

ANALYTICAL RESEARCH ON FRICTION LOSSES IN PISTON - CYLINDER ASSEMBLY OF MOTORED ENGINE

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Abstract

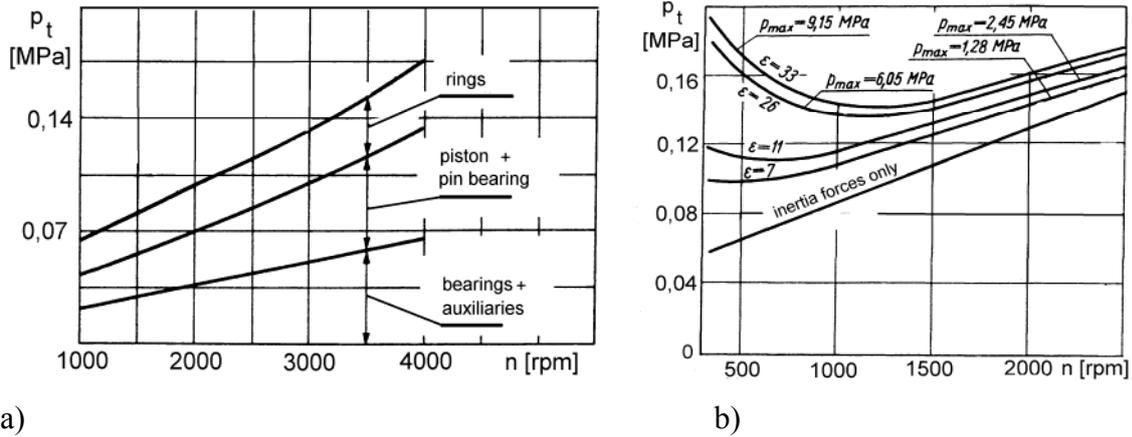
Acquired results and specification of tests presented in the following paper are merely fragment of greater project aimed at development of an engine instantaneous friction torque definition method. A test stand has been developed on which phenomena encountered at kinematic nodes of a motored engine are to be investigated. This paper describes a trial to establish – using analytical model – friction forces and power generated when ring pack moves along cylinder liner of a driven engine. Test have been carried out both for presence and absence of gas forces acting upon crank mechanism, on piston rings in particular. As mentioned above, tests have been carried out on a computer model where technical data of the R5TDI engine were used as input data. A mathematical model of piston-cylinder system developed using the hydrodynamic theory of lubrication has been applied to the computations. Both wedge and squeeze effects have been taken into consideration when determining the position of ring pack relative to the bore.

Results presented in this paper have been achieved using the technical data of VW R5 TDI engine developing about 100 kW power.

Keywords: *IC engine, piston rings, friction, friction losses, oil film*

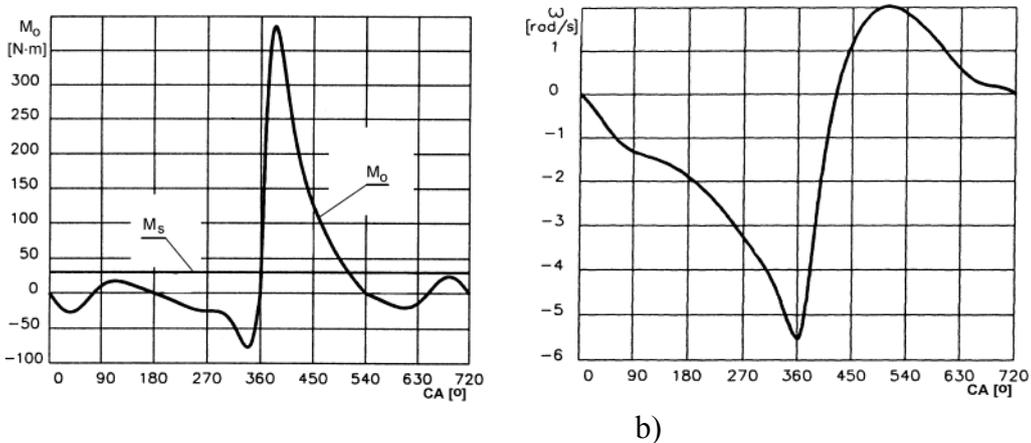
1. Introduction

Operation of certain elements and subassemblies of an engine generate losses of mechanical drag. The most important parts of engine from this point are: piston with rings moving along the cylinder liner, main and crank bearings, timing system and – to a certain extent – fuel, cooling and lubricating systems. As tests reveal, size of these losses depends on design and collaboration of mating elements as well as engine speed and load. Fig. 1 presents exemplary values of friction mean pressure p_t vs. engine speed for different magnitude of gas forces as well as division into individual components (it has been assumed that the mean friction pressure is a difference between the mean indicated pressure and mean effective pressure).



