THERMAL ASSESSMENT ON TRIBOLOGICAL PROPERTIES OF TURBOCHARGER JOURNAL BEARING

Article history
Received
27th November 2019
Received in revised form
8th June 2023
Accepted
8th June 2023
Published
20th June 2023

Chia Wei LOW[a], Siti Hartini HAMDAN[b] and William Woei Fong CHONG[c]*

[a]Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia

[b]Technical Foundation Centre, University Kuala Lumpur, Malaysian Institute of Chemical & Bioengineering Technology, 78000 Alor Gajah, Melaka, Malaysia

[c]Automotive Development Centre, Institute of for Vehicle Systems and Engineering (IVeSE), Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia

*Corresponding author
william@utm.my

ABSTRACT

High speed rotation of turbocharger rotor shaft could lead to significant frictional losses generated in journal bearing, which in turn affects the engine minimum speed. Therefore, this study is to mathematically predict the frictional behaviour for a turbocharger journal bearing, considering lubricant film formation and thermal effect. A mathematical model is initially derived from a 2-D Reynolds equation based on finite difference method and Modified Newton Raphson's Method. Then, a reduced energy equation is integrated, to consider thermal effect. The mathematical model is validated with measured friction power of a typical automotive turbocharger obtained from literature. Through the analysis, it is shown that thermal analysis assumption correlates better with literature experimental data. The significant temperature rise in lubricant leads to the reduction of viscosity as well as the frictional power losses of the turbocharger. A parametric study using Taguchi method is then conducted to optimize parameters, such as bearing width, type of engine oil used as lubricant and bearing radial clearance, for minimizing the friction power loss. The effect of bearing width is found to contribute the most to the frictional power loss, followed by the lubricant oil and lastly, the radial clearance of the bearing.

KEYWORDS

Turbocharger journal bearing, lubrication, tribology, 2-D Reynolds equation, friction, Taguchi method

INTRODUCTION

Engine downsizing using turbocharger has been demonstrated as one of the ways to significantly reduce CO2 emissions while maintaining or even improving the efficiency of an internal combustion engine (Chong et al. 2018). During most of the driven conditions, the engine is running under low load and low speed conditions. This leads to poor engine efficiency. Hence, engine downsizing reduces the engine displacement, making it capable of operating at higher load, increasing engine efficiency (Lecointe and Monnier, 2003). The specific output performance of the small engine must then be increased by a ratio equal to the reduction of engine displacement. Therefore, a well-adapted turbocharger seems be the best solution to feed the engine with the aim of reducing the effective fuel consumption (Lecointe et al, 2003).

During optimization of the turbocharger performance, engineers and researchers usually emphasize on aerodynamic efficiency of the compressor and the turbine while mechanical losses arising from turbocharger bearings system are often neglected. Turbocharger journal bearings are used to reduce friction along the turbocharger rotor shaft. However, in fact, undesirable frictional losses from the turbocharger bearing system could affect the minimum speed of the engine (Vorraro et al., 2017).

Frictional losses in turbocharger journal bearing are determined by the oil film formation of lubricated conjunctions along the journal. Knowledge of tribology is required to understand the formation of the film. Tribology is all about studying the interacting surfaces that are in relative motion, which covers the aspects of friction, wear, and lubrication. Nowadays, more aspects such as contact mechanics, surface engineering, rheology and morphology are added to further investigate the nature of tribology. Besides this, when the turbocharger is operating, the effect of heat transfer will also have a significant impact towards the formation of lubricant film. Thus, it is very important to understand the relationship between the tribological behavior of turbocharger journal bearing to reduce the frictional losses. Therefore, the present study intends to develop a mathematical model to predict the friction power losses generated by a turbocharger journal bearing. The model is then applied for a Design of Experiment (DOE) analysis to identify the sensitive tribological parameters that affect the frictional behaviour of the turbocharger journal bearing.

