11th Australasian Fluid Mechanics Conference
University of Tasmania, Hobart, Australia
14-18 December 1992

STEADY STATE PERFORMANCE OF A TRI-POCKET JOURNAL BEARING

Douglas HARGREAVES and B.M. REID

School of Mechanical & Manufacturing Engineering
Queensland University of Technology
GPO Box 2434, Brisbane, QLD 4001, AUSTRALIA

ABSTRACT

This investigation was undertaken as a result of the tri-pocket journal bearing in a high speed gearbox running 'hot' under normal operating conditions and that the bearing temperature increased markedly when the production rate was increased. The main finding was that the manufacturing tolerances play an important role in determining the bearing performance. Indeed, it was shown that the operating eccentricity can vary from 0.84 to 0.91 depending on the combinations of geometrical dimensions used. The other factor found to be important in the bearing performance prediction was the direction of the applied load in relation to the lubricant supply grooving arrangements. A change of 10° made a 250% change in the load carrying capacity.

NOMENCLATURE

B	length of bearing
C_p	specific heat capacity of the lubricant
C_R	radial clearance
D	mean bearing diameter
E_b	power loss due to viscous friction on the bearing bush
e	eccentricity the distance from shaft centre to bearing centre in the operating condition
F_b	frictional force acting on the bearing bush
h	lubricant film thickness
M	rotor mass per bearing
N_s	journal rotational speed (revs/sec)
p	lubricant pressure
Q	lubricant flow rate from bearing edges, (side leakage)
R	journal radius
Ramp	depth of the tri-taper ramp
T_b	torque due to viscous friction acting on the bearing bush
U	peripheral velocity of shaft (= $R\omega$)
W	total bearing load
W_x	load acting along the line of centres
W_y	load acting normal to the line of centres
ΔT	temperature rise of the lubricant
η	dynamic viscosity of lubricant
θ,y	circumferential and axial co-ordinate directions
ϵ	eccentricity ratio (e/C_R)
Ψ	attitude angle - angle through which the load vector needs to be rotated in the direction of shaft rotation to bring it into the line of centres
ω	angular velocity of journal
M_c^*	dimensionless critical journal mass $\left(\dfrac{M\,C_R\,\omega^2}{W}\right)$
$\bar{h}$	normalised film thickness ratio, (h/C_R)
$\bar{p}$	normalised lubricant pressure, ($p\,C_R^{\,2}/\eta UR$)
Ramp*	normalised depth of the tri-taper pocket ($Ramp/C_R$)
S	Sommerfeld No $\left(\dfrac{\eta\,N_s\,D\,B}{W}\left(\dfrac{R}{C_R}\right)^2\right)$
$\bar{W}$	normalised total load-carrying capacity $\left(\dfrac{W}{\eta UB}\left(\dfrac{C_R}{R}\right)^2\right)$
$\bar{y}$	normalised co-ordinate direction (y/B)

INTRODUCTION

The work undertaken, the results of which are presented here, was initiated from a local industrial situation. The temperature of one of the bearings in a high speed step-up gearbox was considered to be close to the threshold limit for the bearing material. The gearbox increases shaft speed about three times from a reaction turbine to a three-stage centrifugal gas compressor. The problem bearing was on the output end of the output shaft. Thermocouples have been imbedded into the bearing material on the bearing centreline and near the bearing edge each at a circumferential position corresponding to the perceived location of the minimum lubricant film thickness.

