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SIMULATION OF DIESEL ENGINE ENERGY CONVERSION PROCESSES

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In order to keep diesel engines in good working order the troubleshooting methods shall be improved. For their further improvement by parameters of associated processes a need has arisen to develop a diesel engine troubleshooting method based on time parameters of operating cycle. For such method to be developed a computational experiment involving simulation of diesel engine energy conversion processes has been carried out. The simulation was based on the basic mathematical model of reciprocating internal combustion engines, representing a closed system of equations and relationships. The said model has been supplemented with the engine torque dynamics taking into account the current values of in-cylinder processes with different amounts of fuel injected, including zero feed.

The torque values obtained by the in-cylinder pressure conversion does not account for mechanical losses, which is why the base simulation program has been supplemented with calculations for the friction and pumping forces. In order to determine the indicator diagram of idle cylinder a transition to zero fuel feed mode and exclusion of the combustion process from calculation have been provisioned.

Key words: diesel, in-cylinder processes, diesel engine energy conversion, fuel injection, indicator values, torque, piston block friction forces, piston speed, friction in crankshaft bearings, pumping losses, simulation, indicator diagram, zero feed.

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Introduction. Diesel engines are increasingly used as power units of motor vehicles, as have a number of advantages over the petrol rival: they are more cost-effective and eco-friendly, have high efficiency and power, etc.

In order to keep diesel engines in good working order there exists a planned preventive maintenance system, including troubleshooting as its technological component [1, 11, 12].

The existing troubleshooting methods are continually evolving and improving [9, 10, 13]. For further improvement of troubleshooting methods by parameters of associated processes a troubleshooting method based on time parameters of operating cycle has been developed [2].

In the process of its development a computational experiment involving simulation of diesel engine energy conversion processes has been carried out.

Simulation of Diesel Engine Conversion Processes. The simulation was based on the basic mathematical model of reciprocating internal combustion engines, representing a closed system of equations and relationships, which identify with the necessary degree of reliability the quantitative relationships between a certain combination of input actions (factors) and output parameters of these processes [2, 5, 6].

But to determine the engine torque dynamics the current values of in-cylinder processes with different amounts of fuel injected, including zero feed, shall be known [3, 7, 8, 14].

Indicator torque (M) appears due to the impact on the crank of the tangential component (T) of force ($P_{ш}$) acting along the axis of the connecting rod AB (Fig.1),

$$M = TR\,, \tag{1}$$

where M is an indicator torque, N·m; R is a crank radius, m; T is a tangential force, N. In its turn the tangential force

$$T = P_{ш}\sin(\varphi+\beta)\,, \tag{2}$$

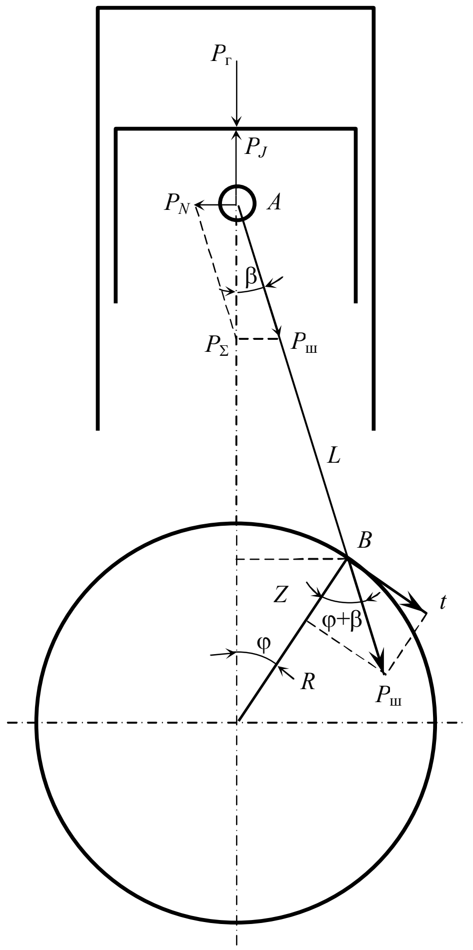

Fig.1. Crank and rod mechanism of internal combustion engine and forces exerted thereupon

where $P_ш$ is the force acting along the axis of the connecting rod, N; β is the angle of the connecting rod deflection from direction of the cylinder axis, deg.; φ is the crankshaft rotation angle, deg.

