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Transient tribodynamic model of piston skirt-liner systems with variable speed effects

Xianghui Meng [a,b,*], Congcong Fang [a,b], Youbai Xie [a,b]

[a] State Key Laboratory of Mechanical Systems and Vibration, Shanghai Jiaotong University, Shanghai 200240, People's Republic of China
[b] School of Mechanical Engineering, Shanghai Jiaotong University, Shanghai 200240, People's Republic of China

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ABSTRACT

A new transient tribodynamic model of the piston skirt-liner system of reciprocating engines is presented in this study. This new model couples the tribological performance of the piston skirt-liner system with the dynamics of the connecting rod, the crankshaft, the flywheel, and the piston. In the model, secondary motion of the piston and transient variable engine speeds are both considered. A case study shows that the transient tribodynamic model is significant in simulating the engine performance under transient conditions. The angular acceleration of the crankshaft is considered, in contrast to models used previously.

1. Introduction

Automakers are faced with increasing challenges to reduce engine emissions and fuel consumption. For reciprocating internal combustion engines, the piston skirt-liner system is crucial to the generation of engine mechanical losses, noise, and exhaust emissions [1,2]. The improvement of the tribological performance of the piston-liner system has been a long-standing research concern.

The performance of the piston skirt-liner system is determined by both the piston dynamics and the engine tribology. In 1983, Li and Ezzat first calculated the piston secondary motion and power lost to friction by coupling the hydrodynamic lubrication model with the piston dynamics equations [3]. In the 1990s, Zhu et al. [4,5] and Liu et al. [6] contributed to simulation analyses of the piston skirt-liner system by adopting the average flow model presented by Patir and Cheng [7]. In these analyses, the elastic and thermal deformation of the piston skirt and liner were also considered. In the 2000s and most recently, further details were considered in theoretical analyses of the system. Gamble, Priest, and Taylor analyzed oil transport in the piston assembly of a gasoline engine [8]. Jang and Cho [9] and Mansouri and Wong [10] investigated the influence of piston design parameters, such as the piston profile and piston-liner clearance, on the lubrication and dynamic performance of the system. Meng et al. analyzed the thermo-elasto-hydrodynamic lubrication of a piston skirt considering the oil-film inertia [11]. McClure and Tian developed a dry-piston secondary dynamics model and achieved faster calculation times [12]. McFadden and Turnbull analyzed the piston dynamics under non-lubricated and fully flooded lubricated conditions [13]. Qasim et al. studied the viscoelastic effects of non-Newtonian engine lubricants on the piston dynamics and the oil-film pressure at small radial clearances [14]. In 2012, two authors of this study, Meng and Xie, built a new numerical model for the lubrication of the piston skirt–liner system by considering the effects of the connecting rod inertia [15]. Dolatabadi, Theodossiades, and Rothberg studied the identification of piston slap events with a transient tribodynamic analysis [16]. They found that crosshatched textures on cylinder liner surfaces might change the pressure distribution and load carrying capacity of the piston-liner system. Biboulet, Bouassida, and Lubrecht examined the influence of crosshatched textures on the load-carrying capacity of oil control rings [17]. More investigation remains necessary to comprehensively model the piston skirt-liner system.

As numerical simulations progressed, experimental techniques were also used to investigate the tribodynamics of the piston-liner system. Taylor and Evans reported several experiments on running engines to measure the piston secondary motion, piston ring oil-film thickness, piston friction, and piston temperatures [18]. Furuhama and Takiguchi presented the floating-liner method measuring the friction force as a function of the crank angle of the piston assembly [19]. The method was direct and accurate, but required major engine modifications. Mufti and Priest measured the piston-assembly friction using the indicated mean effective

* Corresponding author at: State Key Laboratory of Mechanical System and Vibration, School of Mechanical Engineering, Shanghai Jiaotong University, Shanghai 200240, People's Republic of China. Tel./fax: +86 21 34207167.
E-mail address: xhmeng@sjtu.edu.cn (X. Meng).

