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ASSESSMENT OF THE OPERATING CHARACTERISTICS OF THE SSME LOX TURBOPUMP PUMP-END BEARING

FINAL REPORT

DECEMBER, 1984

PREPARED BY
SRS TECHNOLOGIES
UNDER PURCHASE ORDER H-78194B
FOR THE
GEORGE C. MARSHALL SPACE FLIGHT CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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FOREWORD

This report was prepared by SRS Technologies under Purchase Order H-78194B entitled "Assessment of the Operating Characteristics of the SSME LOX Turbopump, Pump-End Bearing" for the George C. Marshall Space Flight Center of the National Aeronautics and Space Administration. The work was administered under the technical direction of the Engineering Physics Division of the Materials and Processes Laboratory with Mr. Fred J. Dolan as Project Manager. This report describes the work accomplished over a six week period from November 9th to December 21, 1984. Mr. Joseph C. Cody was the Project Engineer. A listing of the key contributions is shown below.

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1.0 INTRODUCTION

This report describes the work accomplished during the six week contract period ending in December, 1984. The objective of this activity was to model the SSME LOX turbopump bearings and shaft, and to evaluate effects of specific loading and thermal conditions on the load sharing and operating characteristics of the pump end bearings. Due to the limited time for the investigation, variations in loading and thermal boundary conditions were limited to a few parameters that was considered to be of most interest.

2.0 SUMMARY

A bearing/shaft model of the SSME LOX turbopump has been developed using the SHABERTH bearing/shaft math modeling computer code. A previously developed bearing/shaft thermal model of the SSME LOX turbopump turbine end bearing was used in conjunction with SHABERTH to evaluate the thermo-mechanical operating characteristics of the LOX turbopump end bearings. This model is described in detail in Reference 2. Since time and resources did not allow for development of a detailed thermal model of the pump end bearings, the turbine end bearing thermal mode developed in Reference 2 was used. The close similarity of the bearings justify this approach for rapid and preliminary assessment.

The analysis results show that, for the two unmounted diametrical clearances evaluated (4.0 mils and 6.3 mils) the inboard pump end bearing supports about 81% of the isolator load for the small clearance and 77% of the isolator load for the larger clearance, if uniform bearing temperatures are assumed. The load sharing becomes more severe when thermal effects are included, approximately 89% and 85% of the load supported by the inboard bearing for the clearances of 4.0 mils and 6.3 mils respectively.

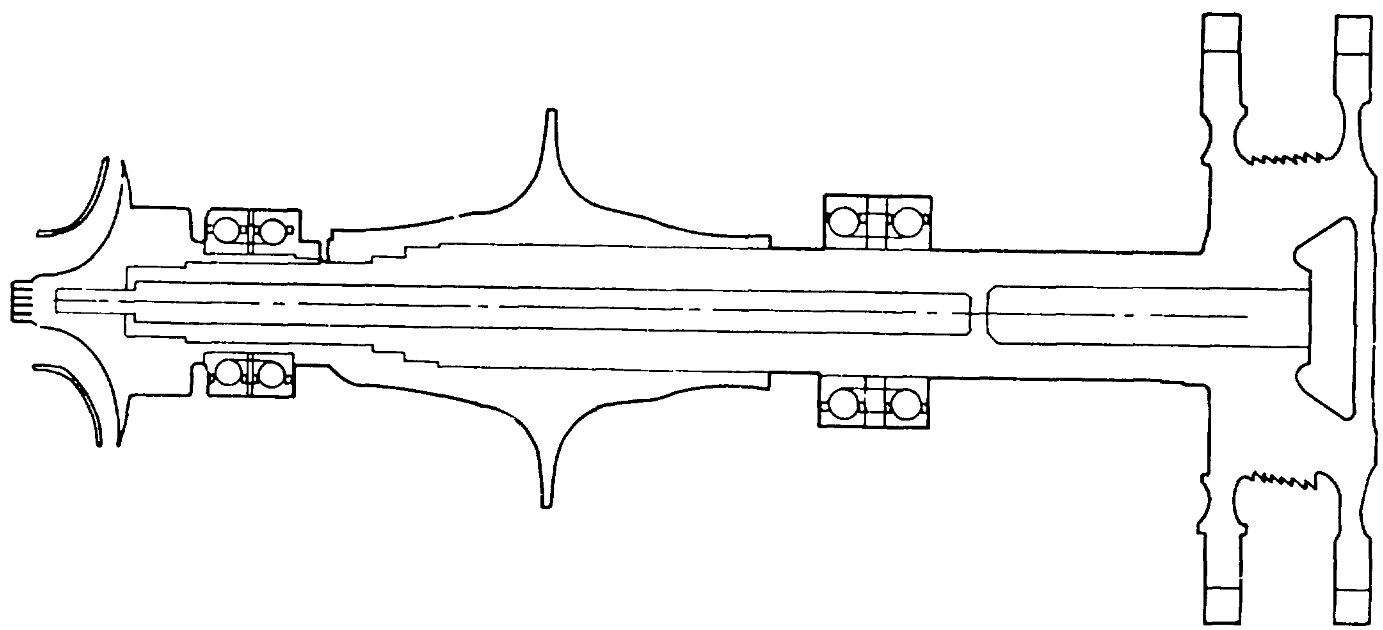
Bearing clearance changes due to thermal effects were; 40% for the 4.0 mil diametrical clearance case and 19% for the 6.3 mil clearance case evaluated. A nominal coolant flow of 7 lbs/sec and inlet coolant temperature of -230 °F was used. Frictional heat generation decreased when the clearance was increased from 4.0 mils to 6.3 mils by 1.6 KW. Also, the maximum hertz stress increased by 35 kpsi when increasing the clearance. The reduction in heat generation and increase in contact stresses was also significantly influenced by a change in the inner race curvature from 0.53 to 0.55.

