

Modeling of thermodynamic processes in internal combustion engine cylinder during cranking in compression measurement tests

ARTICLE INFO

The thermodynamics of in-cylinder processes of the internal combustion engine in cranking mode when measuring compression were examined. Analysis of various theoretical models and comparison of known data showed that the problems of constructing mathematical models of the engine operating cycle remain unresolved. The vast majority of practical data and recommendations for measuring cylinder compression are based on empirical knowledge, numerous experiments, and tests. In accordance with this, there is a need for computational models of the compression measurement process and their theoretical justification. To solve the identified problems, a mathematical model of the thermodynamic processes in the cylinder was developed. Unlike existing ones, the model describes the processes in the cylinder step by step, taking into account the real nature of the intake-exhaust processes, air leakage through the interfaces of parts, and heat exchange with the walls. Using modeling, the main patterns of changes in compression depending on the modes, the nature of damage to the associated parts of the valve mechanism and cylinder-piston group, the amount of leakage, and engine temperature were found. Based on the results of the study, the properties of the model which make its use effective in diagnosing and monitoring the technical condition of engines in operation were concluded.

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1. Introduction

It is known that measuring the maximum pressure – compression in the cylinders of an internal combustion engine is one of the most common methods for diagnosing technical conditions and determining the causes of various faults in operation [9]. This check is usually provided at the preliminary stage of research since it does not require dismantling numerous engine parts and units. Thus, compression measurement is a non-destructive method for monitoring the technical condition in which the object under study fully retains its functions. This property of the method is essential for some types of research, for example, during automotive technical expertise.

Another undoubted advantage of measuring compression is the simplicity and low cost of the measuring device itself, a compression meter, as well as the simplicity of the algorithm for its use (Fig. 1). As a result, the method has become extremely widespread in practice as one of the most universal diagnostic methods, and the maximum pressure in the cylinder has been included in almost all service and repair manuals for most automotive engine brands and models [6].

At the same time, the vast majority of practical data and recommendations for the use of the method under consideration are based on empirical knowledge, numerous experiments, and tests [21]. The consumer is encouraged to trust the recommendations of specialists, who usually associate the amount of compression drop from a certain “normal” level with a specific type of damage and/or fault. However, the consumer does not have any models or methods for calculation or checking any parameters that he measures for its adaptation to his engine. Therefore, the consumer cannot check this or that figure, recommendation, or assumption about the connection of a measurement result with a fault

and also adjust the received data in accordance with his own conditions.

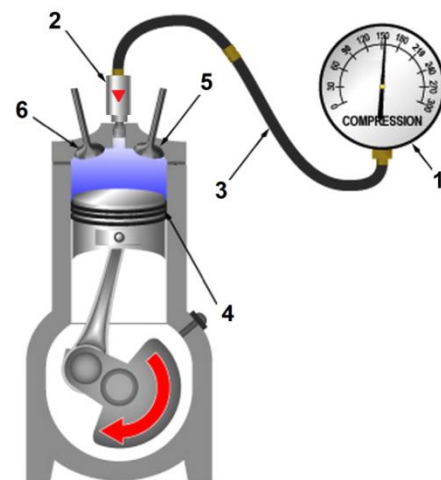


Fig. 1. Scheme for measuring compression in the engine cylinder and the main points of air leakage from the cylinder: 1 – pressure gauge, 2 – check valve, 3 – connecting tube, 4 – piston rings, 5 – intake valve, 6 – exhaust valve

At the same time, such a check may be important for practice due to the design features of the engine under study, which may influence changes in the compression value under various operational damages. However, quantitative assessments and, especially, verification of certain data related to the magnitude of compression are extremely expensive since they require experimental studies of large volume and complexity. Hence, the need arises for computational models of the compression measurement process and their theoretical justification.

2. Sources review and problem statement

The method of measuring compression itself is extremely simple and is accessible not only to a qualified or initial mechanic but also to an ordinary driver. Therefore, a significant number of sources provide numerous practical recommendations on the algorithm and interpretation of the results obtained. Thus, in [11], data are presented that make it possible to determine many malfunctions of the valve mechanism and cylinder-piston group by changes in compression, including burnout of valves, wear and sticking of piston rings, etc.

However, available data indicate that compression measurements should be treated with caution due to the inaccuracy of the method itself and the large variation of measuring results. As a result, erroneous conclusions may be obtained not only about the cause but generally about the presence of a fault as such. This is due to the strong influence of rotation speed on compression noted in many sources [26], which may be caused by differences in test conditions. Such differences are due to changes in ambient temperature, engine temperature, battery and starter condition, oil viscosity, etc.

Moreover, only one compression value measured by a compression meter (and other similar devices) generally does not provide a complete picture of the current engine's technical condition. As a result, other methods have become widespread in diagnostic practice.

One such method is pressure oscillography in cylinders. Indeed, with the help of a pressure transducer [23], an oscilloscope makes it possible to identify deviations of the instantaneous pressure diagram in a cylinder from the normal profile and thereby localize and find the cause [27]. However, the oscillography method has the same disadvantages and features as conventional compression measurements. To apply the method, it is necessary to have detailed information on how the shape of the diagram is affected by various damage and faults. And this knowledge is currently based only on experimental data [11, 21].

An alternative to measuring compression can also be methods such as measuring the power balance of cylinders on the one hand and building an indicator diagram (including its modeling) on the other. These methods require the presence of expensive measuring equipment (for example, motor testers to determine the power balance) or even special preparation of the engine (for example, for installation of a pressure sensor). However, the power balance is not informative enough for fault determination, even compared to compression measurements [32]. The study of the indicator diagram is further complicated by the need to test the engine under load, which for an automobile engine is only possible on extremely expensive special test benches [19]. In addition, large amplitudes of pressure and temperature, as well as the relatively high frequency of the process, characteristic of measurements during the operation of automobile engines, create problems for both measurement and analysis of the resulting diagrams, including the determination of the influence of various faults on them [36]. Similar problems arise when modeling these processes. Moreover, some serious engine damage and faults prevent the ignition of the fuel and operation of the cylinder alto-

gether, which makes it difficult to analyze the indicator diagram of such a cylinder.

These problems impose restrictions on the widespread practical use of power balance and indicator diagram analysis for diagnosing automobile engines in operation. In this regard, the experimental study, modeling, and analysis of a compression diagram in cold cranking mode seem to be a much simpler, technically, and financially accessible diagnostic method. Therefore, there is reason to believe that this method is more promising for further research and practical use of the results obtained.

Another problem that requires a detailed analysis of the diagram of pressure changes in the cylinder during cold cranking of the engine (without fuel combustion) is associated with determining the characteristics of a cold start [20]. Startup simulation allows you to evaluate the starting properties of the engine, especially at low temperatures. This task is close to the problem of measuring compression. However, it is not aimed at diagnosing the technical condition of a particular engine in operation but most often at design work to improve the starting characteristics of the engine [4], determining the starting fuel supply [25], assessing harmful emissions [30], etc.

It was in the study of engine starting characteristics and not in diagnostics and monitoring that various theoretical models of cold cranking became widespread [29]. The next step in this direction can be considered probabilistic-statistical methods [24, 35], in which data arrays are considered, and the probabilistic values of parameters are analyzed. Such methods can be effective for diagnostics, including when used in software diagnostic systems [33]. However, the models that usually form the basis of such methods are quite far from the process model of the engine under study. They can be conditionally called integral since they do not detail the instantaneous processes occurring in the engine but operate only with external signs and parameters [34].

