VARIABILITY OF THE RESISTANCE TO MOTION AT ENGINE CRANK-PISTON SET

Wojciech Serdecki, Piotr Krzymień

Poznań University of Technology
Institute of Combustion Engines and Transport
Piotrowo 3, 60-965 Poznań, Poland
tel. +48 61 6652243
e-mail: wojciech.serdecki@put.poznan.pl

Abstract

The resistance to motion that accompany the combustion engine run affects its technical and economical indices. Appropriate modifications in design of constitutive parts as well as proper selection of kinematic pairs' collaboration data might lead to the minimization of resistance, above all the frictional one. In order to estimate precisely this resistance as well as the area and reasons of their generation certain measurement methods are implemented that facilitate size of resistance and changes occurring within a single cycle.

Beside methods requiring interference into engine's construction other methods are used, among other the motoring method. Despite the advantages like simplicity, there are also problems with the use of this method. The most important are: different measuring conditions (no fires) and measurement of the resistance average value.

In a course of described simulations evaluation of mutual relations between friction forces and torques relative to different sources have been carried out, in particular those relating to the crank – piston assembly. A trial to give the answer if it is possible to distinguish the effect of individual resistances at certain engine subassemblies in the total resistance torque was the main effect of presented analysis.

Exemplary calculations have been carried out for the 170A.000 type engine.

***Keywords**: IC engine, piston-cylinder assembly, friction, friction losses*

1. Introduction

Reduction in motion resistance experienced during engine operation requires a thorough analysis of their sources and eventual prevention. In Fig. 1 the principal forces and torques have been marked on a schematic of the engine piston-cylinder assembly.

a) b)

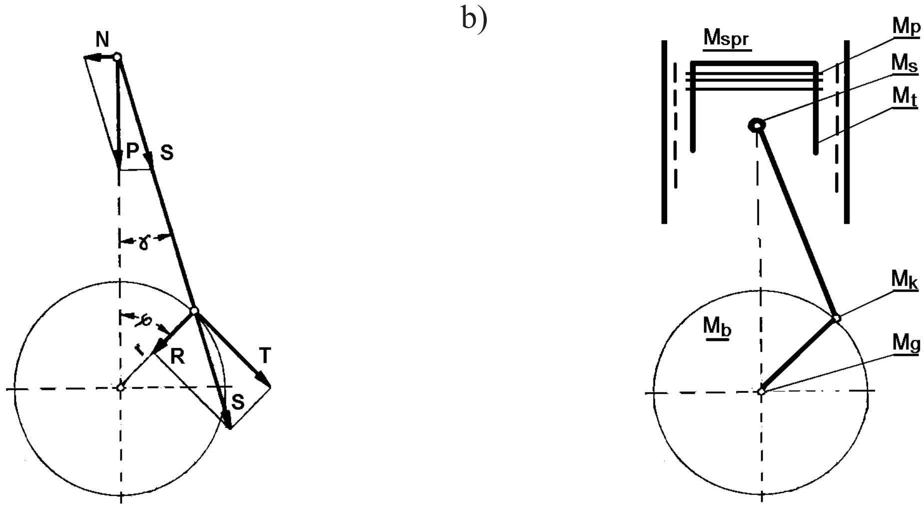


Fig. 1. Forces on engine crank mechanism (a) and components of resistance torque (see text)

Torque produced by an engine is equal to a product of tangential force T and crank radius r

$$M_o = T \cdot r\,, \tag{1}$$

where the tangential force is given according to formula (symbols as in Fig. 1a)

$$T = P\frac{\sin(\varphi + \gamma)}{\cos\gamma}\,. \tag{2}$$

The P force, acting along the cylinder axis, is approximately equal to the vector sum of gas, inertia and friction forces. Its momentary value changes itself continuously along with the change in components relative directly to the engine run conditions, e.g. rotational speed or load. As the results available in literature prove, the contribution of friction in total force P is the minor one which makes that it is mostly neglected.

