SCHEMES FOR APPLYING ACTIVE LUBRICATION TO MAIN ENGINE BEARINGS

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Abstract. The work presented here is a theoretical study that describes two different schemes for the oil injection system in actively lubricated main engine bearings. The use of active lubrication in journal bearings helps to enhance the hydrodynamic fluid film by increasing the fluid film thickness and consequently reducing viscous friction losses and vibrations. One refers to active lubrication when conventional hydrodynamic lubrication is combined with dynamically modified hydrostatic lubrication. In this case, the hydrostatic lubrication is modified by injecting oil at controllable pressures, through orifices circumferentially located around the bearing surface. The pressure distribution of the hydrodynamic fluid film in journal bearings is governed by the Reynolds equation, which is modified to accommodate the dynamics of active lubrication, and which can be numerically solved using finite-difference method. The computed bearing fluid film forces are coupled to the set of nonlinear equations that describes the dynamics of the reciprocating engine, obtained with the help of multibody dynamics (rigid components) and finite elements method (flexible components). The main equations that govern the dynamics of the injection for a piezo-actuated oil injector and a mechanical-actuated oil injector are presented in this study. It is shown how the dynamics of the oil injection system is coupled to the dynamics of the bearing fluid film through equations. The global system is numerically solved using as a case of study a single-cylinder combustion engine, where the conventional lubrication of the main bearing is modified by applying radial oil injection using piezo-actuated injection. The performance of such a hybrid bearing is compared to an equivalent conventional lubricated bearing in terms of the maximum fluid film pressures, minimum fluid film thicknesses and reduction of viscous friction losses.

Keywords: fluid film bearings, active lubrication, machinery dynamics, mechatronics

1. INTRODUCTION

The reduction of energy waste and the improvement of efficiency in vehicles have been a matter of increased concern, being a focus of important research and development particularly during the last three decades. In order to improve functionality, safety, economy and comfort in vehicles, significant developments have been also made, which can be partially reflected by the number of mechatronic systems implemented in today’s cars (Isermann, 2006). However and despite all the progress made until now the energy losses produced by friction and heat are still kept in a significant percent. Some studies have shown that for a light duty vehicle, the frictional losses in the engine could amount to as much as 48% of the total energy consumption. Frictional losses associated with the bearings and crankshaft alone could amount to 15% of the total energy consumption (Spearot, 2000). Moreover, the performance of the bearings affects key characteristics of engines, such as durability, noise and vibrations. In order to address this problem, most of the attention has been put on the use of new lubricants, new composite materials and new geometric bearing designs, but still using conventional lubrication methods. Against this background, the present study aims to show that by combining conventional hydrodynamic lubrication with controllable radial oil injection (active lubrication), significant improvements may be obtained in the lubrication performance of crankshaft journal bearings (i.e. also known as main bearings), in terms of reduction of friction losses and vibrations. In this case, the conventional lubrication is modified by injecting oil through orifices distributed circumferentially along the bearing surface at specific angular positions (hole-entry type bearings), and where the rules for controlling the oil injection pressures are given as a function of the instantaneous crank angle. In order to actively inject the lubricant oil into the bearing, two schemes are described in this work: a piezo-actuated injection system and a mechanical injection system. Considering the main bearing of a single cylinder medium-size combustion engine, in the last section of this paper, some numerical results are presented for the bearing operating with different oil injection control rules, using a piezo-actuated injector. The results are evaluated in terms of the resulting behaviour of the oil film thickness, the viscous friction losses and the journal vibrations. By applying controllable radial oil injection to main bearings it is expected: a) to reduce the friction losses by increasing the fluid film thickness; b) to reduce vibrations (i.e. result in smaller journal orbits); and c) to increase the effective carrying load area by modifying the pressure distribution profile, which can make it possible to use bearings of smaller dimensions with similar load-carrying capacity.
2. DYNAMICS OF MAIN ENGINE BEARINGS

