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Simulating thermo-hydrodynamic lubrication of turbocharger journal bearing

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KEYWORDS

Turbocharger
Journal bearing
Reynold's equation
Energy equation
Lubrication

ABSTRACT

The adoption of turbochargers in engine downsizing is effective to lower the CO2 emissions by internal combustion engines. However, the journal bearings, which are used to provide lubrication to the turbocharger shaft, can rotate up to 150,000 rpm. Thus, the undesirable frictional losses from this bearing system could severely affect the engine minimum speed, where low-end torque can be achieved. Therefore, as a first approximation, the study assesses the tribological characteristics of a typical turbocharger journal bearing. A thermo-hydrodynamic mathematical model is developed to simulate the journal bearing that operates between 10,000 rpm and 120,000 rpm. The model solves for the lubricant fluid film formation along the journal bearing by coupling the 2-D modified Reynolds and the 2-D energy equation. The energy equation considers convection and conduction cooling along with compressive and viscous heating. Surface flash temperatures are solved by simulating the dissipation of heat generated at the interface between the lubricant and the bounding solid through conduction cooling. The mathematical model correlates well with the measured friction power of a typical automotive turbocharger, thus, providing a fundamental numerical platform to better assess turbocharger journal bearing frictional losses that are essential in selecting/formulating suitable lubricant for turbochargers.

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NOMENCLATURE

c	Bearing clearance (μm)
h	Lubricant film thickness (m)
k_f	Thermal conductivity of the lubricant (W/m K)
p	Contact pressure distribution (Pa)
t	Time (s)
u	Sliding along x-direction (m/s)
v	Sliding along y-direction (m/s)
x	Cartesian coordinate along the circumference of journal bearing (m)
y	Cartesian coordinate across the length of journal bearing (m)
z	Direction across lubricant film thickness (m)
C_p	Specific heat capacity (J/K)
F_f	Friction force (N)
L	Bearing length (mm)
R	Bearing radius (mm)
T	Lubricant temperature (K)
W_x	Load carrying capacity of bearing in x-direction (N)
W_y	Load carrying capacity of bearing in y-direction (N)
W	Load carrying capacity of bearing (N)
Z	Lubricant viscosity-pressure dependence coefficient (-)
β_0	Lubricant viscosity-temperature dependence coefficient (-)
β_t	Thermal expansivity (K^{-1})
δ	Surface deflection (m)
η	Lubricant dynamic viscosity at temperature Θ (Pa.s)
η_0	Bulk lubricant dynamic viscosity at temperature Θ_0 (Pa.s)
γ	Temperature–viscosity coefficient (K^{-1})
ϕ	Angular location along the bearing circumference (rad)
ρ_0	Bulk lubricant density at temperature Θ_0 (kg/m^3)
ρ	Lubricant density at temperature Θ (kg/m^3)
τ_v	Viscous shear stress (Pa)

1.0 INTRODUCTION

Global energy demand is expected to increase from 104 quadrillion BTU (110 EJ) in 2012 up to 155 quadrillion BTU (163 EJ) in 2040 (Chong et al., 2018). It also estimated that the transportation sector would experience a similar upward trend in energy demand. As a result of higher energy demand in the transportation sector, decarbonisation must now be the core of international efforts to combat climate change. For the transportation sector, decarbonisation efforts should focus on 1) improving the efficiency of vehicles and 2) moving towards alternative and cleaner fuels.

To achieve better efficiency, engine downsizing is a commonly considered approach. Downsizing engines is one of the most promising technologies in reducing fuel consumption in vehicles. A boosting device, typically turbochargers, is required to compensate for the loss of performance due to downsizing. The selected turbocharger has a significant impact on the performance of a downsized engine, making optimization of this boosting device even more crucial to increase the efficiency of the vehicle. Often, the optimization of turbochargers tends to focus on the aerodynamic efficiency of the compressor and the turbine. However, it should also be noted that the mechanical losses from the bearing system within the turbocharger could affect the minimum engine speed, where low-end torque is desired (Hoepke et al., 2015). Such a deficiency might hinder the engine's potentials for better fuel consumption.