At engine start the starter generated torque should be big enough to overcome the engine resistance torque. On the engine piston-cylinder assembly this torque consists of friction torques M_t and M_p generated during collaboration of piston skirt and piston rings, respectively, with the cylinder liner as well as those generated at main and crank bearings, i.e. M_g and M_k, respectively (see Fig. 1b). An increase in resistance torque most often proves the deterioration of matching elements' collaboration which could be caused by their excessive wear, lube oil shortage, change in oil properties and so on. Particularly considerable effect on friction torque exerts the lube oil temperature, directly affecting its viscosity. An exemplary course of motion resistance components at engine start, depending on oil viscosity, has been presented in Fig. 2.

Beside the friction losses torque, the engine resistance torque constitutes of other torques relative to phenomena accompanying engine operation. These are: compression torque M_s, connected with the work of charge exchange, auxiliaries drive torque M_{up}, connected with operation of devices indispensable for engine run like oil pump, fuel pump, alternator, etc. The torques corresponding to the drive transmission and charger drive can be also classified as constituents of this group. Engine resistance torque is also affected to the considerable degree by the inertia resistance torque M_b present at rotational speed variations.

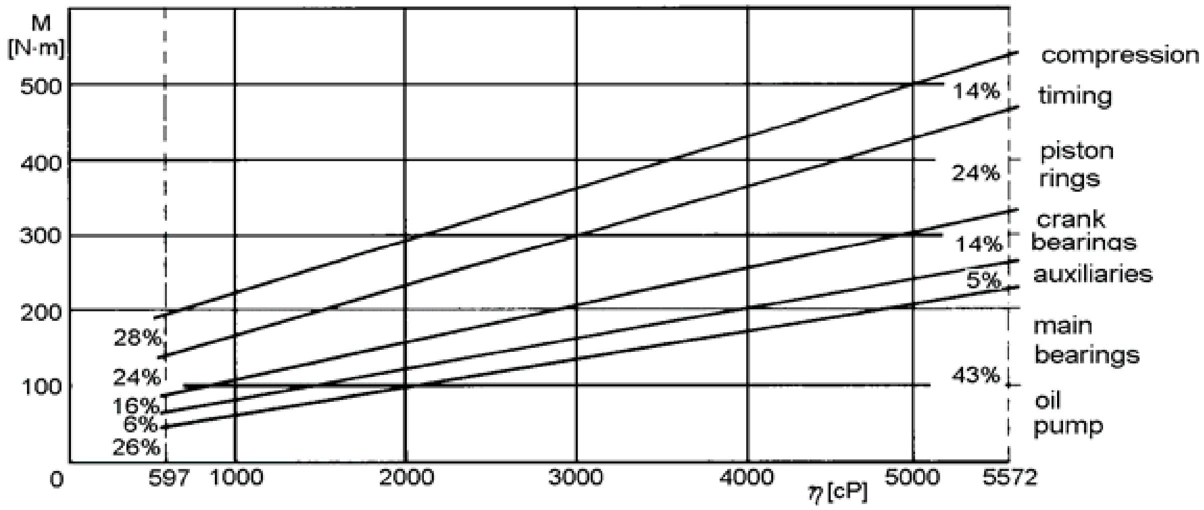


Fig. 2. Individual friction torques and their contribution to engine total friction torque vs. lubricating oil viscosity for n = 120 rpm [2]

A total resistance torque can be expressed as:

$$M_{op} = M_t + M_{up} + M_s + M_b\,. \tag{3}$$

The method of engine motoring is the one serving for engine resistance motion torque evaluation on a test stand. The combustion engine is driven by the electrical motor in a course of measurements. The engine designated for tests may remain complete (only fuel supply is cut off), which means that the pressure in cylinder can reach the value of compression pressure. In that case friction, inertia and compression torques should be taken into account when evaluating the resistance torque (3). Also tests on incomplete engine, i.e. with a cylinder head dismantled can be carried out and in this case the total torque does not include the compression torque.

A suitable test stand as well as the computational program pack that allow to measure and evaluate those torques are at the authors' disposal. Especially those programs are very useful for the introductory analysis of resistance torque generated during engine motoring.

The aim of presented paper is an analytical trial to evaluate the resistance torques present only in piston-cylinder assembly, especially to estimate the friction losses resistance that accompany the piston movement along cylinder when engine is motored with electrical motor. Moreover, we tried to answer if it is possible to connect the value of resistance torque which is complex and instable, with phenomena occurring in engine selected kinematic pairs.