In a typical combustion engine, the crankshaft is supported by journal bearings, as shown in Fig. 1a. The tribology of journal bearings in an engine is affected by factors such as lubricant supply, thermal effects, dynamic loading and elasticity of the bounding solids (Tung and McMillan, 2004). The loads acting on main engine bearings are mainly coming from combustion forces transmitted from the cylinders to the crankshaft and inertia loads. Since these forces change as a function of the crank angle, the thin lubricant films in main bearings are under a dynamic loading condition. Hence, the journal eccentricity $e(t)$, and the attitude angle $\phi(t)$ (see Fig. 1b), vary through the loading cycle, consequently, the oil pressure distribution and likewise the carrying load capacity of the bearing dynamically change. The analysis of the bearings can be done considering them either as rigid bodies or flexible bodies. Since in the present study is more relevant to evaluate the performance of a bearing operating with radial oil injection than to estimate the elastic deformations of the mechanical components, fluid film theory based on hydrodynamic lubrication conditions is used for describing the dynamics of the bearing fluid film. Thus, the governing equation for the fluid film pressure distribution in dynamically loaded journal bearings is obtained from the general formulation of Reynolds equation (Hamrock, 1991).

2.1 Dynamics of radial oil injection

The bearing considered in this study is a hole-entry type bearing with orifices disposed circumferentially along two rows, as shown in Fig. 1b. In these type of bearings, active lubrication is achieved by injecting oil at controllable pressures $P_{inj}(t)$, through radial orifices. The governing equation for the pressure distribution of a dynamically-loaded and actively-lubricated journal bearing, assuming an isoviscous, Newtonian, incompressible fluid operating in laminar regime, is given by Eq. (1) (Santos and Russo, 1998). When the radial oil injection is not operating (i.e. $P_{inj} = 0$ and $F_i(\varphi, z) = 0$), Eq. (1) transforms into the conventional Reynolds equation for hydrodynamic lubrication. Equation (1) can be numerically solved for the pressure distribution ($p(\varphi, z)$) using finite differences, provided that the normal squeeze velocity ($\dot{e}$) and the rotational velocities ($\dot{\phi}$ and $\Omega$) are known from the dynamics of the mechanical system. After solving Eq. (1), the fluid film bearing forces can be calculated by integrating the pressure distribution over the bearing surface. The instantaneous power dissipation and the cyclic averaged power consumption due to the viscous friction forces, can be computed using Eqs. (2) and (3).

\[
W_f = \frac{\Omega \cdot r_h \cdot f_{vf}}{2\pi} \\
(W_f)_{avg} = \frac{1}{2\pi} \int_0^{2\pi} W_f d\theta
\]  

where the viscous friction forces ($f_{vf}$) are obtained by integrating the shear stress around the journal surface using:

\[
f_{vf} = \int_s \lambda^*_y r_h d\varphi dz, \text{ and } \lambda^*_y = \frac{\alpha_s \Omega}{h} + \frac{h}{2\pi} \frac{dp}{d\varphi}.
\]

2.2 Coupling of fluid films to the dynamics of the engine

The dynamics of the actively modified hydrodynamic fluid film is coupled to the multibody dynamic model of a single-cylinder combustion engine, by incorporating at each time step $t$, the instantaneous bearing fluid film forces into the model. The multibody dynamic model of the single-cylinder combustion engine, which is fully described in (Estupinan and Santos, 2008), was developed based on Newton-Euler’s method, and includes the dynamics of the main components of the reciprocating mechanism (piston/connecting rod/crank) and joint constraints. Due to the constraints, the cylinder combustion forces and the bearing fluid film forces, the global mathematical model exhibits a very non-linear behaviour, therefore,
numerical methods have to be used for the solution. In this case, the solution of the nonlinear system of equations is obtained by using a direct iterative method combined with the implicit Newmark method and a predictor-corrector approach (Garcia de Jalon and Bayo, 1994). Using the same bearing geometric properties used in a previous study by Estupinan and Santos (2008), the pressures distributions obtained for a conventionally and hybridly lubricated main bearing are shown in Fig. 2. The fluid film pressure distributions are plotted at three angular positions of the crankshaft (180°, 360° and 540°). In this case, the hybrid lubrication conditions are introduced by injecting oil at a controllable pressure through orifices located at Θ = 180° and Θ = 225°, where Θ is the angle measured in clockwise direction from the X axis, as shown in Fig. 1b.

