Thermodynamic condition of cylinder diesel group for infrastructure objects under high temperatures

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Abstract. The article presents the results of research on the influence of ambient temperature on the thermal state of cylinder-piston groups of diesel engines, carried out by the experimental-calculation method on diesel engines 12ChN 18/20 (M787B) and 8ChN 13/14 (YAMZ-7511.10) of autonomous sources on factory test benches. In the experimental study of the effect of temperature change on the thermal state of cylinder-piston groups, the diesels were equipped with a special water-air heat exchanger for heating the air at the suction to the compressor, and the temperature of the suction air was regulated by changing the flow rate of hot water through the water-air heat exchanger. In the experimental study, along with the thermometry of parts of cylinder-piston groups, the engine operating process was indicated, the heat balance was taken, etc.; and the experimental data of other researchers were analyzed. The results of this experimental study, as well as analyzing the results of earlier work on the effect of temperature on the thermal state of cylinder-piston groups allowed to obtain empirical dependencies of local temperatures of the walls of their parts, the distinguishing feature of which is that as variables are considered their relative (dimensionless) values, which increases the degree of experimental data. The results of the performed research can be used for the calculated estimation of the temperature level in the walls of parts of cylinder-piston groups of diesel engines of a wide class of both stationary and transport diesel engines, while providing satisfactory convergence of the calculation results with the experimental data.

1 Introduction

Diesel engines of diesel generator sets (DGU) used in autonomous power supply sources (APS) of power infrastructure facilities (PIF) are operated in wide ranges of changes in temperature, barometric pressure and humidity of ambient air. Analysis of the results of experimental studies given in [1-4] shows that the greatest influence on the engine operation parameters is exerted by the change in ambient air temperature t_a. At constant cyclic fuel supply b_c, which is typical in the conditions of operation of DGU, with increasing temperature t_a worsens the power and economic performance of the engines. When the temperature t_a

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increases, the engine power *Pe* decreases and the specific effective fuel consumption b_e increases due to the decrease in the indicator efficiency η_i due to a decrease in the excess air ratio at combustion α1.

It is shown in [3, 4] that the temperature increase significantly affects the thermal state of the diesel engine. The latter (first of all, its cylinder piston group (CPG)) when operating at the same speed mode (n = const), at a constant value of b_c and different t_a is determined mainly by the temperature of gases in the cylinder, which depends on the coefficient α_1. An increase in t_a causes an increase in the temperature of gases, and, consequently, the heat balance is established at higher levels of the temperature of the walls of the parts forming the combustion chamber (CC), temperature gradients, and heat fluxes through these walls. The above-mentioned indices characterize the thermal stress of the CCC.

2 Materials and methods

2.1 Influence of temperature on the thermal state of cylinder-piston groups of diesel engines.

The research on the influence of t_a temperature on the thermal state of diesel engines' cylinder piston groups was carried out by experimental and calculation method on 12ChN 18/20 (M787B) and 8ChN 13/14 (YAMZ-7511.10) diesel engines of autonomous sources on factory test benches. During the experimental study, along with thermometry of CKD parts, the engine operating process was indicated, the heat balance was taken and others.

In addition, the experimental data of other researchers were analyzed [1,3,4].

When investigating the influence of t_a temperature change on the thermal state of the CKD, diesels were equipped with a special water-air heat exchanger for heating the air at the suction to the compressor. The intake air temperature was regulated by changing the flow rate of hot water through the water-air heat exchanger. At the same time, the temperature of water t_{cool} and oil t_a at the diesel inlet was maintained at the level of 75÷80 and 78÷80 °C, respectively, and the temperature of air in the charge air manifold tint = 52÷55 °C, i.e. the cooling efficiency coefficient E_p of the charge air cooler was practically constant.

Fig. 1 shows graphs of temperature change of the cylinder head fire bottom of 12ChN 18/20 diesel engine and temperature of aluminum uncooled piston of 8ChN 13/14 diesel engine. The graphs show that the temperature of the bottom of the cylinder head in the zone between the intake valves increases by 10÷12 ° C, and in the zone between the intake valves by 3÷4 ° C for every 10 ° C increase in temperature t_a. At the same time, the temperature difference between the inlet and exhaust valve jumpers increases by 6÷7 °C.