2. Input data

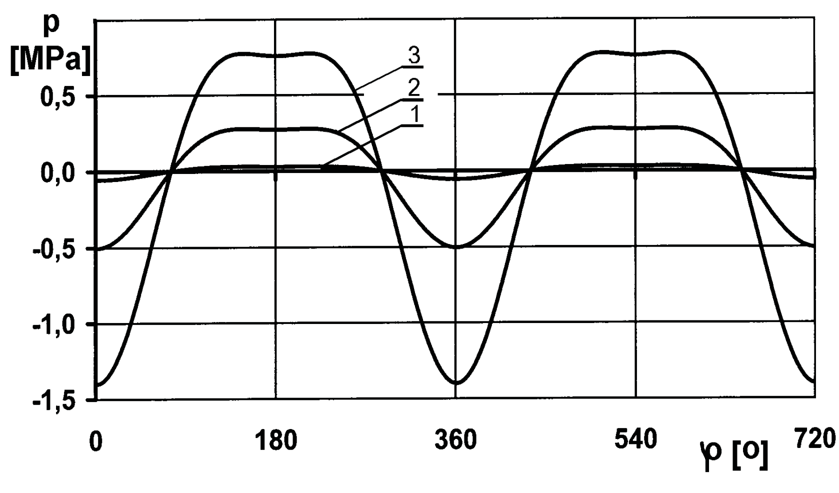
An analytical model of the engine piston-cylinder assembly has been used in simulation tests. Detailed description of models used for calculations of oil film parameters as well as input data one can find in previous publications of the authors, e.g. [3,4]. Technical data of the 170A.000 engine served as input data of simulations, whereas the SAE 5W/40 grade synthetic oil was used as lubricant.

Beside the resistance due to friction in piston-cylinder set and inertia also compression resistance has been taken into consideration in simulation tests. A series of tests without compression (dismantled cylinder head) has been carried out as well. Two values of oil temperature, i.e. 20°C and 80°C were used in order to evaluate the effect of oil viscosity on motion resistance**.**

3. Results and analysis

Courses of specific forces (relative to the piston crown area) presented in Fig. 3 have been determined for a motored engine (calculation have been carried out for a single cylinder). As expected, along with the increase in engine speed the motion resistance also increases, but the increase in friction forces is slower than that of inertia forces. This is caused by the different influence of speed on those forces leading to motored engine motion resistance.

a)



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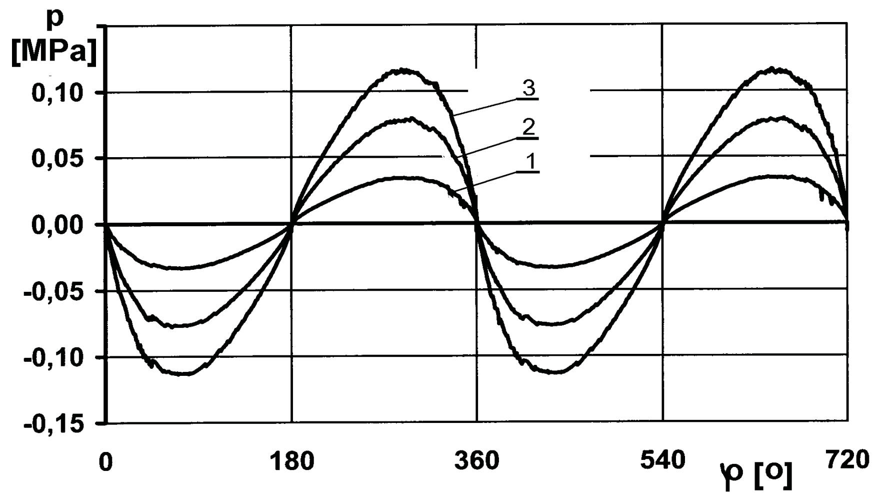

