EFFECT OF FRICTION FORCE DETERMINATION METHOD ON RESULTS OBTAINED FROM MODEL OF GAS FLOW FROM COMBUSTION CHAMBER TO THE CRANKCASE

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Abstract

The paper presents investigation of the influence of the friction force calculation method on the results of simulations run with the use of the model of gas flow from the combustion chamber to the crankcase and piston rings motion in the grooves. The first series of simulations was run using the gas flow and ring motion model in which the friction force was calculated from the empirical equation. The second series was run with the use of the gas flow and ring motion model fully integrated with the ring lubrication model. In that case the friction force was derived from the hydrodynamic model of ring lubrication. Comparison of the results obtained in the first and second series of simulations indicated that despite the fact that the friction forces calculated with these two methods differed considerably, the method of friction force calculation had insignificant effect. This refers to the quantities determined in the simulations which are most important from practical point of view – i.e. displacements of the rings in the grooves, pressure courses in inter-ring regions and blowby rate. It is not necessary to integrate the gas flow and ring motion model with the model of ring lubrication, and doing so to complicate it very much, if the aim of the calculations is simulation of blowby and rings behaviour in the grooves.

Keywords: *combustion engine, blowby, piston ring, friction force, mathematical model*

1. Introduction

In models of the gas flow from the combustion chamber to the crankcase through the piston-rings-cylinder seal most quantities are derived from physical dependences and laws and only some of them are calculated from empirical dependences [1, 2, 4, 5]. Main disadvantage of empirical dependences lies in the fact, that they are correct only for some class of similar objects, while physical dependences are universal and as such can be used in wider range for any object. That is why the empirical dependences are replaced by the physical ones.

In the gas flow and ring motion model used so far by the authors of the paper [2] one of the quantities calculated from empirical equations was the force of friction between the piston ring and cylinder liner. Friction force is one of the forces acting on the ring and influences position of the ring in the groove and intensity of the gas flow through the seal. To determine this force from physical dependences it was necessary to model oil film between the ring and cylinder. The empirical equation was originally utilized because the model of the ring lubrication was very complex and itself constituted independent research problem. But lately the ring lubrication model has been worked out and integrated with the gas flow model. The main motivation for coupling the models was that the values of pressure in inter-ring regions, which can be quite accurately determined with the use of the gas flow model, were essential for the results of the simulations carried out with the use of the piston lubrication model. Considering the gas flow model, integration of these models enables abandoning calculation of the friction force from the empirical equation and usage of its value derived from the ring lubrication model. On the other hand, the integration made the models more complex and extended the time of simulations.

The aim of the work presented in this paper was to check the effect of integration of the gas flow model and ring lubrication model on the results of simulations made with the use of the gas flow model.

2. Models and method of research

Two series of simulations were carried out to investigate the influence of the calculation method of the friction force on the results obtained from the gas flow and ring motion model. The first series was run with the use of the original gas flow and ring motion model in which the friction force was calculated from the empirical equation. The second series was run with the use of the gas flow and ring motion model fully integrated with the ring lubrication model. In that case the friction force was obtained from the ring lubrication model. The results derived from the first and second series of simulations were then compared and used to evaluate how the method of calculation of the friction force affected ring displacements, pressure courses in inter-ring regions and blowby rate calculated with the use of the model.

A previously presented [2] model of the gas flow through crevices of the ring pack and of rings motions in the grooves was used in this research. The model assumed that the gas flows through the seal consisting of several stages linked together by throttling passages. The stages were created by inter-ring and behind ring volumes, which were considered independently. The throttling passages were created by ring end-gaps and crevices between side surfaces of the rings and the grooves (Fig. 1). The model took into account thermal deformations of the cylinder kit components while determining instantaneous (as a function of crank angle) values of the stages volumes and ring end-gaps cross-sections. Cross-sections of the crevices between the side surfaces of the rings and the grooves resulted from instantaneous axial positions of the rings in the grooves. The mass rates of the gas flow through crevices were calculated assuming that the flow was isentropic through the orifice with the consideration for sub-critical and critical flows and taking into account discharge coefficients.