2.0 MATHEMATICAL MODELS

The model is based on the journal bearing design as shown in Figure 2.1. Regions A and B are the actual load-bearing component, where effective lubrication occurs. The rotor shaft is assumed to be perfectly-aligned during the turbocharger operation. Therefore, frictional properties along regions A and B are expected to be the same and only tribological behavior of journal bearing along region A is simulated for computational efficiency.

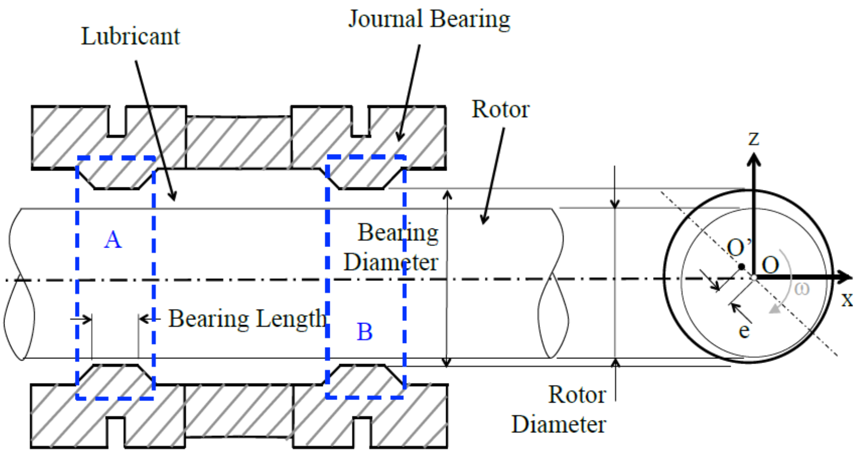


Figure 2.1 Turbocharger journal bearing diagram [1]

2.1 2-D Reynolds' Equation

The Reynolds' equation is commonly used in lubrication models in order to determine the contact pressure distribution and the lubricant film thickness. To apply the Reynolds' equation to predict the contact pressure distribution, several assumptions have to be considered (Dowson, 1962). These assumptions are:

1) Short bearing assumption.
2) Negligible pressure gradient in the direction of film thickness.
3) Newtonian fluid (No slip at boundaries).
4) Constant pressure and lubricant viscosity across the film.
5) Negligible inertia and surface tension forces.
6) Incompressible flow.

7) The radius of curvature of the bearing components is large compared with the film thickness.
8) Average side leakage velocity along y-direction is negligible.
9) Squeeze term is equal to zero. ((∂(ρh))/∂t =0)

The partial differential equation governs the pressure distribution along the lubricated conjunction and is given in Eq. (2.1)

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\cdot\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\cdot\frac{\partial P}{\partial y}\right) = 12\left\{u_{av}\frac{\partial(\rho h)}{\partial x}\right\} \quad (2.1)$$

where h represents lubricant film thickness, ρ is the lubricant density, P is the contact pressure, η is the lubricant dynamic viscosity, u_{av} equal to average lubricant entrainment velocity, x represents Cartesian coordinate along the circumference of journal bearing while y is Cartesian coordinate along the length of journal bearing.

2.2 Lubricant Film Thickness

The lubricant film thickness, h is expressed using bearing coordinates as follow:

$$h = C(1-\varepsilon\cos\theta) \quad (2.2)$$

where θ is angular location of bearing in the range from -π to π, C is radial clearance of bearing while ε represents eccentricity ratio (eccentricity/C).

Considering eccentricity in x and y direction, equation (2.2) can be further modified into:

$$h = C(1-\varepsilon_x\cos\theta - \varepsilon_y\sin\theta) \quad (2.3)$$

where:
ε_x= eccentricity ratio in x-direction
ε_y= eccentricity ratio in y-direction

2.3 Lubricant Viscosity-Temperature Relation

Under high-speed operation, like the turbocharger operation, viscous heating and convection cooling along the lubricated contact can become significant. This will influence the viscosity of the lubricant, leading to varied load carrying capacity and lubrication performance of the journal bearing. To consider for the temperature effect, the lubricant viscosity-temperature relation is predicted using Vogel relation (Deligant, Podevin and Descombes, 2011) as shown in Eq. (2.4).

$$\eta = ae^{\frac{b}{T-c}} \quad (2.4)$$

where T is temperature in unit Kelvin while a, b and c are constant coefficients for different engine oil.