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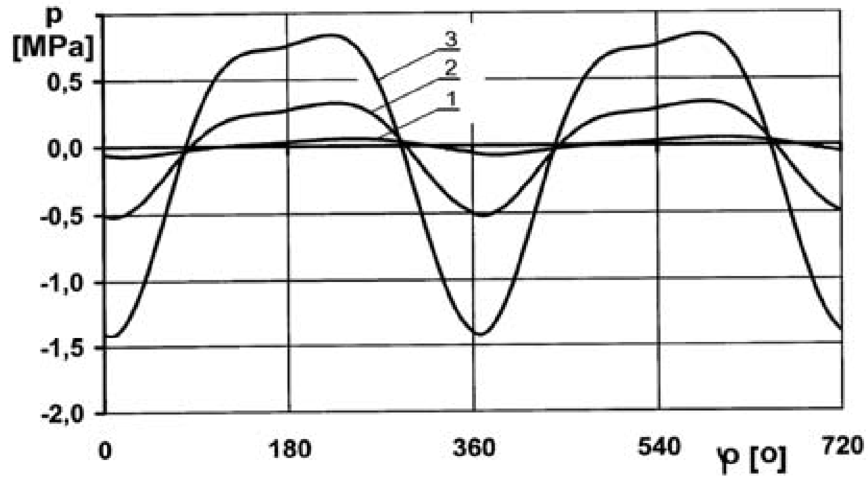


Fig. 3. Courses of inertia (a), friction (b) and total (c) specific force vs. crank angle for selected velocities: 1 – 1000 rpm, 2 – 3000 rpm, 3 – 5000 rpm; t_{ol} = 20°C; gas force neglected

For a coaxial crank mechanism the value of specific inertia force p_b can be defined as:

$$p_b = 4 \cdot \pi \cdot n^2 \frac{m_p \cdot r}{(30 \cdot d)^2} (\cos\varphi + \lambda \cdot \cos 2\varphi), \tag{4}$$

where:

n – crank shaft rotational speed,
m_p – mass of parts in reciprocating movement,
r – crank radius,
φ – crank angle.

Taking into consideration merely the oil wedge effect (one of the effects resulting from hydrodynamic theory of lubrication, present at fluid friction) the following approximate formula facilitates definition of specific friction force p_t accompanying the piston motion along the bore:

$$p_t = 4\pi \frac{\eta \cdot u \cdot b_f}{d \cdot h_m} T_u, \tag{5}$$

where:

u – ring speed,

η – lubricating oil viscosity,

h_m – minimum oil film thickness,

t – friction force of ring against bore,

d – bore diameter,

T_u – shape coefficients.

As it comes from the above formulas, inertia forces present at the piston-cylinder mechanism are proportional to the square of rotational speed (4), while the friction force – simply to the speed of piston (5). It means that the specific contribution of friction force in total resistance decreases with the increase in engine rotational speed.

When the test stand examinations are carried out using motoring method, instead of measurement of forces in the crank mechanism, a measurement of resistance torque is being performed, using for instance a precise torquemeter. The results of introductory torque measurements, carried out using a mathematical model of piston-cylinder assembly have been presented in Fig. 4.

