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Original
An Overview of Engine Components Friction Modeling / Delprete, C., Razavykia, A.. - ELETTRONICO. - (2018).

Availability:
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Publisher:
6th Workshop of Associazione Italiana di Tribologia (AIT) “Tribologia e Industria”, 18-19 April 2018,

Published
DOI:

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11 November 2026

An Overview of Engine Components Friction Modeling

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***Abstract* – Development of efficient internal combustion engines (ICEs) is receiving wide attention due to increasing environmental legislation limits and fuel and emission costs. In order to improve ICEs performance, it is vital to gain insight of lubrication mechanism and tribological performance of engine reciprocating and rotating components. Analytical modeling of engine components such as piston ring assembly (PRA), connecting rod big eye and main journal bearings tribological behavior widely accepted as reliable tool by researchers and car industries. The principle aim of this paper is to briefly describe technical aspects and governing equations as well as numerical approaches apply to simulate the tribological behavior of PRA and connecting rod big eye.**

Keywords – Engine, Piston ring, skirt, modeling

I. INTRODUCTION

Efficiency improvement in internal combustion engines (ICEs) motivates researchers and car industries to make effort to adopt reliable approach to evaluate and compare the basic design configurations and determine important parameters that influence the performance. Due to some drawbacks associated by experimental observation, such as high cost of facilities and being time consuming, analytical modeling becomes reliable tool to evaluate the engine tribological performance.

Analytical and numerical simulation of engine component lubrication characteristics has prime importance that can be applied by designers and companies to understand the contribution of each engine component in overall tribological performance and how and where they must consider the modifications [1, 2]. Piston ring pack and piston skirt which are known as piston ring assembly (PRA) and connecting rod big eye bearing have largest contribution in engine power losses. The devoted analytical studies to engine friction losses can be grouped into two distinct categories; first, empirical models which are based on correlations between influential parameters that are realized by experimental investigation and second one, instantaneous friction models based on lubrication theory and solving Reynolds equation to define pressure field developed by lubricant and consequently calculation of friction losses [3].

The main principle aim of the present paper is to share an overview of constitutive equations and approaches to simulate the instantaneous PRA and connecting rod big eye bearing tribological behavior.

II. PISTON RING ASSEMBLY FRICTION MODELING AND LUBRICATION REGIMES

Lubrication regimes are grouped in four distinct mechanisms with especial characteristics that affect the system performance such as load capacity, thermal and mechanical deformation of bearing bodies and tribological behavior. All loaded reciprocating and rotating bodies enjoy these lubrication regimes when they come to service with respect to operating conditions. In hydrodynamic lubrication (HL), or full film lubrication, the lubricant film is sufficiently thick to sustain loads and to separate the rotating and sliding bodies. Elasto-hydrodynamic lubrication (EHL) is a type of hydrodynamic lubrication associated with significant elastic deformation of coupled surfaces; it extensively alters the shape of the film thickness profile. Mixed lubrication is the transition region between EHL and boundary regimes in which the load is sustained by both lubricant film and asperity contacts. Mixed lubrication occurs at the presence of high load, low speed, or low viscosity due to high temperature. Boundary lubrication deals with conditions where lubricant film is thin and the load is mainly or completely supported by asperity contacts [2]. Piston ring assembly (PRA) friction modeling consists of piston ring pack friction and piston skirt friction. Piston ring pack can be modeled under hydrodynamic, elasto-hydrodynamic, mixed and boundary lubrication. Piston skirt lubrication mechanism can be considered whether hydrodynamic or elasto-hydrodynamic regimes.

A. Piston Ring Pack Friction Modeling

The following assumptions were made during the modeling:

- Ring is fully engulfed and there is no cavity within the oil film thickness and ring experiences HL;
- Oil film thickness is circumferentially uniform;
- Lubricant is Newtonian and incompressible;
- Thermal and elastic deformation of ring and liner are neglected;
- Oil viscosity is constant;

The piston is treated as dynamically loaded reciprocating bearing, considering sliding and squeeze action. Due to circumferential uniform oil film thickness, 1D Reynolds equation (1) is employed to find hydrodynamic pressure at ring face and liner interface.