2.4 Reduced Energy Equation

Considering lubricant viscous heating and convection cooling only, the reduced energy equation proposed by Gohar and Rahnejat (2012) to predict the lubricant temperature change along the journal bearing-shaft conjunction is written as:

$$\eta\left(\frac{\partial u}{\partial z}\right)^2 = \rho u c_p\frac{\partial T}{\partial x} \quad (2.5)$$

where z is direction across lubricant film thickness (m) and c_p represents specific heat of lubricant ($\frac{J}{kg\cdot K}$). Similar approach has also been adopted for piston ring lubrication analysis (Chai and Chong, 2019 and Low and Chong, 2020).

2.5 Load Balance

A finite difference scheme is adopted to solve for the Reynolds' equation. By using the Gauss-Seidel iterative algorithm as discussed by Jalali–Vahid, Rahnejat and Jin (1998) and Chong, Teodorescu and Rahnejat (2014) and more recently by Chong et al. (2019), the criteria of load carrying capacity for the bearing has to be met before proceeding with friction prediction in turbocharger journal bearing. The load carrying capacity for the bearing is calculated using Eq. (2.6),

$$W = \sqrt{{W_x}^2 + {W_z}^2} \tag{2.6}$$

where $W_x = \int_0^L \int_{-\pi}^{\pi} P cos\theta R_b d\theta dy$ and $W_z = -\int_0^L \int_{-\pi}^{\pi} P sin\theta R_b d\theta dy$.

2.6 Friction Force Along Turbocharger Journal Bearing

In view of the turbocharger journal bearing being dominantly of hydrodynamic journal bearing, the friction force along the journal bearing is calculated using the formula below, where only the viscous shearing component is taken into account.

$$F_f = \int_0^L \int_{-\pi}^{\pi} \tau_v R_b d\theta dy \tag{2.7}$$

where F_f = friction force $and\ \tau_v = shear\ stress = \frac{\eta(p)u}{h(\varepsilon_x,\varepsilon_y)}$.

The friction power loss in turbocharger journal bearing is then computed by simply multiplying with lubricant entrainment velocity as shown in Eq. (2.8).

$$F_p = F_f \times u \tag{2.8}$$

3.0 RESULTS AND DISCUSSION

In order to solve for the Reynolds' equation, the average of lubricant entrainment velocity is needed as an input. This is computed by dividing the sliding velocity of rotating shaft by two as the assumption of no slip at boundary is used. The formula can be computed as:

$$u_{av} = \frac{1}{2}\left(\frac{\pi DN}{60}\right) \tag{3.1}$$

where D is diameter of journal bearing while N represents rotational speed of rotor shaft.

To ensure the numerical algorithm is derived correctly, N= 100 rpm is chosen as input to determine whether the algorithm would generate the results similar to the expected properties along the hydrodynamic lubrication regime as shown in Figure below:

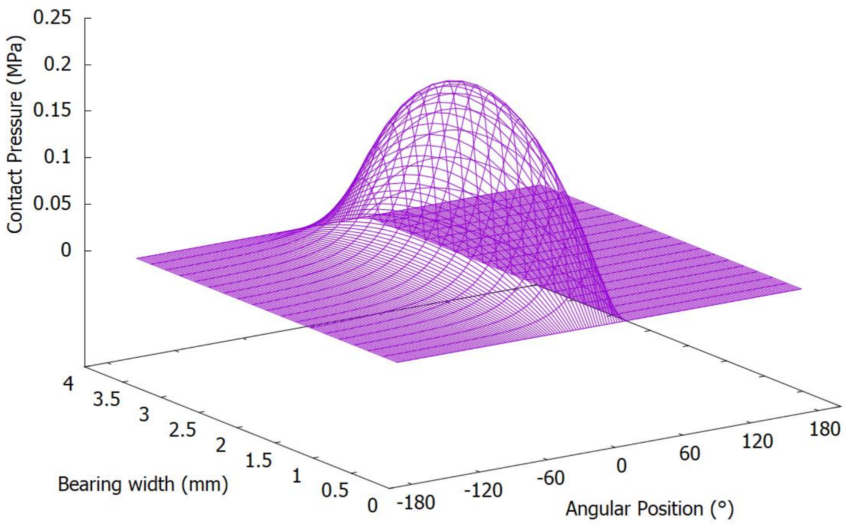
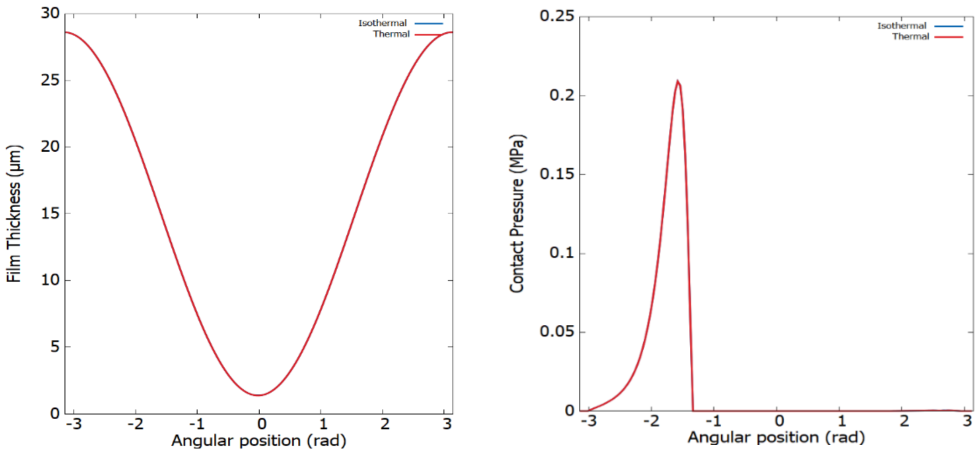


Figure 3.1 Contact Pressure Distribution

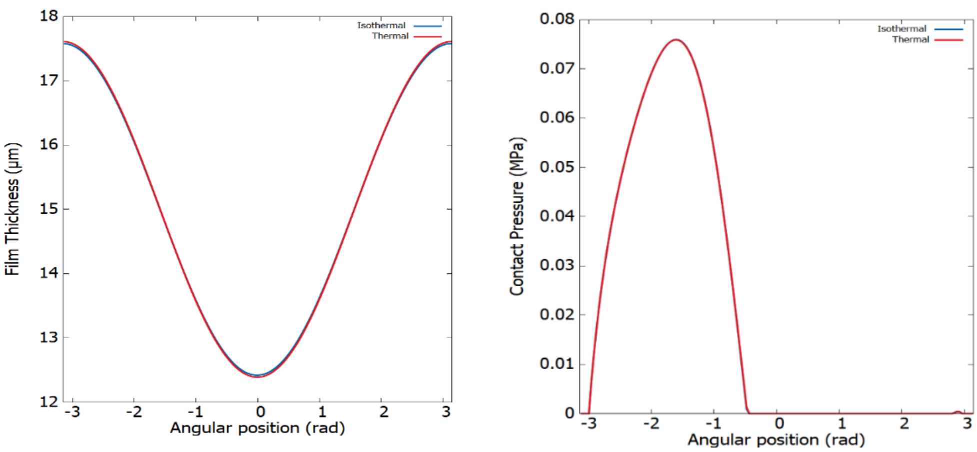
By observing the contact pressure distribution and lubricant film profile along the turbocharger journal bearing in Figure 3.1, it is shown that the hydrodynamic contact pressure build up occurs along the sliding direction in the area of convergent area. The pressure distribution is symmetrical about the mid-plane across the bearing length (y-direction) and it goes down to ambient pressure at divergent region (at 0°) because no pressure would be generated in the cavitated region at the trailing edge of the contact. This simulation results correlate well with the expected properties for a hydrodynamic lubricated journal bearing.