In a turbocharger, the bearing system comprises of 1) journal bearings for shaft rotation and 2) axial thrust bearings to carry axial load due to force imbalance between the compressor and turbine. The mechanical losses of a turbocharger are mainly attributed to the frictional losses of these bearings (Deligant et al., 2012). To date, limited studies have been conducted to better understand the frictional properties of the turbocharger bearing system. Among those include the experimental work by Deligant et al. (2012) in which they measured the friction power of the turbocharger bearings using a torque meter. Deligant et al. (2012) investigated the effect of lubricant inlet temperature and mass flow rate on the frictional behaviour of the turbocharger bearing system.

On the simulation aspect of turbocharger bearing systems, Chun (2008) simulated turbocharger journal bearing using the Reynolds equation to determine the aeration effect on its performances. In a more recent work, Deligant et al. (2011) applied the Computational Fluid Dynamics (CFD) approach to model the turbocharger journal bearing. Both the studies have showed that it is essential to consider the thermal effects of the lubricant towards the frictional properties of the turbocharger journal bearing. Therefore, in this study, as a first approximation, an improved numerical algorithm is derived based on a 2-D Reynolds equation to assess the tribological properties of a turbocharger journal bearing. The algorithm considers the thermal effect towards the lubricant viscosity during turbocharger operation using a 2-D energy equation, considering convection and conduction cooling along with compressive and viscous heating.

2.0 NUMERICAL APPROACH

The study aims to determine the frictional properties of the turbocharger journal bearing, operating between 10,000 rpm and 120,000 rpm. The analysis is based on the journal bearing given in Figure 1. Within the turbocharger journal bearing, the actual load-bearing component is along the regions A and B (see Figure 1), where the effective lubrication occurs. Assuming no misalignment along the rotor shaft during the turbocharger operation, the frictional properties along both regions A and B are expected to be the same. Hence, as a first approximation, the study attempts to simulate only the tribological behaviour of the journal bearing along region A.

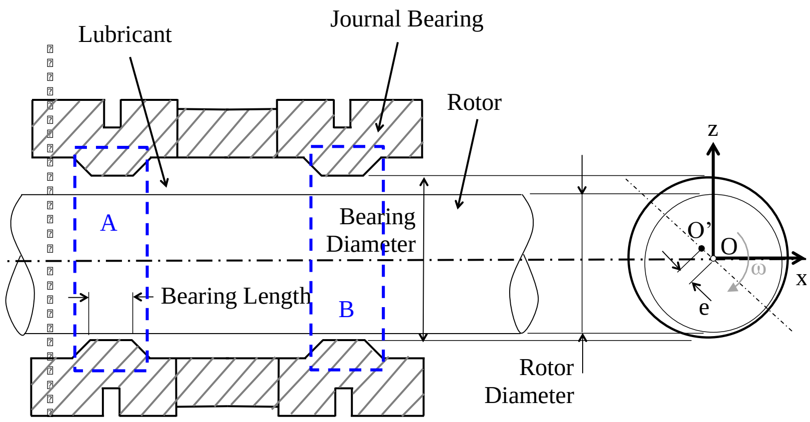


Figure 1: Turbocharger journal bearing diagram.

The journal bearing analysis involves a cylindrical coordinate system to determine the tribological properties. In this study, for ease of setting up the required mathematical tools, a Cartesian coordinate system is adopted instead. The y-direction is the axis along the bearing length. The x-direction is the unwrapped journal bearing along the circumference of the bearing. In a typical journal bearing lubrication system, the bearing operates along with the fluid film lubrication regime, namely the hydrodynamic lubrication regime. To predict the fluid film formation for the investigated lubrication system, a 2-D Reynolds equation is applied. This partial differential equation governs the pressure distribution along with the lubricated conjunction and is given in Eq. (1) (Chong et al., 2019),

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right) = 12\left(\frac{\partial}{\partial x}\rho h u_{avg} + \frac{\partial}{\partial y}\rho h v_{avg}\right) \tag{1}$$

Where u_{avg} represents the entrainment sliding motion along the circumferential direction while v_{avg} represents the sliding motion along the bearing length direction. In this study, sliding along the bearing length direction is taken to be zero ($v_{avg} = 0$) by assuming that the motion in this direction is fully suppressed by the axial thrust bearings. The 2-D Reynolds solution is adopted based on the short-journal bearing assumption, which considers the pressure change due to the side flow (along the bearing length direction) being significant and no longer negligible. The contact geometry, h, in this study is taken as given in Eq. (2),

$$h(x, y) = c - e_x \cos(\phi) - e_z \sin(\phi) \tag{2}$$

The terms e_x and e_z refer to the eccentricity in x- and z-direction, respectively.

