Experimental Study of Temperature Profile in a Journal Bearing

S. Kasolang @ Kasalung[1]*, M. Ali Ahmad[1], R. Dwyer-Joyce[2], A. Jaffar[1], M. A. Abu Bakar[1], N. H. Saad[1] and A. Jumahat[1]

[1]Faculty of Mechanical Engineering, Universiti Teknologi MARA, Malaysia
[2]Department of Mechanical Engineering, University of Sheffield, Sheffield, UK
*E-Mail: salmiahk@gmail.com, alie_76_02@yahoo.com, r.dwyerjoyce@sheffield.ac.uk[2]

Summary: In hydrodynamic lubrication, the viscosity condition of the fluid is critical to ensure good performance of the lubricated machine elements such as journal bearings. In the present study, an experimental work was conducted to determine the temperature distribution around the circumference of a journal bearing. A journal diameter of 100mm with a ½ length-to-diameter ratio was used. Temperature results for different radial loads and speeds were obtained. The experimental results were compared to predict values obtained using the effective temperature. The corresponding viscosity profile for certain cases was also computed as part of the preliminary study of future work on viscosity.

Keywords: Lubrication, Surface, Friction, Monitoring.

1. Introduction

Temperature monitoring is well established technique to detect overheating and to prevent hydrodynamic bearing damage [1]. As reported by Moreno et. al [2], a significant number of numerical studies has been devoted to study the steady-state temperature in journal bearing. In their study, the main difficulty was found to be associated with the exponential dependency of the viscosity on temperature.

Mishra et al [3], in the study of temperature profile of an elliptic bore journal bearing, concluded that the pressure became reduced and duplicated with increasing non circularity specifically after the non-circularity of 0.3. It was also noted that the temperature rise was less in the case of journal bearing with higher non-circularity value. This study assumed that the non-circularity to be elliptical and then the numerical solution of Reynolds equation and Energy equation have been carried out to outline the temperature profile.

An early study on temperature distribution in oil journal bearing shows that load capacity is generally less than that predicted by a classical isothermal theory [4].

An effective temperature is commonly used to calculate an effective viscosity in operating journal bearings. The effective temperature, T_{eff} can be calculated using equation,

$$T_{eff} = T_{in} + \frac{\Delta T}{2} \tag{1}$$

where,

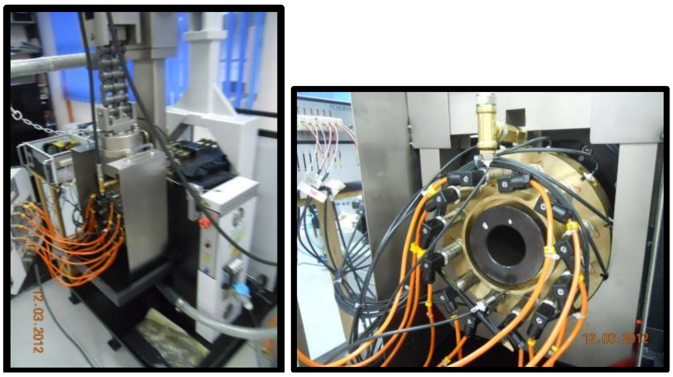
T_{in} = input temperature
ΔT = Temperature rise.

In this study, the validity of using effective temperature has been investigated for a newly develop test rig (Figure 1). A series of experiments to determine the temperature profile and power loss for different operating conditions are described.

2. Apparatus and Experimental Procedures

The Journal Bearing test rig in Figure 1 was used in this experiment. The bearing part was modified to fix 12 PT100 thermocouple wires around its journal bearing circumference at 30 degrees intervals. The journal was then mounted horizontally on the bearing. This journal could run up to a maximum speed up of 1000 rpm. A pneumatic bellow was used to apply the required load.

In the testing, the journal bearing was run at different loads and speeds. Details of test bearing dimensions, lubricant properties and operating parameters are given in Table 1. The bearing temperature profile was measured by the 12 thermocouples inserted in holes bored to within 0.5 mm from the bearing surface [1, 5]. The oil inlet pressure was regulated using a power pack lubrication system and maintained at 0.2 MPa throughout the experiments.



(a) (b)

Figure 1. (a) Journal bearing test rig. (b) Bearing with thermocouple sensors.

Table 1. Dimensions of test bearing, lubricant properties and operating parameters.

Journal diameter, D	100 mm
Bearing Length, L	50 mm
Radial clearance, c	52 µm
Load range, W	5 – 10 kN
Speed	300 – 600 rpm
Lubricant viscosity	68 cSt @ 40°C 8.8 cSt @ 100°C

3. Results and Discussions

In Figure 2 to Figure 5, the temperature profiles obtained for speed values of 300, 400, 500 and 600 rpm are shown and compared with the calculated values by the effective temperature. Generally, all graphs follow the same trend (a non-linear profile) and the temperature value increases with load. Starting from the oil supply whole at 0°, the temperature increases in an increasing rate at the beginning before it reaches a maximum value and then begins to drop again as it approaches the oil supply as the cycle completes.

The higher temperature region observed is believed to be associated with the region near the minimum film thickness which can be calculated from Raimondi and Boyd charts.

The temperature data in Figure 4 were converted to viscosity values in Figure 6. The percentage difference between the computed viscosity values by the temperature profile and those by the effective temperature were shown in Figure 7.

