	-	-	250/96
Storage Stability, F/hr				
Accelerated	230/48/168	-	-	-
Extended	-40 to 140 > 3 yrs	75/12 mos 0/6 wks	-40 to 140 > 3 yrs	75/12 mos 0/6 wks
Corrosion-Oxidation				
Temp/Time/Time, F/hr	392/48/96 347/96	425/72 400/72	464/48 428/48	425/72 400/72
Deposition Bearing Tests				
Standard, mm/hr	100/48	100/100	100/48	100/100
Temperature, F				
Oil Sump	350	390	464	390
Oil In	340	350	455	350
Bearing	500	500	572	500
Other Tests	WADC	-	-	Alcor
Lubrication Tests				
Ryder Gear, Std.				
Temperature, F	165	165	165	165
Load Carry lb/in., approx.	2400/2 ^a	2690/2 ^b	2550/2 ^a	3450/2 ^b
Ryder Gear, Mod				
Temperature, F	-	-	428	-
Load Carry lb/in., approx.	-	-	1150	-
IAE Gear, F	-	-	-	230
Performance Tests				
Engine/hr	J57/100	T63/Specify	J57/100	T63/Specify
Transmission	-	Optional	-	-

^a Air Force Reference Oil.^b Navy Reference Oil.

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- Littmann, W.E. 1968. The Mechanism of Contact Fatigue, NASA SP 237, The Timken Roller Bearing Company.
- Breen, D.H. 1974. Fundamental Aspects of Gear Strength Requirements. Report No. AGMA 229.17. American Gear Manufacturers Association. Presented at the Semiannual Meeting of the American Gear Manufacturers Association, Oak Brook, Ill. Nov. 3-6, 1974.
- Blok, H. 1955. "The Dissipation of Frictional Heat." Applied Scientific Research.

Chapter 4

STATUS OF TECHNOLOGY

A. GEAR STEEL COMPOSITIONS AND PHYSICAL STATIC STRESS CHARACTERISTICS

The composition of experimental and production steels available in quantity for gear manufacture are identified in Table 2.

The standard SAE grades along with the aircraft high production gear steels AMS 6265 (SAE 9310) and AMS 6274 (SAE 8620) are included to provide a comparison of the known properties and to define the properties required and their related testing procedures. The limiting operating temperature of the steels associated with automotive application is 300°F, but the remainder of the tool steel compositions have been specified to achieve a high temperature hardness of R_c 58. At service temperatures of approximately 600°F, they are considered suitable for strength, but some exhibit low impact toughness.

B. HOT HARDNESS

The hot hardness of four grades now under development or in production are presented in Figure 13 to illustrate the extension in service temperature provided by the high temperature gear steel composition. These compositions characteristically are either:

1. High in silicon (except CBS 1000) to resist softening during tempering,
2. High in nickel (except Cartech 53 and CBS 1000) to provide added toughness and ensure fabricability (e.g., piercing and forming);
3. High in molybdenum to ensure hot hardness and, in the presence of vanadium, to provide secondary hardening; and;
4. vanadium to also provide grain refinement.

C. HIGH TEMPERATURE STRENGTH

Table 3 presents core properties of several carburizing grade high temperature steels for the indicated test temperature. The data show a retention in tensile properties for the high alloy compositions with little suggestion of any degradation of uniaxial ductility. To ensure that these mechanical properties are maintained

Table 2. Limiting Operating Temperature
and
Nominal Composition of Carburizing Steels

		Element, weight percent									
		Steel	C	Mn	Si	Cr	Mo	Ni	V	W	Cu
Temperature	300°F	AMS 6274 SAE 8620	0.20	0.80	0.27	0.50	0.20	0.55	-	-	-
		SAE 4820	0.20	0.60	0.27	-	0.25	3.50	-	-	-
		SAE 4320	0.20	0.55	0.27	0.50	0.25	1.80	-	-	-
		AMS 6265 or 6290 [9310]	0.10	0.55	0.27	1.20	0.13	3.25	-	-	-
		EX 55	0.17	0.90	0.30	0.50	0.75	1.80	-	-	-
		EX 32	0.20	0.85	0.25	0.55	0.50	0.85	-	-	-
450° - 600°F		CBS 600	0.19	0.60	1.10	1.50	1.00	-	-	-	-
		CBS 1000	0.13	0.50	0.50	1.10	4.50	3.00	0.35	-	-
		X-2	0.20	0.30	0.90	5.00	1.40	-	0.45	1.40	-
		X-2 (M)	0.15	0.30	0.90	5.00	1.40	-	0.45	1.35	-
		CarTech 53	0.10	0.55	1.00	1.00	3.25	2.00	0.10	-	2.0

Data from Diesberg, Jatczak, Fopiano, and Kula, Standard SAE Specifications.

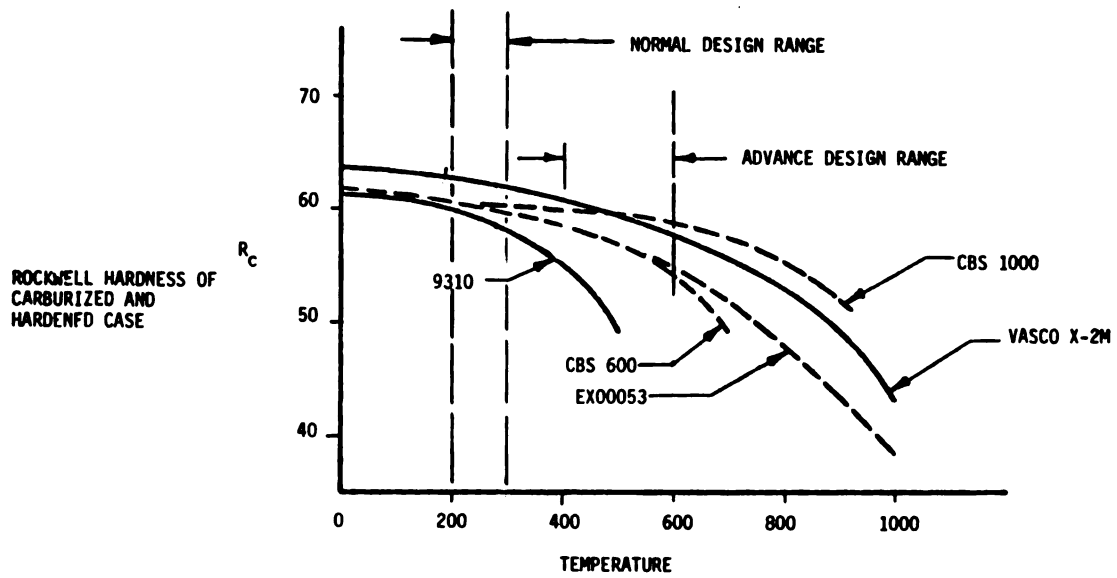


FIGURE 13 Hot hardness of production and experimental gear steels.
(From Jatzak, 1978.)

TABLE 3 Comparison of Core Mechanical Properties of CBS600 and CBS1000M with AISI 4820 and 9310 Steels (from Jatzcak, 1978)

Test Temperature, F(C)	Quenched Size, ^a in. (mm)	Tensile Strength, 10 ³ psi(MPa)	Yield Strength, 10 ³ psi(MPa)	Elongation, %	Reduction in Area, %	Impact Energy, ft - lb(J) Charpy	Hardening/Tempering Conditions, F(C)
CBS600							
70(20)	2.25(57)	220(1515)	180(1240)	12.5	55.0	35(47)	1550/600 (845/315)
600(315)	2.25(57)	215(1480)	152(1050)	18.0	54.0	29(39)	1550/600 (845/315)
700(370)	2.25(57)	205(1415)	144(995)	18.0	53.5	31(42)	1550/600 (845/315)
CBS1000M							
70(20)	1.0(25)	212(1460)	174(1200)	16.0	64.0	10(14)	2000/1000(1095/540)
800(425)	1.0(25)	184(1270)	146(1005)	12.0	52.0	48(65)	2000/1000(1095/540)
900(480)	1.0(25)	168(1150)	141(970)	11.0	52.0	50(68)	2000/1000(1095/540)
1000(540)	1.0(25)	158(1090)	133(915)	12.0	50.0	51(69)	2000/1000(1095/540)
70(20)	4.0(100)	192(1325)	163(1125)	15.0	58.0	32(43)	1750/600 (955/315)
AISI4820^b							
70(20)	0.5(13)	209(1440)	173(1195)	14.0	54.0	12 od 35(47)	1700/300 (925/150)
	1.0(25)	170(1170)	126(870)	15.0	51.0	30(41)	1700/300 (925/150)
	2.0(50)	136(940)	93(640)	20.0	56.0	51(69)	1700/300 (925/150)
AISI9310^b							
70(20)	0.5(13)	179(1235)	143(985)	16.0	59.0	—	1450/300 (790/150)
	1.0(25)	159(1095)	123(850)	16.0	58.0	—	1450/300 (790/150)
	2.0(50)	145(1000)	108(745)	18.0	67.0	—	1450/300 (790/150)

^aIndicated round section hardened in oil, then tested in 0.505 in. (13mm) tensile specimens.

^bData from International Nickel Co. handbook on nickel alloy steels, section 2B.

throughout the anticipated service life of the gear, it is necessary that microstructures remain stable.

This important relationship of high temperature stability can be expressed in room temperature hardness of material after exposure at the indicated temperatures. The data presented in Table 4 include the automotive grades and SAE 9310 (AMS 6265) that clearly show a loss in hardness at temperatures above 300°F.

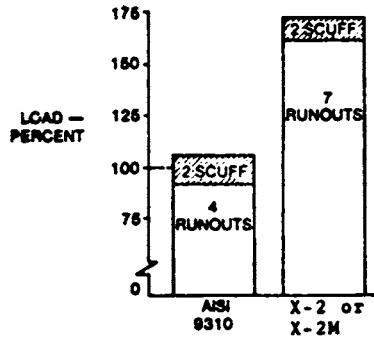
D. HIGH TEMPERATURE SCORE RESISTANCE

Hot hardness coupled with high temperature stability provides required scoring resistance but also eliminate the plastic flow or surface deformation of the gear tooth that can lead to surface fatigue damage. Tests on gear teeth have provided data in terms of percentage load scoring resistance at the flash temperature (Figure 14). The results show the significant advantage of the X-2 steel with high temperature characteristics superior to the high production 9310 composition steel. The spur gear tests cited in Fig. 14 provide data on the scoring load capability of 4.55 P test gears in a 4-square load stand. The numbers of teeth were 33 and 58 (pinion and gear) and input rpm was 2400. Lubrication was by jet-directed MIL-L-23699 oil at an inlet temperature of 195°F. The criteria of failure was scoring on more than 3 percent of the tooth area and the test program provided six valid test points with SAE 9310 steel, and nine with X-2M.

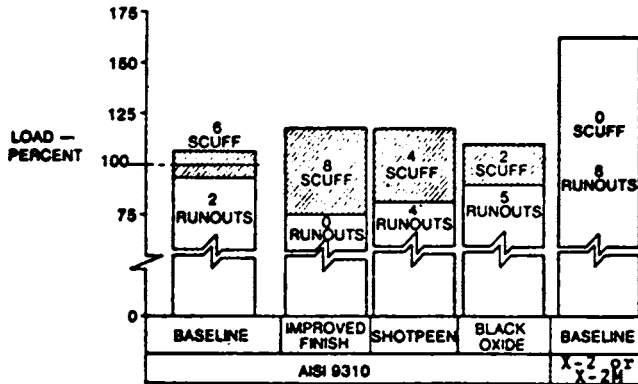
Another comparison scuffing test was conducted for spiral bevel gears (Reference D210-10323-1). Here the gear parameters were: Diametral pitch 5.833; teeth 35 and 43; lubrication was MIL-L-23699 oil, with an average input temperature of 215°F. Variables in this test series included surface finish, black oxide, and shotpeening as well as the basic material difference of 9310 (AMS6265) and VASCO X-2M. Figure 14a summarizes these test results. The VASCO X-2M gears had no process finishes (oxide or peening) and had a surface roughness of 23-27 RMS. This was the same as the SAE 9310 (AMS 6265) gears except that one group with an improved surface (12-15 RMS) was included. The improvement in scoring capability evidenced by VASCO X-2M bevel gears is approximately the same as shown in the spur gear tests (i.e., 70 percent load increase over SAE 9310 steel), all primarily attributed to the hot stability and hot hardness of the material.

E. FRACTURE TOUGHNESS AND DAMAGE TOLERANCE

The fracture toughness of SAE 9310 steel is considered to be very good, and this is one of the benefits of its use



(a)



(b)

FIGURE 14 Scoring tests: (a) spur gears (data from Boeing Vertol Co. Spiral Bevel Gear Manufacturing and Finishing Evaluation Program with Vasco X-2 Steel Contract (DAAJ01-70-C-0453), and (b) spiral bevel gears (data from Improved Manufacturing Process for the Finishing and Surfacing of Spiral Bevel Gears, J. P. Alberti, et al., 1972, Boeing Vertol Document D210-10323-1, Contract DAAJ01-70-C-0453).

as a gear material. During the design stages of a new helicopter transmission, fracture toughness is a design criterion for the newer high temperature steels in development to handle ballistic survival. It generally is accepted that current SAE 9310 technology, because of its exceptional ductility, does not present any problems in this area. Qualitatively, a helicopter transmission must not disintegrate or have other catastrophic side effects even when relatively large metallic chips or other foreign matter are passed through the gear meshes. The ability to continue operation without immediate damage and full functional failure is related to the fracture toughness of the materials used in the transmission. It is desirable that secondary damage be minimized so that a landing can be effected, and, at the very least, continue driving the accessories so that hydraulic power will be available for control in autorotation.

In general, design philosophy has never accommodated gear materials with compositions which are so brittle or strain rate sensitive that one would need to quantify these properties and use them as a selection criteria. It is prudent to use fracture toughness and threshold as a rationale for measuring the degree of toughness and damage tolerant resistance of gears since some measure of this has been sacrificed to achieve hot hardness. The plane strain fracture toughness data in Figure 15 shows more than a 50 percent reduction in K_{IC} values for steels that retain their hot hardness and stability at service temperatures between 400 and 600°F. Some work also has been accomplished in the area of fracture toughness influenced by composition changes of the steel, most specifically the carbon content within the carburized case (Figure 16).

The results of this work indicate that raising chromium levels from 0.2 and 1.2 percent lowered the fracture toughness in the higher carbon content surface layers of the case (Figure 16a) while 1.2 and 1.4 percent Mn lowered the fracture toughness in the case-core interface (Figure 16b). Both conditions lead to low impact fracture resistance, and these steels break with 1 to 2 impacts with a hammer energy of 4J. Figure 16d shows the fracture toughness gradients for EX55, SAE 4817, and SAE 9310 compared to that of SAE 8822. The higher hardenability, lower carbon steels (EX55, SAE 4817, and SAE 9310) exhibit higher fracture toughness values than SAE 8822 in all locations in the case. These latter steels also exhibit high impact fracture stress and impact fatigue properties.