The thermal analysis included evaluation of bearing temperatures for a subcooled case (coolant 5° below saturation temperature at bearing 2 inlet) and a saturated case (coolant at saturation temperature at bearing 2 inlet). The LOX coolant saturation temperature being -214 °F. Analysis results indicated that no drastic temperature change occurred between the two cases. Since the rolling element and race surfaces of the subcooled case were at temperatures sufficiently high enough to be vapor blanketed, exceeding saturation temperature at the bearing inlet did not increase surface temperatures greatly.

3.0 SHAFT/BEARING MODEL DESCRIPTION

A shaft/bearing system model representing the Space Shuttle Main Engine (SSME) Liquid Oxygen (LOX) turbopump shaft and bearing configuration was developed. The model consists of the turbopump shaft, turbine end, and pump end bearings. The major elements of the model are shown in Figure 3.1. As configured radial loads can be applied to the shaft to simulate turbine, main pump impeller and preburner pump impeller loading. Bearing preloads can be varied to investigate the effects of preloads (bearing stiffness) on bearing load sharing. In addition modeling the complete shaft/bearing system allows the effects of shaft deflection on bearing load sharing to be evaluated. These are typical effects that cannot be investigated with a single bearing model. The model has the flexibility to evaluate the effects of various load combinations on bearing operating characteristics such as deflections, contact stresses, ball speeds, etc.

FIGURE 3.1 LOX TURBOPUMP BEARING/SHAFT CONFIGURATION



4.0 SHAFT LOADING

The turbopump shaft loading conditions were generated to produce pre-burner pump bearing reactions consistent with measured loads reported in Reference 1 for 109% power level. Initial analyses were conducted using the load data representing build number 9708R2. Subsequent loads representing build number 2606R1 were evaluated. Table 4.1 contains the load information for 9708R2, and Table 4.2 contains the load data for 2606R1.

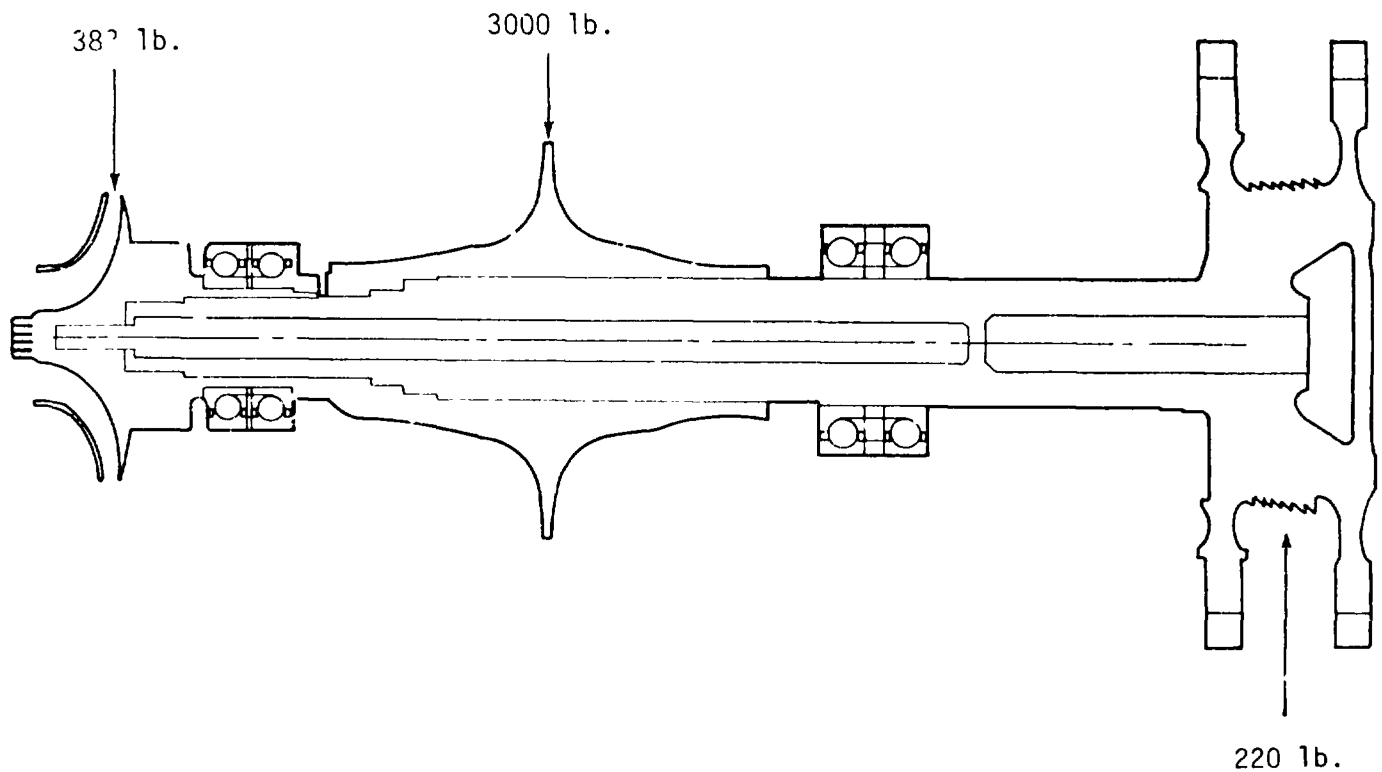
TABLE 4.1 Load Data for Build No. 9708R2

