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DETERMINATION OF CRANKTRAIN FRICTIONAL RESISTANCE USING THE MOTORED ENGINE METHOD

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Abstract

Friction losses generated in engine kinematic nodes have an essential effect on its general efficiency. Measurement of these losses and indication of areas where they are generated can contribute to their minimization at the stage of engine design and during its operation as well. Most of the loss measurement methods requires an intervention in engine construction. The method of engine motoring does not require such intervention and at the same time offers possibility of precise measurement of engine resistance. The results obtained with the use of this method are considered hardly precise (among others because of different conditions during engine normal operation and its motoring), nevertheless they allow to estimate the influence of various effects on resistance observed on an engine.

This study presents the results of tests on the effect of certain quantities characteristic for engine operation as well as properties of lube oil applied on the course of instantaneous values of motored engine torque. Another achievement of the study is the recommendation of test conditions securing highest possible accuracy of engine resistance torque determination. The simulation tests were carried out on the 170A.000 engine.

Keywords: *piston-cylinder assembly, friction, friction loss assessment*

1. Introduction

Torque generated by the engine and transmitted to the power receiver results from following forces: gas force (decisive about torque course), inertia force and friction force (resulting in resistance torque reducing the gas force originated torque). In the case of multicylinder engine the resultant torque is a sum of torques generated in individual cylinders (see Fig. 1b for two-cylinder engine).

For a technically fit engine a typical course of torque instantaneous value corresponding to selected regimes of operation can be easily determined. Presence of any deviation can prove about irregularities that have occurred in operation of engine functional subassemblies (for instance about incorrect collaboration between elements of kinematic pairs). Recognition of interdependencies between the course of torque and engine technical condition and processes proceeding at its kinematic nodes and can be used for engine diagnostics.

Considerations on mutual relations between components of torque (relative to piston-cylinder assembly) have been carried out in [3, 4]. It was proved there that the component relative to the gas force has the most significant effect on the torque while the other components have far lower effect (and could be omitted in typical, less accurate computations).

a) *b)*

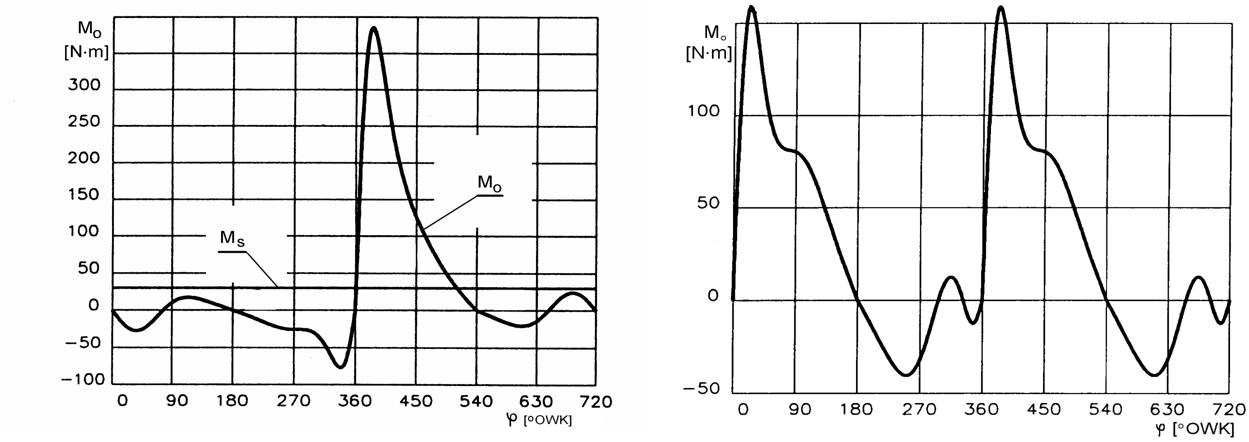

Fig. 1. Exemplary courses of 4-stroke engine torque instantaneous valueone-cylinder (a) and two-cylinder (b), M_s – mean torque [1]

As it can be noticed in Fig. 2, the torque component derived from friction force (important for presented analysis) is far lower than that produced by the gas force. This means that it is very difficult to detect changes in its value during measurement of torque transmitted from engine to the power receiver.

