PAD TEMPERATURE IN HIGH SPEED, LIGHTLY LOADED TILTING PAD JOURNAL BEARINGS

by

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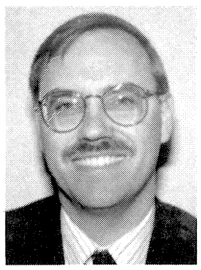

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Dr. Edney received his B.Sc. (1983) and Ph.D. (1990) degrees in Mechanical Engineering from the University of Nottingham, England. He is a member of ASME and the Vibration Institute, and a graduate member of the IMechE. He has coauthored papers on the finite element modelling of rotors, and on the nonlinear response of a flexible rotor-bearing system.

ABSTRACT

The principal results from an extensive experimental investigation of the steady state characteristics of tilting pad journal bearings are presented. The conditions examined were primarily concerned with operation at high speed (>200 ft/sec) and light load (<50 psi specific pressure). The stimulus for this work was the observation of significantly higher than predicted pad temperatures in several steam turbine service applications. The bearings involved were of the conventional pressurized supply (flooded), centrally pivoted, self aligning type, loaded between pad. High temperatures occurred on both four and five inch diameter bearings designed to operate at shaft speeds up to 14,000 rpm.

To investigate this, four and five inch diameter bearings of almost identical design to the service bearings were tested in a laboratory steam turbine at shaft speeds up to 16,000 rpm. The test data acquired show the relationships that exist between the steady state characteristics of pad temperature and power loss, and the parameters of speed, bearing geometry, oil flowrate, and pad material. These results are compared with predicted values from the same tilting pad journal bearing computer program used to evaluate the service bearings.

Of primary interest in the data gathered is the observation of an apparent laminar to turbulent transition region. The results verify the actual behavior of the service bearings and identify limitations of the theoretical model used for predicting steady state performance. It is anticipated that the results presented will be of use to bearing designers, and of general interest to manufacturers of rotating machinery.

INTRODUCTION

The trend towards higher speed for increased performance of turbomachinery has aroused much interest in the subject of bearing influenced rotordynamics. A major contribution to this was the development of the variable geometry tilting pad journal bearing. The inherent stability characteristics of this design of bearing compared with the fixed geometry sleeve type has resulted in its widespread application in modern turbomachinery. Consequently, much research has been dedicated to modelling and advancing its design. This work has been both experimental and theoretical, and ranged from evaluation of steady state performance to dynamic influence on rotor behavior. Nevertheless, with the continuing drive for further increases in operating speed, the application of this type of bearing is reaching a limit whereby, under certain conditions, steady state performance does not match that predicted by established theory.

Background to the Investigation

In several high speed steam turbine service applications utilizing pressurized supply (flooded) tilting pad journal bearings, measured pad temperatures were found to be much higher than predicted. This is of concern to bearing designers, OEMs, and users of turbomachinery, since temperature limits of the babbitt material could be approached and margin against seizure reduced. Thus, a program of work was undertaken to investigate this, identify deficiencies in the theoretical model, and establish design guidelines for future applications.

At the time these high temperature excursions occurred, a program of work was in progress to investigate different tilting pad journal bearing geometries on the dynamic performance of a steam turbine rotor. Since the test bearings were of four and five inches diameter, the program scope was extended to investigate this temperature excursion. The extent of this additional work, however, was limited by the geometry of the bearings procured for the original project scope, which did not permit a detailed evaluation of this problem. Nevertheless, the test data acquired were sufficient to reproduce the high temperatures, and to provide some insight into bearing operation during transition from laminar to turbulent flow.

Literature Perspective

The literature contains a vast amount of information on the design, modelling, and performance of tilting pad journal bearings. Over the years, the traditional pressurized supply design has been extensively studied and is widely reported. Many guidelines and computer programs have resulted from this work, and are generally available to aid design selection and evaluation. Nevertheless, even today, research into this type of bearing is continuing. Last year, two contemporary papers addressing, separately, the design and application of this type of bearing were published by Nicholas [1], and Zeidan and Paquette [2].

In more recent years, new developments in tilting pad journal bearing technology have further broadened their application range. These new developments have primarily focussed on reducing operating bearing temperature and consumed power [3], and reducing tolerance stack up by eliminating the multi piece construction [4]. These new technologies, however, are relatively young, and have comparatively little field experience.

Although a number of papers have been published containing experimental results on turbulent flow transition, these are few in comparison to the many articles on bearings in general. One of the earliest works on turbulent flow in journal bearings was published by Huggins [5] in 1966 on a 24 in diameter plain bearing. Gardner and Ulschmid [6] contributed to this work in 1973 with results on a 19 in diameter sleeve bearing, and a 17 in diameter tilting pad journal bearing. This latter work and other published results on tilting pad journal bearings are summarized in a recent paper by Simmons and Dixon [7]. Herein, additional results are presented from a comprehensive test program designed to complement these earlier studies [6, 8, 9].

All of these works have shown the presence of an inflection in pad temperature with increasing speed. This inflection was attributed to transition from laminar to turbulent flow. These studies, however, were primarily concerned with operation at moderately high bearing loads (specific pressures >150 psi). Here, results are presented for relatively light loads (specific pressures <50 psi) that supplement these published data.

TEST BEARINGS

A typical arrangement of the test bearings is illustrated in Figure 1, which depicts a five pad bearing. The bearings were of the conventional pressurized supply design, consisting of a steel housing supporting center pivoted, babbitt lined pads. The pads were designed with both circumferential and axial alignment capability. Tight clearance floating ring end seals were installed at each end of the housing. Bearing sizes tested were four and five inch diameter, in four and five pad configuration, and 0.5 and 0.75 length to diameter ratio (L/D). Bearing loading was directed between the two lower half pads.

The assembled diametral bearing clearance was obtained by direct measurement using a precision vertical mandrel, horizontal table, and dial indicator. Bump test readings were recorded for every bearing assembly tested at several pad locations. Average values were used in the theoretical calculations.

Pad preload was calculated based on the nominal design pad bore, since it could not be directly measured. In keeping with industry practice, the pads were designed with a positive preload to ensure a converging oil film wedge and hydrodynamic pressure generation. This is particularly important for both lightly loaded and unloaded pads, in the latter case to prevent pad flutter. A preloaded pad will always ensure a converging oil film wedge, even if the journal were to be perfectly centered to the bearing. Pad preload is defined and illustrated by Nicholas [1].

The bearings were designed with separate oil supply and oil drain orifice plugs to control flowrate. These were located in the bearing housing between each pair of neighboring pads. Oil supply was provided by one orifice placed at the axial midposition. Oil drain was controlled using two orifices, one placed near each axial edge of the pad. Their respective locations are identified on Figure 1. The lubricant used was a light turbine oil (equivalent to ISO VG32) with a viscosity of 150 SSU at 100°F.

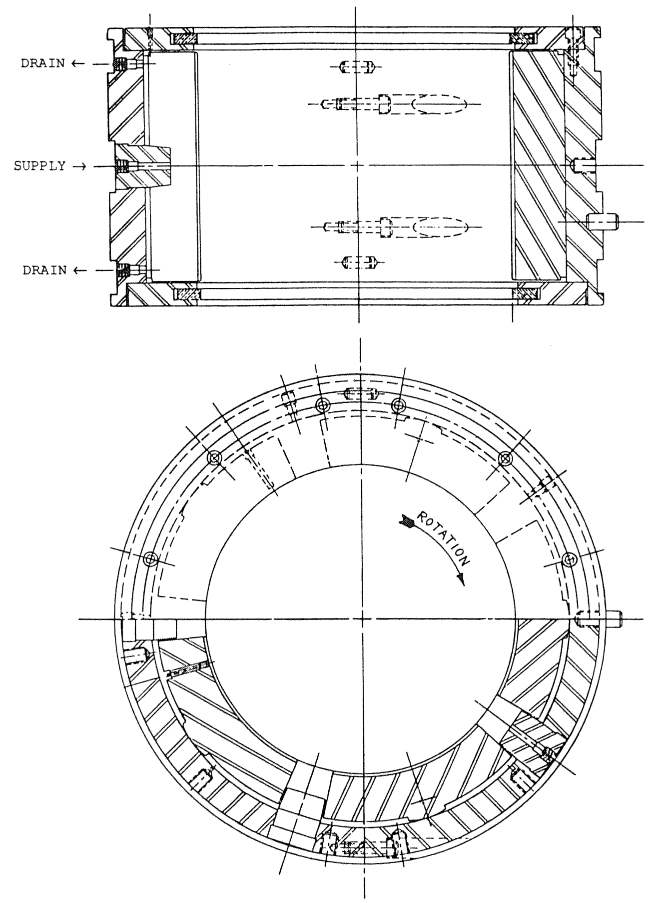

Figure 1. Typical Arrangement of Tilting Pad Journal Test Bearing.