3.1 Isothermal Assumption and Thermal Analysis

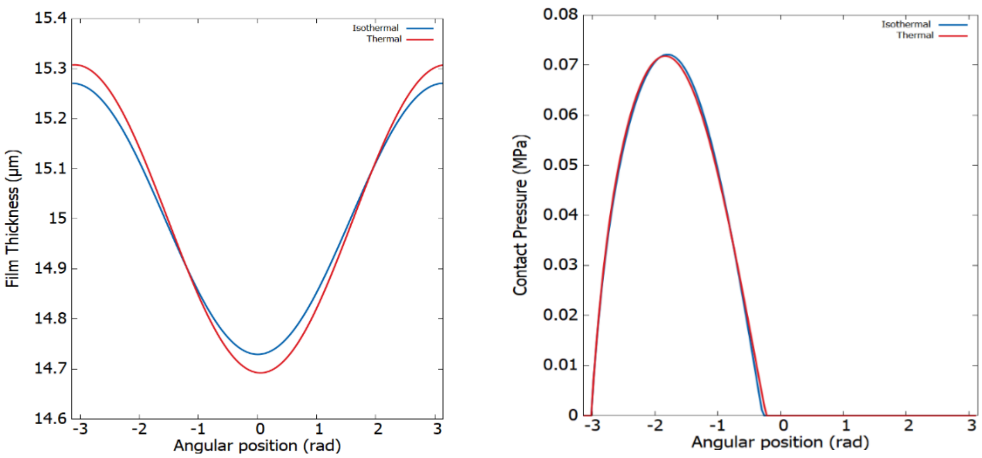
Based on the algorithm, the turbocharger is then simulated for higher rotational speeds at 10,000rpm and 100,000rpm. Along the central section of the sliding direction, the isothermal results are then compared with the results that consider thermal effects, which are shown in Figure 3.2. It is observed that the shape of contact pressure distribution becomes broader when the rotational speed increases. This is because at slow velocity, lubricants are entrained gradually into the contact, allowing for the slow pressure build-up process as observed for 100 rpm. However, at higher velocities, lubricant molecules are somewhat being forced to entrain into the contact, inducing the sudden build-up in pressure as being observed. This could also possibly lead to lubricant starvation.



a) 100 rpm



b) 10000 rpm



c) 100000 rpm

Figure 3.2 Contact pressure and lubricant film profile along the central section of the sliding direction at different rotor shaft speeds

The minimum film thickness also increases with rotational speed. At low speed, the turbocharger shaft tends to fall due to its weight and it would press against the lubricant film layer, resulting in the minimum film thickness being small (approximately 1.50 µm). However, when the shaft starts to rotate at high speed (i.e. 100 000rpm), it tends to float, causing the minimum film thickness to be increased up to 14.7 µm, which is almost the same as the radial clearance of the journal bearing itself. This believed to be because of the high rotational speed, leading to such lift-off effect on the shaft.

The distinct deviation between isothermal and thermal analysis only occurs at high rotational speed. The predicted minimum film thickness for thermal analysis is much smaller. This is mainly due to the significant temperature rise caused by viscous heating, which leads to lower lubricant viscosity during the journal bearing operation. Although the deviation might seem small, in practical, as turbocharger is always running at extremely high speed, the thermal effect towards the lubricant will become critical and must be considered for better prediction of friction power which will be discussed later.

On the aspect of friction power loss in turbocharger journal bearing, the value obtained from the algorithm is doubled to consider effective lubrications at both regions A and B. It is then compared to the experimental data provided by Deligant et al. (2011), which is shown in Figure 3.3. It is shown that the isothermal analysis over-predicts the friction power when compared with the thermal analysis, considering lubricant viscous heating and convective cooling. To be precise, after integrating the reduced energy equation into the existing isothermal algorithm, new temperature distribution along the turbocharger journal bearing is generated, which leads to the change in localised lubricant viscosity. The temperature rise causes the effective viscosity and minimum film thickness to be smaller as compared to isothermal analysis. In short, under pure lubricant viscous shearing, the heat generated is non-negligible and it is shown that the thermal effect could reduce the frictional losses along turbocharger journal bearing.