TEST NO.	LOADS (LBS)		DEVIATION (LBS)	
	FIXED	DYNAMIC	FIXED	DYNAMIC
750-241	2568	3076	175	433
750-242	2049	2563	-344	-80
750-243	2152	2487	-241	-156
750-244	2287	2773	-106	130
750-245	2907	2317	514	-320
	AVERAGE LOADS		STANDARD LOADS	
	2393	2643	311	261
	FIXED LOAD=2393±311		MAX COMBINED LOAD=5608	
	DYNAMIC LOAD=2643±261		MIN COMBINED LOAD=4464	

TABLE 4.2 Load Data for Build No. 2606R1

TEST NO.	LOADS (LBS)		DEVIATION (LBS)	
	FIXED	DYNAMIC	FIXED	DYNAMIC
750-236	565	1400	-99	231
750-237	752	1466	88	297
750-239	738	1022	74	-147
750-240	603	788	-61	-381
	AVERAGE LOADS		STANDARD LOADS	
	664	1169	82	278
	FIXED LOAD=664±82		MAX COMBINED LOAD=2193	
	DYNAMIC LOAD=1169±278		MIN COMBINED LOAD=1473	

FIGURE 4.1
LOX TURBOPUMP BEARING/SHAFT LOAD CONFIGURATION



The following radial loads were used in the shaft/bearing model to provide maximum pump end bearing reactions representative of the combined loads shown in Table 4.1.

Preburner Impeller	1060 lbs
Main Impeller	7000 lbs
Turbine	-950

The radial load distribution shown below (and illustrated in Figure 4.1) was applied to the shaft/bearing model to provide the maximum pump end bearing reaction shown in Table 2 for build 2606R1.

Preburner Impeller	382 lbs
Main Impeller	3000 lbs
Turbine	-220 lbs

5.0 SHAFT/BEARING ANALYSIS RESULTS

Several different loading conditions and bearing temperature profiles were investigated. Different loading conditions with uniform bearing temperatures are described in this section. Results obtained with the integrated thermal-mechanical analysis are described in Section 7, "Thermal Analysis Results".

In evaluating the shaft loading profiles, the following conditions were used:

- o The pump end bearings were preloaded to 800 lbs and the turbine end bearings were preloaded to 1000 lbs.
- o A uniform temperature of 231°R was used for all bearing components - this condition was relaxed for the thermal-mechanical analysis (Section 7).
- o The outer races of all bearings were assumed fixed - this is a worst condition simulating outer race "lock up".
- o A contact friction coefficient of 0.2.
- o Shaft speed of 30,000 RPM.

As previously discussed loads from two pump configurations was evaluated. Although the pump end bearing isolator loads for 9708R2 were among the highest recorded, and not representative of loads recorded with improved configurations, the effects of these loads on bearing and shaft characteristics were judged to be of interest because they do represent an upper limit of the loads experienced.

The following results were obtained from the shaft/bearing model using loading representative of configuration 9708R2 (unmounted diametrical clearance = 4.0 mils).

TABLE 5.1. Bearing Operating Characteristics for Build No. 9708R2

*BGR NO.	REACTIONS (LBS) RADIAL	AXIAL	MOMENTS (FT-LBS)	MAX. HERTZ STRESS (kpsi)	DEFLECTION (INCHES)	HEAT GENERATED (KW)
1	1134	1363	-45	404	.00122	4.7
2	4085	-2518	209	592	.002	12.6
3	3635	2562	-234	517	.0012	26.9
4	-1743	-1407	-116	406	-.0004	11.0

*The bearings are numbered as follows: 1 and 2 are pump end bearings and 3 and 4 are turbine end bearings.

The sum of bearing 1 and 2 radial reactions is 5219 lbs which is slightly less (7%) than the value of 5608 lbs shown in Table 1. This small difference does not warrant rerunning the model to exactly match the maximum measured isolator load. It is significant to note that bearings 1 and 2 do not equally share the total load. Bearing 2 reacts about 78% of the load while bearing 1 supports the remainder. The unequal load sharing is caused primarily by shaft deflection. A contributing factor is the increased axial load induced into bearing number 2. Since the outer races are assumed to be fixed, the increased radial loading of the inboard bearings induces a higher axial load causing the inboard bearings to become "stiffer" in the radial direction. They will therefore support a larger part of the total radial load. For this specific loading condition, the high loads on bearing number 2 also causes four of the balls to become unloaded at the inner race opposite the point of load application. The load for this bearing is therefore carried by 9 balls rather than the total 13, resulting in higher ball loading and contact stresses. Although these loads are considered unrealistically high for current pump configurations they do provide insight to bearing operating characteristics at extreme loads. Thermal gradients resulting from high friction heat generation was not evaluated for this load case. These gradients would undoubtedly cause loss of clearance due to thermal growth and adversely affect bearing operating conditions by increasing loads and stresses.

Results from the loading profile used to produce isolator reactions representing pump configuration 2606R1 (unmounted diametrical clearance = 4.0 mils) are summarized below.

TABLE 5.2 Bearing Operating Characteristics for Build No. 2606R1

BEARING NO.	REACTIONS (LBS) RADIAL	AXIAL	MOMENTS (ft-lbs)	MAX. HERTZ STRESSES (Kpsi)	DEFLECTIONS (INCHES)	HEAT GENERATION (KW)
1	398	1004	-16	324	.00049	2.7
2	1705	-1385	89	443	.00084	4.4
3	1454	1369	-86	389	.00063	7.0
4	-396	-989	-27	224	.00003	3.5

These reactions, moments, stresses, etc. are significantly lower than for pump configuration 9708R2. The radial load sharing for bearings 1 and 2 are approximately 20% and 80% respectively. These loads do not cause the balls of bearing 2 to become unloaded as in the previous loading condition (9708R2).

The above information was developed for bearings with uniform temperature. Thermal effects, with the loading conditions for configuration 2606R1, are evaluated in Section 7.