TEST FACILITY

The test vehicle used was a multistage steam turbine capable of operating up to a maximum speed of 16500 rpm. The rotating element was designed to closely simulate a field service rotor.

The rotor was of an integrally forged design with five wheels, a bearing span of 59.98 in, a midspan shaft diameter of 7.25 in, and 5.0 in diameter journals. The first wheel (control stage) only was bladed and steam provided the motive driving force. All other wheels were dummies. The rotor was designed with shaft end flanges and bored overhangs to accept different weights. This was for the purpose of investigating the effect of overhung moment on rotordynamic behavior, and was part of another test project. A general arrangement of the laboratory steam turbine is shown in Figure 2.

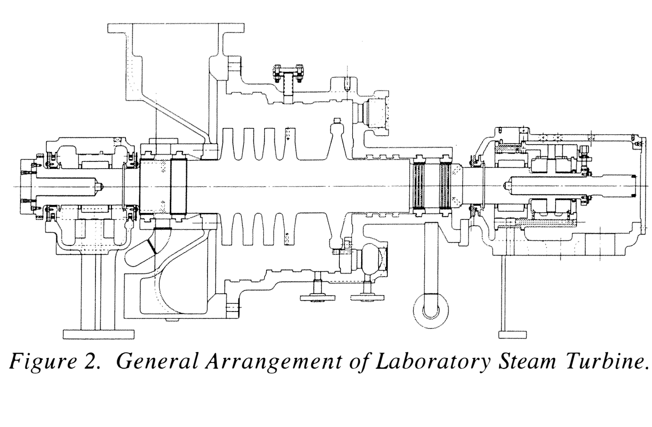

Figure 2. General Arrangement of Laboratory Steam Turbine.

Two series of tests were conducted. The first series was on this base rotor configuration with five inch diameter bearings. The journal diameters were then machined down to four inches, and a second series of tests was conducted.

INSTRUMENTATION

The two lower half (loaded) pads only were instrumented, each with a single thermocouple temperature sensor. The sensors were located at the 75 percent position from the pad leading edge on the axial centerline, in accordance with industry guidelines given in API 670 [10]. This location is generally regarded as corresponding to the vicinity of the maximum pad temperature. The sensors were embedded in the pads to a depth of 50 to 60 thousandths of an inch below the babbitt bond line. The babbitt material was an additional 50 to 60 thousandths of an inch thick.

Thermocouple temperature sensors were also used to measure the oil supply and oil drain temperatures. Turbine type flowmeters were used to measure flowrate. The temperature rise across the bearing and the oil flowrate were used to derive a value for power loss. This was possible only on the exhaust end journal bearing. The steam end journal bearing was located in the same bearing case as the thrust, with a common oil supply and oil drain line.

Eddy current proximity probes were used to measure shaft vibration. These were located 90 degrees apart and axially just inboard of each journal bearing. A key phasor probe was located in the exhaust end bearing case. An electronic governor controlled shaft speed with the reference signal provided by a toothed gear and magnetic pickups.

TEST PROCEDURE

Results from eleven of the bearing configurations tested are presented. A summary of the bearing build data is given in Tables 1 and 2. Each base build is identified by a single alpha character. Variation on a base configuration is identified by a sequential numeral.

Table 1. Build Configurations of the Five Inch Test Bearings.

Build	Figure Number	No Pads	L/D Ratio	Bearing Clearance (mils)	Pad Preload	Specific Pressure (psi)	Pad Material	Oil Flowrate (gpm)
A1	21, 22	5	0.75	6.8	0.409	29.5	STEEL	13.2
B1	3, 4	4	0.75	7.1	0.383	29.5	STEEL	15.7
B2	15, 16	4	0.75	7.1	0.383	29.5	STEEL	8.1
C1	11, 12	4	0.50	7.1	0.398	44.2	STEEL	15.7
D1	5, 6	4	0.75	11.9	0.122	29.5	STEEL	15.7
E1	19, 20	4	0.75	8.2	0.287	29.5	COPPER	15.7

Table 2. Build Configurations of the Four Inch Test Bearings.

Build	Figure Number	No Pads	L/D Ratio	Bearing Clearance (mils)	Pad Preload	Specific Pressure (psi)	Pad Material	Oil Flowrate (gpm)
F1	23, 24	5	0.75	6.6	0.371	44.6	STEEL	13.1
G1	7, 8	4	0.75	6.3	0.405	44.6	STEEL	15.6
G2	17, 18	4	0.75	6.3	0.405	44.6	STEEL	8.1
H1	13, 14	4	0.50	6.0	0.433	66.9	STEEL	15.6
I1	9, 10	4	0.75	9.2	0.268	44.6	STEEL	15.6

All tests were conducted over a speed range from 4000 to 16,000 rpm. The turbine was assembled with the relevant test bearings. The rotor was then run up to a speed of 4000 rpm. This speed was maintained until conditions stabilized. The oil supply temperature was regulated to 120°F at a pressure of 20 psig. The speed was then increased to 16,000 rpm in increments of 1000 rpm. At each increment, two sets of readings were taken: one immediately following a speed increment, and one after pad temperature had stabilized. Data recorded were oil pressure, oil flowrate, and temperature. Temperature measurements were of the four instrumented pads, oil supply, and oil drain.

THEORETICAL RESULTS

The analysis code used for predicting pad temperature and power loss is a modified version of the tilting pad journal bearing computer program PADFEM1 [11]. The theory on which the program is based uses the finite element method to solve the generalized Reynolds' equation for the pressure profiles of individual pads, and a pad assembly method to calculate the assembled bearing characteristics. These values are calculated in dimensionless terms for a given bearing L/D ratio and pad arc length. They are dimensionalized based on the actual bearing geometry, and an effective lubricant viscosity derived for each speed case. This value is derived from an effective temperature, which is determined from a heat and flow balance calculation performed on the loaded pads. This ensures that a realistic value for viscosity is used to determine the bearing stiffness and damping coefficients, which are principally derived from the hydrodynamic pressure profiles of the loaded pads.

The computer program does include a turbulent flow correction option. It is applied only to maximum pad temperature and power loss, and is in the form of two empirical scaling factors. These factors are based on test data from a 17 in diameter tilting pad journal bearing (Gardner and Ulschmid [6]), and are applied only at Reynolds' numbers greater than 1.75 times the critical Reynolds' number. However, these correction factors were found to be inapplicable over the range of parameters investigated. Therefore, this option was not used in the calculation of the theoretical results presented. The values included on the figures are from laminar flow theory.

The calculated values are for a room temperature measurement of the assembled bearing clearance, and the nominal design pad clearance. No change in operating clearance (bearing or pad) was made to account for load and thermal gradients on the individual pads.

EXPERIMENTAL RESULTS

The primary objective of this work was to investigate the effect of shaft speed, oil flowrate, bearing geometry, and pad material, on the steady state performance of lightly loaded tilting pad journal bearings. The steady state characteristics examined were maximum pad temperature and power loss. The geometry variations investigated were assembled bearing clearance, bearing diameter, axial length, and number of pads. Pad materials tested were steel and a chrome-copper alloy.

