KAPITEL 3 / CHAPTER 3 [3]

EXPERIMENTAL RESEARCHES OF MECHANICAL LOSSES IN AN ENGINE WITH AN UNCONVENTIONAL POWER MECHANISM

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Introduction

Designing ICE is technologically complex and time-consuming process, one of the main moments in the design of ICE is to determine the efficiency (engine efficiency).To determine the efficiency, respectively, it is necessary to know the value of mechanical losses in the ICE.

There are two types of methods for determining mechanical losses: theoretical and experimental.

Thus, this article is devoted to the experimental determination of mechanical losses.

3.1. Features of methods for determining mechanical losses in internal combustion engines

As is well known, the experimental determination of mechanical losses in an engine with sufficient accuracy is a rather difficult task. The following methods are used to determine mechanical losses:

1) the method of cranking an internal combustion engine (turning the crankshaft of an idle internal combustion engine driven by a dynamometer motor);

2) Morse method (cylinder shutdown);

3) Willian method (by hourly fuel consumption);

4) method of average indicator pressure pi (cylinder indication);

5) double run-out method (based on the crankshaft rotation decay after the ignition is turned off);

6) (p-ω) method (method of instantaneous values of the mechanical loss torque);

7) a method for determining mechanical losses by estimating gi from the change in Ni and Gp according to the experimentally measured load characteristics of the engine [1].

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The first five methods for determining mechanical losses in an engine are widely known and have their own characteristics. The last two methods have been proposed recently.

The (p-ω) method of instantaneous (current) indicator pressures was developed at Wayne State University and allows determining mechanical losses at different speed and load conditions of engine operation. This method involves testing both in steady-state conditions and in transient modes when the engine crankshaft speed changes.

The method of estimating the specific indicator fuel consumption gi by the change in effective power Ne and hourly fuel consumption Gp according to the experimentally measured load characteristics was proposed at NTU "KhPI" by Doctor of Technical Sciences, Professor V. Dyachenko.

There are other, more accurate, methods for determining the friction losses of a cylinder-piston group, for example, with a sleeve moving in the cylinder in the axial direction.

The Institute of Piston Machines of the University of Hannover, together with Daimler-Benz, has developed an experimental single-cylinder engine with a special movable cylinder liner to study the friction losses of the piston and piston rings, as well as to determine wear and oil consumption in various operating modes of the internal combustion engine. The liner is mounted in a cylinder with axial movement and is supported by piezoelectric sensors that measure the friction forces acting on it from the piston group.

Such an engine has a significantly modified and complex design due to the presence of hydrostatic guides to ensure frictionless movement of the cylinder liner and the need to seal the combustion chamber with a non-contact seal between the cylinder head and the liner. It is worth noting that it is still desirable to measure friction losses in an engine with its parts and assemblies intact.

The peculiarities of the above methods for determining mechanical losses (except for the scrolling method) are the presence of a combustion process and the impossibility of separating mechanical losses. In this case, the sum of mechanical losses due to friction, overcoming aerodynamic and hydraulic drags, pumping strokes, and drive of auxiliary units is determined.

It is worth noting that the runout method can be used to determine the components of mechanical losses to some extent if the crankshaft of an idling engine is spun to a specified speed using a dynamometer motor.

Of all the methods for determining mechanical losses discussed above, the most widely used is the test by cranking the crankshaft of an idling engine. The main disadvantage of this method is the discrepancy between the parameters of the working process during cranking and those in the presence of the combustion process. Lower pressure in the cylinders when the crankshaft is rotated leads to a reduction in friction losses in the engine for the following reasons: lower pressure on the piston rings; lower lateral force acting on the piston skirt (in a classic internal combustion engine); reduced load on the bearings of the crankshaft and connecting rod (wings in a rodless engine); lower oil temperature in the lubrication system.

When the engine is not running, the lower temperature of the cylinder walls changes the viscosity of the oil film on the sliding surfaces. This film and friction losses are affected by the absence of gas flow through the piston rings.