At the same time, it should be noted that the need for detailed calculation of starting processes and characteristics required the development of theoretical methods and models that allow step-by-step modeling of processes based on the angle of rotation of the crankshaft. The simplest solutions are provided by analytical methods built on the basis of dependencies for polytropic compression-expansion processes [28]. However, it is difficult to obtain quantitative data to identify various damages and faults using such methods.

Mathematical modeling methods have received the greatest development in the programs for calculating the operating cycle of engines. A number of programs have acquired well-known status and become standard – these are AVL-Boost [1], Ricardo-Wave [5], GT-Power [3] and Lotus Engine Simulation [18]. These programs are built on the basis of a 0-dimensional thermodynamic representation of a cylinder as a control volume with instantaneous parameters of the working fluid uniformly distributed throughout the volume [7].

This approach makes it possible to calculate instantaneous cycle parameters based on the angle of rotation of the crankshaft at any operating mode [22], that is, exactly what

is required when studying compression in the cylinder and/or cold start. However, when trying to simulate, a problem is revealed: due to limitations, not all standard programs have the function of simulating the cold cranking mode without fuel combustion in the cylinder, taking into account various damages and other necessary functions [28]. As a result, the development of a user model of the engine cold cranking mode, designed to study the patterns of measuring compression in the cylinder, has virtually no alternative.

In accordance with this, the purpose of the work is to study the working process in the cylinder of an internal combustion engine during cold cranking and obtain quantitative characteristics of compression under various operational damages.

To achieve the goal, it is necessary to solve the following tasks:

- 1) develop a mathematical model of the process in the cylinder during engine cold cranking
- 2) carry out modeling in different modes and under different conditions of the cylinder-piston group and valve mechanism, compare the results obtained with experimental data
- 3) validation and setting-up the model
- 4) analyze the results obtained, determine general patterns connecting the amount of compression with various types of damage to engine parts.

3. Mathematical model of the process in the cylinder during engine cold cranking

To derive the design equations, consider the diagram of the cylinder (Fig. 2).

From the 1st law of thermodynamics [7], written for the cylinder, it follows:

$$dU = dQ - dL + idM \quad (1)$$

where dU is the change in the internal energy of the air in the cylinder; dQ is amount of heat supplied (+) or removed (-); L is work of air; i is enthalpy of air; dM is change in air mass in the cylinder due to inflow into the cylinder (+) or outflow (leakage) from the cylinder (-).

Let's transform equation (1):

$$d(MC_V T) = dQ - pdV + idM \quad (2)$$

The value $idM = \sum i_i dM_i$ takes into account the mixing of air flows with different temperatures. In the cold cranking mode at a low rotation speed, when air flows out of the cylinder, it has a temperature T , close to the ambient temperature T_0 . At the same time, air flows into the cylinder when there is a vacuum in it during the intake stroke, when the temperature of the air in the cylinder is also close to the temperature of the environment and the incoming air. In accordance with this, we can neglect the difference in flow temperatures and accept $idM = i \sum dM_i$. From where, taking into account the ideal gas equation of state

$$p = \frac{M}{V} RT \quad (3)$$

from equation (2) we get:

$$MC_V dT + C_V T dM = dQ - \frac{MRT}{V} dV + C_p T dM \quad (4)$$

where p is pressure in the cylinder; V is the volume of gas in the cylinder; R is gas constant; C_V , C_p are heat capacities at constant pressure and volume.

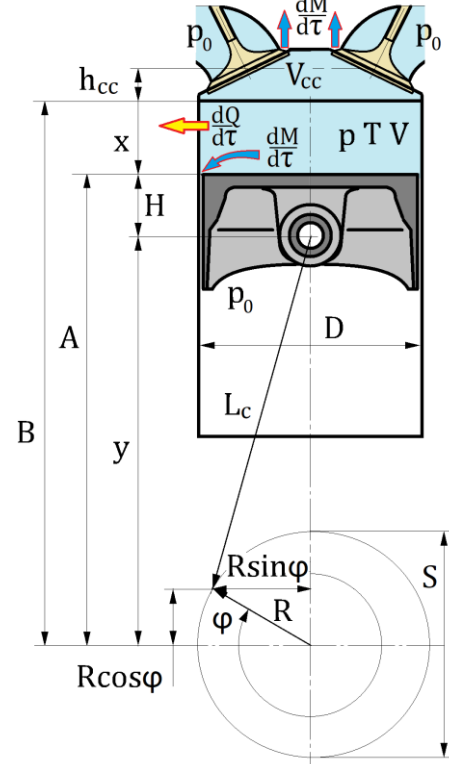


Fig. 2. Calculation diagram of the process in the cylinder during cold cranking

Further transformations of equation (4) give:

$$C_V \frac{dT}{T} = -R \frac{dV}{V} + (C_p - C_V) \frac{dM}{M} + \frac{dQ}{MT} \quad (5)$$

The heat capacities are related by the relation $C_p = \gamma C_V = \gamma R / (\gamma - 1)$, where γ is the air heat capacity ratio (adiabatic index). In addition, the current volume of air in the cylinder is equal to

$$\begin{aligned} V &= V_x + V_{cc} = V_x + \frac{V_h}{\varepsilon - 1} = F_p x + \frac{F_p S}{\varepsilon - 1} \\ &= F_p (x + h_{cc}) \end{aligned} \quad (6)$$

where V_x is current cylinder volume; V_{cc} is volume of combustion chamber; V_h is the working volume of the cylinder; ε is geometric compression ratio; F_p is piston area; x is piston coordinate from top dead center [2]; $h_{cc} = V_{cc} / F_p = S / (\varepsilon - 1)$ is reduced height of the combustion chamber, S is piston stroke.

Obviously, the change in the volume of the cylinder $dV = F_p dx$, according to which equation (5) can be rewritten as:

$$\frac{dT}{T} = \frac{(\gamma - 1)}{R} \left[-R \frac{dV}{V} + \left(\frac{\gamma}{\gamma - 1} - \frac{1}{\gamma - 1} \right) R \frac{dM}{M} + \frac{dQ}{MT} \right] \quad (7)$$

This implies a differential equation for the air temperature in the cylinder:

$$\frac{dT}{T} = \frac{(\gamma - 1)}{x + h_{cc}} \left[-dx + \frac{RT}{pF_p} \left(dM + \frac{dQ}{RT} \right) \right] \quad (8)$$

On the other hand, differentiating the equation of gas state (3), we obtain:

$$dp = \frac{dM}{V} RT + \frac{dT}{V} MR - \frac{MRT}{V^2} dV \quad (9)$$

or

$$\frac{dp}{p} = \frac{dM}{M} + \frac{dT}{T} - \frac{dV}{V} \quad (10)$$

Now, substituting equation (7) into equation (10), we obtain a differential equation for the air pressure in the cylinder in the form:

$$\frac{dp}{p} = \frac{\gamma}{x + h_{cc}} \left[-dx + \frac{RT}{pF_p} \left(dM + \frac{\gamma - 1}{\gamma} \frac{dQ}{RT} \right) \right] \quad (11)$$

As it is known from engine theory [10], the position of the piston in the cylinder x is determined by the angular position of the crankshaft ϕ , measured from top dead center when the valves overlap. In accordance with Fig. 2, the x coordinate can be expressed by the formula:

$$x = A - B = (R + L_c + H) - (y + H) = R + L_c - R \cos \phi - \sqrt{L_c^2 - R^2 \sin^2 \phi} \quad (12)$$

where R , L_c are the radius of the crank and the length of the connecting rod; H is the compression height of the piston.