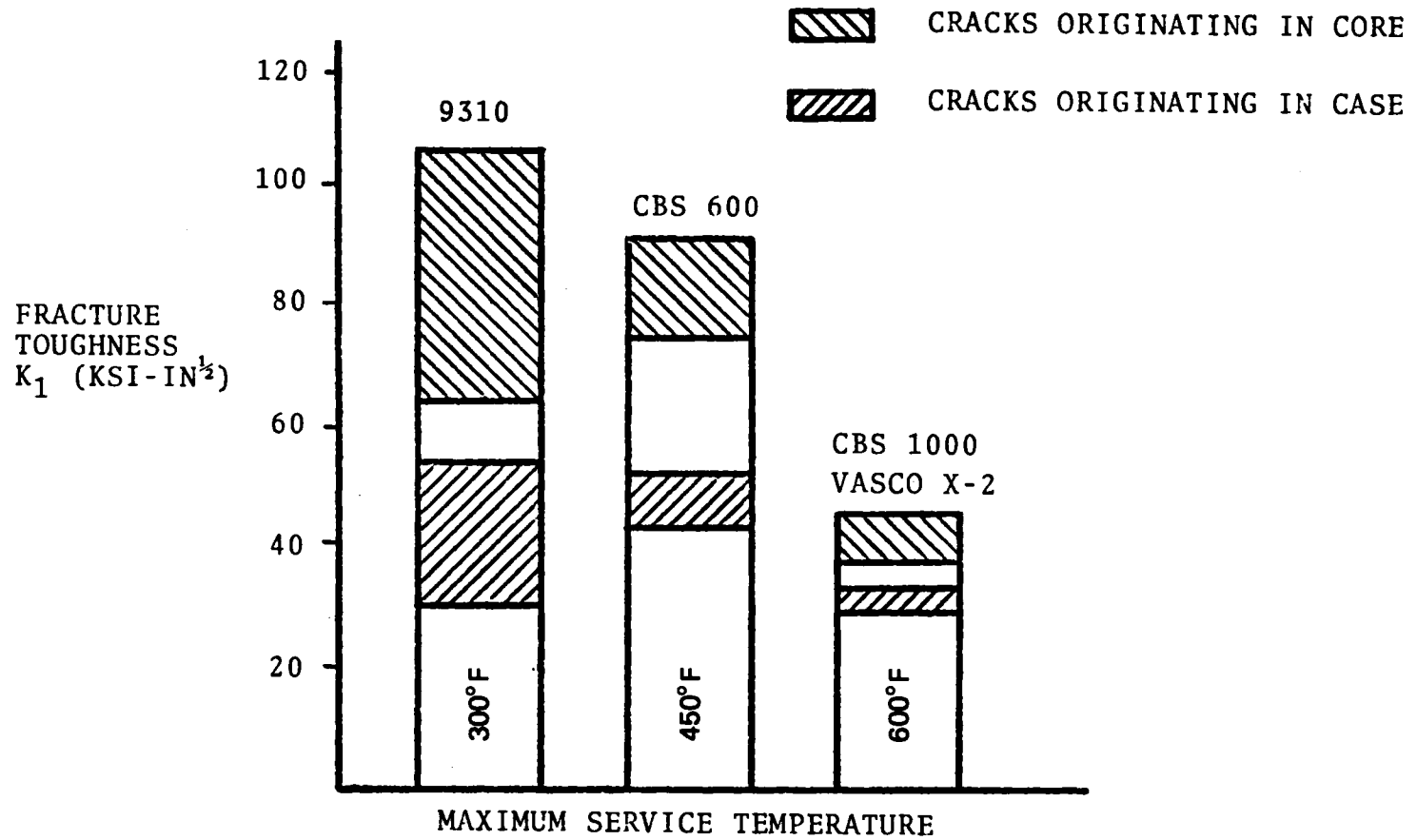


FIGURE 15. FRACTURE TOUGHNESS OF ADVANCED GEAR MATERIALS [7]

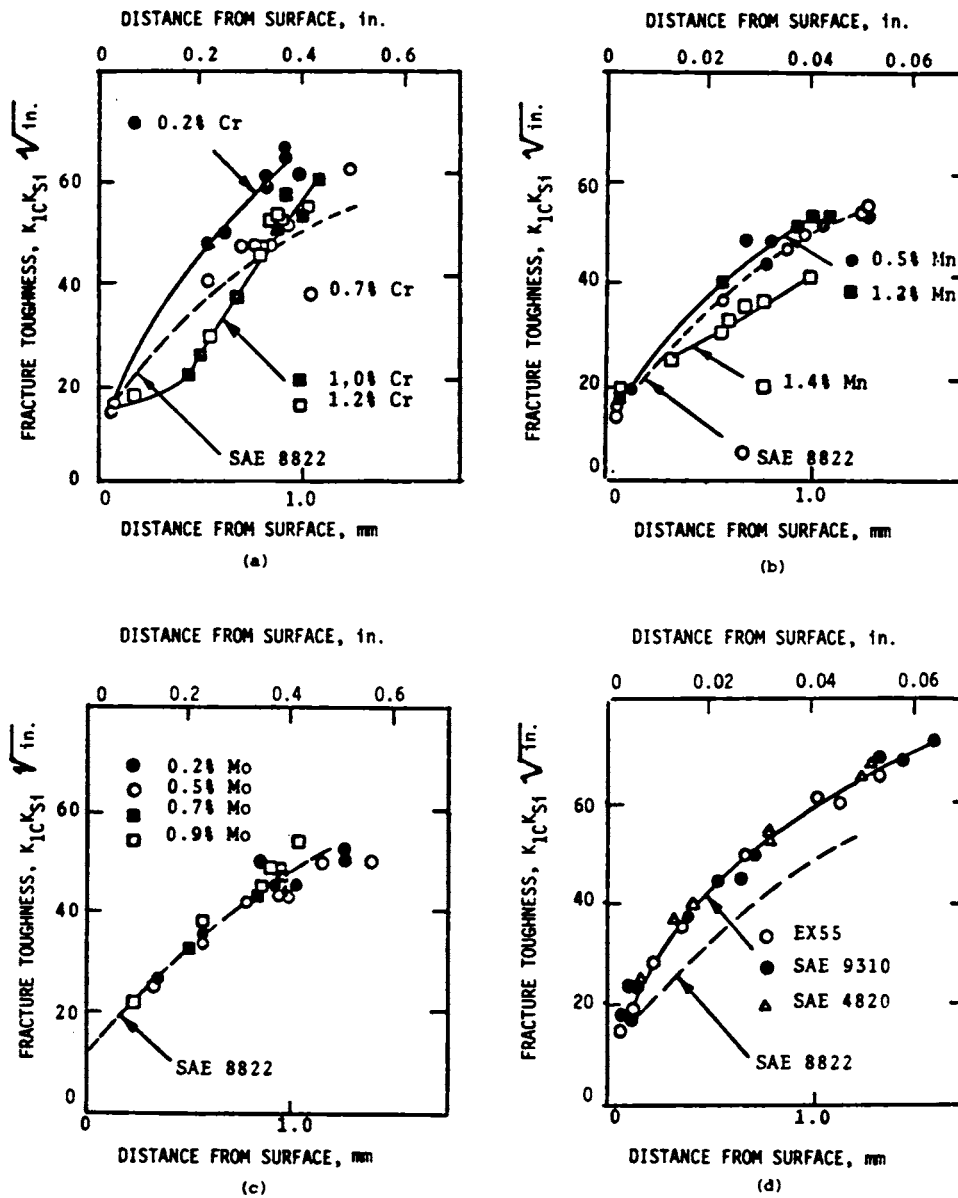


FIGURE 16 Fracture toughness gradients in carburized cases of steels with a base composition of SAE 8822 and steels modified with three elements: (a) chromium, (b) manganese, (c) molybdenum. The results for SAE 8822 are compared to those of three high hardenability steels in (d). (From Y. E. Smith and D. E. Diesburg, 1979.)

F. FATIGUE CRACK GROWTH

While it is agreed that damage tolerance and flaw tolerance are directly related to material toughness, there continues to be an engineering design interest in moving forward with correlations of various fatigue characteristics significant to shafting, rim, and web designs so that related failures will always be eliminated. The results of some fatigue testing under alternating stress in the presence of a flaw comparing SAE 9310 and X-2 steels are presented in Figure 17. The data show a significant superiority in flaw tolerance of double vacuum melt X-2 over the single melt X-2. These are limited data from one heat of material. Some experience with gears manufactured from the high production SAE 9310 steel base have been known to fail as shown in Figure 18. It is, therefore, important that steel cleanliness is controlled to a critical and quantified level through updated aerospace-type cleanliness standards and with specified vacuum melt and ingot processing systems. For comparable size flaws a comparison of SAE 9310 and X-2 or X-2M under fatigue conditions shown in Figure 19 indicates an overlap of crack propagation rate da/dN for a specific stress intensity when tested at room temperature.

Fracture-mechanics-based testing to determine the fatigue crack propagation characteristics of metallic materials indicates the presence of a threshold, below which fatigue crack growth is extremely slow or nonexistent. Examination of fatigue crack growth data presented in a fracture mechanics format (stress intensity versus growth rate) (Figure 20) shows a sigmoidal relationship with three distinct regions. At the high growth rates, crack growth instability is approaching the fracture toughness of the material. The intermediate growth rate range has been explored extensively in the technical literature and is frequently represented by a power-law relationship. At lower rates of growth, the curve tends to asymptotically approach a limiting value of stress intensity that can be viewed as an endurance limit or threshold for fatigue crack growth. The threshold generally is not predictable from a knowledge of the material's fracture toughness (i.e., two materials with significantly different fracture toughness values can have similar fatigue crack propagation thresholds).

The threshold value is established in fatigue crack propagation testing. The number of cycles required to propagate the crack increment of length provides the cyclic growth rate. The corresponding load, crack length, and specimen geometry determine the stress intensity level. The threshold level is primarily a function of material, stress

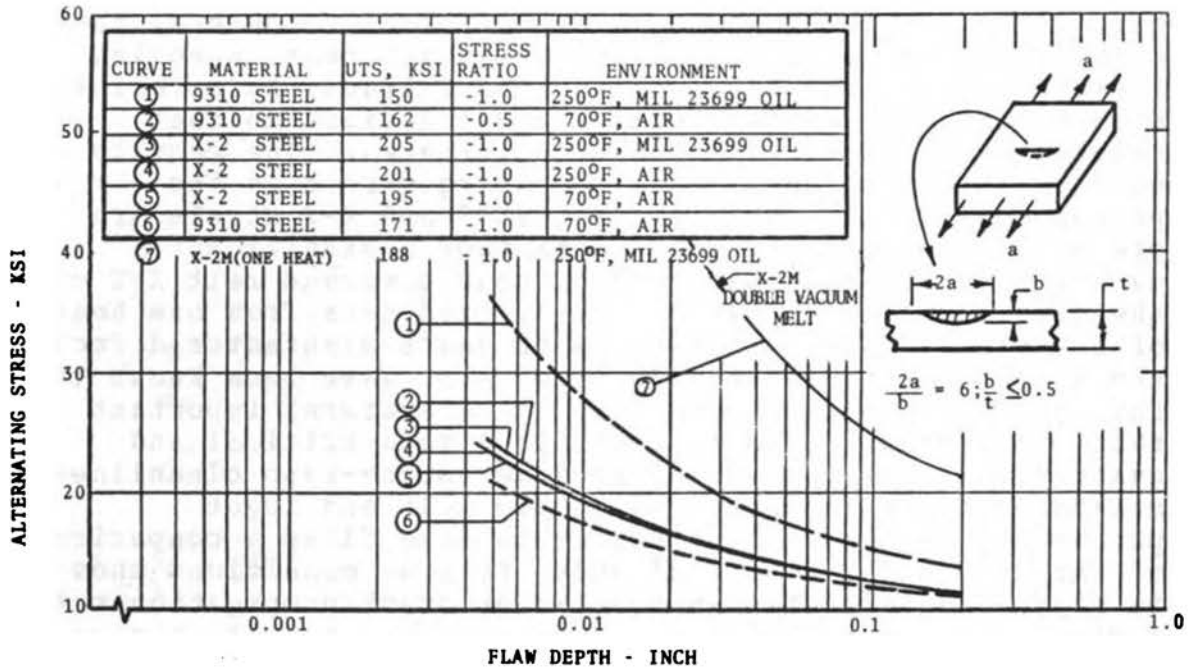


FIGURE 17 Threshold fatigue crack propagation characteristics stress level/flaw size combinations for SAE 9310 and X-2M steel compositoins showing effect of vacuum melting to improve flaw tolerance. (From an internal memorandum, Boeing Vertol Co., relative to work performed in the time period 1975-78.)

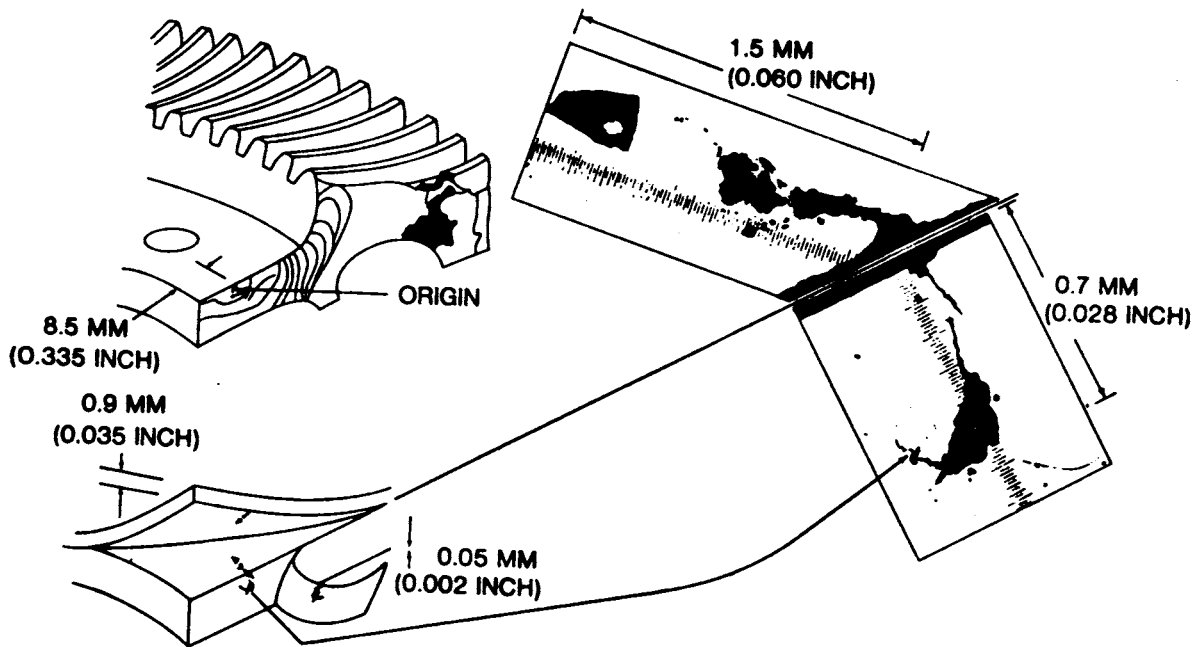


FIGURE 18 Exogenous inclusions associated with gear flange fatigue failure at measured $12 \text{ ksi} \pm 6 \text{ ksi}$ material was AMS 6265 single vacuum melting - heat treated to 190 ksi ultimate strength. (From an internal memorandum, Boeing Vertol Co., relative to work performed in the time period 1975-78.)

