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The Influence Oil Film Lubrication of the Piston-Cylinder Dynamic

An analytical study of the dynamics of a piston in a reciprocating engine was conducted. The equation of Reynolds and moving of piston are derived. The analysis, which incorporates a hydrodynamic lubrication model, was applied to M501 diesel engine. The results of this study indicate that piston dynamics were found to be sensitive to piston-cylinder bore clearance, location of the wrist pin and lubricant viscosity, underscoring their importance in engine design.

Keywords: *oil film, lubrification, dynamic viscosity, piston ring, diesel engine*

1. Introduction

It is award that in the operating process of a diesel engine the reliability and noise of a working piston are determined, to a great extent, by its lubrication condition. One of the purposes of optimized design of a piston is to make it work under full film lubrication condition as could as possible during its reciprocating motion that transfers dynamical forces. Studies indicate that the above purpose is attained by the optimization of the structure and the profile of the piston-skirt under given working conditions.

In this paper the load capacity and moment are calculated by numerical integration of Reynolds equation of lubrication film at the piston-skirt. The behavior of variation of loaded film during a working period is obtained by solving the coupled equations controlling the lateral motion of piston. The results can be used as the basis for design of profile and clearance of a piston-skirt. By this method the piston of M501 diesel engine is improved and its performance is enhanced.

2. The algorithm
Basic equations

The reciprocating motion of working piston along cylinder axis is determined by the gas force, inertia force and reaction force of the connecting rod of mechanism acting on the piston. For the crankshaft-connecting-rod mechanism without offset shown in figure 1, the reciprocating motion of the piston along cylinder axis is,

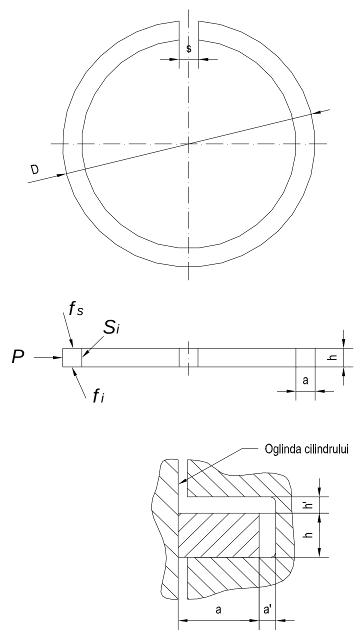

Figure 1 The piston ring shape and its placement into the piston canal

under the usual assumption of constant crankshaft rotating speed, given by the following equations:

$$S = R_1\left[(1-\cos\alpha)+\frac{R_1}{4l_1}(1-\cos 2\alpha)\right] \tag{1}$$

$$U = R_1\omega\left(\sin\alpha+\frac{R_1}{2l_1}\sin 2\alpha\right) \tag{2}$$

$$\dot{U} = R_1\omega^2\left(\cos\alpha+\frac{R_1}{l_1}\cos 2\alpha\right) \tag{3}$$

The lateral motion of a piston within the confinement of the cylinder clearance is determined by the lateral force, the film load capacity and the inertia force acting on the piston, together with the moments of these forces about the wrist-pin axis. The following gives the equations controlling the lateral motion:

$$\begin{pmatrix} m_P\left(1-\frac{a_1}{b}\right)+m_i\left(1-\frac{a}{b}\right) & m_P\frac{a_1}{b}+m_i\frac{a}{b} \\ \frac{I_p}{a}+m_p(a-a_1)\left(1-\frac{a_1}{b}\right) & -\frac{I^p}{b}+m_p(a-a_1)\frac{a_1}{b} \end{pmatrix}\begin{pmatrix} \ddot{e}_t \\ \ddot{e}_b \end{pmatrix}=\begin{pmatrix} F+F_S \\ M+M_S \end{pmatrix} \quad (4)$$

where:

m_p and m_i - are respectively the mass of the piston and that of the wrist-pin;

I_p - is the rotary inertia of the piston about the wrist-pin;

$\ddot{e}_t$ and $\ddot{e}_b$ - are respectively the lateral acceleration of the piston top and that of its bottom;

F and F_s - are respectively the film load capacity and the lateral force of the piston;

M and M_s - are the moment of F and that of F_s about the wrist-pin axis, respectively.