Taking into account the fact that the piston stroke S is equal to two radii of the crank, the position of the piston from top dead center will be

$$x = \frac{S}{2} \left(1 + r - \cos \phi - \sqrt{r^2 - \sin^2 \phi} \right) \quad (13)$$

where $r = 2L_c/S$ is relative length of the connecting rod.

It is known [27] that the value of the desired compression in the cylinder is largely determined by leakage from the cylinder and heat losses, which depend mainly on the process time. In accordance with this, it is convenient to seek a solution to the problem from the time of the process associated with the angular position of the crankshaft by the formula:

$$d\phi = \omega dt = \frac{\pi n}{30} dt \quad (14)$$

where n is the crankshaft rotation speed, rpm (to the 1st approximation, the rotation speed during cranking is assumed to be constant and independent of the process time and the angular position of the crankshaft).

Then, if all equations (8) and (11) are divided by the time increment $d\tau$, we can obtain a mathematical model of the process under study [8, 13] as the system of 1st order differential equations resolved with respect to the derivative, in the form:

$$\begin{cases} \frac{dT}{d\tau} = \frac{(\gamma - 1)}{x + h_{cc}} T \left[-\frac{dx}{d\tau} + \frac{RT}{pF_p} \left(\frac{dM}{d\tau} + \frac{1}{RT} \frac{dQ}{d\tau} \right) \right] \\ \frac{dp}{d\tau} = \frac{\gamma}{x + h_{cc}} p \left[-\frac{dx}{d\tau} + \frac{RT}{pF_p} \left(\frac{dM}{d\tau} + \frac{\gamma - 1}{\gamma RT} \frac{dQ}{d\tau} \right) \right] \end{cases} \quad (15)$$

To solve system (15), in addition to the initial conditions, it is necessary to determine some quantities and variables included in the equations.

Thus, the piston speed can be obtained analytically by differentiating equation (13), which determines the current position of the piston

$$\begin{aligned} \frac{dx}{d\tau} &= \frac{\pi S n}{60} \sin \phi \left(1 + \frac{\cos \phi}{\sqrt{r^2 - \sin^2 \phi}} \right) = \\ &= v_m \frac{\pi}{2} \sin \phi \left(1 + \frac{\cos \phi}{\sqrt{r^2 - \sin^2 \phi}} \right) \end{aligned} \quad (16)$$

where $C_m = Sn/30$ is average piston speed.

The change in air mass in the cylinder is determined by the airflow through the valves and leak points. When the pressure in the cylinder is higher than atmospheric, the airflow from the cylinder through a certain hole (gap, slot) is determined by the well-known formula [12]:

$$\frac{dM_i}{d\tau} = -\mu \frac{\rho f_i}{\sqrt{RT}} \left(\frac{p_0}{p} \right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p_0}{p} \right)^{\frac{\gamma - 1}{\gamma}} \right]} \quad (17)$$

where μ is flow coefficient; f_i is the cross-sectional area through which flow or leakage occurs; p_0 is ambient pressure (it is assumed that at a low speed corresponding to the compression measurement, the pressure in all channels adjacent to the cylinder and in the crankcase is equal to the ambient pressure).

When the pressure drop changes to reverse (vacuum in the cylinder) and air flows into the cylinder, formula (17) takes the form:

$$\frac{dM_i}{d\tau} = \mu \frac{\rho_0 f_i}{\sqrt{RT_0}} \left(\frac{p}{p_0} \right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{p}{p_0} \right)^{\frac{\gamma - 1}{\gamma}} \right]} \quad (18)$$

In general, air can flow into (and out of the cylinder) several flows: through the intake valve $dM_{in}/d\tau$, exhaust valve $dM_{ex}/d\tau$, leaks (gaps, damage) in the interface of intake valves $dM_{lin}/d\tau$ and exhaust valves with seats $dM_{lex}/d\tau$ as well as piston rings with cylinder and ton $dM_{lc}/d\tau$. Rare cases of head gasket failure, cracks in the cylinder head and block can be taken into account by analogy. In accordance with this, the total air flow into or out of the cylinder (change in air mass) is defined as

$$\frac{dM}{d\tau} = \frac{dM_{in}}{d\tau} + \frac{dM_{ex}}{d\tau} + \frac{dM_{lin}}{d\tau} + \frac{dM_{lex}}{d\tau} + \frac{dM_{lc}}{d\tau} \quad (19)$$

To determine the airflow through the valve, consider the connection between the valve and the seat (Fig. 3a).

Let us assume, to the 1st approximation and for simplifying the calculating formulas, that the valve, when lifted by an amount h , opens a section f_v , equal to the area of the lateral surface of the truncated cone, which is perpendicular

to the working chamfer of the valve, of the seat and to the direction of airflow. Then, from the obvious geometric relationships:

$$f_v = \frac{\pi}{2} (d + d_1) h \cos \alpha \quad (20)$$

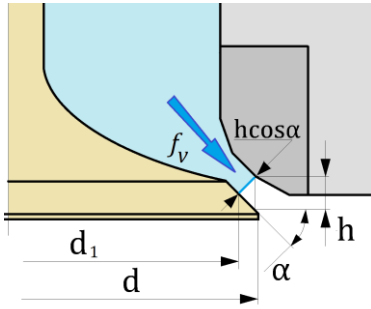
where α is valve chamfer angle; d is valve head diameter; $d_1 = d - 2h \sin \alpha \cos \alpha$ is the diameter of the small base of the cone in Fig. 3a.

After the transformations, we obtain the area opened by the valve (inlet or outlet) depending on the height of its lift:

$$f_v = \pi d h \left(1 - \frac{h}{d} \sin \alpha \cos \alpha \right) \cos \alpha \quad (21)$$

Instead of the approximate formula (21), you can also use more accurate relationships for the flow area that is opened by the valve [14]. However, the expression (21) for the problem under consideration is simpler. The reason is that at low rotation speeds during cold cranking, the pressure drops across the valves are small (this can be seen in the diagrams of compression measuring [26, 34]). Therefore, the airflow should be influenced mainly by the valve timing, i.e., the moments of opening and closing of valves, but not by the valve lift profiles and/or the flow area they open.

a)



b)

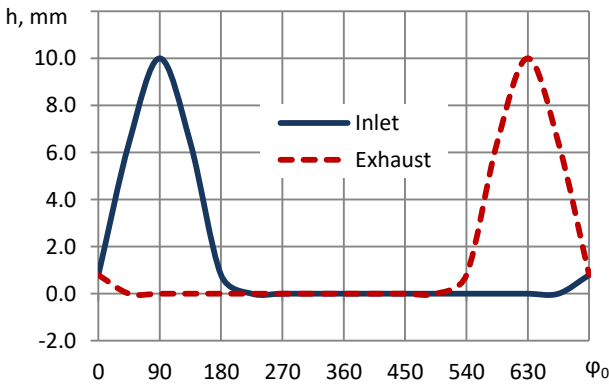


Fig. 3. Calculation diagram for the cross-sectional area opened by the valve (a) and the program for controlling the valve timing (b), adopted when simulating cold cranking