Based on the analysis of the known methods for determining mechanical losses and based on the need to assess the impact of each individual element in the total mechanical loss indicator, as well as taking into account the feasibility of implementing a particular method, the most common method of crankshaft cranking of an idling engine was used in this study. In this case, it is expected to introduce certain adjustments of the measurement results to real conditions.

3.2. Problems and methods of experimental determination of mechanical losses in internal combustion engines

The objectives of the experiments included:

1) Determination of mechanical losses on the timing drive, friction of the piston group and friction in the bearings of the CMM (classic engine) and engine without a connecting rod with a rocker arm (rodless engine) under various engine operating conditions, as well as depending on the compression ratio, piston-cylinder clearance, piston design, oil temperature, etc;

2) measurement of mechanical loss components.

The pressure loss due to gas exchange was determined by the calculation method using a mathematical model, while other components and total pressure loss were obtained experimentally.

The experimental determination of the components of mechanical losses is carried out on an internal combustion engine with an adjustable compression ratio - a classical one with a CMM and a connecting rodless one with an unconventional power mechanism in the following sequence:

a) measurement of total friction losses;

b) measurement of friction losses in individual parts and assemblies by the "stripping" method (by sequentially disconnecting them from operation - disconnecting the IGR, then alternately removing the upper and lower compression rings, oil skimmer ring, piston and connecting rod or rocker panel in a connecting rodless engine).

The general methodology for studying mechanical losses involves the development of empirical formulas based on experimental data to obtain an estimate of the contributions of mechanical loss components to engine performance as a whole [2].

It is known that when determining friction losses by cranking an idling engine, the maximum cylinder pressure is significantly lower than when an internal combustion engine is operating and the combustion process is taking place in the cylinder.

To approximate the loading conditions on the power mechanism in an idling engine cranked from an external source to an operating internal combustion engine, the maximum compression pressure pc in the cylinder was set to correspond to the maximum combustion pressure pz at a given engine operation mode by adjusting the compression ratio within the range of 7 to 17.

3.3. Objects of experimental research

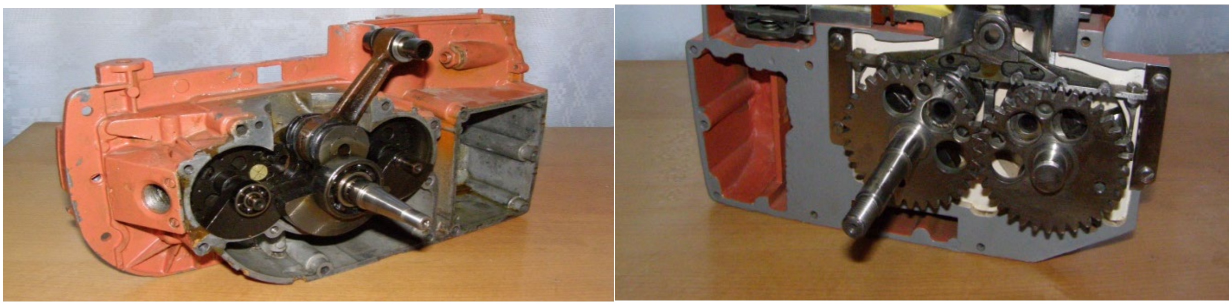
The following objects were chosen as research objects:

- a single-cylinder engine with a crank mechanism;
- a prototype of a single-cylinder connecting rodless engine with an unconventional power mechanism.

Figure 1 shows the experimental crankcases with the power mechanism of the classical (a) and crankless (b) engines, respectively, and figure 2 - their design scheme.

To conduct experimental studies, the engines were equipped with an autonomous electronic ignition system, an autonomous fuel pump, and an independent system for adjusting the ignition advance angle within a wide range [3].