This gearbox and hence the problem bearing is a critical component of the whole production process, that is, if the bearing temperature increases beyond a preset value, the gearbox and the

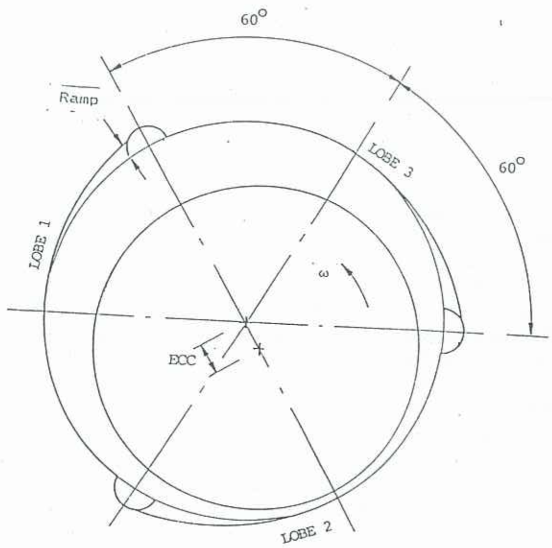

Figure 1
Tri-pocket Journal Bearing

whole plant are shut down. The bearing has been operating successfully for a number of years albeit while under the close scrutiny of maintenance engineers. The possibility of requiring that the plant operate at increased production rates precipitated a more detailed investigation into the bearing performance.

The original gearbox design incorporated two relatively small plain journal bearings on each end of the output (or high speed) shaft. Some years ago, again when production rates were increased, these bearings were completely redesigned. A single tri-taper journal bearing was manufactured and fitted to each end of the output shaft. This type of bearing is similar to a three-lobed bearing as is shown in Figure 1. There are three lubricant supply grooves equi-spaced around the bearing periphery. Each of the three sections consists of equal lengths of the bearing base circle and a section offset from the base circle such that it merges with the base circle and provides a predetermined depth at or near to the lubricant supply groove. This type or arrangement is akin to the composite thrust bearing shown in Figure 2.

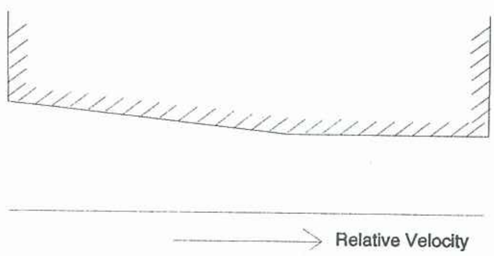

Figure 2
Schematic arrangement of a tapered land composite bearing

The reason for the selected bearing geometry is not known but may be related to the possibility of bearing induced instability. Profile journal bearings such as offset halves and three-lobe bearings are commonly employed to suppress such instabilities.

The objectives of the work undertaken were to

(i) analyse the existing bearing in terms of performance
(ii) identify any actions which may be taken to improve bearing performance.

PRELIMINARY BEARING ANALYSIS

Historical plant records were studied in order to establish relationships between, for example, production rate and measured bearing temperatures and shaft speeds. The magnitude and direction of action of the bearing load were determined as well as an effective or operating lubricant viscosity. Engineering drawings provided details of the bearing. It is pertinent to note here that the conventional geometrical parameters for profiled journal bearings of offset, preload and tilt angle cannot be used for this particular bearing. Definitions of these terms can be found in Abdul-Wahed et al (1982).

A stability analysis was performed based on the criteria presented in Abdul-Wahed et al (1982) and Li (1980). The dimensionless critical journal mass (M_c^*) and the Sommerfeld No (S) were calculated and plotted onto the stability map of Abdul-Wahed et al (1982). From Figure 3, it is clear that the bearing in question does not have any potential stability problems.

A preliminary bearing performance analysis was undertaken using the Engineering Sciences Data Unit (ESDU) Item Nos 84031 and 85028. These design procedures accommodate only circular journal bearings and lubricant supplied from a single axial groove but proved to be instructive especially in relation to whether the bearing would operate in the laminar or turbulent lubrication mode. Calculations were inconclusive in that they showed that the bearing may operate in either mode and this depended to a large degree on what value of radial clearance (C_R) was chosen.

It was decided to write a computer program to obtain a more exact bearing analysis.

