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ELASTOHYDRODYNAMIC PISTON SKIRT LUBRICATION- EFFECT ON TRIBOLOGICAL PERFORMANCES

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ABSTRACT

A model of elastohydrodynamic lubrication of piston skirt is developed in this paper. The secondary motion of the piston, the lubrication and the elastic deformations of skirt and cylinder are described by a transient strongly nonlinear system coming from the coupling of the dynamics equations of piston secondary motion and the Reynolds equation. The iterative Newton-Raphson method in conjunction with Murty's algorithm for cavitation was used to solve the problem. An optimum skirt curved profile, which maintain piston in optimum performance characteristics, is adopted. Using the tribological performance of the lubricated skirt-cylinder, the results - minimum oil-film thickness, maximum pressure in the lubricant film and friction- are compared to the elastohydrodynamic solution and the rigid skirt-cylinder solution. The effect of elastic deformation of both skirt and cylinder show that elastohydrodynamic analysis is necessary for an accurate prediction of a piston- cylinder performance. The computational work, applied to a solid skirt piston of a V-8 direct injection diesel engine truck, should be a powerful design analysis tool, which can be used to increase the available engine power through optimum piston performance characteristics.

INTRODUCTION

The elastohydrodynamic lubrication of pistons has been largely investigated in order to evaluate the available engine power through optimum performance characteristics of piston.
The major requirements for piston performance characteristics are closely related to the secondary movement in conjunction with the lubrication and of the manner in which the complex inter-relationships between skirt stiffness, clearance and profile have been interpreted and recognized in the development of the piston. Of these characteristics, the minimum oil-film thickness is important in estimating the piston-cylinder durability. The maximum pressure in the lubricant film determines the maximum operating stress in the piston-cylinder and the supporting structure. Friction in the system piston-cylinder which represents an important component of the total friction in engine affects the conservation of energy and then the vehicle fuel consumption. In elastohydrodynamic analysis, all these characteristics can be different from those obtained with rigid skirt or rigid cylinder. The elastic deformation due to pressure developed in the lubricant oil film produces a large gap compared to clearance. Know the effect of deformation of the surfaces onto these characteristics is the main issue to increase

the piston performance. Elastic skirt deformations are not considered in the first piston studies where hydrodynamic lubrication of piston skirt model are developped. [1-4]. The first analysis which considers elastic deformation on the skirt, is that presented by Oh and al. [5]. Friction, fluid-film pressure and minimum thickness are analyzed for two profiles in order to show the best profile in elastohydrodynamic lubrication of piston skirt. A very complete analysis of elastohydrodynamic lubrication of piston skirt was presented by Zhu and al. [6-7]. Based on the same formulation developed by Li, the effect of elastic deformation of piston and cylinder bore is considered on piston motion, lubrication and friction. Dursunkaya and al. [8] have developed an elastohydrodynamic model of skirt lubrication. The piston secondary dynamics, skirt lubrication and deformation problems are simultaneously solved. It is shown that the model can be used as a powerful design analysis tool to optimize many design parameters such as skirt profile, oil viscosity and others.

More recently, elastohydrodynamic studies based on the previous models continued to be developed. Balakrishnan and al. [9] presents detailed analysis of the conforming contact between a piston and cylinder liner for a high speed racing engine in order to show critical film lubrication which separate the piston skirt from the cylinder liner. The emphasis in the current investigation is to give detailed analysis of elastic deformation of contiguous bodies and geometric profile of mating members. Mansouri and al. [10] developed a piston secondary motion model to understand the detailed interactions between the piston skirt and the cylinder liner for various design parameters. The authors use a simple formulation considering only the tilt motion of the piston. The importance of piston skirt flexibility with respect to piston dynamics and lubrication regimes is shown.

The current paper describes an elastohydrodynamic lubrication model of solid skirt piston. The iterative Newton-Raphson method, in conjunction with the determination of the cavitation by the Murty algorithm, was used to solve the nonlinear system, which arises from the coupling of universal Reynolds equation with elasticity equations.

An optimum skirt profile [11] , which maintains piston in optimum performance characteristics, was adopted. This analysis tries, in a first step, to show the effect of elastic deformation of the skirt and cylinder on the secondary movement and the key performances characteristics of the skirt-cylinder bearing.