In the discussion that follows, assembled bearing clearance is referred to as either 'standard' or 'open.' Standard refers to a diametral clearance of nominally 1.5 mil/in of journal diameter. Open refers to an increase in this value to nominally 2.0 mil/in of journal diameter. Also, when referring to the size of the bearing tested, 'four inch bearing' refers to a bearing of four inches journal diameter, and 'five inch bearing' to a bearing of five inches journal diameter.

In the figures given for pad temperature, the terms 'left' and 'right' refer to the pad temperature sensor location. Left refers to the sensor located in the downstream pad, and right to the sensor located in the upstream pad. 'EE' refers to the exhaust end bearing. The orifice size specified on each figure is the diameter of a single orifice. One oil supply and two oil drain orifices are provided per pad.

Bearing Clearance

Effect of bearing clearance on the steady state characteristics of maximum pad temperature and power loss are compared in

Figures 3 and 4 (Build B1), and Figures 5 and 6 (Build D1) for a five inch bearing. Equivalent data are given in Figures 7 and 8 (Build G1), and Figures 9 and 10 (Build I1) for a four inch bearing.

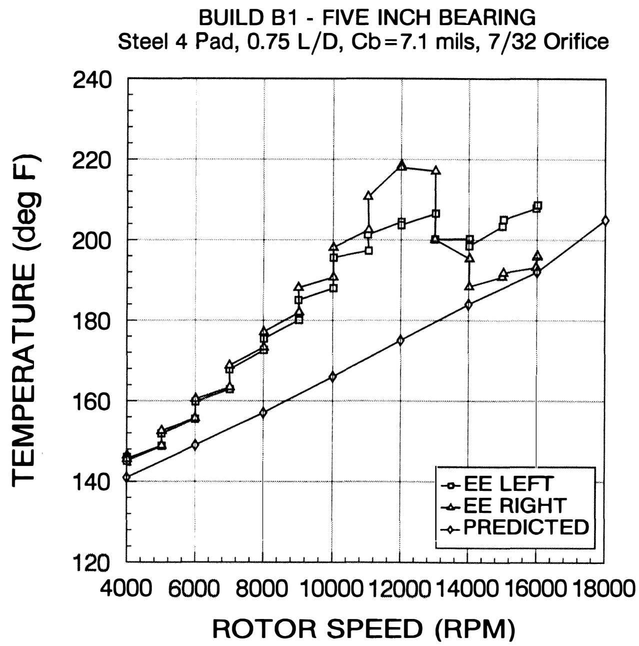

Figure 3. Pad Temperature Vs *Speed for Build B1—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 7.1 Mils, 7/32 In Orifice.*

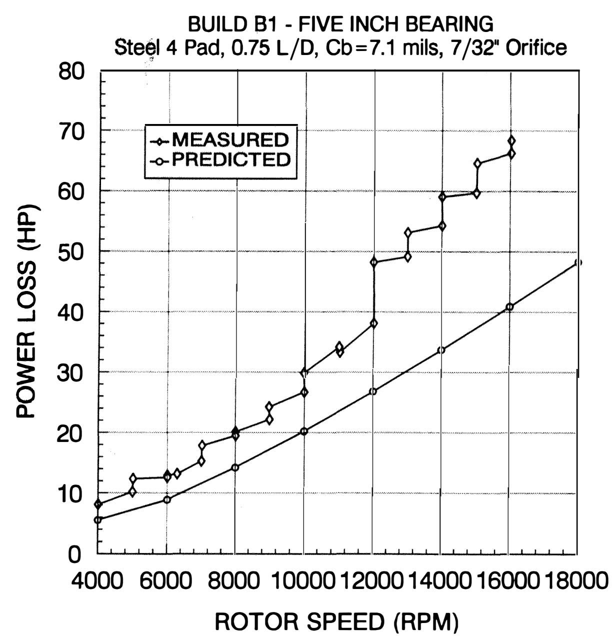

Figure 4. Power Loss Vs *Speed for Build B1—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 7.1 Mils, 7/32 In Orifice.*

The results of both bearing sizes show a significant reduction in pad temperature as clearance is increased. Some decrease in power loss also is apparent. Specifically, at each clearance, the following observations are made.

With standard clearances, the test data are higher than predicted over almost the entire speed range tested. At speeds above

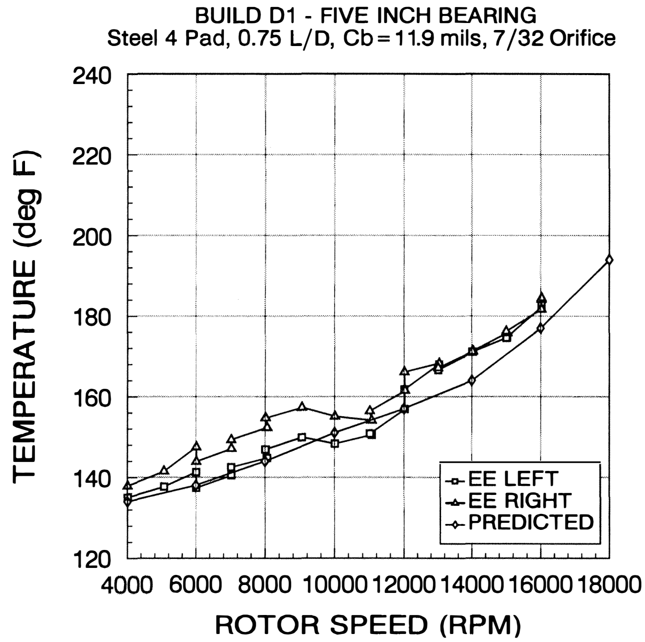

Figure 5. Pad Temperature Vs *Speed for Build D1—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 11.9 Mils, 7/32 In Orifice.*

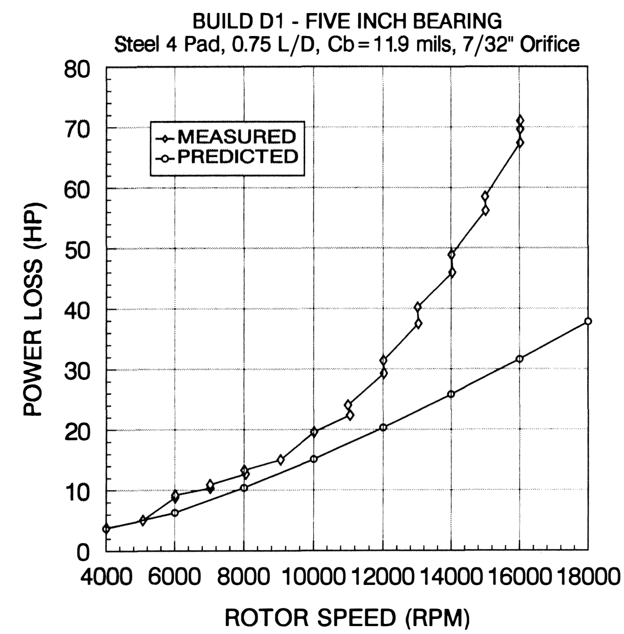

Figure 6. Power Loss Vs *Speed for Build D1—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 11.9 Mils, 7/32 In Orifice.*

6000 rpm, the test data rapidly deviate from theory and exhibit a pronounced temperature peak (or inflection). With the five inch bearing, this peak occurs at about 12,000 rpm and is 50°F higher than predicted. With the four inch bearing, this peak occurs at around 14,000 rpm and is 70°F higher than predicted. This peak is more prominent with the five inch bearing and is followed by a sharp decrease in temperature that stabilizes and then continues to rise. This trend is only partially observed on the four inch bearing, up to the point where the temperature starts to decrease. Speed restrictions limited further investigation of this trend. In contrast, the theoretical results monotonically increase with speed.

