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FINAL REPORT

on

EVALUATION OF AXIAL LOCK-UP OF SPACE SHUTTLE MAIN ENGINE BEARINGS FROM HIGH PRESSURE OXYGEN TURBOPUMP

(Contract No. NAS8-33576 - Task No. 105)

to

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
GEORGE C. MARSHALL SPACE FLIGHT CENTER

by

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EVALUATION OF AXIAL LOCK-UP OF SPACE SHUTTLE MAIN ENGINE BEARINGS FROM HIGH PRESSURE OXYGEN TURBOPUMP

by

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INTRODUCTION

NASA is currently involved in the development of long-life turbopumps for use on the space shuttle. Because of the reusable design of the shuttle, lifetimes of 27,000 second (7.5 hours) are being sought. However, many turbopumps to date have only operated for periods on the order of a few hundred seconds. One persistent problem with the turbopumps has been the mainshaft support bearings. Often these bearings have incurred serious spalls during short-term testing, which makes their long-term reliability extremely questionable. Battelle has been involved with evaluations of these bearings to determine possible causes of failure. This report summarizes the result of the most recent such evaluation.

The purpose of the task reported herein was to

(1) Compare the sensitivity of 15-degree versus 25-degree contact angle bearings with regard to heat damage and thermal runaway.

(2) Evaluate the L/D aspects of the turbine-end bearing pair with regard to hang-up due to cocking.

(3) Compute a quantitative means of hang-up required to support a 44.5 KN (10,000 pounds) axial load without cartridge slipping.

(4) Quantify outer race growth as a result of a 44.5 KN (10,000 pounds) axial load to determine if this might contribute to a lock-up situation.

(5) Experimentally evaluate bearing torque as a function of axial load.

SUMMARY

(1) The 25-degree contact angle bearing does appear to be somewhat superior to the 15-degree angle bearing although the difference between the two is not particularly impressive.

(2) It does not appear that cocking of the bearing cartridge will represent a serious problem with regard to bearing loads.

(3) Hang-up of the cartridge due to bearing heating may be a very serious problem especially if poor lubrication of the bearing occurs. For example, if the coefficient of friction between balls and races is on the order of 0.2 (as a result of poor lubrication), the analysis predicts cartridge lock-up for all bearing axial loads.

(4) Race growth due to axial load can contribute to cartridge lock-up. This growth is on the order of 0.05 mm (.002 inch).

(5) Experimental evaluation of bearing performance under axial loads indicates that the bearing torque is a strong function of surface lubrication. For example, the inferred ball-race friction for a clean bearing is 0.22; whereas, for a bearing with a good PTFE film, this coefficient is on the order of 0.08. This difference in ball-race friction could well be the difference between a "loose" versus a "locked-up" cartridge.

EVALUATION OF EFFECT OF DESIGN CONTACT ANGLE

During the course of the shuttle improvement programs, the bearing contact angle was shifted from 25 degrees to 15 degrees. This change was apparently implemented to improve the radial stiffness of the bearing. However, such a change can also alter internal heating of the bearing and, in turn, the sensitivity of the bearing to thermal lock-up. Calculations have been made using the Battelle bearing dynamic program BASDAP[(1)*] to evaluate the effect of contact angle stresses and

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* Reference presented at the end of the text.

on peak ball temperature. The general inputs for the 7955 bearing used in the calculation are given in Table 1.

Figure 1 shows the effect of load on peak temperature rise for the two configurations. As can be observed, the temperature rise for the 15-degree-contact-angle bearing is somewhat higher than for the 25-degree-contact-angle bearing. These calculations were based on the assumption of reasonably good transfer film lubrication ($f \approx 0.1$) at the ball-race interface.

Higher values of f would result in attendant higher values of peak temperature. The effect of load on ball-race contact stress is shown in Figure 2 for the two contact angles. Again the 15-degree angle is shown to yield somewhat higher values.

A serious temperature rise of the inner race could cause growth of the inner race relative to the outer. This growth would result in increased bearing stresses, as shown in Figure 3. The stresses with the 15-dgreee-contact-angle bearings are somewhat more severe (25 percent) than for the 25-degree-contact-angle bearing.

In general, the effect of changing from a 25-degree contact angle to a 15-degree contact angle results in higher stresses, higher temperatures, and an increased propensity to thermal runaway. However, based on the calculations, the differences associated with the two contact angles do not appear to be particularly severe. On the other hand, it can be argued that any increase in stress is a serious factor and must be avoided, which suggests that a 25-degree-contact angle is definitely to be recommended.

BEARING LOCKING FROM MISALIGNMENT

Since the outer races of the bearings are relatively short cylinders compared with their diameters, the possibility exists for their locking against the cartridge during relative axial motion. The tendency to lock depends on geometric factors as well as the coefficient

TABLE 1. INPUT PARAMETERS FOR BEARING 7955

Parameter	Symbol	Numerical Value
Pitch Radius	R_p	40.5mm (1.595 inches)
Ball Radius	R_B	6.35mm (0.25 inch)
Number of Balls	NB	13
Inner Race Curvation	f_2	0.53
Outer Race Curvation	f_1	0.53
Design Contact Angle	β	15°, 25°
Thermal Conductivity	K	29.2 N/sec-C (3.65 lbs/sec-F)
Thermal Expansivity	α	12.6×10^{-6} cm/cm/C (7×10^{-6} in/in/F)
Operation Speed	Ω	30,000 rpm

