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**BEARING, GEARING, AND LUBRICATION TECHNOLOGY**

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ABSTRACT

Results of selected NASA research programs on rolling-element and fluid-film bearings, gears, and elastohydrodynamic lubrication are reported. Advances in rolling-element bearing material technology, which have resulted in a significant improvement in fatigue life, and which make possible new applications for rolling bearings, are discussed. Research on whirl-resistant, fluid-film bearings, suitable for very high-speed applications, is discussed. An improved method for predicting gear pitting life is reported. An improved formula for calculating the thickness of elastohydrodynamic films (the existence of which help to define the operating regime of concentrated contact mechanisms such as bearings, gears, and cams) is described.

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THE NASA LEWIS RESEARCH CENTER has supported a sustained R&D effort in Mechanical Components technology since the 1940's. This research has been both fundamental and applied, with the applications centering on rocket engine turbopumps, aerospace power generation systems, and aircraft gas turbine engines and transmissions. These are, in the main, high-performance products which require sophisticated research and development programs, and in which cost is usually secondary to performance and reliability. Despite the fact that the goals of NASA's R&D efforts in mechanical components are somewhat different from those of R&D efforts on highly cost competitive commercial products, many of the results of NASA's research are being utilized in the commercial sector.

The objective of this paper is to report results of selected NASA research programs on rolling-element and fluid-film bearings, gears, and elastohydrodynamic lubrication which may be of interest and value to the automotive industry. These results are presented in greater detail in Refs. 1 to 8. In particular, it is hoped that the research reported can be applied to improve the designs of gear systems and high-speed rotating systems such as turbine engines and turbochargers.

#### SYMBOLS

- a semimajor axis of contact ellipse,  
m (in.)
- b semiminor axis of contact ellipse,  
m (in.)
- C journal bearing radial clearance,  
m (in.)
- D journal diameter, m (in.)
- E modulus of elasticity,  $N/m^2$  (psi)
- $E^1$  modified modulus of elasticity,

$$\frac{2}{\frac{(1 - \nu_1^2)}{E_1} + \frac{(1 - \nu_2^2)}{E_2}}$$

- e Weibull slope
- F normal applied load, N (lb)
- f gear tooth face width, m (in.)
- G dimensionless material parameter,  $E'/P_s$

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$H_{\min}$	dimensionless minimum film thickness, $h/R_x$
$h$	film thickness, m (in.)
$k$	ellipticity parameter, $a/b$
$L$	journal bearing length, m (in.)
$L_{gm}$	life of gear mesh, millions of revolutions
$l$	involute profile arc length, m (in.)
$l_c$	length of gear contact line, m (in.)
$M$	rotor mass, kg ((lb sec <sup>2</sup> )/in.)
$\bar{M}$	dimensionless mass, $MP_a (C/R)^5 / 2\mu^2 L$
$N$	rotative speed, rpm
$N_1$	number of teeth, driving gear
$N_2$	number of teeth, driven gear
$P_a$	ambient pressure, $N/m^2$ (psi)
$P_s$	asymptotic isiviscous pressure, $N/m^2$ (psi) (Ref. 3)
$p$	load-life exponent
$R$	effective radius of curvature, m (in.)
$r$	{ journal bearing radius, m (in.) radius of curvature, m (in.)
$r_1, r_2$	pitch circle radii of driving and driven gears, m (in.)
$U$	dimensionless speed parameter, $(u\mu_a) / (E'R_x)$
$u$	surface velocity in x direction, m/sec (in./sec)
$W_P$	dimensionless load parameter, $F / (E'R_x^2)$
$W_t$	gear tangential load, N (lb)
$W_{tm}$	gear mesh dynamic capacity, N (lb)
$\alpha$	fraction of journal bearing lobe that is convergent
$\omega$	angular velocity, rad/sec
$\Gamma$	stability parameter, $(12\mu\omega R^3) / (P_a C^2 L)$
$\mu$	dynamic viscosity, $N \text{ sec}/m^2$ ((lb sec)/in. <sup>2</sup> )
$\phi_t$	gear transverse pressure angle, rad
$\psi_b$	gear base helix angle, rad

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$$\Sigma_D \text{ gear tooth curvature sum,} \\ (\cos \psi_b)(1/r_1 + 1/r_2)/(\sin \phi_t)$$

Subscripts:

1,2 solids 1 and 2

a atmospheric pressure

W whirl

x,y coordinate directions

## RESULTS AND DISCUSSION

ROLLING-ELEMENT BEARINGS - The failure mode for which rolling-element (ball and cylindrical, tapered, spherical, and needle roller) bearings are normally designed is fatigue. The combination of high-contact stresses between the rolling elements and races and the large number of stress cycles which are accumulated in a relatively short period of time can lead to fatigue cracks and surface pitting of the races and/or rolling elements. Thus, rolling-element bearings have a finite fatigue life expectancy in contrast to sliding bearings which are not subject to fatigue unless significant dynamic loads are present. For this reason designers often avoid the use of rolling-element bearings in many applications where they offer advantages which more than offset the potential disadvantages of a finite fatigue life.

