DEVELOPMENT OF NEW MATERIALS FOR TURBOPUMP BEARINGS

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Abstract

The life requirement for the angular contact ball bearings in the Space Shuttle Main Engine (SSME) high pressure oxygen turbopump (HPOTP) is 7.5 hours. In actual operation, significantly shorter service life has been experienced. The objective of this current program is to identify bearing materials and/or materials processing techniques offering significant potential for extending HPOTP bearing performance life.

Interactive thermomechanical analysis of the HPOTP bearing-shaft system was performed with the SHABERTH computer program. Bearing fatigue life, ball-race contact stress, heat generation rate, bulk ring temperatures and circumferential stress in the inner rings were quantified as functions of radial load, thrust load and ball-race contact friction. Criteria established from the output of this analysis are being used for material candidate selection.

Introduction

The space shuttle main engine cryogenic turbopumps, and in particular the high pressure oxygen turbopump, (HPOTP), have been experiencing premature bearing degradation. Bhat and Dolan[1] have reported heavy spalling and prominent wear of the rings and rolling elements of the HPOTP bearings after only

2406 seconds of total running time with only 1090 of those seconds at full power level. Other failure analyses[2,3] have revealed heavy wear, smearing, microcracking and pitting indicative of surface distress associated with inadequate lubrication. The bearings in these analyses experienced total operating times of less than 6000 seconds. In addition to the above mentioned failure observations, surface oxide films indicative of high operating temperatures have been observed on rolling element and race surfaces of the bearings after 100 to 4000 seconds total operating time. These latter observations are consistent with the failure scenario proposed by Bhat and Dolan[1], which attributes the abbreviated HPOTP bearing life to a thermal runaway mechanism.

In an investigative study[4] performed on these bearings, analytical work showed that with full film lubrication the predicted bearing life is on the order of 100 hours (360,000 seconds). The life requirement is 7.5 hours or 27,000 seconds. The current HPOTP bearing lubrication scheme consists of MoS_2 films on the balls and raceways, and a woven glass reinforced Teflon (Armolon) cage intended to provide a transfer film to the balls during bearing operation. The very short life of the MoS_2 films and the lack of Teflon film formation in the liquid oxygen (LOX) environment results in inadequate lubrication. The attendant wear and thermally induced bearing loading have resulted in the much shorter lives experienced to date.

The failure analysis work highlighted above indicates that significant life improvements may be realized by providing low friction, wear resistant rolling contact surfaces. Computer stress analysis indicates that the means by which low friction and wear resistance are obtained must be consistent with metallurgical requirements associated with successful bearing performance in a very severe rolling contact stress environment.

This paper reports the results to date of a study aimed at identifying new bearing materials offering significant potential for extending bearing performance life in advanced cryogenic turbopumps. State-of-the-art bearing analysis techniques were employed to parametrically quantify bearing material property requirements in high pressure cryogenic turbopumps, with emphasis placed on the HPOTP. The results of this analysis were used to establish well defined criteria for the selection of new materials and associated processing techniques. Materials and processing techniques can now be reviewed and rated in compliance with the selection criteria.

HPOTP Rotor/Bearing System

Figure 1 is a cross-section of the high pressure liquid oxygen turbopump (HPOTP) showing the locations of the two pairs of preloaded angular contact ball bearings that support the pump rotor. The bearings are mounted in a back-to-back arrangment with a pair on each side of the main pump impeller. The rotor is designed to act as a floating piston (balance piston) to balance out as much as possible any residual thrust loads on the bearings from the fluid pressures acting on the rotor. Preload springs between the bearings in each pair and apply thrust preloads of 3.8 to 4.5 kN (850 to 1000 lbf) to prevent skidding, depending on bearing size.

The rings and balls in the angular contact ball bearings in these turbopumps are made of CEVM 440C martensitic stainless steel. The cages are made of Armalon, a woven glass reinforced Teflon composite, which is intended to provide lubrication for the ball and race contacts by Teflon transfer films to the balls from the cage pockets. Bearing cooling is achieved by a substantial flow of the LOX through the bearings. The extreme temperatures of the environment preclude use of any conventional type of bearing lubrication.

The bearings are designed with 52% to 53% ball groove conformities and nominal unmounted contact angles of 20° to 28°. The ring mounting fits are designed to be a sliding fit on the outer rings and a tight fit on the inner rings, which is to become essentially line-to-line during operation, taking into account differential thermal growths and centrifugal effects on both the inner ring and shaft.

Analytical Studies

The bearing system analyses were conducted with the SHABERTH[5] (SHaft-BEaring-THermal) computer program. This approach provided an efficient means of calculating the stress, kinematic, and thermal information required to describe the bearing internal operating environment. Bearing stresses, fatigue lives, heat generation rates, temperatures, rolling and sliding velocities were calculated. The analysis incorporates the effects of thermal contractions, shaft stiffness, bearing preload, coolant temperature and flow rate, solid lubricant traction coefficients, and material elastic and heat transfer properties.