According to [14], at a low rotation speed of 200–300 rpm, usually corresponding to compression measurements, the flow coefficient μ included in formulas (17) and (18) can be taken equal to 1.0 with sufficient accuracy. Then, to determine the air flow using these formulas, it remains to

find the valve lift height. To do this, let us set the valve lift profile within the crankshaft rotation angles that correspond to the valve timing. To a first approximation, it is possible to assume for both valves a lift profile corresponding to the sinusoidal law [22]. The simplest form of this is the relationship (Fig. 2b):

$$h = \begin{cases} 0.5h_0(1 + \sin \beta) - \delta, & h > 0, \varphi_{v0} < \varphi < \varphi_{vc} \\ 0, & \varphi < \varphi_{v0}, \varphi > \varphi_{vc} \end{cases} \quad (22)$$

where h_0 is the maximal lift; δ is a clearance in the valve drive; φ_{v0} , φ_{vc} are valve opening and closing angles, $\Delta\varphi_v = \varphi_{vc} - \varphi_{v0}$ is valve opening duration (Fig. 3b); β is auxiliary angle determined by the crankshaft rotation angle φ :

$$\beta = 2\pi \frac{\varphi - \varphi_{v0}}{\varphi_{vc} - \varphi_{v0}} - \frac{\pi}{2} = 2\pi \frac{\varphi - \varphi_{v0}}{\Delta\varphi_v} - \frac{\pi}{2} \quad (23)$$

To calculate the airflow caused by leaks from the cylinder, the area f_i of the elements through which the leak occurs must be determined. Such elements can be a gap between the valve and the seat of size λ_v and cross-sectional area f_c , a gap between the piston ring and the cylinder of size λ_p and area f_p , a sector on the valve head from a chip or burnout of size c , as well as a burnout hole on the piston with a diameter of Δ :

$$f_v = \pi D \lambda_v, f_p = \pi D \lambda_p, f_c = \frac{\pi}{8} c^2, f_\Delta = \frac{\pi}{4} \Delta^2 \quad (24)$$

The last parameter that needs to be determined in the system of equations (15) is the heat flow from the air into the walls of the cylinder and combustion chamber. The assumption that the temperature of the engine walls T_w is constant over time and uniform across sections approximately corresponds to the condition for measuring the compression of a “warm” engine. Then, the heat flow from the air into the wall (and from the wall to the air) can be determined by the formula:

$$\frac{dQ}{d\tau} = (F_p + F_x + F_{cc1} + F_{cc2}) \alpha_w (T - T_w) \quad (25)$$

where α_w is heat transfer coefficient between air and walls; F_x is area of the cylinder side surface opened by the piston; F_{cc1} is area of the side surface of the combustion chamber; F_{cc2} is area of the end surface of the combustion chamber.

If we approximately assume that the area of the end surface of the combustion chamber is equal to the area of the piston F_p , and the area of the side surface of the combustion chamber is proportional to its reduced height h_{cc} , then the formula (25) can be written as:

$$\frac{dQ}{d\tau} = -\pi D \alpha_w \left(\frac{D}{2} + x + h_{cc} \right) (T - T_w) \quad (26)$$

The main difficulty in taking into account heat losses in the process under study is the correct determination of the heat transfer coefficient. Well-known formulas for calculating the heat transfer coefficient for engines are usually obtained for other conditions. In the general case, they are not suitable for the cold scrolling mode, which is characterized by tens of times lower rotation speeds, the absence of combustion, radiant heat transfer, etc. However, in [28] for the cold scrolling mode, the choice of the Woschni formula

was justified, which gives the dependence on the heat transfer coefficient in the form:

$$\alpha_w = 130 \frac{(p \cdot 10^{-5})^{0.8} \omega_m^{0.8}}{T^{0.53} D^{0.2}} \quad (27)$$

where $\omega_m = 2.28C_m$ for gas exchange and compression.

To simplify the debugging of the calculation algorithm, heat transfer in the cylinder was not taken into account. According to the equations of system (15), the heat flow from the air to the walls (and vice versa) affects the temperature and pressure of the air in the cylinder, similar to the effect of airflow during leaks. For example, when pressure increases, leaks from the cylinder are accompanied by heat loss from heated air into the walls. And vice versa, when air flows into the cylinder, heat is supplied to it from the heated walls. This feature made it possible, to a first approximation, not to determine heat losses but to take into account their influence on the process by increasing the area of air leakage from the cylinder. In this case, system (15) will take the following simplest form [15]:

$$\begin{cases} \frac{dT}{d\tau} = (\gamma - 1) \psi T \\ \frac{dp}{d\tau} = \gamma \psi p \\ \psi = \frac{1}{x + h_{cc}} \left(-\frac{dx}{d\tau} + \frac{RT}{pF_p} \frac{dM}{d\tau} \right) \end{cases} \quad (28)$$

System (28) is solved numerically with initial conditions: at $\tau = 0$, $\varphi = 0$, $x = 0$, $T = T_0$, $p = p_0$. For the solution, the 2nd order Runge-Kutta method (modified Euler method) was used [17]. However, when carrying out test calculations, it turned out that the solution had signs of instability (self-oscillations of pressure) at small pressure drops during the intake-exhaust process, especially near valve overlap. Self-oscillations manifested themselves most strongly with significant leaks from the cylinder in the case of modeling serious damage (for example, burnt valves).

It was not possible to suppress the oscillations by reducing the step and increasing the order of the method above the 2nd, and with large leaks from the cylinder, the simple Euler method of the 1st order gave even less instability than the Runge-Kutta method of the 2nd and higher orders. Ultimately, to improve the stability of the solution in the intake-exhaust region, artificial smoothing was applied for pressure using an additional term with a smoothing coefficient ϑ :

$$p_i = p_{i-1} + \frac{\Delta\tau}{2} \left(\frac{dp_i}{d\tau} + \frac{dp_{i-1}}{d\tau} \right) + \vartheta(p_i + p_{i-2} - 2p_{i-1}) \quad (29)$$

As follows from (29), smoothing does not affect the calculated pressure value if the pressure is constant or changes smoothly (straightforward), but in the case of a sharp change in pressure, smoothing affects it in the opposite direction.

Artificial smoothing was introduced only when the intake and/or exhaust valves were open. At a low value of the coefficient ϑ , smoothing did not have any effect on the calculation results (the difference in the maximum compression value did not exceed 0.1%). As a result of test

calculations, the smoothing coefficient was accepted to 0.2 from the condition of maximum suppression of self-oscillations of pressure in the cylinder during intake and exhaust. Self-oscillations were also noted for temperature, but its smoothing was not carried out due to small amplitude.

Debugging of the model and algorithm was carried out in Excel environment. The angle step was set from the condition of 1000 points per 1 cycle (2 revolutions of the crankshaft), which at a rotation speed of 200 min^{-1} corresponded to a time step of 0.0006 s. The choice of step was due to the fact that its decrease did not lead to a change in the accuracy of the calculation and improvement of stability, while its increase caused an increase in instability.

