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A new coupling tribodynamic model of crosshead slipper-guide system and piston skirt-liner system of low-speed marine diesel engines

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ABSTRACT

In this paper, considering the coupling effects of the crosshead slipper-guide system and the piston skirt-liner system, a new tribodynamic model is established for the low-speed two-stroke marine diesel engine. Firstly, a dynamic model including the crosshead slipper, the piston, the piston rod, and the crosshead pin is proposed. Then, considering the disciplinary coupling between the dynamic model and the lubrication models, the stiff and nonlinear tribodynamic model is solved with the modified extended backward differentiation formulae (MEBDF). Finally the tribodynamic performances of the crosshead slipper and the piston skirt are analyzed. The simulation results show that the crosshead slipper bears the thrust force and its friction power loss is obviously larger than that of the piston skirt.

1. Introduction

Low-speed marine diesel engines are main power equipments of vessels over 2000 tons. They are usually two-stroke type with a crosshead slipper-guide system and running at a rotation speed of lower than 300 rpm. The increasing demand for fuel economy and the tighter emissions control of low-speed marine diesel engines make the technologies of reducing engine friction attract the manufacturers' concern. The crosshead slipper-guide system and the piston skirt-liner system are two important coupling friction pairs in low-speed marine diesel engines, especially the former. It bears the large thrust force, and its friction loss constitutes a large part of the total power loss. At the same time, the dynamic behaviors of the crosshead slipper and the piston also have great influences on the noise, vibration and stability of the engines. Therefore, an in-depth study on the tribodynamic characteristics of the crosshead slipper-guide system and the piston skirt-liner system is necessary to improve the engine's performance.

Both the crosshead slipper-guide system and the piston skirt-liner system are affected by the longitudinal reciprocating motion and the transverse motion which is also called the secondary motion. In recent decades, the studies on the numerical simulation of the piston skirt-liner system mainly focus on high speed or medium speed engines without the crosshead slipper-guide system. Li et al. [1] first proposed a method coupling the dynamic model of the piston with the lubrication model of the piston skirt-liner system in 1983. Next, scholars such as Zhu et al. [2] and Liu et al. [3] used the average flow model proposed by Patir and Cheng [4] to consider the influences of the surface roughness. Upon the entry of the new century, scholars have conducted more detailed studies on the piston skirt-liner system. Meng et al. [5] analyzed the thermo-elasto-hydrodynamic lubrication of the piston skirt considering the effects of the oil film inertia. Mcfadden et al. [6] investigated the piston-liner system under fully flooded lubricated and non-lubricated conditions. Qasim et al. [7] analyzed the dynamic and tribological characteristics of the piston considering the viscoelastic effect of the non-Newtonian engine lubricant. Moreover, the effects of piston design parameters and factors on the lubricating performance and dynamic behavior of the piston skirt-liner system were also investigated, such as piston profile, clearance, surface roughness, elastic deformation, thermal distortion, and cylinder liner vibration [8–15]. Meng et al. [16] established a more accurate tribodynamic model of the piston skirt-liner system considering the influence of the connecting rod inertia. Subsequently, Meng et al. [17] proposed a transient tribodynamic model of the piston skirt-liner system by coupling the lubrication model of the piston skirt-liner system and the dynamic model of the piston, connecting rod, crankshaft and flywheel. Livanos et al. [18] established a comprehensive friction model for the piston assembly of a medium speed marine diesel engine and the model can be used to predict the friction of the individual piston component. In addition, the piston slap phenomenon in

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the piston skirt-liner system also attracts many scholars' attention. Dolatabadi [19,20] studied the method to determine the piston slap event and calculated the piston slap noise. He et al. [21] presented the influences of the piston structural parameters on the slapping noise. Zheng et al. [22] investigated the piston slap of the medium speed marine diesel engine. The results indicated that the piston slap can cause greater hull vibration and underwater radiated noise.

For the low-speed marine diesel engine with the crosshead slipper-guide system, however, no published literature has been found to investigate the coupling performances of the crosshead slipper-guide system and the piston skirt-liner system. Most researches focused on the piston rings, crosshead bearings, main bearings and other tribological components in the engine. Abanteriba et al. [23–25] studied the lubrication conditions and friction loss of a single acting crosshead slipper neglecting the coupling tribodynamics between the piston assembly and the crosshead assembly. Wolff et al. [26] presented a comprehensive model of the piston ring-liner system in a low-speed marine diesel engine considering the oil flow, gas flow, piston ring twist, and piston ring axial motion. Mohamad et al. [27,28] analyzed the influences of the cylinder liner surface textures on the piston ring lubrication characteristics numerically. Guo et al. [29] conducted a series of experiments to investigate the influences of the liner surface texture features on the tribological properties of the cylinder liner-piston ring system. The lubrication conditions of the crosshead bearing and main bearing in the low-speed marine diesel engine were also investigated considering the effects of the bearing shell deformation [30,31]. Hassan et al. [32] presented a simulator model of a marine diesel engine which can be used to predict the performance of the lubricating system. Olander et al. [33] performed the scuffing tests to find new piston ring materials and proposed a hypothesis for the low-speed marine diesel engine's scuffing process. As the thermal load of the marine diesel engine continues to increase, the temperature conditions of the critical friction pairs attract the attention of some scholars. Considering the secondary motion of the piston and the lubrication oil film in a marine diesel engine, Lu et al. [34] conducted the thermal numerical analysis on the piston. Stolarski et al. [35] presented the temperature-friction characteristics of the used lubricating oil by the method of five times heating test. In addition, the test techniques for detecting the tribological properties of low-speed marine diesel engines are also developed quickly, especially the acoustic emission (AE) technology in recent years. Using AE technology, Nagata et al. [36] investigated the tribological properties of the bearing materials. Douglas et al. [37] applied AE technology to the research on the tribological behavior of the piston ring-liner system, and pointed out that AE technology has the potential to provide the reference for the optimum oil delivery rate.