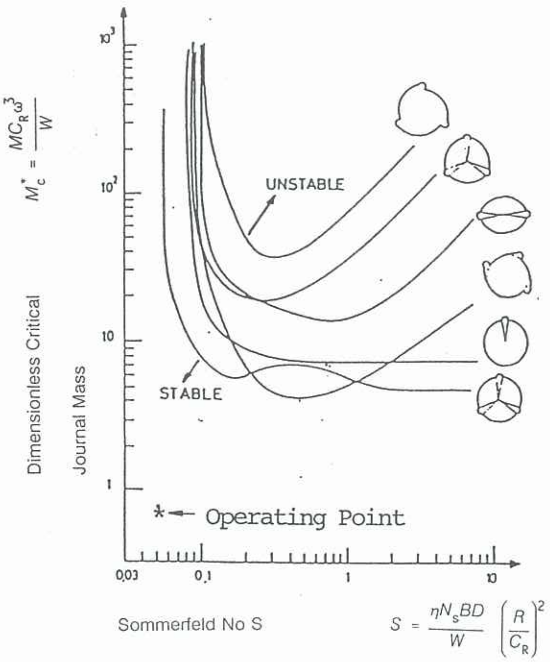

Figure 3
Linearised Bearing Stability Map

THEORY AND COMPUTATIONAL TECHNIQUES

A computer program was developed to determine the performance characteristics of this tri-taper journal bearing, the geometry for which is shown in Figure 1. The appropriate form of Reynolds equation suitably normalised and incorporating the assumptions of the laminar flow in the isoviscous lubricant film is

$$\frac{\partial}{\partial\theta}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\theta}\right) + \left(\frac{R}{B}\right)^2\frac{\partial}{\partial\bar{y}}\left(h^{-3}\frac{\partial\bar{p}}{\partial\bar{y}}\right) = 6\frac{\partial\bar{h}}{\partial\theta} \qquad (1)$$

Pressures were set to zero along the bearing edges and to the normalised supply pressure over each of the three grooves.

The film profile was determined by superimposing the effect of the ramp on the conventional expression for film thickness in a journal bearing ie

$$\bar{h} = 1 + \epsilon\cos\theta + \text{Ramp}^* \qquad (2)$$

where (Ramp*) was a function of the circumferential co-ordinate.

A finite difference approximation scheme was adopted with a solution of the resulting set of simultaneous equations being obtained by a Gauss-Seidel iterative method incorporating successive over-relaxation.

Cavitation was allowed to occur at ambient pressures by setting all calculated negative pressures equal to zero throughout the iterative solution scheme. This implies that the lubricant film ruptures when $\bar{p} = \partial\bar{p}/\partial\theta = 0$.

Having obtained the pressure distribution throughout the bearing, the load-carrying capacity and the attitude angle were evaluated. In practice the orientation of the load direction and groove location was known. To initiate the program, the location of the line of centres in relation to the groove location had to be estimated. The calculated attitude angle was therefore based on this estimation. If the calculated and the initially assumed attitude angles were different, then the program had to adjust the original estimate and recalculate. This iterative scheme proceeded until the calculated and assumed values agreed within a preset tolerance; two degrees of arc were found to be acceptable.

Once the location of the journal in the bearing bush had been established, for the assumed operating eccentricity ratio, the following performance parameters could be determined.

- load-carrying capacity
- lubricant flow rate from the bearing edges
- frictional force (due to viscous shearing of the lubricant on the bearing bush)
- torque on the bearing bush
- power loss (due to viscous shearing of the lubricant on the bearing bush
- average temperature rise of the lubricant (assuming all heat generated is carried away by the lubricant)

THEORETICAL PREDICTIONS

Operating Envelopes

Early in this investigation, it became clear that manufacturing tolerances played a significant role in the performance predictions, in particular the radial clearance (C_R) and the depth of the tri-taper ramp (Ramp*). For each of the maximum and minimum values of (C_R), there would be a corresponding set of (Ramp*) values. For each of these conditions, an operating envelope was developed.