NOMENCLATURE

- a piston axis position (m)
- b position of the center of mass piston (m)
- c radial clearance (m)
- C_p wrist-pin offset (m)
- d_1, d_2 deformation dues to hydrodynamic pressure of the skirt and the cylinder respectively
- e_t , e_b eccentricities of the piston at the top and the bottom of the skirt (m)
- F load capacity of the hydrodynamic fluid film (N)
- F_G force due to combustion gas pressure (N)
- F_{IC} , $\overline{F}_{IC}$ inertia forces due to piston mass (N)
- F_{IP} , $\overline{F}_{IP}$ inertia forces due to wrist-pin and connecting small end mass (N)
- f_c f_s sliding friction of the cylinder and the skirt (N)
- $h(y)$ skirt profile parameter (m)
- h fluid-film thickness (m)
- I_{pist} piston inertia moment about its center of mass ($Kg.m^2$)
- L piston-skirt length (m)
- ℓ connecting-rod length (m)
- M hydrodynamic moment about wrist-pin (N.m)
- m_{pin} wrist-pin mass (Kg)
- m_{pist} piston and connecting small end masses (Kg)
- p hydrodynamic fluid-film pressure (Pa)
- P_s instantaneous power loss for the skirt (W)
- R piston radius (m)
- $\bar{r}$ engine half stroke (m)
- r filling ratio at cavitation point
- t time (s)
- U piston sliding velocity (m/s)
- y fluid-film axial coordinate measured from top of skirt (m)
- α piston tilt angle about wrist-pin (rad)
- ρ_c , ρ_f density of fluid-film in cavitation and active zone
- θ circumferential coordinate (rad)
- θ_1 Active pressure fluid-film angle of the skirt (rad)
- $\bar{\theta}$ crankshaft rotation coordinate
- μ lubricant viscosity (Pa.s)
- ω crankshaft rotational speed (s^{-1})

BASIC EQUATIONS

The piston skirt EHL problem is formulated by a nonlinear system which contains:

- The two strongly non linear second order differential equations of the unsteady piston secondary motion which is described by the two degrees of freedom $e_t(t)$ and $e_b(t)$. The dynamic equations are derived from the second Newton law applied on the piston body subjected, at each time of engine cycle, to the action of external loads and moments as shown in figure 1

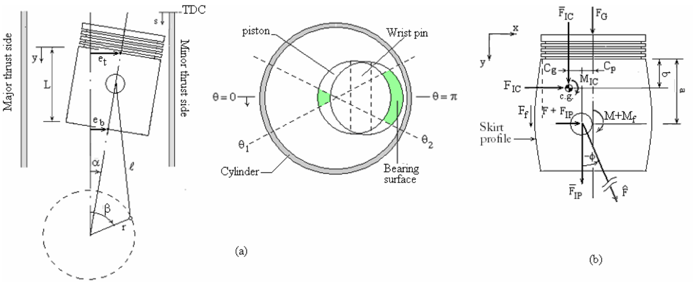

Fig. 1 Piston cylinder system (a) and forces and moments acting on the piston body

$$\begin{bmatrix} m_{pist}\left(1-\frac{b}{L}\right)+m_{pin}\left(1-\frac{a}{L}\right) & m_{pist}\frac{b}{L}+m_{pin}\frac{a}{L} \\ \frac{I_{pist}}{L}+m_{pist}(a-b)\left(1-\frac{b}{L}\right) & -\frac{I_{pist}}{L}+m_{pist}(a-b)\frac{b}{L} \end{bmatrix}\begin{Bmatrix}\ddot{e}_t \\ \ddot{e}_b\end{Bmatrix}$$

$$=\begin{Bmatrix} F+\left(F_G+\bar{F}_{IP}+F_{IC}+F_f\right)tg\phi \\ M+F_G C_P-\bar{F}_{IC}C_g+M_f \end{Bmatrix} \qquad (1)$$

-The universal Reynold equation which takes into account the cavitation in the fluid film

$$F\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu}\frac{\partial D}{\partial\theta}\right)+F\frac{\partial}{\partial y}\left(\frac{h^3}{\mu}\frac{\partial D}{\partial y}\right)=6U\frac{\partial h}{\partial y}+12\frac{\partial h}{\partial t}+$$

$$(1-F)\left(6U\frac{\partial h}{\partial y}+12\frac{\partial h}{\partial t}\right) \qquad (2)$$

Where

$$\begin{cases} D=p \\ F=1 \end{cases} \qquad D\geq 0 \text{ for active zones} \qquad (3)$$

$$\begin{cases} D=r-h \\ F=0 \end{cases} \qquad D\prec 0 \text{ for cavitation zone} \qquad (4)$$