The timing was assumed to be the same for both engines.



a) classical engine, b) crankless engine

Figure 1 - Experimental crankcases with a power mechanism [3]

The engines differ in their power train and crankcase design. In the classical engine with a crankcase, the crankcase, together with the power mechanism, is used from a serial engine of the Taiga-214 gasoline-powered saw (Russia). The crankcase of the crankless engine contains two crankshafts kinematically connected to the piston through a rocker arm and a rod.

The cylinder, together with the cylinder head and the gas distribution mechanism, was used in bench tests for both classical and crankless internal combustion engines by installing it on the corresponding crankcase.

This made it possible to more closely compare the two types of engines, including differences in the wear of the working surfaces of the cylinder-piston group, the gaps between the piston and cylinder, the condition of the surfaces of the piston and cylinder mirror, as well as differences in the technological errors of the IGR elements.

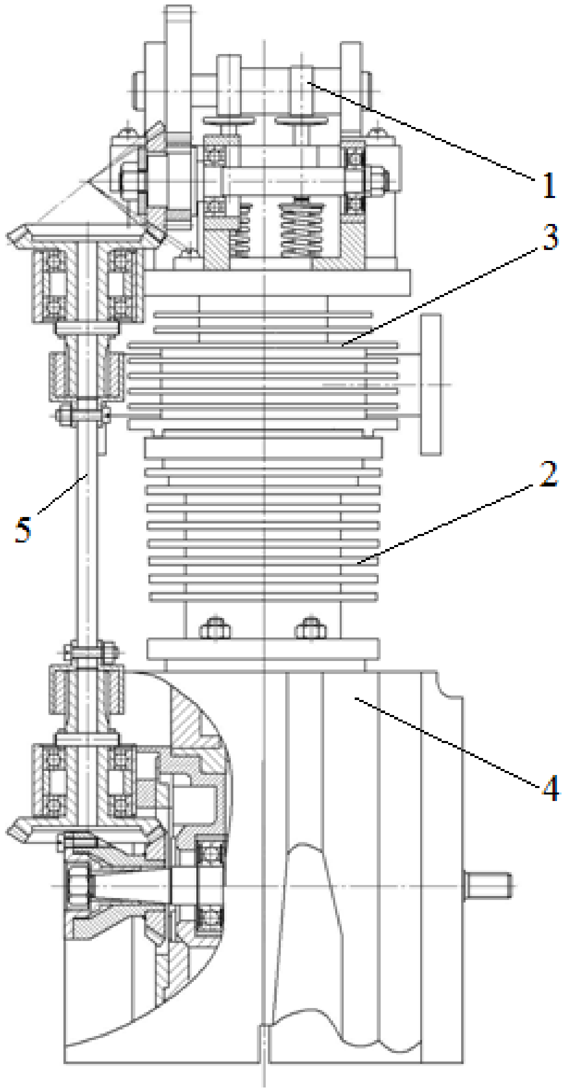
Obviously, due to different changes in the temperature in the cylinder of a running engine and when adjusting the compression ratio in an idling engine, even when $p_c = p_z$, the obtained values of friction losses cannot exactly match the losses of a running engine.

A serial set of piston rings from a Brigs engine (USA) with a diameter of D = 52 mm consists of two compression rings and an oil ring, which is a box slotted ring with converging chamfers, a chrome coating, and a twisted spring extender.

In the experiments on idling engines, the cylinder wall temperature was maintained within a given temperature range using a nichrome coil wound around the cylinder and connected to a 30V DC source.

The coil was electrically isolated from the cylinder by ceramic insulators in the form of beads.

The cylinder temperature was monitored using a chromel-copper thermocouple installed in the middle part of the cylinder.



1 - camshaft; 2 - cylinder; 3 - cylinder head;
4 - crankcase; 5 - IGR drive

Figure 2 - Schematic diagram of the experimental engine

The engines use an independent system for lubricating the crankshaft bearings and regulating the oil temperature from 40 to 120°C.