4. Validation and setting up the model

To validate the model, we used experimental data obtained by measuring compression on different engines of the same type, having different technical conditions [31]. The main patterns of cold cranking in the process of measuring compression were studied using the example of an engine with dimensions $D \times S = 76 \times 71 \text{ mm}$, compression ratio $\varepsilon = 9.9$, with a 2-valve cylinder head, with valves with a diameter of 37 and 32 mm, a working chamfer angle $\alpha = 45^\circ$, valve lift of $h_0 = 10 \text{ mm}$ and valve timing: the intake valve opening $\varphi_{OV} = 696^\circ$ (24° to the top dead center) with a duration of 264° and the exhaust valve opening $\varphi_{OV} = 486^\circ$ (54° to the bottom dead center) with a duration of 258° .

During the experiments, instantaneous pressure in the cylinder was measured over time using a strain gauge of the MD-10B model, which was screwed into the hole for the spark plug (Fig. 4).

a)



b)

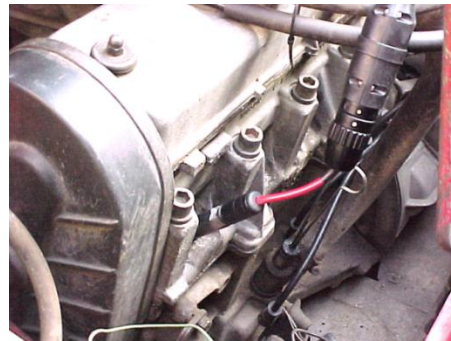


Fig. 4. MD-10B pressure sensor (a) and its connection to the spark plug hole on the engine (b)

The sensor signal, after amplification and conversion in an analog-to-digital converter, was recorded in the comput-

er memory. To process the recording, a user program was used [31], which presented the measurement results in the form of tables (Fig. 5) and a time sweep (oscillogram) of pressure.

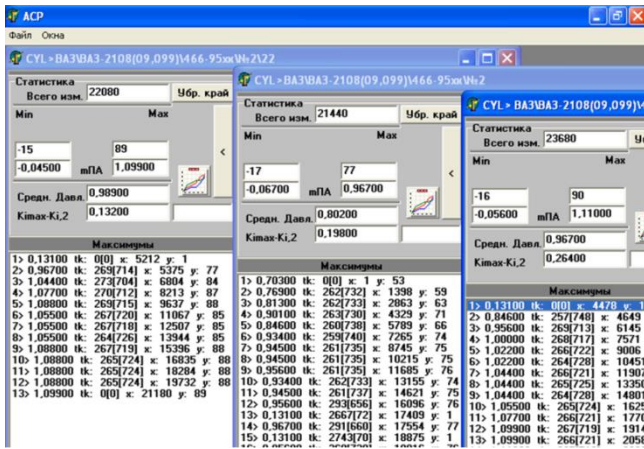


Fig. 5. Program interface for recording and processing compression measurement results

Under the assumption of a constant rotation speed when measuring compression, the pressure diagrams were saved both over time (Fig. 6a) and over the crankshaft rotation angle (Fig. 6b) [31].

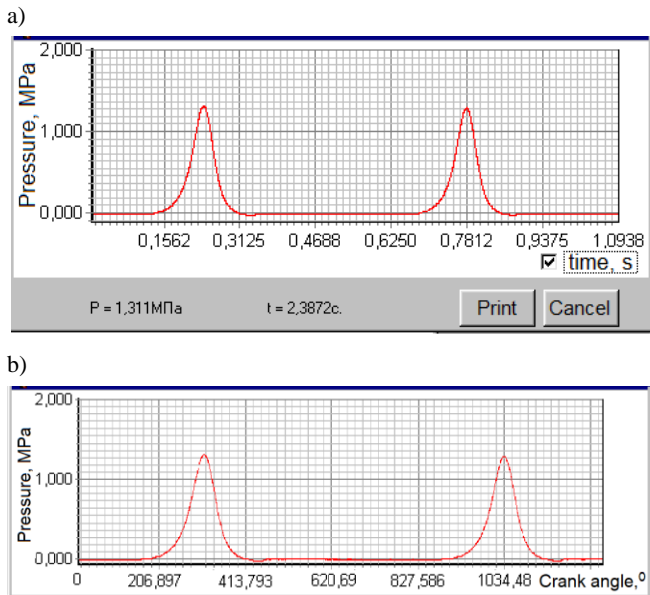


Fig. 6. Diagrams of cylinder pressure over time (a) and by crankshaft rotation angle (b), obtained using a data processing program

This data was used to set-up the model. For this purpose, parametric modeling was performed, in which the amount of leakage through the piston rings was changed by changing the conditional gap in the interface between the ring and the cylinder. From the condition that the maximum compression value corresponds to the experimental data shown in Fig. 6, the value of the conventional gap between the piston ring and the cylinder was determined to be $\lambda_v = 0.028$ mm.

In Figure 7a, the results of calculating the cold cranking cycle of an engine with a leak specified using the model settings are presented. After this, the experimental and calculated diagrams were superimposed (Fig. 7b).

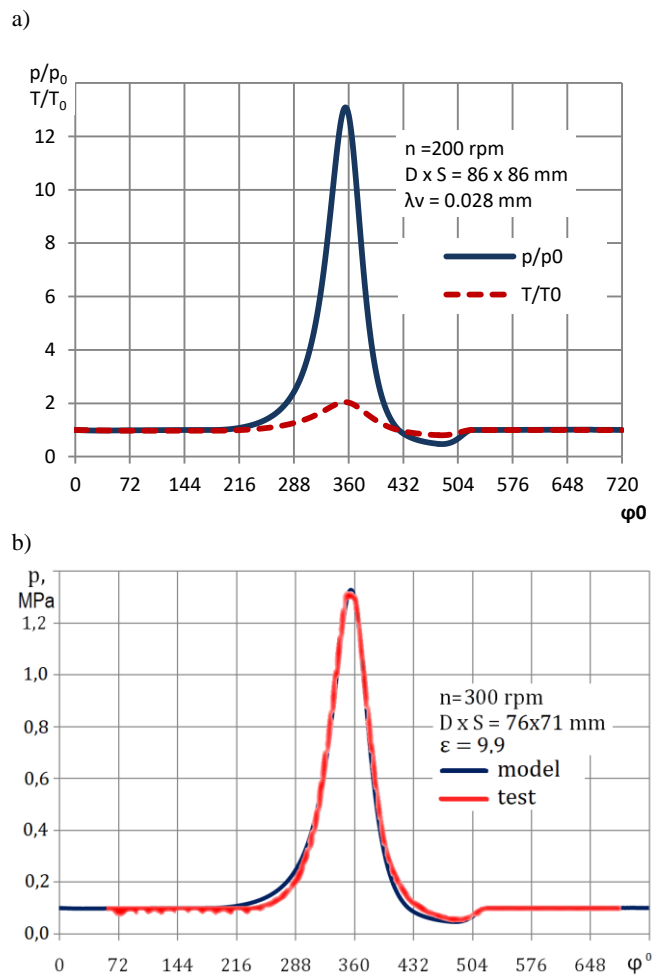


Fig. 7. Results of validation and setting-up the model: modeling pressure and temperature in the cylinder for a given leakage through the piston rings (a) and comparison of calculated and experimental diagrams by superimposing them on each other with the same coordinate axes (b)

The resulting curves show, on the whole, a satisfactory agreement between the calculations and the experiment. It is noteworthy that the obtained curve corresponds to the experimental diagram both in form and quantitatively. However, the clearest results were obtained when various types of damage were introduced into the model. For this purpose, several of the most characteristic ones were selected from the existing database of compression tests [31] (Fig. 8).