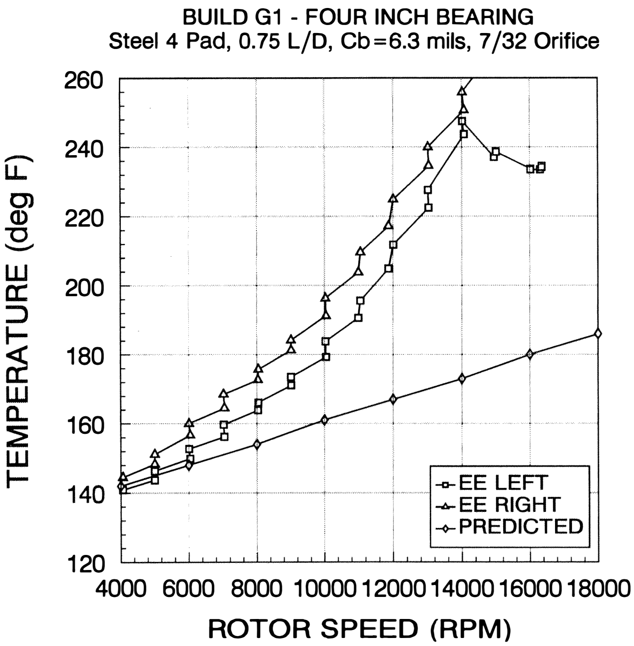

Figure 7. Pad Temperature Vs *Speed for Build G1—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 6.3 Mils, 7/32 In Orifice.*

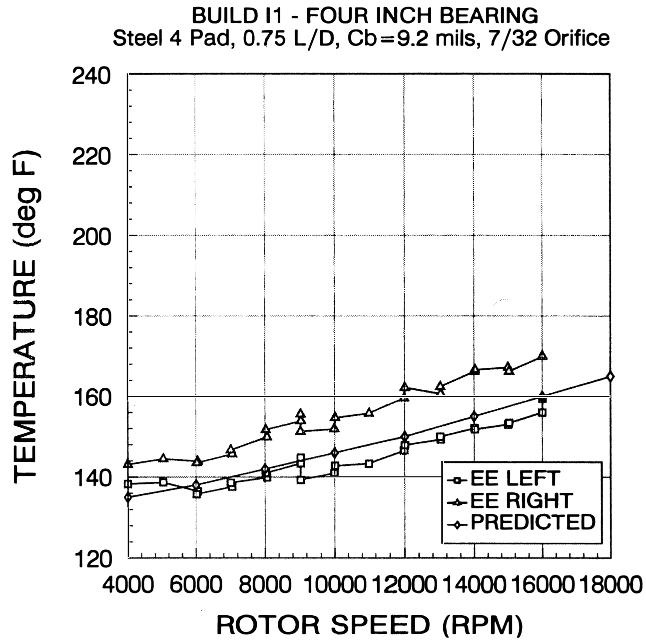

Figure 9. Pad Temperature Vs *Speed for Build I1—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 9.2 Mils, 7/32 In Orifice.*

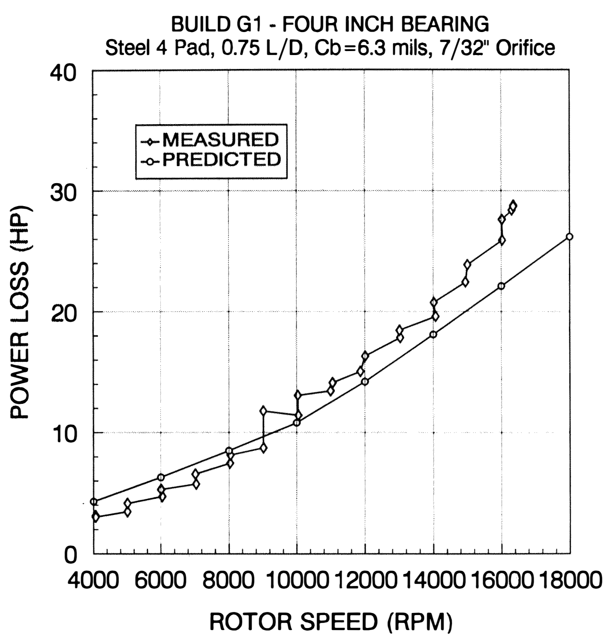

Figure 8. Power Loss Vs *Speed for Build G1—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 6.3 Mils, 7/32 In Orifice.*

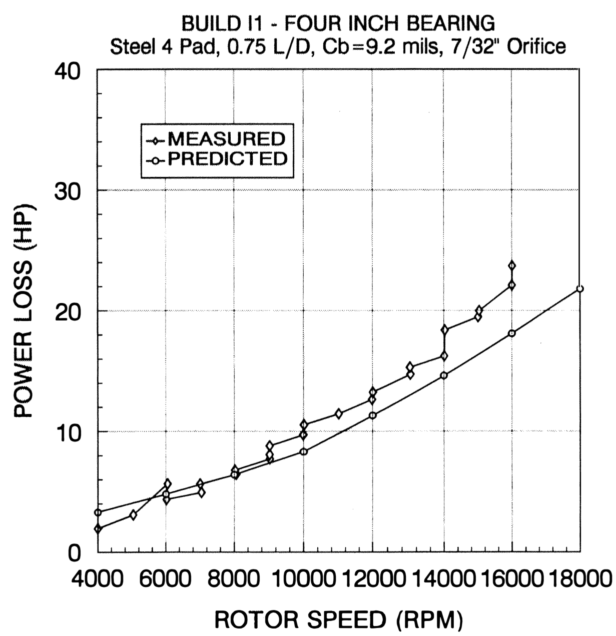

Figure 10. Power Loss Vs *Speed for Build I1—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 9.2 Mils, 7/32 In Orifice.*

With open clearances, the test data for both bearings closely follow theory and are, at worst, higher by no more than 20°F. There is some evidence of a slight temperature peak, particularly with the five inch bearing results, although this could have been influenced by variation in oil supply temperature. The theory does predict some reduction in temperature, although this is not nearly as dramatic as is actually observed.

Regarding power loss, the theory correlates well with the four inch bearing results at both clearances. Nevertheless, some deviation from theory is evident at the higher end of the speed range, at about 14,000 rpm. With the five inch bearings, agreement is good only up to around 12,000 rpm. Beyond this value, power loss increases much more rapidly than predicted. A close inspection of the results shows that this increase starts at a slightly lower speed (11,000 rpm) with the larger clearance. The location of this change in slope is generally regarded as corresponding to the region of transition from laminar to turbulent flow.

Bearing Length

Effect of axial length was investigated by comparing 0.5 and 0.75 L/D ratio bearings. Results for a five inch bearing are given in Figures 3 and 4 (Build B1), and Figures 11 and 12 (Build C1). Results for a four inch bearing are given in Figures 7 and 8 (Build G1), and Figures 13 and 14 (Build H1).

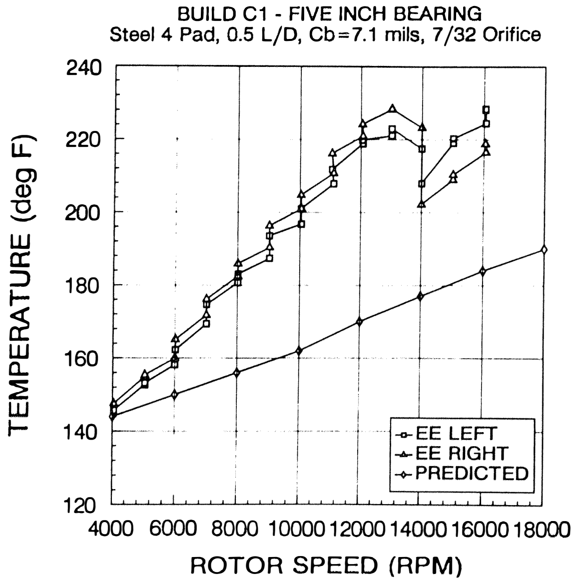

Figure 11. Pad Temperature Vs *Speed for Build C1—Five Inch Bearing, Steel 4 Pad, 0.5 L/D, C_b = 7.1 Mils, 7/32 In Orifice.*