Parameter r is the filling ratio or equivalent film thickness at cavitation point

$$r=h\frac{\rho_c}{\rho_f} \qquad (5)$$

This leads to the governing equation for cavitation zones

$$U\frac{\partial r}{\partial y}+2\frac{\partial r}{\partial t}=0 \qquad (6)$$

The lubricant film thickness is givenby

$$h=c+e_t\cos\theta+\frac{y}{L}(e_b-e_t)\cos\theta+f(y)+d_1+d_2 \qquad (7)$$

c is the nominal radial clearance between piston and cylinder bore, $f(y)$ is the piston skirt surface profile measured from the cylindrical reference surface, d_1 and d_2 are the elastic deformation of the skirt and the cylinder wall due to the pressure developed in the lubricant film.

From the piston motion, the skirt and cylinder frictional forces f_s and f_c are computed with the same expression for cavitation and active zones as below:

-Skirt friction

$$f_s=2R\int_0^L\int_0^\pi\left(\frac{h}{2}\frac{\partial p}{\partial y}+\frac{\mu U}{h}\right)\frac{r}{h}dyd\theta \qquad (8)$$

- Cylinder friction

$$f_c=2R\int_0^L\int_0^\pi\left(-\frac{h}{2}\frac{\partial p}{\partial y}+\frac{\mu U}{h}\right)\frac{r}{h}dyd\theta \qquad (9)$$

The instantaneous power loss for the skirt is given by

$$P_s=f_sU \qquad (10)$$

The same equation is written for the cylinder.

METHOD OF SOLUTION

Using a rectangular grid for the finite difference method to derive the Reynolds equation, the iterative scheme of Newton-Raphson is used to solve simultaneously the above dynamic equations and the universal Reynolds equation. The numerical solution is obtained at the i^{th} iteration from the precedent solution by

$$\begin{Bmatrix}\{p\} \\ e_t \\ e_b\end{Bmatrix}^{(i+1)}=\begin{Bmatrix}\{p\} \\ e_t \\ e_b\end{Bmatrix}^{(i)}-\left[J^{(i)}\right]^{-1}\{R\}^{(i)} \qquad (11)$$

[J] is the Jacobian matrix which dimension is $(n+2)(n+2)$, where $n=m_\theta m_y$, m_θ and m_y are respectively the total number points in circumferential and axial direction of the lubricant film grid.and $\{R\}$is the residual vector of equations (2-3)..

After the solution is obtained, a check is made to determine the zone of cavitation by solving the complementarity conditions

- $D\geq 0$ is satisfied in the active zone Ω_a otherwise the correspondent point is placed in the cavitation zone Ω_c and D is prescribed to zero
- $R\prec 0$ and $D=0$ are satisfied in the cavitated zone Ω_c otherwise the correspondent point is placed in the active zone Ω_a

The above procedure is repeated until the free boundary $\delta\Omega$ become stable. Then the trial decomposition is correct and the required solution can be found at this step time. This needs four or five iterations. Once the solution is obtained at this step Euler method is used to determine the solution at the next step, This process is repeated until the engine cycle is completed. The whole process is repeated for each engine cycle until convergence of the solution.

RESULTS

Effect of piston-cylinder elasticity

The movement is simulated for a 2000 rpm engine speed. The following cases are discussed - rigid skirt and cylinder -elastic skirt and rigid cylinder - elastic cylinder and rigid skirt and elastic skirt and cylinder.

Figures 2-3 show the effect the elasticity of the skirt and the cylinder on the eccentricity at the top of the skirt $e_t(t)$. The same curve shape is obtained for the eccentricity at the bottom $e_b(t)$. The skirt motion increases in amplitude when the effect of skirt and cylinder elasticity is considered. The piston is pushed constantly most of the time against the major thrust side of the cylinder wall where a maximum pressure was observed in the lubricant film at 395 degree crank angle. This leads to decrease the film thickness as shown in figure 4. The results indicate that there is no effect of the elasticity of the cylinder on the secondary movement.