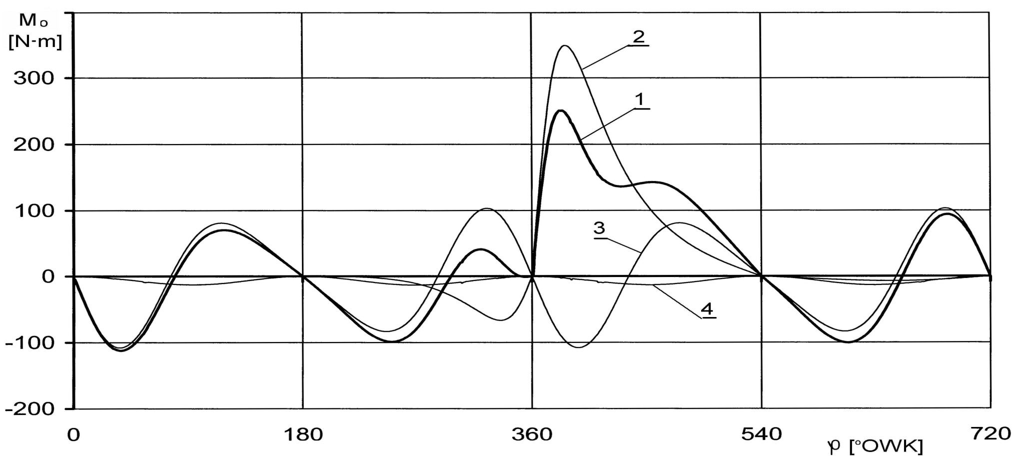

Fig. 2. Theoretical course of torque (1) and its components caused by following forces: 2 – gas force, 3 – inertia force, 4 – friction force, vs. crankshaft angle

An effective evaluation of the course of resistance torque caused by friction forces could be possible if the torque component relative to gas force (resulting from compression or combustion) was eliminated. Negligence of gas force means the need for tests on unfired engine which can happen when the engine is motored and does not generate the torque. Such method of investigation is called “engine motoring method”. This method consists in making unfired engine’s crankshaft rotate by the external drive. The drive train most often consists of electric motor, which can also operate as power generator and constitute the engine load. During the measurements a torque necessary for performing the rotation of driven engine crankshaft is being settled. The torque value can be find measuring the energy transmitted to the motoring electric motor or measuring the moment of housing reaction. Although the results obtained using this method are considered less accurate (due to e.g. different conditions of fired and motored engine) still they allow to estimate the effect of various quantities on resistance to motion in engine individual subassemblies. The engine selected for tests should be deprived of fuel supply which means that there is no combustion and the pressure in cylinder corresponds to the compression pressure.

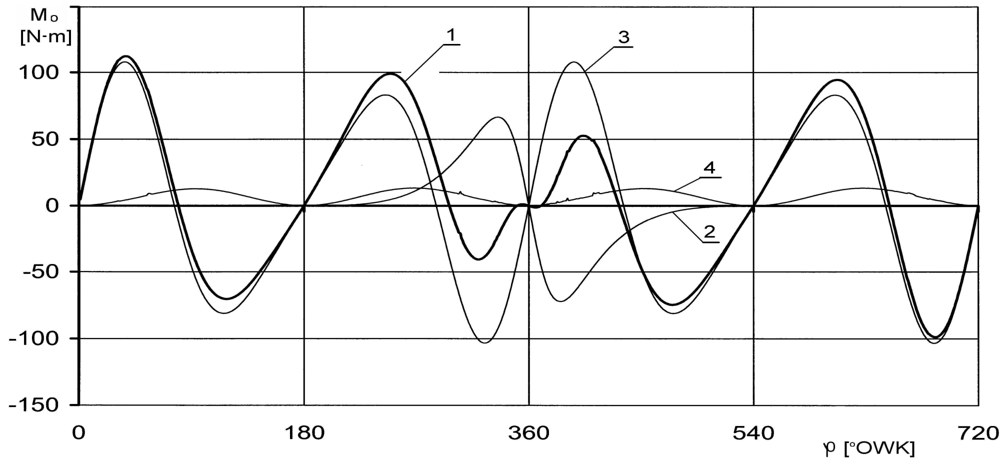

Fig. 3. Theoretical course of torque (1) and its components caused by following forces: 2 – gas force (no firing), 3 – inertia force, 4 – friction force, vs. crankshaft angle

However, the component relative to gas force is far lower for a case of unfired engine than for fired one it is still substantially higher than the component relative to friction force (see the course of resistance torque in Fig. 3). A complete exclusion of gas force is possible until after dismantling the cylinder head. Then the total resistance torque consists exclusively of components dependent on inertia and friction forces.