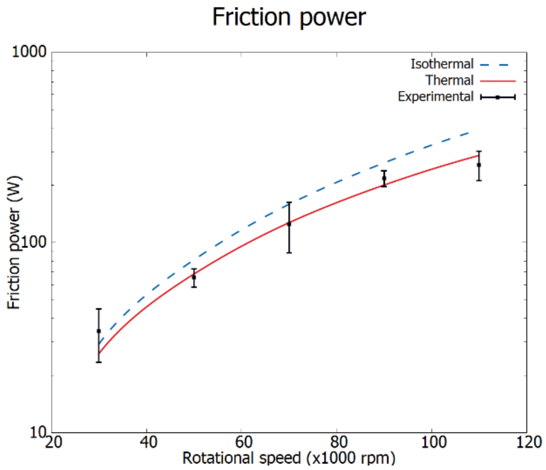


Figure 3.3 Simulated and measured friction power comparison for a turbocharger journal bearing (Experimental data from Deligant et al. (2011))

Most importantly, the friction power predicted based on the thermal analysis correlates well with the experimental data of measured friction power in a typical automotive turbocharger. The algorithm considering thermal analysis, which is more accurate is successfully validated and can now be utilized to optimize the parameters of turbocharger journal bearing to achieve minimum friction power loss.

3.2 Turbocharger Journal Bearing Optimization

Table 3.1 Parametric studies use in optimization simulation

Parameters \ Levels	1	2	3
Type of Engine oil	SAE 5W20	SAE 5W30	SAE 5W40
Bearing Width (mm)	3.8	5.4	7.0
Radial clearance (µm)	1.0	8.0	15.0
Bulk Temperature (°C)	25	60	100

The selected parameters for the optimization are shown in Table 3.1 above. The sliding velocity is made constant at typical rotational speed of an operating turbocharger (100,000rpm) while the materials and the diameter for bearing and shaft remain the same as the optimization is made for the tested turbocharger only.

Taguchi method, as Design of Experiments (DOE) approach, is being used in the present study to optimize these parameters. There are four factors listed but the bulk temperature is considered as a signal factor as it is not controllable. Therefore, the responses (friction power loss) for each combinations of other parameters are run at each level of signal (bulk temperature) to simulate the situation of operating the turbocharger immediately after the engine is started, in normal operating condition and in extreme condition, where lubricant starts to boil, in order to have better understanding of their relationships. An orthogonal arrays of L_{27} is used and each factor is assigned to their respective column by referring to the linear graph. A total of 81 cases have been simulated with each combination of factors at each level of bulk temperatures. Using Minitab, the generated results are shown below:

Table 3.2 Response Table for Signal to Noise Ratios

Level	Type of Engine Oil	Bearing Width	Radial Clearance
1	-52.89	-51.93	-55.83
2	-55.08	-54.97	-54.64
3	-56.54	-57.60	-54.03
Delta	3.66	5.67	1.80
Rank	2	1	3

Table 3.3 Response Table for Means

Level	Type of Engine Oil	Bearing Width	Radial Clearance
1	418.4	365.1	612.9
2	535.5	516.4	515.6
3	625.8	698.2	451.2
Delta	207.4	333.1	161.7
Rank	2	1	3