Force acting along the axis of the connecting rod,

$$P_ш = \frac{P_\Sigma}{\cos\beta}, \tag{3}$$

where P_Σ is the resultant force applied to the piston pin axis and acting along the direction of the cylinder axis,

$$P_\Sigma = P_i F_п + P_{ji}, \tag{4}$$

P_i is the in-cylinder pressure in the i^{th} point of the cycle; F_n is the area of the piston; P_{ji} is the inertia force of the masses moving forward in the i^{th} point of the cycle.

The inertia force P_{ji} was determined using the dynamic equation of axial crank and rod mechanism:

$$P_{ji} = -m_A R\omega^2(\cos\varphi_i + \lambda\cos 2\varphi_i), \tag{5}$$

where m_A is the mass of forward moving parts of CRM; R is the crank radius; $\omega = \pi n/30$ is the angular speed of the crankshaft; λ is the crank radius to rod length ratio.

The mass m_A was calculated by the following formula:

$$m_A = m_{p.b} + m_ш^A, \tag{6}$$

where $m_{п.к.}$ is the mass of the piston block; $m_ш^A$ is the mass of the upper part of the rod referred to the reciprocating moving parts,

$$m_ш^A = k_ш m_ш, \tag{7}$$

where $k_ш$ is the mass fraction of the rod referred to the reciprocating moving parts.

Through step-by-step calculation using formula (4) the resultant force $P_{\Sigma i}$ has been determined for each value of the crankshaft angle φ_i.

The torque generated on the crankshaft due to the effect of this force was calculated for each φ_i under the below formula

$$M_{ki} = P_{\Sigma i}\frac{\sin(\beta_i + \varphi_i)}{\cos\beta_i}R, \tag{8}$$

where β_i is the angle of the rod deflection from direction of the cylinder axis.

In order to obtain values of β_i angle, the following empirical formulas were arrived at:

$$\beta_i = 60\sin\beta_i - 26(\sin\beta_i)^2, \tag{9}$$

where $\sin\beta_i \approx \lambda\sin\varphi_i$.

The torque values obtained by the in-cylinder pressure conversion shall not be viewed as indicator values, as they do not account for mechanical losses: friction forces and forces required to enable pumping strokes.

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In view of this, the calculation of friction forces and pumping processes was introduced into the basic simulation program.

Current values of the friction force (in Newtons) of the piston group can be found by Newton's formula:

$$T = \mu F \frac{dc}{dz}, \tag{10}$$

where μ is the dynamic coefficient of engine oil viscosity, Pa·s; F is the frictional area of the piston group; dc/dz is the gradient of velocity in the lubricant layer, 1/s.

It is assumed that the velocity gradient in the i^{th} point is

$$\frac{dc}{dz} = \frac{c_{\max}}{\delta_{м}}, \tag{11}$$

where $c_{\max i} = c_{n_i}^{0,5}$ is the maximum velocity of the lubricant layer in the i^{th} point, m/s; $\delta_м$ is the lubricant layer thickness, m,

$$\delta_{м} = \frac{1}{\sqrt{P_i}}. \tag{12}$$

With account of formula (12) and based on the conducted experimental studies an empirical formula has been built to determine the piston group friction force:

$$T_{п_i} = 1.76510^{-3} D^2 \sqrt{P_i C_{пi}},$$

where P_i is the resultant force of gas pressure and inertia forces acting on the piston group, N.

The piston speed in this point is

$$C_{п_i} = R\omega(\sin\varphi_i + \frac{\lambda}{2}\sin 2\varphi_i).$$

The friction in the crankshaft bearings was taken into account together with the losses associated with actuation of auxiliary units in the form of a torque constant by the angle of crankshaft rotation subtracted from the resultant value of the torque M_{k_i}

$$M_{к_{gi}} = M_{Ri} - \Delta M,$$

$$\Delta M = (34.43\sqrt{nP_{г}} + 53.46\sqrt{n}) i V_h,$$

where P_r is the average force acting on the CRM; n is the crankshaft speed, min^{-1}.

Pumping losses were reflected in the program by numerical integration of expression $(P_i - P_0)dV$ within the bounds of inlet φ_i = 0-180 degrees of CRA and outlet φ_i = 540-720 degrees of CRA processes.

The main block of simulation program provides for building the indicator diagram of a single operating cycle. For operating cylinders the torques were summed up at each φ_i value with account of their phase position in the cycle.

In order to build the indicator diagram of idle cylinder the program implies automatic transition to zero fuel feed mode and exclusion of the combustion process from the calculation. Meanwhile the indicator processes of idle cylinder were distributed within the general range of crankshaft rotation angles φ_i in accordance with their operation order.