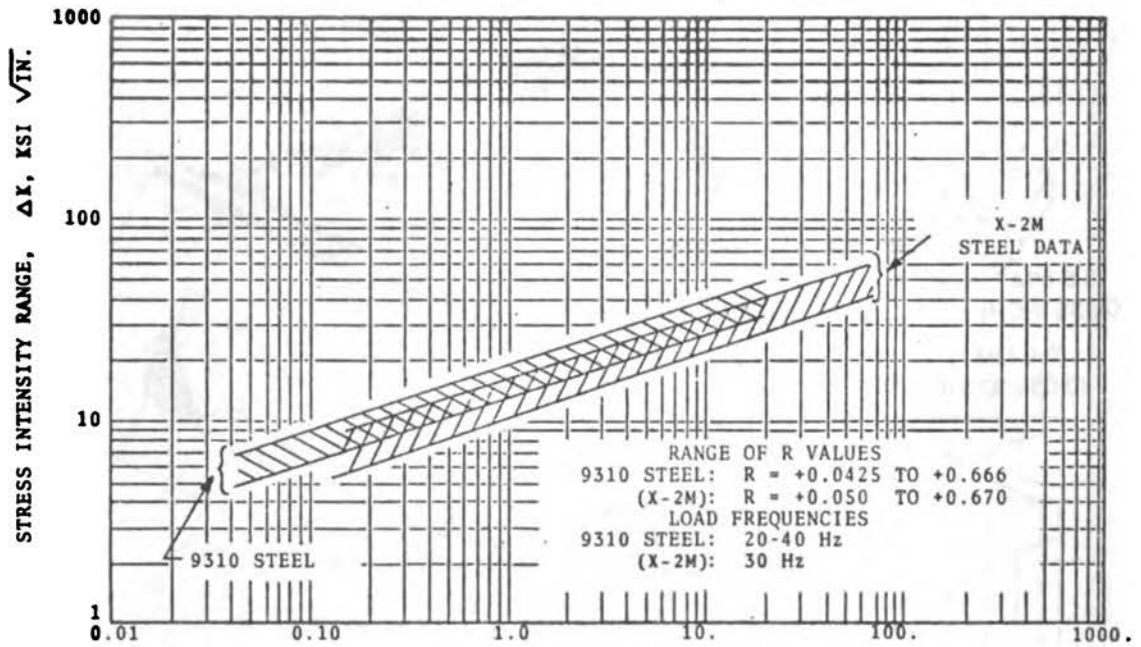


FIGURE 19 Comparison of room temperature fatigue crack propagation rates for SAE 9310 and X-2M steels for crack propagation rate, $\frac{da}{dN}$, micro-inch. (From Cunningham, et al., 1974.) $\frac{da}{dN}$ cycle

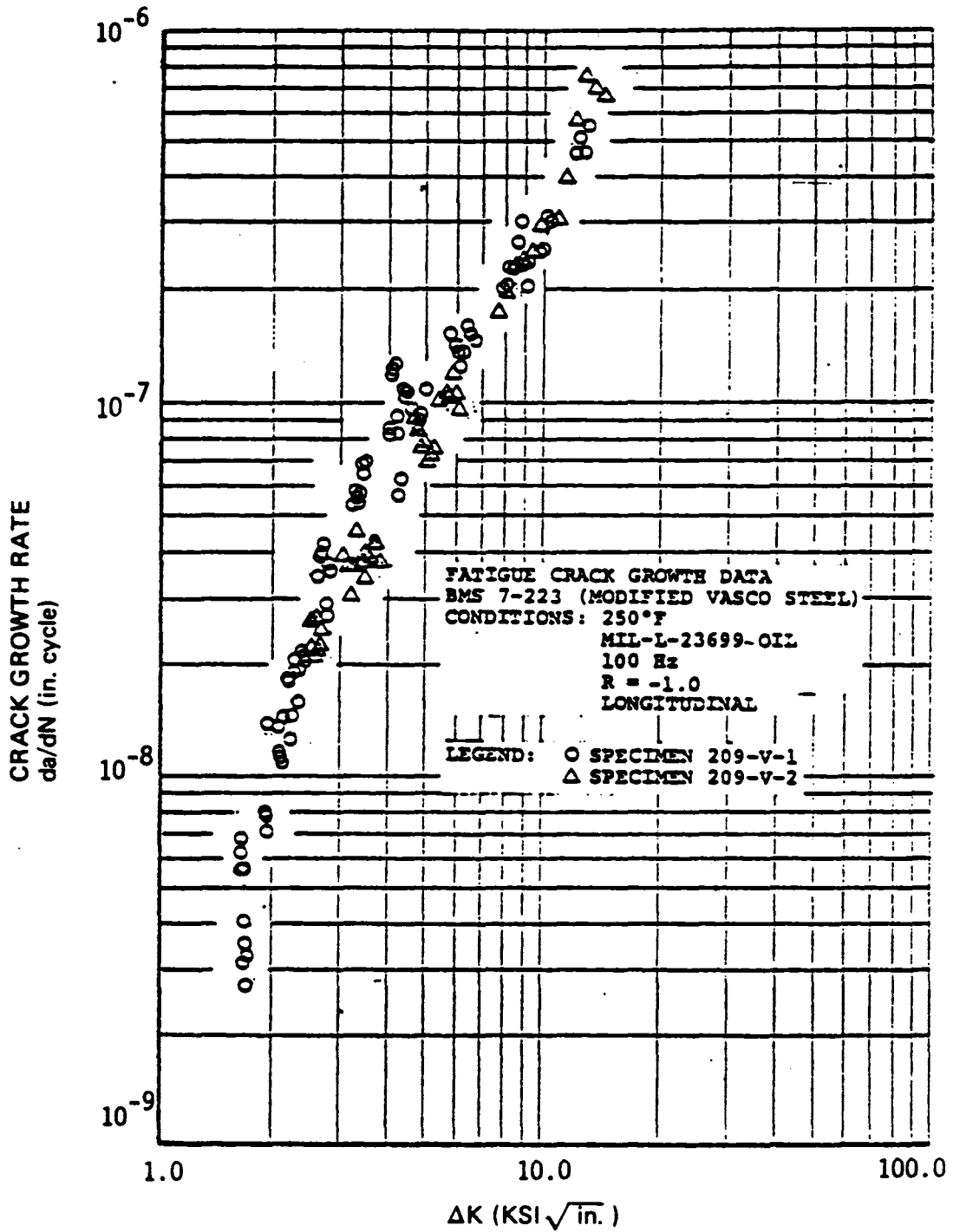


FIGURE 20 Fatigue crack growth as influenced by intensity of stress.

ratio, and environment. The American Society for Testing and Materials (ASTM) has an active program for threshold testing and data production, which presently is working to standardize definitions and test procedures relative to determination of fatigue crack propagation threshold values.

For a particular material, loading condition and geometry, the threshold stress intensity level, ΔK_{TH} yields (via fracture mechanics relationships) combinations of fatigue-stress levels and crack sizes below which crack growth is negligible. With this approach it is possible to develop parametric charts for selected crack geometries that relate steady stress, alternating stress, and crack size at the crack growth threshold condition and to assess one aspect of the damage tolerance of a material/structural system. The operating stress levels consistent with nonpropagation of given size damage (flaw or crack) and inspection requirements can be determined.

Threshold fatigue crack propagation tests have been conducted on more than 10 samples of X-2M and SAE 9310 core material. Testing has encompassed a range of loading and environmental conditions related to transmission operation. Figure 21 shows basic fatigue crack growth data for both single and double vacuum melt X-2M steel. Both samples were tested at a stress ratio of -1.0, a temperature of 250°F, and in a MIL-L-23699 oil environment. The data are presented in terms of stress level-flaw size relationships in Figure 22. A significant increase in damage tolerance is indicated for the double vacuum melt X-2M steel. Attention is again directed to Fig. 17 which presents additional threshold data for single vacuum melt X-2M and 9310 steels covering a number of environmental and stress conditions. The threshold fatigue crack growth characteristics of X-2M and 9310 core material generally are similar. This testing has indicated that double vacuum melting improves the threshold for fatigue crack propagation and that X-2M and SAE 9310 steels exhibit similar threshold values for conditions typical of transmission operation.

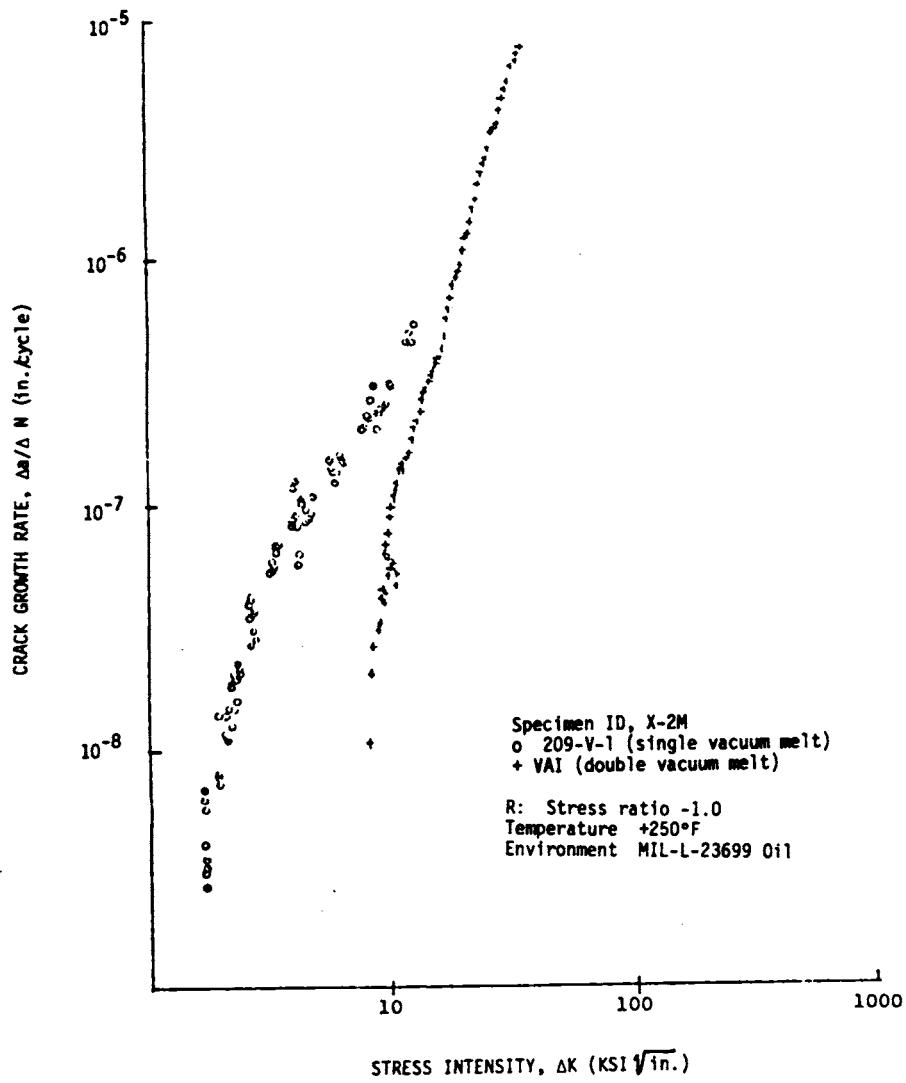


FIGURE 21 Comparison of single vacuum and double vacuum melt XM-2 steel to illustrate improved damage tolerance provided by double vacuum in terms of ΔK (KSI $\sqrt{\text{in.}}$).

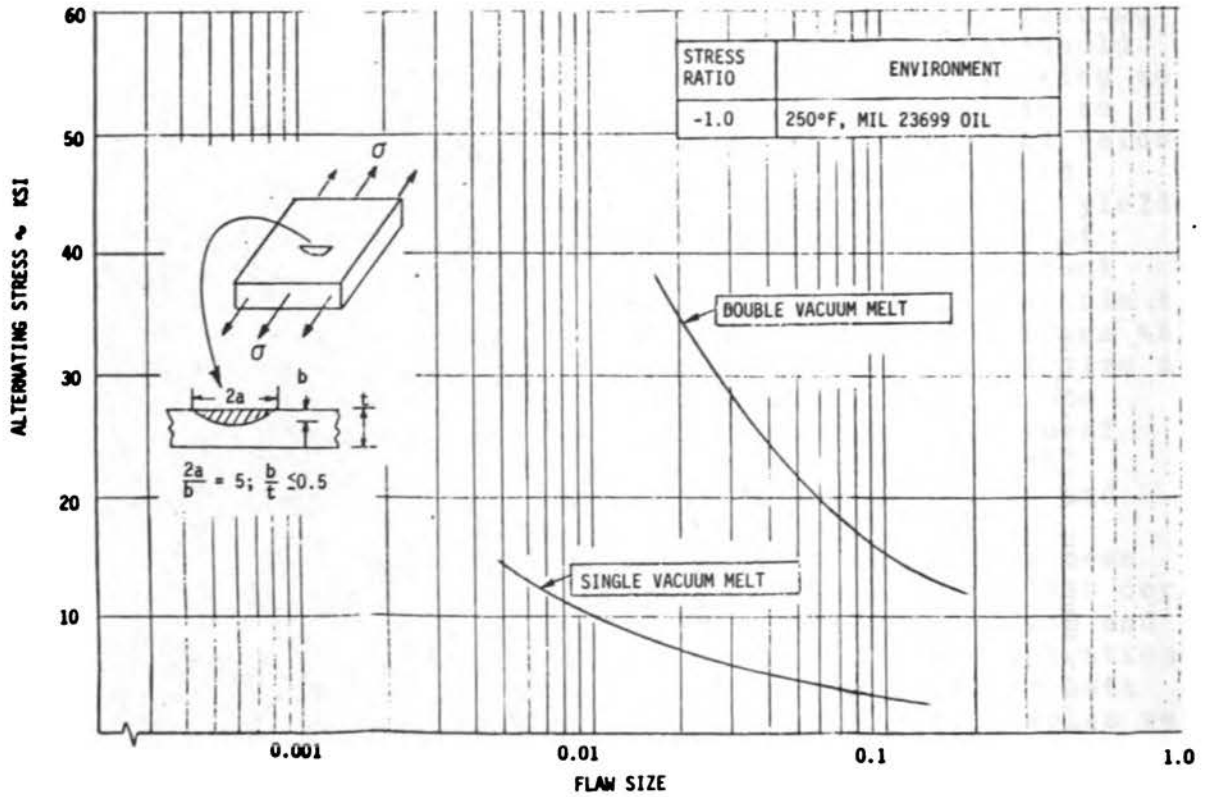


FIGURE 22 Threshold fatigue crack propagation characteristics for single and double vacuum melt BMS 7-233 steels.

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- Jatczak, CF. 1978. "Specialty Carburizing Steels for Elevated Service." Metal Progress, April, 1978 pages 70-78.
- Cunningham, R. J., et al., Vasco X-2, 0.15% Carbon (BMS 7-223) Steel HLH/ATC Transmission Gear Materials Evaluation, Test Results and Final Report. Boeing Vertol Co. Document D301-10036-2, Contract DAAJO1-71-8840 (P6A) July 1974.
- Figures 14-22 are derived from internal memoranda, Boeing Vertol Co.

Chapter 5

PROCESSING

A INTRODUCTION

The carburizing of specialty high temperature gear steels presents some unique requirements to achieve a uniform and controlled response during the carburizing process. The formation of chromium oxide or tightly adhering dense spinel that block the carburizing action have been a common experience by helicopter manufacturers. Some observations relating to this phenomena may be significant to the successful exploitation of these high chromium content steels.

1. Carburization enhancement of high Cr steels by preoxidation has been shown to be caused by the depletion of Cr from the surface of the steel to the surface of the preoxidation layer. The presence of scale breakaway after preoxidation, while originally thought to be important, was found to be insignificant as a factor in carbon penetration during carburizing.

2. A method based on atomic gas composition of C-O-H mixtures has been applied to establish the metallurgically important equilibria between carburizing atmospheres and metal, oxide, and carbide systems.