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=-6\eta U\frac{\partial h}{\partial x}+12\eta\omega\frac{\partial h}{\partial \theta} \qquad (1)$$

where h is the nominal oil film thickness (m), x the coordinate along the cylinder axis (m), ω the engine spin speed (rad/s), θ the crank angle (rad), p the hydrodynamic pressure (Pa), η the oil dynamic viscosity (Pa·s), and U the instantaneous piston velocity (m/s). The ring motion in the piston groove is negligible in comparison to piston primary

motion, therefore same speed of piston can be considered for the ring:

$$U = \omega r\left[\frac{\cos\theta(\Lambda\sin\theta-\delta)}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}+\sin\theta\right] \qquad (2)$$

where r is the crank radius (m), δ the overall offset including wrist pin and crankshaft offset (m), and Λ the elongation ratio (ratio between crank radius and connecting rod length). In order to define axial coordinates, piston is considered stationary and liner reciprocates but in opposite direction. Oil film thickness at ring/liner interface which is function of time or crank angle and ring profile (Figure 1) can be written as following [4]:

$$h(x,\theta) = h_{\min}(\theta) + \frac{c}{(\frac{b}{2}+o)^2}(x-o)^2 \qquad (3)$$

where h_{min} is minimum oil film thickness (m) varies with time, c the ring crown height (m), o ring offset (m).

With two times integration from (1), the hydrodynamic pressure at ring/liner interface (4) can be defined and in to define integration constants, full Sommerfeld boundary conditions can be applied. Based on full Sommerfeld, there is no rupture within oil film thickness and ring face completely engulfed by oil and the boundary pressures are pressure at the upper and lower edges of the ring (Figure 1) [5].

$$p = -6\eta U I_0(x) + 12\eta\omega\frac{\partial h}{\partial\theta}I_1(x) + C_1 I_2(x) + C_2 \qquad (4)$$

where C_1 and C_2 are integration constants and:

$$I_0(x) = \frac{1}{2h_{\min}\sqrt{Bh_{\min}}}\tan^{-1}\left(x\sqrt{\frac{B}{h_{\min}}}\right) + \frac{x}{2h_{\min}\left(h_{\min}+Bx^2\right)}$$

$$I_1(x) = -\frac{1}{4B\left(h_{\min}+Bx^2\right)^2}$$

$$I_2(x) = \frac{3}{8h_{\min}^2\sqrt{Bh_{\min}}}\tan^{-1}\left(x\sqrt{\frac{B}{h_{\min}}}\right) + \frac{x\left(5h_{\min}+3Bx^2\right)}{8h_{\min}^2\left(h_{\min}+Bx^2\right)^2}$$

In order to calculate minimum oil film thickness, h_{min}, the problem is solved in steady-state that means the generated hydrodynamic force at ring/liner interface is in equilibrium with the gas force acting on the ring back side, and with the ring tension force at each crank angle:

$$b\left(p_{el}+p_{back}\right) - \int_{x_{in}}^{x_{out}} p\,dx = 0 \qquad (5)$$

where p_{el} is the ring elastic pressure (Pa), p_{back} the gas pressure (Pa) acting on the backside of the ring (note that $p_{back} = p_1$ if $p_1 > p_2$, $p_{back} = p_2$ and if $p_2 > p_1$).

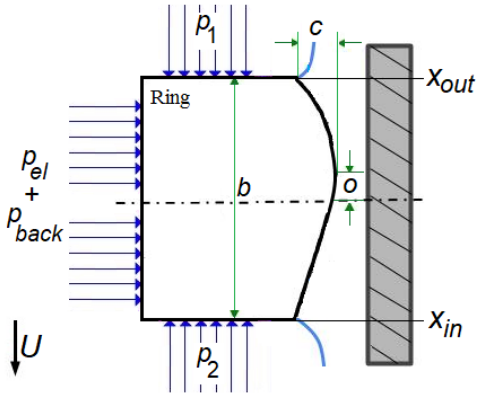

Figure 1. Schematic of piston ring profile and boundary pressures

The integral in (5) has analytical solution, but there are two unknowns, $h_{\min}$ and $\frac{dh}{dt}$ as well as the shortage of one equation with respect to number of unknowns. Therefore, numerical and iterative methods must be applied and an initial estimation of $h_{\min}$ where the film thickness is expected to change slightly must be made by elimination of $\frac{dh}{dt}$. Start from mid phase of stork where hydrodynamic lubrication is predominant, and squeeze action ($\frac{dh}{dt}$) can be ignored and calculate $h_{\min}$, at the next angle calculate $\frac{dh}{dt}$ using $h_{\min}$ estimated at previous angle, and then calculate $h_{\min}$ at current angle. Using this iterative approach and Newton-Raphson numerical method due to its fast convergence, it is possible to calculate $h_{\min}$ for whole engine cycle (Figure 2) and consequently calculate friction force and power losses applying following equations:

$$F_f = \int_0^{L_r}\int_{x_{in}}^{x_{out}}\left(\frac{h(x,\theta)}{2}\frac{dp}{dx} + \eta\frac{U}{h(x,\theta)}\right)dxdy \qquad (6)$$

$$P_f = \int_0^{L_r}\int_{x_{in}}^{x_{out}}\left(\frac{\eta}{h(x,\theta)}U^2 + \frac{h^3(x,\theta)}{12\eta}\left(\frac{\partial p}{\partial x}\right)^2\right)dxdy \qquad (7)$$

where L_r is the ring circumferential length.

Same approach can be applied to evaluate second ring and oil control ring tribological performance, but it is necessary to perform Blow-by analysis to determine inter ring gas pressure which are applied as boundary conditions for each ring as shown by Figure 3.

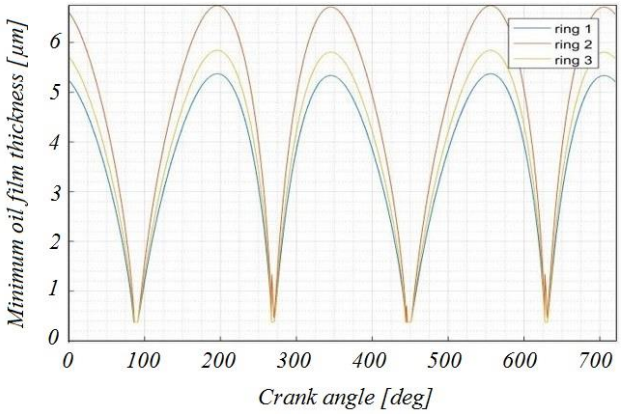

Figure 2. Minimum oil film thickness for 1 engine cycle with crank angle increment of 0.5 degree

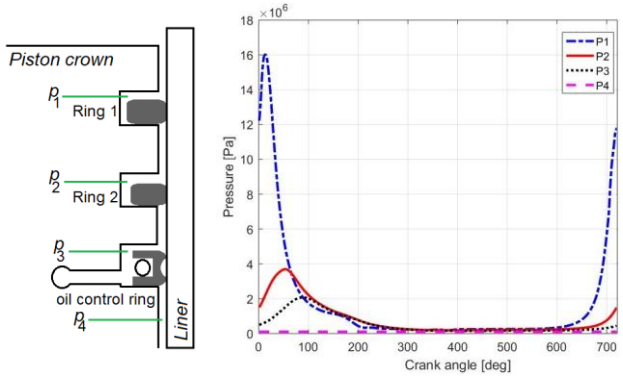

Figure 3. Inter-ring gas pressure analysis (Blow-by)

B. *Piston skirt*

Piston skirt is recognized as lower part of the piston containing wrist pin boss bearings, with the task of keeping piston from rocking excessively in the cylinder which is typically machined with small grooves to improve the lubrication of skirt/liner junction. Due to unbalanced forces and moments acting on plane perpendicular to wrist pin axis including thrust and anti-thrust sides, piston executes small translation and rotational motions within define clearance. These motions are known as piston secondary motion which affects the piston skirt tribological performance and defined by means of eccentricities at top (e_t) and bottom (e_b) of skirt [6] as shown by Figure 4.

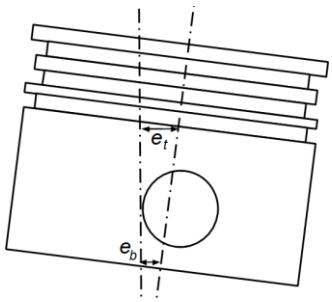

Figure 4. Eccentricities at the top and bottom of skirt

The following assumptions were made during the modeling:

- The piston skirt is fully engulfed by the oil and there is no cavitation;
- The oil is a Newtonian fluid;
- Thermal and elastic deformation of skirt and liner are neglected;
- There is full hydrodynamic lubrication at piston/liner interface.