6.0 THERMAL MODEL DESCRIPTION

Resources and time did not allow for development of a thermal model specifically for the LOX turbopump end bearing. A thermal model of the turbine end 57 mm bearing has been developed for the Bearing Materials tester program under contract NAS8-34686 under the technical direction of Mr. Fred J. Dolan. This model is described in Reference 2. Since the pump end bearing, 45 mm bore, and the turbine end bearing, 57 mm bore, have many similarities, it was judged that a reasonable estimate of the pump end bearing component temperatures could be made by using the turbine bearing thermal model. Friction heat generation was determined for specific loading and operating conditions from the shaft bearing model. This data was used as input to the bearing thermal model to estimate bearing component temperatures. Iterations were made between the bearing thermal model and the shaft/bearing model to arrive at a consistent set of component temperatures, heat generation rates, and operating clearances.

Since there was interest in evaluating the effects of two phase flow on the thermal characteristics of the number 2 pump end bearing, a thermal analysis of this bearing was conducted using saturated coolant at the entrance to bearing number 2. Loading conditions of build number 2601R1 were used in developing the heat generation rates for bearing number 2.

7.0 THERMAL ANALYSIS RESULTS

The thermal-mechanical analysis was performed using loading conditions representative of build number 2601R1. Two different diametrical clearances were evaluated, 4 mils and 6.3 mils. The fixed operating conditions are listed below.

- o Shaft Speed (30,000 RPM)
- o Coolant Flow (7.0 lb/sec)
- o Friction Coefficient (.2)
- o Preloads:
  - Pump End (800 lbs)
  - Turbine End (1000 lbs)

The following results were obtained from the shaft/bearing model using an unmounted diametrical clearance of 4.0 mils (inner race curvature = . outer race curvature = .52). This data was developed for the pump bearings with thermal effects considered

TABLE 7.1 Bearing Operating Characteristics Considering Thermal Effects (clearance = 4.0 mils)

BEARING NO.	REACTIONS (LBS) RADIAL	AXIAL	MOMENTS (ft-lbs)	MAX. HERTZ STRESSES (Kpsi)	DEFLECTIONS (INCHES)	HEAT GENERATION (KW)
1	241	1811	-6	362	.0003	5.5
2	1886	-2105	98	460	.0005	7.1

In comparing operating characteristics in Table 4.2 which shows the data for not thermal effects considered and Table 7.1 which shows the data with a thermal gradient across the bearing, the deflection is less for bearing number 2 in Table 7.1. This is due to the thermal growth of the ball with respect to the races which reduces the operating clearance. A reduced operating clearance results in a larger axial reaction causing the bearing to become stiffer", or able to support a larger radial load.

Figure 7.2 illustrate the variation in maximum and average heat flux for bearing number 2 for each of the 13 rolling elements in the bearing. In Figure 7.2 the dashed lines indicate friction heat generation for an unmounted diametrical clearance of 4.0 mils (O.R. curvature = .52, I.R. curvature = .53) and the solid lines show the same data for an unmounted diametrical clearance of 6.3 mils (O.R.. curvature = .52, I.R. curvature = .55). The average heat flux is approximately 55% less than the maximum heat flux. This data reflects the unsymmetrical heat geneated in the bearing due to radial loading and the influence of clearance and curvature changes.

The thermal analysis performed provided the bearing operating temperatures shown in Table 7.6. The subcooled case provided 16 degrees of subcooling at the bearing 1 inlet and 3 degrees at the bearing 2 outlet with a ΔT across the bearing of 13°F. In order to evaluate the effects of two phase flow on the thermal characteristics of the number 2 pump end bearing, coolant at saturation temperature (-214°F) was modeled entering bearing number 2. The average component temperatures for the inner race, ball, and outer race were 4, 6, and 22 degrees higher respectively for the saturated case with 4.0 mil clearance, and 8, 13, and 25 degrees higher respectively for the saturated case with 6.3 mil clearance. The larger ΔT for the outer race is due to the assumption of a fixed outer race which simulates a worst case of outer race "lock up". These results indicate that no drastic temperature change occurred between the saturated and subcooled cases as might be expected. This is due to the fact that the surfaces of the bearing components were already vapor blanketed before the coolant entering bearing 2 became saturated.

Temperature distributions and gradients for the number 2 pump end bearing operating in saturated coolant are shown for the rolling element and inner and outer races in Figures 7.3, 7.4 and 7.5 respectively. All bearing component surface temperatures are running well above the LOX coolant saturation temperature indicating that the bearing is operating in vapor, also the temperature gradients in the bearing components are quite severe.

Results from the shaft bearing model using an unmounted diametrical clearance of 6.3 mils (inner race curvature = .55, outer race curvature = .52) are presented below. This data is for the pump end bearings with thermal effects considered. A representative bearing/shaft load and reaction configuration is shown in Figure 7.1. Note the opposing bearing reactions on the turbine end bearing set.

TABLE 7.2 Bearing Operating Characteristics Considering Thermal Effects (clearance = 6.3 mils)

BEARING NO.	REACTIONS (LBS) RADIAL	AXIAL	MOMENTS (ft-lbs)	MAX. HERTZ STRESSES (Kpsi)	DEFLECTIONS (INCHES)	HEAT GENERATION (KW)
1	313	1553	-13	382	.0004	3.9
2	1811	-1888	100	495	.0007	5.5

The effects of diametrical clearance and inner race curvature changes in terms of frictional heat generation and stresses are summarized below for bearing number 2.