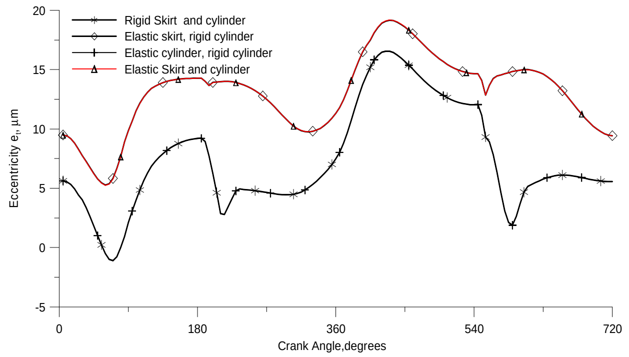

Fig. 2 Effect of the elasticity of the skirt and the cylinder on the upper skirt

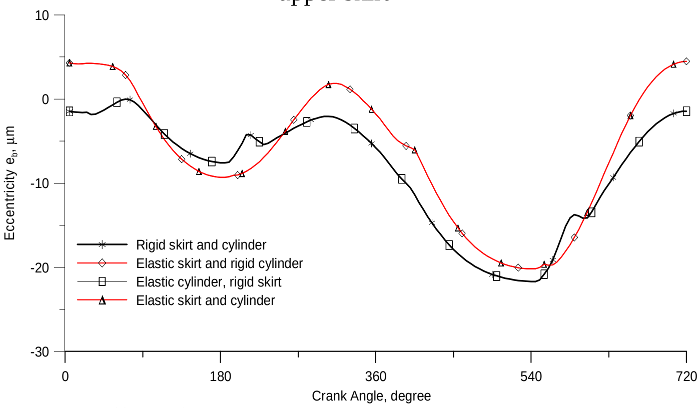

Fig. 3 Effect of the elasticity of the skirt and the cylinder on the bottom eccentricity of the skirt

Effect on the minimum film thickness

The minimum film thickness as a function of crank angle for the four cases is shown in Fig. 4.

The fluid-film keep a constantly value of about 20 µm in intake-compression-exhaust strokes and decrease rapidly at the beginning of the power stroke to reach the minimum value of 13.63 µm for both elastic skirt and cylinder surfaces and 11.93 µm for rigid surfaces at the same deg. crank angle 545, after the peak loading has occurred. The delay of the fluid-film reaction is due to damping inherent in the fluid-film, which prevents the piston bearing reacting rapidly to the combustion load change.

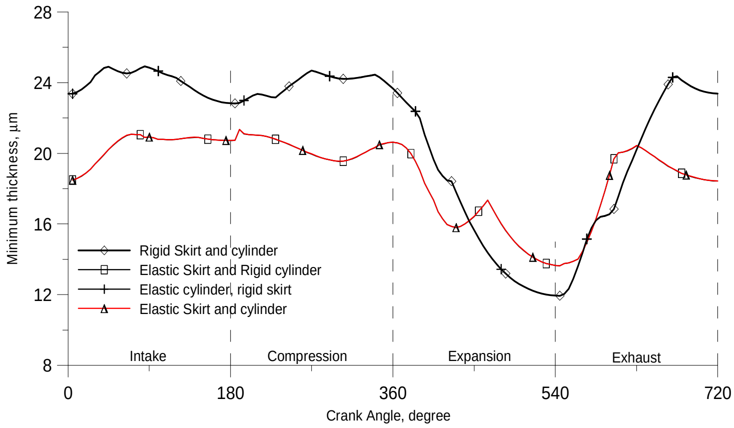

Fig. 4 Effect of the elasticity of the skirt and the cylinder on the minimum film thickness

Effect on the maximum pressure

Figure 5 shows the instantaneous maximum pressure developed in the lubricant film for the four cases.