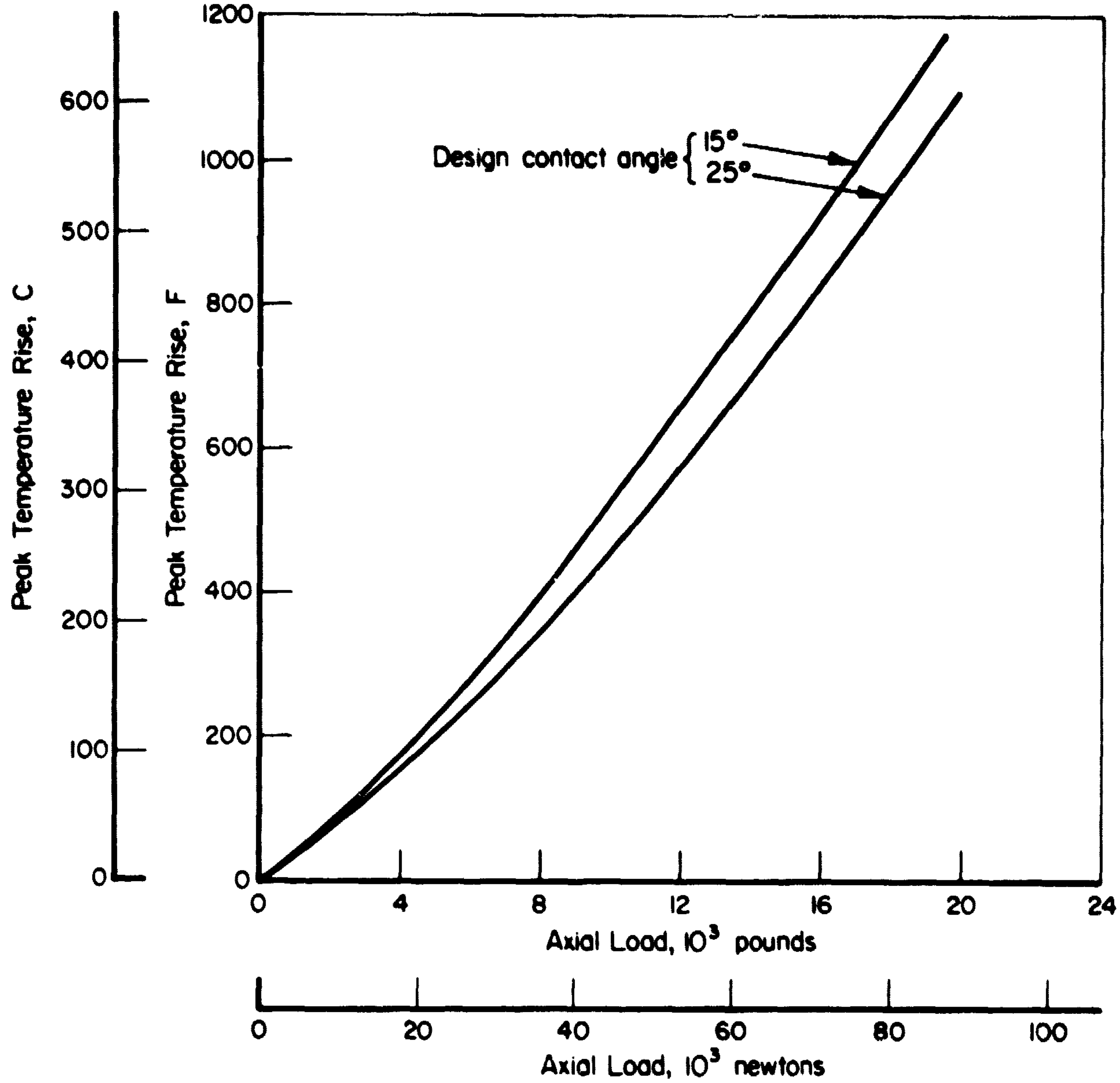


FIGURE 1. EFFECT OF LOAD ON PREDICTED PEAK TEMPERATURE RISE AT BALL-RACE INTERFACE

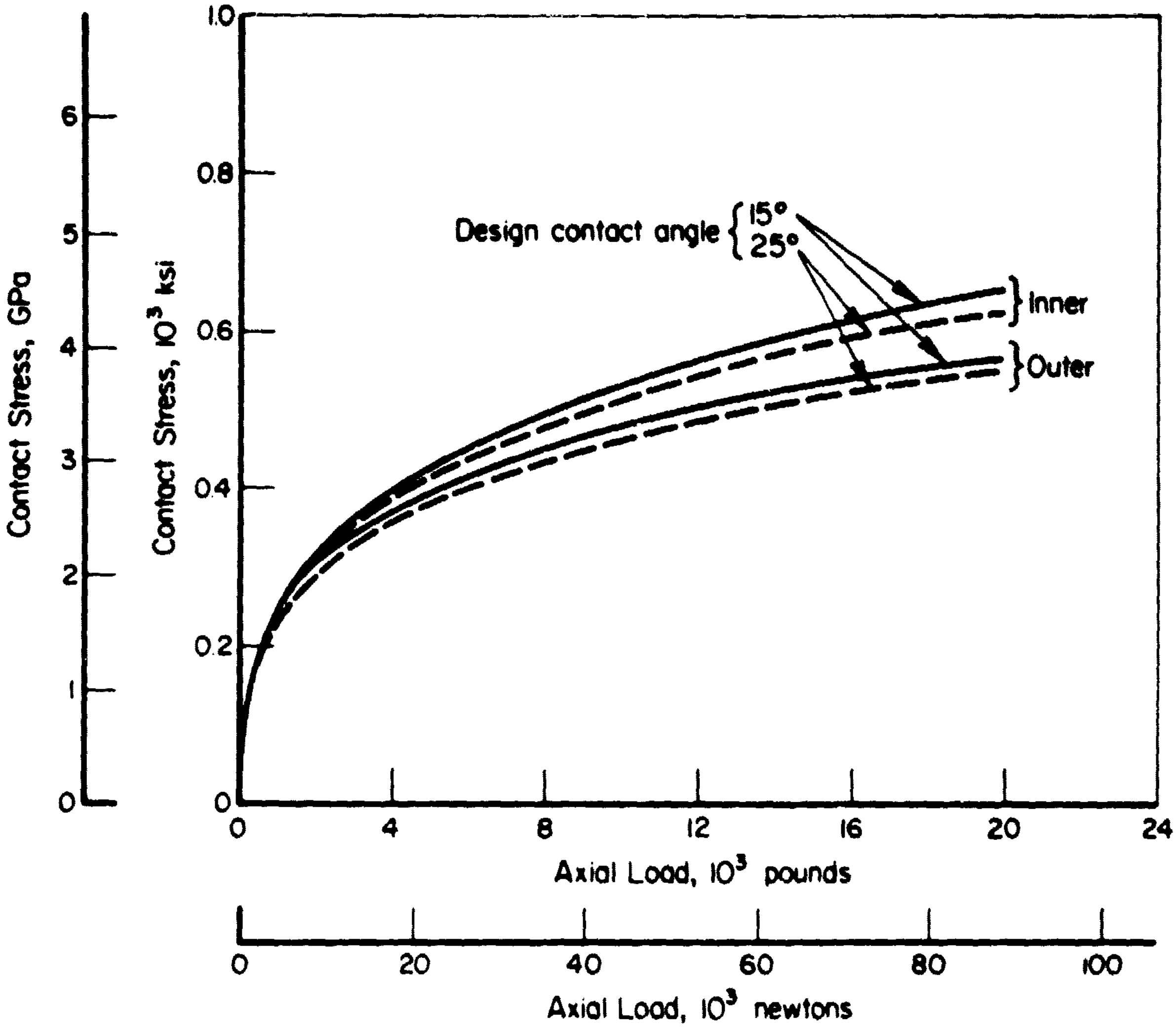


FIGURE 2. EFFECT OF LOAD ON BALL-ANGLE CONTACT STRESSES

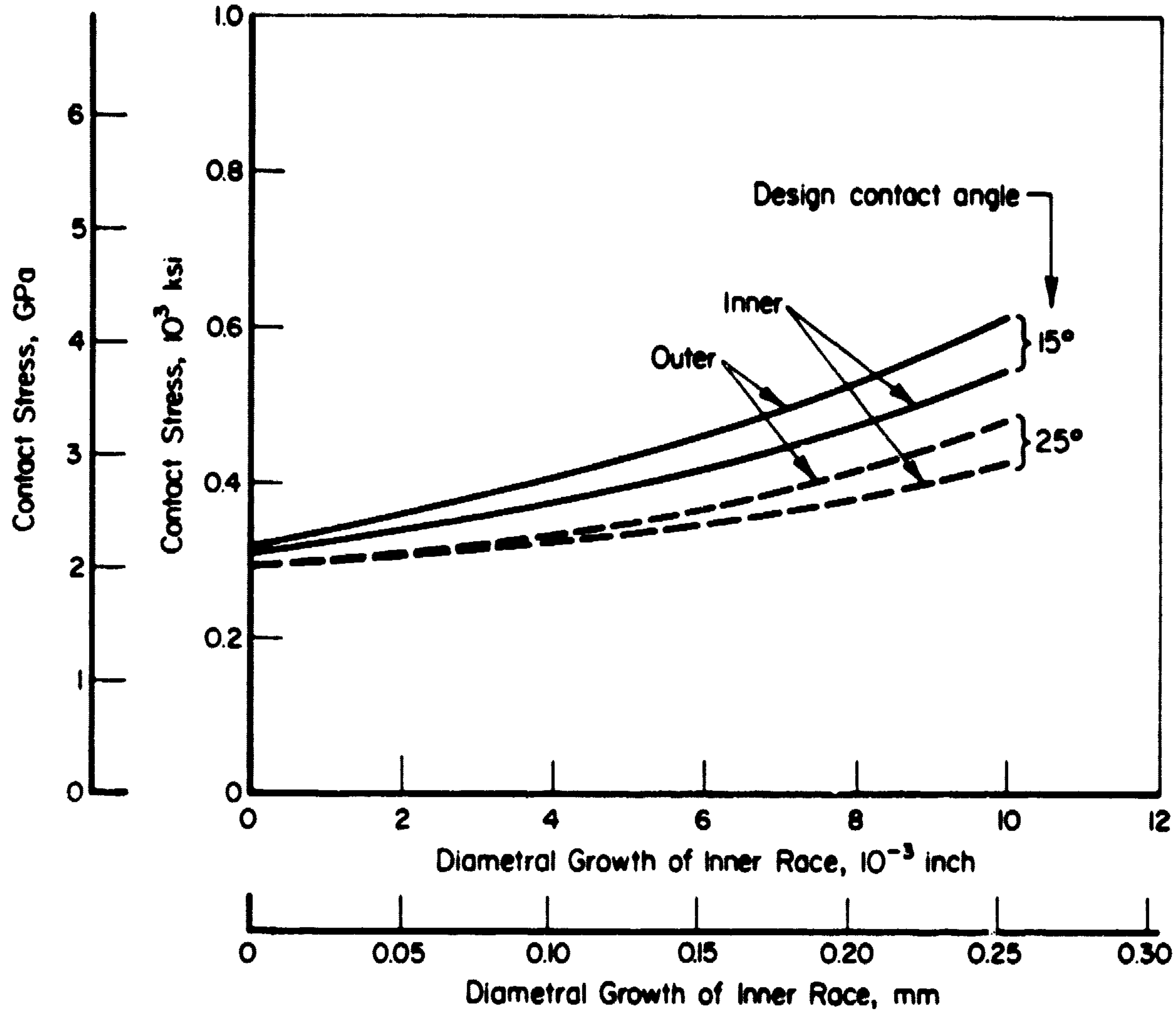


FIGURE 3. EFFECT OF INNER-RACE GROWTH ON BEARING CONTACT STRESSES

of friction between the outer race and cartridge. The important considerations are shown in Figure 4, which permit the following calculation of the threshold for locking.