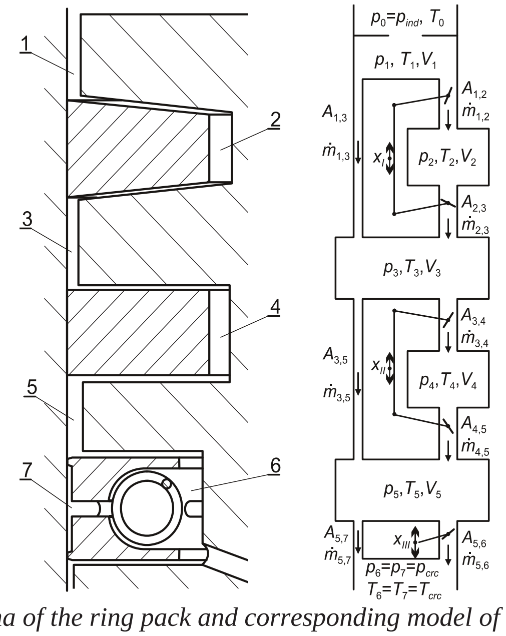

Fig. 1. Schema of the ring pack and corresponding model of labyrinth seal

Axial positions of the rings in their grooves were calculated with the consideration of the forces of inertia F_i, gas pressure F_p and ring friction against the cylinder F_f from the following equation:

$$F_p + F_i + F_f = F_{net} = m_r \frac{\mathrm{d}^2 x}{\mathrm{d}t^2}, \tag{1}$$

where:

m_r - mass of the ring,

x - ring displacement relative to the piston.

The pressure force was a resultant of forces generated by the pressure acting on the upper and lower surfaces of the ring (Fig. 2). The inertia force was calculated assuming that the acceleration of the ring was equal to the acceleration of the piston.

In the first series of calculations presented in this paper the friction force between the ring and the liner was calculated using empirical relationship [4]:

$$F_t = -f\pi D_c H\left(p_b + p_s\right), \tag{2}$$

where friction coefficient f was defined as:

$$f = 4{,}8\left(\mu \frac{v_p}{H\left(p_b + p_s\right)}\right)^{\frac{1}{2}}, \tag{3}$$

where:

μ - dynamic viscosity of oil,

H - height of the ring,

D_c - cylinder diameter,

v_p - piston velocity,

p_b - -pressure behind the ring,

p_s - ring pressure on the liner resulting from self-elasticity of the ring.

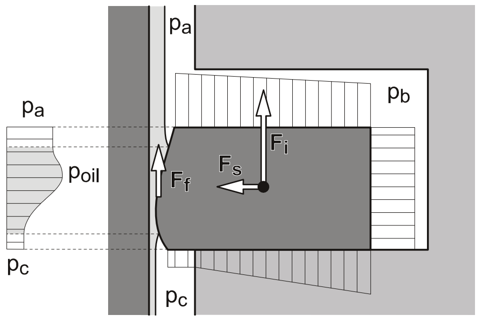

Fig. 2. Assumed pressure distribution and forces acting on the ring

In the second series of calculations the above described original model of the gas flow and ring motions was integrated with the ring lubrication model. The ring lubrication model used in the research was a conventional model of hydrodynamic lubrication [6] in which the flow of oil in the gap between the ring and cylinder was described by the Reynolds equation in the following form:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p_{oil}}{\partial x}\right) = 6\mu u \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t}, \tag{4}$$

where:

$h = h(x)$ - gap between the ring and cylinder liner,

u - oil flow velocity,

x - coordinate in the direction of oil flow (axial),

$p_{oil} = p_{oil}(x)$ - pressure in oil film.

Using the Reynolds equation (4) for modeling of the oil flow allowed for determination of the friction force between the ring and cylinder as the viscous friction force from the following equation:

$$F_f = \int_{x_a}^{x_c}\left(\frac{h}{2}\frac{\mathrm{d}p_{oil}}{\mathrm{d}x} - \frac{\mu u}{h}\right)\mathrm{d}x, \tag{5}$$

where:

x_a and x_c - boundaries of the wetted area of the ring.

