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Research Article

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Oil film pressure distribution of combined piston skirt considering the function of instantaneous clap force

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ABSTRACT

When the piston moves in the cylinder, on the one hand influenced by the explosion pressure of the gas, on the other hand due to the large mass of the piston, the piston is subjected to a large inertial force, then the force caused by the two side force on the piston. The change of lateral force will cause a slight lateral movement of the piston and the rocking motion (secondary movement) in the cylinder. The change of instantaneous clap force has a great impact on the form of oil film, which will affect the distribution of the oil film distribution. The instantaneous clap force of the piston skirt which is measured on a test machine will be considered to modular calculation of the dynamic equations. the distribution of oil film pressure will be analyzed and compared under the circumstances of considering the instantaneous clap force and not considering the instantaneous clap force, which will lay a solid foundation for the design of profile of the piston skirt as well as improve the lubrication conditions and reduce the impact on the cylinder.

Key words: Instantaneous Clap Force, Oil Film Pressure, Secondary Movement, Combined Piston Skirt

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INTRODUCTION

With the rapid development of ICE industry, high-power diesel engine piston is one of the most demanding parts of diesel engine, which would pass driving force in the high-speed reciprocating motion and withstand high mechanical and thermal loads [1]. The piston is one of the bottlenecks of the constraints to further diesel development. The piston design quality and machining accuracy level would directly affect the economic reliability and service life of the diesel engine. In the modern piston design process, considering the contact and friction performance of the design would become increasingly important throughout the entire design process.

Based on the hydrodynamic lubrication theory and system dynamics, the research of teratology characteristic on the combined piston of the type of 16V280 [2] high-power diesel engine was established. According to the secondary movement of the combined piston and to calculate the piston transverse and swing movement. The oil film pressure would be calculated by the means of the coupled hydrodynamic lubrication theory and the convex body elastic contact model.

The deduction and establishment of the dynamic equation which is suitable for the Combined Piston Skirt will be obtained. On the basis of the squeezing effect, the Reynolds equation has been obtained. The partial differential equation has been transformed into numerical solution with a five-point difference meshing, and the difference equation has been solved by using the adaptive SOR method. The establishment of the coupled model of the secondary movement and the cylinder-piston Reynolds equation would be solved by the means of DFP algorithm. Ultimately, the results concerning the oil film pressure distribution would be obtained in the entire system.

EXPERIMENTAL MODELS AND NUMERICAL CALCULATION

The instantaneous clap force of the piston skirt which is measured on a test machine will be considered to modular calculation of the dynamic equations, given the basic parameters of the piston skirt and the equation iteration of the initial displacement, velocity and acceleration. In Table 1, the profile and dynamics parameter of piston of 16V280 diesel engine are shown on. In the case of speed 1000rpm, the gas pressure would be given at the top of piston head, which would be defined as Fig.1. Based on the Reynolds and dynamics Equation, under the circumstance of the conformity with iteration residuals, the results concerning the oil film pressure distribution could be obtained.

Table 1 Profile and dynamics parameter of piston of 16V280 diesel engine

Profile and dynamics parameter	Values
Cylinder Bore	280
Piston Throw	300
Cylinder Matching Gap	0.4
Connecting Rod Length	590
Crank Radius	150
Piston Center of Gravity to Pin Center Distance	120
Piston Mass	39.089 [kg]
Piston Pin Mass	17 [kg]
Speed	1000 [rpm]
Inertia of Piston Rotating Center of Gravity	5.233 [N.m^2]
Dynamic Viscosity	0.016 [Pa.S]
Oil Film Bearing Angle	35^o

Notes: all the unmarked units of table are millimeter.