3. Carburization on non-preoxidized high chromium steels at normal potentials, using CO control at 1750°F (955°C) and 1850°F (1010°C), can result in a situation where the initial part of the cycle will be carburizing while the remainder of the cycle will be non carburizing. When the surface is Cr-depleted by preoxidation, the above situation cannot occur and carburization throughout the entire cycle is experienced.

4. High chromium steels may be carburized on a production basis in CH -enriched atmospheres (e.g., in simple carburize-diffuse cycles in which the ratio of RX gas to natural gas is kept at 20/1 or richer), but temperatures above 1850°F (1010°C) will be required for uniform carbon penetration.

5. These overly rich atmosphere compositions always will seem to produce massive carbides in Cr-rich steel, but rehardening from temperatures higher than the carburizing cycle when furnace or induction hardening will serve to redissolve the carbides.

6. If gas carburization must be done at 1750°F (955°C) and below, preoxidation at 1750°F or above should always be employed on steels containing more than 2 percent Cr. The

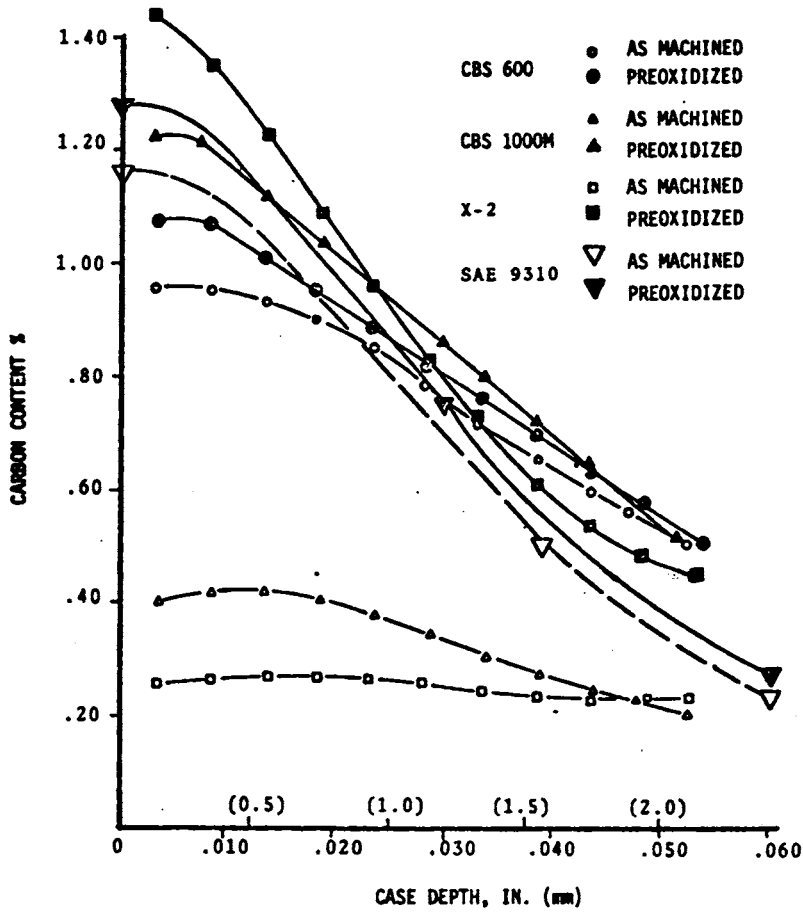
preoxidation treatment may be conducted in either a separate furnace or during heat-up in air in the carburizing furnace itself. Both of these techniques have been conducted on the high temperature carburizing steels, X-2, CBS600 and CBS1000M, as shown in Fig. 23.

7. C-O-H atomic gas composition equilibrium plots have contributed to understanding of the effects of carburizing and processing atmospheres, such as RX, DX, NX on oxide surfaces affected by chromium in the alloy.

8. As machined high chromium surfaces can be carburized directly providing the atmospheres contain controlled low oxygen levels and there is now some understanding on alternate methods of by-passing a preoxidation processing step by employing a vacuum or gas mixture of $\text{NH}_3 + \text{H}_2 + \text{CH}_4$.

This report is not intended to establish specific carburizing controls and inform on techniques but rather to cite selected results achieved in some industrial heat treating operations. The importance of the chromium content and temperature effect upon equilibrium between carbon and complex iron chromium compounds is now recognized, and Figure 24 provides temperature guidelines for the carburizing process to achieve specified carbon equilibrium. The required gas compositions and their relationship of equilibria with the steel have been thoroughly explored and, in practice, are employed as rectangular plots with nitrogen/reactive gas ratio = 0. As these ratios are increased to between 0.2 and 1.0 for controlled carbon ratios in the reactive C-O-H reactive gas, there are specific CH_4 weight fractions necessary to achieve and maintain a carbon driving force and accommodate the demands of $\text{Cr}(\gamma)/\text{CrO}_3$, Fe_3C , and $\text{Cr}(\gamma)\text{Cr}_7\text{C}_3$. A chart illustrating this effect is shown in Figure 25.

In the light of this special requirement to step up carbon potential and circumvent the resistance to carbon penetration by chromium/iron oxide adherent layers, comparisons were run on the efficiency of preoxidized vs non-preoxidized surface layers on carburizing characteristics. This is illustrated in Figure 23 which compares carbon concentration after 1750°F cycle to achieve about a 0.050 - 0.060 in. case depth. The data suggest that when dealing with steels such as X-2 and CBS1000 preoxidation is extremely beneficial in maintaining a uniform carbon penetration. Further, it appears that CBS1000, although containing slightly lower chromium than SAE9310, nevertheless benefited similarly to the preoxidized surface. It is suggested that vanadium and/or high molybdenum in combination with chromium also may provide an inhibiting surface spinel with iron. Some combinations for achieving preoxidation showing the relative carbon



CORE COMPOSITION:

	C	Mn	Si	Cr	Ni	Mo	W	V
CBS 600	.22	.68	1.05	1.60	-	0.96	-	-
CBS 1000M	.13	.50	0.10	1.15	2.90	4.67	-	0.41
Vasco X-2	.24	.32	0.86	4.89	-	1.32	1.41	0.48
9310	.10	.55	0.22	1.20	3.25	0.10	-	-

FIGURE 23 Influence of preoxidation (1850°F/½ hr) on carburizing characteristics of CBS 600, CBS 1000M, and X-2 Steels. Carburized 2 + 9 hrs at 1750°F with CO₂ controlled at 0.035 percent. Note: For 9310, the CO₂ was controlled at 0.050 percent. (Based on Internal Report, The Timken Co.) (1977)

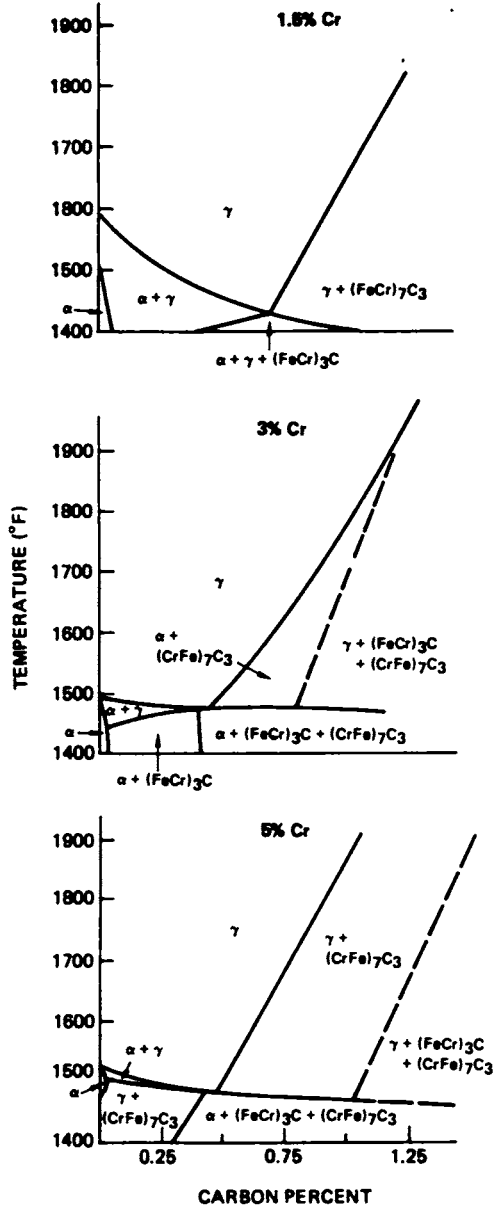


FIGURE 24 Iron-carbon equilibrium diagrams at various chromium compositions.

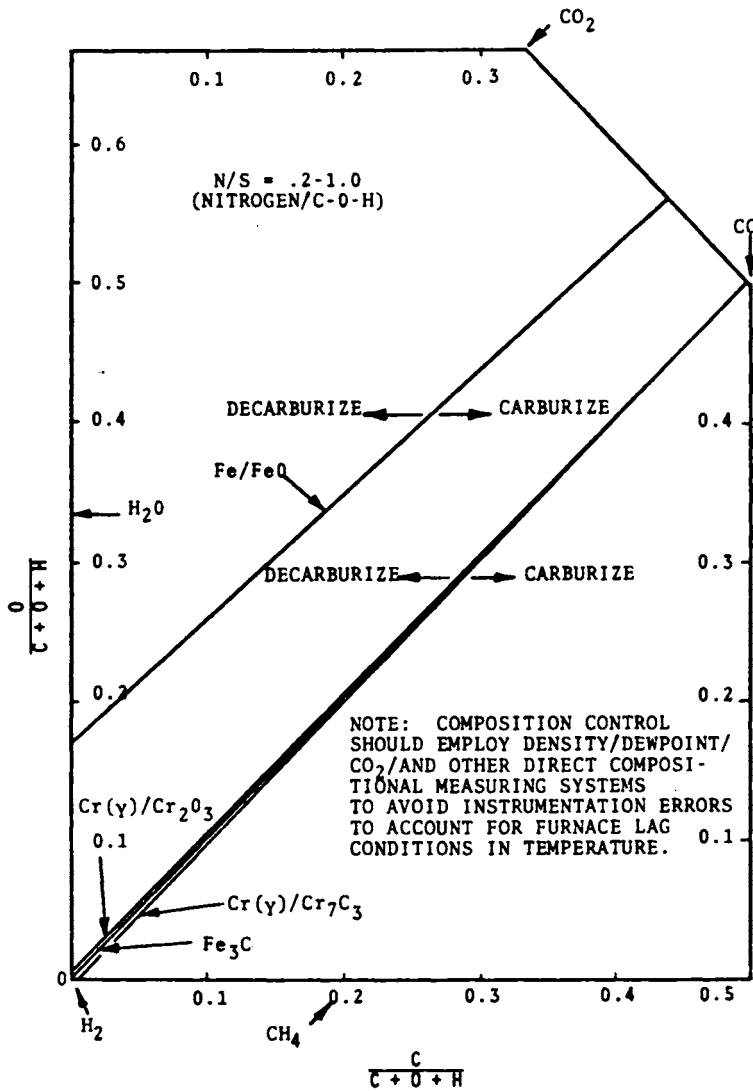


FIGURE 25 Gas compositions in equilibrium with various metal systems at 1850°F (1010°C). (Based on data provided by the Timken Co.)

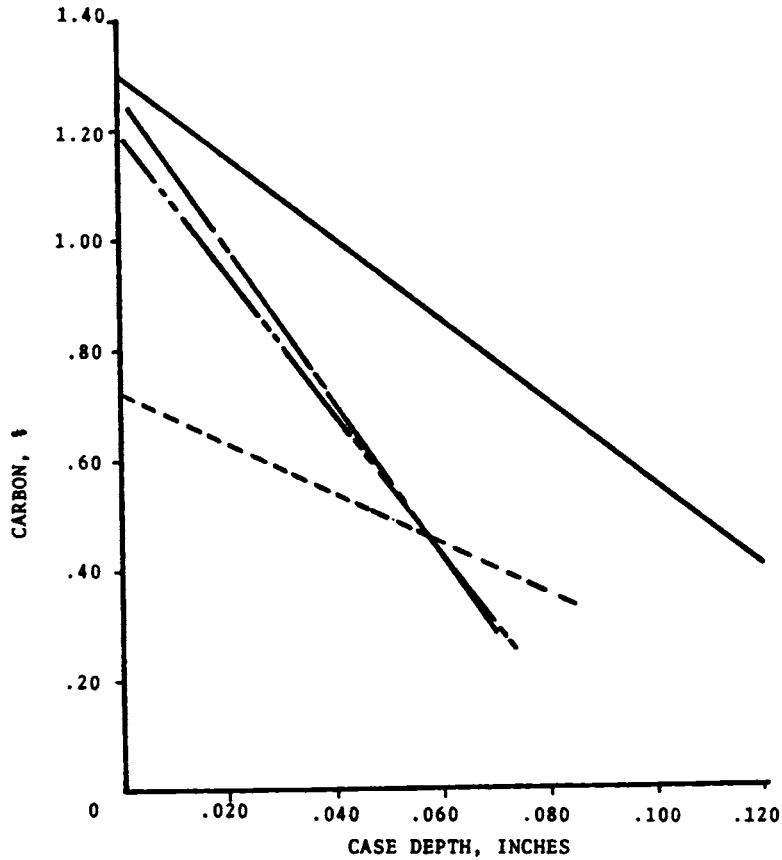
penetration are illustrated for CBS1000 in Figure 26. The data again show the advantage of the preoxidation effect, whether performed in situ or in separate furnaces from the carburizing operation. The additional surface carbon content and the case depth increase in comparing the 0.125-in. case depth cycles indicates that the non-preoxidized surface has only about 50 percent of the desired case at approximately 0.40 percent C. It then would be necessary to interrupt the process, clean the parts, and further return to the carburizer at high expense and poor predictability of results. A typical processing cycle employing X-2 (5% Cr) steel for gear application is shown below:

Preoxidize	1800°F	40-50 Minutes
Al ₂ O ₃ Blast		
Copper Plate		
Carburize	1700°F	
Temper	1100°F	2 Hours
Nickel-Copper Plate		
Bake	1150°F	
Preheat	1450°F	
Austenitize	1850°F	15-30 Minutes
Oil Quench		
Deep Freeze	-120°F	
Double Temper	600°F	

The cost relationship of machining these selected surfaces below the case depth after carburizing and further, to copper/nickel plate for protection of the entire part during hardening can be a rather important added cost driver in the process.

B. TESTING

Criteria significant to gear design include a number of standard tests that provide guidelines to the designer in material and process selection, but it is now recognized that new and innovative testing useful to future high temperature applications are now required. A few standard tests are already available and described in the ASTM procedures and other not standardized are listed below to illustrate the surprisingly small amount of standardization and to identify the tests that helicopter gear manufacturers and users employ.



- Preoxidized in air at 1750°F (945°C) in separate furnace (to 0.125 in. case depth)
- As machined, not preoxidized (to 0.125 in. case depth)
- - - - Preoxidized in air at 1750°F (945°C) in separate furnace (to 0.062 in. case depth)
- - - - Preoxidized in carburizing furnace, by blowing air during heat up to carburizing temperature of 1750°F (945°C) (to 0.062 in. case depth)

FIGURE 26 Influence of preoxidation on carburizing characteristics of CBS 1000M Steel. All specimens carburized in 1.25 percent C potential at 1750°F (945°C). (Based on data furnished by The Timken Co.)

