NOTATION

τ, time step; h, spatial step; U(u, v), gas velocity; $\varkappa$, thermal diffusivity; ν, kinematic viscosity; c_s, sound velocity; φ, unknown vector function; A, a nonlinear differential operator matrix; t, time; x and y, Cartesian coordinates; ρ, density; P, pressure; T, temperature; γ, adiabatic index; M, Mach number; Re, Reynolds number; Pr, Prandtl number; Ra, Rayleigh number; Nu, Nusselt number; Fr_x and Fr_y, Froude numbers; $\hat{A}$, grid operator; E, unity operator; K, Courant number; H_0 and L_0, sizes of the calculation domain; T_1 and T_2, side wall temperatures; ρ_1, gas density at the cold wall; g, gravitational acceleration; R, gas constant; and η, dynamic viscosity.

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EFFECT OF CARBON ON HEAT TRANSFER THROUGH A PISTON OF AN INTERNAL COMBUSTION ENGINE

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UDC 536.248

By analyzing and correlating the results of thermometrization of a 11D45 diesel we estimate the effect of carbon in oil-cooling channels on parameters for heat transfer through a piston.

When boosted diesel locomotives operate on group B motor oils (M-12B, M-14B, etc.), carbon deposits form on cylinder and piston components. Observations show that carbon deposits in the inner cavity of an oil-cooled piston head are particularly harmful and dangerous. Heavy carbon deposits and ineffective decrustation in the oil-cooling region shorten the useful life of pistons in 2D100, 10D100, 11D45, etc. diesels by a factor of three to four [1, 2].

It is well known that carbon deposits have an appreciable effect on heat transfer, yet in treatises on heat transfer in internal combustion engines the effect of carbon deposits is generally ignored. This naturally hampers their use in practical problems of increasing the reliability of diesel locomotives.

On the basis of analysis and correlation of the results of thermometrization of a 11D45 diesel we estimate the effect of carbon on heat transfer through a piston [3-5].

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 38, No. 3, pp. 538-542, March, 1980. Original article submitted April 19, 1979.

0022-0841/80/3803-0323$07.50

TABLE 1. Temperature of Piston Head for an Oil Temperature of 65°C

Piston head material	State of oil-cooling channel	Surface temp. of piston head, °C	
		t_h	t_c
Cast iron	No carbon	435	255
2Kh13L	No carbon	465	265
2Kh13L	0.3-mm carbon	555	250

TABLE 2. Effect of Thermal Resistance of 11D45 Diesel Piston Head on Heat Transfer

Piston head material	Carbon thickness, mm	Thermal resistance, $m^2 \cdot deg\ C/W$	Heat flux, kW/m^2	Av. resulting temp. of gases, °C	Heat-transfer coeff. $W/m^2 \cdot deg\ C$	
					α_b	α_o
Cast iron	0	$3{,}87 \cdot 10^{-4}$	465	680	1900	2550
2Kh13L	0	$4{,}91 \cdot 10^{-4}$	407	680	1900	2030
2Kh13L	0,3	$11{,}42 \cdot 10^{-4}$	267	750	1370	1440
2Kh13L	1,2	$30{,}8 \cdot 10^{-4}$	162	850	1080	1200

Dual thermocouples were embedded in the central portion of a 12-mm-thick piston head at a distance of 40 mm from the piston axis. The part of the piston head near the thermocouples can be considered a portion of a flat wall with the heat flux normal to the surface. The piston heads were made of high-strength cast iron and of a heat-resistant steel alloy (2Kh13L). Table 1 lists the temperatures of the heated t_h and oil-cooled t_c surfaces of the piston heads when the 11D45 diesel operates at nominal power (N_e = 3000 hp, n = 750 rpm) with no carbon deposit on the piston heads.

The temperature of the steel head in the experiment was somewhat higher than that of the cast iron head as a result of the difference in thermal conductivities of the materials. In the first hours of experiment with the diesel there was a tendency for the temperature of the steel head to rise higher than that of the cast iron. The temperature of the cooled side of the cast iron piston head was 10-12°C lower than that of the steel head, but this difference is important for the carbon deposition process in the oil-cooling channels of the pistons of a boosted diesel locomotive [1, 2]. For this reason carbon was deposited more rapidly on the steel than on the cast iron piston head in the first hours of operation.