Since the crosshead slipper and the piston components are coupled together, a tribodynamic model coupling the crosshead slipper-guide system and the piston skirt-liner system will be established in this paper. Within the author's knowledge, this model will be presented for the first time. Based on the tribodynamic model, the dynamic behavior, oil film lubrication characteristics and friction of these two friction pairs will be analyzed. This research will lay a theoretical foundation for improving the performance of low-speed marine diesel engines.

Nomenclature

a	vertical distance between the center of mass of crosshead pin and the top of the crosshead slipper	L	length of piston-rod
b	vertical distance between the center of mass of crosshead slipper and the top of the crosshead slipper	L_1	vertical distance between the COM of piston and the top of skirt
B	width of the crosshead slipper	L_2	vertical distance between the COM of piston and the bottom of skirt
C	nominal clearance between guide and crosshead slipper	L_3	length of crosshead slipper
C_1	nominal radial clearance between liner and piston	L_4	length of connecting rod
e_t, e_b	eccentricities of crosshead slipper at the top and bottom end respectively	M_1	moment of bolt to piston
e_{tp}, e_{bp}	eccentricities of piston at the top and bottom of skirt respectively	M_2	moment of bolt to piston rod
F_c	total normal force acting on crosshead slipper	M_c	moment of F_c
F_{cf}	total friction force acting on crosshead slipper	M_{ic}	inertial moment of crosshead slipper
F_g	combustion force acting on piston	M_{ip}	inertial moment of piston
F_{ic1}	reciprocating inertia force of crosshead slipper	M_{irod}	inertial moment of piston rod
F_{ic2}	transverse inertia force of crosshead slipper	M_{icp}	inertial moment of crosshead pin
F_{icp1}	reciprocating inertia force of crosshead pin	m_c	mass of crosshead slipper
F_{icp2}	transverse inertia force of crosshead pin	m_p	mass of piston
F_{ip1}	reciprocating inertia force of piston	m_{cp}	mass of crosshead pin
F_{ip2}	transverse inertia force of piston	m_{rod}	mass of piston rod
F_{irod1}	reciprocating inertia force of piston rod	p	oil film pressure
F_{irod2}	transverse inertia force of piston rod	p_c	contact pressure
F_l	connecting rod force acting on crosshead pin	R	piston radius
F_p	total normal force acting on piston skirt	R_{pin}	crosshead pin radius
F_{pf}	total friction force acting on piston skirt	R_c	crankshaft radius of gyration
F_x	reaction force of crosshead pin to slipper in the X direction	U	velocity of crosshead slipper
F_{x1}	reaction force of bolt to piston in the X direction	w	half of the height of crosshead slipper
F_{x2}	reaction force of bolt to piston rod in the X direction	x_1, y_1	local coordinate system on the crosshead slipper
F_y	reaction force of crosshead pin to slipper in the Y direction	x_2, y_2	local coordinate system on the piston skirt
F_{y1}	reaction force of bolt to piston in the Y direction	X, Y	global coordinate system as shown in Fig. 1
F_{y2}	reaction force of bolt to piston rod in the Y direction	α	angular coordinate of piston skirt
G_c	gravity of crosshead slipper	θ	connecting rod angle
G_{cp}	gravity of crosshead pin	φ	crack angle
G_p	gravity of piston	ϕ_x, ϕ_y	pressure flow factors
G_{rod}	gravity of piston rod	ϕ_c	contact factor
h_{c1}	oil film thickness of crosshead slipper in the thrust side	ϕ_s	shear flow factor
h_{c2}	oil film thickness of crosshead slipper in the anti-thrust side	$\phi_f, \phi_{fs}, \phi_{fp}$	shear stress factors
h_{pis}	oil film thickness of piston skirt	μ	dynamic lubricant viscosity
I_c	rotary inertia of crosshead slipper about its COM	τ	shear stress
I_p	rotary inertia of piston about its COM	ω	angular velocity of crankshaft
I_{cp}	rotary inertia of crosshead pin about its COM	μ_f	friction coefficient of asperity contact
I_{rod}	rotary inertia of piston rod about its COM	ρ	lubricant density

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