Two key features of SHABERTH with respect to the cryogenic turbopump bearing analyses are the inclusion of a general thermal analysis model and a dry lubrication option. The high bearing heat generation rates due to the use of solid lubrication make consideration of thermal and mechanical interactions very important in analyzing the bearings. The dry lubrication option makes use of a coulomb friction model to calculate bearing heat generation rates.

All of the possible heat transfer mechanisms including conduction, free and forced convection, and fluid flow are considered in the SHABERTH thermal model. The thermal analysis is coupled with the bearing mechanical analysis enabling consideration of thermomechanical interplay. Steady state or time transient temperatures may be calculated.

An interactive thermomechanical analysis of the aforementioned HPOTP shaft/bearing system was performed with the objective of parametrically quantifying HPOTP bearing operating environments from a materials viewpoint. Table 1 is a listing of the loading conditions modelled in the analytical studies.

Bearing fatigue life calculated in SHABERTH is based on classical subsurface initiated failure. Empirically derived multipliers are often applied to this calculated value to reflect observed effects of bearing material quality and lubrication conditions in an adjusted rating life, or expected life. In the current analytical studies no multipliers were used, so the values indicated are the unadjusted, calculated lives that may be achieved if surface integrity is protected against wear and lubrication related surface distress. The surface characteristic that was parameterized was coefficient of sliding friction. This permitted characterization of the effect of coefficient of friction on bearing heat generation and the associated thermally induced loading, and the resulting effect of thermal loading on bearing fatigue life.

Figure 2 is a plot of bearing fatigue life versus friction coefficient for Bearings #2 and #3, for two of the radial load conditions that were analyzed. Under the worst case radial load conditions (i.e. 22.2 kN (5000 lbf) total radial shaft load) the life of Bearing #3 decreases from a calculated 9 hours at coefficient of friction (μ) = 0.08 to a life of less than 1 hour at μ = 0.5. Under the minimum radial load condition plotted (i.e. 8.5 kN (1915 lbf) total radial shaft load), the calculated life for Bearing #3 shows a very dramatic decline from approximately 44 hours at μ = .08, to 3 hours at μ = 0.5. (A coefficient of friction for unlubricated 440C-440C sliding of 0.65 is reported by Spalvins[6]).

The life degradation is severely aggravated by the application of thrust load (possibly caused by the failure of outer race axial movement) resulting in the trends in bearing life reduction plotted in Figure 3.

Figures 2 and 3 indicate that even if surface initiating mechanisms can be ruled out as causes for failure, the HPOTP bearings will have difficulty achieving the required fatigue lives unless friction coefficient can be minimized and the occurrence of applied thrust load can be eliminated.

Figures 4 and 5 present bearing element-to-raceway stress levels versus friction coefficient and applied thrust load, respectively. Again, a dramatic sensitivity to friction coefficient is indicated. With the application of thrust load the contact stress levels can exceed 3447 MPa (500 ksi).

These high stress levels and low fatigue lives are a result of thermally induced loading. The radial expansion of the bearing created by the heat generated within the bearing is causing a radial preloading effect. Even though the bearings are operating in a cryogenic environment and coolant flow is present through the bearings, the ball/race contacts are still a source of considerable heat generation. Figures 6 and 7 illustrate the heat generation rates for the HPOTP bearings.

The figures (6 and 7) show total bearing heat generations, however, it should be noted that 55% to 65% of this generated heat is at the inner race contacts. This is due to the fact that the computer modeling has assumed outer race control, which means that all the ball spinning takes place at the inner race contact. Since there is no gross sliding present (axial spring preload are sufficient to prevent sliding) the spin at the inner race is the major contributor to the heat generation. Spin-to-roll ratios for the inner race contacts ranged from 0.1 to as high as 0.6 under light loads. A spin/roll ratio of 0.3 was typical for nominal operating conditions.

Other bearing performance parameters examined included, operating contact angles, subsurface stresses, and circumferential ring stresses. The trends in the results for these parameters were consistent with those already examined.

In summary, the analytical studies highlighted the following key points about the HPOTP bearing operating environment:

1) Due to the short duration of the MoS_2 films on the balls and raceways, and the inability of the Teflon transfer to be effective in the LOX environment, the HPOTP bearings most likely operate with friction coefficient values approaching that of dry metal-to-metal contact (i.e. $\mu > .35$).

2) At these high friction levels and under maximum operating conditions the bearing heat generation rates reach levels on the order of 20 to 50 kilowatts (19 to 47.4 Btu/sec).