a) b)
 Fig. 1. Change in friction mean pressure p_t vs. rotational speed n for selected values of compression ratio ϵ and maximum compression pressure p_{max} (a), and divided into individual components of losses (b) [6]

Courses presented in Fig. 1 correspond to the changes in mean friction losses, while the phenomena performed during engine run change themselves within a single operation cycle, as the course of engine torque, for example (Fig. 2a). Its momentary value depends on the course of cylinder pressure, inertia forces as well as dynamics of intake and exhaust. Because the combustion is not identical for each cycle and differs for individual cylinders one might assume that the value of gas force has stochastic character. The changes in rotational speed have similar character. It means that in real engine speed experiences considerable fluctuations which corroborate both experimental and analytical tests (Fig. 2b).



a) b)
 Fig. 2. Exemplary variations of torque M_o instantaneous value (a) and crankshaft rotational speed ω (b), determined for a single cylinder 4-stroke engine; M_s – mean value of torque [1]

Knowledge of torque and speed fluctuations, i.e. quantities relative to friction torque generated in kinematic nodes of an engine can be employed to design efforts aimed at improvement of engine general efficiency.

Several methods of engine torque measurement have been developed, among which the following seem to be most important:

- methods employing the torsion angle of drive shaft section measurement,
- methods of measurement of reaction in engine support,
- methods employing the inertia of engine moving parts,
- methods of torsion angle and crankshaft vibrations measurement.

However, stand tests prove that application of those methods to the measurement of torque instantaneous value is but simple even on the stand furnished with measurement equipment of high accuracy. The point is that most often measured torque (M_s) is the average value contrary to the momentary value M_o that fluctuates substantially within a single operational cycle (see Fig.2a). Torsional vibrations of the engine – power receiver set are the additional factor responsible for disturbances in torque measurement. As a result of those phenomena pretty considerable difficulties are being encountered when one tries to set apart friction relative torque from the measured coupling torque.

The method of engine motoring is one of the possible methods that could be employed in order to estimate friction losses. In that method the tested engine is driven by the external drive, e.g. electric one. The tested engine could be complete and cylinder pressure is the compression pressure or incomplete (dismantled head, for example) but the operational conditions are far from typical ones. On the other hand, incomplete engine could be suitable in order to locate certain losses.

Evaluation of friction losses in combustion engine can not be carried out directly on the ground of engine-to-power receiver coupling torque fluctuation analysis because of inertia and gas forces that affect the momentary value of torque beside friction force. Taking it into account, one can assume with the high degree of simplification, that the friction torque generated during operation of crank mechanism elements equals the difference between total measured torque and torques relative to inertia and gas forces (assuming no auxiliaries introducing additional friction torques).

As mentioned earlier, friction torque generated during cylinder assembly operation consists of friction torques from piston rings piston skirt and both main and crank bearings. A suitable test stand and a set of simulation programs that facilitate measurement and evaluation of these torques at full engine operational cycle are at authors' disposal [3, 4, 5].

The aim of this paper is a trial of model assessment of friction losses accompanying the move piston ring over cylinder liner. Results acquired in a course of planned tests should facilitate the verification of test stand results.

2. Input data

Five cylinder in-line diesel type R5 TDI (AXD code) (engine basic technical data are in Table 1) is equipped with a ring set of two compression rings and one oil control ring (see Table 2). Measurements of rings geometry and elasticity have been carried out and exemplary profilograms of their faces have been presented in Fig. 3 (rings worked before tests and their face are worn, especially the second compression one).

Computation have been carried out assuming the tested engine motored by an electric machine which serves as AC dynamometer as well.

Two cases of stand operation have been recognized, namely
 the case I – motored engine piston is not subjected to gas forces (dismantled head),
 the case II – gas forces relative to compression pressure act on engine piston.

