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# Piston dynamic characteristics analyses based on FEM method Part I: Effected by piston skirt parameters ☆

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ABSTRACT

The dynamic and lubrication characteristics are all very complex problem in piston-liner analysis, and they have great effect on the power output, vibration, noise emission. In this paper, the numerical model which concludes lubrication part and dynamic motion is established, the lubrication is solved by the finite element method, and dynamic equation is solved by Runge–Kutta. The effect of piston skirt parameters on dynamic characteristics are compared based on a typical inline six-cylinder engine, such as: clearance, offset of piston pin, length of piston skirt, position of bump, curvature parameter and ellipticity of the piston, all the result mainly focus on the slap noise of the engine. All the analyses are very useful to design of piston-liner at the development of the engine, and it can provide the guidance for the design of the low noise engine.



## 1. Introduction

Piston skirt in an internal combustion engine (ICE) plays an important part in the way of power transmission, and piston set is one of the most important components in an internal combustion engine. Its characteristics have great effect on the sealing performance, friction loss, oil consumption and noise. Today, the technology of the reciprocating engine is very mature, having been developed extensively over the past century as the prime mover for the automotive industry, significant improvement has led to in power density, efficiency, and emissions in recent decades. Reducing friction can improve efficiency without harming emissions, while simultaneously reducing wear and improving the reliability of the engine. The research showed that 5% of total engine fuel energy is dissipated through the piston assembly friction loss, mainly due to piston skirt and piston ring pack loss, piston assembly accounts for 20–30% of mechanical loss for a typical gasoline engine [1]. At the same time, the whole loss from the whole piston contributes to 58–75% loss of the whole engine friction loss [2]. Richardson provides a summary of recent literature and testing experience in sources of engine friction, the research showed that 45–55% mechanical friction loss is dissipate from piston system, and piston skirt contributes 25–47% in this component [3]. Priest and Taylor, in their analysis of engine friction conclude a 10% reduction in mechanical losses would lead to a 1.5% reduction in fuel consumption. Fox found that the piston skirt and rings had the greatest potential for improvement from a reduction in boundary friction [4]. It is necessary to quantitatively assess the piston assembly friction in order to optimize the piston ring pack and the piston skirt geometries. A well-designed skirt will result in the optimal operation of the engine, low friction, low noise, low oil consumption, low emissions and ultimately long engine life.

Apart from big friction loss coming from the piston skirt, one of the major noise and vibration sources in internal combustion engine is impacted by the secondary motion between the piston and cylinder wall [5], the clearance between the piston and liner is very small but large enough to induce the piston's secondary motion periodically and finally generates unwanted sound and vibration [6]. Secondary motion across the clearance between piston and cylinder inner wall and then side thrust force induced by the connecting rod, the piston moves from one side to opposite side in the cylinder. Eventually, the piston collides against the

*Abbreviations:* ICE, internal combustion engine; FEM, finite element method; CA, crank angle; MOFT, the maximum oil film thickness; FEM, the finite difference method; TS, thrust side; ATS, anti-thrust side; TDC, top dead centre.

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**Nomenclature**

| | |
|---|---|
| $R$ | diameter of piston |
| $C_p$ | offset of the piston pin |
| $M_p$ | mass of the piston |
| $M_{pin}$ | mass of the piston pin |
| $M_{sl}$ | mass of small conrod side |
| $l$ | length of the conrod |
| $n$ | speed of the engine |
| $\Omega$ | wave of the piston |
| $m_{r1}$ | mass of the first piston ring |
| $m_{r2}$ | mass of the second piston ring |
| $m_{r3}$ | mass of the third piston ring |
| $r$ | radius of the crank-pin |
| $b_1$ | width of the first piston ring |
| $b_2$ | width of the second piston ring |
| $b_3$ | width of the third piston ring |
| $F_{Ic}$ | initial force of the piston system |
| $L$ | length of the piston skirt |
| $a_p$ | the length from the third ring to the pin center |
| $I_{pin}$ | the moment initial of the pin |
| $I_p$ | the moment initial of the piston |
| $u$, $v$, $w$ | the velocity along $x$, $y$, $z$ direction |
| $\mu$ | dynamic viscosity of oil film |
| $\varphi_x$, $\varphi_y$ | pressure flow factors along $x$, $y$ direction |
| $\varphi_s$ | shear flow factors |
| $\sigma$ | roughness of surface |
| $p$ | oil film pressure |
| $h$ | oil film thickness |
| $t$ | time |
| $U$ | velocity of the piston |
| $e_0$ | the offset distance relative the mass of the piston |
| $\gamma$ | the tilt angle of the piston |
| $a_p$ | the distance from piston mass center to top of piston skirtedge. |
| $x_0$, $y_0$ | the coordinate of the element center |
| $\xi$–$\eta$ | the transformed coordinate |
| $m$, $n$ | the long and short axis of oval |
| $\phi$ | the crank angle |
| $e_t$ | the lateral displacement of the top skirt |
| $e_b$ | the lateral displacement of the bottom skirt |
| $P_g$ | the cylinder pressure |
| $P_{b1}$ | the piston groove pressure in the first ring |
| $P_0$ | the piston groove pressure in the second ring |
| $D$ | the diameter of the cylinder |
| $F_g$ | the cylinder pressure force |
| $M_f$ | the friction moment |
| $\gamma$ | the tilt angle |
| $\eta_f$ | the friction coefficient |
| $M_h$ | the moment caused by oil film |
| $F_f$ | the friction force |
| $L_1$ | the distance from the first ring to pin center |
| $L_2$ | the distance from the second ring to pin center |
| $F_L$ | the connect rod force |
| $K$ | the elliptic coefficient |
| $\Delta$ | the ellipticity of the piston skirt |
| $C_s$, $r_s$ | the curve fitting coefficient |