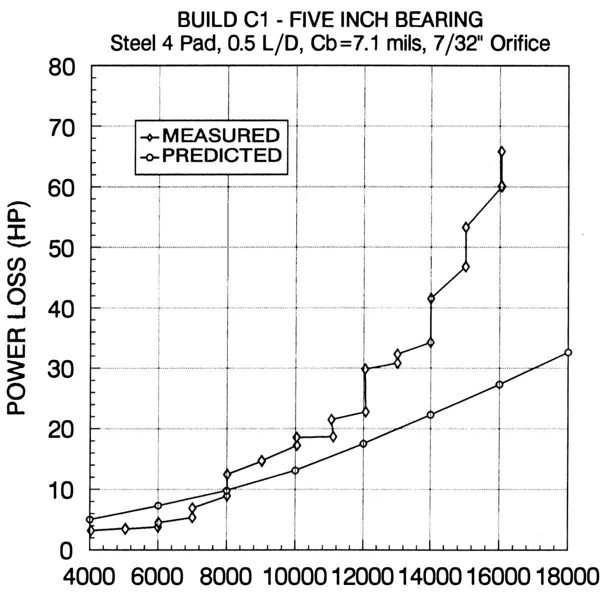

Figure 12. Power Loss Vs *Speed for Build C1—Five Inch Bearing, Steel 4 Pad, 0.5 L/D, C_b = 7.1 Mils, 7/32 In Orifice.*

Both sets of five inch bearing results exhibit similar pad temperature trends with the peak exceeding that predicted by up to 50°F. The four inch bearing results loosely follow a similar trend, although it is questionable as to whether a peak is reached with the 0.5 L/D data. All of these results do, however, deviate significantly from the predicted values. Nevertheless, the observation that at light load, bearing length has little influence on pad temperature is consistent with the theory.

In contrast, both the four and five inch bearing results show some decrease in power loss with reduced bearing length. This

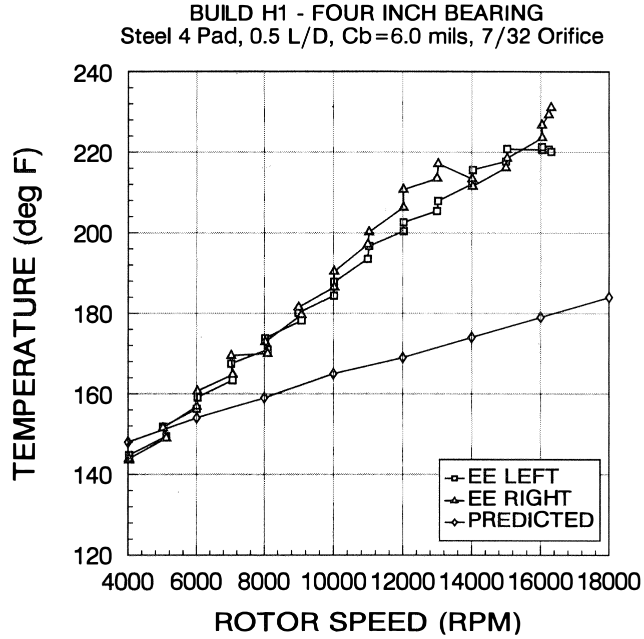

Figure 13. Pad Temperature Vs *Speed for Build H1—Four Inch Bearing, Steel 4 Pad, 0.5 L/D, C_b = 6.0 Mils, 7/32 In Orifice.*

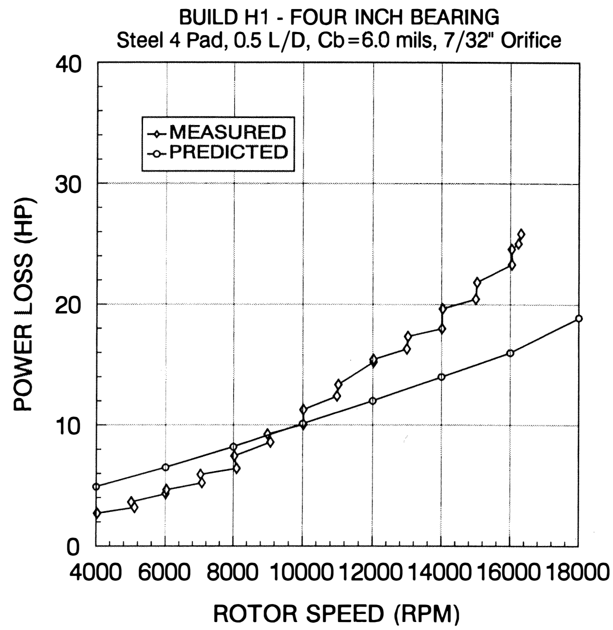

Figure 14. Power Loss Vs *Speed for Build H1—Four Inch Bearing, Steel 4 Pad, 0.5 L/D, C_b = 6.0 Mils, 7/32 In Orifice.*

is primarily due to the difference in bearing surface area and associated frictional losses. The 0.5 L/D five inch bearing data exhibits a similar increase in power loss gradient to that observed on the 0.75 L/D bearing, and at about the same speed. This tends to indicate that transition to turbulent flow is independent of pad area.

Oil Flowrate

Variation in oil flowrate is shown in Figures 3 and 4 (Build B1), and Figures 15 and 16 (Build B2) for a five inch bearing, and in Figures 7 and 8 (Build G1), and Figures 17 and 18 (Build G2) for a four inch bearing.

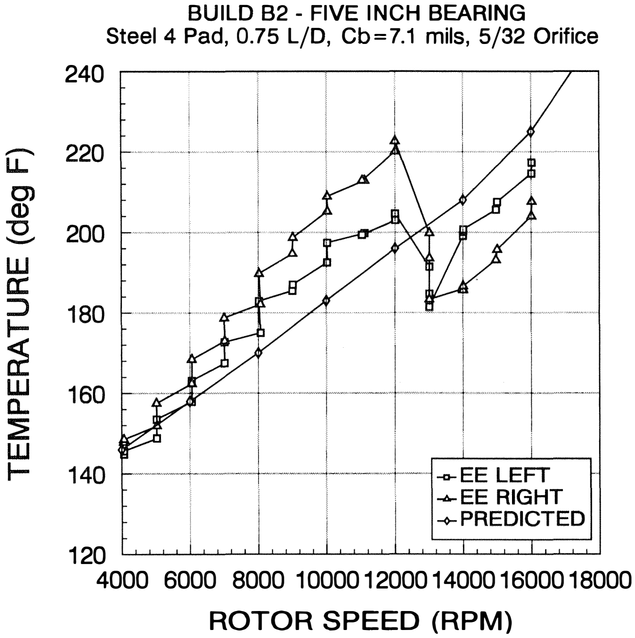

Figure 15. Pad Temperature Vs *Speed for Build B2—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 7.1 Mils, 5/32 In Orifice.*

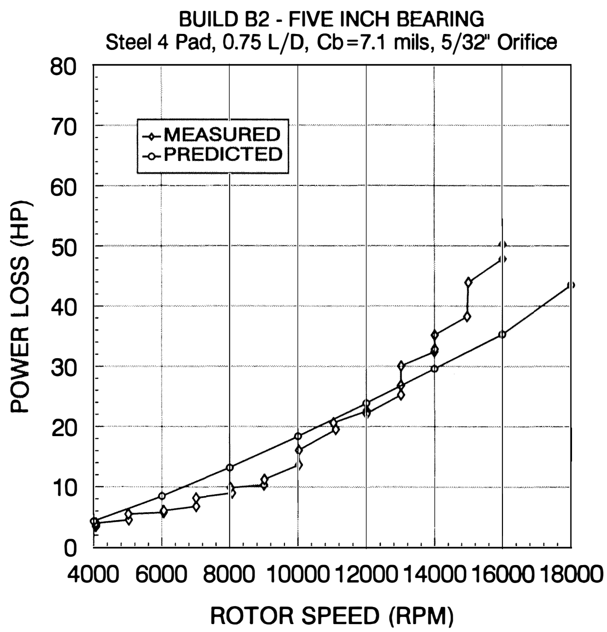

Figure 16. Power Loss Vs *Speed for Build B2—Five Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 7.1 Mils, 5/32 In Orifice.*

