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Fluid inertia and energy dissipation in turbocharger thrust bearings

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ABSTRACT

The present study introduces a new transient modified Reynolds equation by considering a detailed model to represent multigrade engine oils. Elastic effects introduced by long polymeric chain additives are studied. This thin film equation also accounts for well known rheological effects such as the viscosity dependence on temperature and shear rate. The application of this model to turbocharger (TC) thrust bearings is presented and includes the consideration of fluid inertia. Performances predictions of a typical TC thrust bearing are compared to experimental data obtained on a test rig. Rheological parameters are varied to determine the influence of current engine oils on the phenomenon of turbo lag.

1. Introduction

With increasing focus on engine downsizing, high performance and low fuel consumption, turbocharger technology must address several design challenges to reach consumer expectations of improvements to automotive engine performance. One of such challenges consists in addressing the phenomenon of 'turbo lag' that delays the boost response and penalizes the customer drivability. As a device to increase power in internal combustion engines, the TC increases pressure in the combustion chamber only after having been provided sufficient energy to increase its rotational speed. However, the bearing system located between the turbine wheel and the compressor wheel is the origin of a parasitic frictional drag torque that reduces the energy transmitted to the compressor wheel by the turbine. Several studies report the origin of friction loss in turbocharged engines [1,2]. Among them, Deligant et al. [3] study the respective influences of the thrust bearing and the journal bearing on the TC global power loss. It turns out that thrust bearings play a larger role in frictional losses than radial bearings. These authors observe that the impact of the axial load on power loss is relatively small and fairly linear. More recently, Hoepke et al. [4] conducted a similar study analyzing the impact of thrust bearing on friction losses. They quantified the axial bearing contribution to the overall power loss to be as high as 38% and also linearly increasing with axial load.

In order to always shorten the turbo lag, the turbochargers companies are putting their efforts in optimizing the hydrodynamic bearings to reduce bearing losses. In addition, the industry looks to alternative technologies such as ball bearing or e-boost engine. It has been shown by De Araujo that substituting the axial and radial bearings with ball bearings decreases the turbo lag up to 25% for an equivalent TC size [5]. The advantage of such a system depends on the rolling contact which dissipates less energy than a shear film. This leads to the availability of more engine torque at low engine speeds. However, these solutions remain more expensive than hydrodynamic bearings TCs.

The present study aims at analyzing the influence of complex engine oil rheology on the energy dissipation of the thrust bearing contact. Long polymeric chain additives such as viscosity index (VI) improvers are known to improve base mineral oil characteristics, and can be described by the very general and commonly used Phan-Thien and Tanner (PTT) model. This model is based on the description of the liquid microstructure [6]. It is used in numerous applications and can represent, for instance, the rheological behavior of crude oils [7], blood [8], synovial fluid [9], and engine oils [10]. In the present study PTT is coupled with two additional rheological laws accounting for the temperature effect on viscosity, and the shear-thinning effect; and is thus called the modified PTT or MPTT model. Actual TC rotational speeds up to 300 krpm require one to consider fluid inertia in the contact as modified Reynolds numbers locally reach up to 40. This rheological combination results in an original modified Reynolds equation solved simultaneously with a 3D energy equation to compute the temperature in the fluid film. As presented by Heshmat and Pinkus [11], the use of a recirculation coefficient helps set a better mixing inlet temperature by accounting for hot oil dragged between two successive pads.

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Nomenclature

x, y, z	cartesian coordinates [m]
u, v, w	velocities $\left[\mathrm{m\,s^{-1}}\right]$
h	film thickness [m]
σ	stress tensor [Pa]
I_{ij}	inertia terms $\left[\mathrm{m^3\,s^{-2}}\right]$
C_p	heat capacity $\left[\mathrm{J\,kg^{-1}K^{-1}}\right]$
λ	oil relaxation time [s]
$\dot{\gamma}$	shear rate $\left[\mathrm{s^{-1}}\right]$
k	thermal conductivity $\left[\mathrm{W\,m^{-1}\,K^{-1}}\right]$
$\bar{W}$	dimensionless axial load [dimensionless]
c_0	minimum film thickness [m]
L	pad length [m]
t	time [s]
ρ	density $\left[\mathrm{kg\,m^{-3}}\right]$
P	pressure [Pa]
q_i	averaged flow rate $\left[\mathrm{m^2\,s^{-1}}\right]$
η	dynamic viscosity [Pa s]
T	temperature [K]
ε, ξ, n	models parameters [dimensionless]
$\dot{\gamma}_c$	critical shear rate $\left[\mathrm{s^{-1}}\right]$
De	deborah number [dimensionless]
$\bar{P}$	dimensionless power loss [dimensionless]
B	pad width [m]
U	tangential velocity $\left[\mathrm{m\,s^{-1}}\right]$

2. Analysis

As commonly used in hydrodynamic contacts, the thin film theory assumptions are invoked. For the thrust bearings considered, the ratio of the thickness to radius varies between 10^{-2} and 10^{-3} allowing all second order terms such as $(h/R)^2$ to be neglected. The flow is assumed incompressible and laminar. In our case, Reynolds number varies from 10 to 2100 for the most severe operating conditions – 300 krpm, 50 μm – and still remains in the lower range of the laminar to turbulent transition. All external forces are negligible and there is no slip of the fluid at the bearing walls. Note that x, y and z, respectively, are the tangential, vertical and radial directions of the contact, as shown in Fig. 1.