Research over the past three decades has, however, resulted in new rolling-element bearing materials which have significantly greater resistance to pitting fatigue. As a result, materials are now available on a production basis, for the manufacture of rolling-element bearings which will provide service lives as long as two orders of magnitude greater than those of bearings manufactured in 1940. Figure 1 shows the chronological improvement in relative bearing life and some of the factors that have resulted in the dramatic life improvement. The major portion of the improved bearing life has resulted from the use of vacuum melted steels. Vacuum melting significantly reduces or eliminates the formation of hard oxide inclusions. That, plus careful control of the melt process to eliminate refractories and other foreign matter, results in bearing materials which are free of the hard inclusions and stringers which serve as nucleation sites for fatigue cracks.

As shown on Fig. 1, consumable electrode vacuum melting (CEVM, introduced in 1962) resulted in a five-fold improvement in bearing

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fatigue life. Then vacuum induction melting followed by vacuum arc remelting (VIMVAR) resulted in a further, very significant improvement in life. Details of the VIMVAR bearing testing are reported in Ref. 8. All of these tests were conducted with bearings of M-50 steel (0.8C, 4Cr, 4.3Mo, 0.25Si, 1V, Bal. Fe), which has become the standard material for aircraft turbine engine mainshaft bearings. VIMVAR M-50 steel is available at a small premium in cost over that of SAE 52100 steel.

The principal advantage in using rolling-element rather than sliding (fluid-film) bearings in high-speed turbo machines such as gas turbines and turbochargers is in reduced power loss. In a typical automotive size turbine engine, bearing power loss would be reduced by several horsepower. A lesser advantage would be lower starting torque. The disadvantages would be in cost, since rolling-element bearings are more expensive than sliding bearings. Fatigue life would not be a factor with today's improved materials.

FLUID-FILM BEARINGS - Conventional fluid-film journal bearings, although they offer cost and possibly life advantages over rolling-element bearings, are subject to destructive, self-excited whirl instabilities when they support rotors running at speeds above twice the first critical speed of the rotor-bearing system. Self-excited fractional frequency whirl can result in almost instantaneous bearing failure. The trend toward smaller, more compact higher power density machines has led to higher operating speeds, and many cases of rotors operating at several times the first critical speed of the rotor-bearing system. Figure 2 illustrates three common types of whirl resistant journal bearings: elliptical, lobed, and tilting pad. The most effective of these is the tilting pad bearing, but it is expensive and its complexity sometimes leads to assembly problems. NASA carried out a research program (Refs. 4 and 5) to thoroughly investigate the whirl characteristics of several types of fixed geometry journal bearings. The bearing tests were conducted in water to simulate the viscosity characteristics of liquid metals used in Rankine cycle power systems, and in a low viscosity oil. They are applicable to bearings operating in any fluid.

Tests were run on herringbone grooved journals (Fig. 3), Rayleigh step journals (Fig. 4), lobed journals (Fig. 5), and lobed bearings with both converging-diverging sectors or lobes and fully converging sectors or tilted lobes (Fig. 6).

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The whirl onset speed for each of these bearing geometries was determined for a range of clearances in each bearing type. Tests were run with the bearing axis vertically oriented and at zero radial load. As shown in Fig. 7, the tilted-lobe bearing with fully convergent lobes ( $\alpha = 1.0$ ) was the most stable of the five bearing types investigated. The region of stable operation for each bearing is below and to the left of each curve. These test conditions represent the most severe conditions possible for inducing bearing instability. It is recognized that some bearing loads exist in all practical applications so that the results obtained are conservative. It must also be recognized, however, that periods of bearing unloading, even though very brief, can occur in any application and can be of sufficient duration to induce whirl and cause failure.

With the results of Fig. 7 in hand, it was decided to conduct a thorough investigation of the tilted-lobe bearing to determine the influence of the number of lobes and the length-diameter ratio as well as the clearance on stability. Bearings with 3, 5, and 7 lobes and length-diameter ratios of 0.2, 0.5, 0.75, and 1.0 were tested in both oil and water (Ref. 5). This range of length-diameter ratios encompasses those which would be expected in almost all applications. The results of these tests are illustrated in Fig. 8. The results are plotted as dimensionless mass  $M$  vs dimensionless stability parameter  $\Gamma$ . A journal bearing stability analysis (Ref. 9) indicated that these are the important parameters to consider. Variables included in these parameters are rotor mass  $M$ , ambient pressure  $P_a$ , bearing radial clearance  $C$ , bearing radius  $R$ , bearing length  $L$ , fluid viscosity  $\mu$ , and rotative speed  $\omega$ . In terms of  $M$  and  $\Gamma$ , the data for all bearings can be approximated as a straight line on a log-log plot. Although it was intended that these bearings all have fully convergent tilted lobes, the average offset factor was 0.77 rather than 1.0. This means that the average lobe was 77% convergent and 23% divergent instead of fully convergent. Test results with tilted-lobe bearings indicated that bearings with fully converged lobes were the most stable. Therefore, the results obtained with these bearings are conservative.

The experimental bearings used in this program were fabricated by offset milling. In production lots the bearings would be broached from an undersized circular bore to the finished geometry. They can be produced at a very nominal

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cost premium above conventional circular sleeve bearings and offer a practical economically competitive whirl-resistant bearing for many high-speed applications.