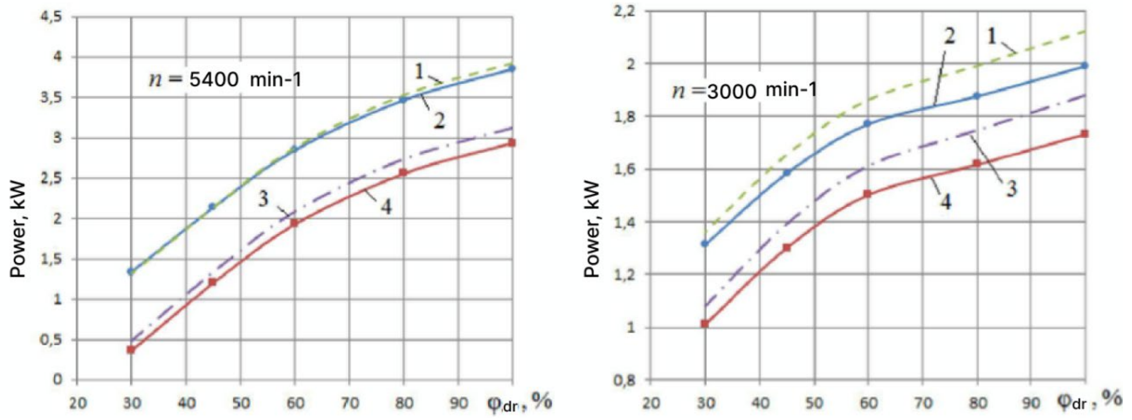
3.4. Research results

Parameters of a connecting rodless and classical internal combustion engine by load characteristics

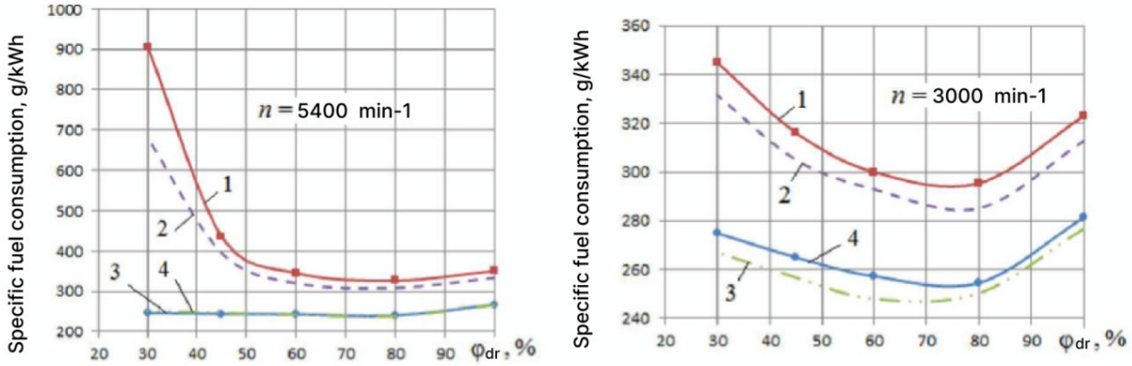
Figure 3 and 4 show some parameters of the crankless and classical internal combustion engines by load characteristics.

It can be seen from the data that at the rated load (φ_{dr} = 100% and n = 5400 min^{-1}), the indicator values such as N_i, g_i, and ηi are almost the same for the crankless and classical ICEs. As the load and rotational speed decrease, the difference between these indicators increases in favor of the crankless engine. This difference is explained by the influence of a number of factors that depend on the piston kinematics.

For example, at a constant rotational speed of n = 3000 min^{-1}, the indicator power at φ_{dr} = 100% in the crankless engine is 0.11 kW (3.7%) higher than in the classical engine, and at φdr = 30%, the N_i values in both engines are almost equal. At n = 2000 min^{-1} and φ_{dr} = 100 %, the difference in N_i is 0.09 kW (3 %), and at φ_{dr} = 30 %, the difference in indicator power is 1 % less.