The data of Table 1 were subjected to graphicoanalytic processing. The numerical values of the temperature of the heated and cooled surfaces of the piston head were transferred (on an appropriate scale) to a plotting table whose coordinate axes were thermal resistance and temperature. The quantities in Fig. 1 with a single prime refer to the cast iron head, and those with a double prime to the steel head. The width of the rectangles on the plotting table is proportional to the thermal resistance of the cast iron ($\delta_1 : \lambda_1$) and steel ($\delta_2 : \lambda_2$) heads near the dual thermocouples. The values of the temperature t_h and t_c plotted on rectangles are connected by straight lines. The slope of the straight lines is proportional to the heat flux q in the vicinity of the thermocouples [6].

The heat flux and the heat-transfer coefficients α_b from the hot gases to the piston head, and α_o from the head to the cooling oil are calculated from the Fourier equation for a flat wall:

$$q = \alpha_b (T_{av} - t_h) = \frac{\lambda}{\delta} (t_h - t_c) = \alpha_o (t_c - t_o). \tag{1}$$

The results of graphicoanalytic processing of the data of Table 1 and Fig. 1 are listed in Table 2.

Since the straight line for the cast iron piston head is steeper than that for the 2Kh13L steel head ($q_1 > q_2$), these lines intersect at the point T_{av} when prolonged (Fig. 1). This point characterizes the average equivalent temperature of gases in the 11D45 diesel cylinder with respect to heat transfer in the absence of carbon deposition on the piston surfaces. The problem of determining this temperature and the heat-transfer coefficients α_b and α_o is rather complicated. The graphicoanalytic method described above significantly simplifies finding the main characteristics of steady-state heat transfer in internal combustion engines. To do this it is necessary to perform a brief thermometrization of two or three pistons with different thermal

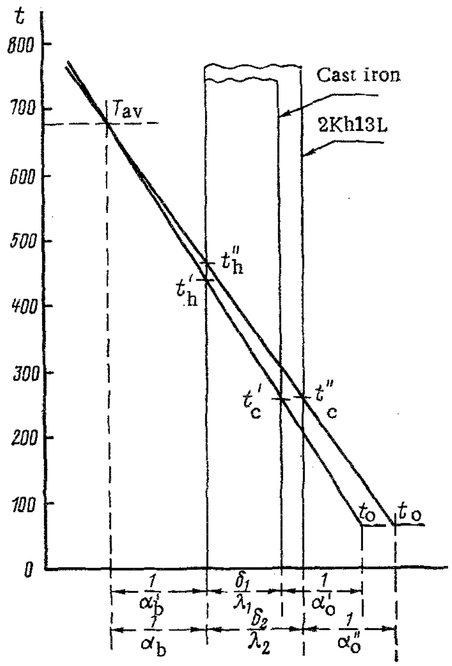

Fig. 1. Effect of thermal resistance of 11D45 diesel piston head on heat transfer.

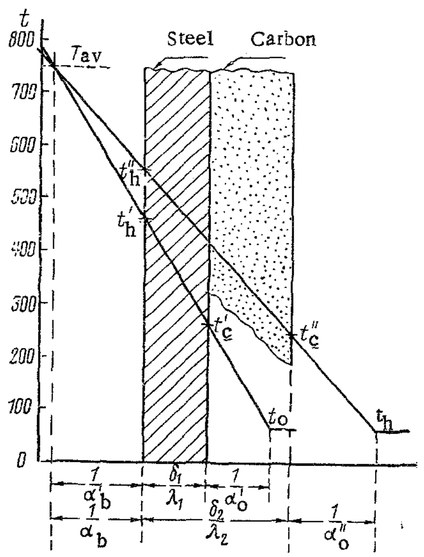

Fig. 2. Effect on heat transfer of carbon in oil-cooling channels of a 11D45 diesel piston.

resistances of the piston heads in the same engine. After this the results obtained are analyzed and correlated by the procedure discussed above.