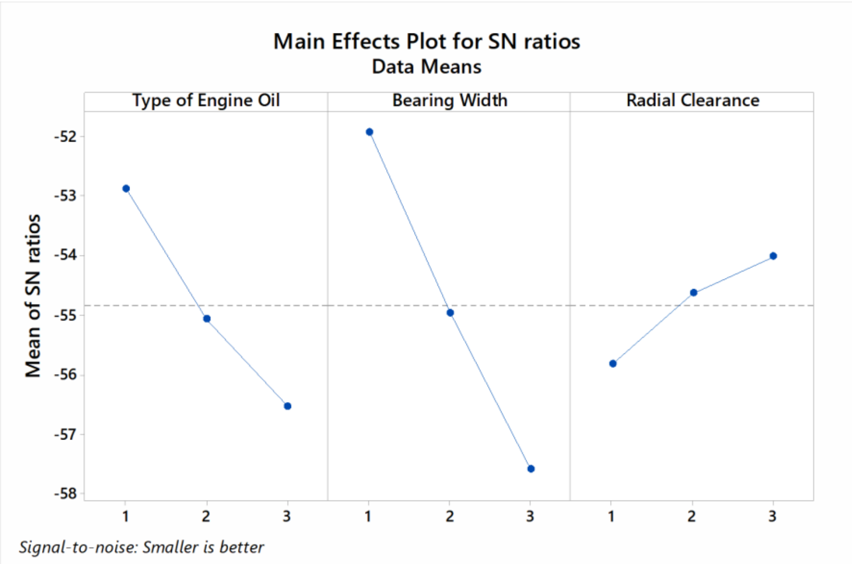


Figure 3.4 Main Effects Plot for S/N ratios

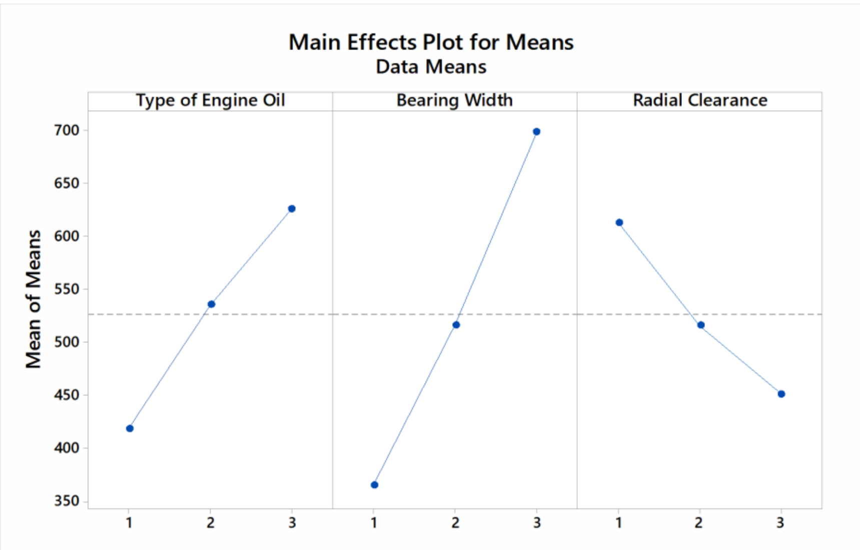


Figure 3.5 Main Effect Plot for Means

Based on the results in both tables, the greater the delta, the higher the rank, the greater the effect of that particular parameter towards frictional power loss in turbocharger journal bearing. It is shown that bearing width has the highest ranking, so, in order to minimize friction loss, it should be made as short as possible because the total surface area of the rotating shaft shearing the lubricant would increase if the bearing is long, resulting in greater loss. For lubricant oils, all of them are from the SAE 5W series but the number after the "W" indicates how thick the oil would be when it is at operating temperature. Referring to shear stress equation in Eq. 2.7, SAE 5W20 is still better than others although the film is thinner when the temperature is high. This is because the effective viscosity is much lower and this result has highlighted the dominance of effective viscosity over the oil film thickness in affecting the friction loss of turbocharger journal bearing. Meanwhile, the radial clearance would have an impact towards the minimum film thickness based on Eq. 2.2 and 2.3. Again, referring to shear stress equation, the smaller the radial clearance, the smaller the oil film thickness and the greater the shear stress, which leads to greater friction power loss. However, the effect of radial clearance is not as obvious as compared to effective viscosity.