Table 1. Basic technical data of the R5 TDI engine

Parameter	Value
Engine swept volume, ccm	2460
Cylinder diameter, mm	81
Stroke, mm	95.5
Compression ratio	18.5
Number of cylinders	5 (in-line)

Cooling	water
Torque, Nm	320/2000
Power, kW	96/3500
Firing order	1 – 2 – 4 – 5 – 3

Table 2. Piston rings of the R5 TDI engine

Ring	Total height	Total height	Total width	Elasticity
First compression ring	2.5 mm	2.1 mm	3.0 mm	120 kPa
Second compression ring	2.0 mm	1.8 mm	3.0 mm	140 kPa
Oil control ring	2.7 mm a=0.38 mm	2.7 mm a=0.38 mm	2.5 mm	1.4 MPa (spring)

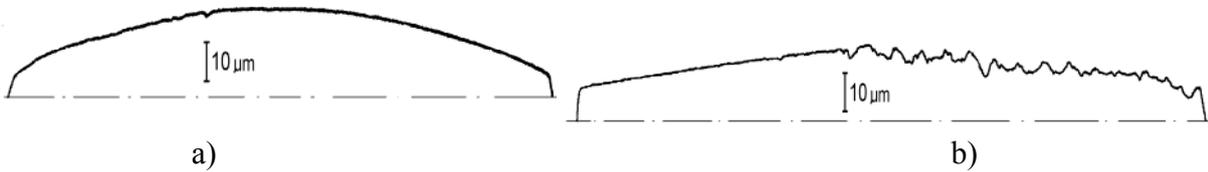


Fig. 3. Profiles of ring face: first compression one (a) and second compression one (b)

The synthetic lube oil 5W/40 of dynamic viscosity 0.1 Pa's (at 20°C) [2] has been selected for lubrication. The carried out tests concerned the relation between crankshaft angular velocity and oil film parameters on cylinder surface. The speed has been changed within the range from 500 to 4000 rpm.

3. Results and analysis

Courses shown in Fig. 4 have been drawn for the case I, i.e. no gas forces acting upon piston and rings. This assumption leads to the observed repetition of oil film thickness course of 360°CA period characteristic for two stroke engine.

On the other hand, for the case II computations (results have not been presented here) this period was 720°CA which corresponds to the 4-stroke engine cycle. As the analytical tests revealed, there is a minor effect of compression pressure on oil film thickness and noticeable differences in oil film thickness occur closely after TDC where a slight drop is observed (curve 2 in Fig. 5). Courses of oil film mean thickness $h_{m,s}$, calculated as arithmetic mean over the entire engine cycle, for both cases have been presented in Fig. 6.