Total summation of current torques was carried out with account of specific phase position of in-cylinder processes of all cylinders:

$$M_{\Sigma i} = M_{кi}^{\mathrm{I}} + M_{кi}^{\mathrm{II}} + M_{кi}^{\mathrm{III}} + M_{кi}^{\mathrm{IV}} + M_{кi}^{\mathrm{V}} + M_{кi}^{\mathrm{VI}} + M_{кi}^{\mathrm{VII}} + M_{кi}^{\mathrm{VIII}},$$

where $M_{кi}^{\mathrm{I}} - M_{кi}^{\mathrm{VIII}}$ are the current values of the torque generated through operation of the corresponding cylinders.

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Mathematical model of changes in pressure of the engine's idle cylinder by the crankshaft rotation angle was developed based on the equation of state (Clapeyron-Mendeleev equation) [14, 15]:

$$PV = GRT,$$

where P, V, T are the current pressure (Pa), volume (m^3) and air temperature (K); R is the gas constant, J/(mol K); G is the amount of air in the cylinder, kg.

The model was developed based on the following assumptions:

• for the pressure calculations gas inside the cylinder is assumed to be the perfect gas (the discharged gas, such as the gas in normal environmental conditions);

• at each step of calculation by the crankshaft rotation angle the thermodynamic system is in the state of equilibrium (gas quasi-static state).

The ultimate purpose of simulation is determination of current pressure values in the engine cylinder, i.e. getting relationship expressed as

$$\frac{dP}{d\varphi} = f(\varphi),$$

where φ is the crankshaft rotation angle, degrees of CRA.

In addition the system of equations includes thermodynamic and kinematic relationships describing changes in the volume, temperature and pressure inside the cylinder.

The change in pressure inside the cylinder during compression and expansion on the basis of the equation of state can be found by the below formula

$$P_{ri} = \frac{RGT_i}{V_i},$$

where R is the gas constant, J/(kg·deg.); G is the amount of air in the cylinder, kg; T_i is the current temperature of the cylinder charge, K; V_i is the current cylinder volume, m^3.

Current temperature value

$$T_i = T_{i-1}\left(\frac{V_{i-1}}{V_i}\right)^{n-1},$$

where n is the polytropic index determined by pressure and volume at the beginning (i-1) and at the end (i) of the segment:

$$n = \frac{\log \frac{P_{i-1}}{P_i}}{\log \frac{V_i}{V_{i-1}}}.$$

Current cylinder volume

$$V_i = V_h\left[\frac{1}{\varepsilon - 1} + 0.5\left((1-\cos\varphi_i) + \lambda/4(1-\cos 2\varphi_i)\right)\right],$$

where V_h is the cylinder capacity, m^3; ε is the compression intensity; λ is the crank radius to rod length ratio.

For simulating air leakage through the ring seal the computational model shall be examined [4, 15] (Fig. 2).

With the flow of the working medium through the ring seal system the gas load is distributed between all rings in a certain way. It's characterized in quantitative terms by the pressure in the volume behind each ring (pressure in the behind-the-ring volume). Let's make a number of assumptions concerning the process under examination:

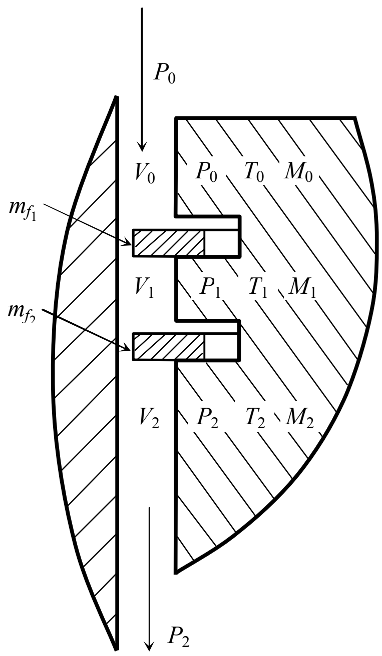

Fig. 2. Computational model of piston ring seal

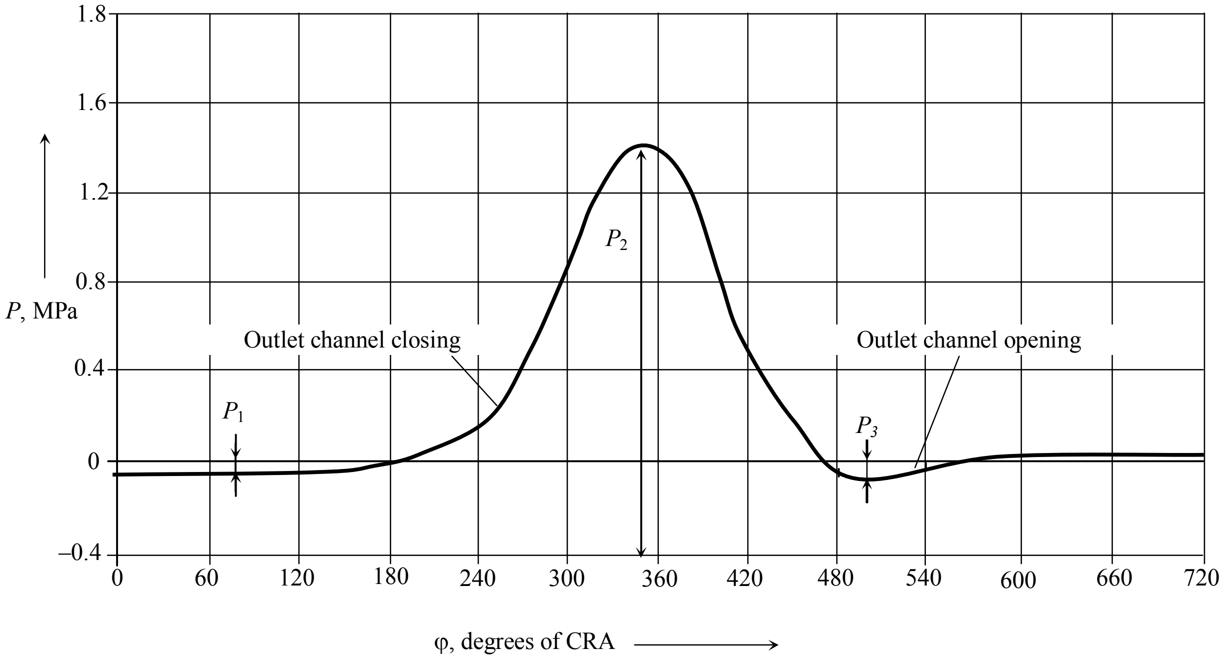

Fig.3. Graph of changes in pressure inside cylinder by crankshaft rotation angle

• the air charge flows from the cylinder working chamber through the locks of piston rings;

• the time of air charge flow through the ring seal is short, therefore the process can be regarded as adiabatic;

• the Strouhal number, describing the flow unsteadiness, in this case has the order of 0 (10^{-2}), therefore the task can be regarded as quasi-stationary.

In general terms the gas flow Q (in kilograms per second) through the valve opening and ring seal looseness is determined based on the known Saint-Venant dependence, derived from the equation for subcritical escape ($P_i / P_{i-1} > 0.53$) [4, 15]:

$$Q = \mu f \sqrt{2\rho\Delta P} \,,$$

where μ is the flow coefficient; f is the flow section area, mm^2; ΔP is cylinder differential gas pressures in the ring seal, MPa; ρ is the air density, kg/m^3.

The amount of air ΔG (in kilograms), passing through the cross-section μf for a period of time corresponding to the step of calculation $\Delta\varphi$, is found by the below formula

$$\Delta G = Q \frac{\Delta\varphi}{6n}.$$

Fig.3 shows a graph of changes in pressure inside the cylinder by crankshaft rotation angle, produced as a result of simulation. The graph demonstrates changes in pressure inside the cylinder by crankshaft rotation angle. Here the following indicative points for assessing the state of cylinder-piston group can be identified:

1) the inlet pressure P_1 characterizes the intake tract tightness and condition of the cylinder liner;

2) pressure P_2 corresponds to the piston position at the top dead center (TDC) of the cylinder and characterizes its tightness and condition of valves;

3) pressure P_3 is lower than the atmospheric pressure and characterizes condition of the piston rings and valves.

It is simulated by changing the flow cross section f. When the piston goes up to the top dead center on the compression stroke some part of air charge goes into the oil pan through the ring seal looseness, thereby due to such leakage a short-term underpressure is created on the expansion stroke lasting until the opening of the exhaust valve. This underpressure shows how well the piston rings seal the overpiston space.

Conclusion. Thus, the simulation of diesel engine energy conversion processes based on the basic mathematical model of reciprocating internal combustion engines was supplemented with the engine torque dynamics taking into account the current values of in-cylinder processes with different amounts of fuel injected, including zero feed. In order to account for mechanical losses when determining the torque value the basic simulation program was supplemented with calculations for the friction and pumping forces. In order to determine the indicator diagram of idle cylinder a transition to zero fuel feed mode and exclusion of the combustion process from calculation have been provisioned. The simulation results are proven by satisfactory convergence with the graphs and oscillograms produced during diesel engine troubleshooting with the divergence not exceeding 10%.

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