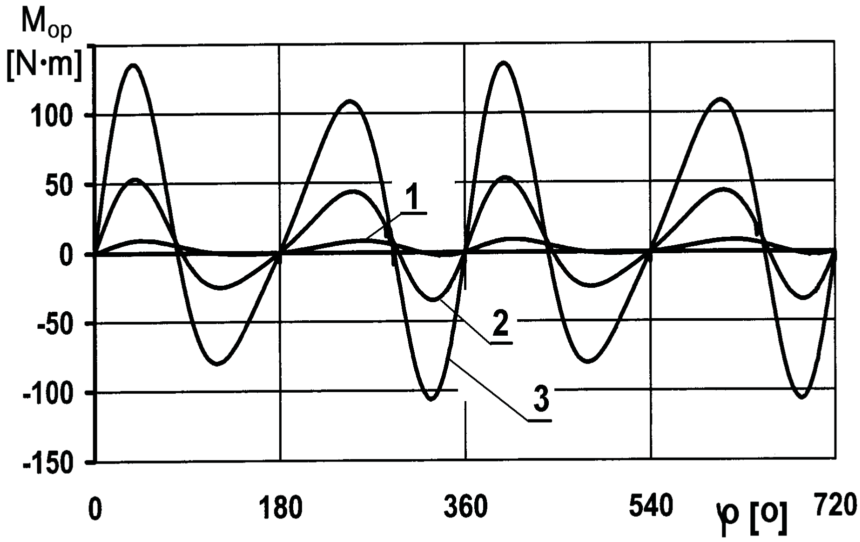
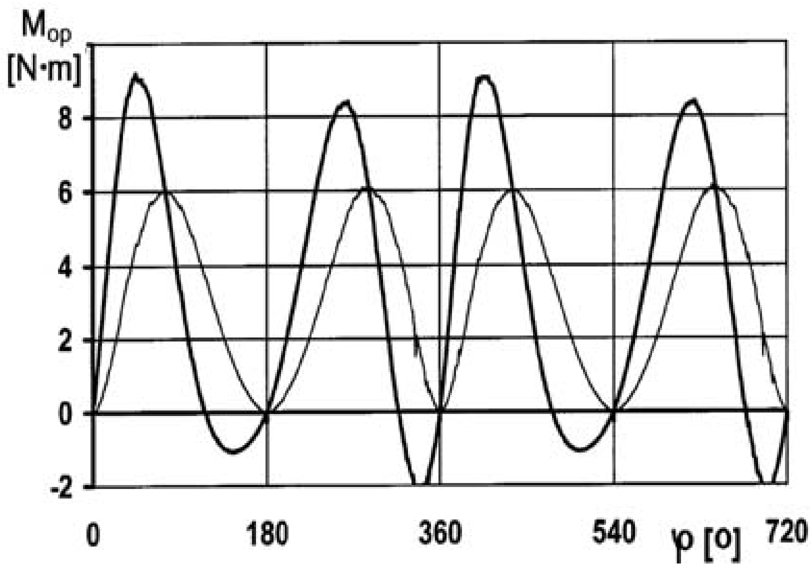


Fig. 4. Course of resistance torque vs. crank angle for selected velocities: 1 – 1000 rpm, 2 – 3000 rpm, 3 – 5000 rpm; t_{ol} = 20°C; gas force neglected

a)



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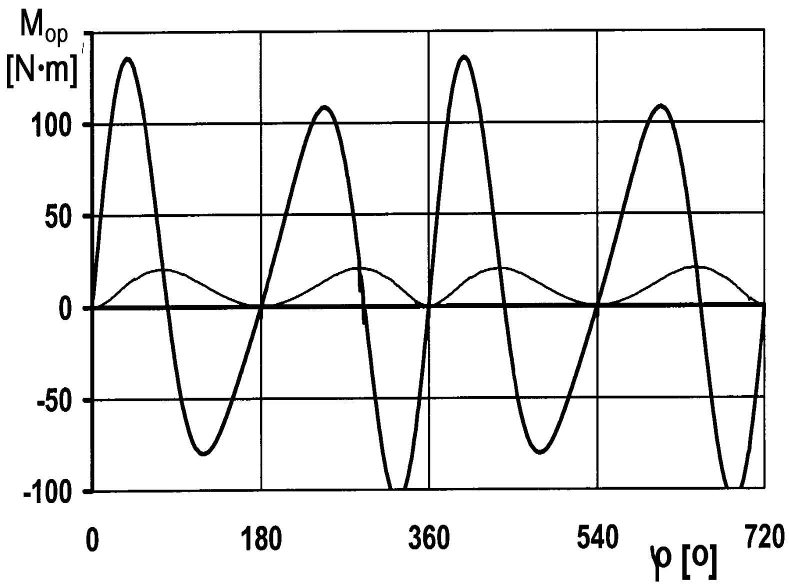


Fig. 5. Course of total resistance torque (bold line) and torque due to friction (thin line) vs. crank angle for velocities of 1000 rpm (a) and 5000 rpm (b); t_{ol} = 20°C; gas force neglected

For the presented examinations it is particularly essential to determine mutual relations between courses of resistance torque and the total torque, i.e. this torque recorded by the torquemeter.

As it follows from tests performed for two exemplary speeds, at the speed of 1000 rpm the value of friction torque is comparable to the total torque (Fig. 5a), while for the higher speed this torque is few times lower than the total one (Fig. 5b).

