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A new numerical method for piston dynamics and lubrication analysis

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ABSTRACT

A numerical model for piston dynamics and lubrication analysis is proposed by coupling the lubrication model of piston skirt–liner system and the dynamics model of multibody system consisting of crank, connecting rod and piston. The lubrication model is solved by the Finite Element Method, while the multibody dynamics equations are established with Lagrange multipliers and constraint Jacobian matrix. The validation of the method is verified through comparing with a well-known existing work. Apart from this, some design parameters of piston skirt were investigated so as to reveal their influence on the slap noise and lubrication performances of the piston.

1. Introduction

With the ever-increasing requirements for improving fuel economy and reducing emissions from the automobile, the reduction of the friction in internal combustion engine (ICE) has attracted more and more attention from the public. Piston assembly frictional loss accounts for the highest fraction of the total power loss of an ICE [1]. Such friction loss is mainly caused by the relative motion of piston rings and skirt against the cylinder bore. In addition, piston skirt is closely related to the piston-slap noise, vibration and durability. A well-designed piston skirt will facilitate the optimal operation of the engine with low oil consumption, low noise and low emission. Therefore, an in-depth study to the effects of the piston skirt on the engine performance is important for optimizing the piston design parameters.

Considerable studies on the lubrication of the piston-liner system have been carried out since 1980s. To illustrate, by incorporating the hydrodynamic lubrication model into the equations of the piston dynamics, Li et al. [2] investigated the dynamics of a piston and the piston skirt friction power loss for the first time. Zhu et al. [3,4], Keribar and Dursunkaya [5], Wong et al. [6] and Liu et al. [7] established, in a more complete way, the dynamic models of the piston based on Patir and Cheng's Average Reynolds equation [8,9]. These equations took into account the effects of surface roughness and waviness on the hydrodynamic lubrication by introducing the pressure- and shear-flow factors into the equation. In addition, for the optimization, the effects of some factors on the lubrication were also discussed, such as bulk elastic deformation and thermal distortion, the availability of lubricant, surface waviness and roughness [3,4,6,10–15]. The aforementioned studies were conducted on the assumption that the inertia of the connecting rod could be ignored. However, the inertia of the connecting rod will bring extra influence on the lubrication of the piston, especially when the engine runs at a high speed. Later on, Zhang et al. [16] developed a novel mathematical model for piston-connecting rod–crankshaft dynamics by considering the variation in the system inertia. They found that the piston secondary motion and side force would increase with the mounting system inertia. Meng et al. [17,18] investigated the effects of the connecting rod inertia on the lubrication performance of the piston-liner system, and concluded that the connecting rod inertia did have some influence on the system lubrication as well as the piston dynamics, especially when engine ran at high speeds.

In the previous studies, only the dynamics models for the piston are established to investigate the piston secondary motion and lubrication. These models are established by taking the piston-liner system as a two-degree-of-freedom system, and choosing e_t and e_b (i.e., the lateral displacements of the piston at the top and bottom of the skirt, respectively) as the degrees of freedom of the system. Then, the balance of forces and moments about the wrist yields the set of governing differential equations of the piston secondary motion. However, this approach makes it difficult to consider the effects of other components of the ICE. Although some scholars have studied the effects of the connecting-rod inertia [16–19], the deducing process of piston dynamics equations is complex. Moreover, some other affecting factors, such as the clearance at connecting rod big end bearing and flexibility of the connecting rod, might be very difficult

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Nomenclature

a_p	vertical distance from the piston pin to the top of the piston skirt edge
b_p	ertical distance from the center of mass (COM)of piston to the top of the piston skirt edge
C_g	Horizontal distance between piston COM and piston pin
C_p	piston pin offset
c	nominal radial clearance between the piston skirt and liner
e_0	the lateral displacement of the piston
F_n	total normal force
F_f	total friction force
F_G	combustion gas force
h	oil film thickness
h_{skt}	iston skirt profile
L_{skt}	length of the piston skirt
$\mathbf{M}$	system mass matrix
M_n	total moment of F_n about piston pin axis
M_f	total friction moment about piston pin axis
$\boldsymbol{N}$	shape function of discrete elements
p	oil film pressure
p_c	asperity contact pressure
$\boldsymbol{N}$	generalized force vector
$\boldsymbol{q}$	generalized Cartesian coordinates of the system
R	radius of the piston
r	radius of the crankshaft
u	velocity of the piston
x, y	local coordinate system on the piston skirt
μ	dynamic viscosity of oil
ϕ_x, ϕ_y	pressure flow factors
ϕ_s	shear flow factor
Φ_{fs}, Φ_{fp}	tilt angle of the piston
ϕ_c	contact factor
Φ_f	term to average the sliding velocity component of the shear stress
σ	roughness of the surface
α	angular coordinate of the piston skirt
ω	angular velocity of the crankshaft
β_1 and β_2	the location of the mass center of the crank and rod, respectively
Γ	boundary of integral domain
μ_f	friction coefficient of the asperity contact
$\boldsymbol{\Phi}$	kinematic constraints
λ	lagrange multipliers
ϑ, ς	feedback parameters in Baumgarte's approach