A further part of this study will present results of tests aimed at definition of the best conditions for carrying out the measurements of engine resistance to motion using the motoring method.

2. Course of measurement

Exemplary simulation tests presented in the further part of this study, have been carried out for the 170A.000 engine (Table 1) motored with a motor of adjustable rotational speed. Assumption on the lack of cylinder head made that the model takes into consideration only selected parameters that influence the changeable inertia and friction forces acting in piston-cylinder set. The synthetic oil of SAE 5W40 grade as well as semisynthetic SAE 10W40 and mineral SAE 15W40 ones have been selected for lubrication. Assumed temperature was 0°C, 20°C and 40°C while the rotational speed was limited to 100 rad/s.

Tab. 1. Basic technical data of the Cinquecento 170A.000 engine [6]

Parameter	Cinquecento ED 700
Engine swept volume, cm^3	704
Cylinder diameter, mm	80
Stroke, mm	70
Compression ratio	9
Max. torque (n = 3000 rpm), Nm	52
Max. power (n = 5000 rpm), kW	23
Number of cylinders	2 (in-line)

The assumption on temperature various values led to the need for definition of viscosity at those temperatures (oil viscosities defined according to [3] are collected in Table 2).

Tab. 2. Dynamic viscosity of the Elf lube oils at selected temperatures [5]

Rodzaj oleju	T = 0°C	T = 20°C	T = 40°C
SAE 5W/40 – synthetic	0.424	0.163	0.0717
SAE 10W/40 – semisynthetic	0.686	0.211	0.0822
SAE 15W/40 – mineral	0.979	0.262	0.0932

An analytical model of the piston-crank mechanism constructed for computations concerns the engine with dismantled cylinder head. The effect of following factors on collaboration of engine drive train elements, especially on changes in friction force and power has been taken into consideration during tests:

- changes in crankshaft speed,
- various types of lubricating oil
- fluctuations in oil temperature.

Masses of engine drive train elements performing the reciprocating motion like piston, rings, pin and connecting rod have been taken into account when calculating the inertia forces.

3. Results of resistance torque computations

Fig. 4 presents exemplary results of calculation of resistance torque and its components for selected crankshaft angular speeds and different lubrication oils (data of measurement conditions are presented in caption). An advantage of torque relative to friction force over that relative to inertia forces is easily noticeable for angular speed values selected for computations. For other input parameters these relations could be different. The analysis presented in [3] proves that inertia force connected with the operation of power train elements is proportional to the square of crankshaft angular speed while the friction force is proportional to the square root of this speed. However, this correlation should be treated merely as the approximation because only the wedge effect has been taken into account. It should be noted here that the presented results have been obtained using a complex model taking into consideration oil film phenomena of greater importance [2].

Variations in the course of resistance torque relate to the fluctuations of forces in engine power train. Fig. 5 presents a collective results of resistance torque calculations brought about for selected crankshaft speeds (at constant oil temperature) and courses obtained for different temperatures (at constant crankshaft speed).