Intensive government-sponsored programs for the development of gas bearings for application in turbine engines are underway (Refs. 10 and 11). Gas bearings will be required in locations where the temperatures exceed the maximum capability of engine oils. A complete gas-bearing engine would offer the obvious advantages of reduced complexity through the elimination of several oil-lubricated bearings. This would, however, require a significant advance in gas-bearing technology.

**GEAR LIFE PREDICTION** - Gears may fail by scoring, tooth breakage, or tooth pitting (surface fatigue). Both scoring and tooth breakage can be avoided by proper design and lubrication practice. Under such circumstances, surface fatigue will determine the life of the gear set.

Current methods of designing gears to resist surface fatigue are based on the concept of a surface fatigue endurance limit. The current method (Refs. 12 to 14) of predicting gear tooth pitting failures is similar to that used for predicting tooth breakage. The Hertzian contact stress is estimated, and then modified with service condition and geometry factors to become the stress number. When the stress number is less than the surface fatigue endurance limit, the pitting fatigue life is assumed to be infinite. There is controversy among gear researchers regarding the existence of an endurance limit (Refs. 15 to 17).

In contrast, rolling-element bearing endurance life prediction, as developed by Lundberg and Palmgren (Refs. 18 and 19), assumes that fatigue life is a function of

- (1) The maximum orthogonal shear stress
- (2) The depth below the surface at which the maximum shear stress occurs
- (3) The stressed volume
- (4) The number of stress cycles

It assumes that an endurance limit does not exist. An easily applied gear life prediction method, based on the Lundberg-Palmgren theory, is developed in Ref. 6.

In essence, the life prediction method of Ref. 6 states that the life of the gear mesh in terms of the dynamic capacity of the mesh  $W_{tm}$  and the transmitted load  $W_t$  is

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$$L_{GM} = \left( \frac{W_{tm}}{W_t} \right)^p$$

The dynamic capacity of the gear-pinion mesh is calculated as follows:

$$W_{tm} = K_2 I_e \cos \psi_t (\cos \psi_b)^{11/9} (C_p)^{-35/27} \\ \times \left[ f N_1 \left\{ 1 + \left( \frac{N_1}{N_2} \right)^e \right\} \right]^{-2/9}$$

In Ref. 6 values of  $p = 3/2$ ,  $e = 3$ , and

$$K_2 = 132\,000 \text{ English units (lbf-in.)} \\ = 5.28 \text{ SI units (N-m)}$$

were derived based on theory and rolling-element bearing experience.

Experiments were conducted in Ref. 7 with vacuum arc remelted (VAR) 9310 steel spur gears. The results were used to modify the exponents  $p$  and  $e$  and the constant  $K_2$ . To obtain these values the gears were run to pitting failure at three tangential load levels. Results are shown in Fig. 9, plotted on Weibull coordinates. The data indicate that life is inversely proportional to load to the 4.3 and 5.1 power at the  $L_{10}$  and  $L_{50}$  life levels, respectively, as shown in Fig. 10. Thus, the value of  $p = 3/2$  derived in Ref. 6 appears to be too low for gears. The correlation of the revised NASA gear life prediction theory with test results is shown in Fig. 11. Life prediction at all three load levels is excellent. Comparison of the AGMA Standard 411.02 (Ref. 17) with experiment is shown in Fig. 12. The test data show a steeper slope for the load-life diagram than assumed by the AGMA Standard 411.02.

ELASTOHYDRODYNAMICS - Elastohydrodynamic (EHD) lubrication deals with the development of thin films of lubricant in the contacts between nonconformal machine elements in relative motion. Examples of nonconformal machine elements are the balls or rollers and raceways in ball and roller bearings, gear teeth and cams, and cam followers. The development of hydrodynamic lubrication theory, which deals with conformal machine elements such as sliding bearings (journal and

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thrust) began in the 1880's. In contrast, the first significant publication on EHD lubrication theory was in 1949 (Ref. 21). This was followed by further work on the line contact problem (Refs. 22 and 23) and publication of methods for calculating film thicknesses in point (elliptical or circular) contacts (Refs. 24 and 25).

None of the research reported in Refs. 21 to 25, however, deals with the detailed contour of the contact area, and, in particular, with the minimum film-thickness region. The minimum film thickness developed is critical to the performance, reliability, and life expectancy of the machine element. The thickness of the separating film must exceed, by a factor of two or three, the composite surface roughness of the contacting surfaces if the full fatigue life potential is to be achieved. Otherwise, surface distress such as micropitting, smearing, or scuffing occurs and the machine-element life is drastically shortened.