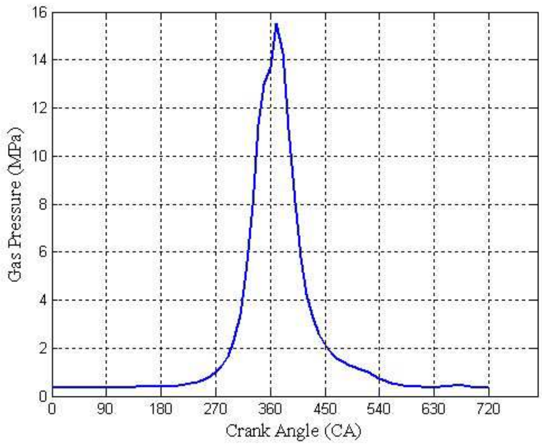

Fig.1 gas pressure

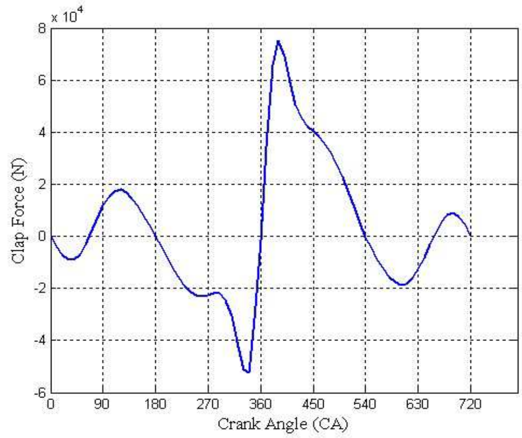

Fig.2 clap force

The average two dimensional Reynolds equationof piston skirt and cylinder[3]is defined as Eq.1.

$$\frac{\partial}{\partial x}(\phi_x h^3 \frac{\partial P_h}{\partial x}) + \frac{\partial}{\partial y}(\phi_y h^3 \frac{\partial P_h}{\partial y}) = 6\mu U(\frac{\partial h_T}{\partial y} + \sigma \frac{\partial \phi_s}{\partial y}) + 12\mu \frac{\partial h_T}{\partial t} \qquad (1)$$

Oil film bearing area is calculated in the region of $(-\theta_1, \theta_1)$ and $(\pi - \theta_1, \pi + \theta_1)$, Assuming the oil film pressure is zero in other region. The oil film bearing area is defined as Fig.3.

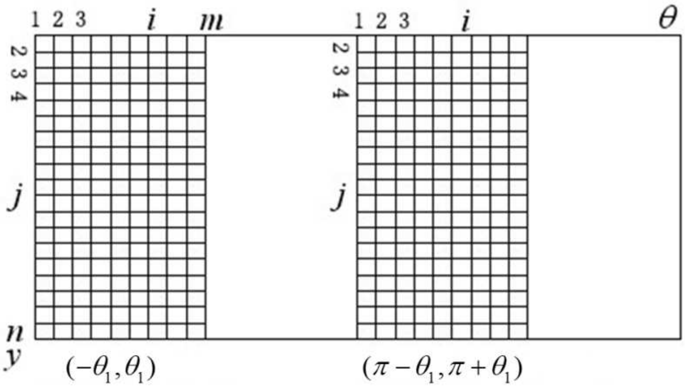

Fig.3 oil film bearing area

The grid area is oil film bearing area,the other blank space is non-bearing area. After the calculating region is determined,using different axial vertical grid spacing and the same number as the standard to mesh grid [4].The oil film bearing area of the piston skirt is circumferentially divided into m equal parts, which is divided into n parts along the direction of the skirt. Similarly in the circumferential direction is divided into m equal parts divided along the direction of the skirt and n equal parts.

Each grid cell of the skirt portion in the longitudinal direction and the circumferential spacing of the gridis defined as Eq.2.

$$\Delta\theta = \frac{2\theta_1}{m} \qquad (2)$$
$$\Delta y = \frac{L}{n}$$

SIMULATION RESULT AND ANALYSIS

According to the above numerical method [5], which involves the corresponding calculation Matlab code, maximal oil film pressure distribution of the four strokes of the piston skirt modular, which would be defined as Fig.4, Fig.5, Fig.6, Fig.7 in the case of not considering the instantaneous clap force.

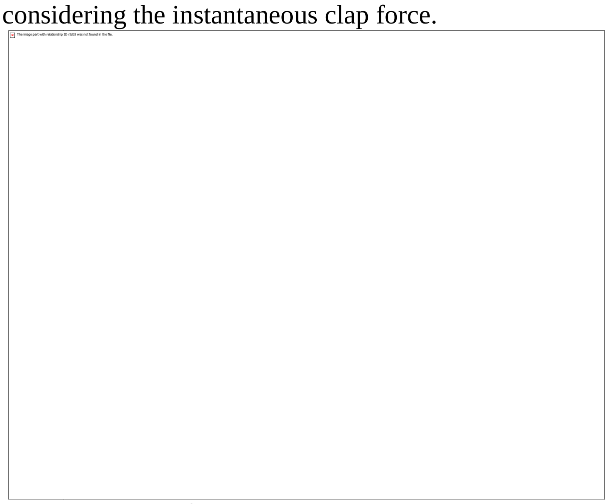

Fig4. oil film pressure crank angle 110(deg.)