TABLE 7.3 Frictional Heat Generation and Hertz Stresses for Bearing Number 2

CLEARANCES (mils) AND INNER RACE CURVATURE	MAXIMUM HERTZ STRESSES (KPSI)	AVERAGE FRICTION HEAT GENERATION (KW)
4.0 & .53	460	7.1
6.3 & .55	495	5.5

Bearing load sharing and operating clearance changes with and without considering thermal effects are shown in Table 7.4 and 7.5 as a function of both 4.0 mils and 6.3 mils unmounted diametrical clearance. As can be seen, the percentage of the isolator load reacted by bearing 2 at the pump end increases when thermal effects are considered. The bearing operating clearance changes for bearing 2 due to thermal effects are approximately 40% for the small clearance and 19% for the large clearance.

FIGURE 7.1
LOX TURBOPUMP BEARING/SHAFT LOAD AND REACTION CONFIGURATION

- Unmounted Diametrical Clearance = 6.3 mils
- Bearing Operating in Saturated Coolant

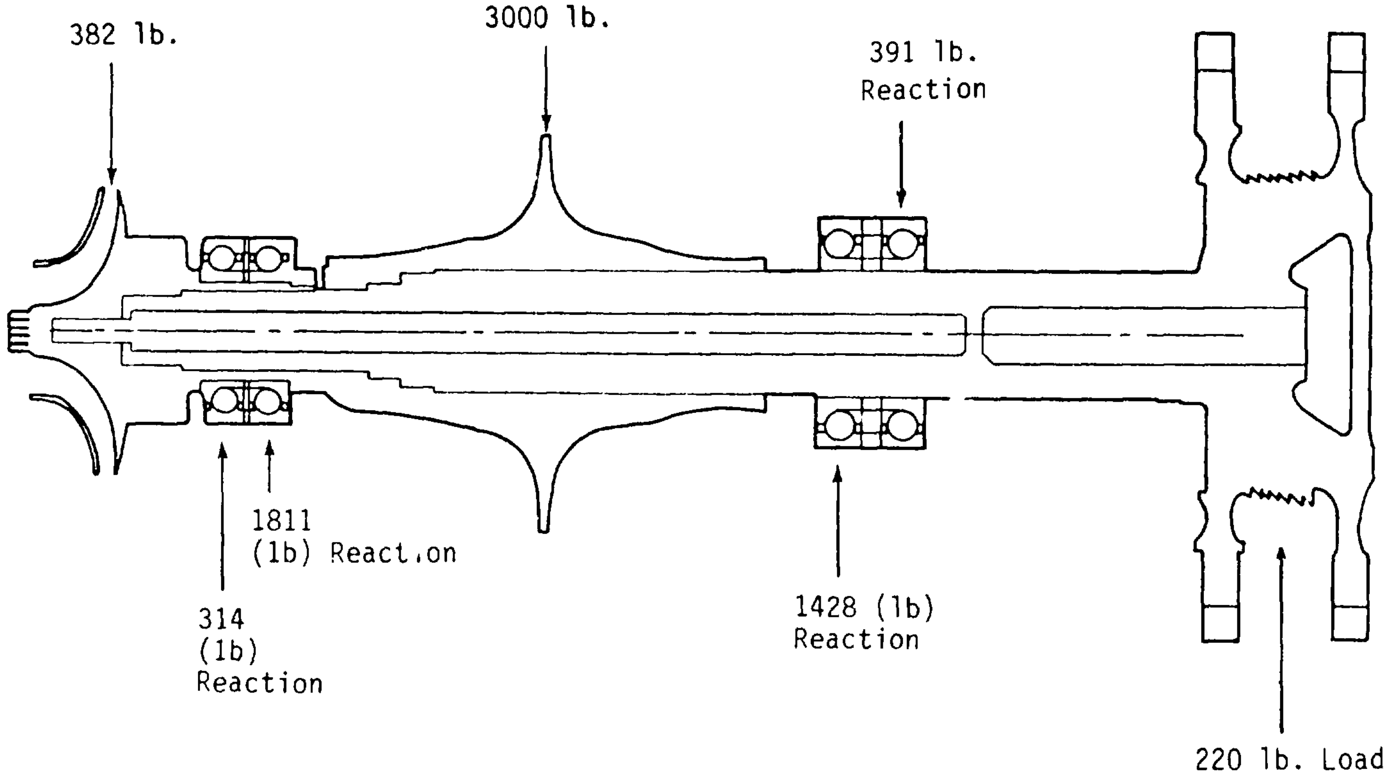


TABLE 7.4 BEARING LOAD SHARING

UNMOUNTED DIAMETRICAL CLEARANCE (mils)	PERCENT OF ISOLATOR LOAD REACTED AT PUMP END			
	NO THERMAL EFFECTS		WITH THERMAL GRADIENT	
	BEARING NO. 1	BEARING NO. 2	BEARING NO. 1	BEARING NO. 2
4.0	19	81	11	89
6.3	23	77	15	85

TABLE 7.5 BEARING CLEARANCE CHANGES

UNMOUNTED DIAMETRICAL CLEARANCE (mils)	BEARING 2 OPERATING CLEARANCES (mils)		
	NO THERMAL EFFECTS	WITH THERMAL GRADIENT	PERCENTAGE CHANGE
4.0	2.5	1.49	40
6.3	4.9	3.97	19