A comprehensive elastohydrodynamics analysis was undertaken at the NASA Lewis Research Center. The analysis (Refs. 3 and 26 to 28) was done in partial fulfillment of the requirements for a doctorate degree at Leeds University, England. A coupled solution to the elasticity and hydrodynamics equations was obtained (Ref. 26) and applied to the calculation of film thickness in EHD contacts with a wide range of ellipticity ratios (Ref. 27). Easily applied film-thickness equations for the cases of fully flooded contacts (Ref. 3) and contacts partially starved of lubricant (Ref. 28) were developed. For the fully flooded case the dimensionless minimum film thickness is given by

$$H_{\min} = 3.63U^{0.68}G^{0.49}W_p^{-0.073}(1 - e^{-0.68k})$$

A simplified expression for the ellipticity parameter is

$$k = 1.03 \left( \frac{R_y}{R_x} \right)^{0.64}$$

For a discussion of the consequences of partial starvation in the contact, the reader is referred to Ref. 28. While lubricant starvation and its effect on film thickness can be easily defined in a mathematical sense, there is no easy way to predict when it occurs in real machine elements.

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For well-lubricated bearings and gears, it is suggested that fully flooded conditions be assumed.

The critical parameter of importance is the ratio of film thickness to composite surface roughness

$$\Lambda = \frac{h}{\sigma}$$

where

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$$

For full film conditions without asperity contact,  $\Lambda$  should exceed three. For values of  $\Lambda$  between one and three some asperity contact will occur and some surface distress, with a consequent reduction in pitting fatigue life, may occur. For values of  $\Lambda$  less than one, gross surface distress is likely to occur with a very significant reduction in component life.

#### SUMMARY

Significant advances have been made in rolling-element bearing materials technology over the past three decades. These have resulted in improvements in bearing fatigue life of two orders of magnitude and open up many new possibilities for applying rolling-element bearings. The relative whirl resistance of several journal bearing geometries was investigated experimentally. The tilted-lobe journal bearing was found to have superior whirl resistance. It can be made inexpensively in production quantities, and offers promise for turbine engine and turbocharger applications. A new method for predicting gear pitting life, based on the Lundberg-Palmgren bearing fatigue life theory, was developed. It agrees well with pitting life data obtained with spur gears.

An improved formula for calculating elasto-hydrodynamic (EHD) film thickness in concentrated contact machine elements was developed. Proper calculation of EHD films is necessary in order to predict the operating regime of bearings and gears. The lack of a sufficiently thick EHD film can result in surface distress and reduced life.

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TABLE I. - TEST CONDITIONS FOR EXPERIMENTAL DATA

Fig- ure	Refer- ence	Nozzle type (area ratio)	Inner stream				Outer stream				Ambient					
			Veloc- ity, $V_1$ , m/sec	Tem- pera- ture, $T_1$ , K	Pres- sure ratio	Area, $A_1$ , cm <sup>2</sup>	Hydrau- lic di- ameter, $2H_1$ , cm	Veloc- ity, $V_2$ , m/sec	Tem- pera- ture, $T_2$ , K	Pres- sure ratio	Area, $A_2$ , cm <sup>2</sup>	Hydrau- lic di- ameter, $2H_2$ , cm	Spac- ing, $L_2$ , cm	Appar- ent distri- bution, $R_a$ , m	Stream veloc- ity, $V_a$ , m/sec	Tem- pera- ture, $T_a$ , K
5	2	Coaxial (0.75)	407	697	1.55	72.26	9.6	583	1.80	54.19	2.88	3.3	4.57	0	302	101
6	→	→	611	812	2.45	→	→	716	2.48	→	→	→	→	→	301	→
7	→	→	400	714	1.51	→	→	870	4.11	→	→	→	→	→	301	→
8	→	(1.20)	441	844	1.53	57.48	8.5	859	4.06	68.97	3.84	→	→	→	296	→
9	3	(.75)	296	389	1.52	14.32	4.3	575	2.51	10.77	1.26	1.7	3.05	9	298	→
10	3	(.75)	301	402	1.52	14.32	4.3	573	2.51	10.77	1.26	1.7	3.05	62	298	→
11	10	(.65)	367	553	1.57	109.1	11.8	548	2.42	70.66	3.26	0	12.19	0	284	100
12	10	Coannular (.65)	374	556	1.60	111.0	8.2	552	2.46	71.38	2.17	8.3	12.19	0	286	100

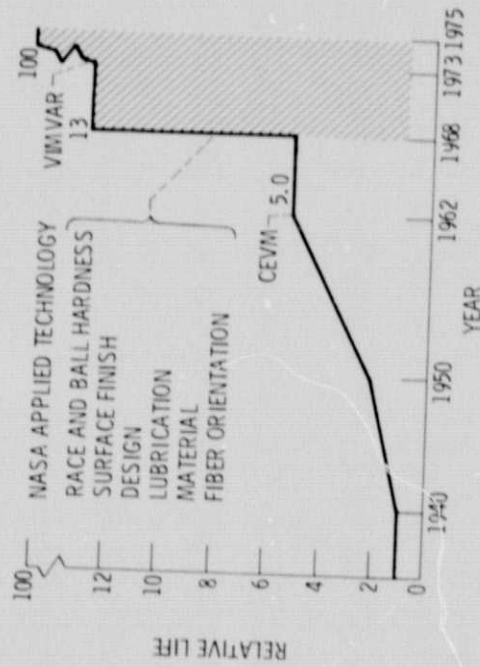
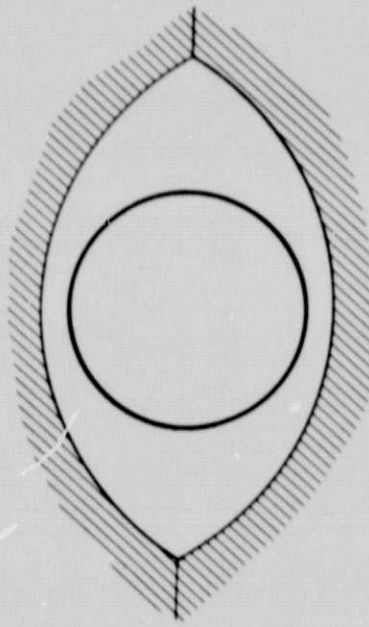
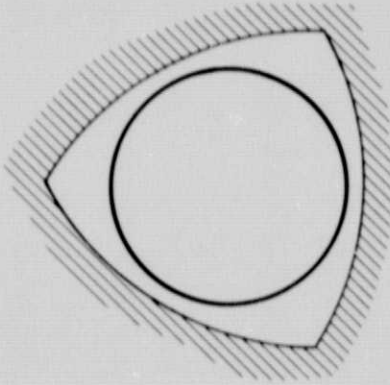