To generate the plots in Figure 3.4 and 3.5, the values from Table 3.2 and 3.3 are required. The x-axis indicates the level of each parameter studied while y-axis stands for mean of S/N ratio for Figure 4.5 and mean of means for Figure 4.6, respectively. Graphically, it is possible to rank the parameters based on the height of the graph, which indicates the delta parameter from Table 4.6 and 4.7. For Figure 3.4, the greater the signal-to-noise (S/N) ratios, the better the signal or the lesser the loss due to noise, which indicates a good sign towards the desired output. Meanwhile, for Figure 3.5, as the desired output is the minimum friction power loss in turbocharger journal bearing, the smaller could mean the better the results. Therefore, for both plots, the Taguchi method suggested that in order to obtain minimum friction power loss in turbocharge journal bearing, it is recommended to use SAE 5W20 oil (lower high temperature viscosity), shorter bearing width (3.8mm) and as large as possible for radial clearance of journal bearing (15 µm).

CONCLUSION

The study proposes a mathematical model based on the 2-D Reynolds equation and a reduced energy equation to determine the frictional power of turbocharger journal bearing, operating at speeds up to 110,000 rpm. The thermal analysis considering lubricant viscous heating and convective cooling is shown to predict friction power that correlates well with experimental data obtained from literature. The validated mathematical model is then used to conduct a parametric study using Taguchi Method to optimize selected parameters, such as bearing width, type of engine oil used as lubricant and bearing radial clearance, for minimizing the frictional power loss in turbocharger journal bearing. The generated results has shown the ranking of each parameter towards the friction loss is as the sequence mentioned above. To minimize the friction power loss, the recommended settings based on the selected parameters are by using lower high temperature viscosity lubricant oil with shorter bearing width and as large as possible for radial clearance of journal bearing. The mathematical model proposed in this study has been shown to provide a numerical platform to allow for a more detailed assessment of the turbocharger frictional losses, such as considering squeeze film effect, side leakage and rheology of lubricants or using a full energy equation which considers conduction cooling term as the future work of this study.

Acknowledgement:

The author acknowledges the technical support from Universiti Teknologi Malaysia (UTM) under the Transdisciplinary Research Program (TDR) of "Innovative Materials for Resilient Contact Systems" (Project Vot. No. 06G19 and 05G56).

References:

1. Chai, B.Y.B. and Chong, W.W.F. (2019). Predicting frictional losses generated by piston connecting-rod big end journal bearing for an internal combustion engine. Journal of Transport System Engineering, 6(1).
2. Chong, W.W.F., Teodorescu, M. and Rahnejat, H. (2014). Mixed thermo-elastohydrodynamic cam–tappet power loss in low-speed emission cycles. International Journal of Engine Research, 15(2), 153-164.
3. Chong, W. W. F., Ng, J. H., Rajoo, S., & Chong, C. T. (2018). Passenger transportation sector gasoline consumption due to friction in Southeast Asian countries. Energy conversion and management, 158, 346-358.
4. Chong, W. W. F., Hamdan, S. H., Wong, K. J., & Yusup, S. (2019). Modelling transitions in regimes of lubrication for rough surface contact. Lubricants, 7(9), 77.
5. Deligant, M., Podevin, P., & Descombes, G. (2011). CFD model for turbocharger journal bearing performances. Applied Thermal Engineering, 31(5), 811-819.
6. Dowson, D., 1962. A generalized Reynolds equation for fluid-film lubrication. International Journal of Mechanical Sciences, 4(2), pp.159-170.
7. Gohar, R., & Rahnejat, H., 2012. *Fundamentals of tribology*. London: Imperial College Press
8. Jalali-Vahid, D., Rahnejat, H. and Jin, Z.M., 1998. Elastohydrodynamic solution for concentrated elliptical point contact of machine elements under combined entraining and squeeze-film motion. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 212(6), pp.401-411.
9. Lecointe, B. and Monnier, G., 2003. Downsizing a gasoline engine using turbocharging with direct injection (No. 2003-01-0542). SAE Technical Paper.
10. Low, H.J. and Chong, W.W.F., 2020. Thermo-elastohydrodynamic analysis of piston connecting-rod big end journal bearing lubrication system. Journal of Transport System Engineering, 7(2)