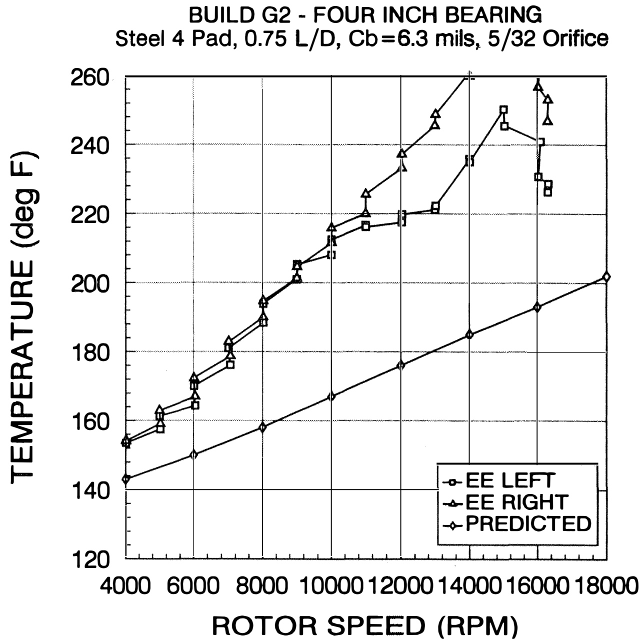

Figure 17. Pad Temperature Vs *Speed for Build G2—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 6.3 Mils, 5/32 In Orifice.*

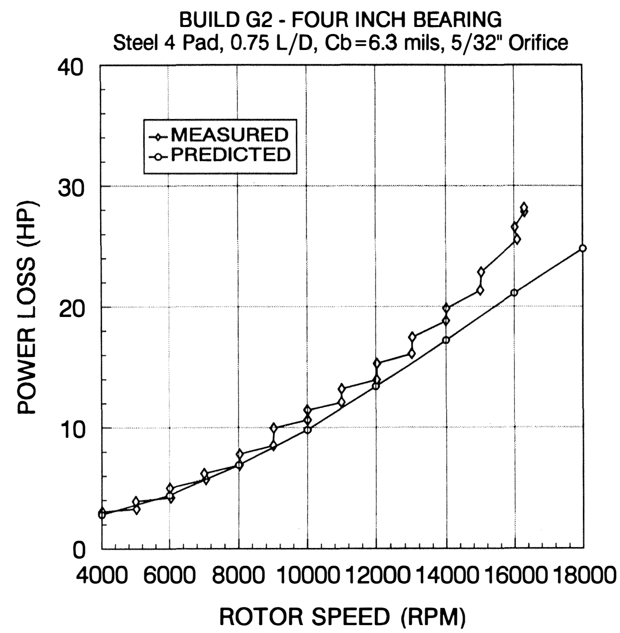

Figure 18. Power Loss Vs *Speed for Build G2—Four Inch Bearing, Steel 4 Pad, 0.75 L/D, C_b = 6.3 Mils, 5/32 In Orifice.*

It can be inferred from the five inch bearing test data that at these light loads, pad temperature is largely independent of oil flowrate. This is because both sets of data follow very similar paths. These results indicate that there could be a limit beyond which increasing the oil flowrate to the bearing has little or no effect on pad temperature. Power loss, however, is noticeably affected by oil flowrate. A considerable reduction in consumed power is evident at the lower flowrate. Correlation with theory also is better. The increase in slope at about 13,000 rpm again indicates transition to turbulent flow.

These trends are not as obvious with the four inch bearings. Some increase in pad temperature is evident as the oil flowrate is reduced, particularly at speeds below the peak. Near the peak, however, the pad temperatures do appear to converge to approximately the same value. The power losses are very similar over the entire speed range. Close inspection of the results, however, does reveal a slight decrease in consumed power with reduced flowrate.

Pad Material

Steel and chrome-copper backed pads are compared for a five inch bearing only. The results are given in Figures 3 and 4 (Build B1), and Figures 19 and 20 (Build E1). The chrome-copper bearing build had a slightly larger assembled bearing clearance.

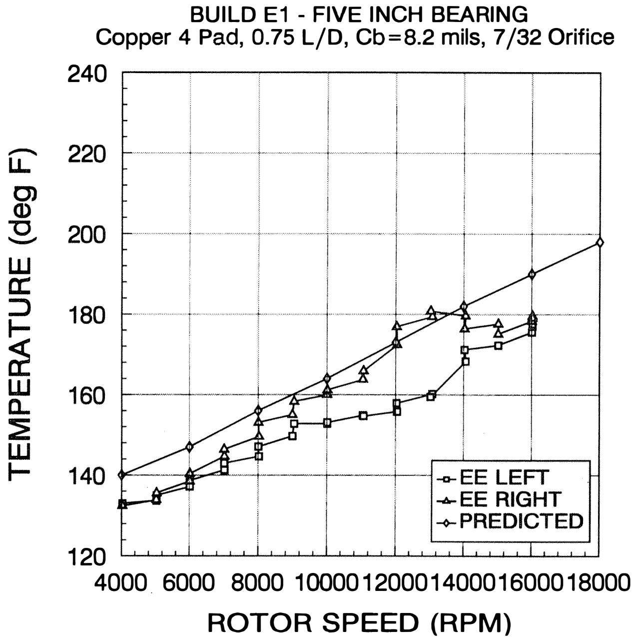

Figure 19. Pad Temperature Vs *Speed for Build E1—Five Inch Bearing, Copper 4 Pad, 0.75 L/D, C_b = 8.2 Mils, 7/32 In Orifice.*

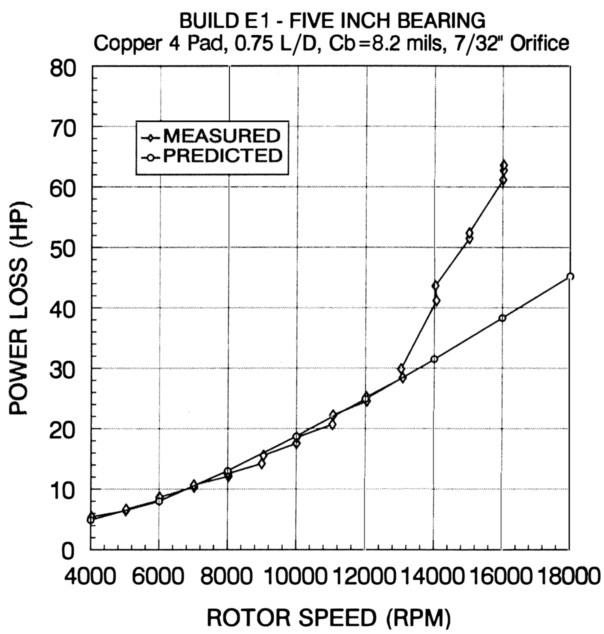

Figure 20. Power Loss Vs *Speed for Build E1—Five Inch Bearing, Copper 4 Pad, 0.75 L/D, C_b = 8.2 Mils, 7/32 In Orifice.*

The chrome-copper test data do not show the severe temperature peak as observed on the steel pads. There is some evidence of a slight inflection, however, at about 13,000 rpm on one pad. Overall values for temperature are lower than for steel by up to 50°F. This difference is primarily due to the higher thermal conductivity of chrome-copper and its ability to remove heat from the oil film more effectively. Although the chrome-copper pad results correlate well with theory, the theoretical model does not allow for pad material conductivity.

Results for power loss with the chrome-copper pads show excellent agreement with theory up to about 13,000 rpm. Beyond this speed, the power consumed by the bearing increases rapidly, thus indicating transition from laminar to turbulent flow, even at a lower overall oil film temperature.