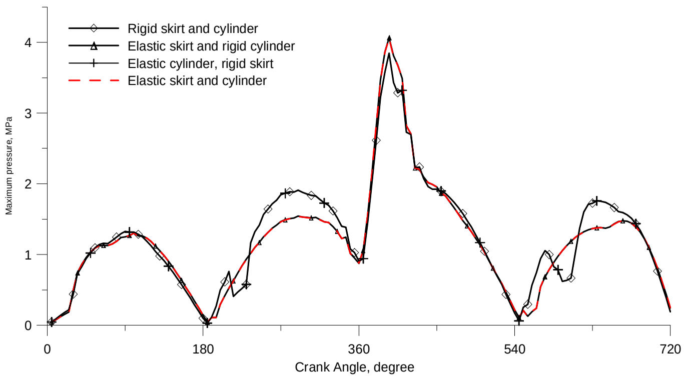

Fig. 5 Instantaneous maximum pressure in the fluid-film

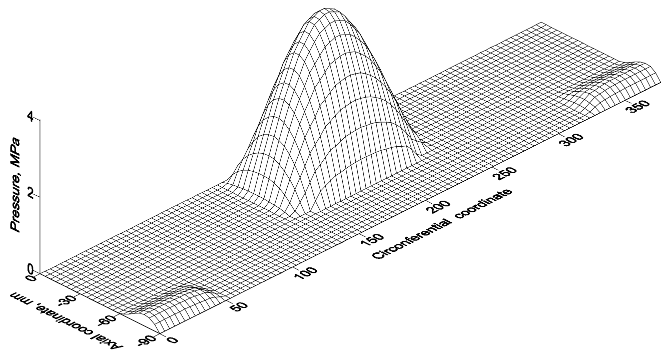

Fig.6 Maximum pressure distribution at 395 deg. Crank angle

The results indicate that there is no effect of the elastic surfaces on the maximum pressure, the pressure curves of hydrodynamic and elastohydrodynamic are approximately the same. The maximum pressure reaches the higher value after the top dead center at 395 deg. crank angle. At this position, the pressure distribution in the fluid film is given in figure 6. The positive value represents a radially outward and the negative value a radially inward deformation. The maximum deformations, which occur at the upper skirt, produce peak outward deformations on this surface of the skirt.

Effect on friction

The results of the four cases, including instantaneous skirt and cylinder-friction are plotted in Fig. 7. It is shown that the friction force, for the skirt and the cylinder, is similar in hydrodynamic and elastohydrodynamic motion. This indicates that the EHL solution has no effect on the friction loss. A high level of friction, about 30N is observed in the middle between TDC and BDC, where the piston sliding speed is high. The corresponding friction power loss is shown in Figure 8. Power losses are slightly higher in elastohydrodynamic lubrication than in the case of hydrodynamic lubrication. It can be seen from Figure 8 that the skirt and cylinder-frictional power loss are the same in the EHD lubrication.

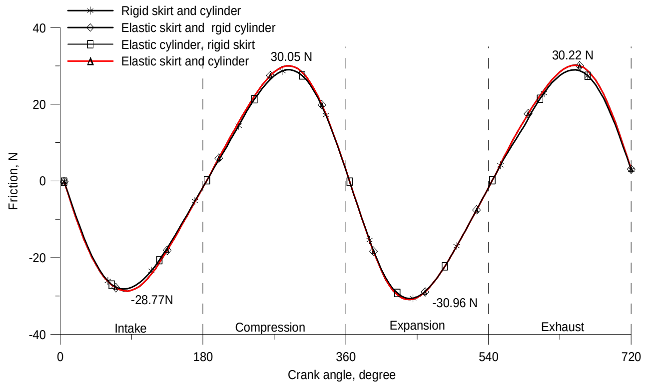

Fig. 7 Instantaneous skirt friction

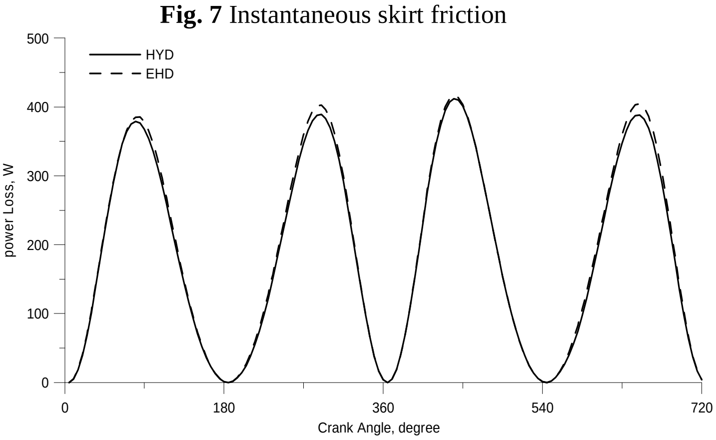

Fig. 8 Instantaneous friction power loss on the skirt for hydrodynamic and EHD lubrication

CONCLUSIONS

In this study computational work of elastohydrodynamic lubrication of piston skirt is developed. The study takes into account the elastic deformation of both skirt and cylinder and the cavitation generated in the lubricant film. The main results are as follows:

- The comparisons between the hydrodynamic and elastohydrynamic skirt lubrication shows that elastohydrodynamic's analysis is necessary for an accurate prediction of piston motion and optimum tribological performances.
- The obtained results indicate that there is no effect of the elasticity of the cylinder on the secondary movement.
- The results indicate that there is no effect of the elastohydrodynamic lubrication on the maximum pressure, the reults of hydrodynamic and elastohydrodynamic models are approximately the same
- The study permits to develop a powerful design analysis tool which can be used to increase the available engine power through optimum piston performance characteristics

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