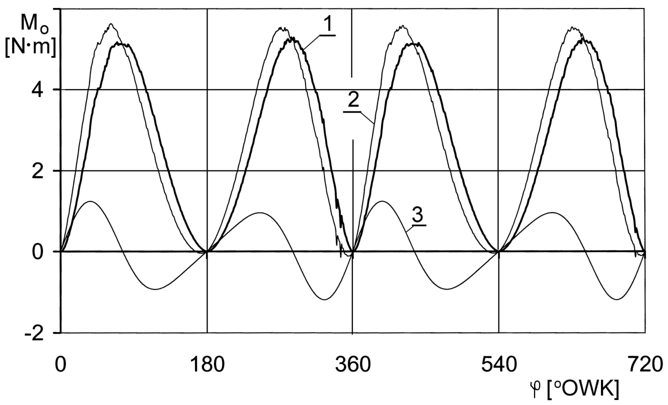

Fig. 4. Course of resistance torque (1) and its components relative to friction (2) and inertia (3) forces vs. crank angle, defined for SAE 5W40 grade lube oil; T = 20°C, ω = 50 rad/s

As expected, the resistance to motion increases with an increase in crankshaft speed and increase in oil viscosity while torque increase due to friction is slower than that due to inertia (what corresponds to presented previously dependencies). Courses presented in Fig. 5 show that for the analyzed range of input data inertia forces exceed the friction forces only on short sections of piston displacement and for the highest angular speed taken into account (total resistance torque is negative).

a) *b)*

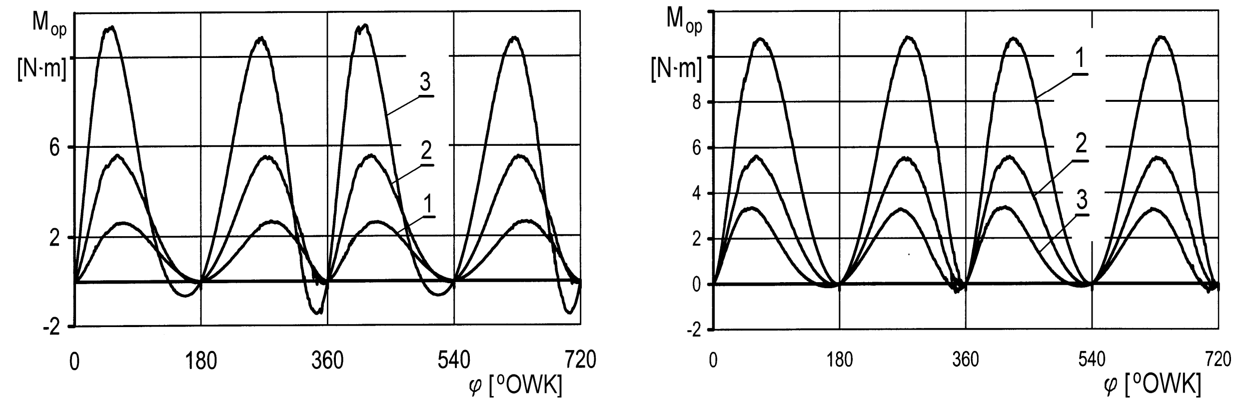

Fig. 5. Course of resistance torque vs. crank angle for selected angular speeds: 1 – 20 rad/s, 2 – 50 rad/s, 3 – 100 rad/s; for SAE 5W40 grade lube oil, t = 20°C (a), and different temperatures: 1 – 0°C , 2 – 20°C, 3 – 40°C;ω = 50 rad/s (b) [3]

Comparison of the course of resistance torque momentary value has a considerable diagnostic value but is not sufficiently precise for a global evaluation of individual factors contributing to motion resistance. Further part of this study compares not momentary values of torques but their mean values. Fig. 6 presents comparison of average values of motion resistance caused by friction (M_t) and inertia (M_b), for selected crankshaft speeds ω and oil temperature T. Computations were carried out for a synthetic oil of SAE 5W40 grade (Fig. 6a) and for mineral oil of SAE 15W40 grade (Fig. 6b). The results confirm earlier remarks on mutual proportions between these torques and about dependencies connecting those torques with crankshaft angular speed and lubricating oil viscosity.