In order to define the e_t and e_b, equilibrium equations of forces and moments about the wrist pin axis must be written and solved (8-10).

$$F_g + F_{ap} + F_{awp} + F_{cr}\cos\beta = 0 \quad (8)$$

$$F_h + F_{zwp} + F_{zp} + F_{cr}\sin\beta = 0 \quad (9)$$

$$M_h + M_f + M_{ip} - F_g z_{owp} + \\ - F_{ap}(z_{Gp} + z_{owp}) + F_{zp}(a-b) = 0 \quad (10)$$

These forces and moments are; gas force F_g, connecting rod force F_{cr}, forces and moment resulting from piston inertia (F_{ap}, F_{zp} and M_{ip}), wrist pin inertial forces (F_{awp} and F_{zwp}) and hydrodynamic force and moment (F_h and M_h) as illustrated by Figure 5 and can be calculated using equations 11 to 19.

$$F_h = \int_0^{l_{ps}} \int_0^{2\pi} p(x,\alpha) R\cos\alpha \, d\alpha dx \quad (11)$$

$$M_h = \int_0^{l_{ps}} \int_0^{2\pi} p(x,\alpha) R(a-x)\cos\alpha \, d\alpha dx \quad (12)$$

$$F_f = \int_0^{l_{ps}} \int_0^{2\pi} \left(\frac{h}{2}\frac{\partial p}{\partial x} + \eta\frac{U}{h}\right) R \, d\alpha dx \quad (13)$$

$$M_f = \int_0^{l_{ps}} \int_0^{2\pi} \left(\frac{h}{2}\frac{\partial p}{\partial x} + \eta\frac{U}{h}\right) R^2\cos\alpha \, d\alpha dx \quad (14)$$

$$F_{ap} = -m_p\ddot{x} \quad (15)$$

$$F_{zp} = -m_p[\ddot{e}_t + \frac{b}{l_{ps}}(\ddot{e}_b - \ddot{e}_t)] \quad (16)$$

$$F_{awp} = -m_{wp}\ddot{x} \quad (17)$$

$$F_{zwp} = -m_{wp}[\ddot{e}_t + \frac{a}{l_{ps}}(\ddot{e}_b - \ddot{e}_t)] \quad (18)$$

$$M_{ip} = -J_p\frac{\ddot{e}_t - \ddot{e}_b}{l_{ps}} \quad (19)$$

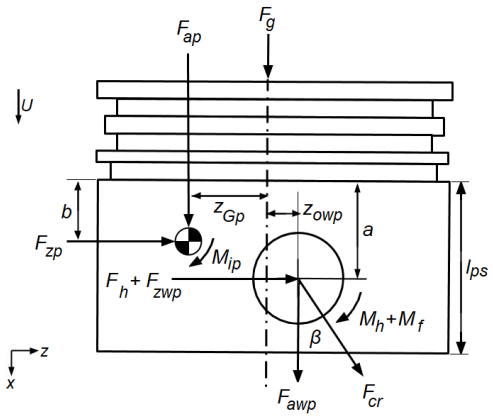

Figure 5. Piston free diagram

Substitution of equations from (11) to (19) in piston dynamic equations (8), (9) and (10) yields to the governing equation of piston secondary motion:

$$\begin{bmatrix} m_p(1-\frac{b}{l_{ps}}) + m_{wp}(1-\frac{a}{l_{ps}}) & m_p\frac{b}{l_{ps}} + m_{wp}\frac{a}{l_{ps}} \\ \frac{J_p}{l_{ps}} + m_p(a-b)(1-\frac{b}{l_{ps}}) & -\frac{J_p}{l_{ps}} + m_p(a-b)\frac{b}{l_{ps}} \end{bmatrix} \quad (21)$$

$$\begin{Bmatrix} \ddot{e}_t \\ \ddot{e}_b \end{Bmatrix} =$$

$$= \begin{Bmatrix} F_h + F_s \\ M_h + M_s \end{Bmatrix}$$

where F_s and M_s are summation of forces and moments about the wrist pin axis.

Iterative method and Newton-Raphson can be adopted to calculate instantaneous eccentricities at top and bottom of the skirt. After defining eccentricities, oil film thickness extended over piston skirt can be calculated as following:

$$h(x,\alpha) = c + (e_t + (e_b - e_t)\frac{x}{l_{ps}})\cos\alpha \quad (22)$$

where c is the piston liner clearance, l_{ps} the skirt length and α the circumferential coordinate.

The hydrodynamic pressure distributed over the piston skirt can be determined using 2D Reynolds equation due to variation of oil film thickness axially and circumferentially:

$$\frac{\partial}{\partial x}(\frac{h^3}{\eta}\frac{\partial p}{\partial x}) + \frac{1}{R^2}\frac{\partial}{\partial \alpha}(\frac{h^3}{\eta}\frac{\partial p}{\partial \alpha}) = 6U\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t} \quad (23)$$

where R is piston radiuses.