The pressure distribution within the loaded film obeys the Reynolds equation:

$$L(\bar{p})\equiv\frac{\partial}{\partial\theta}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\theta}\right)+\frac{1}{\lambda^2}$$

$$\frac{\partial}{\partial y}\left(\bar{h}^3\frac{\partial\bar{p}}{\partial\bar{y}}\right)=-\frac{1}{\lambda}\left[(\varepsilon_b-\varepsilon_t)\cos\theta+\frac{\partial\overline{f(\bar{y})}}{\partial\bar{y}}+q_b\,\bar{y}\cos\theta+q_t\left(1-\bar{y}\right)\cos\theta\right] \quad (5)$$

Where

$$\bar{h}=\frac{h}{c}=1+\varepsilon_t\cos\theta+\bar{y}(\varepsilon_b-\varepsilon_t)\cos\theta+\bar{f}c\bar{y},$$

$$q_b=\frac{D\dot{\varepsilon}_b}{U},\ q_t=\frac{D\dot{\varepsilon}_t}{U},\ \bar{p}=\frac{p\psi^2D}{12\mu U},\ \bar{y}=\frac{y}{b},$$

$$\psi=\frac{2C}{D},\ \varepsilon_b=\frac{e_b}{C},\ \varepsilon_t=\frac{e_t}{C},\ \lambda=\frac{2b}{D},$$

and:

C - is the radial clearance between piston and cylinder;

M - is the dynamic viscosity of the lubricant;

$\bar{f}(\bar{y})$ - the function describing the profile of piston skirt along its axis;

e_t and e_b - are the eccentricities of the piston measured at the top and the bottom of the skirt, respectively; p - is the pressure of the film.

The boundary conditions of E_q. (5) are:

$$\bar{p}(\theta,0) = \bar{p}(\theta,1) = 0. \qquad \left.\frac{\partial \bar{p}}{\partial \theta}\right|_{\theta = 0, \pi} = 0 \tag{6}$$

The numerical method

By the linear superposition principal, the solution of E_q. (5) under boundary conditions (6) can be expressed as:

$$\bar{p} = \bar{P}_1 + q_b \bar{p}_2 + q_t \bar{p}_3 \tag{7}$$

Where $\bar{p}_i$ are the solutions of the following equations under the same boundary conditions (6).

$$L(\bar{p}_1) = g_i \quad \text{(i=1,2,3) and}$$

$$g_1 = -\frac{1}{\lambda}\left[(\varepsilon_b - \varepsilon_t)\cos\theta + \frac{\partial \bar{f}(\bar{y})}{\partial \bar{y}}\right], \quad g_2 = \bar{y}\cos\theta, \quad g_3 = (1-\bar{y})\cos\theta$$

The Reynolds equation is solved numerically by the finite difference method and the dimensionless film load capacity $\bar{F}$ and its moment $\bar{M}$ about the wrist-pin axis are calculated by the following formulas:

$$\bar{F}\left(e_t, e_b, \dot{e}_t, \dot{e}_b\right) = \frac{F\psi^2}{6\mu U b^2} = 2\int_0^1 \int_0^{\pi} \bar{p}\cos\theta k(\bar{p}) d\theta d\bar{y} \tag{9}$$

$$\bar{M}\left(e_t, e_b, \dot{e}_t, \dot{e}_b\right) = \frac{M\psi^2}{6\mu U b^2} = 2\int_0^1 \int_0^{\pi} \bar{p}(\bar{a} - \bar{y})\cos k(\bar{p}) d\theta d\bar{y} \tag{10}$$