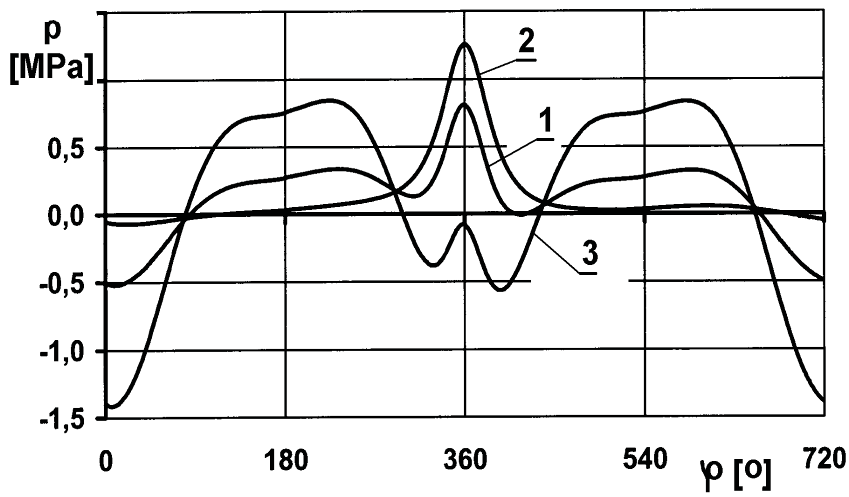


Fig. 6. Course of specific force of resistance vs. crank angle for selected velocities:
1 – 1000 rpm, 2 – 3000 rpm, 3 – 5000 rpm; t_{ol} = 20°C; compression taken into account

Taking compression into consideration significantly affects the course of specific motion resistance (see Fig. 6). Comparison of courses presented in Fig. 3 and Fig. 6 proves that differences are considerable for crank angle corresponding to the compression.

At the same time, the obtained results show that within the analyzed area the contribution of friction forces in resultant force is quite minor (Fig. 7) regardless of speed range. Similar conclusions concern the contribution of friction torque to the total resistance torque (Fig. 8).