Charpy impact	ASTM E23
Plane strain fracture toughness	ASTM E399
Tensile testing	
Room temperature	ASTM A370
High temperature	ASTM E21
Ryder gear score testing	ASTM D1947
Dynamic gear tester	Not standard
R.R. Moore rotating beam	No standard document available
Bending fatigue	Not standard
Single tooth bend	Not standard
Pitting fatigue	Not standard
Hot hardness	No standard document available
Impact bending	Not standard

1. Fatigue Tests

Final product testing under controlled stress and environment conditions is, of course, the best method of product verification short of full gearbox application. For example, the R.R. Moore fatigue tests of X-2 steel in the smooth and notched condition (Figure 27) show a normal level of notch sensitivity for the core condition of the material. The single tooth bending tests on the same material under conditions where a case hardened surface compressive residual stress is built into the specimen (Figure 28) show an excellent fatigue strength. Under rotating contact at various stress levels (Figure 29), a satisfactory contact stress fatigue strength was demonstrated on a pinion gear. For combined stress testing the Goodman diagram affords the best opportunity to assess gear performance and express design allowables which can then provide a base for future applications. The various points on the curve are collector gears on the helicopters designated in Figure 30. The design allowables are expressed as an arc between the full alternating and full static tensile stress values.

Unfortunately, the correlation of gear tests during the design and development process does not always permit such extensive and time consuming tests. More recent work employing specimens to stimulate a gear tooth and used to measure high cycle fatigue, impact fatigue, and impact fracture stress has provided a useful composite approach for evaluating gear steels.

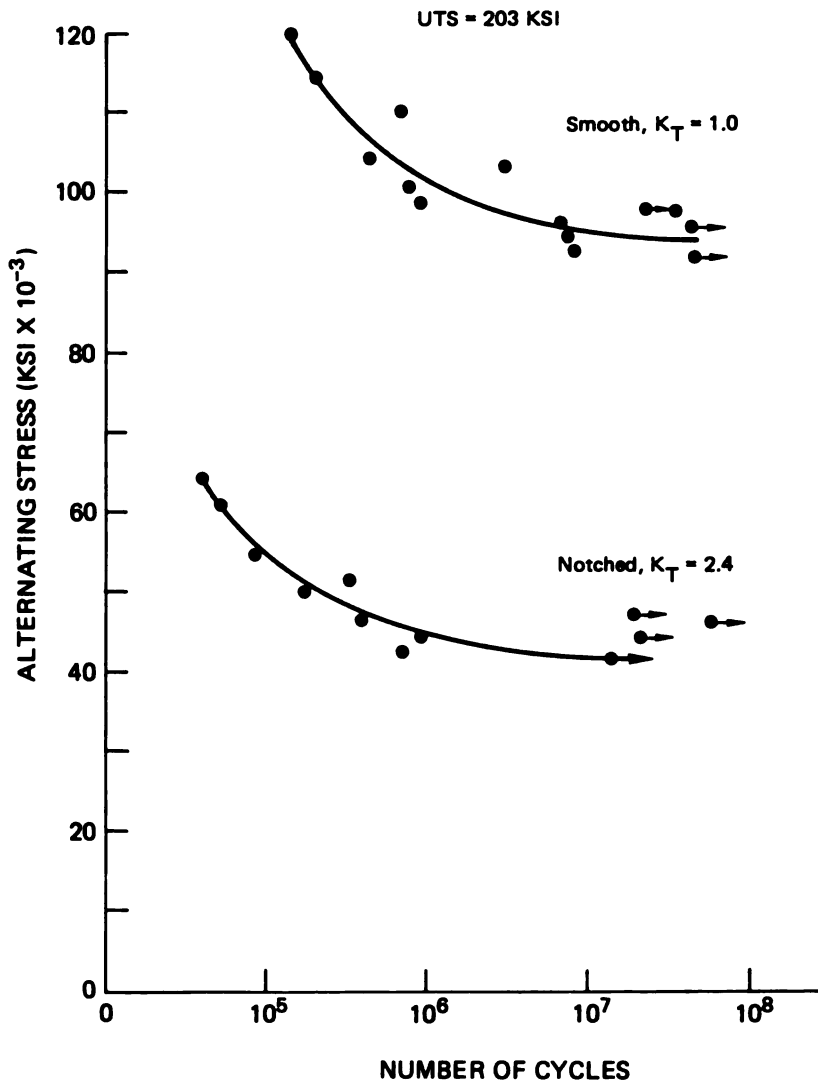


FIGURE 27 Fatigue characteristics of X-2 steel smooth and notched rotating beam. (From Cunningham, *et al.*, 1974.)

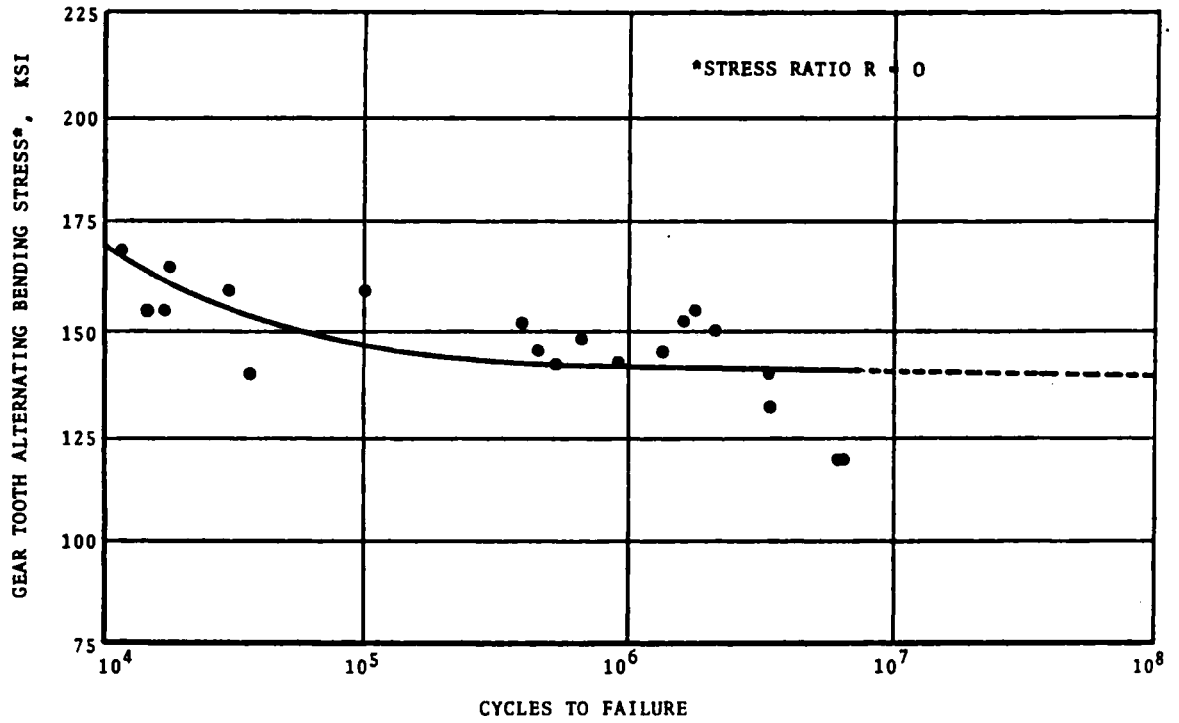


FIGURE 28 Single tooth bending fatigue of X-2M (0.15C) vacuum melted steel carburized and hardened R_c 61 case, R_c 46 core. (From Lemanski, et al., 1971.)

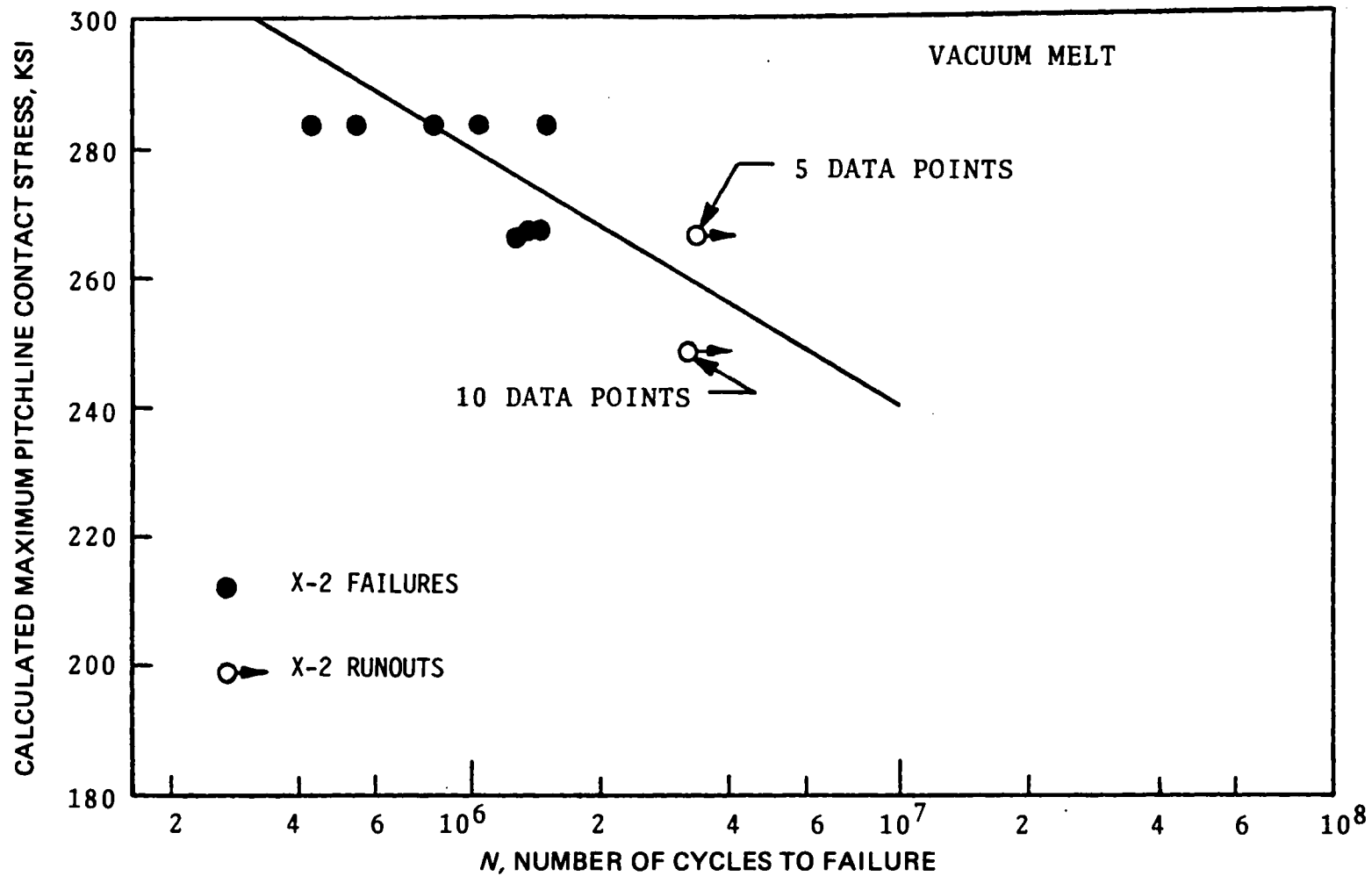


FIGURE 29 Rotating tooth contact durability testing for testing the combination of pitting and bending fatigue. Material was X-2 at 0.20 and carburized/hardened to R_c 60-62 case, R_c 53 core.

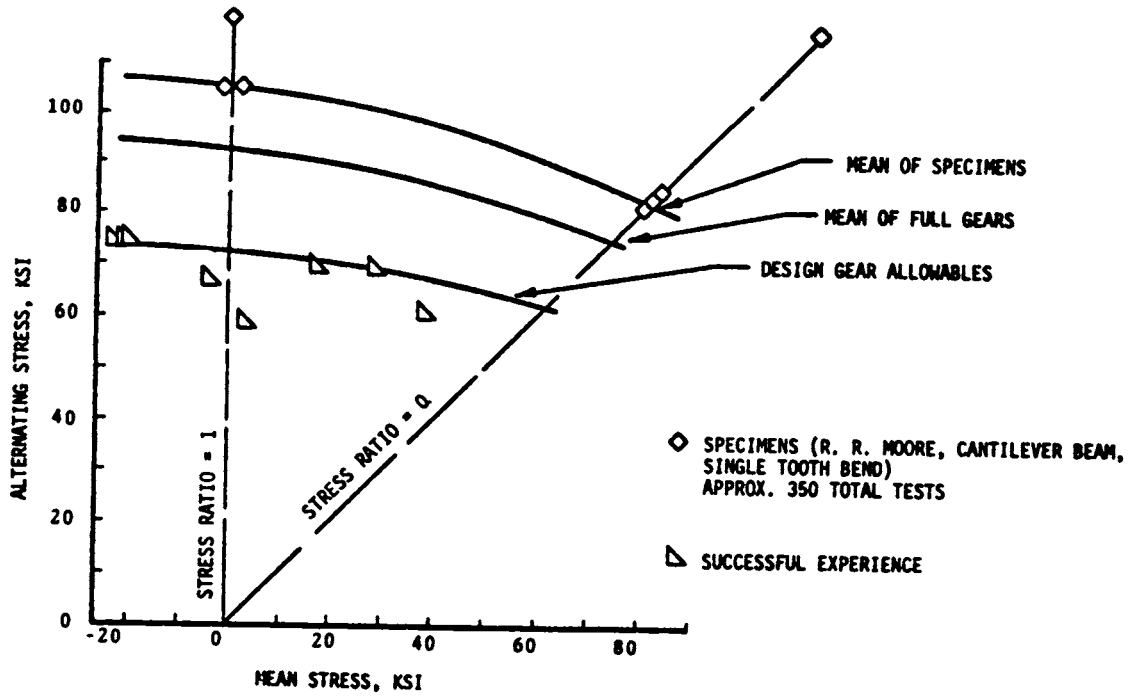


FIGURE 30 Combined stress Goodman diagram (comparing carburized, hardened, and shot peened gear and laboratory test specimens for X-2M or 9310). (Based on internal report, Boeing Vertol Company.)

2. Bend Testing

Although the Charpy V-notch (CVN) impact test is recognized as a common test for evaluating relative fracture resistance, it is best used to evaluate quenched-and-tempered properties of only the core material. The test unfortunately does not lend itself to evaluating the steels in the carburized condition. There also are several unanswered questions about extending the behavior observed in small uncarburized CVN specimens to the behavior expected in larger section sizes, especially sections surrounded by a hard carburized case. The alternatives to handle the section-size extrapolation and provide valid laboratory test results are to use plane-strain fracture toughness or to use test specimens having a geometry similar to that of the final component. Both techniques have been used in development laboratories; either can give acceptable results.

The testing approaches that have been employed to evaluate various aspects of the fracture properties of carburized and hardened specimens include high-cycle fatigue properties, impact fatigue and impact fracture stress properties, and fracture toughness. The test specimens used in these three basic tests were kept as consistent in cross-sectional area and geometry as possible. The cyclic and impact tests are performed on specimens having the same critical dimensions given in Figure 31 whereas the fracture toughness specimens are precracked slow-bend specimens having the dimensions of an unnotched standard-size Charpy impact specimen and tested in the carburized condition.

The fracture properties evaluated are primarily case fracture properties, independent of the strength properties of the core. The energy required for fracture was not considered as a fracture criteria in this study because of the large dependency of this measurement on the hardness and carbon content of the core. All fracture properties are expressed in terms of the stress required to cause fracture of the carburized case. The case depths are thick enough to insure against yielding in the core prior to fracture of the case. In such specimens, it is believed that strength of the core influences the fracture stress of the case is through its influence on the resulting residual stress distribution in the case. It is oftentimes necessary to measure residual stress distribution in the carburized and hardened case to explain apparent discrepancy in results.