Where

$$\bar{a} = \frac{a}{b}, \qquad K(\bar{p}) = \begin{cases} 0, & \bar{p} \le 0 \\ 1, & \bar{p} > 0 \end{cases}$$

By solving simultaneously the coupled equations (4), (8) and (10), we can obtain the corresponding $e_t, e_b, \dot{e}_{t,}\ \dot{e}_b, \ddot{e}$ and $\ddot{e}_b$. The solution procedure is:

1) Set the right hand side of E_q.(4) at zero and obtain the periodic solutions of e_t and e_b by iteration method;

2) Calculate $\ddot{e}$ and $\ddot{e}_b$ by numerical differences, and then substitute them into the right hand side of E_q.(4) and solve E_q.(4) to obtain the periodic solution of e_t, e_b and the corresponding velocities and accelerations.

3. Exampled of improved designs

By the above method we improved the design of the piston of M501 diesel engine. The parameters of structure and operating of the engine are:
D=0.1m, b=0.063m, a_1=0.021m, a=0.0255m, l_1=0.21m, R_1=0.0575m, ω=232.38s^{-1}, $\mu = 0.7401 \times 10^{-2}\ Pa \cdot S$, $m_p = 1.045kg$, $m_i = 0.8565kg$.

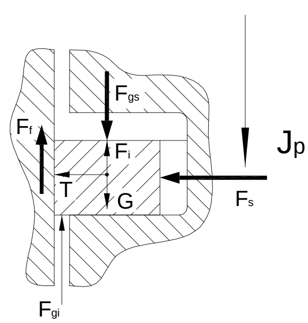

Figure 2 The forces action on the piston ring

Figure 2 shows the trajectories of piston lateral motion of original design at ψ=0.00145. It is found that the skirt bottom came into contact with the cylinder once, and scrape of the skirt occurred accidentally.

Figure3 shows piston trajectories for the improved design Radial clearance of piston ψ =0.00075 and 0.00095, respectively. It is found that the minimum film thickness $h_{t\,min} = 0.1616 \times 10^{-4}\,m$ and 0.143×10^{-4}, $h_{b\,min} = 0.1783 \times 10^{-4}\,m$ and 0.1377×10^{-4} are enhanced respectively. During 6000h duration test of the engine of improved design the rate fuel consumption of engine reduced 178g/ps h by 1.1g/ps h, and operating of the piston was good.

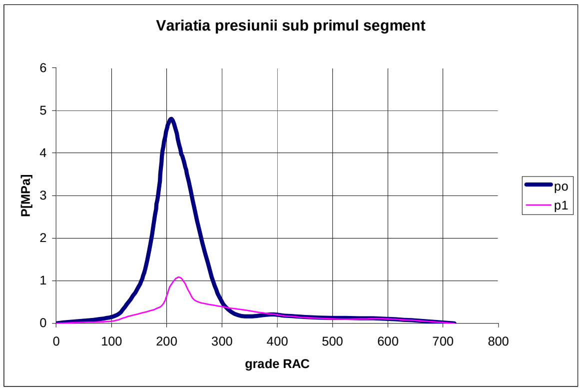

Figure 3 The pressure variation under the first piston ring

4. Conclusions

A usual method of solution was obtained in this study. Calculation showed that the effect of piston-skirt parameters on lubrication characteristics is significant.

The piston-skirt of optimized design improves significantly working conditions on piston-cylinder impact, piston trajectory and the degree of lubricant availability in the cylinder, therefore reliability of the piston, cylinder and working characteristics of the engine are improved substantially.

References

[1] Negrea V.D., *Procese in motoare cu ardere interna*, Editura Politehnica, ISBN: 973-9389-89-9

[2] Wang H.S., *A study on the mathematical model of a real friction surface for piston ring by using the conjugate curved surface principle,* SAE paper 76008.
[3] Qui H., *Theoretical anlysis on the lubrication between piston ring and cylinder liner lubrication and seal,* 1985, 4.

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