a) *b)*

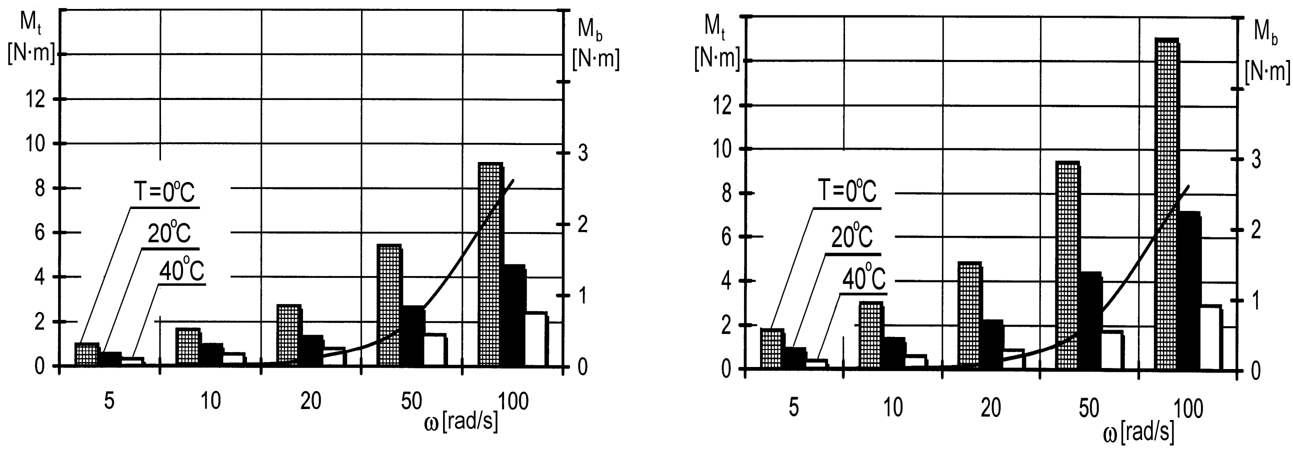

Fig. 6 . Arrangement of resistance torques mean values relative to friction M_t and inertia M_b (continuous line) for selected crankshaft angular speeds ω and oil temperature T: a –SAE 5W40 grade oil, b – SAE 15W40 grade oil

The measurement of friction torque would be more precise if the value of torque due to inertia was lower. In order to monitor mutual relations between those torques, a conception of AM factor has been introduced, that expresses relation of friction and inertia mean values. Higher value of this factor, more advantageous conditions for evaluation of friction torque.

When analyzing the course of A_M factor in Fig. 7 one can conclude that its value decreases with the increase in crankshaft angular speed regardless lube oil grade. This shows that the region of low crankshaft speeds is the most advantageous one for the foreseen investigations.

On the other hand, one should remember that with a drop in shaft speed conditions of collaboration between rings and cylinder liner deteriorate (an oil film rupture can happen and part of piston stroke in conditions of mixed lubrication extends). The quality of such collaboration can be described with relative displacement S expressed as the relation of stroke section covered in conditions of fluid friction (a sum of microunevennesses is smaller than the oil film thickness) to the entire stroke. As it outcomes from the hydrodynamic theory of lubrication, the length of this displacement increases with the increase in crankshaft speed and with drop in oil temperature (increase in oil viscosity).