The problem under consideration can also be solved by using a single piston with dual thermocouples. To do this the temperatures t_h and t_c are first determined for a clean piston without a carbon deposit. Then the experiment is continued until the temperature of the heated surface of the piston head increases 100-120°C as a result of carbon deposition in the oil-cooling channels.

Under the actual experimental conditions with the 11D45 diesel it was found that a 0.3-mm-thick carbon deposit on the cooled side of the steel piston head increased the temperature of the heated surface by 90°C. The thermal conductivity of the carbon deposit was $\lambda = 0.46$ W/m · deg C [1]. Consequently, the total thermal resistance of the piston head in the vicinity of the dual thermocouples was increased by $\delta : \lambda = 6.5 \cdot 10^{-4}$ m^2 · deg C/W.

The temperature of the inner surface of the piston head under the carbon layer is not included in the thermometrization readings. The temperature of the carbon surface toward the cooling oil gradually decreased during the operation of the engine. Studies [1, 2] showed that in the initial period of operation of the engine, when the inner surface was free of lacquer and carbon deposits, carbon deposition was particularly rapid when the cooling oil came in contact with the metal surface of the piston head. But as soon as the metal surface of the piston head was separated from the cooling oil by a carbon layer, the rate of carbon deposition slowed down. As the thickness of the carbon layer on the inside of the piston head increases, the temperature of the metal increases, and the temperature of the carbon on the side of the cooling oil falls, which leads to further retardation of the carbon formation process. Finally there comes a time when the carbon layer exerts so large a thermal insulating effect that carbon formation practically stops. If the diesel operates on M-14B oil, carbon formation in the oil-cooling channels stops at a temperature of 200°C, and when operating with M-14V_2 and M14Vts carbon deposition stops at 218 and 258°C, respectively [1, 2].

Observations show that when the carbon deposit in the oil-cooling channels of a 11D45 diesel piston is 0.3 mm thick the temperature of the carbon surface toward the cooling oil is 250°C instead of 265°C when there was no carbon (Table 1).

As in the first case, the values obtained for t_h' and t_c' (clean piston) and t_h'' and t_c'' (piston with carbon) were plotted on the plotting table and connected by straight lines. It is clear from Fig. 2 that the straight lines intersect at the point T_{av} = 750°C instead of 680°C for a clean piston. The results of the graphicoanalytic calculations given in Table 2 show that a 0.3-mm carbon deposit in the oil-cooling channels of a 11D45 diesel led to a decrease in the heat flux by a factor of 1.5, and to an increase in the average resulting temperature by 70°C.

The 500-600 h tests on 2D100 and 11D45 diesels using low-sulfur fuel, and oils with commercial additives, showed that the main operating parameters of the engines were not changed. On the other hand, in tests on diesels operating with high-sulfur fuel, rapid carbon deposition was observed on cylinder and piston components, the specific fuel consumption was increased by 2-3 g/hp-h (eff.), and the temperature of the exhaust gases was increased by 60-70°C. It is quite clear that in this case the average resulting temperature of the gases was increased still more. Consequently, the worsening of heat transfer because of carbon deposition on cylinder and piston components is the main reason for the decrease in heat flux through the wall of the engine combustion chamber and the increase in the average resulting temperature of the gases in the cylinder.

After 2000-2500 h of operation of the 11D45 diesel on M-14B oil, and fuel with a 0.3-0.5% sulfur content, the thickness of the carbon deposit in the oil-cooling channels reached 1.2-1.3 mm, and the thermal resistance of the piston head was six to seven times as large as that of a clean piston [1]. The heat-insulating effect of the carbon deposit increases the temperature of the piston to a value at which burnout of the heat-resistant 1Kh13L steel alloy occurs (~700°C). Simultaneously with the increase in temperature of the metal piston head there is a decrease in the temperature of the surface of the carbon deposit toward the cooling oil to 200°C [1].