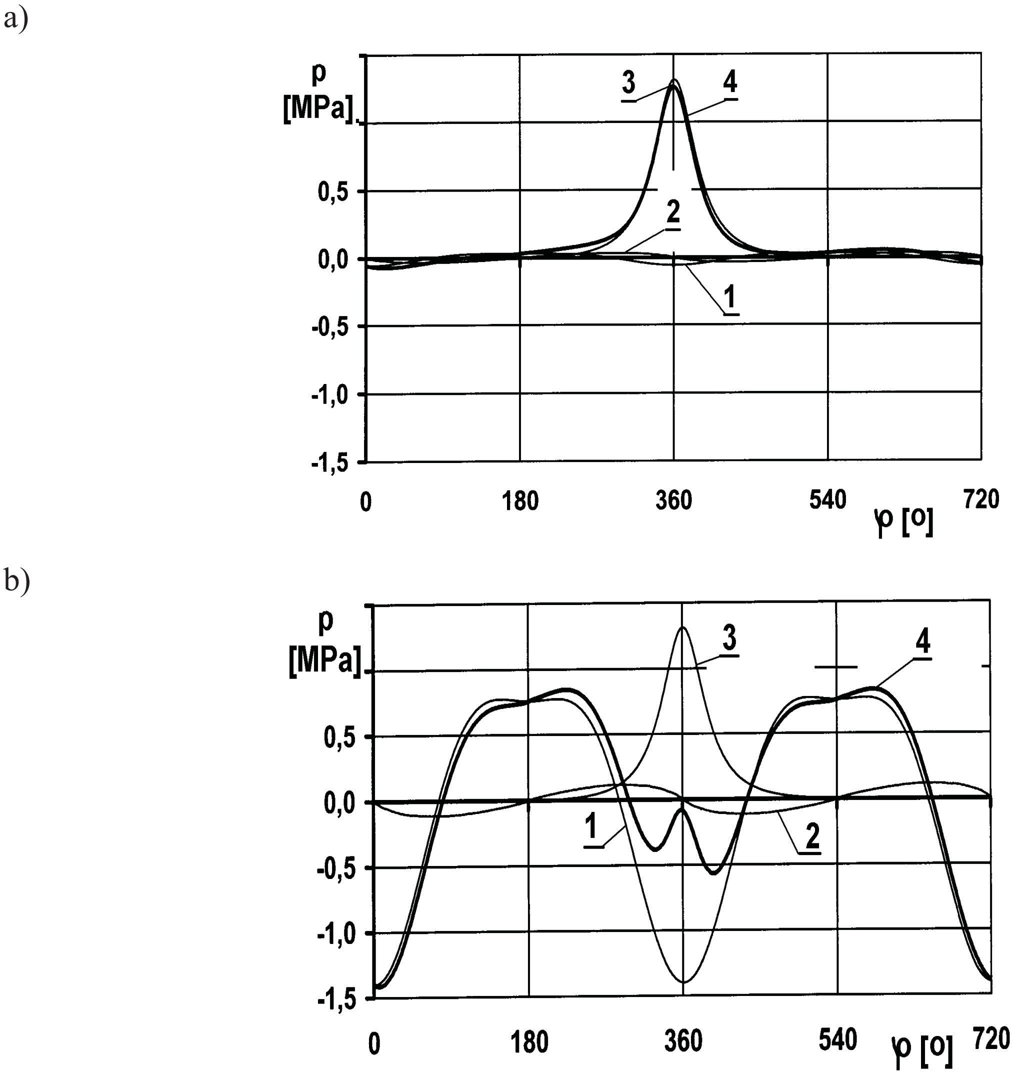


Fig. 7. Course of inertia (1), friction (2), gas (3) and total (4) specific force vs. crank angle for velocities of 1000 rpm (a) and 5000 rpm (b); t_{ol} = 20°C; compression taken into account

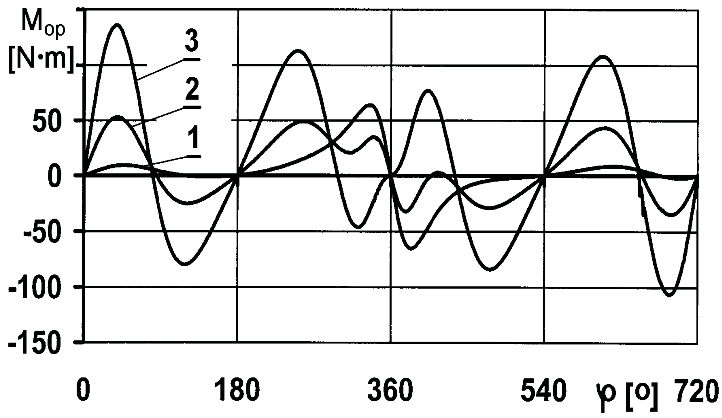


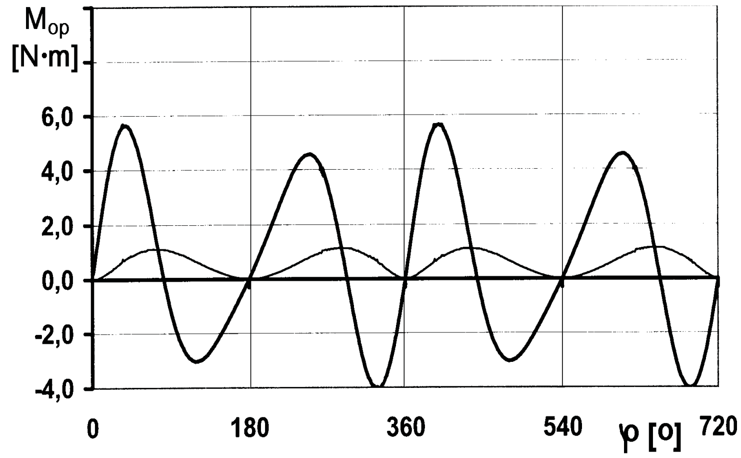
Fig. 8. Course of resistance torque vs. crank angle for velocities of: 1 – 1000 rpm, 2 – 3000 rpm, 3 – 5000 rpm; t_{ol} = 20°C; compression taken into account

Model computations presented above were carried out for 20°C of lube oil temperature, which means that oil viscosity was quite high. The computation were repeated for the temperature of 80°C, which corresponds to the oil dynamic viscosity of about 0.01 Pa·s, in order to determine the effect of oil temperature on the magnitude of resistance torque. Some selected, crucial for the performed analysis results have been presented in Fig. 9.