With increasing t_a the temperature in different points of the piston surface has different intensity of growth, and for every 10 °C of increasing t_a the temperature of the piston at the periphery increases by 12÷14 °C, in the center of the piston by 5÷6 °C, in the area of the upper piston ring on the exhaust side by 8÷10 °C and on the inlet side by 5÷7 °C.

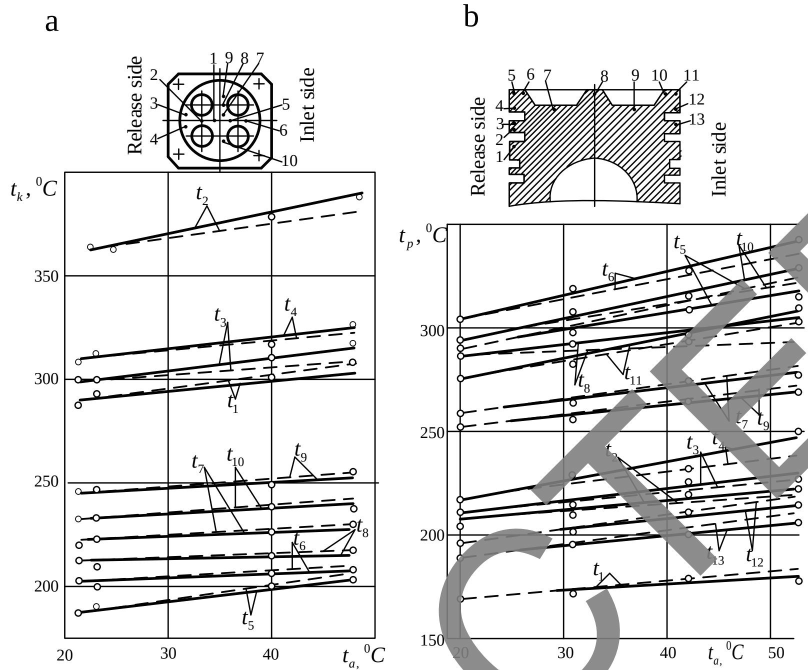

Fig.1. Variation of the temperature of the cylinder head fire bottom of 12ChN 18/20 diesel engine (a) and the temperature of the piston of 8ChN 13/14 diesel engine (b) from the temperature t_a at constant b_c and n: 1-13 - numbers of temperature measurement points; —— - experience; - - - - - calculation.

The results of this experimental study, as well as analyzing the results of earlier work on the effect of temperature t_a at constant b_c and n allowed us to obtain empirical dependencies of local temperatures of the walls of the parts of the CPG. The distinctive feature of these dependences is that their relative (dimensionless) values are considered as variables, which increases the degree of experimental data. Below are the dependencies for determining the local temperatures of the bottom of the cylinder head, piston and cylinder sleeve:

$$T'_c = T_c \cdot \left[0.15\bar{r}\,(1-\frac{T'_a}{T_a}) + 0.4\frac{T'_a}{T_a} + 0.6 \right] \quad (1)$$

$$T'_p = T_p \cdot \left[0.5\bar{r}\,(\frac{T'_a}{T_a} - 1) + 0.2\frac{T'_a}{T_a} + 0.8 \right] \quad (2)$$

$$T'_r = T_r \cdot \left[1.4\bar{h}\,(1-\frac{T'_a}{T_a}) + 0.7\frac{T'_a}{T_a} + 0.3 \right] \quad (3)$$

$$T'_{CS} = T_{CS} \cdot \left[0.7\bar{l}\,(1-\frac{T'_a}{T_a}) + 0.8\frac{T'_a}{T_a} + 0.2 \right], \quad (4)$$

where T_c, T_p, T_r, T_{CS} - and are the temperatures in the cylinder head fire bottom, piston head fire bottom, piston in the ring zone and cylinder sleeve respectively at standard ambient air temperature T_a = 300^{0}K;

T_a - is the current value of ambient air temperature;

$\bar{r}$ - relative radius of the cylinder equal to *r/R* (*r* is the current value of the radius, *R* is the radius of the cylinder);

$\bar{h} = h / h_p$ - the relative distance along the piston's piston's form from the outer surface of the piston's bottom to the point at which the temperature is to be determined,
where h - is the current value;

h_p - height of the piston);

$\bar{l}$ - relative distance from the junction plane of the sleeve and cylinder head to the point at which it is necessary to determine the temperature of the cylinder sleeve mirror, equal to *l/s* (*l* - current value, *s* - piston stroke).