FIGURE 7.2   AVERAGE AND MAXIMUM FRICTIONAL HEAT GENERATION

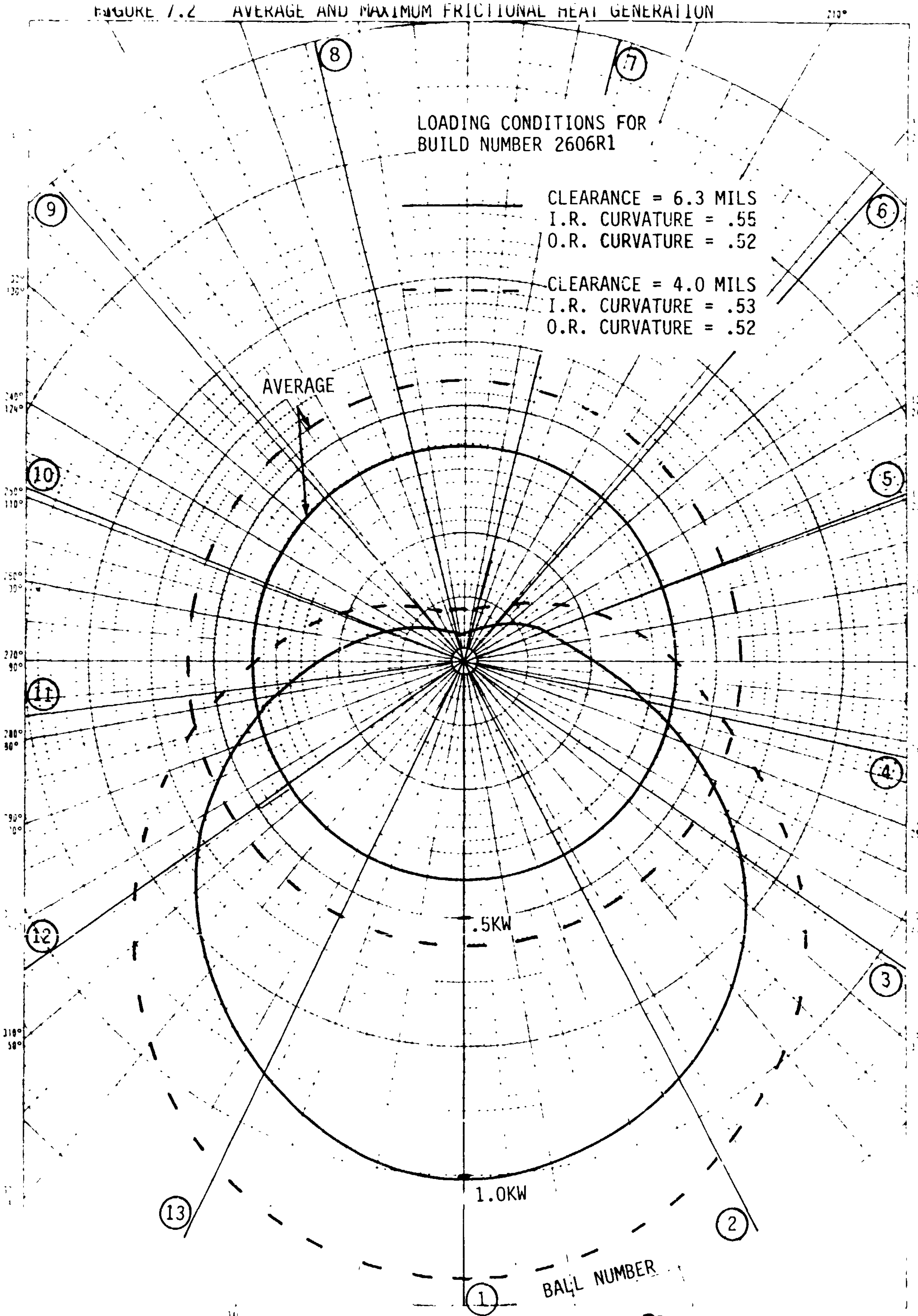


TABLE 7.6 BEARING OPERATING TEMPERATURES

COOLANT OUTLET CONDITION	UNMOUNTED DIAMETRICAL CLEARANCE (mils)											
	4.0						6.3					
	AVERAGE TEMPERATURE (°F)			MAXIMUM TRACK TEMPERATURE (°F)			AVERAGE TEMPERATURE (°F)			MAXIMUM TRACK TEMPERATURE (°F)		
	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE
SUBCOOLED	-136	71	-134	242	304	138	-159	-2	-152	125	190	95
SATURATED	-132	77	-112	246	309	146	-151	11	-127	134	199	106

FIGURE 7.3
ROLLING ELEMENT TEMPERATURE DISTRIBUTION FOR LOX TURBOPUMP BEARING (PUMP END)
(TEMPERATURES IN DEGREES FARENHEIT)

Unmounted Diametrical Clearance = 6.3 mils
Bearing Operating in Saturated Coolant