3. SCHEMES FOR APPLYING RADIAL OIL INJECTION

In order to advantageously modify the hydrodynamic fluid film, by applying controllable radial oil injection (i.e. hybrid lubrication), suitable positions of the injection orifices along the bearing surface and appropriate control rules for the oil injection pressures have to be derived, which has been investigated in previous studies (Estupinan and Santos, 2009a,b). Thus, the curves of the loads acting on the bearings, the curves of the combustion forces inside the cylinders, and the plots that represent the performance of the bearings under conventional hydrodynamic lubrication are important sources of information when attempting to optimize the oil injection. Although in those cited previous studies appropriate locations for the injection orifices and convenient rules for changing the oil injection pressures as a function of the instantaneous crankshaft angle were found, details related to the oil injection system were not discussed.

In this section, two alternatives for applying the oil injection are presented: a) a piezo-actuated injection system; and b) a mechanical unit injection system, as schematically illustrated in Figs. 3a and 3b. These two types of oil injector are proposed based on similar working principles of those type of injectors used in fuel injection systems, described in more detail in (Lee et al., 2002; Lino et al., 2007; Chung et al., 2008), but with lower injection pressures required. Concerning to the modelling of oil injection systems, the focus has been mainly put on the derivation of the equations that govern the dynamics of the injector and their coupling to the fluid dynamics of the hybridly lubricated main bearing, presented in the previous section.

3.1 Piezoelectric injection system

An schematic diagram of a piezo-actuated injector is shown in Fig. 3a. In this type of injector, the injection is controlled by the activation of a piezoelectric actuator. If the piezoelectric stack is activated by a voltage signal, the control valve opens, which results in a pressure drop in the control chamber and subsequently the opening of the injection nozzle due to the lift of the needle valve, starting the injection. When the feeding voltage applied to the piezoelectric stack is deactivated, the control valve returns to the closed position, the servopiston and needle valve move downwards and consequently the injection ends. For the simplicity of the modelling, a constant oil supply pressure (P_o) is assumed and the dynamics of the oil supply pump are not included. The main equations for the modelling of the piezoelectric actuator and the hydraulic and mechanical subsystems of the injector are presented next.

Modelling of the piezoelectric stack: Considering that a stack actuator has several layers of piezoelectric material, the number of layers (N_{pz}), with each layer of an area of \( A_{pz} \) and thickness \( d_{pz} \), the resultant piezo-stack force (\( F_{pz} \)) and electric charge can be derived from the constitutive equations for piezoelectric materials (Feenstra et al., 2008):
injector (see Fig. 3a), one obtains:

\[ F_{ps}(t) = -k_{33}x_{ps}(t) + k_{33}d_{33}N_{pz}v_{pz}(t) \]  
\[ Q_{ps}(t) = k_{33}d_{33}x_{ps}(t) + \epsilon_{33}N_{pz}E_{pz}^{-1}v_{pz}(t) \]

where \( x_{ps}(t) \) is the total displacement of the piezo-stack; \( k_{33} \) is the elastic stiffness of the piezo-stack; \( d_{33} \) is the piezoelectric coupling coefficient; \( v_{pz} \) is the feeding voltage; \( E_{pz} \) is the elastic modulus of the piezo-stack; and \( \epsilon_{33} \) is the dielectric permittivity.

**Modelling of hydraulic subsystems:** Considering compressibility of the oil, the pressure change in each one of the control volumes \( v_i \) of the hydraulic subsystems: control valve \((cv)\), control chamber \((cc)\) and injector valve\((iv)\), see Fig. 3a, can be calculated by:

\[
\dot{P}_{vi} = -\frac{\beta}{V_{vi}} \left( \dot{V}_{vi} - Q_{vin} + Q_{vout} \right)
\]

where for each control volume, \( \beta \) is the bulk modulus of elasticity; \( V \) is the instantaneous volume of the fluid; \( \dot{V} \) accounts for the volume change due to the motion of mechanical parts. \( Q_{vin} \) and \( Q_{vout} \) are the intake and outtake flows through each control volume, which can be calculated by applying the energy conservation law:

\[
Q = sgn(\Delta P) \cdot C_d \cdot A_o \cdot \sqrt{\frac{2}{\rho}} |\Delta P|
\]

where for each control volume at the inlet and outlet section, respectively: \( C_d \) is a discharge coefficient; \( A_o \) is the orifice section; \( \Delta P \) is the oil pressure difference across the section \( A_i \); and \( \rho \) is the oil density. The discharge coefficient \( C_d \) accounts for the flow losses through orifices and it is usually obtained experimentally. In the literature there is a wide spread of values for discharge coefficients, however, for short and sharp-edged orifices a value of \( C_d = 0.6 \) is commonly assumed for numerical simulations (Merritt, 1967). Applying Eqs. (6) and (7) to the hydraulic subsystems of the piezo-actuated injector (see Fig. 3a), one obtains:

**Control valve \((cv)\):**

\[
\dot{P}_{cv} = -\frac{\beta_{cv}}{V_{cv}} \left( V_{cv} - sgn(P_{ce} - P_{cv})C_{dv_{cv}} A_{cv_{in}} \sqrt{\frac{2}{\rho}} |P_{ce} - P_{cv}| + sgn(P_{ce} - P_{cv})C_{dv_{cv}} A_{cv_{out}} \sqrt{\frac{2}{\rho}} |P_{ce} - P_{cv}| \right)
\]

**Control chamber \((cc)\):**

\[
\dot{P}_{cc} = -\frac{\beta_{cc}}{V_{cc}} \left( V_{cc} - sgn(P_s - P_{cc})C_{dv_{cc}} A_{cc_{in}} \sqrt{\frac{2}{\rho}} |P_s - P_{cc}| + sgn(P_s - P_{cc})C_{dv_{cc}} A_{cc_{out}} \sqrt{\frac{2}{\rho}} |P_s - P_{cc}| \right)
\]

**Injector valve \((iv)\):**

\[
\dot{P}_{inj} = -\frac{\beta_{iv}}{V_{iv}} \left( V_{iv} - sgn(P_s - P_{inj})C_{dv_{iv}} A_{iv_{in}} \sqrt{\frac{2}{\rho}} |P_s - P_{inj}| + sgn(P_{inj} - P_{ib})C_{dv_{iv}} A_{iv_{out}} \sqrt{\frac{2}{\rho}} |P_{inj} - P_{ib}| \right)
\]
where $P_{cv}$ is the pressure in the control valve; $P_{cc}$ is the pressure in the control chamber; $P_s$ is the supply pressure; $P_t$ is the return pressure; $P_{nj}$ is the injection pressure; $P_{jb}$ is the hydrodynamic journal pressure computed at the bearing orifice, which is obtained from the solution of the modified Reynolds equation (Eq. (1)). In other words, for each one of the $i^{th}$ oil injection orifices, $P_{jb}$ would be equal to $p$ evaluated at the coordinates $\Theta_{o_i}, a_{o_i}$, illustrated in Fig. 1b. Since the value of $P_{jb}$ depends on the dynamics of the journal bearing fluid film, its value must be updated for each time step in Eq. (10).

### Modelling of mechanical subsystems

In order to model the main mechanical parts of the piezoelectric injector, they are grouped in two mechanical subsystems $s1$ and $s2$. The first mechanical subsystem consists of the push rod, valve piston, control valve and piezoelectric stack, and it is dynamically described by Eq. (11). The second subsystem includes the control piston and the needle valve, and it is described by the Eq. (12).

\begin{align}
 m_s x_{cv1} &= F_{pv} - k_s x_{cv} - c_s x_{cv} - A_{cc} P_{cc} \\
 m_s x_{cv2} &= -k_s x_{cv1} - c_s x_{cv1} - A_p P_{cc} + (A_{nv} - A_{ns}) P_s
\end{align}

where $m_{s1}$ is given by the equivalent mass of the control valve cluster and the piezoelectric stack; $m_{s2}$ is given by the total mass of the needle valve and control piston; $c_s$ is the viscous friction coefficient of the valve piston; $k_s$ is the combined stiffness of the push rod and valve springs; $c_s$ is the viscous friction coefficient of the needle injector; $k_s$ is the servo piston spring constant; $A_{nv}, A_{ns}, A_p, A_{cc}$ are the sectional area of the needle valve, needle valve seat, servo piston and control valve respectively, as shown in Fig. 3a.