The models were identified for the naturally aspirated 4-cylinder diesel engine with displacement of 2.4 dm^3 and rated power of 51.5 kW at 4200 rpm. The piston ring pack of the engine comprised a keystone barrel-shaped face top ring, a taper face second ring and a twin-land spring-backed oil control ring. The input data necessary for the calculations was obtained from the engine technical documentation and from the measurements of the components. In case of the dimensions determining the volumes of stages and cross-sections of the throttling passages, thermal deformations were added to the cold-measured values. Temperatures and thermal deformations of components for the different conditions of engine operation were calculated using Finite Element Methods. The indicated pressures for different engine operation conditions indispensable for the calculations came from the measurements made on the engine during test stand research [3].

Simulations were done for several engine operating conditions: full load at 2000, 3000 and 4200 rpm. Results of the calculations for 2000 and 4200 rpm are presented in Fig. 3. Those two rotational speeds were selected because corresponding behaviours of the rings were very different, what, in consequence, influenced courses of the gas pressure in the inter-ring spaces. At 2000 rpm rings changed their axial position in their grooves only a few times, and the second compression ring was not changing its position at all – adhered to the bottom shelf of the groove during the whole engine cycle. At 4200 rpm rings, especially the second one, changed their position in the grooves repeatedly, what generated rapid changes of pressure in the corresponding inter-ring spaces. Friction forces used in the simulations presented in Fig. 3 were calculated from the empirical equation (2).