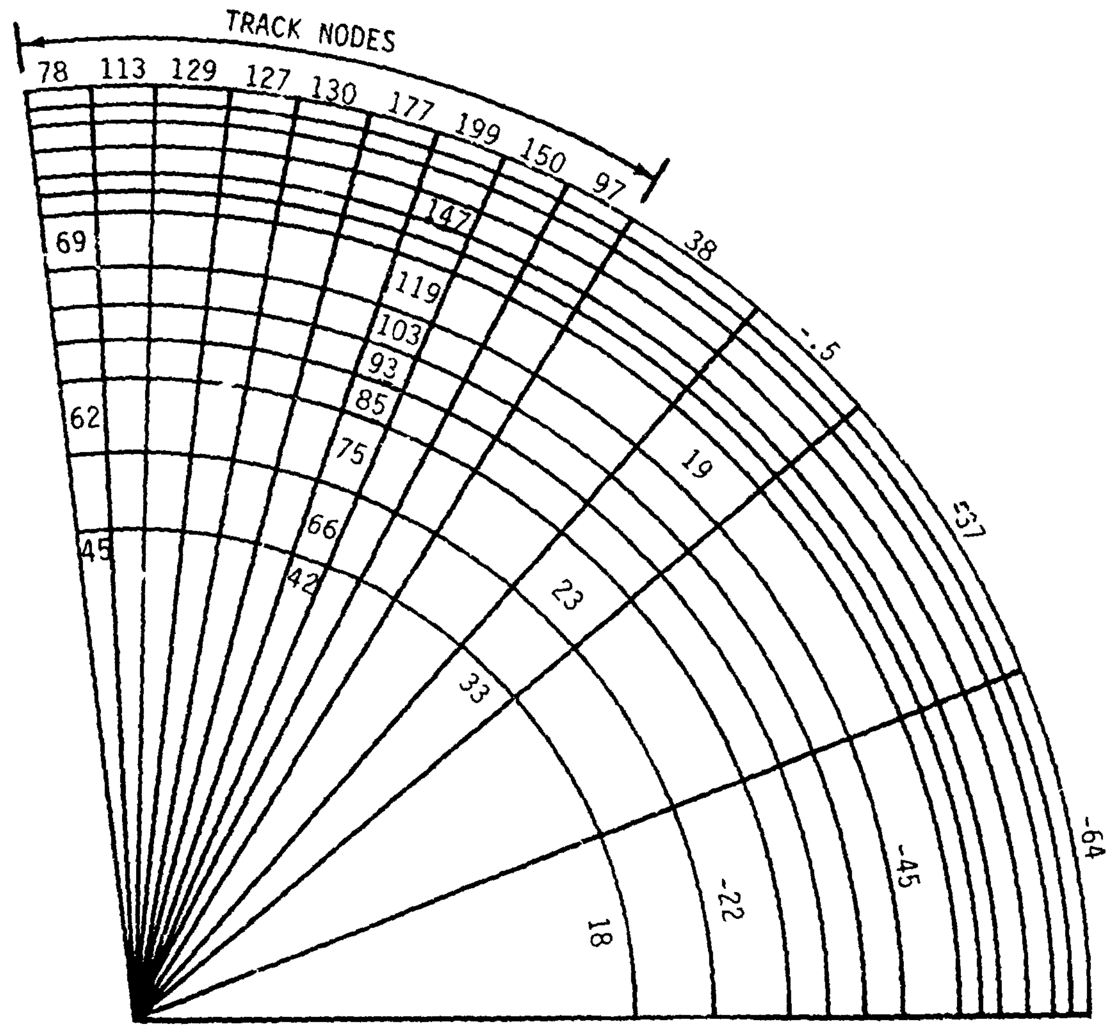


FIGURE 7.4

INNER PACE TEMPERATURE DISTRIBUTION FOR LOX TURBOPUMP BEARING (PUMP END)

(Temperatures in Degrees Farenheit)