The maximum friction torque is far lower than that presented in Fig. 5, which makes that its contribution to the total resistance torque is also lower. This has been confirmed by the analysis of the *K* parameter (this parameter is a product of maximum friction torque and total torque expressed as percentage) variations for different conditions of computations (Fig. 10).

The emphasized courses show that the highest contribution of friction torque to the resistance torque appears at low crankshaft speed and high oil viscosity.

a)



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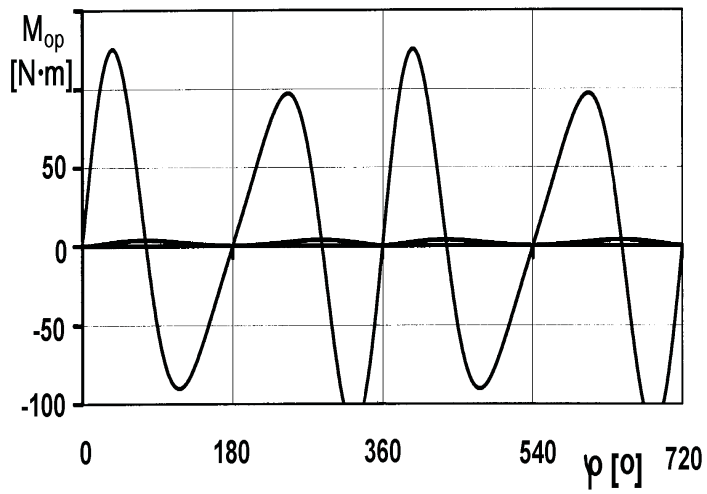

Fig. 9. Course of total resistance torque (bold line) and friction torque (thin line) vs. crank angle for velocities of: 1000 rpm (a) and 5000 rpm (b); t_{ol} = 80°C; gas force neglected

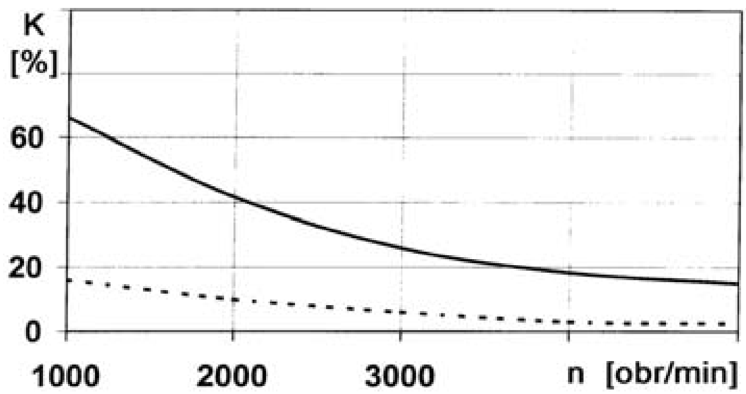

Fig. 10. Percentile contributions of friction torque in total resistance torque calculated for selected values of rotational speed; t_{ol} = 20°C – continuous line; t_{ol} = 80°C – dot line

4. Summary and conclusions

The model analysis presented in the paper allows concluding:

- the value of resistance torque caused by the inertia and compression forces exceeds the resistance torque due to friction within the range of speed variations covered with the tests,
- both friction torque and inertia torque increase along with the increase in rotational speed, however it happens differently (the increase is far faster for torque related to inertia),
- magnitude of friction torque considerably decreases along with the increase in lubricating oil temperature,
- the highest contribution of friction torque to the total resistance torque takes place for low speed and high oil viscosity,
- this contribution decreases, especially within the region of compression, for the case where the compression is taken into account.

Presented conclusions and observations resulting from the model analysis should contribute to the choice of the best torque measurement conditions directly at the test stand.

Literature

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