Under high-speed operation, synonymous with the turbocharger operation, heat transfer along the lubricated contact can become significant. The generated heat will influence the viscosity of the lubricant, leading to varied load carrying capacity and lubrication performance of the journal bearing. Therefore, in this study, to consider for the temperature effect, the lubricant viscosity-temperature relation is predicted using Roeland's equation as shown in Eq. (3) (Roelands et al., 1966),

$$\eta(p) = \eta_o e^{(\ln \eta_o + 9.67)\left\{\left(1 + {}^{p}/{}_{p_o}\right)^{z} - 1\right\} - \gamma(T - T_o)} \tag{3}$$

Where $Z = 0.5885$, $1/p_o = 5.1 \times 10^{-9}\ Pa^{-1}$ and $\gamma = 0.04\ K^{-1}$. Even though the viscosity-pressure effect for the simulated problem is negligible, it is still included in the model for completeness's sake.

Considering thin-film flows at steady conditions, the 2-D energy equation is expressed as in Eq. (4) (Pai, 1956 and Kim et al., 2001) is adopted in the present study to determine the change in lubricant operating temperature.

$$\rho C_p \left(U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial z}\left(k_f \frac{\partial T}{\partial z}\right) + \beta_T \left(U \frac{\partial p}{\partial x} + V \frac{\partial p}{\partial y}\right) + \eta \left\{\left(\frac{\partial U}{\partial z}\right)^2 + \left(\frac{\partial V}{\partial z}\right)^2\right\} \tag{4}$$

Where $U = -\left\{\frac{z(h-z)}{2\eta}\right\}\left(\frac{\partial p}{\partial x}\right) + \frac{u_1(h-z)}{h} + \frac{u_2 z}{h}$, $V = -\left\{\frac{z(h-z)}{2\eta}\right\}\left(\frac{\partial p}{\partial y}\right)$ with T being the lubricant temperature. On the left-hand side of equation (4), the energy equation considers the convection cooling effect. On the right-hand side of equation (4), the energy equation considers conduction cooling (first term) with compressive (second term) and viscous heating (third term). The approach adopted in the present study in solving the energy equation is further explained by Kim et al. (2001). It is to note that the steady-state heat transfer simulated in the present study considers the amount of heat generated via the viscous shearing effect to be the same as the heat dissipated via convection and conduction cooling. Surface flash temperatures are then solved by simulating the dissipation of heat generated at the interface between the lubricant and the bounding solid through conduction cooling.

The turbocharger journal bearing is simulated based on the specifications given in Table 1 by Deligant et al. (2012). Figure 2 illustrates the flow chart for the numerical model to simulate the turbocharger bearing. Once the simulated parameters are provided, the Reynolds equation as given in equation (1) can then be solved. It is to note that a complete analytical solution, such as the one proposed by Ng et al. (2018), could be considered for the derivation of Reynolds' equation. However, the integration of the energy equation would then be much more complicated. Hence, to ease the integration of the energy equation, a finite difference scheme is instead adopted to solve the Reynolds equation, considering lubricant viscous heating and convection cooling, in predicting the fluid film formation of lubricated conjunctions. The finite difference scheme is solved using the Gauss-Seidel iterative algorithm (Chong et al., 2014 and Chong et al., 2019). Referring to Figure 2, the iteration is executed considering the pressure and the lubricant temperature balance loop. The converged solution has to fulfill the criteria, where the change of contact pressure and lubricant temperature as compared to the previous iteration is less than 1%. Once the converged pressure is obtained, the load-carrying capacity for the bearing is calculated using Eq. (5),

$$W = \sqrt{W_x^2 + W_z^2} \tag{5}$$

Where $W_x = \int p \cos\phi \, L.Rd\phi$ and $W_z = \int p \sin\phi \, L.Rd\phi$. The friction along the journal bearing is calculated using Eq. (6),

$$F_f = \int \tau_v.L.Rd\phi \tag{6}$$

Where $\tau_v = \eta u/h$.