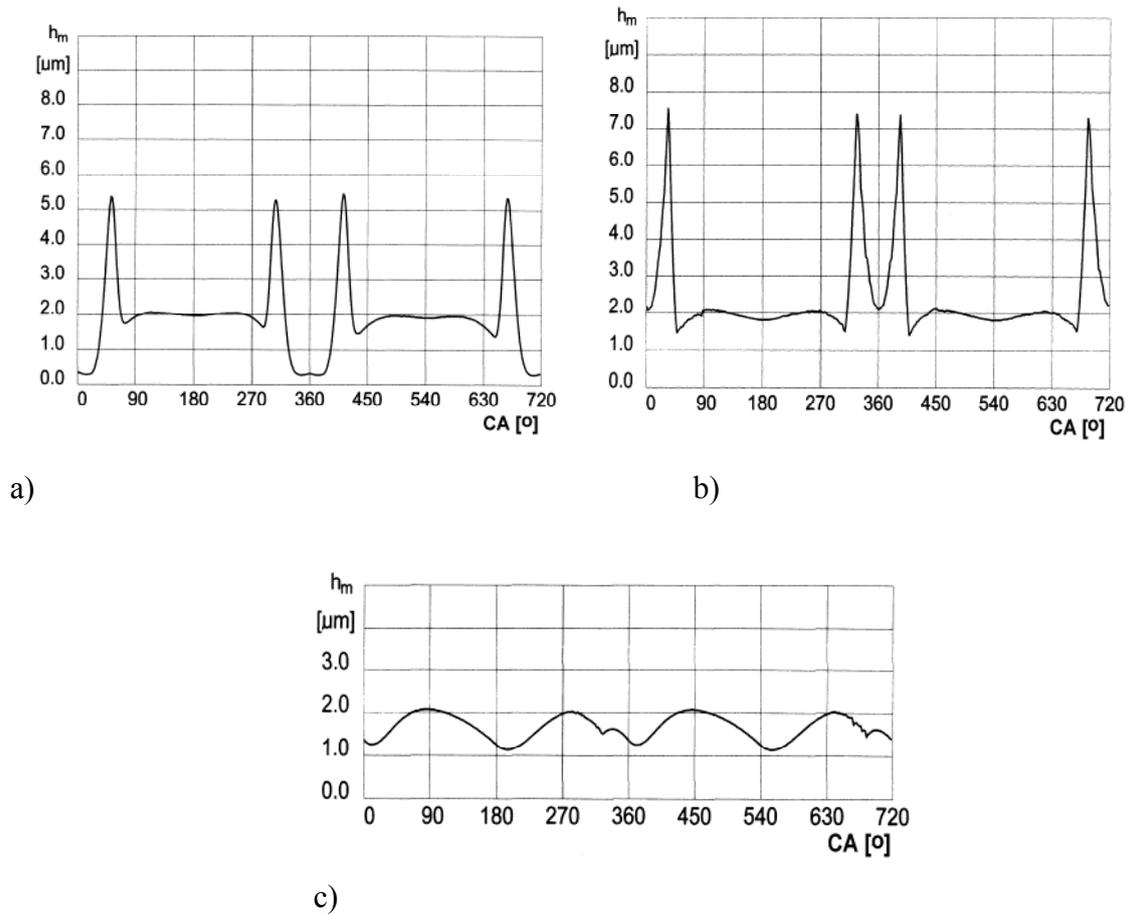


Fig. 4. Exemplary courses of oil film instantaneous thickness h_m under first compression ring (a) second one (b) and oil control ring (c) of the R5 TDI engine; case I, $n= 1000$ rpm

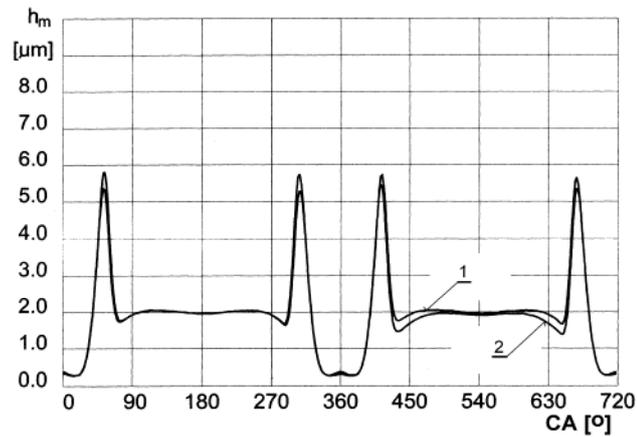


Fig. 5. Comparison of oil film momentary thickness h_m course under first ring, for case I (1) and case II (2), $n= 1000$ rpm

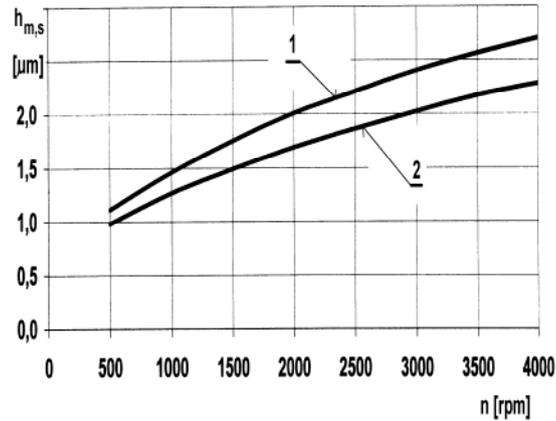


Fig. 6. Course of oil film average thickness $h_{m,s}$ under first compression ring, for case I (1) and for case II (2) of test stand operation

The instantaneous values of friction force t accompanying the movement of ring pack along liner have been calculated parallelly with oil film thickness (Fig. 7). The shape of these courses reflects such quantities as instantaneous friction force, ring speed and range of lubricating gap filling.