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Nomenclature

a	vertical distance between the center of mass (COM) of piston and the top of the skirt
b	vertical distance between the piston pin and the top of the skirt
C	nominal radial clearance between piston skirt and liner
C_p	piston pin offset
C_g	horizontal distance between piston COM and piston pin
d	increase of gap between skirt and liner due to deformations
e_b,e_t	eccentricities of piston at the bottom and top of the skirt, respectively
F_B	assumed connecting rod force acting on piston
F_{BX}	reaction of piston pin in the X direction
F_{BY}	reaction of piston pin in the Y direction
F_{CX}	reaction of big end in the X direction
F_{CY}	reaction of big end in the Y direction
F_{DX}	reaction of crankshaft in the X direction
F_{DY}	reaction of crankshaft in the Y direction
F_G	combustion gas force acting on the piston
$\overline{F}_{ic}$,$\overline{F}_{ip}$	reciprocating inertia force of piston and piston pin respectively
F_{ic},F_{ip}	lateral inertia force of piston and piston pin due to secondary motion of piston
F_S	force, intermediate variable
F'_S	force, intermediate variable when considering inertia of connecting rod
F_{SK}	total friction force acting on the piston skirt
h	oil film thickness
h_c	ratio of DE to DC as shown in Fig. 2(c)
h_{skt}	piston skirt profile
I_{pis}	rotary inertia of piston about its COM
I_R	rotary inertia of connecting rod about its COM
I_C	rotary inertia of crankshaft and flywheel about its COM
j	ratio of CO to CB as shown in Fig. 2(b)
L	length of piston skirt
l_c	length of connecting rod
M	moment of S about the piston pin
M_{fSK}	moment of F_{SK} about the piston pin
M_{pis}	inertial moment of piston
M_S	moment, intermediate variable
m_p	sum of m_{pis} and m_{pin}
m_{pis}	mass of piston
m_{pin}	mass of piston pin
m_R	mass of connecting rod
m_C	mass of crankshaft and flywheel
p	oil film pressure
p_c	contact pressure
R	piston radius
r	crankshaft radius of gyration
S	side force acting on the piston skirt
T_{load}	output torque delivered by the crankshaft
u	velocity of piston
X, Y	global coordinate system as shown in Fig. 1
x, y	local coordinate system on the piston skirt
X_R	displacement of connecting rod COM along the X direction
X_E	displacement of crankshaft COM along the X direction
Y_P	displacement of piston COM along the Y direction
Y_R	displacement of connecting rod COM along the Y direction
Y_E	displacement of crankshaft COM along the Y direction
α	angular coordinate of piston skirt
$\theta_0=\arcsin(\frac{C_p}{l_c+r})$	tilt angle of piston
ϕ	connecting rod angle
ϕ_c	contact factor
ϕ_{fs}, ϕ_{fp}	shear stress factors
ϕ_s	shear flow factor
ϕ_x, ϕ_y	pressure flow factors
η	lubricant viscosity
ρ	density of oil
θ	crank angle
$\dot{\theta}$	angular speed of crankshaft
τ	shear stress
μ_f	friction coefficient of asperity contact

pressure (IMEP) method [20]. Compared to the floating-liner method, the IMEP method could measure the friction force at realistic engine speeds and loads without major engine modifications. On a motored test rig with a single-cylinder engine, Tan and Ripin estimated that the piston skirt friction force contributes as much as 44% of the overall friction force of the piston assembly [2]. Masaaki et al. measured the oil-film thickness and velocity map by combining the laser-induced fluorescence method and the particle image velocimetry method [21]. Littlefair et al. measured the lubricant film thickness by a non-invasive ultrasonic-assisted method [22]. Kamiya et al. measured the oil-film pressure on the piston skirt using small thin-film pressure sensors [23]. Using a reciprocating test rig, Demas et al. examined the tribological behavior of the piston skirt and cylinder liner interface with different surface coatings and oil additives at various loads, speeds, and temperatures [24]. Dolatabadi, Theodossiades, and Rothberg measured the surface acceleration of the engine block to validate the predicted piston slap events [16]. Because piston temperature has a significant influence on the piston-liner performance, Buono, Iarrobino, and Senatore developed an optical method to measure the piston temperature during different engine operations [25].

Most of these studies focused on the piston dynamics alone in examining the tribodynamic behavior of the piston-liner system, and assumed stable engine speeds during analysis. Although the authors of this study have previously accounted for the effects of the connecting rod inertia [15], this adjustment is insufficient for accurate modeling. First, the piston-liner system is coupled with not only the connecting rod, but also the crankshaft and the flywheel. Second, internal combustion engines often work under variable speeds. Engines often increase or decrease speed while running. Even for engines operated under stable conditions, the transient engine speed varies instantaneously because of the variable cylinder pressure. Therefore, a new transient tribodynamic model of the piston-liner system is necessary to reveal more details of the system. This new model is the main contribution of this study.

2. Theory

To understand the transient tribodynamic model presented in this study, two previous models are first shown here. Eq. (1) is the model used prior to 2012, in studies [4,6,9–11], to express the