Finite difference scheme can be applied to solve 2D Reynolds equation:

$$\frac{h^3_{i+\frac{1}{2},j}\frac{p_{i+1,j}-p_{i,j}}{\Delta x} - h^3_{i-\frac{1}{2},j}\frac{p_{i,j}-p_{i-1,j}}{\Delta x}}{\eta\Delta x} +$$

$$+\frac{1}{R^2}\frac{h^3_{i,j+\frac{1}{2}}\frac{p_{i,j+1}-p_{i,j}}{\Delta\alpha} - h^3_{i,j-\frac{1}{2}}\frac{p_{i,j}-p_{i,j-1}}{\Delta\alpha}}{\eta\Delta\alpha} \quad (24)$$

$$= 6U\frac{h_{i+\frac{1}{2},j} - h_{i-\frac{1}{2},j}}{\Delta x} + 12\frac{h^{t+\Delta t}_{i,j} - h^{t}_{i,j}}{\Delta t}$$

To determine the pressure value at each node of the grid, boundary conditions must be applied. In particular, crankcase pressure acts on upper and lower edges of skirt and circumferential pressure at reference points along a line parallel to the liner must be same, or $\partial p/\partial\alpha=0$. After defining the pressure distribution over the skirt (Figure 6), the friction force F_f can be calculated using (25).

$$F_f = -\int_0^{l_{ps}}\int_0^{2\pi}\left(\frac{h(x,\alpha)}{2}\frac{dp}{dx} + \eta\frac{U}{h(x,\alpha)}\right)Rd\alpha dx \quad (25)$$

Figure 7 shows the calculated friction force at piston skirt and liner interface at 2000 rpm under motored condition.

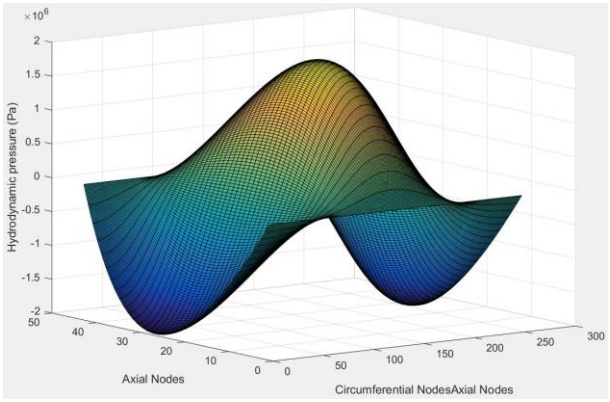

Figure 6. Hydrodynamic pressure over piston skirt (2000 rpm under motored condition)

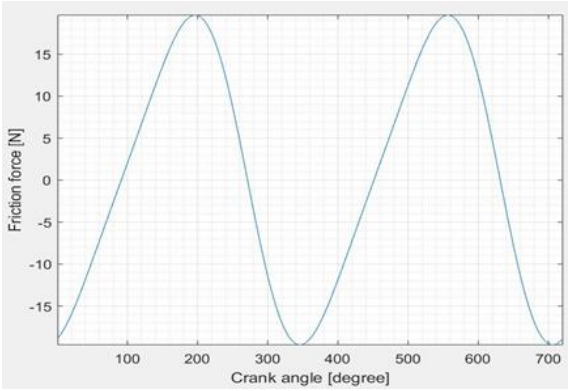

Figure 7. Friction force contributed by piston skirt

C. Connecting rod big eye bearing friction modeling

Connecting rod big eye bearing does not considered as part of PRA but can be recognized as one of the piston group component. 2D Reynolds equation (26) can be applied to determine the pressure distribution at big eye and pin junction.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{1}{R^2}\frac{\partial}{\partial \alpha}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial \alpha}\right)=6U\frac{\partial h}{R\partial \alpha}+12\frac{\partial h}{\partial t} \quad (26)$$

Oil film thickness at bearing surfaces interface is defined as:

$$h(x,\alpha)=c(1+\varepsilon\cos\alpha) \quad (27)$$

where c is the clearance and ε the ratio of eccentricity between bearing bodies axes to clearance ($\varepsilon=\frac{e}{c}$).