A comparison of modeling results and experimental data was carried out for several types of damage, including leakage of the valve-seat interface (Fig. 9).

The obtained data not only shows the consistency of the simulation data with the tests but also allows for adjustment of the model in case the characteristic size of the damage is not accurately determined. This makes it possible to model a variety of damage to specific engines, including those for which there is currently insufficient data.

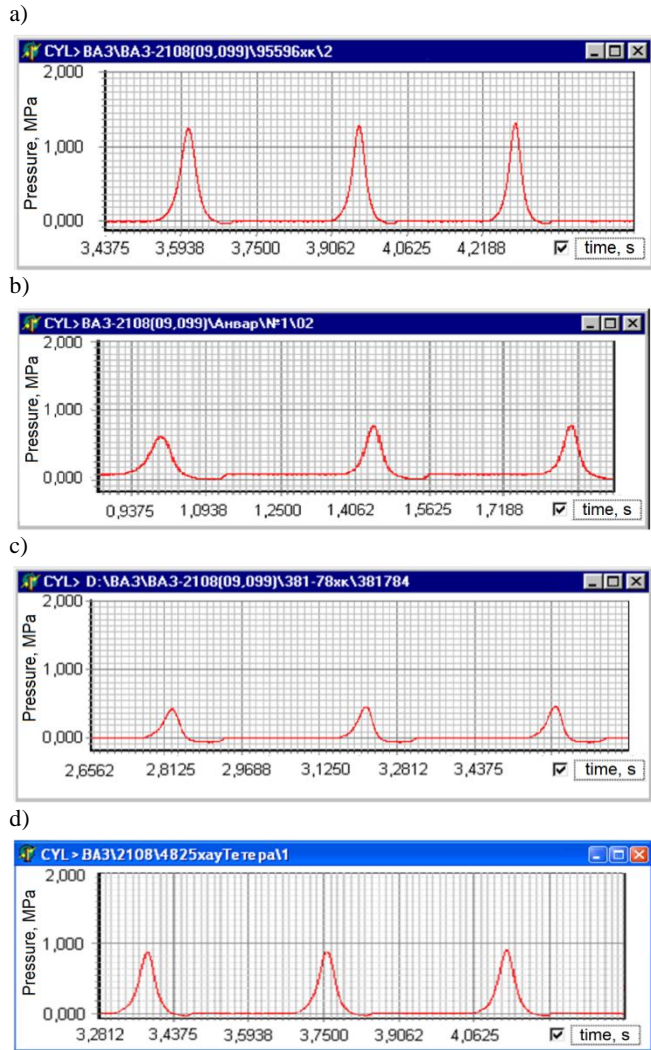


Fig. 8. Diagrams of pressure in the cylinder obtained during testing of one type of engine [33]: a) cylinder-piston group and valve mechanism in normal condition, b) severe wear of the piston rings, c) leakage of the valve-seat interface; d) presumably a drop in pressure due to hydrolock and deformation of the connecting rod

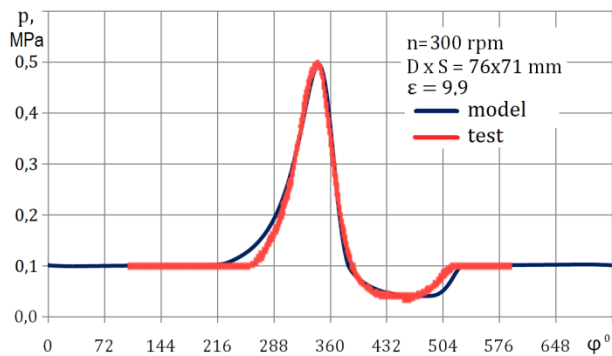


Fig. 9. Comparison of damage modeling results with experimental data for leaking valve-seat interface (diagram c in Fig. 8)

5. Simulation results for different conditions of the cylinder-piston group and valve mechanism, comparison with experimental data

The general picture of the influence of various types of damage on compression, calculated using the model, is presented in Fig. 10.

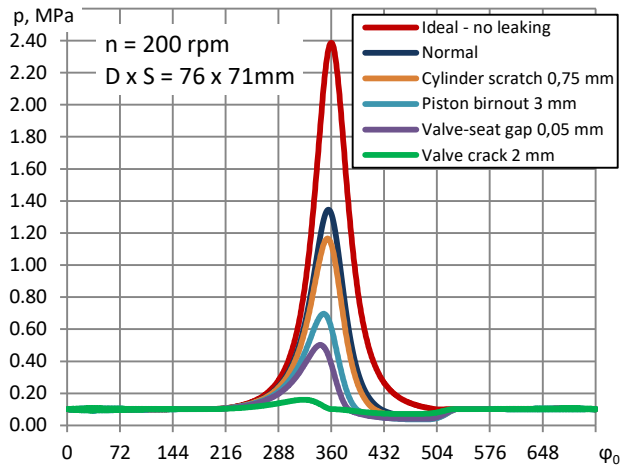


Fig. 10. The influence of various types of damage on compression obtained by modeling

Pressure change curves during cold cranking are shown sequentially for the following damage:

- 1) deep vertical scratch on the cylinder 0.75 mm wide
- 2) burnout of the piston with an equivalent flow area corresponding to a hole with a diameter of 3 mm
- 3) non-adhesion of the valve to the seat by 0.05 mm
- 4) burnout or chip on the valve head 2 mm wide.

For comparison, the diagram (Fig. 10) also shows pressure curves in a normal state (with a given leakage through the rings) and in an ideal engine with no leaks at all.

The capabilities of the developed model are demonstrated in Fig. 11, in which you can see the calculated effect of a scratch (scuff) on the cylinder surface on compression (Fig. 11a). At the same time, the most important for operational practice and diagnostics is the dependence of compression on rotation speed during cold cranking (Fig. 11b).

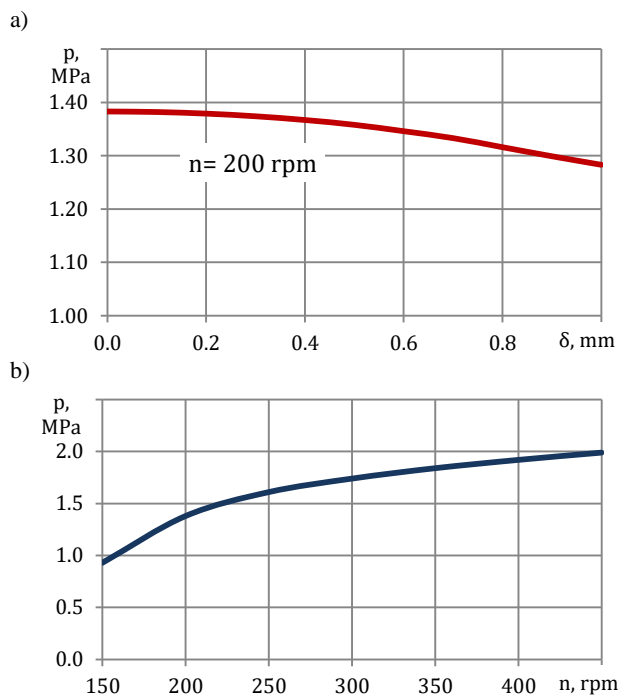


Fig. 11. Results of modeling the influence of the size of a vertical scratch on the cylinder (a) and rotational speed (b) on compression

The introduction of an additional module for calculating heat losses into the calculation model and the use of the system of equations (15) instead of (27) made it possible to further identify the effect of heat losses on compression (Fig. 12).