1, 2 - indicator power, respectively, of an engine with an unconventional power mechanism and a crank mechanism;
3, 4 - effective power, respectively, of an engine with an unconventional power mechanism and a crank mechanism

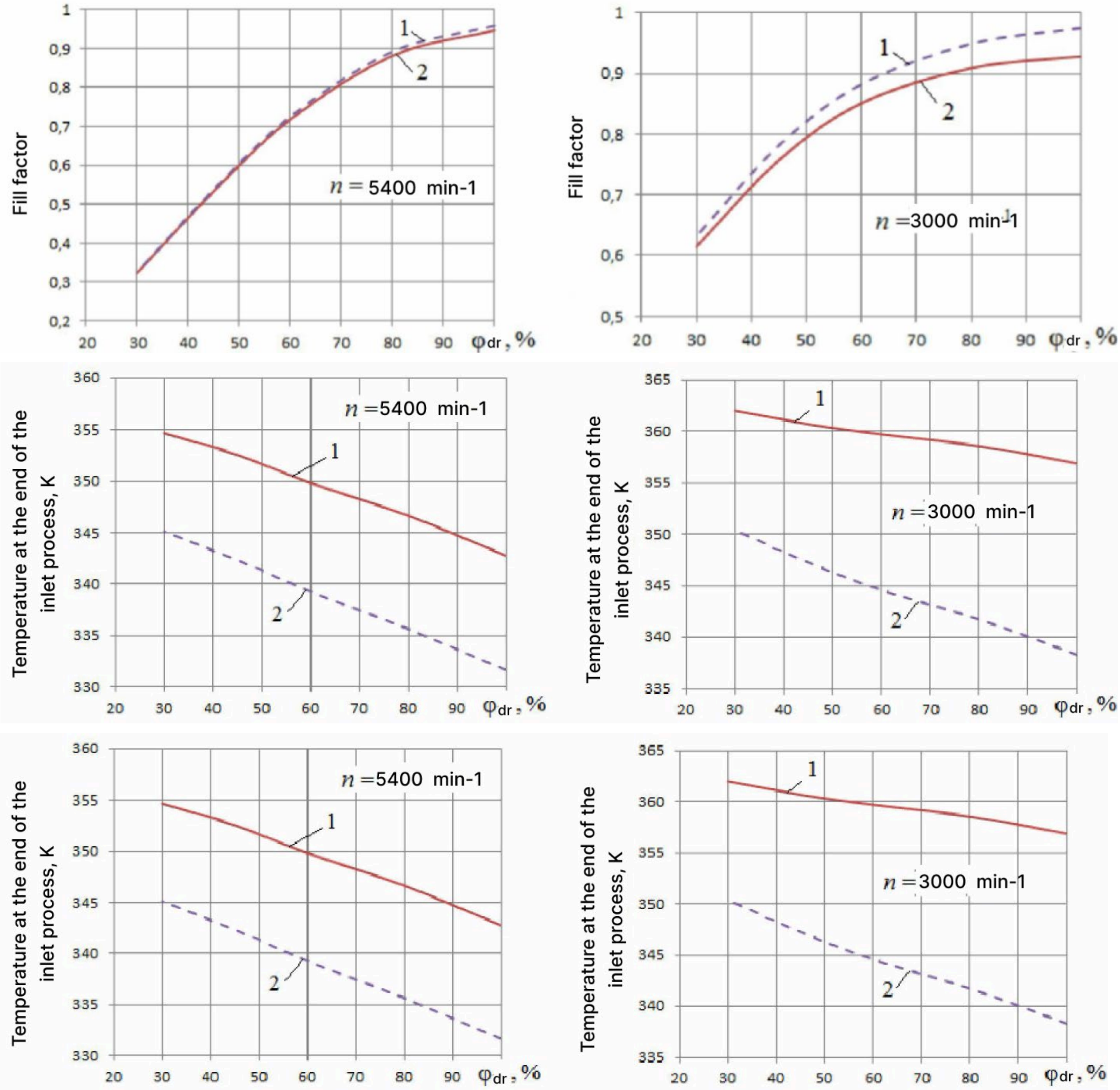


1, 2 - mechanical efficiency, respectively, of an engine with an unconventional power mechanism and a crank mechanism;
3, 4 - indicator efficiency, respectively, of an engine with a crank and a non-traditional power mechanism;