3) The high rate of heat generation creates a situation of inner ring thermal expansion which creates high [up to 3447 MPa (500 ksi)] bearing contact stress levels and greatly reduced bearing rolling contact fatigue lives.

4) Any applied thrust load dramatically aggravates the above scenario and provides increased potential for a thermal runaway type failure.

5) The analysis indicates that much improved bearing life may be realized at low (<0.15) friction coefficient levels coupled with minimized bearing wear and thrust loading.

Bearing Material Consideration

Bearing material candidate eligibility is to be assessed on the basis of compatibility of their property offerings with the property requirements identified in the analytical studies.

The required material properties include:

- rolling contact fatigue life consistent with the 7.5 hour design goal
- dimensional stability at cryogenic temperatures
- fracture toughness sufficient to preclude catastrophic component fracture in service
- corrosion resistance at least equal to that of 440C steel
- ball-race contact friction coefficient less than 0.15
- wear resistance consistent with maintaining the required contact friction through the life goal of 7.5 hours.

Bulk material requirements for hardness, rolling contact fatigue life, dimensional stability and fracture toughness are adequately provided by conventional, high quality, through hardening bearing steels such as 440C, 52100 and M50. Of these materials, 440C is the only one offering corrosion protection, but with a sacrifice in rolling contact fatigue performance compared to 52100 or M50. The required friction and wear resistance properties are not provided by these materials.

Coefficients of friction obtained from unlubricated 4-ball wear tests are reported by Kannel, et al[7], for 52100, 440C, BG-42 and Star J. The values are in the range of 0.4 to 0.6 for all of the materials. It is doubtful that appreciably different values would be demonstrated with any other bearing steel composition. As previously discussed, the solid film lubrication methods used in the current

HPOTP system appear inadequate to prevent metal-to-metal contact between the balls and raceway surfaces. With the constraints of the HPOTP operating environment, and restrictions on the types of lubricating substances that can be introduced into the LOX system, no outstanding replacement candidates for Armolon have been identified. Consequently, friction and wear control of the required magnitude necessitates measures offering significantly more reliability and efficiency than that provided by the current lubrication scheme.

There are several surface treatment techniques, compatible with hardened steel substrates, that provide substantial reductions in friction, and improved wear and corrosion resistance. Some have been tested in rolling contact fatigue and have demonstrated excellent performance. These include reactively sputtered (RS) TiN, a reactively sputtered material referred to as "Hard Molybdenum" and electrodeposited chromium via the Armoloy process[8,9]. Although the rolling contact tests were oil lubricated, specimens of M50 steel coated with RS TiN survived 167.5 hours (86.4×10^6 cycles) at a contact stress of 5419 MPa (786 ksi) without failure, demonstrating remarkable coating adherence[8].

Coefficient of friction values in the range of 0.05 to 0.1 are reported by Jamal, et al[10], for unlubricated test couples when both surfaces are coated with TiC or TiN. Ramalingam[11] reports coefficient of friction values in the range of 0.13 to 0.16 for TiN coated AISI 1018 steel (annealed) running against 52100 steel (RC62). Wear rate reductions of approximately two orders of magnitude were obtained with TiN coated 1018 steel as compared with uncoated 1018 steel. A value of 0.16 is reported by the Armoloy Corporation for uncoated steel against Armoloy coated steel.

Increased adhesive and abrasive wear resistance of both carbide and high speed steel cutting tools coated with TiN and TiC is credited by Bunshah[12] for life improvement factors of 2 to 8 over uncoated tools.

Exceptional corrosion and wear resistance, reported coefficient of friction values in the vicinity of 0.15 demonstrated adherence under extremely high rolling contact stress, and compatibility with conventional, high performance bearing steel add up to provide solid endorsement of hard coatings as candidates for evaluation in HPOTP bearing services.

An enduring surface coating exhibiting low friction in unlubricated conditions (compared to an unlubricated steel-steel contact) not only offers potential for significant reductions in bearing heat generation and the associated thermal loading, but also provides for lower surface tractions, thereby diminishing life reduction effects described by Smith and Liu[13]. Additionally, substrate material selection can be based on rolling contact fatigue performance, uncompromised by bulk material requirements for corrosion and wear resistance.

Information is being compiled concerning application techniques, properties and performance characteristics of hard coatings as well as that pertaining to alternate surface treatments. Specific substrate-surface treatment combinations will be identified and recommended for further evaluation as HPOTP bearing material candidates.

Summary and Conclusions

1) The current HPOTP bearing lubrication methods are inadequate, resulting in ball-race friction approaching that of unlubricated metal-to-metal contact (i.e. μ = .35 to .60).

2) Under these high friction conditions and maximum load, bearing heat generation rates reach levels

on the order of 20 to 50 kilowatts (19 to 47.4 Btu/sec).