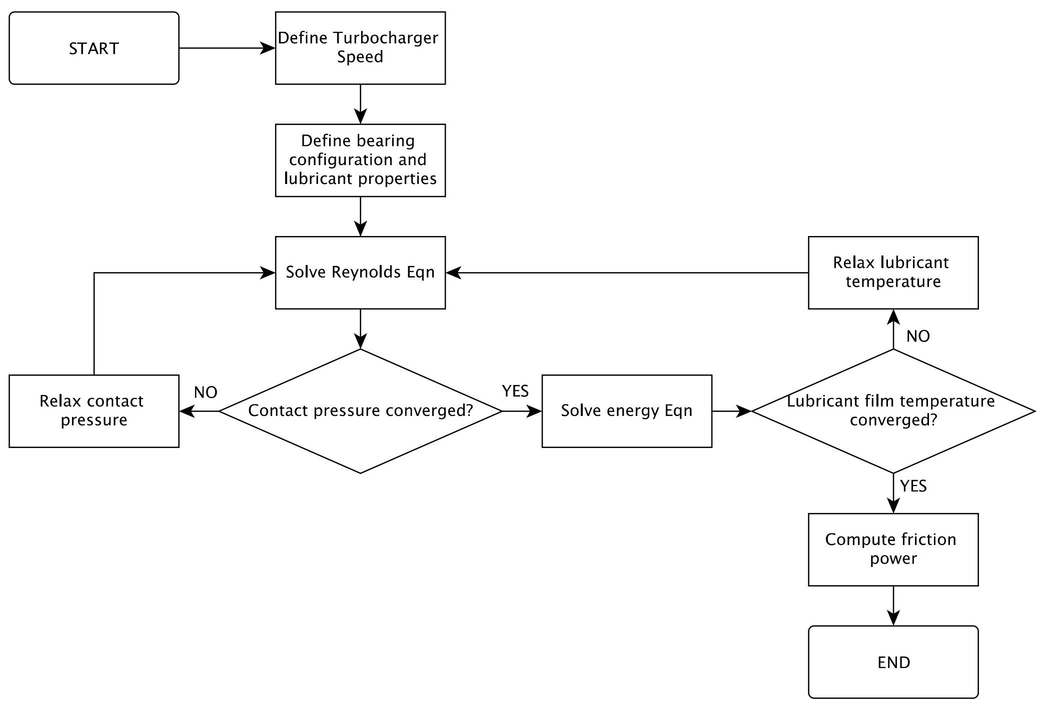


Figure 2: Flow chart.

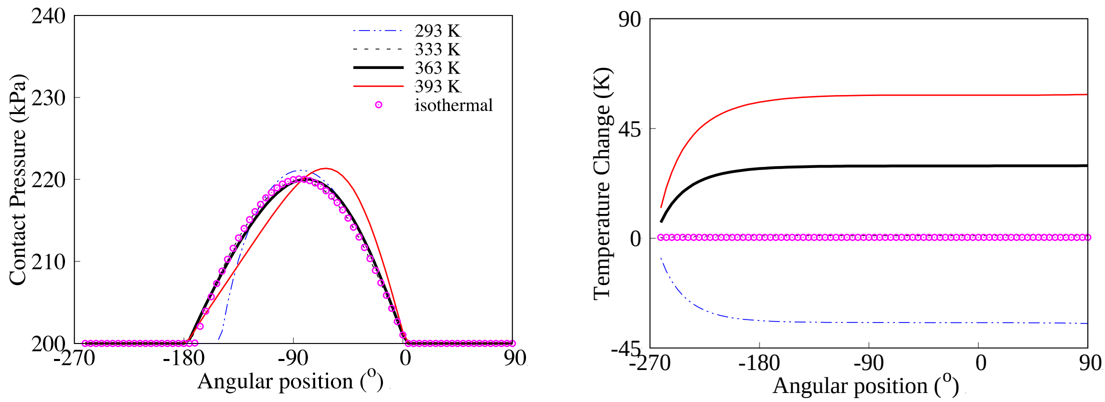
Table 1: Simulated turbocharger journal bearing lubrication system.