Fig. 1 shows that the calculated and experimental values of temperatures have quite satisfactory coincidence similar to the experimental and calculated by the dependences (1 - 4) values of temperatures of parts of the CKD for other engines. Divergence of calculated and experimental values of temperatures for the cylinder head diesel 8ChN 13/14 in characteristic points is up to 1.6 %. For cooled composite piston diesel 8ChN 13/14 divergence of calculated and experimental values of temperature is slightly greater - 2.5 ÷ 4.5%. Also satisfactory coincidence have experimental and calculated values of temperatures for cylinder sleeves of other brands of diesel engines 6ChN 21/21 and 10DN 20.7/(2 X 25.4), where the discrepancy is up to 2.5%. Thus, the above dependencies allow to determine with a sufficient degree of accuracy the change in local temperatures of the walls of the parts of the CKD at a change in temperature t_a and constants b_c and *n*.

In more detail, the computational study of the thermal state of CKD parts at temperature change t_a can be performed by modeling the temperature fields in the walls using boundary conditions of the third kind. The local heat transfer coefficients α_{cr} and the resulting gas temperatures $t_{g.res}$ at temperature change ta can be calculated using the methodology described in [5]. For this purpose, it is necessary to have initial data (first of all, indicator diagrams). To calculate the indicator diagrams, it is necessary to determine the value of pressure p_{int} and temperature T_{int} of air in the supercharging receiver.

The analysis of the experimental data obtained and given in the literature on the influence of temperature t_a on the main parameters of diesel engines operation at constant b_c and *n* shows that the rotor speed of the turbocharger rotor n_{tc} remains practically constant. This allowed us to consider the adiabatic work of air compression in the compressor $L_{c.ad}$ constant at temperature change t_a. Taking $L_{c.ad}$=const and neglecting the resistance of the charge air cooler, from the known equation for determining the adiabatic work of compression of 1 kg of air in the compressor after simple transformations, we can obtain the dependence for calculating the pressure pint at temperature change t_a, i.e., from the known equation for determining the adiabatic work of compression of 1 kg of air in the compressor.

$$p_{\text{int}} = \left[\frac{T_a}{T_a^{'}}(\pi_c^{\frac{k-1}{k}} - 1\right]^{3,5} p_a \qquad (5)$$

where π_c - is the degree of pressure increase in the compressor at initial T_a and p_a.

The discrepancy between the values of pressure value p_{int}, calculated by dependence (5), and the experimental values obtained on different diesel engines is about 1%, which is shown in Fig.2.

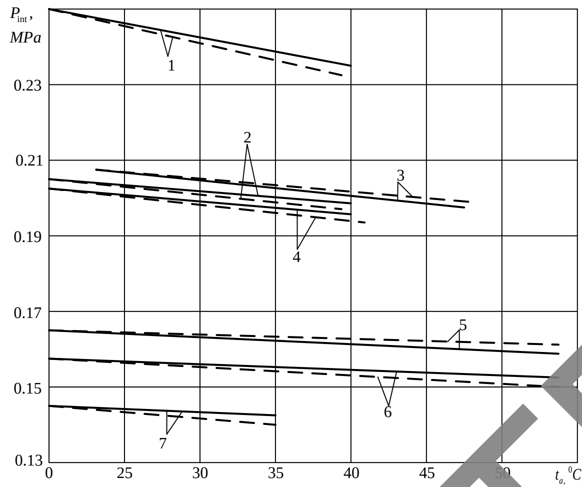

Fig. 2. Variation of pressure p_{int} from temperature t_a at constant b_c and n diesel engines: 1 - 16ChN 25/27 (P_e = 2070 kW); 2 - 10DN 20,7/(2x25,4) (P_e = 2210 kW); 3 - 12ChN 18/20 (P_e = 883 kW); 4 - 7D100 (P_e = 1100 kW); 5 - 64N 13/11,5 (P_e = 125 kW); 6 - 6ChN 31,8/33 (P_e = 993 kW); 7 - 8ChN 13/14 (P_e = 200 kW); Values of curves see in Fig.1.

The temperature T_{int} can be determined by considering the degree of efficiency of the charge air cooler.