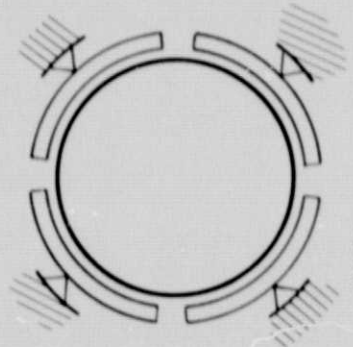
Figure 1. - Improvements in relative life of bearings in aircraft turbine engine service.



(a) ELLIPTICAL BEARING.



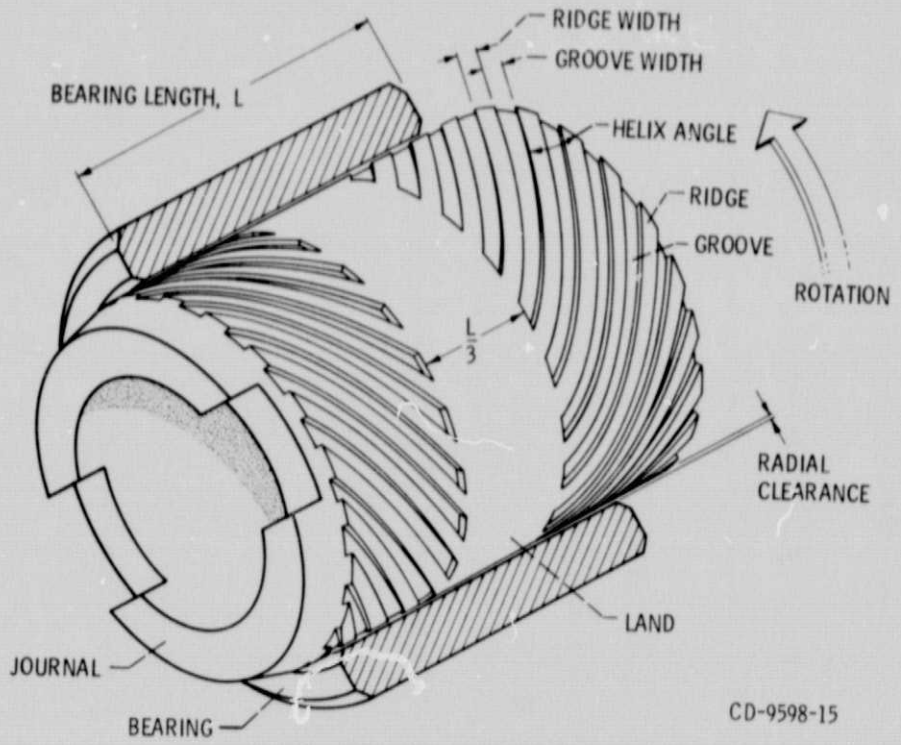
(b) THREE-LOBE BEARING.



(c) PIVOTED-PAD BEARING.

Figure 2. - Three types of whirl-resistant journal bearings.

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φ<sub>1</sub> BEARING ASSEMBLY.

Figure 3. - Spiral groove journal bearing geometry.

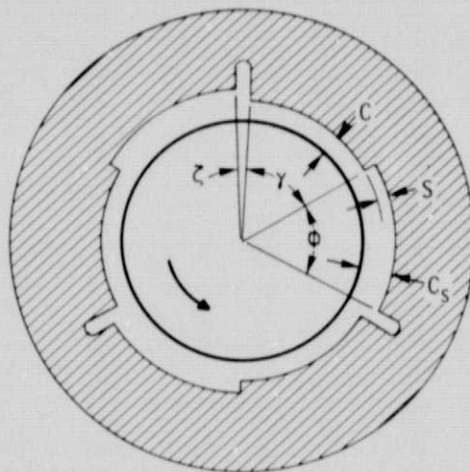


Figure 4. - Rayleigh-step bearing geometry.