#### 3.1.1 A simplified model of piezo-injector

The modelling of the injector can be simplified by considering it as a control volume without considering the dynamics of the piezo-stack and control chambers. It can be also assumed that the volume of the injector chamber remains constant and that the injection section area switches between zero and its maximum, neglecting opening and closing transients of the injection orifices. Hence, the instantaneous pump volume and the rate of change of volume due to piston motion can be calculated by knowing the position of the piston with respect to the camshaft angle - $z_p(\theta_{cs})$, which is known from the cam profile. Consequently, the instantaneous pump volume and the rate of change of pump volume as a function of piston motion are given by:

\begin{align}
 V_e &= V_{e0} - A_c \cdot z_p(\theta_{cs}) \\
 \dot{V}_e &= A_c \cdot \dot{z}_p = A_c \cdot \Omega_{cs} \cdot \frac{dz_p}{d\theta_{cs}}
\end{align}

where $A_c$ is the cylinder bore; $z_p$ is the piston instantaneous axial displacement; $V_{e0}$ is the cylinder total volume; $\Omega_{cs}$ is the camshaft speed.

### 3.2 Mechanical injection system

An schematic diagram of a mechanical injection system is shown in Fig. 3b. In this type of injector the injection is basically controlled by an inline cam-pump system, where the design of the cam profile determines the oil injection events. The governing equations for the dynamics of the hydraulic subsystems (i.e. needle injector valve and pump) can be deduced from Eqs. (6) and 7. In this case, the dynamics of the pump must be included in the model of the injector since the oil supplied to the injector (flow and pressure) strongly depends on the performance of the pump. The governing equation for the pressure change in the pump is given by:

\begin{align}
 \dot{P}_p &= -\frac{\beta_P}{V_e} \left( V_e - sgn(P_p - P_{nj}) C_{in}^n A_{in}^n \sqrt{\frac{2}{\rho}} |P_p - P_{nj}| + sgn(P_{dv} - P_p) C_{out}^n A_{out}^n \sqrt{\frac{2}{\rho}} |P_{dv} - P_{jb}| \right)
\end{align}

The change of volume due to piston motion can be calculated by knowing the position of the piston with respect to the camshaft angle - $z_p(\theta_{cs})$, which is known from the cam profile. Hence, the instantaneous pump volume and the rate of change of pump volume as a function of piston motion are given by:

\begin{align}
 V_e &= V_{e0} - A_c \cdot z_p(\theta_{cs}) \\
 \dot{V}_e &= A_c \cdot \dot{z}_p = A_c \cdot \Omega_{cs} \cdot \frac{dz_p}{d\theta_{cs}}
\end{align}

where $A_c$ is the cylinder bore; $z_p$ is the piston instantaneous axial displacement; $V_{e0}$ is the cylinder total volume; $\Omega_{cs}$ is the camshaft speed.

### 4. NUMERICAL IMPLEMENTATION AND RESULTS

The numerical implementation and the results described in this section are for the main journal bearing of a single-cylinder medium-size combustion engine, operating with controllable radial oil injection and piezo-actuated injectors. All the equations that describe the mathematical model of the piezo-actuated injector can be written in space state form as: $\dot{x} = f(x, \psi_{pi})$, where the state vector is given by $x = [Q_{pv}, x_{cv}, \dot{x}_{cv}, P_{cc}, x_{nv}, \dot{x}_{nv}, P_{vi}]$ and the input signal is given by the driving voltage and its derivative acting on the piezo-stack: $\psi_{pi} = [V_{pv}, \dot{V}_{pv}]$. The equations of the dynamics of the injector and the system of equations that describes the dynamics of journal bearing and reciprocating mechanism (see section 2.2) have to be simultaneously solved, since the oil injection pressures ($P_{nj}$) are influenced by the hydrodynamic pressure of the fluid film at the bearing side of the orifices ($P_{jb}$), as seen in Eq. 10. At the same time, the updated values of $P_{nj}$ have to be introduced into Eq. 1 for each time step.
conventional hydrodynamic lubrication, \( P_{\text{inj}} = 0 \)

(b) \( \Theta_{\text{inj}} = 180^\circ \)
\[ P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 \]

(c) \( \Theta_{\text{inj}} = 210^\circ \)
\[ P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 \]