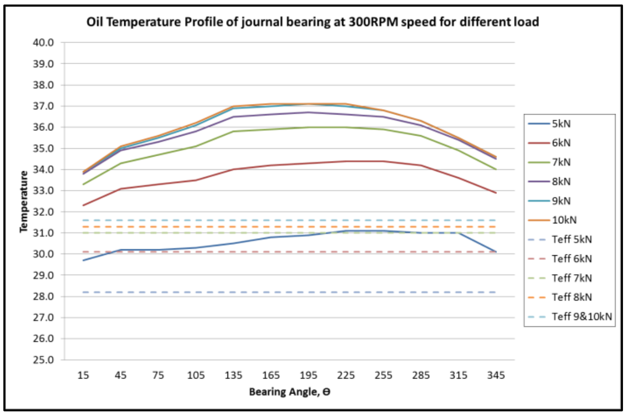


Figure 2. Oil temperature profile in journal bearing at 300 rpm for different loads.

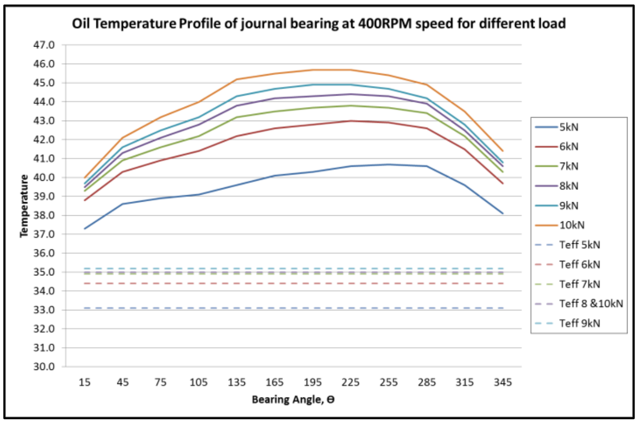


Figure 3. Oil temperature profile in journal bearing at 400 rpm for different loads.

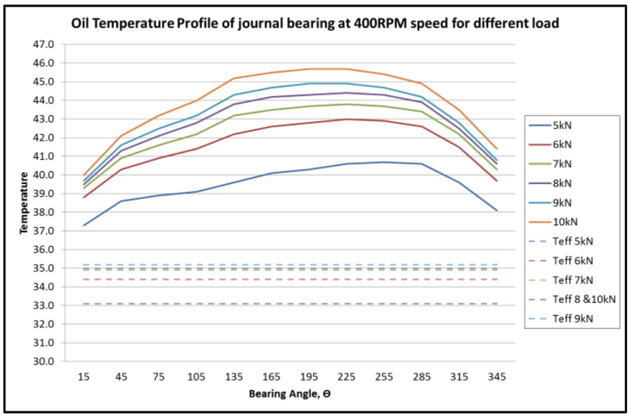


Figure 4. Oil temperature profile of journal bearing at 500 rpm speed for different loads.

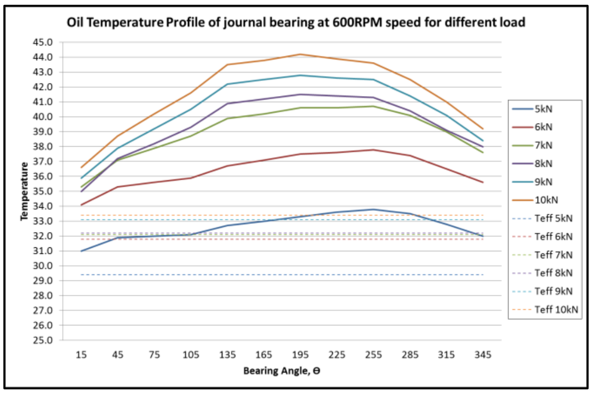


Figure 5. Oil temperature profile in journal bearing at 600 rpm for different loads.

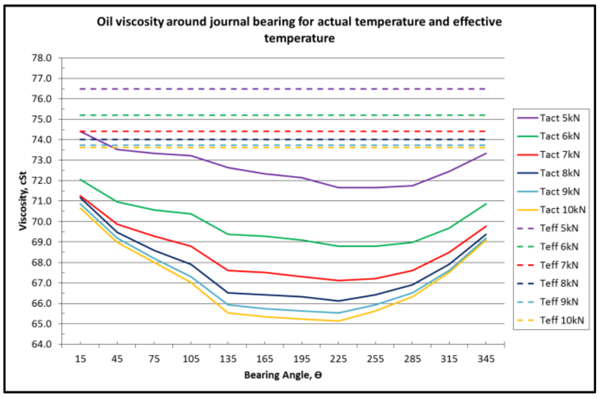


Figure 6. Computed Viscosity Profiles from actual temperatures and effective temperatures for the case of 500 rpm.

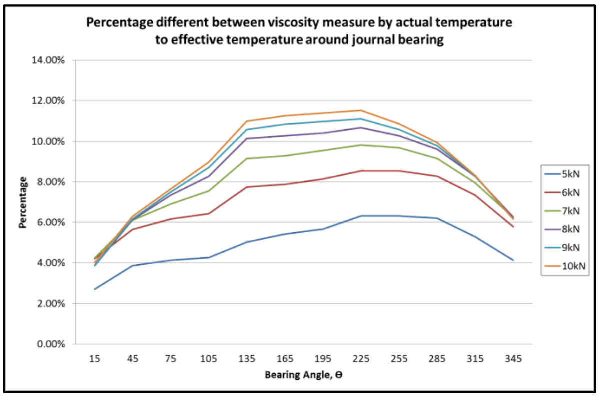


Figure 7. Percentage difference between computed viscosity values by actual temperature and effective temperature for the case of 500 rpm.

4. Conclusion

Temperature profiles around a journal bearing test rig was established for different loads and speeds. Based on the measured data of temperature profile, the corresponding viscosity values were computed and compared with viscosity values determined from the effective temperature.

Generally, the temperature increases with increasing loads for a given speed. In all cases, the temperature profile follows a similar trend that is non linear. Based on the observation, it was noted that the validity of the classical isothermal theory is less significant and hence, a further work on this respect is recommended.

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