In the case under consideration, with a given geometry and cold cranking mode at an engine temperature of 50°C, heat losses calculated using the Woschni formula caused a decrease in compression from 1.21 to 1.11 MPa, i.e. by approximately 8%.

The model also makes it possible to simulate and predict the magnitude of compression in the event of faults for which there is currently no reliable statistical data. For example, during a hydraulic lock from liquid entering the cylinder [16], longitudinal deformation (shortening due to buckling) of the connecting rod is possible. In this case, the engine geometric compression ratio will be reduced, which will cause an obvious decrease in the maximum pressure in the cylinder. The model makes it possible to fairly accurately describe this process on a pressure diagram when measuring compression (Fig. 13).

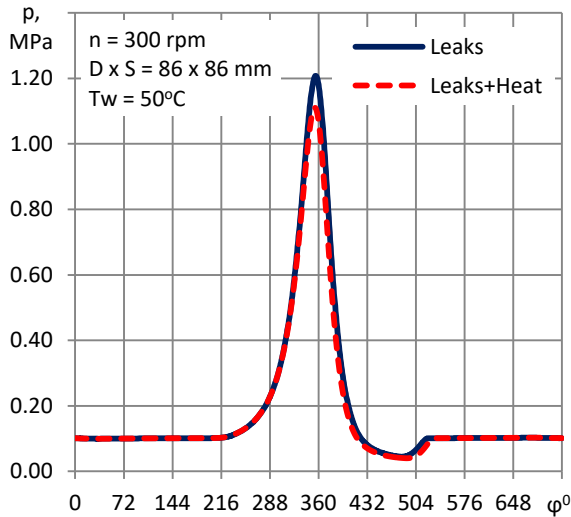


Fig. 12. Calculated effect of heat exchange on compression (engine temperature 50°C)

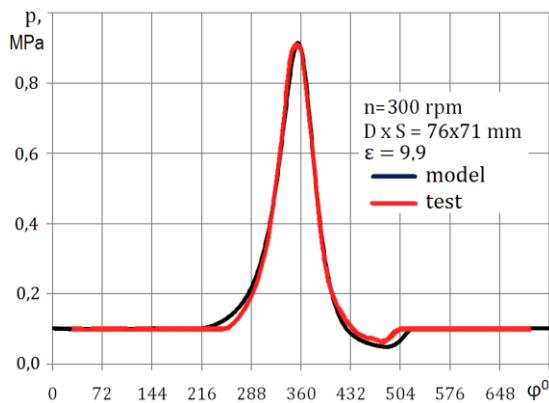


Fig. 13. Comparison of the calculated and experimental diagram of the pressure in the cylinder after hydrolock (diagram d in Fig. 3d). This diagram corresponds to longitudinal deformation (shortening) of a connecting rod of 2.6 mm

A comparison of diagrams for different damage, but with approximately close maximum compression values (Fig. 14), shows the following. During a hydrolock, if the parts of the cylinder-piston group do not receive significant damage (the engine remains operational), the difference in the diagrams of normal and damaged engines is observed only in the upper part of the pressure curve. But if the drop in compression is caused by damage associated with air leaks from the cylinder, the differences in the diagrams will also be significant in the lower part, namely, in the shape and size of the compensation pocket. The reason for such differences requires analysis and explanation.

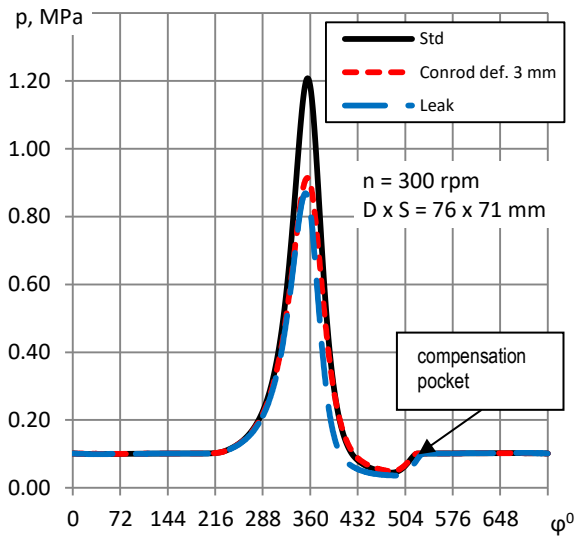


Fig. 14. Influence of the type of damage on the pressure diagram: deformation (shortening) of the connecting rod during hydrolock affects only the upper part of the curve, while additional air leaks from the cylinder also affect the size and shape of the compensation pocket.

Encouraging simulation results also allow us to study the effect of repair defects on compression. Figure 15 shows the results of modeling the variation of the compression diagram due to an error in setting-up the valve timing.

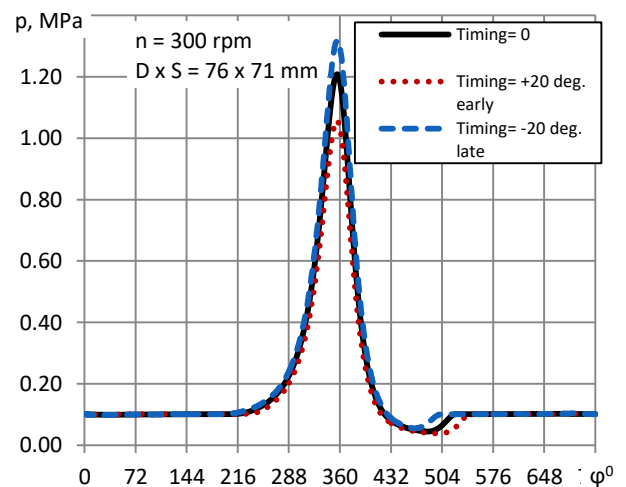


Fig. 15. The influence of the valve timing shift on the compression diagram

The effect of the maximum pressure in the cylinder of a shift in the setting angle of the valve timing from the nominal value is shown in Fig. 16.

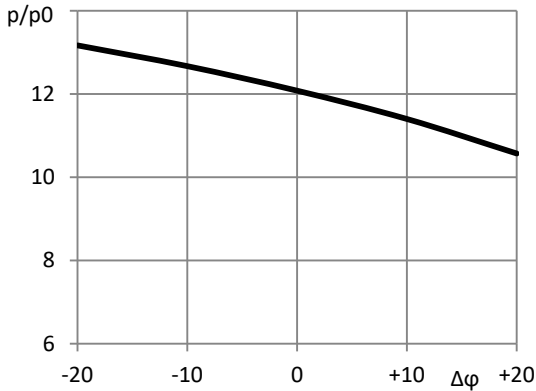


Fig. 16. Maximum pressure in the cylinder as a function of changing the valve timing angle from the nominal zero value

The model as a whole well confirms the nature of compression changes known from practice. So, a clockwise shift of the camshaft relative to the crankshaft (early valve timing, for example, due to a repair error) causes a decrease in compression. A shift in the opposite direction (later valve timing, for example, due to driving belt or chain jumping), on the contrary, leads to an increase in compression. This is important for some engines of previous years of production that did not have a camshaft position sensor when an error in setting up the valve timing is not shown by the control system. However, the most important is quantitative data, including changes in the width of the compensation pocket (Fig. 15), which may be important for the correct diagnosis of faults.