(d) \( \Theta_{\text{inj}} = 225^\circ \)
\[ P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 \]

(e) \( \Theta_{\text{inj}} = 180^\circ \)
\[ P_{\text{inj}} = 12 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 5 \]

\( \Theta_{\text{inj}} = 225^\circ \)
\[ P_{\text{inj}} = 12 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 5 \]

The results obtained for the bearing operating with conventional lubrication are compared to the results obtained when the bearing operates with radial oil injection using four different strategies for controlling the injection pressures as a function of the instantaneous crankshaft angle, as seen in Tab. 1. For the piezo-actuator, a square voltage signal was assumed to be applied to the piezo-stack, considering maximum voltage of \( v_{\text{max}} = 100V \) during the injections, and minimum voltage equal to zero. The main geometric properties of the bearing considering in the numerical simulations are the same as in (Estupinan and Santos, 2008).

In order to illustrate the injector behaviour in terms of the movement of the needle and the variation of the injection pressure as a function of crank angle, Fig. 4 shows the results obtained for case (b), after solving the global system of equations. It can be observed that the main role of the injector is played when the needle moves upwards, opening the outlet orifice of the nozzle which is connected directly to the bearing orifice, as shown in Fig 3a. Thus, the high oil pressure is injected to the main bearing only during a portion of time of every cycle, in order to contribute with the conventional hydrodynamic lubrication and to increase its load-carrying capacity. The minimum film thickness and maximum pressure for cases (b) and (e) are plotted in Fig. 5. It can be seen in this figure a significant increase in the minimum film thickness compared to the bearing of case (a). The performance of the bearing for all the cases in terms of different parameters can be compared from the graphs in Fig. 6. Plots of the lowest film thicknesses, maximum fluid film pressures, journal vibrations and cyclic averaged power consumption due to the viscous frictional forces are included in Fig. 6. It can also be seen that the bearing of case (e) has the best bearing performance compared to the other cases analyzed.

5. CONCLUSIONS

This paper has shown that the lubrication performance of main engine bearings can be enhanced by combining conventional hydrodynamic lubrication with controllable radial oil injection. The numerical results presented in this paper were for the main bearing of a single-cylinder combustion engine operating with active lubrication conditions, by using different strategies for controlling the oil injection pressures, and by using a piezo-actuated injection system. It was shown from the results that the fluid film thickness can be significantly increased and at the same time the friction losses and vibrations can be reduced. Two different schemes for injecting the oil into the bearing were described, coupling by equations the dynamics of the injectors to the dynamics of the bearing. However, further studies should be carried out in order to experimentally evaluate the oil injection systems presented in this work. Although the numerical simulations were carried out considering the main bearing of a single-cylinder combustion engine, similar oil injection systems may be applied to main bearings of multiple cylinder combustion engines and in general to other type of reciprocating machines using fluid film bearings, in order to enhance their tribological performance.

Table 1: Cases considered in the study - controllable lubrication.

<table>
<thead>
<tr>
<th>Case</th>
<th>( \Theta_{\text{inj}} )</th>
<th>Rules for controlling the radial oil injection (pressures in MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>( 180^\circ )</td>
<td>conventional hydrodynamic lubrication, ( P_{\text{inj}} = 0 )</td>
</tr>
<tr>
<td>(b)</td>
<td>( 180^\circ )</td>
<td>( P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 )</td>
</tr>
<tr>
<td>(c)</td>
<td>( 210^\circ )</td>
<td>( P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 )</td>
</tr>
<tr>
<td>(d)</td>
<td>( 225^\circ )</td>
<td>( P_{\text{inj}} = 15 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 6 )</td>
</tr>
<tr>
<td>(e)</td>
<td>( 30^\circ )</td>
<td>( P_{\text{inj}} = 12 \text{ if } 320^\circ \leq \theta_{\text{inj}}(t) \leq 420^\circ; \text{ otherwise } P_{\text{inj}} = 5 )</td>
</tr>
</tbody>
</table>

Figure 4: (a) Displacement of the injector needle; (b) Oil injection Pressure - case (b)

Figure 5: (a) Minimum film thickness; (b) Maximum pressure - controllable lubrication (\( \Omega = 3000 \text{ rpm} \)).
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7. REFERENCES


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