3) The high rate of heat generation results in thermally induced bearing loading and greatly reduced bearing fatigue life.

4) Applied thrust load significantly increases bearing heat generation rates.

5) Bearing life approaching the design goal of 7.5 hours may be realized with a coefficient of friction for ball-race contact of 0.15 or less, coupled with minimized bearing wear and thrust loading.

6) Hard coatings (e.g. reactively sputtered TiN) used in conjunction with a high performance bearing steel (e.g. M50) offer significant potential to provide the combination of properties indicated by the analytical studies as requisite for HPOTP bearing design life attainment.

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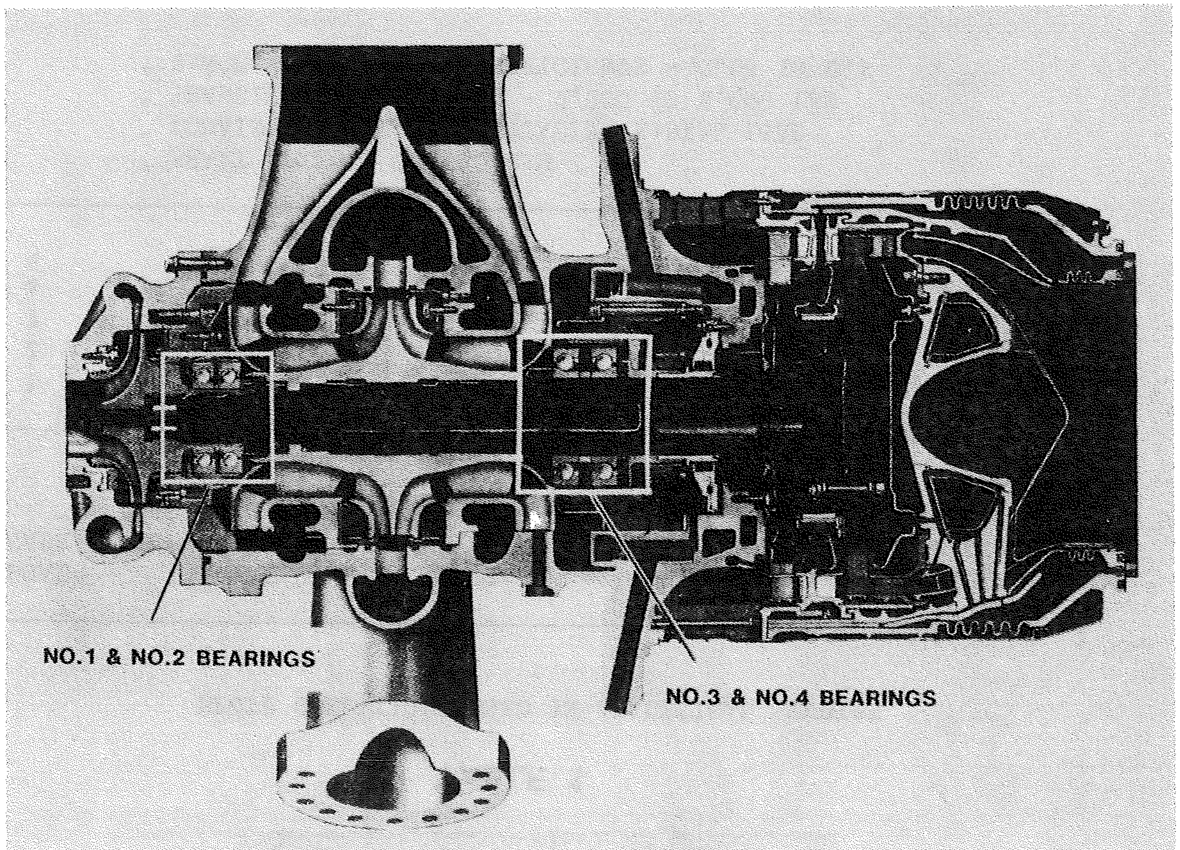

FIGURE 1. SSME HPOTP

TABLE 1

HPOTP CONDITIONS USED IN ANALYTICAL STUDIES

LOAD CASE	RADIAL NON-ROTATING LOAD COMPONENT (LBS)	RADIAL ROTATING LOAD COMPONENT (LBS)	ORIENTATION OF ROTATING & NON-ROTATING LOAD COMPONENT (DEG)	RESULTANT RADIAL LOAD (LBS)
1	2500	2500	0	5000
2	2500	2500	45	4620
3	2500	2500	90	3535
4	2500	2500	135	1915
5	2500	2500	180	0

- SHAFT SPEED - 30,000 RPM
- COMBINED STATIC AND ROTATING RADIAL LOAD
- TRANSIENT AXIAL LOADS - 2,000 TO 8,000 LBS
- RANGE OF FRICTION COEFFICIENTS - 0.08 TO 0.5