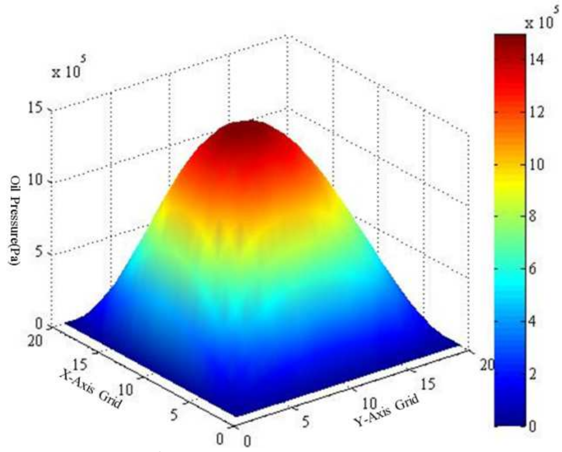

Fig5. oil film pressure crank angle 330(deg.)

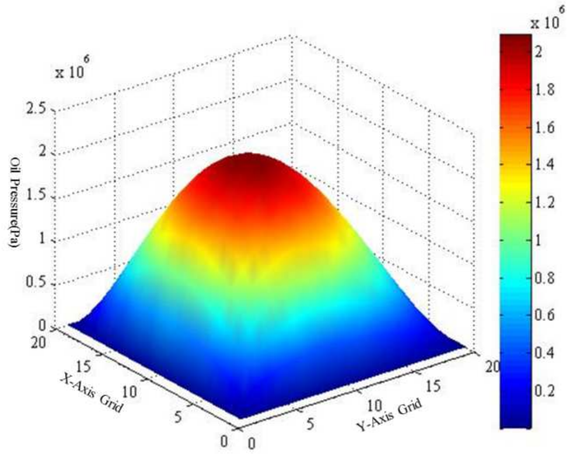

Fig6.oil film pressure crank angle 380 (deg.)

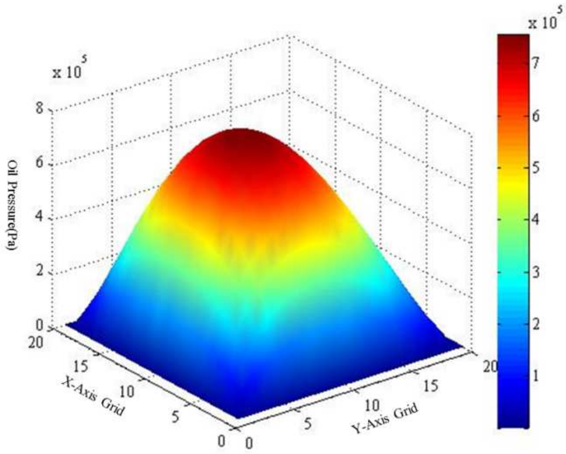

Fig7. oil film pressure crank angle 610(deg.)

As can be seen from the figure, the pressure distribution along the circumferential direction of the film is symmetrical, a peak value is in the center of the contact area. Fig.6 shows at the time when the crank angle is 380 degree, the maximum oil film pressure reaches 2.11MPa in the expansion stroke. The gas pressure is also a great peak top on the piston, the piston combined joint action by the force of inertia, friction.

The calculating result considering the instantaneous clap force is defined as Fig.8, Fig.9, Fig.10, Fig.11.

Fig.10 shows at the time when the crank angle is 380 degree, when the oil film pressure distribution of the skirt portion, the maximum oil film pressure reaches 4.2 MPa in the expansion stroke. The volume of the combustion gas swelling pressure and temperature rises rapidly. Under the function of the gas pressure, the piston moves to the upper dead point, the wedge due to the presence of two-way, the film thickness due to the lateral force becomes small extruded film while its carrying capacity is increased only.