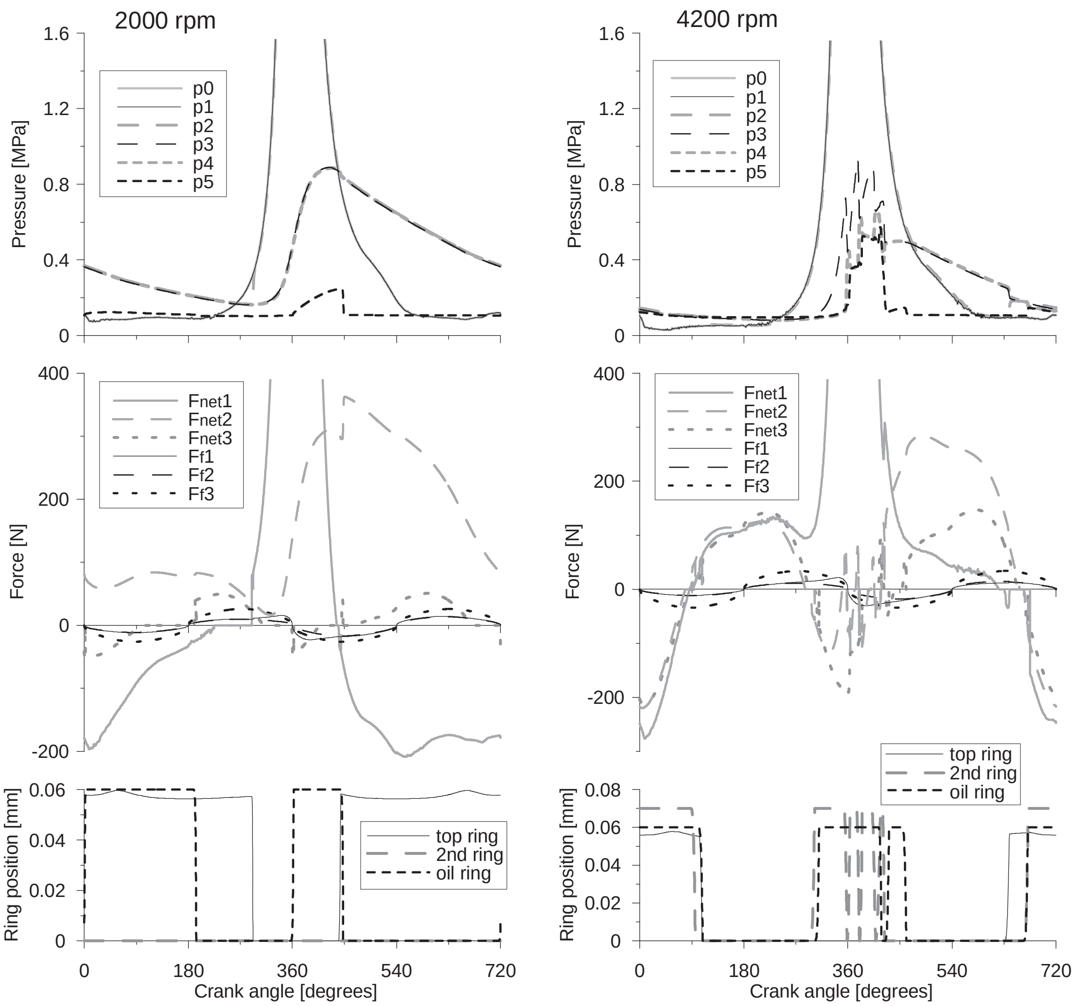

Fig. 3. Pressures in inter-ring spaces, friction and net forces acting on rings and axial positions of rings in piston grooves determined for engine full load at 2000 rpm (left) and at 4200 rpm (right)

3. Results

The courses of friction forces as function of crank angle calculated from empirical equation and derived from ring lubrication model were similar to each other, but the magnitudes differed significantly in the case of the top and second ring (Fig. 4). In contrary to the ring lubrication model, empirical dependence did not take into consideration the effect of the face surface profile on the friction force (see equations 2 and 3). This is especially visible in the case of the second ring which face surface was strongly asymmetrical (taper face ring). The friction force for this ring calculated with the use of the oil film model was much smaller when ring moved in the direction of the TDC than when it moved downstroke (Fig. 4).

Comparing the determined friction forces it could be also noticed that the ring lubrication model with regard to calculations of the friction force was less sensitive to the pressure of the ring on the cylinder than the empirical dependence. This is especially visible in the case of the top ring near TDC (Fig. 4). However, it should be reminded that the ring lubrication model used in this research considered only hydrodynamic lubrication. Therefore, the actual friction force could be bigger than calculated with this model because of the mixed lubrication which could occur in this range of crankshaft angles and the bigger values of friction force obtained from empirical dependence can be reasonable.