If it is assumed that the temperature of the heated surface of the piston head is 700°C and that of the surface toward the cooling oil is 200°C, for a sixfold increase of the thermal resistance of the piston head the heat flux through the piston is 162 kW/m^2 (Table 2). Calculations with Eq. (1) show that in this case the heat-transfer coefficients α_b and α_o are 1080 and 1200 W/m^2·deg C, respectively. Thus, a carbon deposit in the oil-cooling channels of a piston in boosted diesel locomotives causes significant changes not only in the temperature conditions and the heat flux through the piston head, but also in the heat-transfer coefficients α_b and α_o as compared with operation with a clean piston.

NOTATION

T_{av}, average resulting temperature of gases in engine cylinder; t_h, temperature of heated surface of piston head; t_c, temperature of oil-cooled surface of piston head; t_o, temperature of cooling oil; q, heat flux through piston; δ, thickness of piston head; λ, thermal conductivity of piston head; α_b, heat-transfer coefficient from gases to head; α_o, heat-transfer coefficient from piston head to oil.

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SOLUTION OF NONAUTOMODELED PROBLEMS OF BOUNDARY-LAYER THEORY TAKING INTO ACCOUNT NONSTATIONARY CONJUGATE HEAT EXCHANGE AND BLOWING

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UDC 532.526.2

The results of an investigation of conjugate heat exchange when a supersonic flow of gas flows around a spherical shell when gas blows from the surface of the material are presented.

Theoretical and experimental investigations [1] of the effect of blowing on heat flows to the surface of bodies lead to problems of the supersonic flow of a perfect gas around a porous or perforated spherical shell. Because of the need to take into account the inertia of the heat transfer in the shell material one must solve a combined heat and mass transfer problem, since when the blowing law of the contour of the body is assigned arbitrarily, the heat-transfer coefficient will be the required function of the process and it is difficult to use a separate formulation.

As in [2-4], which are devoted to calculating conjugate heat exchange in the boundary layer, we considered a system of nonautomodeled equations of the boundary layer, we used the nonstationary equation of heat conduction for the material of the body, taking porosity into account, and on the boundary of separation of the media we used the condition of conservation of energy.

Consider the system of equations of the laminar boundary layer [4]

$$\frac{\partial}{\partial\eta}\left(l\frac{\partial^2 f}{\partial\eta^2}\right)+f\frac{\partial^2 f}{\partial\eta^2}+\beta\left[\frac{\Theta}{\Theta_e}-\left(\frac{\partial f}{\partial\eta}\right)^2\right]=\alpha\left(\frac{\partial f}{\partial\eta}\frac{\partial^2 f}{\partial\eta\partial s}-\frac{\partial^2 f}{\partial\eta^2}\frac{\partial f}{\partial s}\right), \tag{1}$$

$$\frac{\partial}{\partial\eta}\left(\frac{l}{\mathrm{Pr}}\frac{\partial\Theta}{\partial\eta}\right)+f\frac{\partial\Theta}{\partial\eta}=\beta\gamma\frac{\Theta}{\Theta_e}\frac{\partial f}{\partial\eta}-l\gamma\left(\frac{\partial^2 f}{\partial\eta^2}\right)^2+\alpha\left(\frac{\partial f}{\partial\eta}\frac{\partial\Theta}{\partial s}-\frac{\partial f}{\partial s}\frac{\partial\Theta}{\partial\eta}\right). \tag{2}$$

The equation of conservation of energy in a solid porous body, assuming the process to be one-dimensional, and that the medium is at the same temperature, has the form

$$\pi_\rho\frac{\partial\Theta_1}{\partial\tau}=\frac{\partial}{\partial y_1}\left(\pi\frac{\partial\Theta_1}{\partial y_1}\right)+\frac{\partial\Theta_1}{\partial y_1}\left[\sqrt{\mathrm{Re}}\mathrm{Pr}\frac{\lambda_{e0}}{\lambda_{1*}}(\overline{\rho v})_w\frac{1}{(1-y_1)^2}-\pi\frac{2}{(1-y_1)}\right]. \tag{3}$$

We used a natural system of coordinates when writing Eqs. (1)-(3). The coordinate y_1 for the body is directed into the material normal to the surface. We assumed that $(1-y_1)^2(\rho v)_g\varphi=(\rho v)_{gw}\varphi_w$ within the pores of the material because of our assumption of the quasistationary nature of the equation of continuity [5].

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