The operating envelopes for each of 90%, 100% 120% and 125% production rate were plotted onto a common set of axes and is shown here as Figure 4.

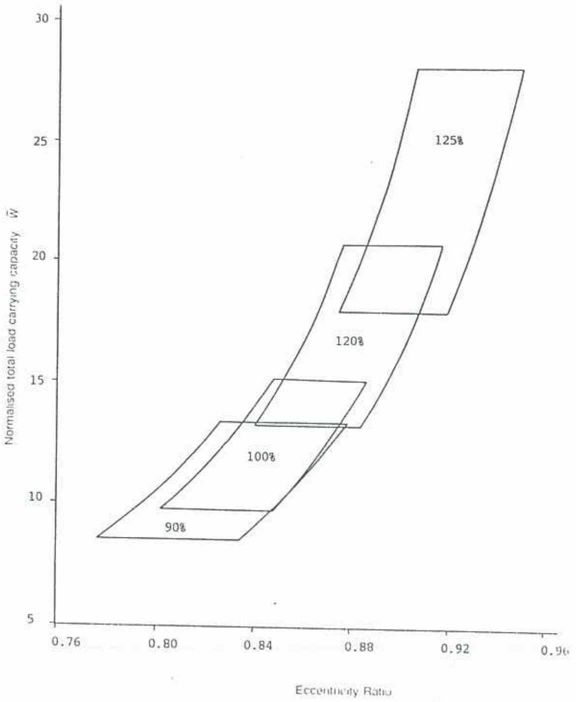

Figure 4
Operating Envelopes for 90%, 100%, 120% and 125% production rates

Pressure Profile

The calculated pressure profile for the 125% production rate is shown in Figure 5. The four profiles correspond to the four extremities of the operating envelope. As expected, the pressure achieves significant positive values only in one of the three tapered sections of the bearing. This implies that the total load-carrying capacity is derived from only one-third of the bearing circumference.

Analysis of Variations

The data used in the development of the overall analysis were taken from recorded values in the field. It was not known how accurate some of the data were so a series of predictions were made to vary each of a number of parameters in turn.

A ten percent increase in the magnitude of the applied load at the 125% production rate showed that the maximum bearing operating eccentricity increased from $\epsilon = 0.95$ to $\epsilon = 0.96$.

A five percent increase in the effective lubricant temperature and the consequent change in effective lubricant viscosity increased the maximum operating eccentricity beyond 0.96 for the 125% production rate.

Changes to the lubricant supply pressure had an insignificant effect on the load-carrying capacity but the total side leakage flow was greatly affected.

The importance of the applied load direction on multi-lobe profile bore journal bearings has been highlighted by Gethin et al (1990). The bearing examined in this study has the angle between the load line and the first lubricant supply groove at 125°. Figure 6 shows that the load carrying capacity can be significantly increased by changing this angle to 105°. The increased load capacity results in a reduction in eccentricity ratio from 0.915 to 0.855 and a consequent reduction of about 25% in the temperature increase for the mean 125% production rate test conditions.

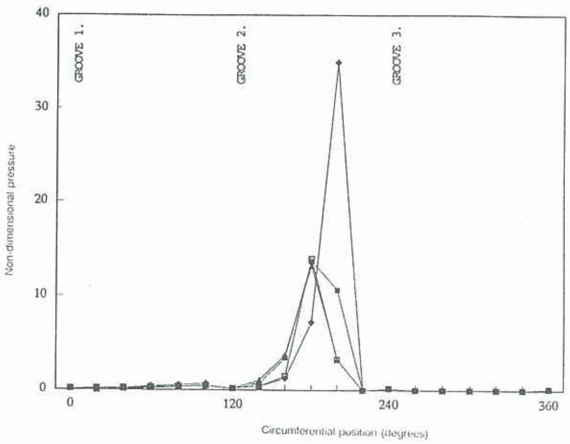

Figure 5
Pressure Profiles (125% Production Rate)