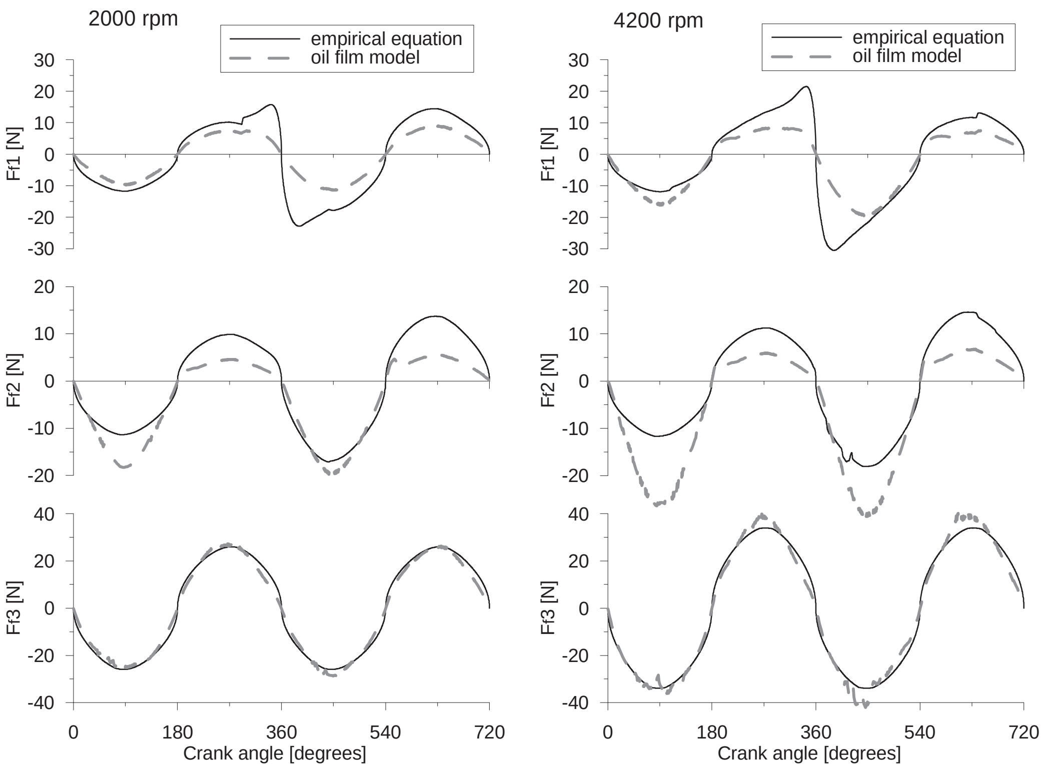

Fig. 4. Friction forces acting on top (F_{f1}), second (F_{f2}) and oil ring (F_{f3}) as function of crank angle calculated using empirical equation and oil film model for the engine full load at 2000 rpm (left) and at 4200 rpm (right)

Axial position of the rings in their grooves and pressure of the gas in inter-ring spaces as functions of crank angle calculated with the use of the original gas flow and ring motion model and calculated with the use of this model coupled with the ring lubrication model were practically the same for engine speeds of 2000 and 3000 rpm (Fig. 5 and Fig. 6). For engine rotational speed of 4200 rpm the crank angles at which the pistons changed their position in the grooves and corresponding changes of the pressure were slightly shifted (up to 2° CA).

The consequences of such small differences in ring positions and pressure courses were very small differences in the blowby rates obtained in two schemes of calculations – the biggest difference of 0.5% was obtained for 4200 rpm.

Such small differences in ring positions, pressures and blowby rates, even though the friction forces differed significantly in the two schemes of calculations, were related to the fact that the friction force was the smallest of all the forces taken into consideration, which acted on the rings in axial direction. The courses of friction forces and resultant forces acting on the rings in axial direction are presented in Fig. 3. Only in the case of the oil ring at small engine speed the friction force had an important influence on the net force, but friction forces calculated with the use of the empirical equation and derived from the ring lubrication model were very similar to each other in the case of the oil ring (Fig. 4).