At the threshold for locking, the friction forces equal the axial force, or,

$$F_1 = 2fF_2$$

when locked, a summation of moments about point "A" yields

$$F_1 e - F_2 m - F_2 (D+C) = 0$$

By geometry, $m = \sqrt{\ell^2 + D^2 - (D+C)^2}$.

Combining the equations eliminates the forces and results in

$$f = \frac{\sqrt{\ell^2 + D^2 - (D+C)^2}}{2e - D - C} .$$

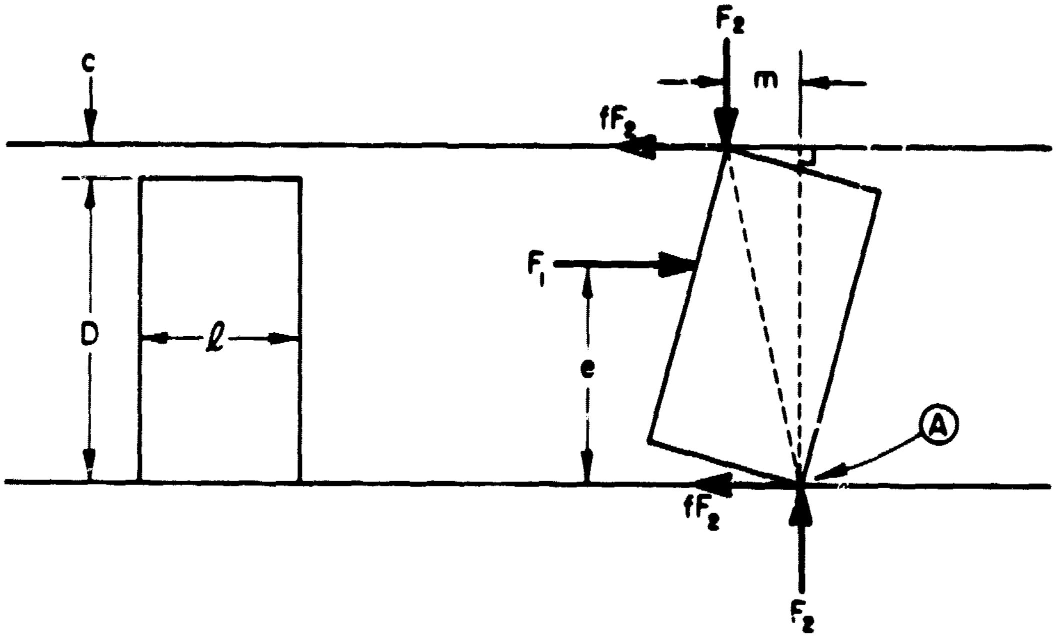
The worst case occurs when the axial force is applied at the edge of the bearing, or when $e = D$,

or $$f = \frac{\sqrt{\ell^2 + D^2 - (D+C)^2}}{D-C} .$$

A plot of this equation as a function of bearing length with the dimensions pertinent to the turbine end bearing is presented in Figure 5. Since the clearance will always be small relative to the diameter in this application, the equation becomes

$$\mu \doteq \frac{\ell}{D} .$$

With the current bearing length of 0.766 inch (19.5mm), a coefficient of friction greater than 0.2 would be required to cause axial locking. Since a coefficient of friction below this value can be obtained with the MoS_2 lubricant being used on the bearing outer diameter, locking by this mechanism is probably not a problem



where D = Bearing diameter (4.06 inch [103 mm])

ℓ = Bearing length (0.766 inch [19.5 mm])

c = Diametral clearance (0.0025 inch [0.064 mm])

e = Distance from edge of bore to force

f = Coefficient of friction

F_1 = Axial force

F_2 = Reacted force

FIGURE 4. GEOMETRY CONSIDERATIONS FOR CALCULATING AXIAL BEARING LOCKING

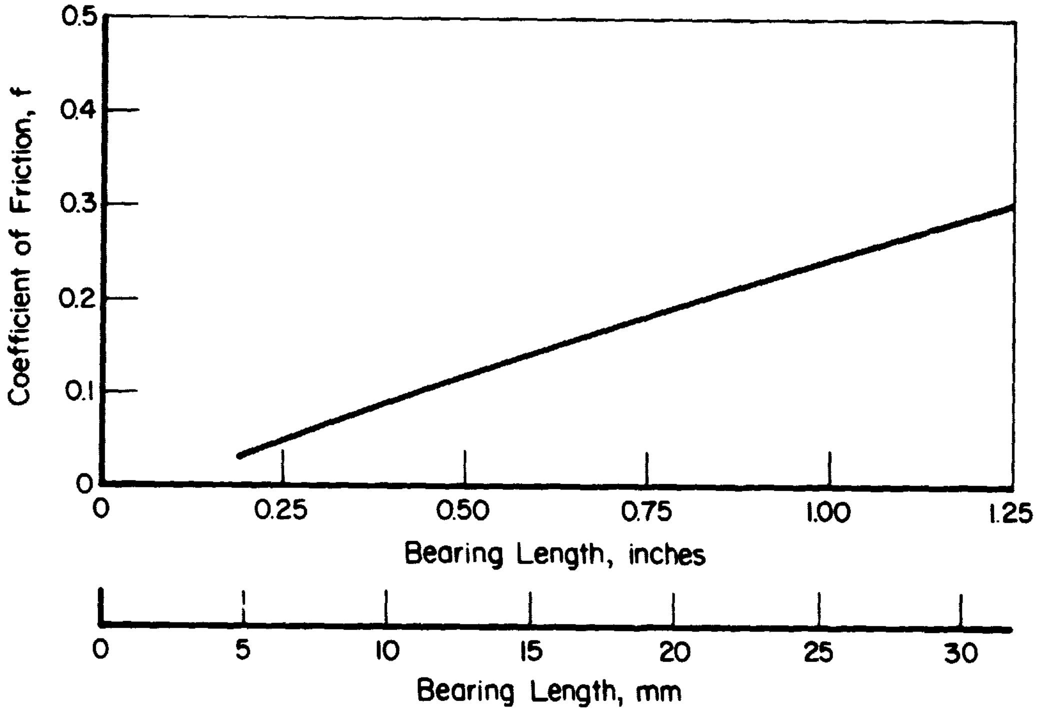


FIGURE 5. FRICTION REQUIRED TO LOCK BEARING AS A FUNCTION OF BEARING LENGTH

However, this result emphasizes the need for maintaining a competent solid lubricant film at the bearing-cartridge interface.

EVALUATION OF CARTRIDGE HANG-UP DUE TO BEARING HEATING

Evaluation of Thermal Growth of Bearing System

The purpose of this analysis was to evaluate the possibility of internal heating in the oxygen-pump bearing causing "lock-up" of the bearing cartridge. The approach taken involved estimating the heating due to ball spinning on the bearing outer race and computing the effect of this heat on bearing outer race and cartridge growth (see Figure 6).