Finite difference can be applied to solve the Reynolds equation and nodes over the pin can be considered as shown by Figure 8.

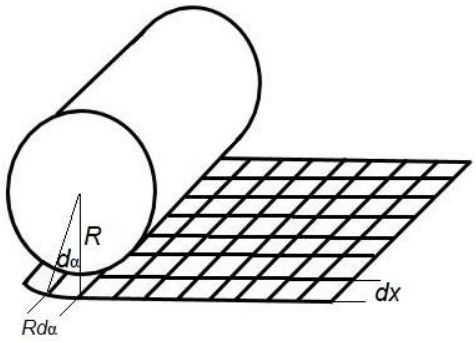

Figure 8. Finite difference mesh

As the forces acting on big eye bearing and pin are dynamic and vary by the time or crank angle, eccentricity between big eye axis, *o*, and pin axis, *o´* (Figure 9) also alters angle by angle.

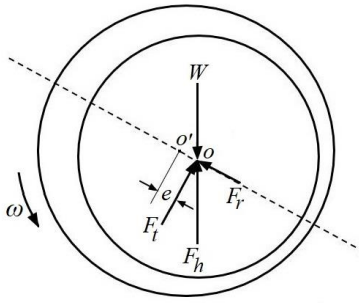

Figure 9. Eccentricity between journal and bearing

In order to estimate *e*, hydrodynamic force due to oil pressure at journal and bearing interface, F_h, which is the resultant force contributed by radial and tangential forces, F_r and F_t, equations (28) and (29) respectively, must be equal to force acting on pin, *W*, contributed by gas, connecting rod and rotational mass inertial forces.

$$F_r=\int_0^{\pi}\int_{-\frac{L}{2}}^{\frac{L}{2}} p(x,\alpha)R\cos\alpha dx d\alpha \quad (28)$$

$$F_t=\int_0^{\pi}\int_{-\frac{L}{2}}^{\frac{L}{2}} p(x,\alpha)R\sin\alpha dx d\alpha \quad (29)$$

where L is the connecting rod big eye bearing width and R the pin radius and *p* the pressure expression.

Hydrodynamic pressure expression, *p*, to solve equations (14) and (15) can be written as following [7-8]:

$$p=\frac{6\eta\left(\frac{1}{C^2}\right)\left(\frac{U}{2}\varepsilon\sin\alpha+\dot{\varepsilon}\cos\alpha\right)}{(1+\varepsilon\cos\alpha)^3}x^2+D_1x+D_2 \quad (30)$$

where D_1 and D_2 are the integration constant. To find the integration constant, boundary conditions at both edges of big eye bearing are crank case pressure. The obtained force equilibrium equation ($W - F_h = 0$) is a first order differential equation which can be solved using analytical and numerical methods. After calculation of centers eccentricity for whole engine cycle, hydrodynamic pressure at big eye bearing and pin interface can be determine as shown by Figure 10 (1000 rpm).

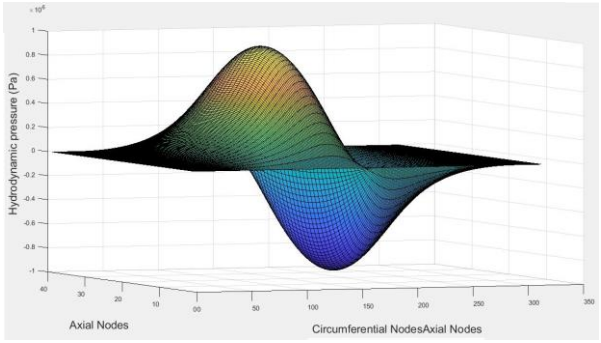

Figure 10. Hydrodynamic pressure profile

Friction force can be calculated for complete engine cycle using:

$$F_f=-\int_{-\frac{L}{2}}^{\frac{L}{2}}\int_0^{2\pi}\left(\frac{h(x,\alpha)}{2}\frac{dp}{Rd\alpha}+\eta\frac{U}{h(x,\alpha)}\right)Rd\alpha dx \quad (31)$$

III. CONCLUSION

Most power loss occurs at bearing and lubricated surfaces; piston ring assembly (PRA) and connecting rod big eye and

journal main bearings are the most important contributors to total frictional power loss in internal combustion engines. An overview of analytical formulation and numerical approaches to evaluate the piston ring assembly and connecting rod big eye bearing tribological performance is presented. General purpose analytical models s proposed to encourage the examination of engine components independently.

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