FIGURE 2
SSME HPOTP
Fatigue Life vs Friction Coefficient

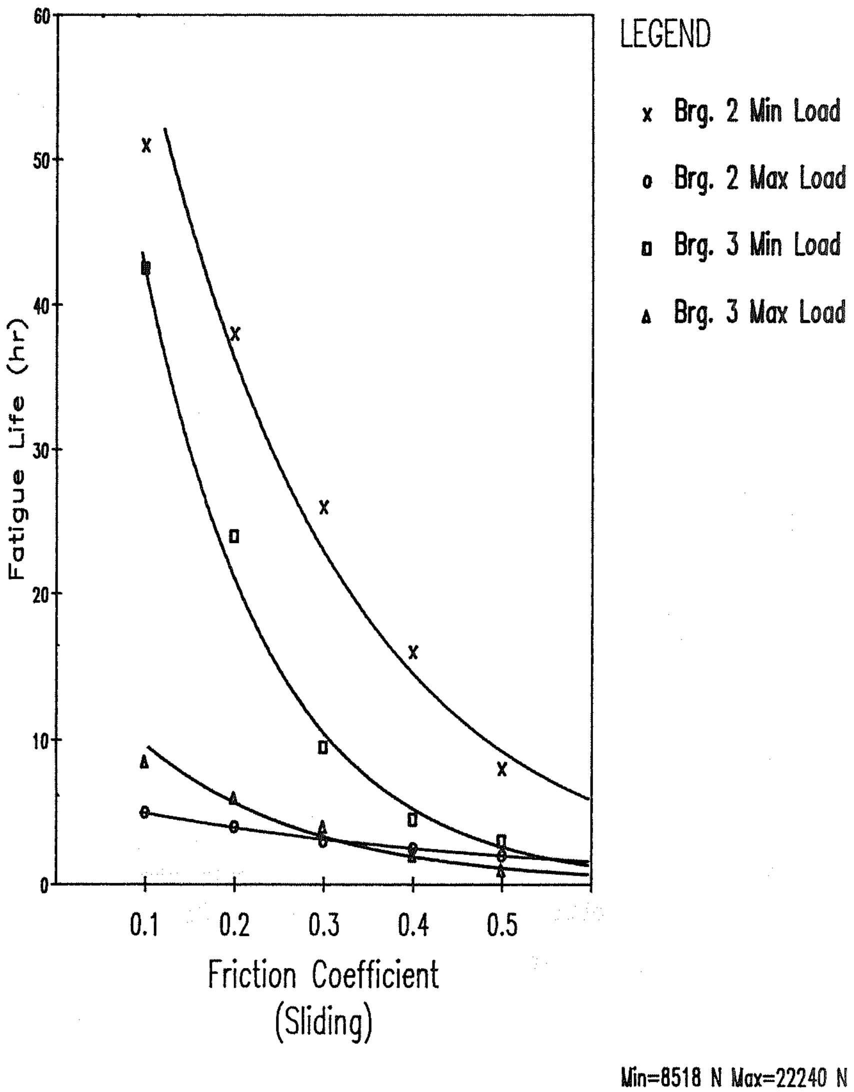

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