2.1. Modified Reynolds equation

As inertia effects are considered, a direct integration of the momentum equations is not possible. All equations are averaged across the film thickness, following the method developed by Tichy and Bou-Saïd [12]. Using Leibniz integral rule, one obtains:

$$\rho\left(\frac{\partial q_x}{\partial t}+\frac{\partial I_{xx}}{\partial x}+\frac{\partial I_{xz}}{\partial z}\right)=-h\frac{\partial P}{\partial x}+\sigma_{xy_h}-\sigma_{xy_0} \tag{1}$$

$$0=\frac{\partial P}{\partial y} \tag{2}$$

$$\rho\left(\frac{\partial q_z}{\partial t}+\frac{\partial I_{xz}}{\partial x}+\frac{\partial I_{zz}}{\partial z}\right)=-h\frac{\partial P}{\partial z}+\sigma_{yz_h}-\sigma_{yz_0} \tag{3}$$

where

$$q_x=\int_0^h u\,dy \qquad q_z=\int_0^h w\,dy \qquad I_{xx}=\int_0^h u^2\,dy$$

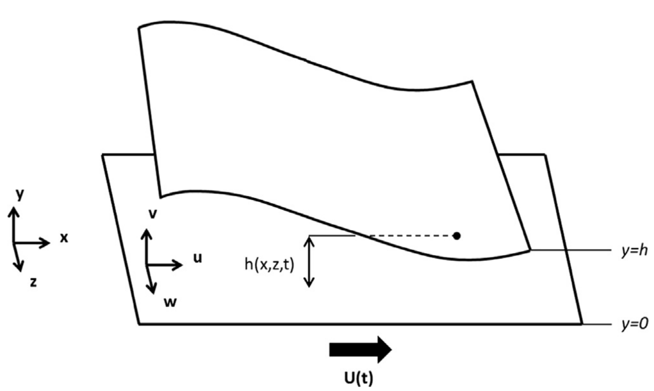

Fig. 1. Coordinate system used in the contact.

$$I_{xz}=\int_0^h uw\,dy \qquad I_{zz}=\int_0^h w^2\,dy$$

In these expressions, σ_{ij} is the deviatoric stress tensor. Its components are expressed as the sum of Newtonian stress contributions σ_s and polymeric stress contributions σ_p based on the MPTT model.

The continuity equation is also averaged across the thickness:

$$\frac{\partial q_x}{\partial x}+\frac{\partial q_z}{\partial z}+\frac{\partial h}{\partial t}=0. \tag{4}$$

The main assumption of this type of method is to assume the velocity profiles. Given the boundary conditions ($u=U; v=0; w=0$ at $y=0$ and $u=0; v=\frac{\partial h}{\partial t}; w=0$ at $y=h$), a Couette flow is superimposed on a Poiseuille flow determined without inertia terms. Note that for our application, rotational speeds are high. Therefore, Couette flow prevails over Poiseuille flow to great extent. The tangential speed profile varies almost linearly in the film thickness. Hence the shear rate σ_{xy} is close to constant in the vertical direction. This allows us to average the lubricant viscosity across the film without penalizing the thermal analysis. Following a development similar to Tichy and Bou-Saïd using the averaged momentum and continuity equations, a modified Reynolds equation is obtained:

$$\begin{aligned}
&\frac{\partial}{\partial x}\left(h^3\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial P}{\partial z}\right)\\
&=12\eta\frac{\partial h}{\partial t}+6\eta U\frac{\partial h}{\partial x}\\
&\quad-\rho\left[2h\frac{\partial h}{\partial x}\left(\frac{\partial q_x}{\partial t}+\frac{\partial I_{xx}}{\partial x}+\frac{\partial I_{xz}}{\partial z}\right)+2h\frac{\partial h}{\partial z}\left(\frac{\partial q_z}{\partial t}+\frac{\partial I_{xz}}{\partial x}+\frac{\partial I_{zz}}{\partial z}\right)\right]\\
&\quad-\rho h^2\left[-\frac{\partial^2 h}{\partial t^2}+\frac{\partial^2 I_{xx}}{\partial x^2}+2\frac{\partial^2 I_{xz}}{\partial x\partial z}+\frac{\partial^2 I_{zz}}{\partial z^2}\right]\\
&\quad-12h\left(U_1\frac{\partial\eta}{\partial x}+W_1\frac{\partial\eta}{\partial z}\right)+2h\left[\frac{\partial h}{\partial x}(\sigma_{xy\,P\,h}-\sigma_{xy\,P\,0})+\frac{\partial h}{\partial z}(\sigma_{yz\,P\,h}-\sigma_{yz\,P\,0})\right]\\
&\quad+h^2\left[\frac{\partial}{\partial x}(\sigma_{xy\,P\,h}-\sigma_{xy\,P\,0})+\frac{\partial}{\partial z}(\sigma_{yz\,P\,h}-\sigma_{yz\,P\,0})\right]
\end{aligned} \tag{5}$$

with $U_1=\frac{1}{12\eta}\frac{\partial P}{\partial x}h^2$ and $W_1=\frac{1}{12\eta}\frac{\partial P}{\partial z}h^2$ representing the averaged Poiseuille velocities respectively in the radial and tangential directions.

The difference of this equation relative to the classical Reynolds equation is observed in the second part of the right hand side term. The third and fourth terms account for the inertia effect and the surface acceleration. The fifth term represents the variation of the viscosity in the contact. Finally, the polymeric contribution to the pressure field stands in the two last terms. At this state of the development, the polymeric stress tensor needs to be known to obtain the pressure field. We choose to solve the Modified Phan-Thien and Tanner model.