Parameter	**Value**	**Unit**
Bearing		
Clearance, c	15	μm
Radius, R	3.485	Mm
Length, L	3.8	Mm
Modulus Young	110	GPa
Poisson's ratio	0.34	-
Thermal conductivity	121	W/mK
Specific heat capacity	377	J/kg.K
Density	8530	kg/m3
Shaft		
Modulus Young	200	GPa
Poisson's ratio	0.29	-
Thermal conductivity	47	W/mK
Specific heat capacity	460	J/kg.K
Density	7850	kg/m3
Lubricant		
Inlet lubricant temperature, T_o	333	K
Inlet lubricant bulk viscosity, η_o	0.14	Pa.s
Thermal conductivity	0.125	W/mK
Specific heat capacity	2100	J/kg.K
Density	884	kg/m3

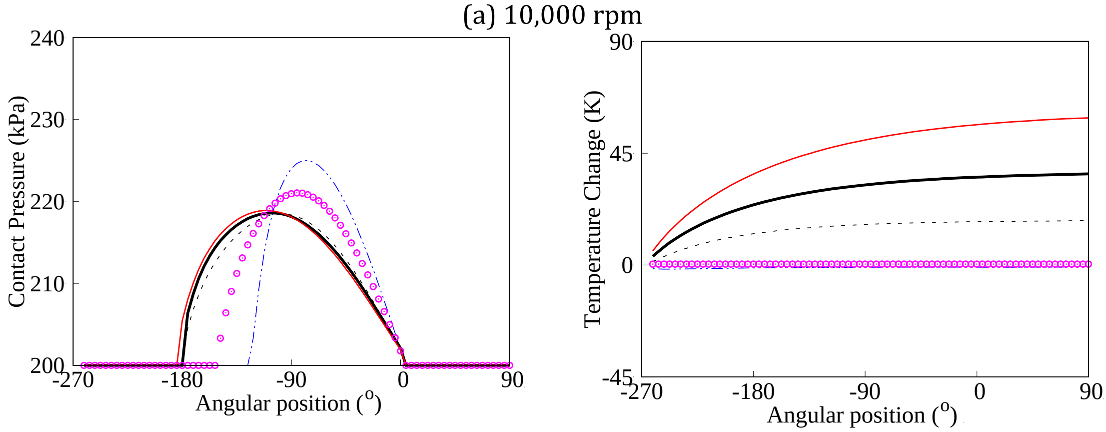
3.0 RESULTS AND DISCUSSION

The current study attempts to determine the frictional losses in a turbocharger journal bearing using a numerical 2-D Reynolds solution along with a 2-D energy equation. The turbocharger journal bearing is simulated for higher speeds, ranging from 10,000 rpm to 120,000 rpm with the lubricant inlet temperature taken as 333 K. The contact pressure distribution and the lubricant film profile along the central section of the sliding direction are given in Figure 3 at different assumed surrounding ambient temperatures. The contact pressure predicted from an isothermal assumption is also compared to the thermal analysis using the 2-D energy equation. It is observed that at relatively low speeds of rotation (i.e., 10,000 rpm), the generated pressures indicate a less starved inlet at higher surrounding ambient temperature. Across the angular position along the bearing, it is also shown that the lubricant temperature tends to saturate towards the surrounding ambient temperature. This change in lubricant temperature is expected because of the conduction cooling/heating across the turbocharger journal that is considered via the 2-D energy equation.

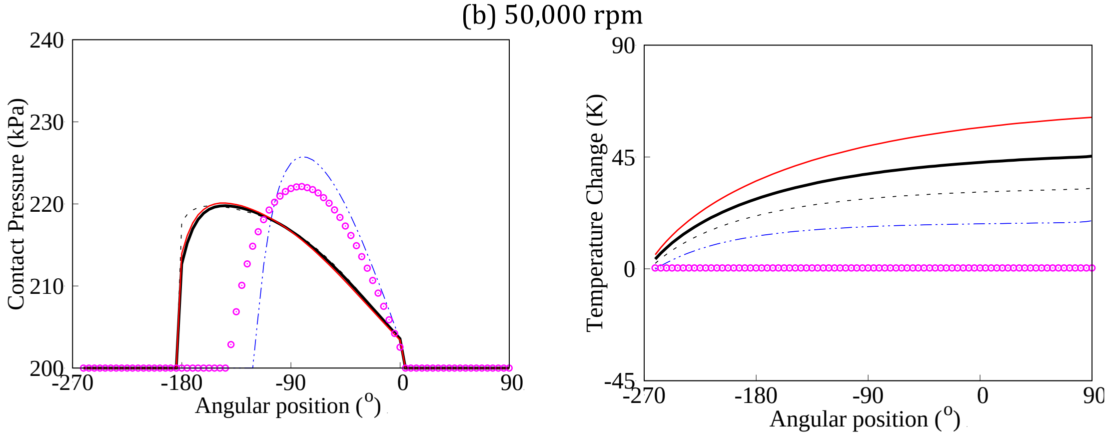
When the speed increases to 100,000 rpm, a significantly starved inlet can be observed from the simulated contact pressure distribution, especially at lower assumed surrounding ambient temperature conditions. Interestingly, at 293 K ambient temperature, the lubricant temperature is shown to not converged towards the assumed surrounding ambient temperature, indicating dominant viscous shear heating at such a high-speed operating condition. Figure 4 illustrates the change in minimum film thickness along the journal bearing at different rotor shaft speeds. The film thickness is simulated to saturate towards the bearing clearance at higher rotor shaft speeds, clearly indicating thick hydrodynamic film lubrication with no boundary interaction expected.