Number of Pads

A direct comparison cannot be made on the effect of number of pads with either the four or five inch bearing results. Due to the initial project scope, four and five pad bearings were procured to meet the original test objectives. No consideration was given to comparing the thermal characteristics of these bearings. Hence, these bearings were designed with different total pad areas and oil supply flowrates.

Nevertheless, the results are of general interest and, therefore, are included. The test data of a five pad bearing arrangement are given in Figures 21 and 22 (Build A1) for a five inch bearing, and in Figures 23 and 24 (Build F1) for a four inch bearing. Similar temperature and power loss trends are observed as with the four pad bearings. In both cases, the power loss data show a change in slope, thus indicating an apparent transition to turbulent flow.

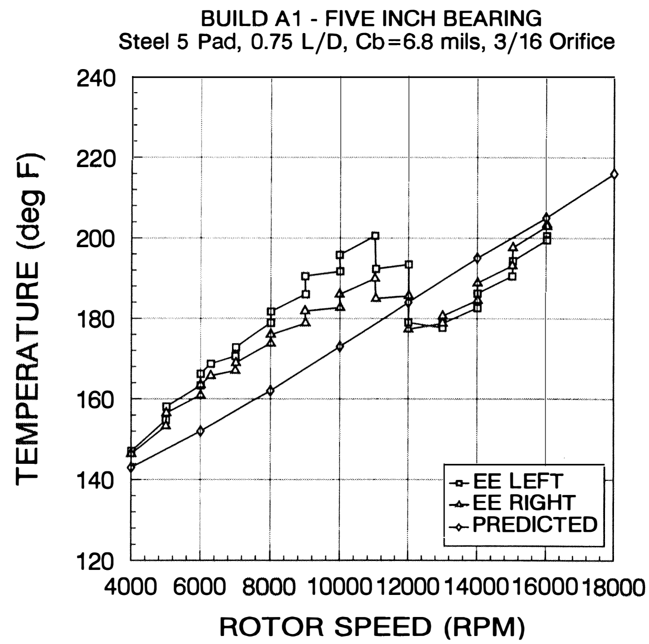

Figure 21. Pad Temperature Vs *Speed for Build A1—Five Inch Bearing, Steel 5 Pad, 0.75 L/D, C_b = 6.8 Mils, 3/16 In Orifice.*

DISCUSSION

Temperature Peak Phenomenon

One observation of particular interest from this investigation is the existence of a temperature peak in pad metal temperature. The location and degree of this peak appears to be dependent on bearing geometry and operating condition. This peak is generally followed by a reduction in temperature, a stabile zone, and then a further rise.

The observed temperature peak is consistent with results from other published works. This peak is generally regarded as corresponding to the region of transition from laminar to turbulent flow. Booser [12] speculates that the decrease in temperature at the onset of turbulent flow is due to an increase in heat transfer from the pads to the cool oil in the supply grooves between adjacent pads. He adds that transition is considered to occur at a value of Reynolds' number above a critical value. This value is termed the critical Reynolds' number, which is defined as:

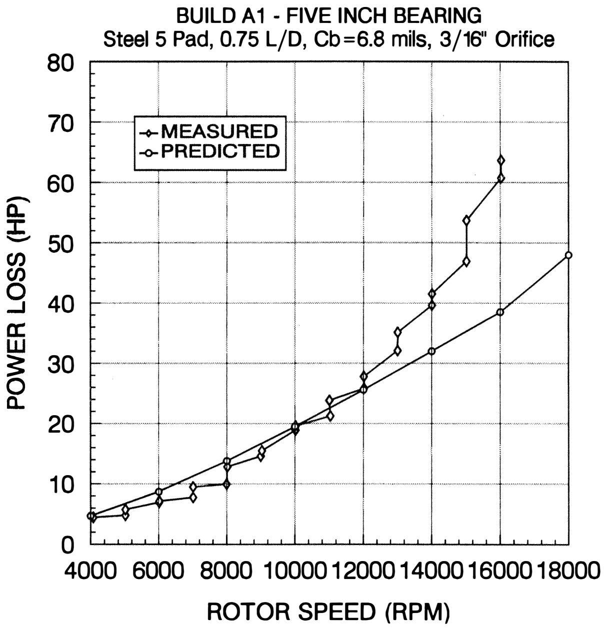

Figure 22. Power Loss Vs *Speed for Build A1—Five Inch Bearing, Steel 5 Pad, 0.75 L/D, C_b = 6.8 Mils, 3/16 In Orifice.*

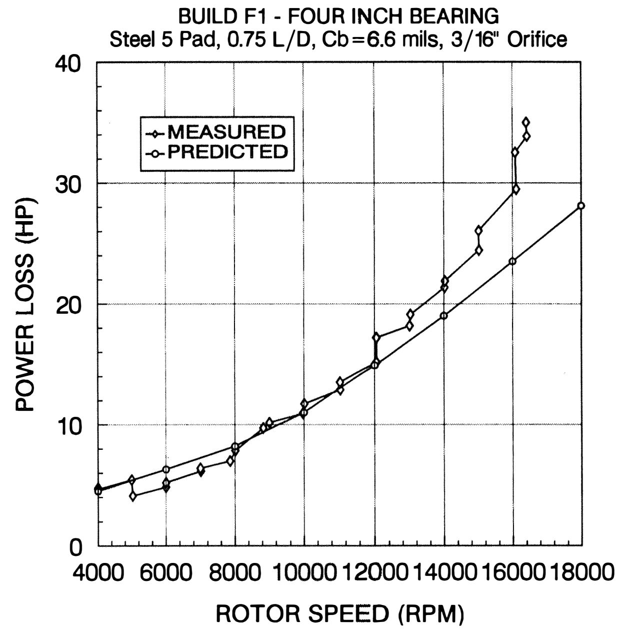

Figure 24. Power Loss Vs *Speed for Build F1—Four Inch Bearing, Steel 5 Pad, 0.75 L/D, C_b = 6.6 Mils, 3/16 In Orifice.*

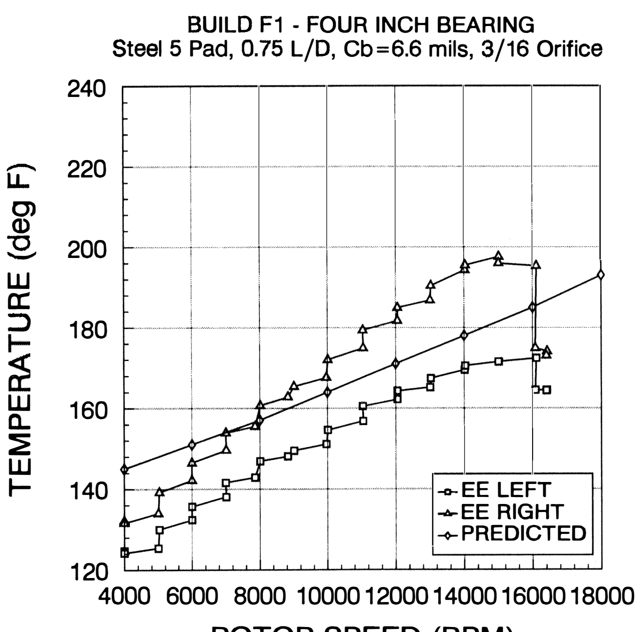

Figure 23. Pad Temperature Vs *Speed for Build F1—Four Inch Bearing, Steel 5 Pad, 0.75 L/D, C_b = 6.6 Mils, 3/16 In Orifice.*

$$Re_c = 41.1\ (R/c)^{0.5}$$

where R is the bearing radius and c is the radial bearing clearance. At Reynolds' number above Re_c, Booser states that the flow instability may be characterized by either Taylor vortices or turbulence. If turbulence occurs first, the value of Reynolds' number above which it is fully developed is generally accepted as 2000. If Taylor vortices develop first, that is when $Re_c < 2000$, fully developed turbulence occurs shortly afterward. However, since turbulence is the dominant instability, most turbulent flow lubrication theories tend to neglect the effect of Taylor vortices. The previous equation shows that the critical Reynolds' number reduces with increasing bearing clearance. This implies that the onset of turbulent flow should occur at a lower value of Reynolds' number, which is defined as:

$$Re = \frac{\pi\ \rho\ c\ R\ N}{30\ \mu}$$

where ρ is the lubricant density, N is the rotational speed, and μ is the lubricant absolute viscosity. Since Reynolds' number increases with bearing clearance, it would seem logical that turbulent flow should occur at a lower value of rotational speed. The observed change in pad temperature, however, could lead one to a different conclusion. With increased clearance, the temperature peak was almost entirely flattened out. Since this inflection is generally regarded as corresponding to the onset of turbulence, this could pose the question of whether such a transition actually occurs. In contrast, however, the observed change in power loss tends to support the theory. The dramatic change in power loss gradient, which also is widely regarded as corresponding to the onset of turbulent flow, is very apparent and does infant occur at a slightly lower value of speed.