Heating at the Ball-Race Interface

The heat generated, Qo, at the outer race contact can be estimated[1] to be

$$Qo = \frac{.93\ f\ pc\ \omega_s\ ba^2}{A_R} \tag{1}$$

where

f is the coefficient of friction between ball and race,

ω_s is the ball spin relative to the outer race,

a and b are the major and minor axes of the ball-race contact ellipse,

A_R is the area over which the heat is assumed to be dissipated.

Since we are interested in the thermal growth of a large portion of the bearing, we must allow the heat to be spread out over a region that is significantly larger than the ball-race contact zone.

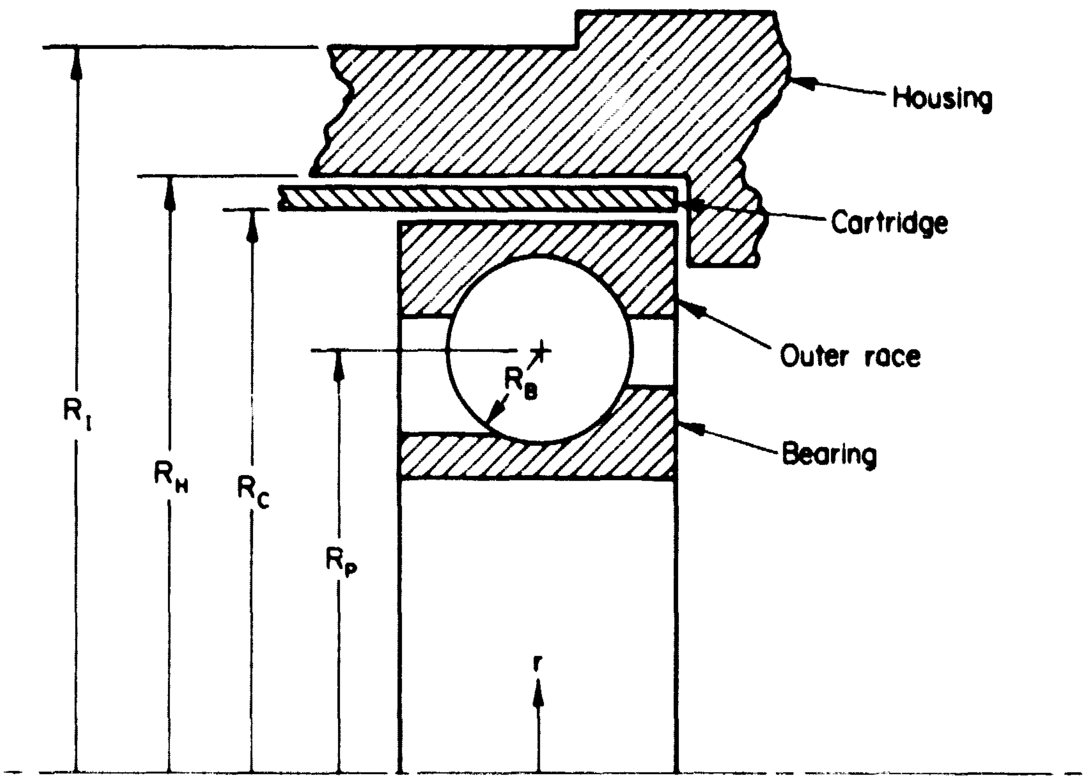


FIGURE 6. NOMENCLATURE FOR BEARING THERMAL GROWTH ANALYSIS

A somewhat conservative assumption is that

$$A_R = \left(\frac{2\pi R_P}{N_B}\right)\left(2\ R_B\right) \tag{2}$$

where R_P is the ball pitch radius

N_B is the number of balls

R_B is the ball radius.

Combining Equations (1) and (2)

$$Q = \frac{.074\ f\ P_o\ \omega_S\ b\ a^2\ N_B}{R_B\ R_P} \quad . \tag{3}$$

<u>Temperature at Ball-race Contact</u>

For a steady-state heating situation, it can be shown that

$$K = \frac{\Delta T}{\Delta y} = nQ \quad , \tag{4}$$

where $\frac{\Delta T}{\Delta y}$ is the temperature gradient out of the outer race,

K is the thermal conductivity,

n is the fraction of heat entering the outer race,

1-n is the fraction of heat entering the ball.

The maximum temperature (above ambient) at the ball-race contact can be written

$$T_{mx} = \frac{nQ}{K}\left(L - r_o\right) \tag{5}$$

where $(L - r_o)$ is the path length.

From Reference (1), it was shown that the ball temperature could be estimated by

$$T_{mx} = (1\text{-}n)\ 0.96\ \frac{Q}{K}\ a \ , \tag{6}$$

where (1-n) was assumed to be 0.5.

Equating (5) and (6) shows that

$$0.96\ (1-n)a = n(L-r_o)\ , \tag{7}$$

or,

$$n = \frac{0.96\ a}{(L-r_o) + 0.96\ a}\ . \tag{8}$$

Finally, the temperature distribution can be written

$$T = T_{mx} \frac{(L-r)}{(L-r_o)}\ , \tag{9}$$

where r_o is the radial position; where $T = T_{mx}$ ($r_o \doteq R_p + R_B$).

Thermal Growth Analysis

Timoshenko and Goodier(2) show that radial growth, u, due to temperature, can be written

$$u = \frac{1+\nu}{1-\nu}\,\alpha\,\frac{1}{r}\int_{r}^{r} Trdr + C_1\,r + \frac{C_2}{r} \tag{10}$$

where δ is the thermal coefficient of expansion

ν is poissons ratio,

C_1 and C_2 are constants.

Likewise, the radial stress, σ_r, can be written

$$\sigma_r = \frac{-\alpha E}{1-\nu}\,\frac{1}{r^2}\int_{r}^{r} Trdr + \frac{E}{1+\nu}\;\frac{C_1}{1-2\nu} - \frac{C_2}{r^2}\ , \tag{11}$$

where E is Young's modulus.

If we assume that $\sigma_r = 0$ at $r = r_o$ and $r = L$, then C_1 and C_2 can be determined as

$$C_1 = \frac{0.779\alpha}{L^2-r_o^2}\int_{r_o}^{L} Trdr\ , \tag{12}$$

and

$$C_2 = 2.31\, r_o^2\, C_1 \quad , \qquad (13)$$

where it has been assumed that $\nu = 0.284$. The integrals in Equations (11) and (12) can readily be determined using the temperature distribution of Equation (9).

Specific Examples

The primary bearing-related factors affecting thermal lock-up of the cartridge are the ball-race contact dimensions, the ball-spin on the outer race and the lubrication conditions controlling ball-race friction [see Equation (3)]. The ball-race contact dimensions are readily computed from the BASDAP computer model. However, the friction and ball-spin are strongly dependent on the lubrication parameters and are much more difficult to quantify. The lubrication model used for solid films with BASDAP computation was described previously[1] and is based on limited experimental data.

Ball-spin in the BASDAP model is computed based on a spin torque balance between inner- and outer-race ball contacts. This torque balance results in spinning on both races as opposed to just one race as assumed in the classic A.B. Jones[3] bearing dynamics model. Jones used a model considering only the friction coefficient for ball-race traction and, to circumvent the need for a torque balance, assumed that one race (the outer race for the shuttle bearing) controlled spin. That is, the ball does not spin on the "control" race. This race-control assumption would inherently preclude any outer race heating since ball-spin is an essential aspect of energy input from frictional heating. However, the BASDAP model does predict that significant spin will occur on both races, albeit inner-race spin is somewhat greater than outer-race spin.