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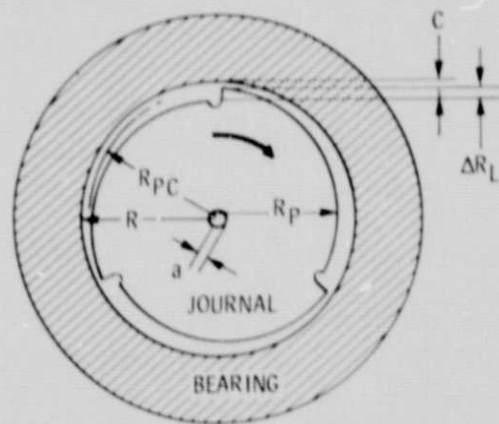


Figure 5. - Three-lobe journal geometry.

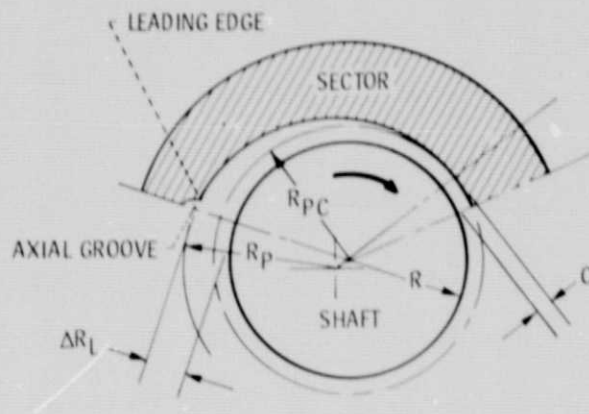


Figure 6. - Geometry of single lobe or sector of three-tilted-lobe bearing.



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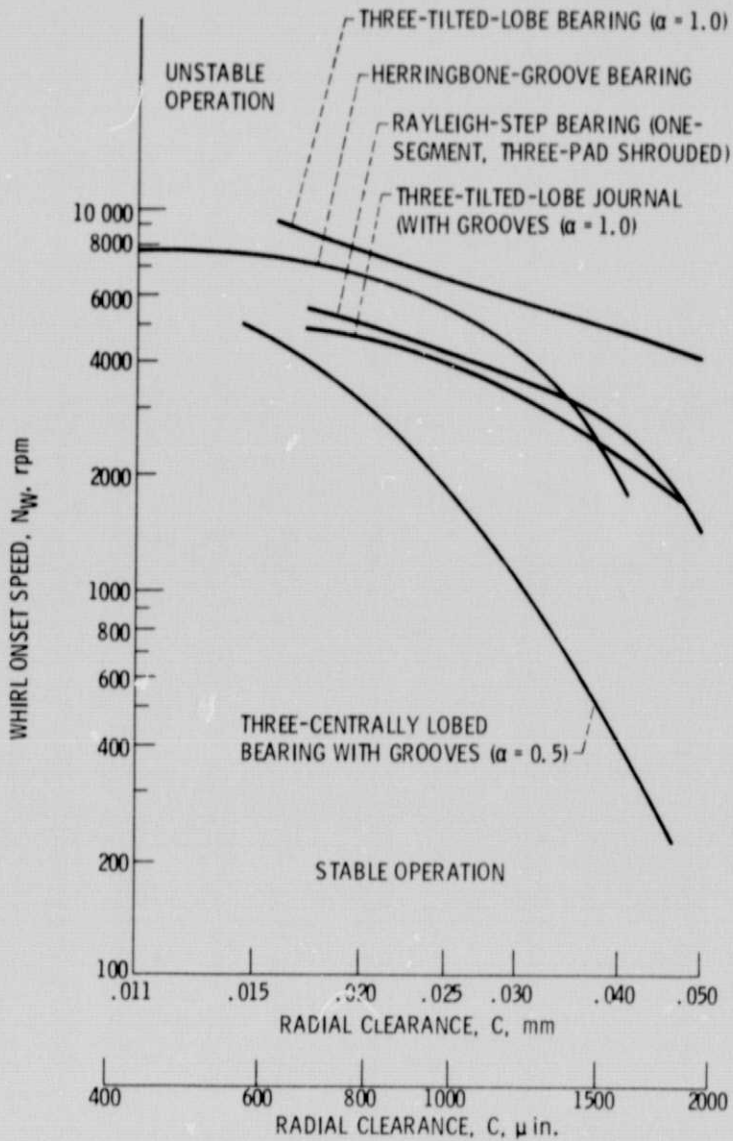


Figure 7. - Stability of five fixed geometry bearings. Zero radial load.