This anomaly was also observed and discussed by Simmons and Dixon [7]. They speculate that possibly the transition from laminar to turbulent cooling may occur in the surrounding oil rather than in the hydrodynamic film. The results of this study are too limited to concur with this statement or to provide an alternative explanation. Nevertheless, this is a very interesting phenomenon and one that should be investigated in a rather more controlled manner.

Power Loss at Onset of Turbulent Flow

The five inch bearing results provide better insight into the effect of high speed and light load on power loss. With standard clearances, the test data are generally in good agreement with that predicted up to a point considered to be the laminar to turbulent transition region. At the onset of turbulent flow, the power loss is observed to sharply increase in slope, rapidly diverging from the values predicted using laminar flow theory.

This trend also is evident with the open clearance bearings, but with the region of transition occurring at a slightly lower speed. The total power consumed by the bearing also is slightly reduced.

The chrome-copper pad results exhibit a similar trend to the steel pad results. In this case, however, the change in rate of increase in consumed power occurred at a slightly higher speed than for steel.

Peak Temperature and Pad Location

Although not specifically studied, one observation of noteworthy interest from the results is the location of the pad with the highest measured temperature. Of the two pads instrumented, the pad located in the upstream position (labeled 'right' in the figures) consistently indicated a higher trailing edge temperature. This is contradictory to the generally accepted view that the pad located in the downstream position will operate with a higher oil film temperature due to mixing of hot oil carried over from the upstream pad. Variation in individual pad clearance could account for some of the observed differences, although this would be highly coincidental in virtually every case.

A similar observation also was reported by Brockwell and Kleinbub [13] from tests conducted on a five pad bearing loaded between pad. They presented test data on two L/D ratio bearings at five shaft speeds and identical specific pressure load (300 psi). The smaller (0.46) L/D ratio bearing exhibited slightly higher upstream pad temperatures, and the larger (0.75) L/D bearing noticeably higher downstream temperatures. These measurements were recorded at the 65 percent position on each pad.

A detailed study and explanation of this observation is beyond the scope of this work. Nevertheless, from measurement of shaft locus, Brockwell and Kleinbub [13] observed the equilibrium position of the journal to deviate from the vertical axis. At low eccentricities, the shaft locus was observed to deviate, although small, from the vertical axis in a direction consistent with that in a plain sleeve bearing (i.e. with rotation). At high eccentricities, however, the shaft locus was observed to deviate to the opposite side of the vertical axis. Deviation consistent with that in a plain sleeve bearing also was observed by Tripp and Murphy [14] on a five pad bearing loaded on pad.

From these observations, it is speculated that the operating position of the journal within the bearing is the controlling factor on pad temperature. Since the journal equilibrium position deviates from the vertical axis, the load on the two lower half pads will not be distributed equally, and thus both pads will operate at slightly different attitude angles with respect to the journal. Consequently, the magnitude and angular location of the maximum oil film temperature on each pad will be different, with the peak likely ahead of or behind the instrumented position. The measured temperature, therefore, is likely not to be an indication of the true maximum oil film temperature. This could account for some of the observed differences.

Accuracy of Theoretical Model

In general, the theoretical results did not compare favorably with the test data for both pad temperature and power loss. Some improvement in correlation for pad temperature only was evident at the larger clearance values. The primary reason for this is that the calculated values were based on laminar flow theory. Although the computer program used to derive these data does include a turbulent flow correction option, it was found to be inapplicable over the range of parameters investigated. A principal objective of this investigation, however, was to generate test data from which a more applicable empirical scaling factor could be derived.

The turbulent flow Reynolds' equation is only moderately more difficult to solve numerically than the laminar model. It was originally developed by Ng [15], and Ng and Pan [16], and is based on eddy viscosity effects. However, a computer program based on a turbulent lubrication theory that yields maximum pad temperature and power loss was not available to the author. Thus, no comparison with a turbulent model for predicting pad temperature and power loss was possible.

CONCLUSIONS

The test data, although limited in some areas, are sufficient to show the general relationships that exist between the steady state characteristics of maximum pad temperature and power loss, and the parameters of shaft speed, oil flowrate, and bearing geometry. The significant conclusions of this investigation are as follows:

- Under conditions of light load and high speed, bearing clearance is the primary factor affecting pad temperature. Increasing bearing clearance can significantly reduce pad temperature.

- With certain bearing geometries and operating conditions, a distinct peak was observed in the relationship between pad temperature and shaft speed. Beyond this peak, the temperature decreased, stabilized, and then continued to rise. This phenomenon is characteristic of the reported behavior that occurs during transition from laminar to turbulent flow. This peak is observed to be less dramatic as bearing clearance increases, to the point where it almost entirely flattens out.

- At the onset of turbulent flow, power loss increases rapidly. With standard bearing clearances, this increase coincided with the observed pad temperature peak. However, a sharp increase in power loss also was observed with open bearing clearances, even though a temperature inflection was hardly noticeable. It is speculated that at larger bearing clearances transition does occur, as indicated by a lower value of critical Reynolds' number, but that turbulent cooling is highly effective in reducing peak operating temperature. Whether this cooling occurs in the oil film or in the surrounding oil cannot be concluded from this work.

- There is a significant increase in power loss at shaft speeds above the turbulent regime onset speed. This increase in loss could be of concern to designers of high performance turbomachinery where efficiency is marginal. This observation also is consistent with other experimental research reported in the literature.

- Chrome-copper backed pads operate with significantly lower pad temperatures than steel backed pads. A difference of up to 50°F was observed in the test data presented. With standard bearing clearances, the chrome-copper pads did not exhibit the dramatic temperature peak as observed with the steel pads. However, a speed was reached beyond which the power loss gradient sharply rose, thus indicating probable transition to turbulent flow, even at a lower overall oil film temperature.

- The limited data available indicate that, for a given set of conditions, there is an optimum flowrate to give the lowest maximum pad temperature. Increasing the flowrate beyond this optimum increases the power consumed by the bearing without reducing pad temperature.

- Shortening the bearing axial length reduces the consumed power. However, at the light loads investigated, this has little effect on pad temperature.

- At standard bearing clearances, laminar flow theory under predicts maximum pad temperature and power loss at speeds approaching and into the turbulent regime. As bearing clearance is increased, correlation with maximum pad temperature only is improved.

NOMENCLATURE

c bearing radial clearance (in)
D journal diameter (in)
L bearing axial length (in)
N journal rotational speed (rpm)
R journal radius (in)
Re Reynolds' number (dim)
Re_c critical Reynolds' number (dim)
μ oil viscosity (lbf-s/in^2)
ρ oil density (lbf-s^2/in^4)

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ACKNOWLEDGEMENTS

The author would like to thank Dresser-Rand Engineering Laboratory Staff for building and operating the test turbine, John Waite for coordinating the testing and reducing much of the data, and Jatin Katariya for reviewing the manuscript. Thanks also are extended to John Nicholas at Rotating Machinery Technology, Inc., for technical insight and discussion on this work. Finally, he would like to thank the Directors of Dresser-Rand Steam Turbine Division for permission to publish the material contained in this paper.