Figure 7 presents thermal growth for a 7955 bearing loaded to 8890 N (2000 pounds) operating at 3000 rpm. Curve (a) is based on an

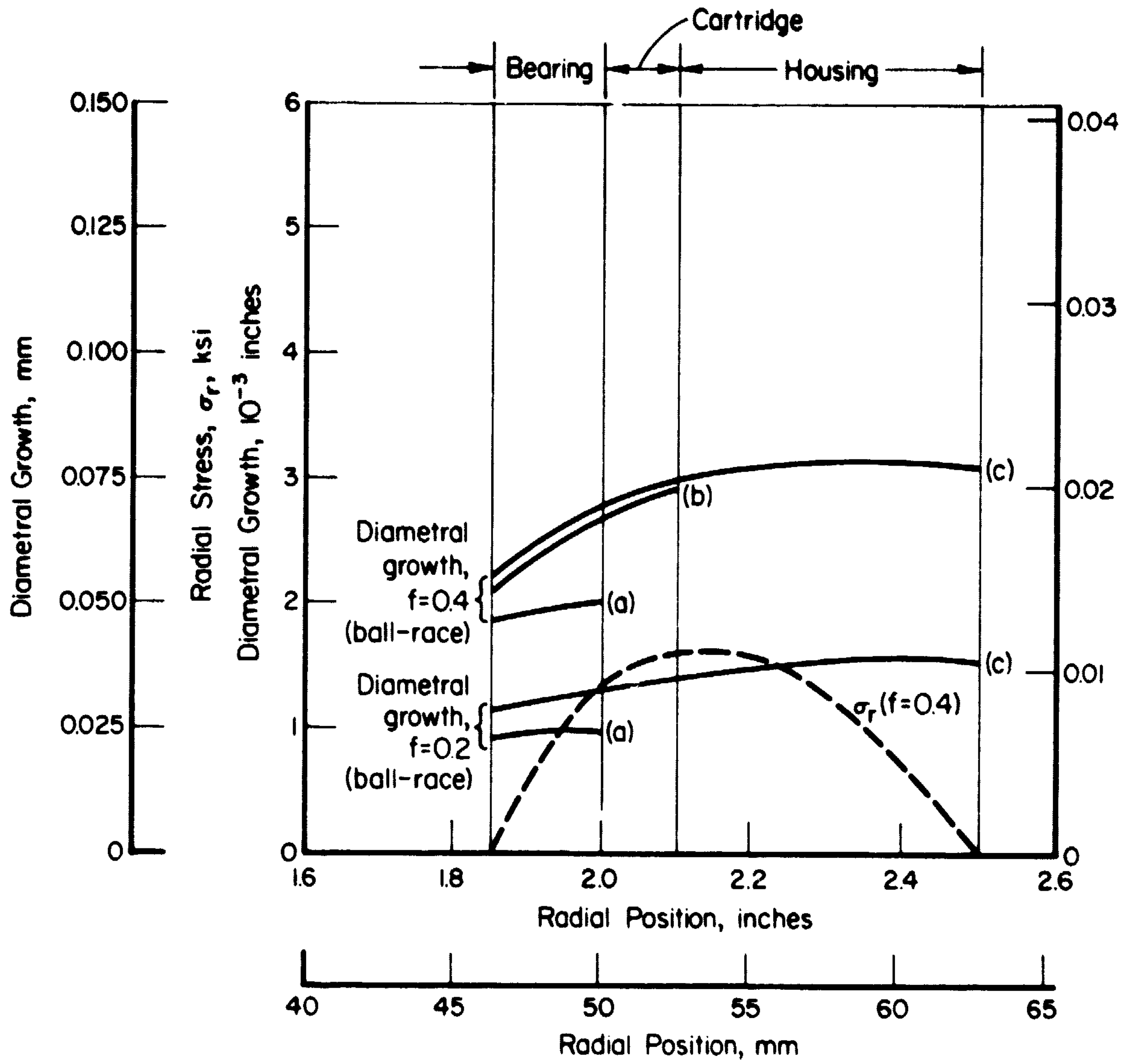


FIGURE 7. THERMAL GROWTH OF 7955 BEARING OVER RACE AND CARTRIDGE WITH AN AXIAL LOAD OF 8890 N (2000 POUNDS)

assumption that the outside of the outer race is at ambient temperature, curve (b) is based on the assumption that the outside of the cartridge is at ambient temperature, and curve (c) assumes that the outside of the housing is at ambient temperature.

Note that for a coefficient of friction of 0.2, the outer race of the bearing can grow about 0.033 mm (0.0013 inch) which is on the order of the bearing-cartridge clearance. If the friction coefficient were as high as 0.4, the growth would be sufficient to eliminate the clearance. At a load of 53,400 N (12,000 pounds), as shown in Figure (8), the bearing and cartridge clearance would be eliminated even at a very low friction coefficient.

Also shown in Figures 7 and 8 are the radial stresses associated with the temperature distribution. These stresses assumed a continuous surface for the race-cartridge-housing system and did not consider the clearance between them. The stress computations were based on the curve (c) heat path condition only.

The effect of axial load on stress at the cartridge-housing interface is shown in Figure 9, and Figure 10 presents the mean temperatures associated with these loads. Also shown in Figure 9 is the estimated stress required to lock the cartridge, assuming a Vitrolube coating at the cartridge-housing interface. The general observation is that if good ball-race lubrication occurs such that $f \leq 0.1$, then thermal lock-up of the cartridge is unlikely. If, however, poor lubrication occurs ($f > 0.2$), then such lock-up looms as failure-inducing mechanism.

EXPERIMENTAL MEASUREMENTS

As a means for confirming the calculations made of contact angle and deflections and for measuring bearing torque, a series of experimental measurements was conducted using two turbine end bearings. The bearings were S/N 8517932 and S/N 8517834. Both had reportedly been used in an engine but were in excellent condition as determined by visual inspection.

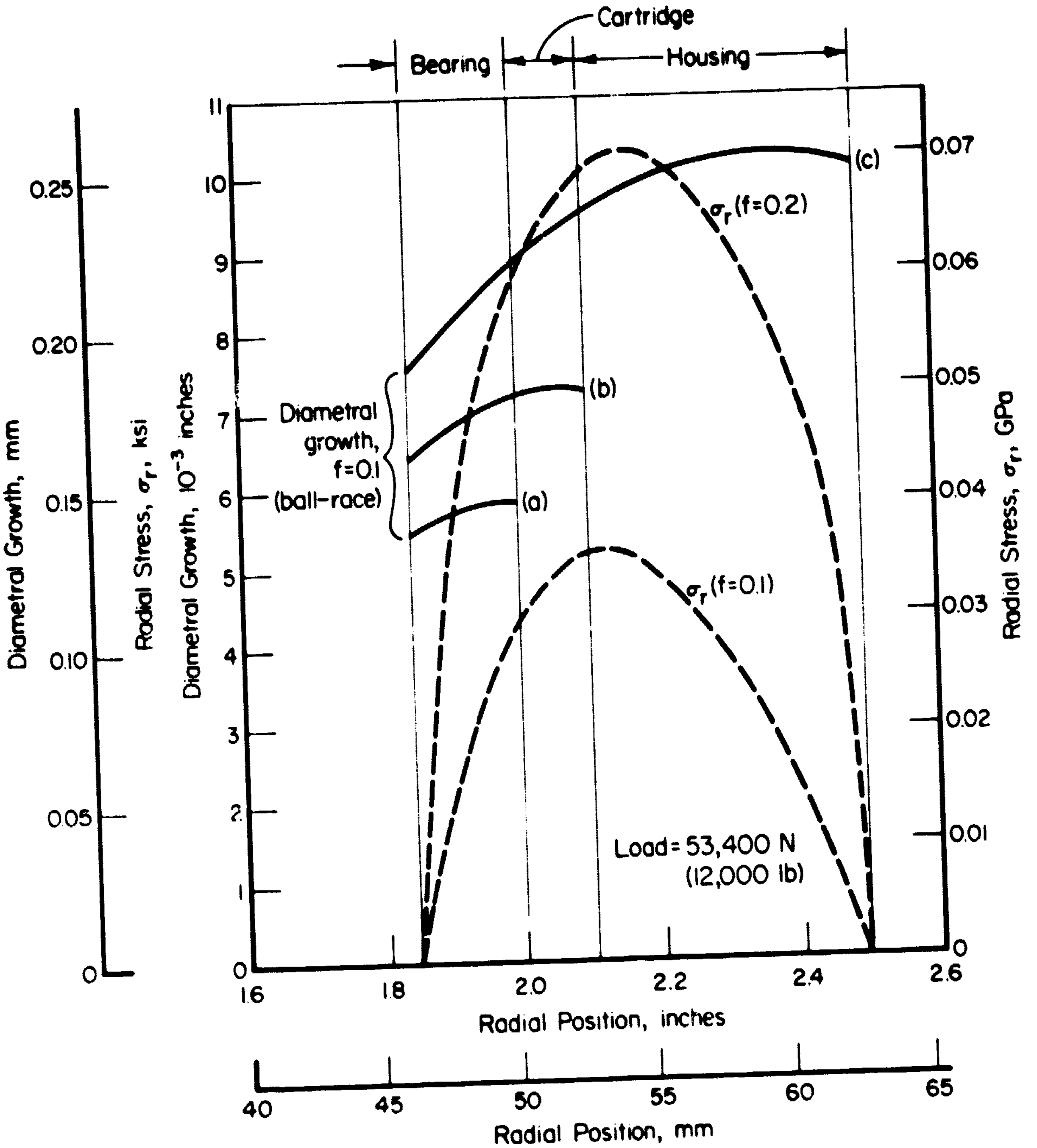


FIGURE 8. THERMAL GROWTH OF OUTER RACE AND CARTRIDGE WITH AN AXIAL LOAD OF 53,400 N (12,000 POUNDS)