(a) 10,000 rpm



(b) 50,000 rpm



(c) 100,000 rpm

Figure 3: Contact pressure and lubricant temperature distribution along the central section of the sliding direction at different rotor shaft speed.

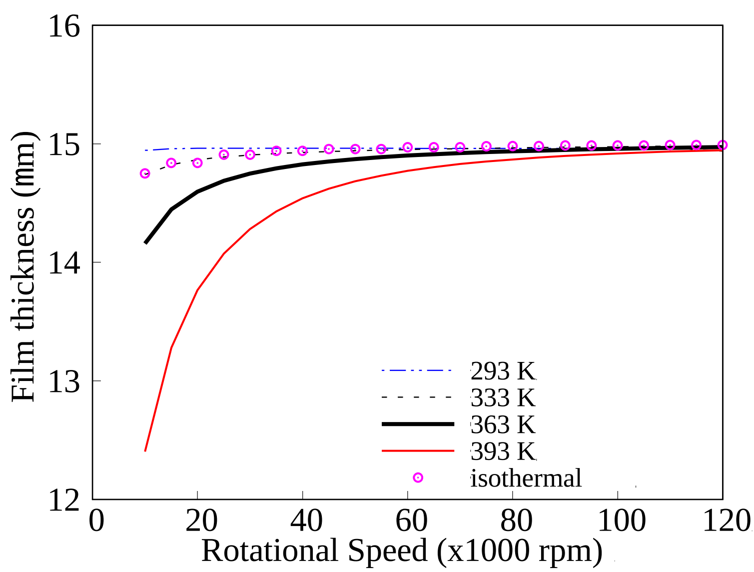


Figure 4: Minimum film thickness for the turbocharger journal bearing at different rotor shaft speed.

The friction power for the turbocharger journal bearing is then computed based on the friction force determined as in Figure 5. The friction power is compared with the experimental data provided by Deligant et al. (2012). Thus far, the simulated journal bearing friction power is based only on region A (see figure 1). However, the effective lubrication for the turbocharger journal bearing takes place also along region B (see figure 1). Taking both regions to have the same tribological properties (assuming no misalignment of the rotor shaft), the friction power is doubled and compared with the experimental data. From the comparison, the isothermal analysis over-predicts the friction power when compared with the thermal analysis using the 2-D energy equation. More importantly, the friction power predicted by assuming the surrounding ambient temperature at 363 K showed a better correlation with the measured friction power given by Deligant et al. (2011). It is to note that the experimental data from the literature did not mention/measure the surrounding ambient temperature. Thus, it can only be surmised that the heat dissipated by the high-speed operation of the turbocharger could have also led to the increased surrounding ambient temperature during the experiment, which was not factored by Deligant et al. (2012).