3. High-Cycle Fatigue

Since carburized components can fail by fatigue at low stresses, surface finish and carburizing processing (case

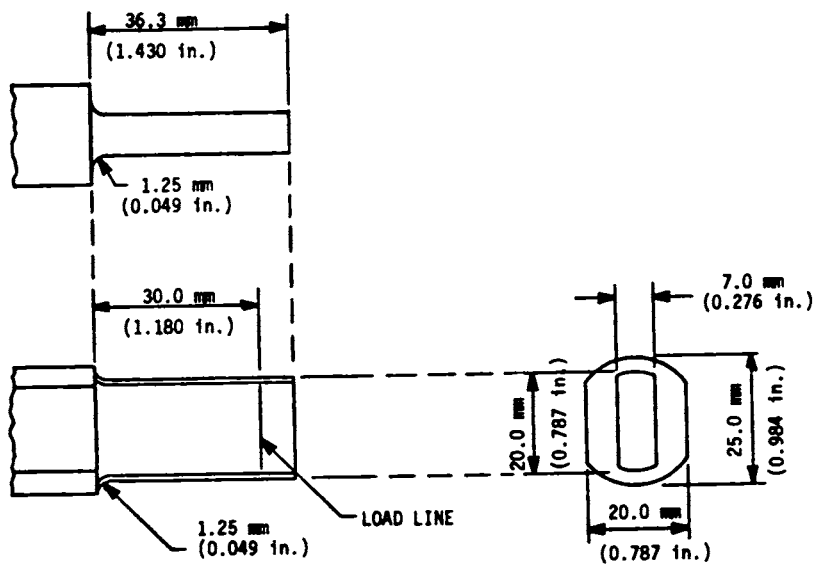


FIGURE 31 Dimensions of the critical "tooth" portion of high-cycle fatigue, impact fatigue, and impact fracture stress specifications. (Based on data furnished by D. E. Diesburg, 1979).

depth and surface carbon) have been found to influence the high-cycle fatigue limit more than does alloy content. Steels having adequate surface strength (approximately HRC 60) exhibit similar high-cycle fatigue limits as determined by three-point bend tests on specimens with a "tooth" geometry (Figure 31). As long as the candidate steels exhibit a high surface hardness and sufficient case depth, the fatigue properties of the steels should be compatible with design allowables.

4. Impact Fatigue and Impact Fracture Stress

Impact fatigue properties of carburized specimens are obtained with specimens having the same "tooth" dimensions as the high-cycle fatigue specimens; however, the specimens are cantilevered instead of being loaded in three-point bending as they are in high-cycle tests. The specimens are held rigidly in the anvil of a Charpy impact machine in a manner similar to that used in testing Izod impact specimens. The specimens are impacted with an instrumented Izod striker attached to a 27-kg pendulum hammer. The drop height is reduced to less than an inch to prevent complete fracture on the first impact. The total range of energy input levels is between 4 and 6 J. The energy level that is chosen will fracture the specimen in a reasonable length of time. In most instances, the instrumentation of the fracture event allows the number of cycles for crack initiation to be separated from the cycles required for complete fracture. The typical fatigue curves shown in Figure 32 illustrate the ability of this test to differentiate materials that can absorb impact under fatigue conditions.

The impact fracture stress of each steel is determined using the same setup as that used for the impact fatigue tests. The drop height is raised to cause fracture with one impact. The instrumented striker measures the fracture load which is then used to calculate a stress using bending equations and a stress concentration factor of 1.56. It has been shown that impact fatigue properties are related to the impact fracture stress. Figure 33 shows the correlation between the two tests for a broad range of steels. The higher the impact fracture stress, the more resistant the steels are to fracture in impact fatigue. EX55 and SAE 4817 are the most resistant to fracture of any of the steels evaluated in this series of experiments.

Work (see Figure 16) dealing with fracture toughness using slow-bend Charpy specimens provides the necessary explanations relating the broad spread of cycles for a specific fracture stress under the impact loading systems. It has been noted that the SAE 4820 and EX32 steels with

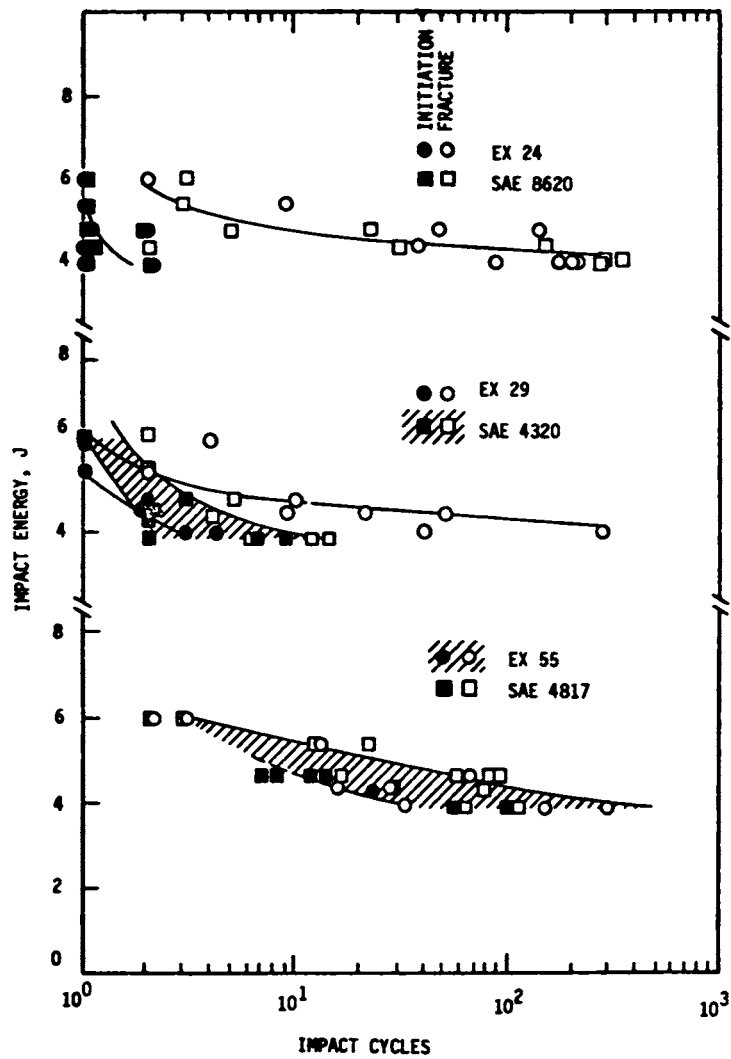


FIGURE 32 Comparison of the impact fatigue properties of carburized steel specimens. (Based on data furnished by Climax Molybdenum Co. of Michigan.)

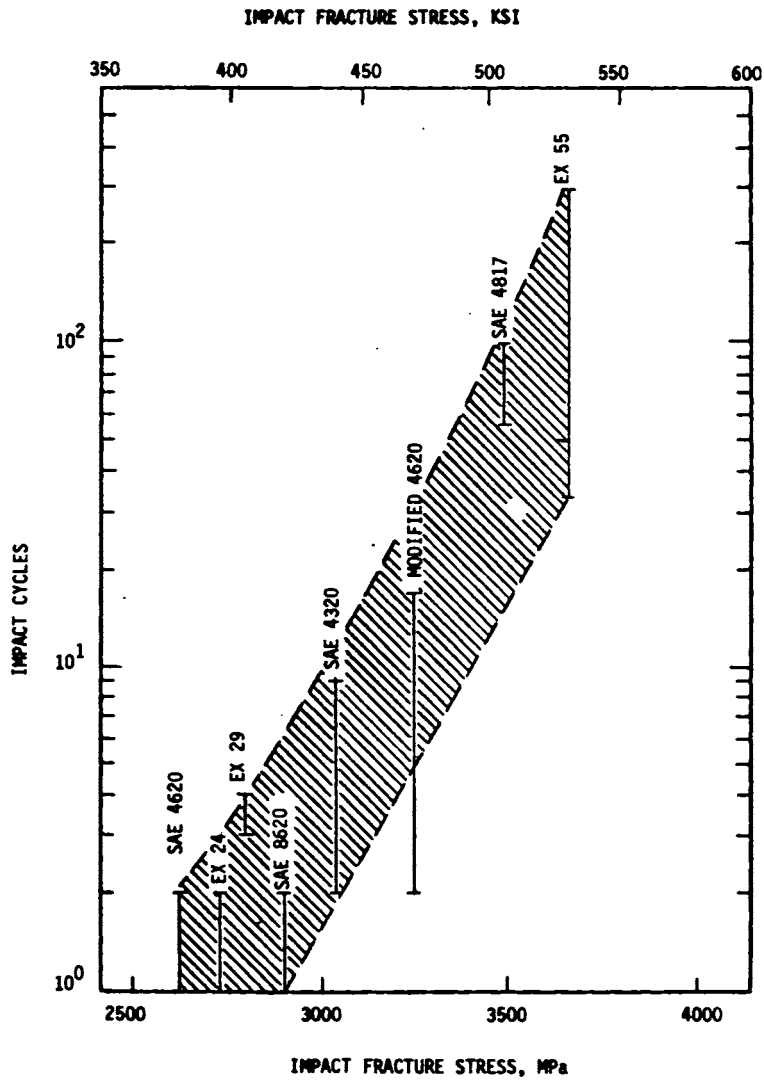


FIGURE 33 Relationship between impact fracture stress and the number of impacts required for crack initiation under repeated impacts at an energy level of 4.0 J (35 in.-pounds). (Based on data furnished by Climax Molybdenum Co. of Michigan.)

similar hardenability but with large differences in nickel and, therefore, austenite content showed very little difference in fracture toughness in this carburized case even with the influence of residual stress eliminated as shown in Figure 34. A possible explanation illustrated in Figure 35 shows how the impact fracture stress was observed to be dependent upon the compressive residual stress in the carburized case.

When cantilever stressed samples of SAE 9310 and X-2H were tested under fatigue vibration, the high temperature application (350°F) seemed to show no advantage of hot hardness (strength) and stability (resistance to softening). These data are presented in Figures 36 and 37.

It is interesting to note that the slow-bend Charpy specimen has been used to differentiate the advantages in vacuum and/or special melting systems to improve the steel toughness. Work conducted on SAE 4320 composition is summarized in Figure 38. Similar data on high temperature gear steel is not available at this time.

In summary, bending/impact fatigue testing techniques on carburized and hardened samples representing section sizes of actual gears offer an excellent opportunity to discriminate between materials and processes leading to actual gear manufacture and scheduled product testing.

5. Nondestructive Evaluation

There is an urgent need for nondestructive diagnostic equipment capable of detecting surface and near subsurface defects as well as residual stress in gears. Such an apparatus has been developed for rolling element bearings at Southwest Research Institute (SWRI) for the Army. It is designated CIBLE (Critical Inspection on Bearings for Life Extension) by SWRI.

The application of the CIBLE system to gears will be straightforward since the major concerns are the same for both gears and bearings: surface and near subsurface flaw detection and residual stress evaluations. While problems of implementation still exist, the CIBLE system is a viable piece of diagnostic equipment for bearings. The application of CIBLE techniques to gears is complicated by the more intricate and varied scanning procedures required for gears but, in principle, the same techniques should be applicable.

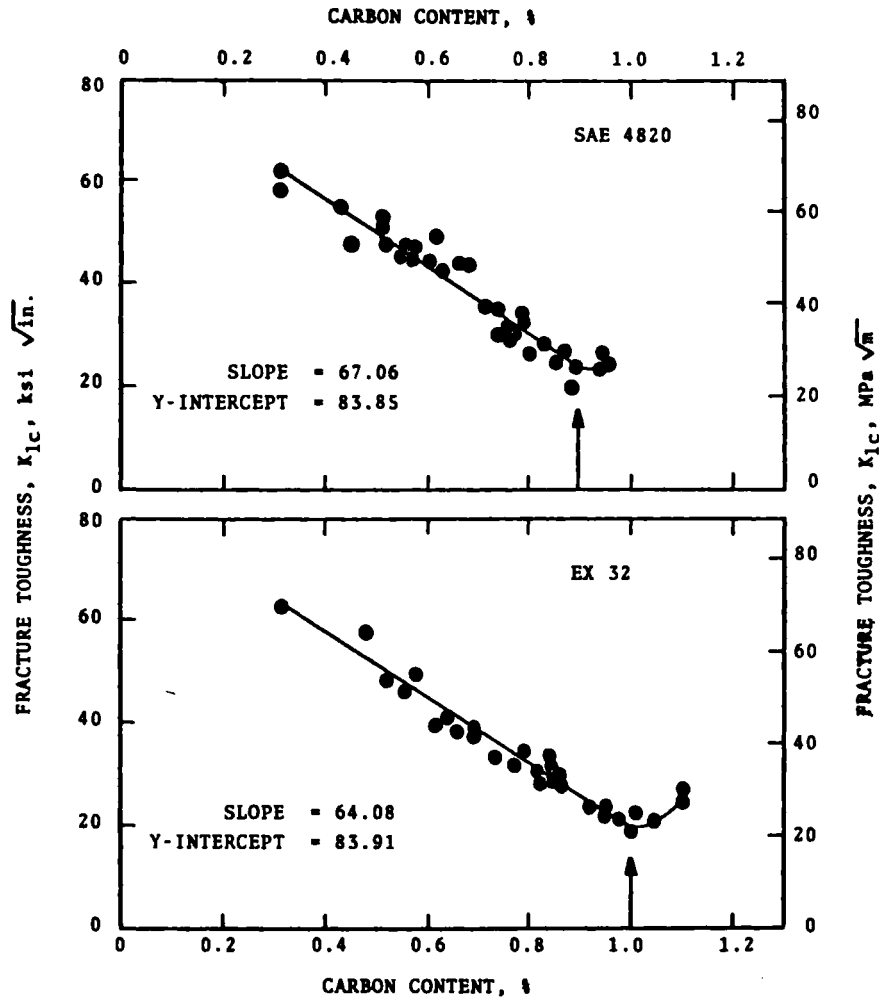


FIGURE 34 Influence of carbon content on fracture toughness (influence on residual stress has been eliminated). (Based on data furnished by Climax Molybdenum Co. of Michigan.)

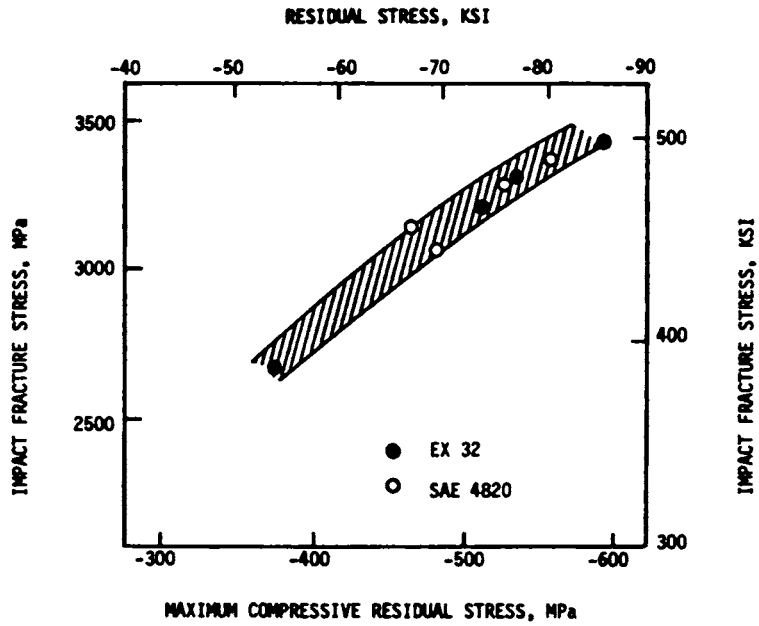


FIGURE 35 Relationship between impact fracture stress and compressive residual stress in the outer 0.76 mm (0.030 in.) of the carburized case. (All cases contained <50% retained austenite.) (Based on data furnished by Climax Molybdenum Co. of Michigan.)