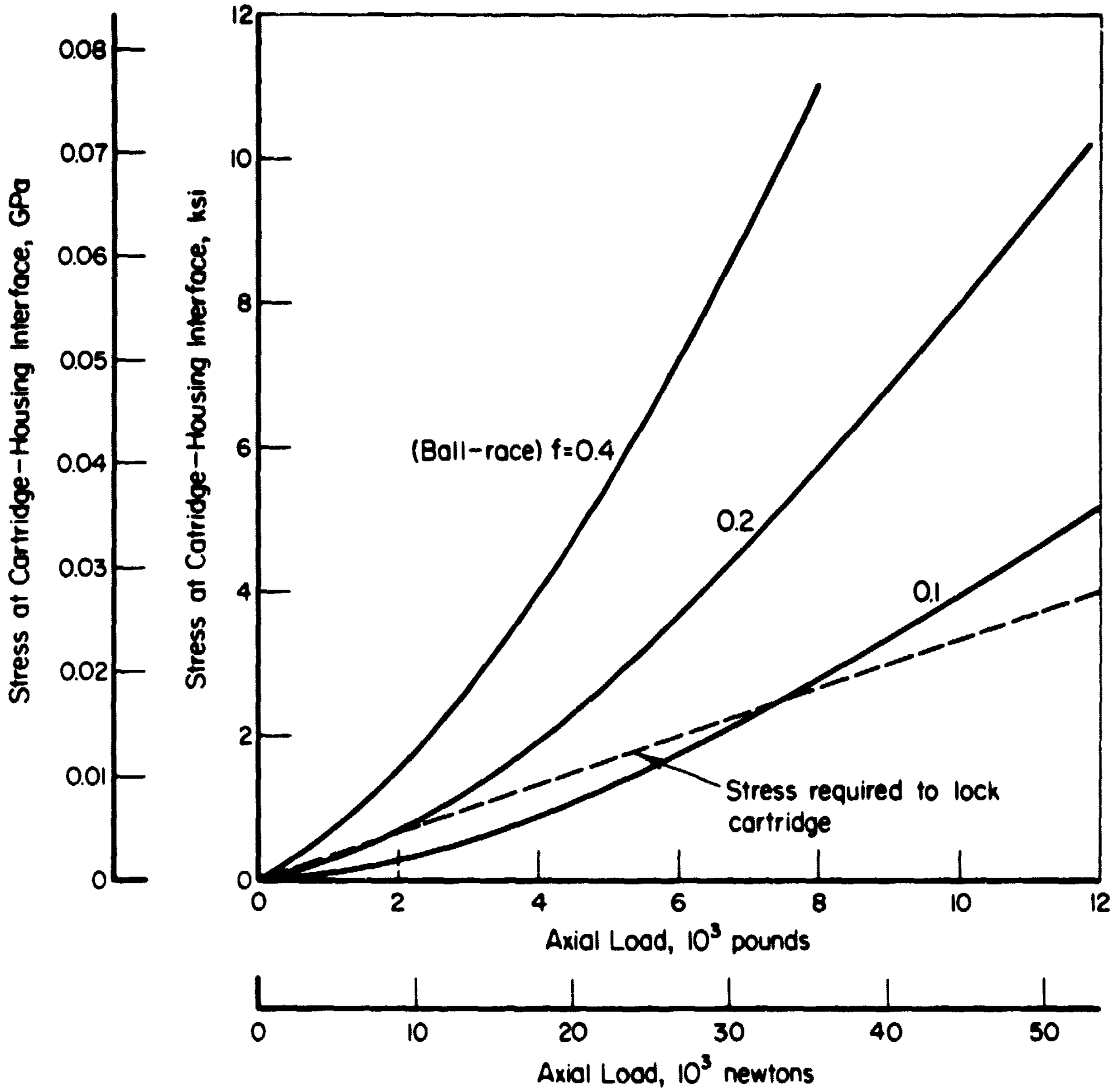


FIGURE 9. EFFECT OF AXIAL LOAD ON CARTRIDGE HOUSING STRESSES

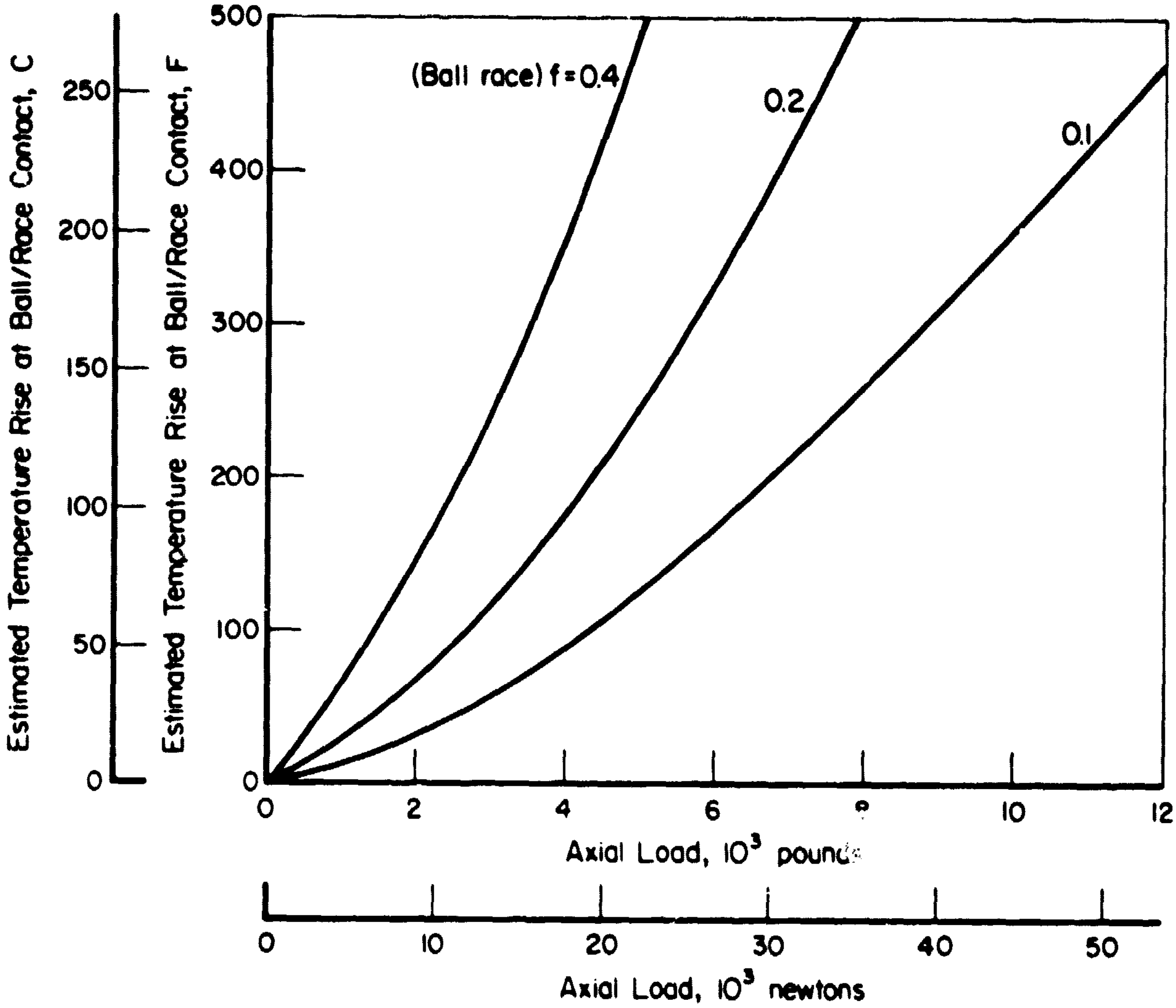


FIGURE 10. ESTIMATED MEAN TEMPERATURE RISE AT BALL-OUTER-RACE CONTACT

Procedure

Pedestals were constructed with a shoulder for mounting the inner races with the same interference fit used on the actual turbopump. A simple short cylinder was used to couple the outer races, as shown in Figure 11. With this arrangement, axial loads were applied in a tensile-compression testing machine, as shown in Figure 12.