6. Analysis of the results obtained, determination of general patterns connecting the amount of compression with various types of damage to engine parts and measurement conditions

When analyzing the reliability of the developed model, two features that confirm the results obtained should be especially noted. Thus, the convergence of the model is determined by the coincidence or at least very close values of the input and output cyclic values of pressure and temperature in the cylinder. In the case under consideration, such a coincidence occurs in both pressure and air temperature (Fig. 7, 9, 10), and the convergence in temperature is more important, and the discrepancy here does not exceed 1%.

In addition, an important confirmation of the correctness of the obtained results is the shape and size of the characteristic compensation (expansion) pocket on the curves after the expansion section (downward stroke of the piston after compression). This pocket occurs as a result of air leaking from the cylinder during compression, causing the cylinder pressure at the bottom dead center to be lower than ambient pressure.

As follows from the experiments [31], the real engines have a compensation pocket that is close to the calculated one not only in shape, but also in the value of the parameters (Fig. 7, 9). It is also obvious that the larger the pocket,

the higher the air loss due to leakage from the cylinder [27]. Conversely, as leakage decreases, the size of the pocket will decrease. In the limit, in the absence of leaks and heat losses, a pocket does not form (Fig. 10). All these features can be easily simulated using the model.

The influence of various factors in the form of operational damage on the compression value generally corresponds to real practice [26, 27]. Indeed, damage to the valve mechanism has the greatest impact on compression, while damage to the cylinder-piston group is less critical. The simulation results shown in Fig. 7 and 10 generally confirm the known experimental data [31, 34].

Separately, the effect of heat losses from the cylinder on compression (Fig. 12) should be noted. As follows from the data obtained, any type of leakage from the cylinder reduces compression noticeably more than the heat loss calculated using the Woschni formula (Fig. 10). On the one hand, this result confirms the initial assumption that it is possible to simplify the model by taking into account heat losses using additional leakage, which constitutes only a small fraction of the total. On the other hand, there is currently no reliable data on the effect of heat loss under cold cranking conditions. Therefore, perhaps in the future the model should be refined based on heat losses from the cylinder.

From the point of view of the model's practical use, the dependence of compression on rotation speed is of the greatest importance (Fig. 11b). As it is possible to see, an increase in rotation speed from 150 to 450 rpm causes a 2-fold increase in compression. The simulation data actually confirms the fact known from practice and research [26] that compression always increases noticeably with increasing rotation speed. This dependence is quite obvious, since any leaks from the cylinder occur in a time process which decreases in inverse proportion to the frequency. It turned out to be difficult to introduce such a correction using empirical and statistical methods [24, 33]. Thus, the model allows what is difficult to do with the help of experiments, namely, to predict with sufficiently high accuracy how the compression in a particular engine will change when the rotation speed changes.

That means, the model makes it possible to introduce a correction for any compression meter readings, if the rotation speed at which the test was performed is known. Moreover, this amendment is extremely important for correct engine diagnostics. This allows you to determine the effect on the readings of the device of changes in speed for any reason, including changes in the characteristics of the battery, starter, engine oil, engine temperature, pressure and ambient temperature, etc. That is, the model allows you to almost completely exclude (or, conversely, take into account) the influence of any operational factors on compression as an important diagnostic parameter. And this is something that usually cannot be done in other ways.

As shown in Fig. 13 and 14, modeling processes during cold cranking can significantly expand the diagnostic capabilities of certain types of faults, including hydraulic lock in the cylinder [16]. If the sealing properties of the piston rings do not deteriorate during hydrolock (and this is so what usually happens in practice), then leaks from the cylinder do not increase with increasing pressure. In accord-

ance with this, despite a noticeable drop in the maximum pressure due to deformation (buckling) of the connecting rod, the lower part of the pressure diagram in general and the compensation pocket in particular will not change during hydrolock. This fundamentally distinguishes this type of damage from any others, which are usually associated with an increase in air leaks from the cylinder and, accordingly, with a significant expansion of the compensation pocket (Fig. 14).

The shape of the compensation pocket is also characteristic if the valve timing is shifted (Fig. 15a), when the pocket width actually changes only by the movement of the exhaust valve opening point. Then, if there is a certain reference compression diagram for a given engine, the indicated defect can be detected by a disproportionate change in the width of the pocket.

It follows that with a correct comparative analysis of the compression diagram and comparing it with the simulation results, it becomes possible not only to localize the damaged area (cylinder), but also to obtain reliable data on the nature and cause of serious damage in the operation. All this is possible even without engine disassembly and can prevent the destruction of a damaged engine [16], including significantly, many times, to reduce the cost of restoration repairs.

At the same time, it should be noted that the results obtained represent only the first experience of using the developed model. They show that it is necessary, among other things, to further improve the model, and, firstly, to improve the stability of the calculation algorithm when describing intake-exhaust processes at low rotation speeds. In addition, it is necessary to remove some simplifications that may affect the results. For example, it seems necessary to introduce and take into account a variable crankshaft rotation speed, since uneven rotation at low speed can be significant. In addition, it is necessary to continue research of the model itself for various types of leaks. For full compliance with the real process, you can also include the volume of the connecting tube in the model and take into account the hydraulic resistance of the check valve of the compression meter. That is, in this case, the greatest importance is to compare the calculation results with experimental data in order to identify quantitative characteristics and patterns

that could make the model suitable for use in diagnostic and monitoring practice.

7. Conclusions

A thermodynamic model has been developed to calculate the in-cylinder processes of an internal combustion engine in cranking mode when measuring compression. The model describes the processes in the cylinder step by step, taking into account the real patterns of the intake-exhaust processes, air leakage through the interfaces of parts and heat exchange with the walls.

To test the developed model, control mathematical modeling was performed. The results obtained were compared with known experimental data on oscillography of pressure in the cylinder during cold cranking. When analyzing the accuracy and reliability of the model, it was found that the model has convergence in pressure and temperature up to 1%. After setting up the model, this made it possible to obtain the shape of the pressure diagram in the cylinder, including a characteristic compensation pocket in the curve, consistent with the experimental data.

Using modeling, the main patterns of changes in compression depending on the modes, the nature of damage to the parts of the valve mechanism and cylinder-piston group, the amount of leakage and engine temperature were found. It has been established that heat losses reduce compression by approximately 8%, and when the cold cranking frequency increases from 150 to 450 rpm, the compression value can double. As a result, the model allows you to introduce a correction to any compression meter readings if the rotation speed at which the test was performed is known, and thereby eliminate the influence of operational factors on the measurement results. These properties of the model make its use effective in diagnosing and monitoring the technical condition of engines in operation.

As a prospect for further research, it is expected to improve the model to improve the stability of the calculation algorithm for simulating intake-exhaust processes at low rotation speeds, taking into account variable crankshaft rotation speed, as well as continuing the study of the model itself for various types of damage and associated leaks.

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