Fig. 3 shows the indicator diagrams of 12ChN 18/20 and 8ChN 13/14 diesel engines at constant b_c and n of the diesel crankshaft and temperature change t_a, which served as initial data for calculating the boundary conditions of the third kind in modeling the temperature fields in the walls of the cylinder head fire bottom and the bottom of the piston head. Calculations of indicator diagrams were performed according to the methodology described in [6]. From the graphs of Fig. 3 shows that the calculated and experimental indicator diagrams have a satisfactory coincidence, creating prerequisites for obtaining, to a certain extent, reliable values of local heat transfer coefficients and the resulting heat transfer temperature of gases. t_{res}

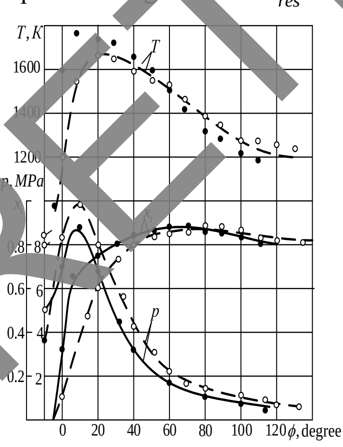

Fig. 3. Indicator diagrams for constant values of bc and n of diesels:
—- 8ChN 13/14, ta = 48 °C;
--- - - 12ChN 18/20, ta = 50 °C.
Experimental points - experience; curves - calculation.

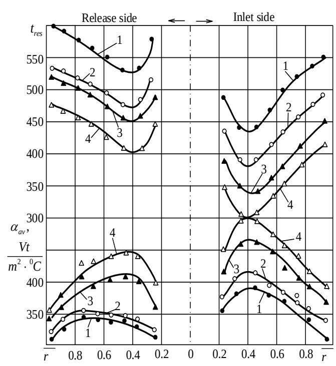

Fig. 4. Variation of the coefficient αsr and tg.res from the temperature ta:
1 and 2 - ta = 48 and 20 °C, diesel 8ChN 13/14;
3 and 4 - ta = 50 and 23 °C, diesel 12ChN 18/20.

The graphs shown in Fig. 4 graphs show that the values of the cycle-averaged local heat transfer coefficients αcr practically do not depend on the temperature t_a. This is due to the following. A change in temperature t_a (e.g., its growth) leads to an increase in the temperature of gases in the cylinder and the CW wall, as well as to a certain decrease in the gas pressure. At the same time, the thermal conductivity and heat capacity of the gas slightly increase, and the gas density decreases. The combined effect of these parameters causes an insignificant change in the heat penetration coefficient $\sqrt{\lambda_m C_{pm} \rho_m}$.

Approximately similar changes occur with other parameters determining the coefficients of convective and radiative heat transfer. Thus, the intensity of lecal heat transfer with temperature increase from 20÷23 to 48÷50 °C decreases by 4÷6 %, i.e. practically it can be considered unchanged in the specified range of ambient air temperature change.

The resulting heat release temperature of gases $t_{g.res}$, as can be seen from the graphs of Fig. 4, depends to a large extent on the temperature t_a. This is due to the change in the temperature of the gases in the cylinder, which depends to a large extent on the coefficient α'_1, the variation of which at a constant value is due to the change in temperature t_a. [7-19]

3 Results and discussions

The results of the performed research allowed us to establish the value of the reduction in the rated power of diesel engine 12ChN 18/20 (M787B) autonomous source of DEI when operating it at elevated temperatures T_a = 313^{0}K, provided that the level of temperatures of the bottom of the cylinder lid will not exceed the level of temperatures in the mode P_e = 883 kW (n = 750 min^{-1}) at T_a = 293^{0}K.

It was found that in order to limit the cylinder lid firing bottom temperature at t_a = 40°C, it is necessary to set the rated power P_e = 790 kW at t_a = 20°C. In this case, the temperature of the cylinder head will not exceed the temperature in the most heated places (the jumper between the windows of exhaust valves, t_{c2} = 368°C) in the mode of rated power at t_a = 20 °C.

4 Conclusion

The results of the performed research can be used for the calculated estimation of the temperature level in the walls of the parts of the CPG of a wide class of both stationary and transport diesel engines, while ensuring satisfactory convergence of the calculation results with the experimental data.

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