The loading of the two bearings against each other permitted a measurement of the total torque at the axial loads of interest. Axial deflections were measured with a dial indicator and diametral deflections were measured with a micrometer. The contact angle was determined from the following equation by counting revolutions of the retainer relative to the outer race.

$$\cos \beta = \frac{E}{d}\left(1 - \frac{2N_r}{N_o}\right)$$

where β = contact angle

E = pitch diameter [81.0mm (3.19 inches)]

d = ball diameter [12.7mm (0.5 inch)]

N_r = revolutions of the retainer

N_o = revolutions of the outer race.

In all cases, 50 revolutions of the outer race were made for the determinations.

Results

Measurements made of the deflections and contact angles as a function of load are presented in Figure 13. The measured contact angles compared very well with those predicted by the BASDAP calculations. The prediction for diametral growth was made on the basis of the balls applying a pressure to the inside of a thick cylinder. It did not consider

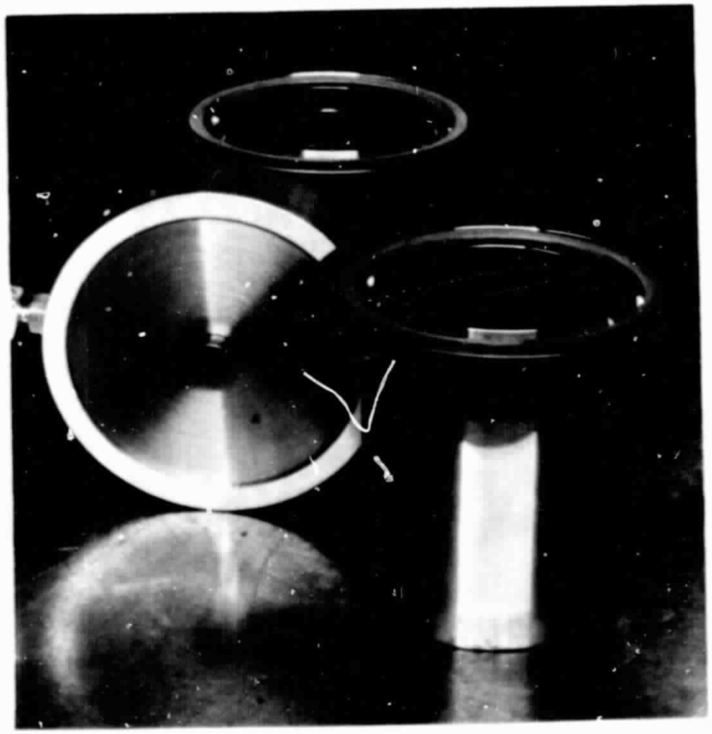


FIGURE 11. BEARING MOUNTING PEDESTALS AND OUTER RACE COUPLER

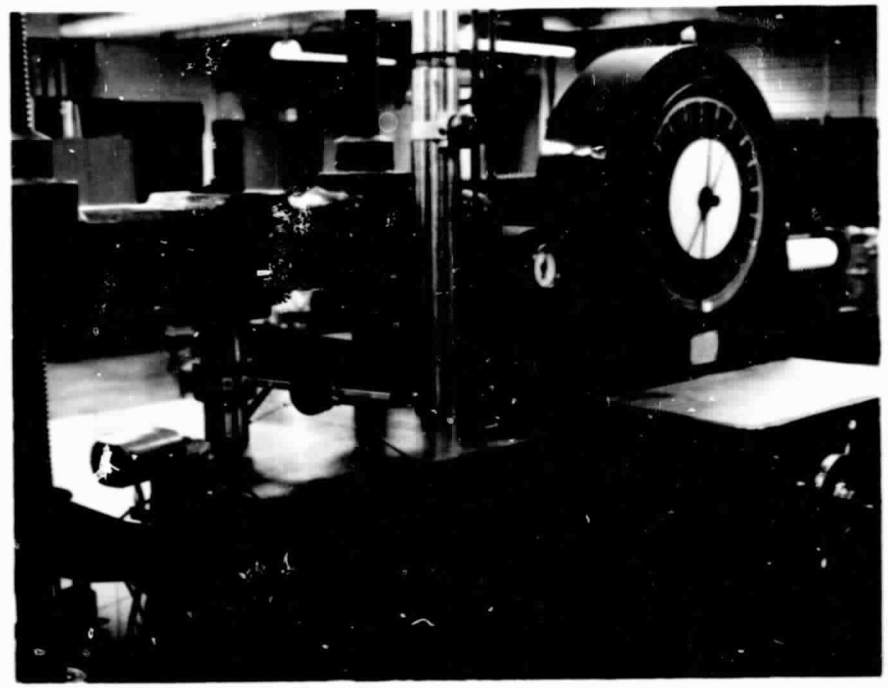


FIGURE 12. BEARING TEST SETUP IN TESTING MACHINE FOR APPLYING AXIAL LOADS

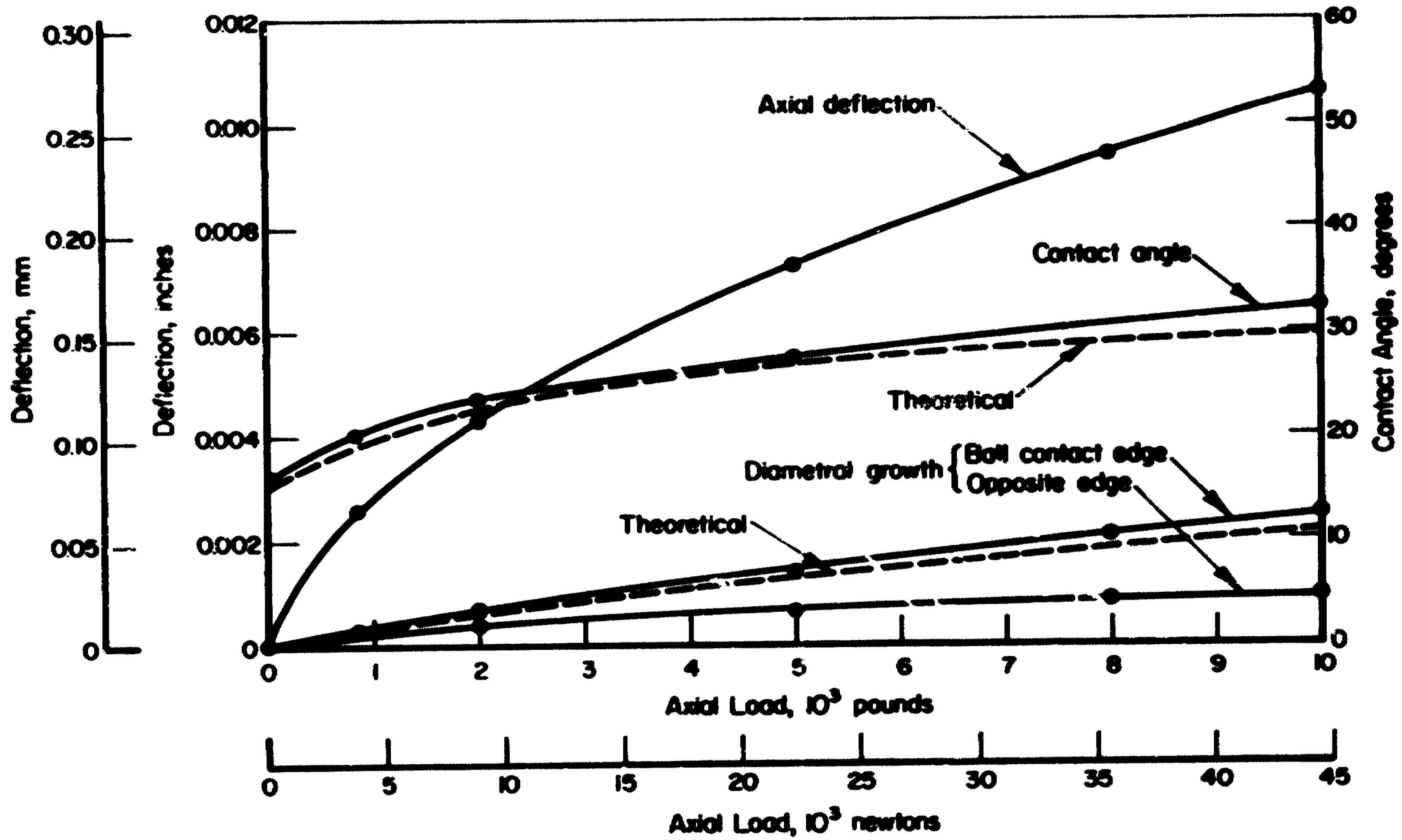


FIGURE 13. BEARING DEFLECTIONS AND CONTACT ANGLE AS A FUNCTION OF AXIAL LOAD

that the balls contacted the outer race closer to one edge. Because of this, the actual deflection was greater than that predicted at the edge of contact and less on the opposite edge. The measurements showed that the outer race becomes a section of cone as the result of the loading. Also, at 44.5 KN (10,000 pounds), the outer race deflection of 0.064 mm (0.0025 inch) consumes the minimum clearance between the cartridge and outer race.

The measurements of bearing torque as function of axial load are presented in Figure 14. The torque readings are presented for a single bearing and were obtained by dividing the measured readings in half, which assumes that the two bearings had equal torque. The torque curve for the as-received bearings presumably represents the expected torque after an equilibrium of PTFE has been transferred to the