a) *b)*

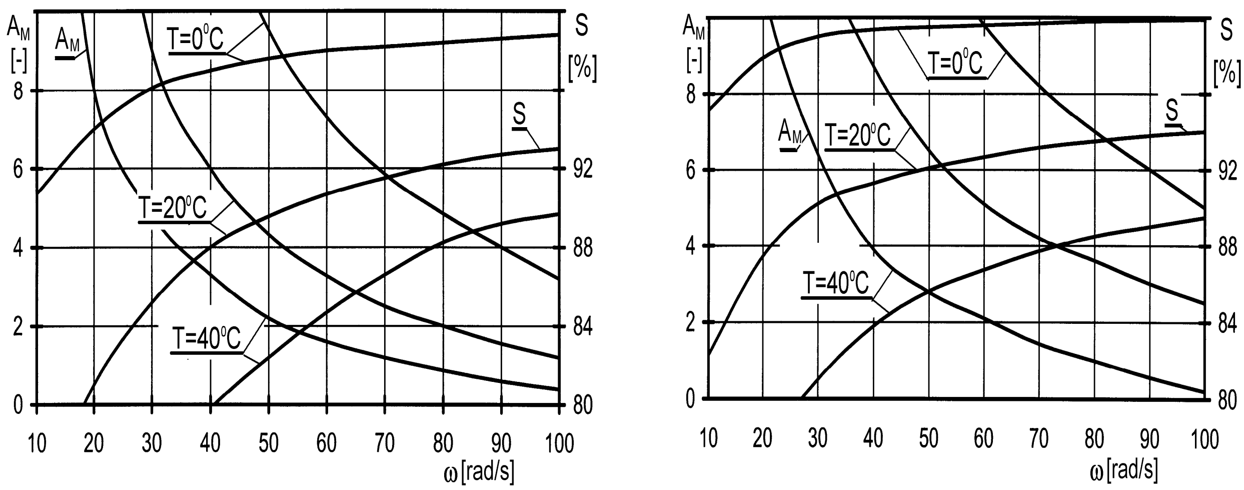

Fig. 7. Changes in A_M factor and S relative stroke vs. crankshaft angular velocity for selected temperatures of SAE 5W40 grade (a) and SAE 15W40 grade (b) lubricating oils

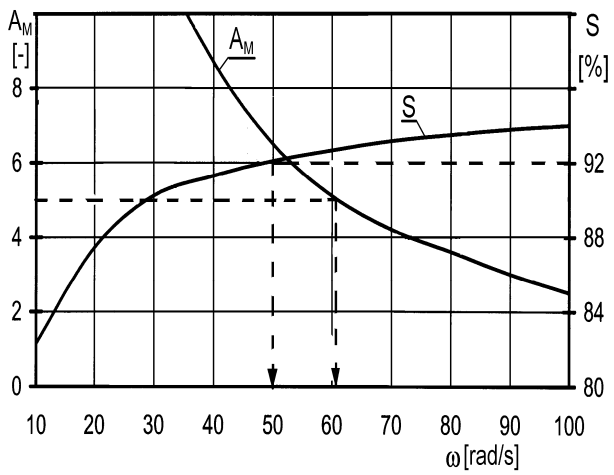

Fig. 8. Example of determination the crankshaft angular velocity corresponding to assumed input parameters of the test stand

Construction of diagrams presented in Fig. 7 is very time consuming (requires carrying out a number of simulation computations) but considerably eases a choice of best measurement conditions from the point of accuracy. A procedure of the choice has been presented in further example.

Assuming the input data as follows:

– measurement temperature: 20°C,

– A_M factor: > 5,

– S displacement: > 92%

one can read from the graph in Fig. 8 that for the SAE 15W40 grade oil the crankshaft angular velocity satisfying these requirements lies within the limits of 50 and 61 rad/s.

These graphs could serve also as a tool for estimation of A_M and S parameters for introductory assumed values of crankshaft angular velocity.

4. Summary and conclusions

Most important conclusions from the presented investigations are as follows:

– gas forces relative to the processes performed in engine cylinder are far higher than other forces acting in engine power train; similar proportions are observed in the case of torques relative to the forces mentioned,
– decrease in crankshaft angular speed results in decrease in contribution of inertia related torque in total resistance torque,
– lowering the measurement temperature leads to the increase in oil viscosity and eventually to the increase in resistance torque relative to friction,
– drop in crankshaft speed results in longer distance covered by piston in conditions of mixed lubrication.

The analyses presented in this paper are based on analytical model investigations and the only way to verify them is to carry out tests on a test stand (including models of engine power train).

Literatura

[1] Iskra A., *Dynamika mechanizmów tłokowych silników spalinowych*, Wydawnictwo Politechniki Poznańskiej, Poznań 1995.
[2] Serdecki W., *Badania współpracy elementów układu tłokowo-cylindrowego silnika spalinowego*. Monografia, Wydawnictwo Politechniki Poznańskiej, Poznań 2002.
[3] Serdecki W., *Analiza zmienności momentu oporowego silnika spalinowego małej mocy*, Materiały konferencji KONMOT-AUTOPROGRES 2008 (w druku).
[4] Serdecki W., *Zmienność sił i momentów tarcia w układzie korbowo-tłokowym podczas rozruchu silnika spalinowego*. W: Eksploatacja Silników Samochodowych. Zeszyt nr 16, Polska Akademia Nauk – Oddział w Lublinie, Szczecin 2007.
[5] Materiały informacyjne firmy ELF.
[6] Materiały informacyjneFIAT.