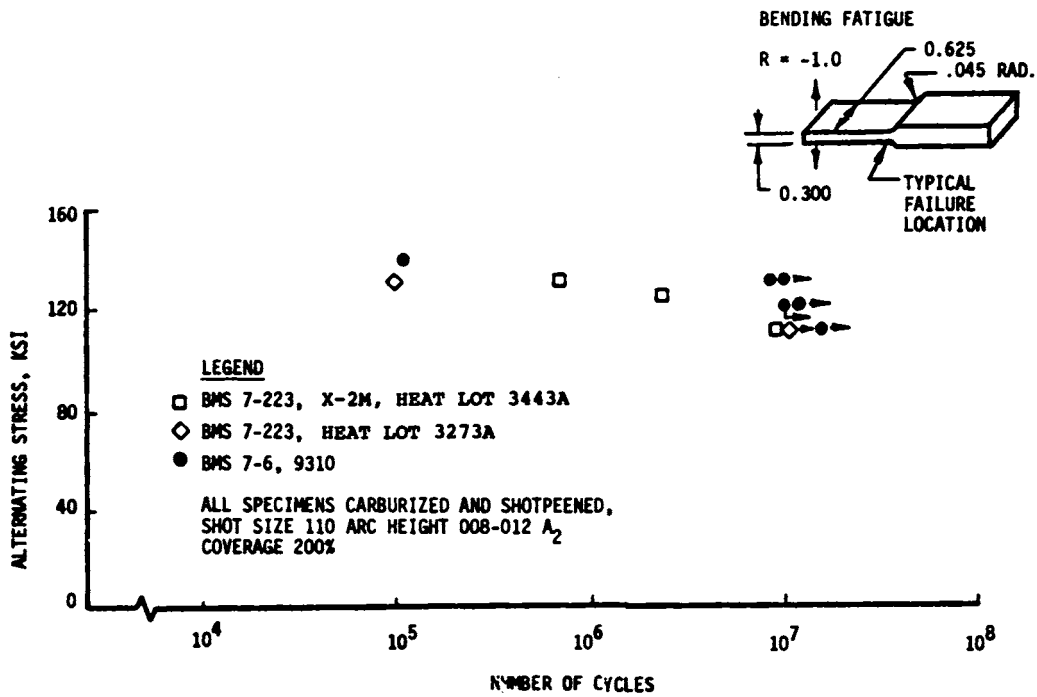


FIGURE 36 S/N data: gear materials evaluation tested at room temperature for the steels shown. (From Mack, 1977.)

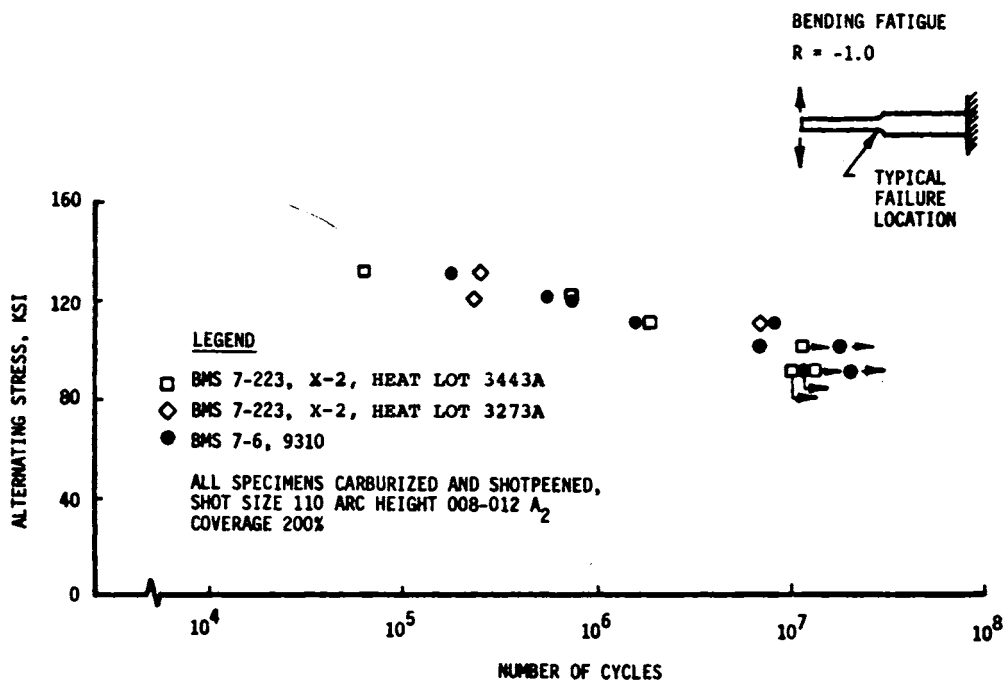


FIGURE 37 S/N data: gear materials evaluation tested at 350°F for the steels shown. (From Mack, 1977.)

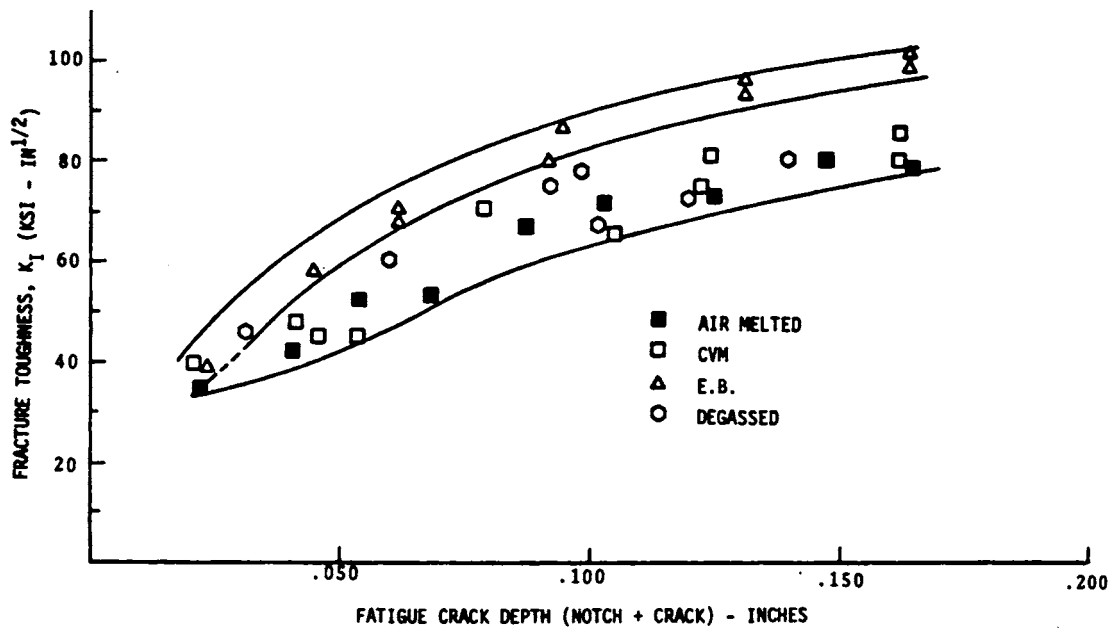


FIGURE 38 Fracture toughness (K_I) versus fatigue crack depth for carburized 4320 steel on slow bend Charpy specimens at 0.50 carbon and 0.030 to 0.035 in. case depth.

APPENDIX A
PITTING FATIGUE ANALYSIS

1. TEST PROCEDURES

a. Specimens

A group of test rollers usually consists of 10 roller/specimen pairs that have been accurately ground before heat treating as a single batch. Specimens for bearing type tests also are ground after heat treatment. Differences in surface carbon content and hardness and geometry errors within the batch are minimized.

b. Geared Roller Test Machine

The load, lubricant type, lubricate temperature, and the amount of sliding between the roller and specimen is held constant for each pair tested. The load range for the test group is determined by the application (i.e., bearings or gears).

The oil temperature and surface velocity are controlled over a range that ensures a ratio of oil film thickness to surface roughness equivalent to that experienced by the application. This ratio is significant because it is a measure of the amount of metal-to-metal contact in the elliptical contact zone. Complete asperity separation would ensure extremely long pitting life. Pitting life decreases as asperity contact increases.

c. Data Analysis

Data analysis involves three basic steps: (1) a plot of the uncorrected data for maximum Hertz contact stress vs cycles to failure; (2) a plot of the ratio of oil film thickness to composite surface roughness vs cycles to failure for a constant contact stress; and (3) a Weibull plot of percent failures vs cycles to failure for a constant contact stress and a constant ratio of oil film thickness to surface roughness.

Step 1--The roller/specimen geometry and the load determine the contact stress. The maximum Hertz contact stress (S) is given by:

$$S_C = 2.92 \times 10^4 P^{1/3},$$

where P = load on a 1-in. diameter specimen mated with a 5-in. diameter roller with a 10-in. radius crown.

A graph of contact stress (ordinate) vs life (abscissa) is plotted for the raw data in which some variables, notably oil film thickness and surface roughness, are not constant. A multiple stepwise regression analysis determines the best fit curve by least squares. A contact stress is selected (e.g., 400 ksi), and the data points are projected parallel to the median curve to that level. A corrected life is determined. This might be repeated for other stress levels as needed.

Step 2--Dawson's equation for calculating minimum oil film thickness is:

$$h_0 = 1.6R G^{0.6} U^{0.7} W^{-0.13},$$

where R = relative radius of curvature, $G = \alpha E'$ where α = pressure-viscosity coefficient of the lubricant, and E' = reduced modulus of elasticity ($E' = 32 \times 10^6$ psi), and $G = 5000$ for steel lubricated with a paraffinic oil.

$$U = \mu_0 V / E' R$$

where μ = absolute viscosity of the lubricant corresponding to the surface temperature of the test specimen and

$$V = \frac{V_{\text{spec}} + V_{\text{roller, in./sec}}}{2}$$

$$W = \frac{\omega}{E' R}$$

where ω = load/unit length of contact = $P/(4a/3)$ and a = semi-major axis of elliptical contact.

Average velocity, oil viscosity, modulus, Poisson's ratio, load, and specimen geometry are accounted for in Dawson's equation.

Composite surface roughness $\sigma_c = [(\text{spec roughness})^2 + (\text{roller roughness})^2]^{1/2}$ longitudinal centerline average values are used for these calculations.

The ratio of h_0/σ_c = corrected life (abscissa) data from Step 1 is plotted and a multiple stepwise regression analysis, determines the mean curve for which contact stress is a constant. A reasonable value for the h_0/σ_c ratio (0.5) is selected and the corrected data points are projected to that h_0/σ_c value along lines parallel to the mean curve.

Step 3--The corrected data points for a constant stress (400 ksi) and a constant h_0/σ_c ratio (0.5) ratio are plotted on Weibull paper and the least squares best fit curve is drawn through the points. A point on a graph of corrected

stress (ordinate) vs corrected life (abscissa) then can be plotted for any percent failure life (e.g., B_{10}) for the corrected h_0/σ_C ratio. Repetition of steps 1 and 2 allows the life for any stress and h_0/σ_C value to be plotted on one graph to define the separate effects of contact stress and the h_0/σ_C ratio for any desired failure probability.

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APPENDIX B

FRICTIONAL FORCES AND SHEAR STRESSES

Excerpt from a Paper Prepared by
B.W. Kelley for the Caterpillar Tractor
Symposium on Wear, Fatigue, and
Fretting, Indianapolis Chapter, 1960

Examination of shear stresses and ranges of shear on fixed planes as the contact load passes are extremely significant because no failure will occur under even very high compression of a hydrostatic nature (equal compression in all directions). One therefore must look for large differences in compression that are capable of producing high enough shearing stresses to fail the material.

Investigations of this type can be conducted using mathematical tensor analysis or Mohr circles (Zizicas, 1955). The work reported here was aided by the use of an automatic digital computer. A number of planes were examined for maximum shear, and ranges of shear, for the negative sliding surface are shown in Figure B-1. The angle that each plane makes with the surface in contact is illustrated simply by the position of each small plane on a quadrant representing as its surface all possible angles which could be examined. The magnitude of the maximum shears is shown on each small plane. The maximum range of shear was found to be about 0.394 times the maximum Hertz stress or 118,000 psi which occurred on the same plane showing the maximum shear of 95,000 psi. It is reasonable that shears of this magnitude, when subject to some reversal, will cause fatigue fracture of even the strongest commercially used steels.

Considerable study must be given to qualitatively understand the physical causes of the stresses in each of the planes. However, it is particularly interesting to note that many of the planes illustrate shear stresses that are very close to the maximum. In a case where the shear stresses are large in many planes, it is not difficult to surmise why even very hard steels, which are not generally considered to be ductile, will flow at the surface. Such surface metal flow frequently is seen on hypoid gears, for instance, where thermal stresses can play an important part because of high sliding velocities even though the coefficient of friction is not exceptionally high (Almen, 1950; Barwell, 1958).

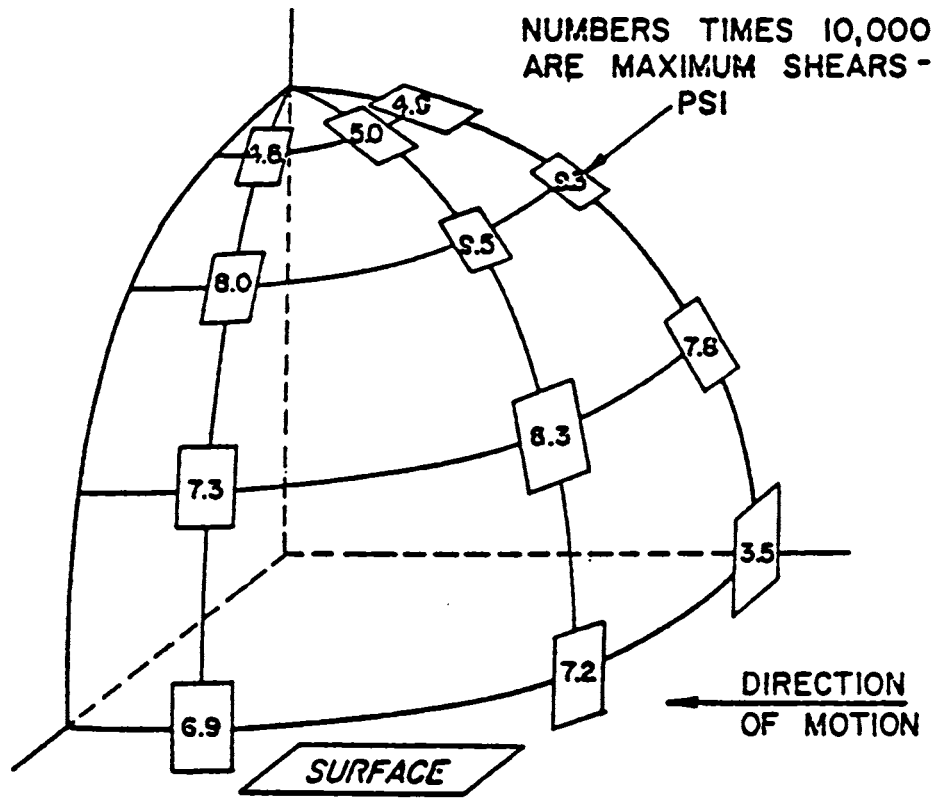


FIGURE B-1. Maximum shear stresses occurring at various angles to surface in thermal stress example.