cylinder inner wall and slaps the liner, so the sound is radiated to the environment. The oil film between the piston skirt and the cylinder inner wall would experience reaction forces induced by the vertical piston movement and the lateral piston movement. In the traditional isolation design of piston, only the lubrication of piston is considered, while piston-slap induced exciting components of force and moment are of much less concern or neglected to simplify the analysis and design. It has been demonstrated by both analytically and experimentally that piston-slap is a major excitation source of air-borne noise from an internal combustion piston engine, especially from turbo-charged diesels. Joseph [7] eliminated piston slap through a design for robustness CAE approach, in the paper, these factors were investigated by DOE method, factors that affected the slap noise weight was analyzed. Cho [8] and Geng [9]used the simple model to estimate the impact force induced by piston slap, but the detail parameters was not considered. Gerges [10]found oil film was an important factor on the impact behavior, Reynolds' theory for fluid film squeezing can be applied for oil film damping determination, the effective damping was found to be inversely proportional to the oil film thickness cubed, the influence of cylinder lubrication on the piston slap was investigated.

To make deeper research on the wear mechanism and secondary motion of the piston or validate the piston motion model, many kinds of methods are used to test the lubrication or wear, which can be concluded that: (1) Bench test [11]; (2) Floating liner [12]; (3) Realistic engine test [13], but researches mainly focused on the piston ring. Compared with lubrication and wear test, researches of experiment focused on mechanism of the secondary motion and slap noised by the impact are few, an indirect measurement of the piston secondary motion of a diesel engine was reported to analyze the piston slap [8]. Tan and Ripin developed an experimental technique to measure frictional behavior and the distinct modes of piston secondary motion, the piston motion and tilt angle were captured using two laser displacement sensors, the lateral motion are verified by comparing the frequency components of the lateral motion with the frequency components of the lateral acceleration measured using a triaxial accelerometer [14,15]. Later, a nonlinear model of the piston with reciprocating, lateral and rotational degree of freedom was developed to investigate the piston secondary motion and the induced vibration behavior of the engine block by the piston slap, the model was validated by experimental data obtained from three laser displacement sensors which captured the distinct modes of the piston secondary motion directly from the piston assembly under motorized conditions [16].

In fact, the piston skirt dynamic motion model can be divided as Reynolds model and secondary model, Reynolds model that considered the roughness was put forward Patir and Cheng [17], but it was first used in piston ring lubrication problem, then Knoll and Peeken [18] studied the lubrication, it was used in bearing, until Li [19] put out the mathematic model of the lubrication. Richmond [20] found that the clearance and ellipicity of piston had great effect on the slapping noise, too small gap will affect the formation of the oil film, bigger clearance will intensify the noise. Zhu [21,22] established the dynamic model of the piston skirt, he coupled the Patir and Cheng model to analyze the secondary motion and lubrication, thermal deformation was considered to the model. In practice engineering, the lubrication and noise is a trade-off problem. Piston design for noise analysis mainly focuses on the parameters: clearance, piston skirt stiffness, skirt profile, piston pin offset, and piston skirt length, ellipicity of piston, cylinder pressure, and initial force and so on. Nakashima changed the piston pin offset and centroid height to improve slapping noise at every speed, the noise was reduced clearly [23]. Kim found that the geometry that decided by the type line can reduce the wear of the piston [24]. Malagi [25] had done the lubrication before, and made comparison for the lubrication characteristics after the