to be considered in the existing model. In order to address this problem, it would be a favorable approach to establish a dynamics model of the piston-connecting rod–crank system in a framework of multibody mechanical system. The studies on dynamics of multibody systems have made great progress during the past several decades. For example, Flores et al. [20,21] proposed the model for a slider–crank mechanism with clearance in the revolute joint, and found that the existence of the clearance made the system highly nonlinear and the dynamic behavior tended to become chaotic as the clearance increased. Muvengei et al. [22] numerically studied the parametric effects of clearance joints at different locations on the dynamic response of multibody systems. Bai and Zhao [23] proposed a quantitative analysis method for multibody systems with clearance joints. Zhao et al. [24–26] also investigated the effects of clearance joint in slider–crank system on the dynamic performance of the system and the wear at the joint. Wilson et al. [27] proposed a method of predicting the detailed motion of the slider in a slider crank mechanism with dry slider-guide clearance, and evaluated the effects of changes in geometry and mass distribution on the trajectory of the slider and the energy losses at the points of impact. However, the analysis of mechanical system with the lubricated clearance at the slider (i.e., the clearance between the piston and the liner) has been rarely reported in a framework of multibody mechanical system.

In addition, the clearance between the piston and liner is small, but it is large enough to induce the piston's secondary motion and thus generate unwanted noise and vibration [28,29]. In practicing the ICE design, the reduction of the friction power loss may be accompanied with the increase of slap noise. Therefore, apart from friction power loss in the piston-liner system, the engine noise and vibration induced by the piston's secondary motion should also be given enough concern. Unfortunately, the above studies just focused on the lubrication of the piston, and therefore, the noise was often neglected to simplify the analysis and design. It has been demonstrated both analytically and experimentally that a piston-slap produced by the piston's secondary motion is the major excitation source of the vibration and noise of an ICE [30–32]. To be specific, Zheng et al. [33] analyzed the diesel piston–slap induced ship hull vibration and noise, and found that the piston–slap exerted excitation on the engine frame might cause a high level of ship hull vibration and underwater radiated noise. Kazuhide [31] clarified that decrease of the piston clearance of upper part could reduce the high frequency components of engine noise. He et al. [34] established a numerical model and analyzed the slap noise from the view of the slap energy.

In this study, a new numerical model is proposed for piston lubrication and dynamics analysis. This model is established by coupling the lubrication model of the piston–liner system and the multibody dynamics model of the piston–rod–crank system. The lubrication model for the piston–liner system is solved based on the Finite Element Method (FEM) due to its advantage of the flexibility in dealing with irregular domains [35]. Then, by integrating the pressure distribution evaluated with the aid of Reynolds' equation, the hydrodynamic forces (oil-force and oil moment) are obtained and taken as external forces acting on the piston. The hydrodynamic forces built up by the lubricant fluid are evaluated from the piston state variables and included into the equations of motion of the multibody system. In the end, the proposed approach is applied to a four-stoke gasoline engine. Some piston skirt design parameters, such as the clearance, the bulge position and curvature parameter of piston skirt profile, are investigated for their influence on the slap noise and lubrication of the piston skirt and liner system.

2. The FEM-based hydrodynamic lubrication model of piston skirt–liner system

2.1. Hydrodynamic lubrication model

Considering the effect of surface roughness, the average Reynolds equation [8,9] is used for solving oil pressure between the skirt and liner. This equation can be expressed as

$$\frac{\partial}{\partial x}\left(\phi_x \frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\phi_y \frac{h^3}{12\mu}\frac{\partial p}{\partial y}\right) = -\frac{u}{2}\left[\phi_c\frac{\partial h}{\partial y}+\sigma\frac{\partial \phi_s}{\partial y}\right]+\phi_c\frac{\partial h}{\partial t} \tag{1}$$