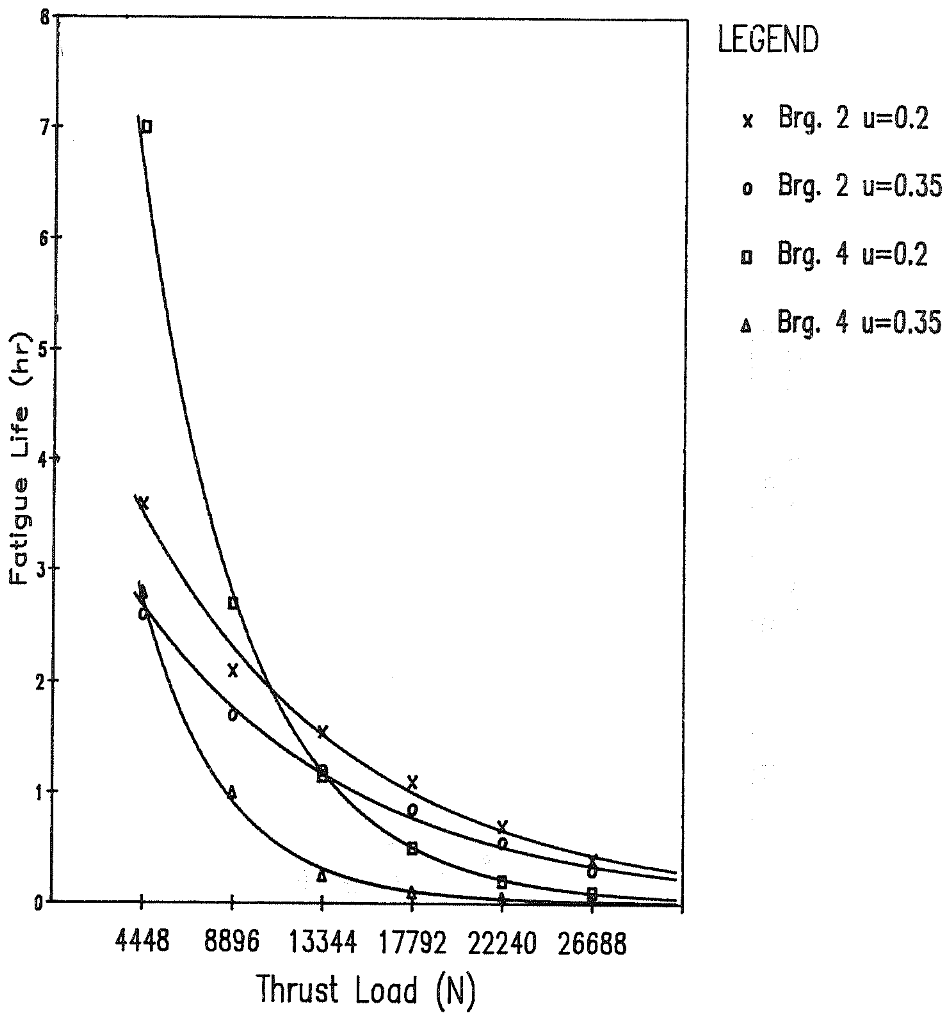
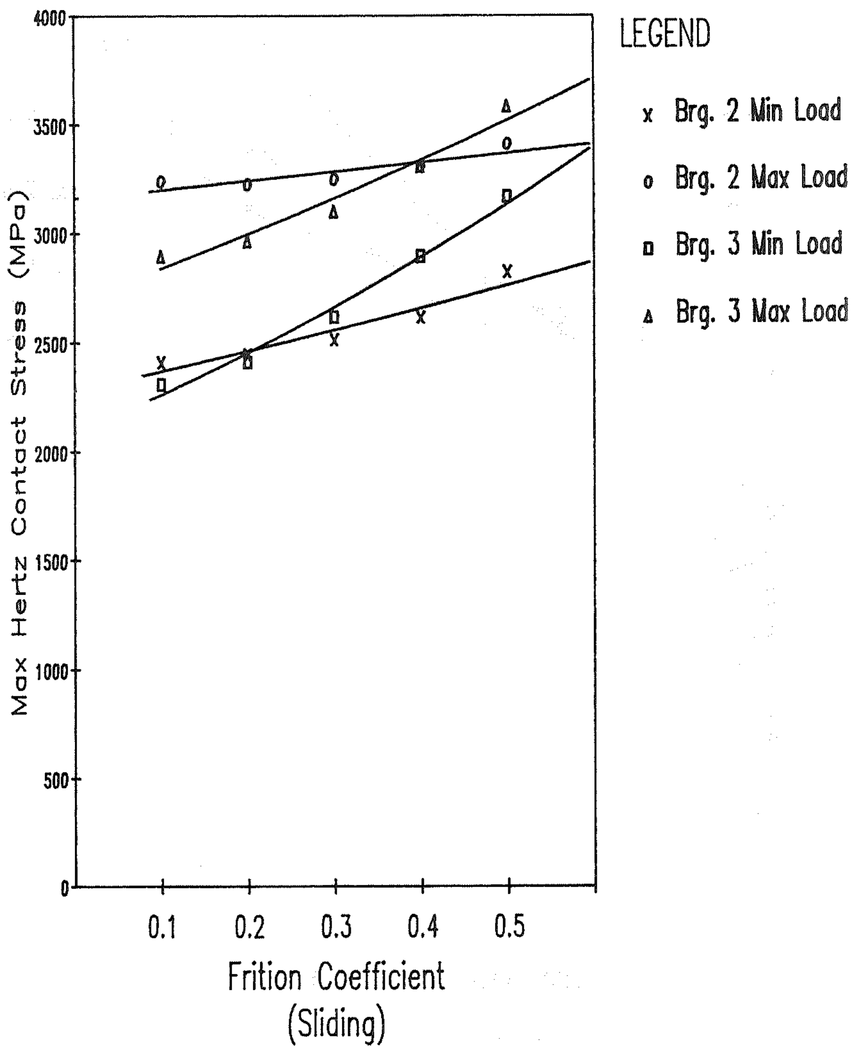
FIGURE 3
SSME HPOTP
Fatigue Life vs Thrust Load