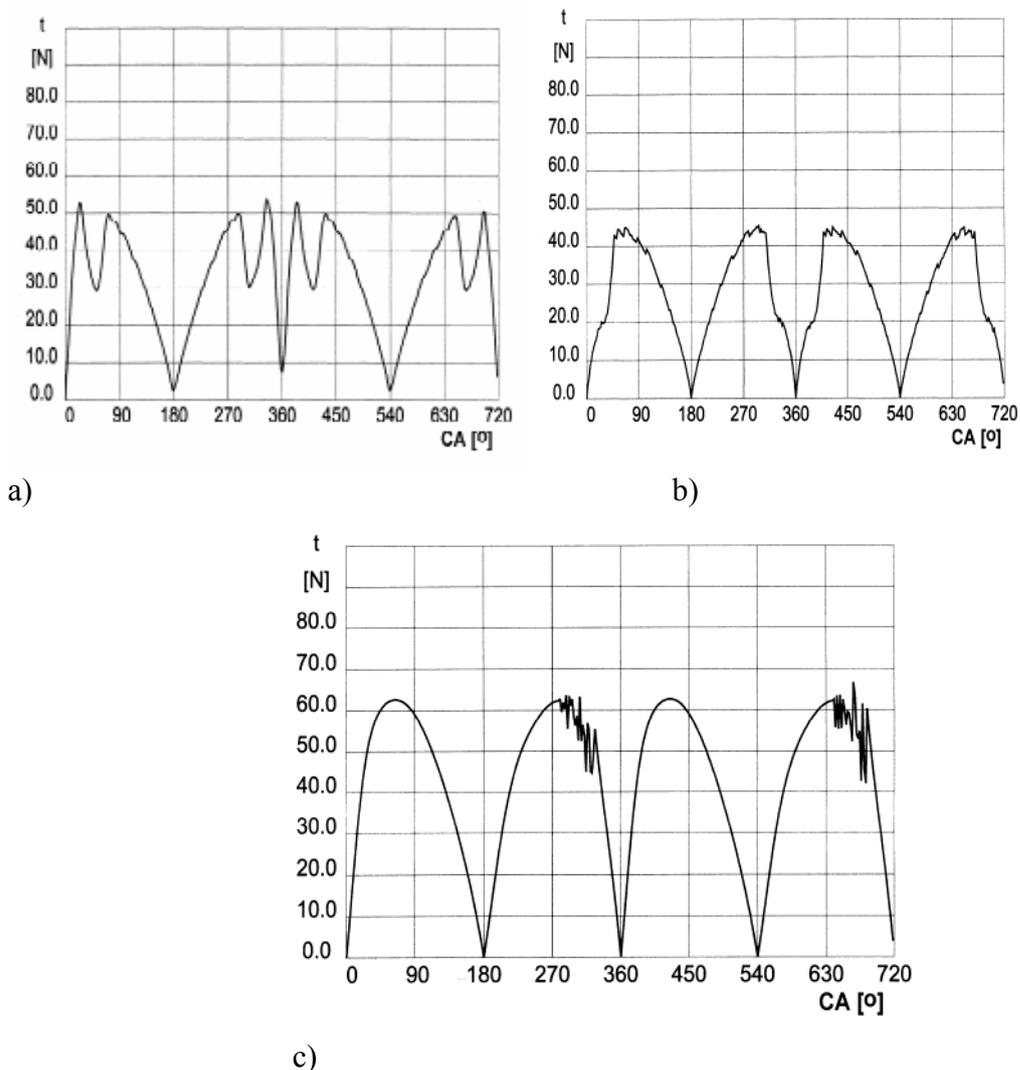


Fig. 7. Exemplary courses of momentary friction power t relative to movement of first (a) second (b) and oil control ring (c) of the R5 TDI engine; case I, $n = 1000$ rpm