The examination explained in this report is admittedly abbreviated. It is intended to impress upon the engineer the importance of increased studies on the effect of surface temperatures not only on scoring and pitting, but also on wear of various forms. It is felt that studies of this nature combined with further studies on the coefficient of friction will bring about increased understanding of surface damage and that this, in turn, will bring about increased efficiency in the attention to surface damage problems.

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APPENDIX C

SCORING PHENOMENA

Prepared for the Committee by
B.W. Kelley
Caterpillar Tractor Co.

1. INTRODUCTION

One of the most difficult forms of machine part failure to examine and understand is that of surface damage because it is a function of so many forms of activity that occur on, in, and under the surface proper. One cannot, within the scope of any paper of reasonable length, deal with more than one of these variables, and in this consider only a few forms of failure. The variable chosen for discussion here is surface temperature, which is primarily caused by friction heating.

This paper will deal principally with two forms of failure--scoring and pitting--and application will be primarily to very hard surfaces, such as those that are found on gears, anti-friction bearings, and camshafts. Although a thorough study of the subject requires a fair amount of theoretical background, it is felt that the thermal characteristics created during friction heating are not well understood by the engineer and that a qualitative feeling for at least the principles involved can be useful in developing a respect for the importance of this subject to the damage of surfaces.

2. PRINCIPLES

For the purposes of this discussion, several assumptions are made. The first is that perfect thermal contact occurs between the two parts that are involved. This assumption is valid if thickness of the oil film between the two mating parts is not great enough to allow a significant temperature differential or if the quantity of lubricant will not materially remove heat directly from the contact area. Perfect thermal contact therefore means that the temperature on one of the inner faces can be assumed to be the same as the temperature on its mating surface and that the total primary dissipation of heat is into the two elements in contact and not through the lubricant in the contact area. The second assumption made is that the bulk temperature of the two parts in contact is stable (i.e., the

amount of heat being removed by external dissipation is equal to the heat input due to friction). This assumption is valid in many practical applications for machine elements. A third assumption is that the velocity of the surfaces is sufficiently high so that the temperature that occurs in contact will not significantly precede the area of contact. This velocity actually is not very high; therefore, even though it is somewhat less than required on an existing part, the error is still small. These principles have been clearly set forth by Blok (1955 and 1937) and will not be discussed further, but for those who are more interested in the subject, the references should be required background reading.

The assumptions that have been made can be readily fulfilled in the case of cylinders, such as those shown in Figure C-1, pressing together with a rectangular shape contact area and having rotation velocities V and V . The band of contact of width b constitutes the shape of the heat source. The heat intensity which is proportional to the friction at each point in the contact pressure is set forth in the elastic analysis by Hertz (1895). Blok's approximate solution, which will be used here, assumes a parabolic distribution for mathematical simplification, but this produces an insignificant error. The distribution of temperature at the surface shown in Figure C-2 is of interest and importance. A lag in the maximum temperature point occurs in this case about midway between the center of contact and the trailing edge. The reader will later see that this trailing characteristic is important in the consideration of thermal stresses.

3. SCORING

Scoring of surfaces is gradually becoming more recognized as a direct function of surface temperature. In the case of unlubricated surfaces, this temperature generally is considered to be the softening or melting point of the materials. In the case of solid lubricant films, it is directly related to the softening point of these films (Bowden and Tabor, 1950). More recently it has been found that surface temperatures far lower than melting points of the materials in the surfaces bring about the onset of scoring of liquid lubricated surfaces (Rabinowitz and Tabor, 1951). Thus, it is becoming more recognized that nonreactive (no extreme pressure additives) mineral oils and, very likely, most other commercial oils and synthetics (Murray et al., 1954; Cowley et al., 1956) have a critical temperature beyond which they are no longer capable of satisfactory lubrication. Many speculations on the exact mechanism over which the temperature has control have been

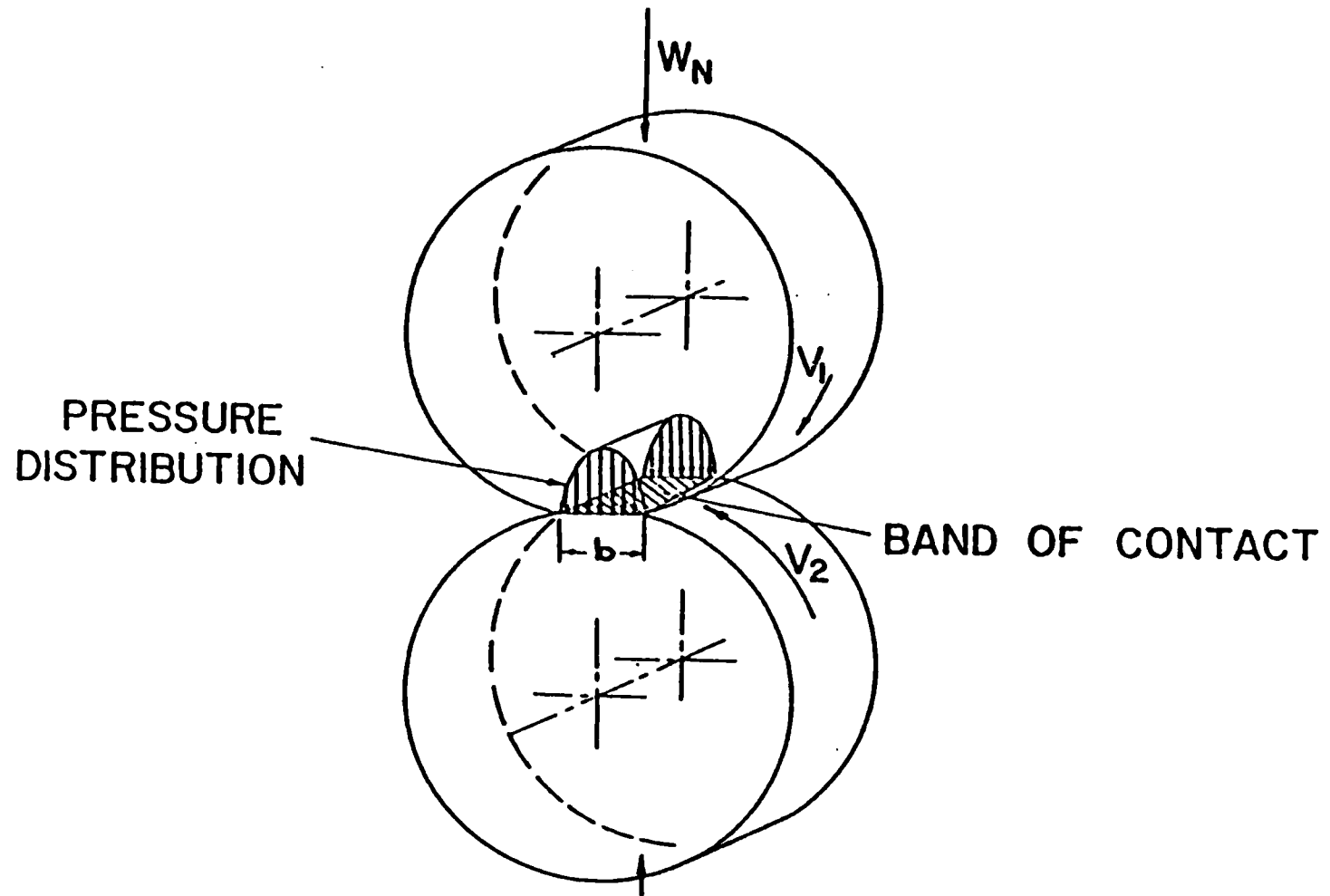


FIGURE C-1. Contact conditions.

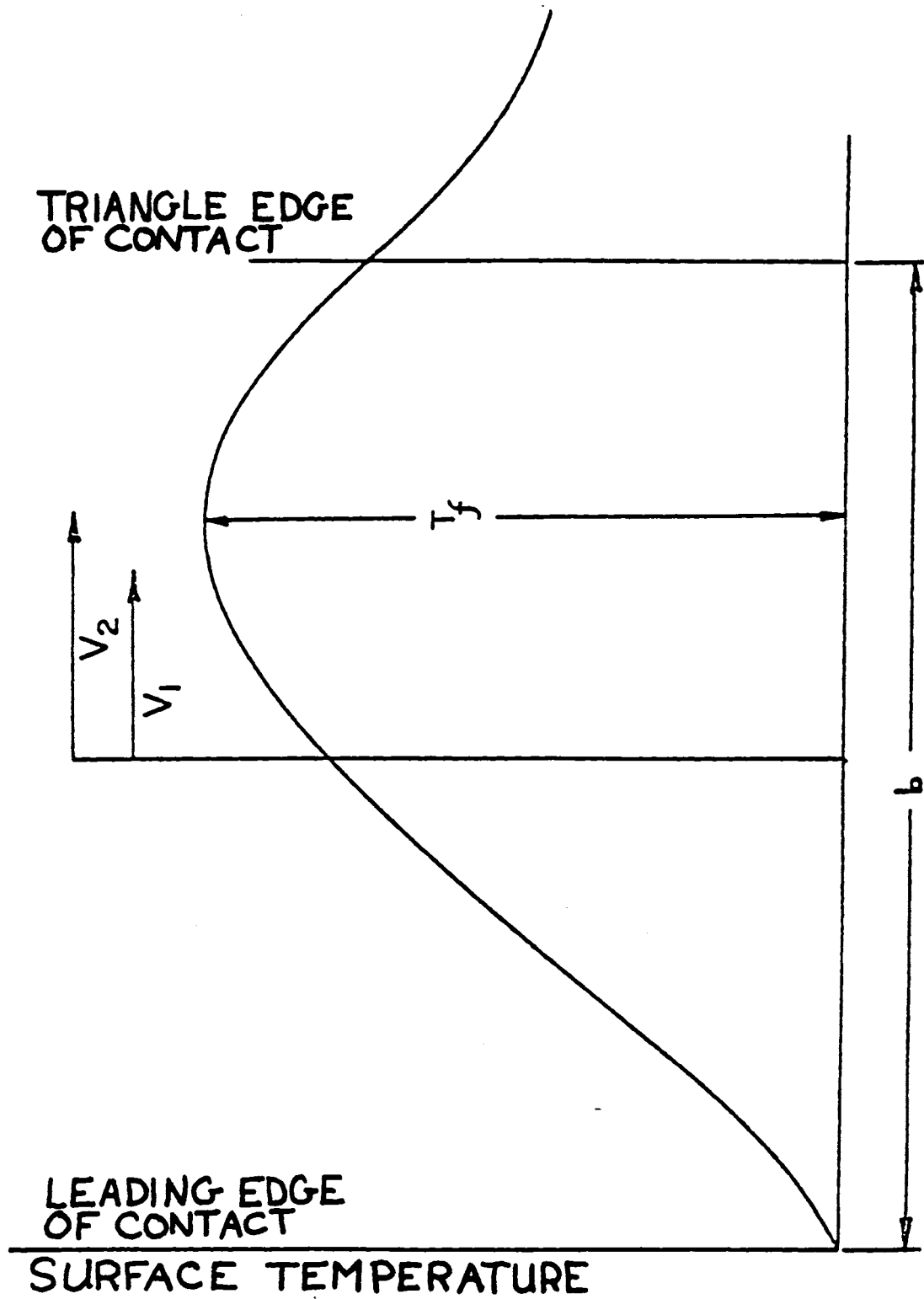


FIGURE C-2. Shape of surface temperature rise in band of contact.

made. The most promising of these hypotheses perhaps is one concerning the failure of the adsorption characteristics of the oil, called desorption. Roughly this may be looked upon as the failure to "wet" the surface.

Scoring takes on different appearances depending principally on the characteristics of the material, its surface characteristics, and the lubricant. The carburized and hardened surface scored in a nonsurface reactive mineral oil is badly torn and appears "roped" (Figure C-3). The same material scored in an extreme pressure (EP) lubricant appears softer with more material flow. Very likely, the lack of tearing that occurs with EP oil is associated with the protective contamination of the surface by the additives and the higher surface temperature required to produce the failure. A similar appearance frequently will be seen on a surface that has been treated (e.g., on one that has been Parko-Lubrited).

With regards to the maximum mean temperature due to frictional heating, T_f (Figure C-2), that is produced, a qualitative examination of the formula derived by Block (1937) will be revealing:

$$T_f = K \frac{fW_n (V_1 - V_2)}{(C_1 \sqrt{V_1} + C_2 \sqrt{V_2} \sqrt{b/2})}$$

where K = constant,
 f = coefficient of friction,
 W_n = normal load per unit length,
 V_1 and V_2 = surface velocities,
 C_1 and C_2 = constants of material which include thermal conductivities, specific heats and densities,
 b = width of band of contact, and
 T_f = maximum mean surface temperature (sometimes called "flash temperature").

It is important to notice the velocity relationship that is obvious in the formula. From any given condition, if V_1 and V_2 are increased but the difference remains the same, then the resulting temperature is reduced. This fact often has been overlooked in previous scoring criteria (e.g., PV and PVT) that are proportional to the heat intensity but not the surface temperature.

It will be found that if V represents the sliding velocity or the difference between V_1 and V_2 , then the scoring resistance will vary as \sqrt{V} . This velocity relation has been used, for instance, by Lane and Hughes (1951) to



FIGURE C-3. Scoring of carburized and hardened gears in straight mineral oil.

further verify the temperature criteria for scoring on gears.

Another interesting relation is found between maximum Hertz pressure, P_0 (Figure C-1), and temperature. It can be shown, for instance, that the maximum surface temperature varies as $P_0^{3/2}$ instead of directly proportional to P_0 for a fixed values of V and geometry such as the roller radii. Increases in radii of curvature, however, will affect the surface temperature much less than might be expected and, as in the case of gears where the use of larger radii of curvature such as in the adoption of higher pressure angles, frequently will be accompanied by higher values of V that can easily create no improvement in surface temperature and scoring resistance at all.

In previous publications on gear tooth scoring (Kelley 1953), it has been shown that distribution of the load and speeds on gear teeth can be expressed as a formula based on the temperature flash formula which includes an empirical factor of surface roughness. It was assumed that the coefficient of friction was a constant for all of the tests and was recognized that elements foreign to the fundamental nature of the formula were introduced.

Since this work, investigations into the coefficient of friction as influenced by velocity have provided the formula presented, which can be used with improved success. Allowing for the similarity of material and simplifying, the formula can be written as:

$$T_f = T_b + K \frac{fW_n \sqrt{V_1 - V_2}}{(1 - s/50) \sqrt{b/2}}$$

where T_f = total surface temperature,

T_b = bulk stable temperature of the part, and

S = surface finish r.m.s. micro inches

It is obvious that the empirical value of S for surface finish has not yet been removed. The work on the coefficient of friction is not complete, and it may be found that the surface finish will affect this value strongly. As might be anticipated, the coefficient of friction is reduced as the velocities increase thereby giving more advantage to the higher speed gears than would have been predicted by the previous formula, which assumed friction was constant. The reader will find additional verification of the critical temperature criteria throughout the literature (Lane and Hughes, 1951; Thomas and Hoersch). It is important to note that the proponents of this hypothesis are increasing as time goes on.

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