Figure 3 - Load characteristics of experimental engines by throttle opening angle φ_{dr}

The difference between the engines' efficiency (N_e, g_e, η_e) is more significant due to the higher mechanical efficiency ηm in the connecting rodless ICE

The residual gas coefficient γr in the crankless engine is lower in all partial modes compared to the classical engine (Figure 4).



1, 2 - respectively, an engine with a crank mechanism and an unconventional power mechanism

Figure 4 - Intake process parameters

This can be explained by the longer piston residence time in the area of the timing belt in the connecting rodless engine, which contributes to better cleaning of the combustion chamber from combustion products. The difference in γr between a connecting rodless and a classical internal combustion engine increases with increasing load (throttle opening) and decreasing crankshaft speed, and at rated operation, the values of γr of these engines are practically equal (the difference is no more than 1.3 %). With a decrease in the rotational speed n, the difference in γ_r increases significantly and reaches 45 % at the mode: φ_{dr} = 100 % and n = 2000 mi^{n-1}.

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From Figure 4 shows that as the load increases (throttle opening), the difference in the value of the filling factor ηV increases. At the same time, the absolute value of the filling factor in the connecting rodless engine is greater than in the classical engine due to the lower residual gas coefficient. A similar pattern of change in ηV is observed with a decrease in rotational speed.

For the load mode φ_{dr} = 100%, the value of η_V of the crankless engine varies from 0.936 at n = 5400 min^{-1} to 0.782 at n = 3000 min^{-1}; in a classical ICE, η_V varies, respectively, from 0.923 to 0.731. Such a difference in η_V with a decrease in rotational speed (1.4 and 7 % at n = 5400 min^{-1} and 3000 min^{-1}) is due to the fact that in this case the effect of piston speed on ηV in a crankless engine is greater than in a classical ICE.

A crankless engine with an unconventional power mechanism is interesting from the point of view of using supercharging or increasing the compression ratio because it has a lower temperature at the end of the intake Tah at all load and speed conditions. At the nominal mode n = 5400 min^{-1} and φ_{dr} = 100 %, the temperature difference is ΔT = 11 K; at n = 2000 min^{-1} and φ_{dr} = 100 %, ΔT = 8.95 K, and at φ_{dr} = 30 %, ΔT = 13 K.

The calculations show that due to the lower temperature at the end of the intake, ceteris paribus, the compression ratio can be increased by 1...2 units relative to the basic compression ratio of a classical internal combustion engine. This allows to increase the overall expansion ratio and, accordingly, to raise the indicator efficiency.

The value of the ignition advance angle in a crankless engine compared to a classical engine is less on average by 10 °p.e.m. at n = 5400 min^{-1}, and at n = 2000 min^{-1} it is less by 2.5 °p.e.m. This indicates that in a crankless engine at the moment of sparking the piston is closer to the timing, and, accordingly, the degree of expansion is greater and the efficiency of using the supplied heat is greater.

Conclusions

1. The empirical dependences obtained on the basis of experimental data reflect the regularities of the flow of mechanical loss components in internal combustion engines and are confirmed by statistical data of modern automobile engines.

2. In a connecting rodless engine, the components of mechanical losses, especially gas exchange and friction in the piston group and crank mechanism, have not yet been studied. Therefore, in order to develop constructive measures and create a theoretical basis for reducing the mechanical losses of a crankless engine, it is necessary to conduct

extensive research on the components of mechanical losses, including losses on the drive of auxiliary mechanisms.

3. A crankless engine, in comparison with a classical engine, provides 10...20 % better fuel efficiency and higher specific effective power (by 7...12 %) due to lower mechanical losses and better thermodynamic cycle.