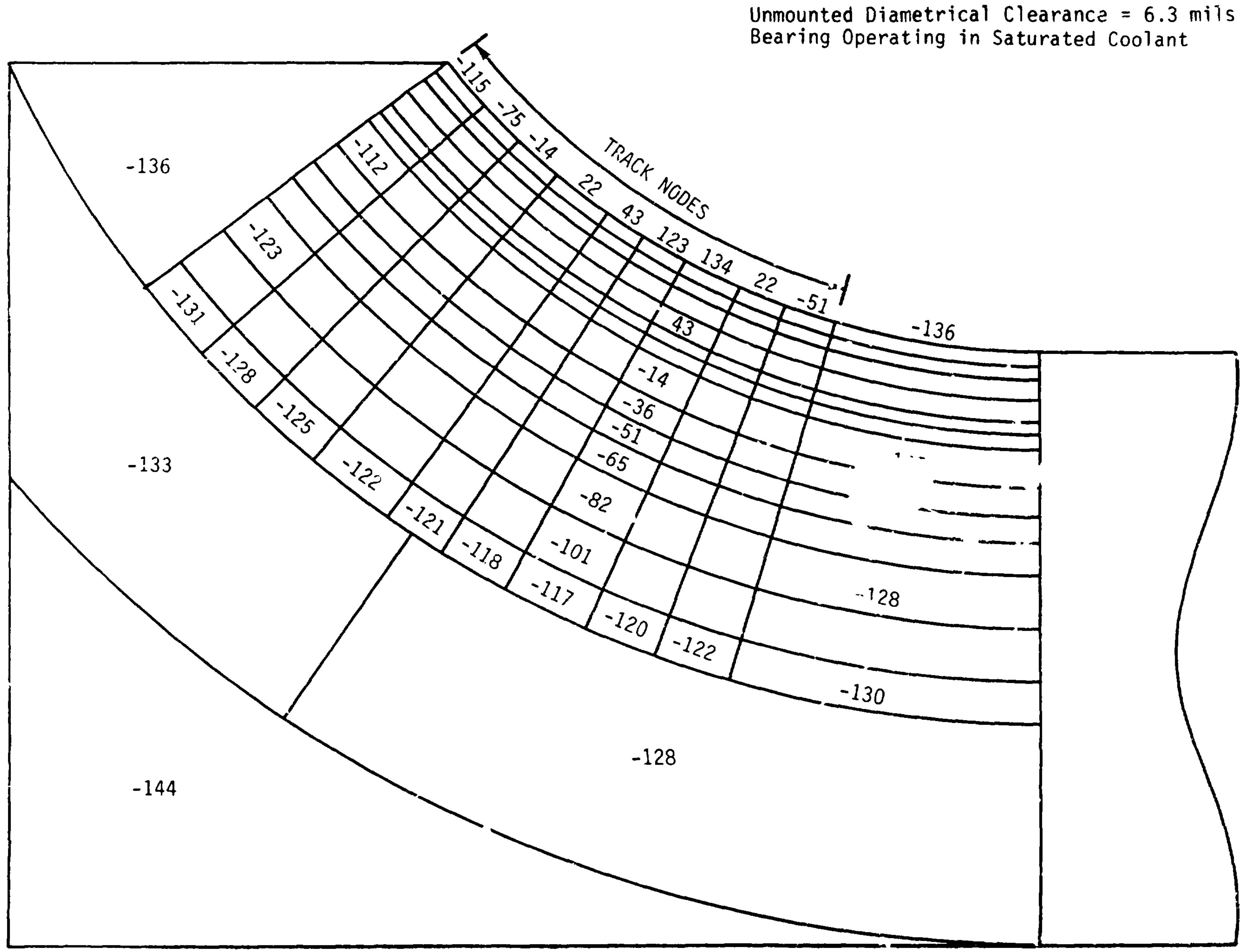
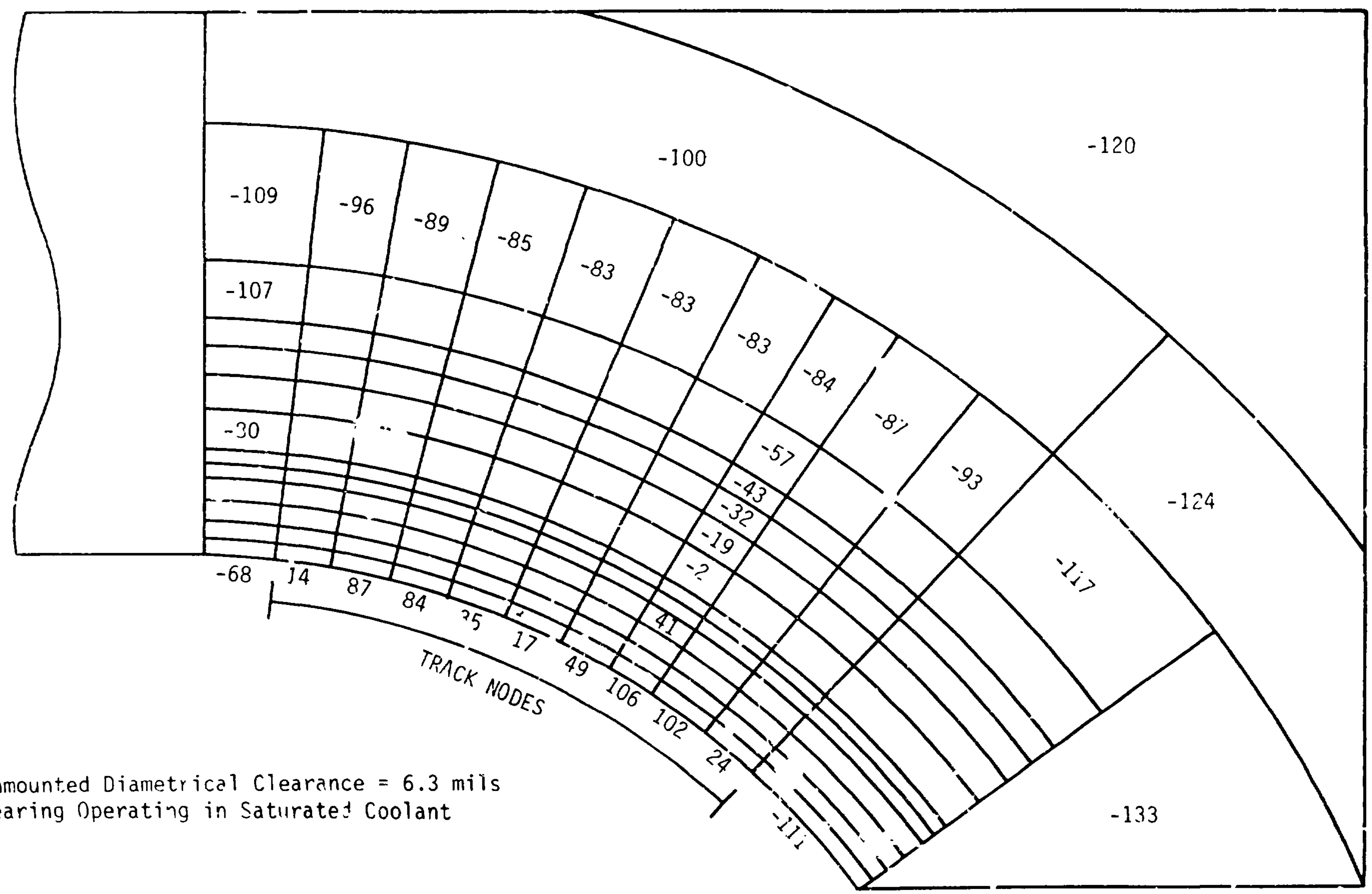


FIGURE 7.5

OUTER RACE TEMPERATURE DISTRIBUTION FOR LOX TURBOPUMP BEARING (PUMP END)

(Temperatures in Degrees Farenheit)



Unmounted Diametrical Clearance = 6.3 mils
Bearing Operating in Saturated Coolant

8.0 RECOMMENDATIONS

The results of this limited analysis used the 57 mm bearing thermal model to estimate thermal conditions for the 45 mm pump end bearing. Although this is adequate for preliminary analysis to define trends and sensitivities, a model of the 45 mm bearing is needed for more detailed analysis.

Bearing load sharing is expected to be significantly influenced by the bearing "suspension system" or preload springs that allow the outer races to move axially to adjust for components of axial loads. Since the current analysis assumed "fixed" outer races, additional analysis should be conducted to investigate this effect on bearing load sharing and operating characteristics.

Only one value of coolant flow rate was investigated. The sensitivity of bearing component temperatures, and operating characteristics to coolant flow should be further developed.

REFERENCES

1) Instrumented High Pressure Oxygen Turbopump (HPOTP) Isolator Load Charts Rockwell International Internal Letter 4128-0231 SSME-84-2140, dated 11/11/84.

2) Bearing Tester Data Compilation, Analysis, and Reporting and Bearing Math Modeling, Final Report; SRS Technologies TR84-022; Contract NAS8-34686, May 1964.