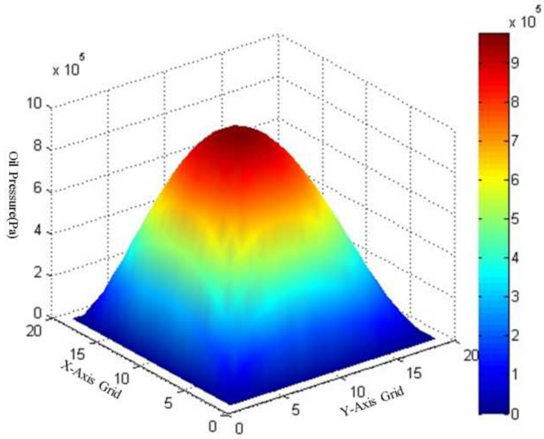

Fig.8 oil film pressure crank angle 110(deg.)

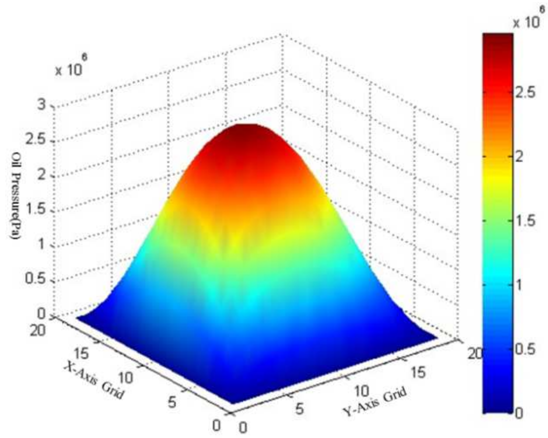

Fig.9 oil film pressure crank angle 330(deg.)

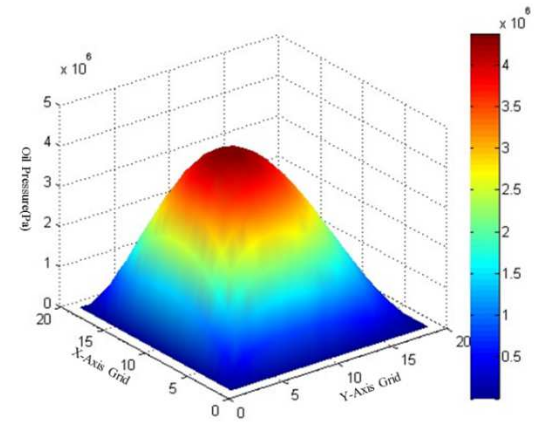

Fig.10 oil film pressure crank angle 380 (deg.)

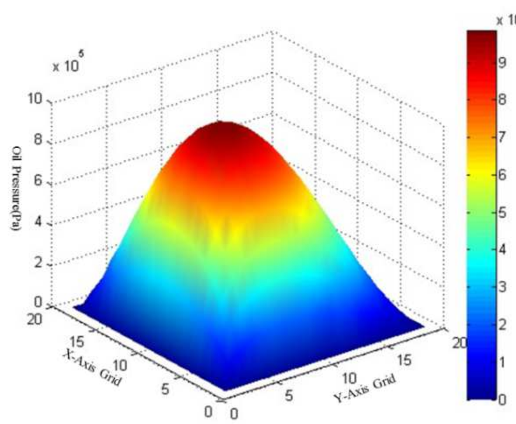

Fig.11 oil film pressure crank angle 610(deg.)

CONCLUSION

According to the calculation results and analysis, comparison of the oil film pressure distribution of the piston skirt in the case of not considering the instantaneous clap force and considering the instantaneous clap force. The three-dimensional pressure distribution can reflect the actual value of the oil film pressure and status of pressure distribution. The results show the maximal value of the oil film pressure reaches 2.11MPa without considering the instantaneous clap force. On the contrary, the oil film pressure of the piston skirt reaches 4.2MPa in the case of considering instantaneous clap force. By Comparison, it sharply increases 2.09MPa in the oil film pressure, which is mainly attributed to the instantaneous clap force in the momentary action and the decreasing thickness of the oil film, which increases in the bearing capacity of the oil film [6]. With the requirements of the withstanding the higher mechanical and thermal load, the oil film pressure of the piston skirt would be considered, which is of benefit to form bidirectional oil wedge.

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