balls and races from the retainer. When PTFE was burnished onto the races by rubbing the ball contact area with a PTFE block, the torque was cut approximately in half. Rotating the bearing for 50 revolutions at 44.5 KN (10,000 pounds) resulted in a slight further reduction in torque. After removing the PTFE by 2 hours of burnishing in a Sweco vibratory polishing unit, higher torque readings were recorded. Since the torque increased with increasing revolutions, a total of 250 revolutions was made at 44.5 KN (10,000 pounds) prior to obtaining the curve presented in Figure 14. Although the rate of torque increase had decreased after 250 revolutions, it still had not stabilized. This indicates that very high torques and associated frictional power losses can be expected if poor ball-race lubrication exists and further that transfer of PTFE from the retainer does not occur rapidly under conditions of thrust loading. Ball speed variations are minimal with thrust loading, which minimizes the ball-retainer forces probably required for the transfer to occur. This suggests that the bearing must operate on residual PTFE films during periods of high thrust loading and that the longevity of these films will be important in determining the resulting race wear and damage.

Some analyses were conducted to relate the measured torque to a ball-race coefficient of friction. This analysis indicated that for a burnished film $f \simeq 0.08$ for the as-received bearing $f \simeq 0.12$, and for

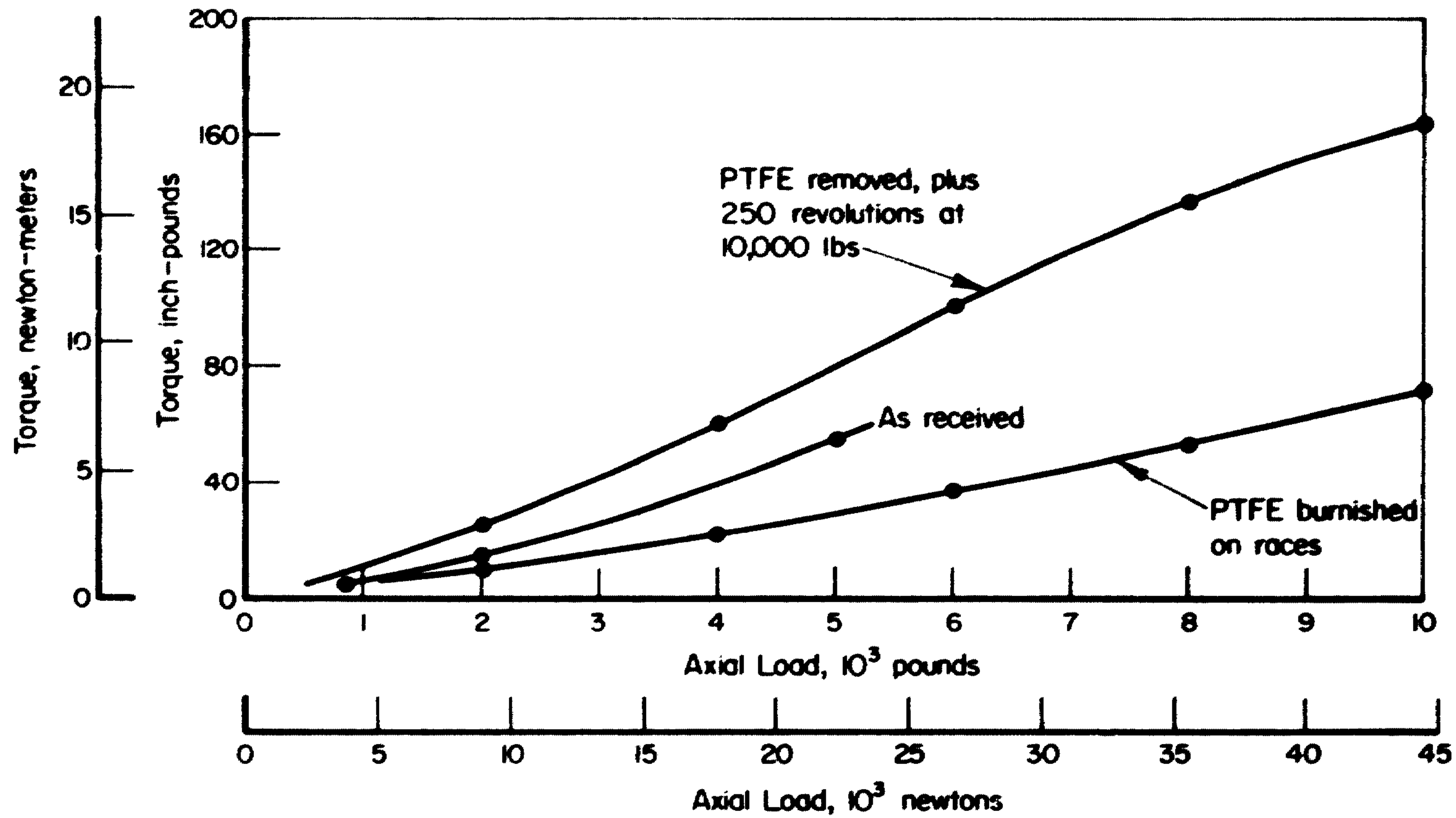


FIGURE 14. BEARING TORQUE AS A FUNCTION OF AXIAL LOAD

the clean bearing $f \approx 0.22$. These analyses further underscore the need for achieving good ball-race lubrication. For example, (as discussed earlier), if $f > 0.2$, thermal lock-up could be a serious problem.

References

(1) Kannel, J. W. and Merriman, T. "SSME Turbopump Bearing Analytical Study." August 20, 1980.

(2) Timoshenko, S. and Goodier, J. R. Theory of Elasticity. McGraw-Hill Book Company, New York, 1951, p. 409.

(3) Jones, A. B., "A General Theory for Elastically Constrained Ball and Roller Bearings Under Auxiliary Load and Speed Conditions. Trans. ASME, J. Basic Eng., Vol. 82, Ser. D., No. 2, June 1960, pp. 309-320.