Max Shaft Load=22240 N

FIGURE 4
SSME HPOTP
Contact Stress vs Friction Coefficient



Min=8518 N Max=22240 N

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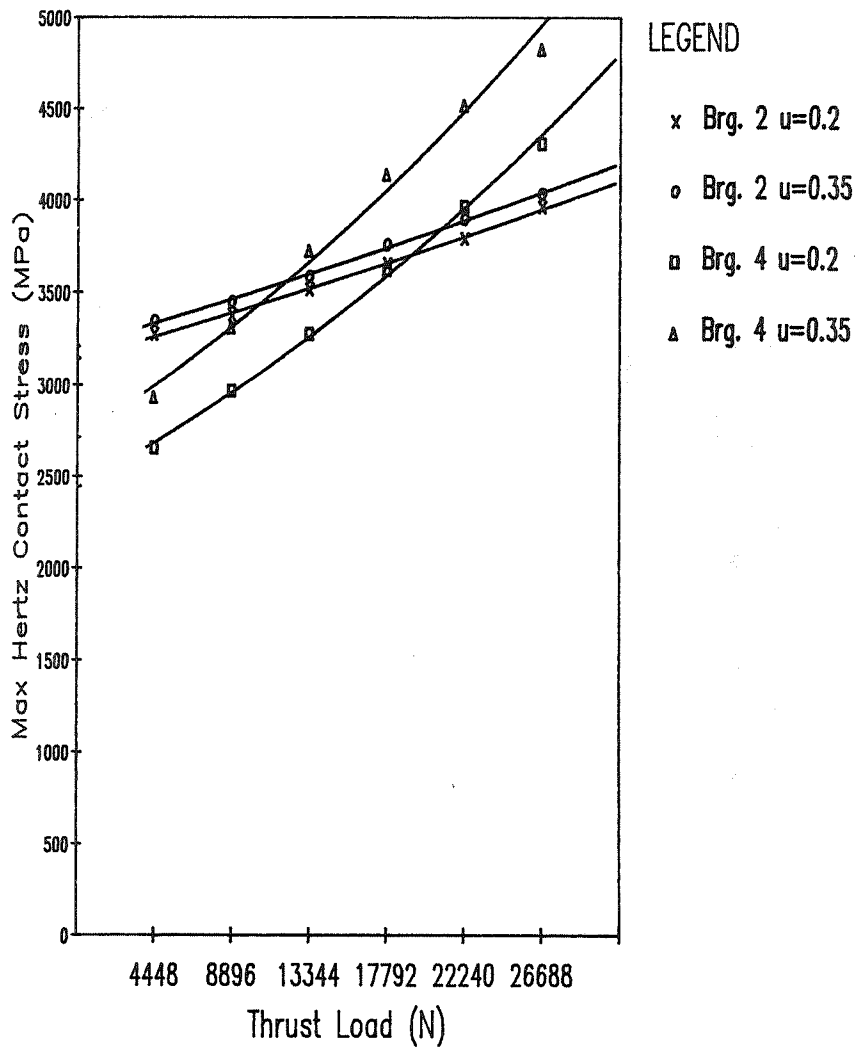
FIGURE 5
SSME HPOTP
Contact Stress vs Thrust Load