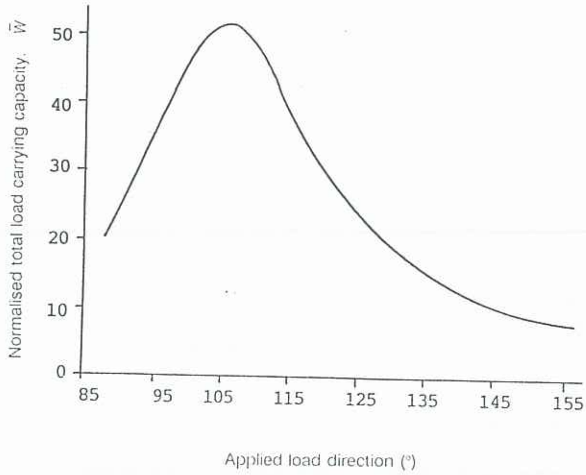

Figure 6
Bearing Load Capacity versus Applied Load Direction

Maximum side leakage flow rate was predicted when the value of (Ramp*) was 0.5. Values both greater than and less than 0.5 produced lower flow rates and hence increased temperature rises within the bearing.

Comparison With a Three-Groove Journal Bearing

A plain circular three-groove journal bearing of equivalent dimensions was modelled for comparison. Figure 7 compares the load carrying capacity, side leakage and the temperature increase. The tri-taper bearing has a lower load-carrying capacity, larger side leakage and hence smaller temperature increases in the bearing. These are all consistent with expected results.

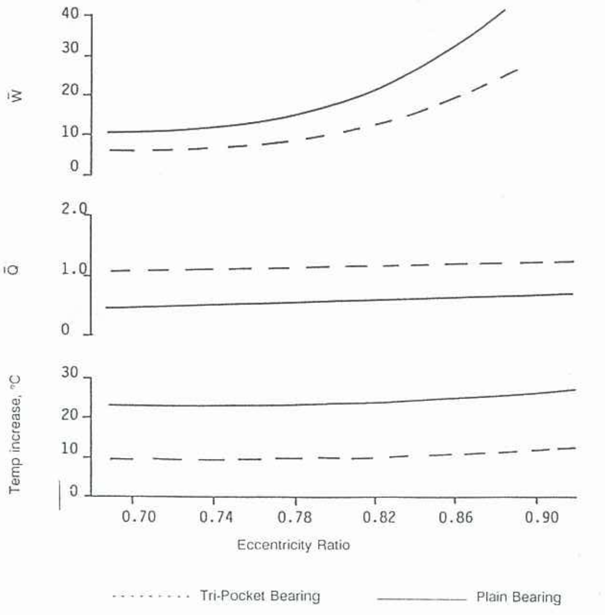

Figure 7
Load Capacity, Oil Flow Rate, and Temperature Differential vs Eccentricity (for Plain Journal and Original Tri-pocket Bearings)

Manufacturing Tolerances

The importance of manufacturing tolerances cannot be overemphasised when working with heavily loaded bearings (Martin et al (1984), Kirk (1978)). The operating envelopes detailed in Figure 4 quantify the possible variations in operation due to the manufacturing tolerances pertaining to this bearing. The bearing profile was measured on a "spare" and was found not to be within the tolerance and the ramp geometries were not well formed. The resultant bearing was oversized and the profile was more akin to a tri-lobe bearing than the tri-taper bearing as designed.

The effect of the poor manufacturing tolerances attained was to produce a bearing whose load carrying capacity was considerably reduced compared to the design geometry. Figure 8 shows the actual operating envelope with the design tolerances for the 125% production rate. If the bearing tolerances had been maintained, substantially lower operating eccentricities would have led to safer bearing operation. The deeper ramps manufactured in the bearing, while increasing the oil flow rate, reduced the load capacity and increased the working eccentricities. Subsequently, the substantially higher shear rates resulted in an increased heat generation rate which overshadowed the increased cooling associated with the increased oil supply rate.