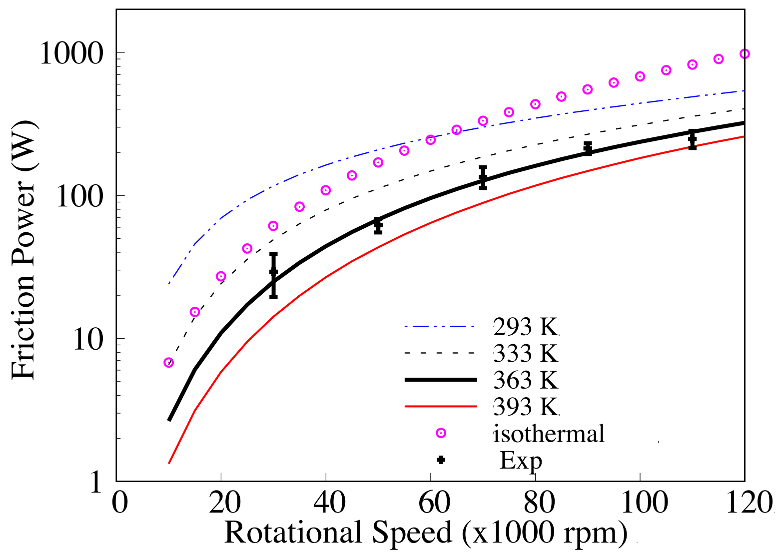


Figure 5: Simulated and measured friction power comparison for a turbocharger journal bearing (Experimental data taken from Deligant et al., 2011).

The simulated journal bearing above emphasizes the need to consider the thermal effect when assessing the frictional losses in turbochargers. During operation, it is shown in this analysis that the heat generated by the journal and the rotor shaft contact is non-negligible. It is also shown that the surrounding ambient temperature plays a role in influencing the frictional power loss of the turbocharger. Even though higher surrounding ambient temperature produces lower frictional power loss, care must be taken as the lubricant viscosity drop with increased operating temperature might still lead to undesirable lubricant film formation. The lower lubricant viscosity could otherwise bring adverse effects to the turbocharger, which is not yet considering other potential drawbacks, such as lubricant property dilution/degradation (Hamdan et al., 2018). Hence, the analysis shows the need to include an intensive turbocharger heat transfer analysis to better assess journal bearing frictional losses.

4.0 CONCLUSION

The study proposes a mathematical model using the 2-D Reynolds equation and a 2-D energy equation to determine the frictional power of turbocharger journal bearing, operating at speeds up to 120,000 rpm. The thermal analysis predicts friction power that correlates well with experimental data. Through this study, it is important to consider thermal effects to more accurately understand the tribological performance of a turbocharger journal bearing, where normal operating speeds can be as high as 150,000 rpm. Overall, the mathematical model proposed in this study provides a numerical platform to allow for a more detailed assessment of the turbocharger frictional losses, which is important, especially when deciding on the type of engine lubricant to be used for the system.

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REFERENCES

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.

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.

Chong, W. W. F., Teodorescu, M., & Rahnejat, H. (2014). Mixed thermo-elastohydrodynamic cam–tappet power loss in low-speed emission cycles. International Journal of Engine Research, 15(2), 153-164.

Chun, S. M. (2008). Aeration effects on the performance of a turbocharger journal bearing. Tribology international, 41(4), 296-306.

Deligant, M., Podevin, P., & Descombes, G. (2011). CFD model for turbocharger journal bearing performances. Applied Thermal Engineering, 31(5), 811-819.

Deligant, M., Podevin, P., & Descombes, G. (2012). Experimental identification of turbocharger mechanical friction losses. Energy, 39(1), 388-394.

Hamdan, S. H., Chong, W. W.F., & Din, M. H. (2018). Frictional analysis on engine lubricant dilution by coconut oil and soybean oil derived biodiesel. Jurnal Tribologi, 18, 149-158.

Hoepke, B., Uhlmann, T., Pischinger, S., Lueddecke, B., & Filsinger, D. (2015). Analysis of thrust bearing impact on friction losses in automotive turbochargers. Journal of Engineering for Gas Turbines and Power, 137(8).

Johnson, K. L. (1987). Contact mechanics. Cambridge university press.

Kim, H. J., Ehret, P., Dowson, D., & Taylor, C. M. (2001). Thermal elastohydrodynamic analysis of circular contacts part 1: Newtonian model. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 215(4), 339-352.

Ng, Y. C., Hamdan, S. H., & Chong, W. W. F. (2018). Development of a mathematical tool to predict engine in-cylinder friction. Jurnal Tribologi, 17, 29-39.

Pai, S. I. (1956). Viscous Flow Theory, Vol. 1. Laminar Flow. D. Van Nostrand Company.

Roelands, C. J. A., Winer, W. O., & Wright, W. A. (1971). Correlational aspects of the viscosity-temperature-pressure relationship of lubricating oils (dr in dissertation at technical university of delft, 1966).