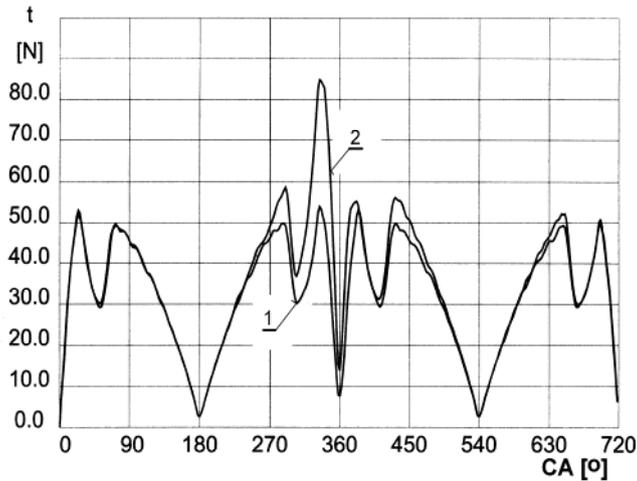


Fig. 8. Comparison of friction power t course for first compression ring; case I (1) and case II (2); $n= 1000$ rpm

Taking the gas force relative to compression into account results in an increase in friction force and the most noticeable differences could be observed closely to the TDC (Fig. 8).

Although the momentary differences in friction force value for both cases analyzed are of considerable size in the region of TDC, differences in friction power must not have been great because of short displacement. Courses shown in Fig. 9 illustrating changes in friction power N_t vs. crankshaft rotational speed for both cases corroborate this conclusion. The courses shown prove that the compression pressure affects the increase in friction power only to a limited extent and this increase is smaller when the analyzed ring is more distant from the combustion chamber.

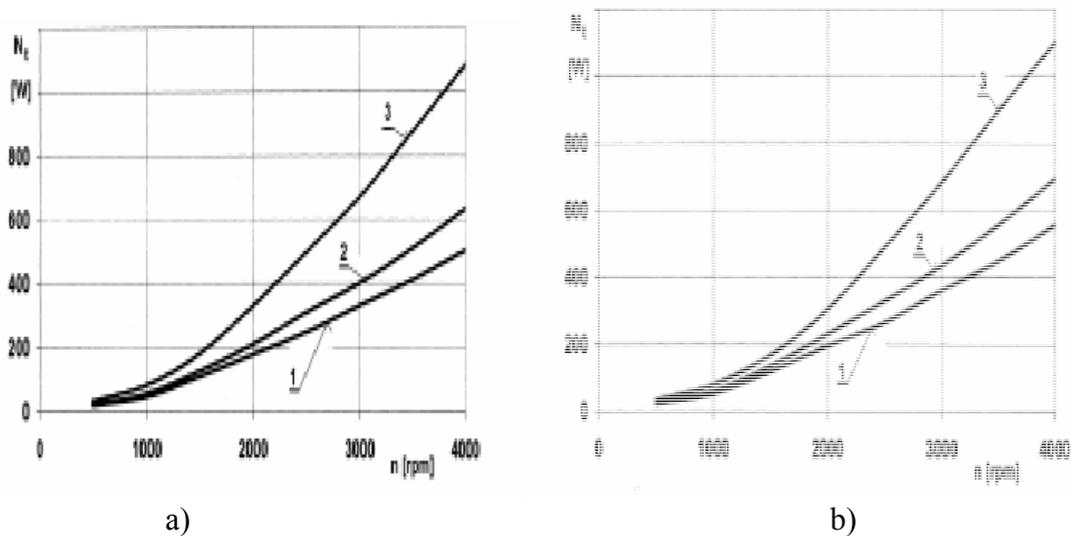


Fig. 9. Comparison of mean friction power N_t course relative to crankshaft rotational speed n , for case I (a) and for case II (b); 1 – first compression ring, 2 – second compression ring, 3 – oil control ring

4. Summary and conclusions

Considerations presented at the beginning as well as the analysis of carried out simulations allow to draw conclusions of which the most important ones are as follows:

- oil film thickness under the piston ring depends on its design and conditions of collaboration with mating surface; the speed of ring motion directly relative to the crankshaft speed is particularly influential,
- an increase in crank mechanism load caused by compression pressure leads to the drop in oil film thickness,
- friction force brought about by the ring motion depends on ring's construction and operational conditions; the effect of ring speed directly relative to crankshaft speed is particularly substantial,
- changes in friction force lead to changes in friction power that increases along with the increase in engine rotational speed; higher load of crank mechanism caused by gas force leads to an increase in power value.

Presented model investigations have been carried out assuming the constant lubricating oil viscosity. It should be stressed that during the tests on real motored engine the oil temperature increases with time. This increase brings about a fall in oil viscosity and eventual fall in motion resistance. Authors' opinion is that certain conducted tests should be repeated taking into account changes in oil temperature.

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