CONCLUSIONS

The developed computer program was able to predict the same trends in performance as those known to exist in practice. The absolute values of performance parameters could not be directly compared.

Bearing induced instability was shown not to be likely and this agreed with field performance.

The following conclusions were made from the predicted performance of the bearing:

- the major factor contributing to the reduced operating capacity of the bearing (at a given production rate) was the poor manufacturing tolerances. Measurement of a "spare" bearing showed that the tolerances were outside the designed tolerance. The oversized ramps resulted in an increased oil flow rate and a decreased load-carrying capacity. The increased heat generation associated with the reduced load capacity overrides the cooling effect of the increased oil flow rate.
- although the load direction is fixed in gearbox applications, it was found that by re-orientating the bearing, a significant increase in load-carrying capacity could be obtained by optimising the bearing orientation.
- the existing operating conditions indicate that the bearing is at its limit of performance so that increased production rate would not be recommended.

The major limitations of the existing computer analysis is the inability to (1) accommodate a misaligned journal in the bearing and (2) turbulent flow in the lubricant in the bearing. It is intended to incorporate these phenomena into the analysis in the near future.

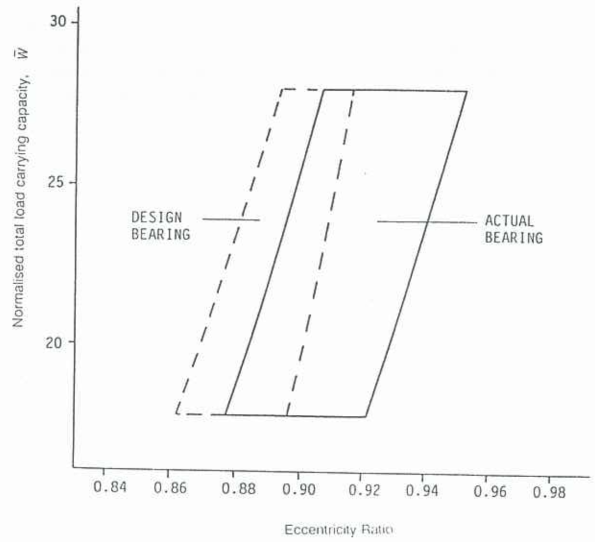

Figure 8
Actual Operating Envelope/Design Envelope 125% Production Rate

REFERENCES

ABDUL-WAHED, N et al (1982): Stability and Unbalance Response of Large Turbine Bearings, Trans ASME, Jan (104) pp 66-75.

ESDU (1984): Engineering Sciences Data Unit Item No 84031, "Calculation Methods for steadily loaded axial groove hydrodynamic journal bearings", Engineering Sciences Data Unit, 251-259 Regent St, London, W1R 7AD.

ESDU (1985): Engineering Sciences Data Unit Item No 85028, "Calculation methods for steadily loaded axial groove hydrodynamic journal bearings (Superlaminar operation)", Engineering Sciences Data Unit, 251-259 Regent St, London, W1R 7AD.

GETHIN, DT et al (1990): Loading direction effects in a three-lobe profile bore bearing, Proc of Conf "Developments in plain bearings for the 90's", I Mech E, pp 33-44.

KIRK, RG (1978): The influence of manufacturing tolerances on multi-lobe bearing performance in turbomachinery, in "Topics in Fluid Film Bearing and Rotor Bearing System Design and Optimisation" edited by SM Rohde, PE Allaire, CJ Maday, American Society of Mech Engrs, pp 108-129.

LI, DF (1980): Stability and Transient Characteristics of Four Multilobe Journal Bearing Configurations, J Lub'n Tech, July, pp 291-299.

MARTIN et al (1984): The effect of manufacturing tolerances on the stability of profile bore bearings, Conf "Vibrations in Rotating Machinery, I Mech E, Sept, York, UK, pp 287-293.