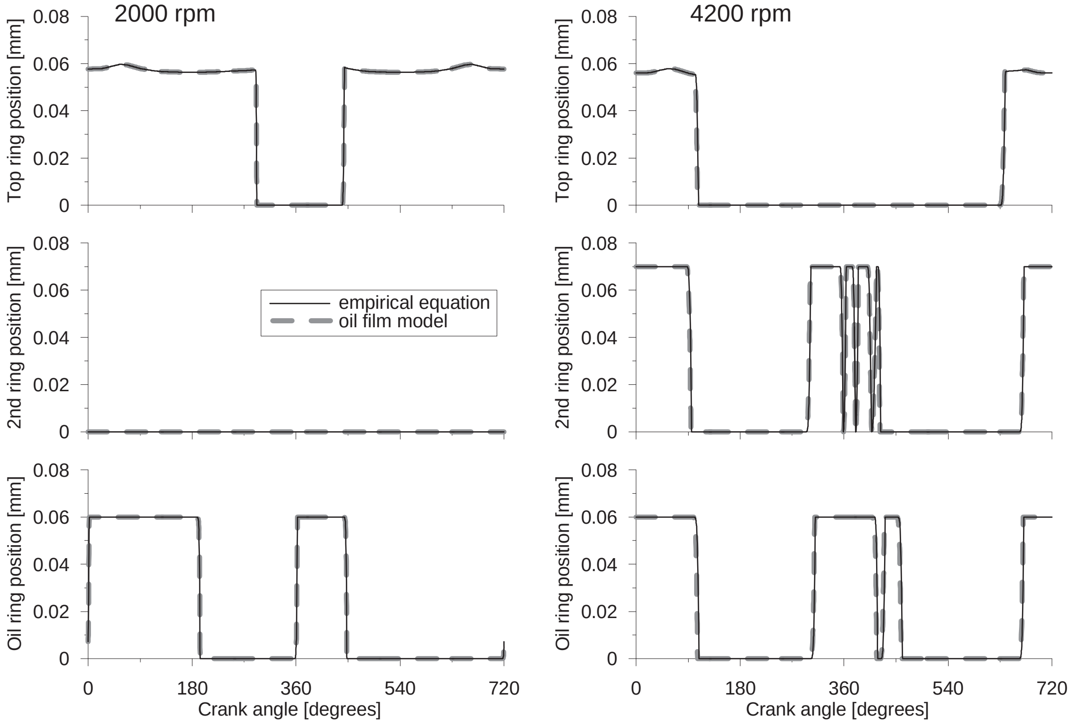

Fig. 5. Axial displacements of rings in the piston grooves for friction forces calculated using empirical equation and oil film model for the engine full load at 2000 rpm (left) and at 4200 rpm (right)

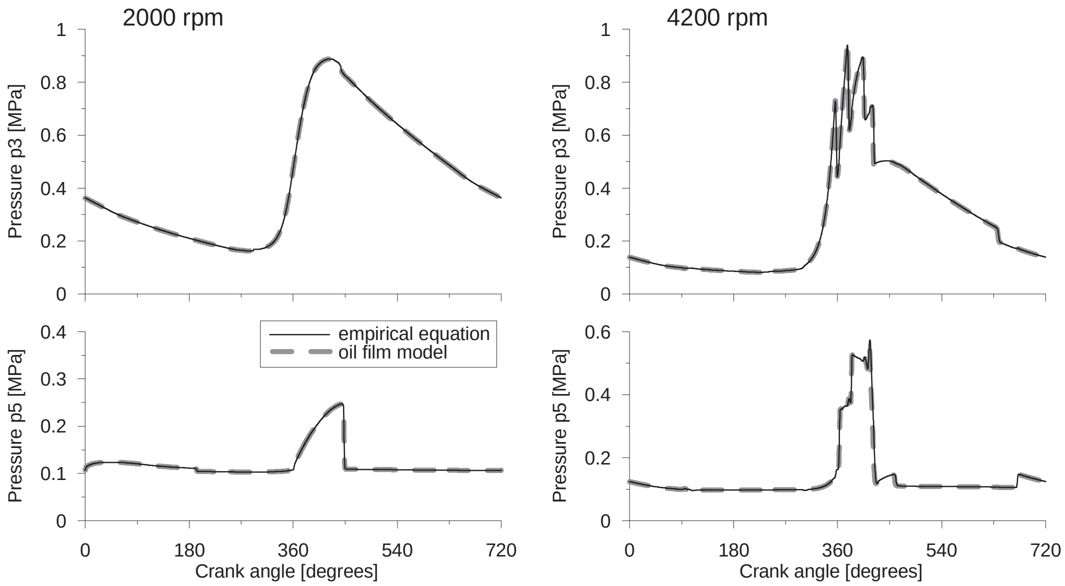

Fig. 6. Pressure courses in the inter-ring spaces calculated for friction forces calculated using empirical equation and oil film model for the engine full load at 2000 rpm (left) and at 4200 rpm (right)

4. Conclusions

The aim of the numerical investigation was to determine the effect which the method of determination of the friction force between the ring and cylinder liner can have on the results of the simulations run with the use of the model of the gas flow from combustion chamber to the crankcase and piston ring motion. In the first method the friction force was calculated from the simple empirical dependence. In the second one it was derived from the hydrodynamic model of lubrication which had to be earlier fully integrated with the original model of gas flow and ring motion.

Despite the fact that the friction forces calculated with these two methods differed considerably in certain ranges of crankshaft angles, differences between values of the gas pressure in inter-ring spaces, axial displacements of rings in their grooves and blowby rates calculated with the two methods were considered as insignificant. This was because the friction force was the smallest of all the considered forces acting on the ring.

Therefore, it can be concluded that it is not necessary to integrate the gas flow and ring motion model with the model of ring lubrication, and doing so to complicate it very much, if the aim of the calculations is simulation of blowby and rings behaviour in the grooves.

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