Max Shaft Load=22240 N

FIGURE 6
SSME HPOTP
Heat Generation vs Friction Coefficient

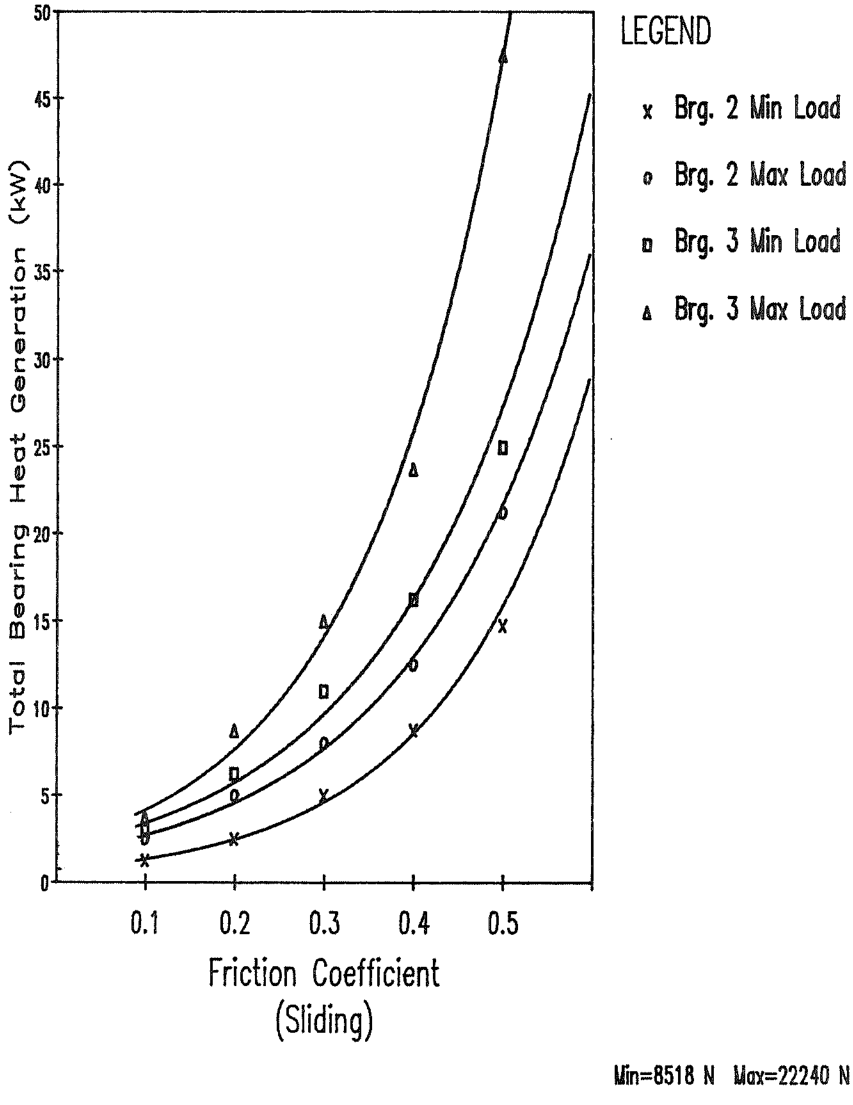
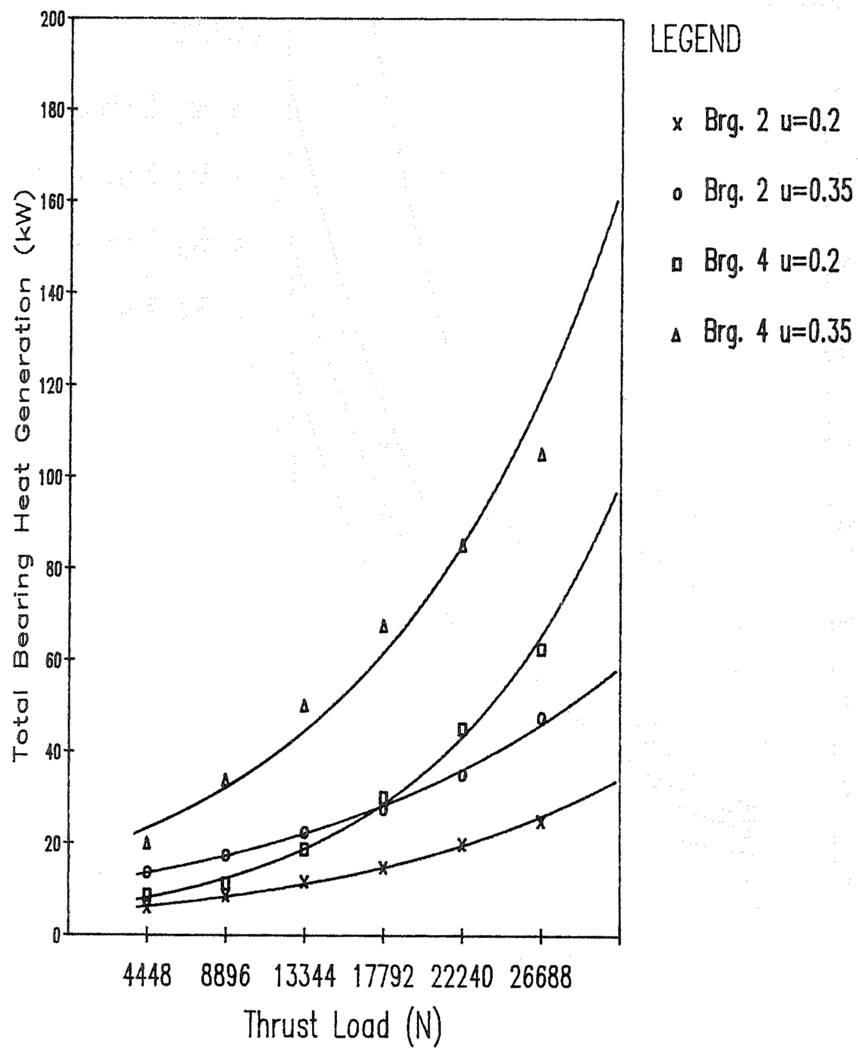


FIGURE 7
SSME HPOTP
Heat Generation vs Thrust Load



Max Shaft Load=22240 N

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