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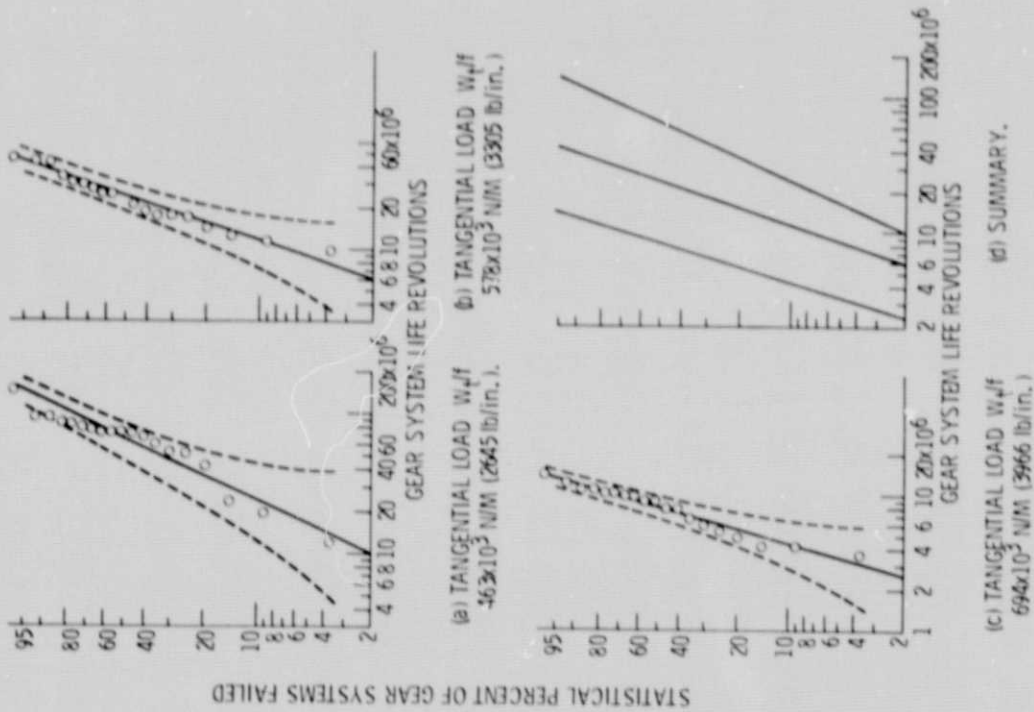


Figure 9. - Weibull plots of pitting lives of vacuum arc remelt (VAR) AISI 9310 steel spur gears.

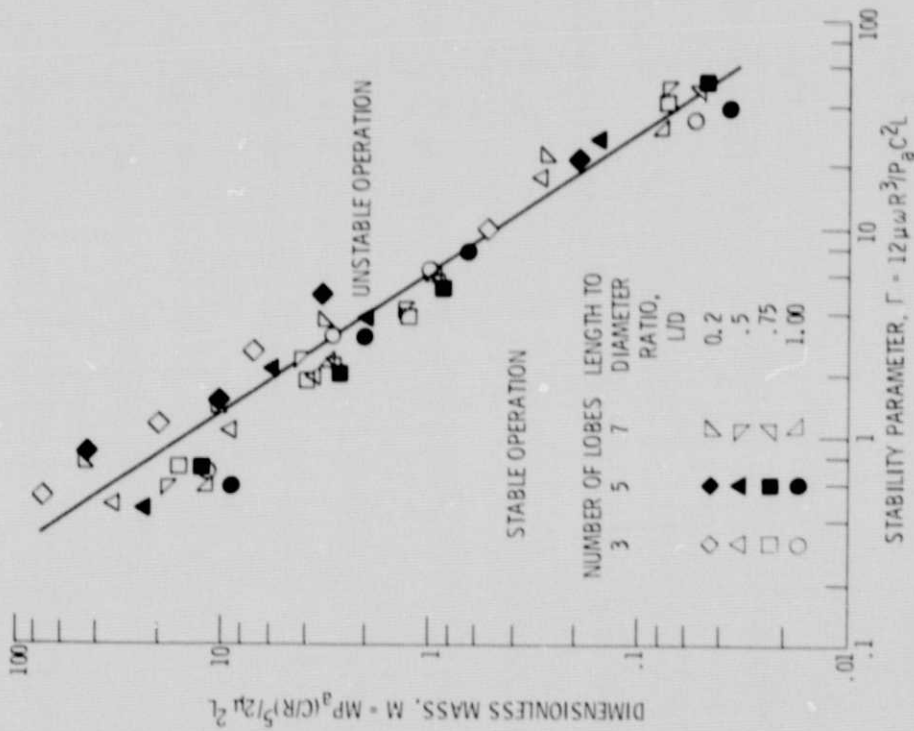


Figure 8. - Stability of tilted-lobe bearings with various numbers of lobes and L/D ratios. Zero radial load.



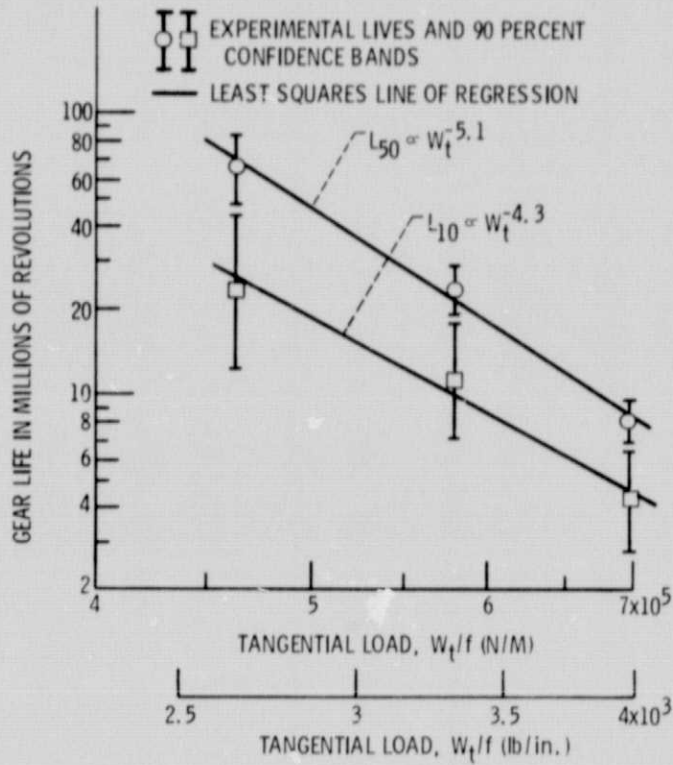
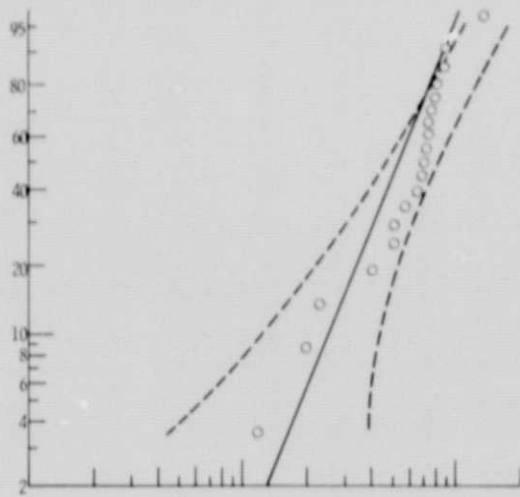
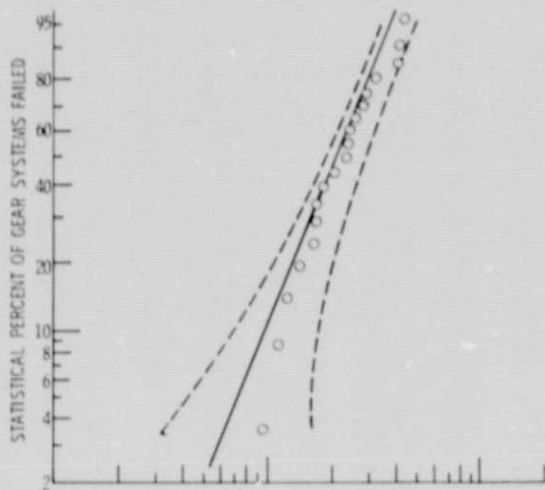


Figure 10. - Load-life relationship for (VAR) AISI 9310 steel spur gears.

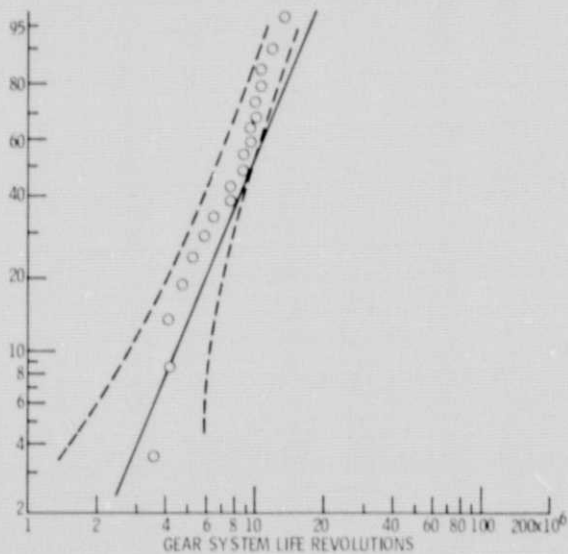
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(a) TANGENTIAL LOAD  $W_t/f 46 \times 10^3$  N/M (2645 lb/in.).



(b) TANGENTIAL LOAD  $W_t/f 578 \times 10^3$  N/M (3305 lb/in.).



(c) TANGENTIAL LOAD  $W_t/f 694 \times 10^3$  N/M (3966 lb/in.).

Figure 11. - Comparison of revised life prediction theory with experimental results for (VAR) AISI 9310 spur gears.

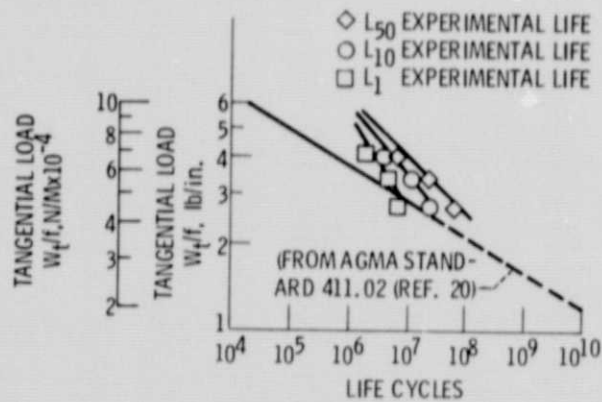


Figure 12. - Comparison of experimental life of (VAR) AISI 9310 steel spur gears with AGMA life prediction.

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