GRAPHICOANALYTICAL METHOD FOR CALCULATION OF HEAT TRANSFER
THROUGH THE PISTON OF AN INTERNAL COMBUSTION ENGINE

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Using a quantitative estimate of deposit formation on the pistons of a diesel engine and results of engine thermometry, a graphicoanalytical method is developed for calculation of heat transfer through an oil-cooled piston.

Heat transfer plays a uniquely important role in the processes which occur in internal combustion engines. It has a significant effect not only on engine economy, but also on reliability and lifespan. Therefore, determination of the characteristics of heat transfer from the gases to the cooling fluid is one of the important problems in the present-day science of internal combustion engines. Despite the existence of a large number of formulas for heat exchange and transfer, until the present there has been no reliable method for calculating heat transfer in internal combustion engines. Such a method is necessary for design and development of reliable, long-lived engines.

The majority of researchers have assumed that the dominant factor in heat transfer in internal combustion engines is the heat-transfer coefficient from the hot gases to the combustion chamber walls α_W. For this reason the efforts of researchers for the last 50 years have been directed to determining the magnitude of α_W as a function of construction features and operating regimes. There exist a number of equations for calculation of the instantaneous and average values of α_W: those of Nusselt, Brilling, Eichelberg, Lenin, Pflaume, Chirkov, Zitkey, Annand, Voshin, Rozenblit, Sibirkin, Central Scientific-Research Diesel Institute, et al. [1-5]. The maximum values of the coefficient α_W calculated by the various formulas may vary by a factor of four, while heat losses through the combustion chamber walls may vary by a factor of three.

Studies performed by the present author have shown that the dominant factor in heat transfer in internal combustion engines is not this quantity α_W, but rather the specific thermal flux q through the combustion chamber walls, together with the thickness and thermal conductivity of these walls. This conclusion was reached by generalizing the results of a thermometric study of Soviet-made and foreign diesel engines and many years of observation of locomotive diesel engines under actual railway use conditions [6].

The essence of this approach to solution of the heat-transfer problem is a quantitative consideration of the influence on heat-transfer parameters of the screening effect of the combustion chamber walls (piston, cylinder sleeve, head, etc.). The gas temperature in the engine cylinder, T_g, changes sharply over the course of the operating cycle. The temperature of the combustion chamber walls heated by the gas, t_h, also oscillates, but not as much. Because of the screening effect, the temperature of the cooled surfaces of the chamber, t_c, remains stable.

Years of observations of types 2D100, 10D100, and 11D45 locomotive diesels (as well as others) have shown that the piston bottom has the largest screening effect as compared to the other combustion chamber walls. This is true because the thermal resistance of the piston bottom increases many times as a result of carbon deposit formation in the oil-cooling channels, while the thermal resistance of the cylinder sleeve and head remains practically constant even after lengthy operation.

The thickness of an oil-cooled piston bottom is an order of magnitude less than its diameter. This permits us to consider heat transfer through the piston as heat transfer through a plane wall without introducing large errors.

In the present study, heat transfer will be considered using a physical model composed of two chambers separated by a plane screen (Fig. 1). The engine piston represents a movable

screen, separating the hot gases (with variable temperature T_g) from the cooling oil (with constant temperature T_0). Virtual or actual change in the thickness or thermal conductivity of the piston leads to a change in the screening effect.

Thus, with an insignificant increase in thermal resistance of the piston (not more than 50%) the temperature of the heated side increases ($t''_h > t'_h$), while the temperature of the cooled side remains practically the same ($t''_c \approx t'_c$). The specific thermal flux then decreases ($q'' < q'$) (Fig. 2). With a significant increase in thermal resistance (by a factor of several times), e.g., because of deposit formation, not only the temperature of the heated surface and the thermal flux, but also the temperature of the cooled side of the piston change significantly. These phenomena have actually been observed during thermometry of a cast iron and steel piston with and without deposits in an 11D45 diesel engine.

The slope of the straight lines joining the points T_m and t_0 in Fig. 2 can be determined from the condition

$$\alpha_w(T_m - t_h) = \frac{\lambda}{\delta}(t_h - t_c) = \alpha_0(t_c - t_0) = q. \tag{1}$$

Physically, Eq. (1) implies that in a stationary heat-transfer process the specific thermal flux through the piston q is not discontinuous on the phase boundaries (gas–piston, piston–oil). The boundary conditions specified in calculating heat transfer through the piston must statisfy this relationship.

The specific thermal flux through the piston (local or mean) may be determined from Eq. (1). However, to do this it is necessary to make temperature measurements in a real engine. In the design procedure when the concrete engine does not yet exist, this is naturally impossible.

Analysis of data obtained by thermometry of locomotive and marine diesel engine components has established the dependence of specific thermal flux through the piston on engine design parameters [6], producing an expression of the form

$$q = c \frac{P_e C_m T_i g_e}{Z P_i}\left(\frac{\lambda}{\delta}\right)^{0.5}. \tag{2}$$

Equation (2) permits determination of the thermal flux through the piston with satisfactory accuracy even in the design stage, and thus permits analysis of thermal loading on the engine with variation of loading parameters. For railway and marine diesels with oil-cooled pistons the coefficient c lies within the interval $(2.4-2.6)\cdot10^7$. The mean value of this coefficient for 17 types of railway and marine diesels is $2.45\cdot10^7$ $kg^{0.5}\cdot m/deg^{0.5}\cdot sec^{2.5}$. It follows from Eq. (2) that the local thermal flux through some portion of the piston is directly proportional to the engine supercharging coefficient and proportional to the thermal conductivity of that part of the piston raised to the 0.5 power. In order to calculate the mean specific thermal flux through the entire piston surface, it is necessary to introduce a reduced piston thickness in Eq. (2), equivalent in heat transfer to the thickness of a plane wall.

The mean specific thermal flux through the piston and the fraction of total heat removed through the piston into the cooling oil are characterized by the data of Table 1.

When type B motor oils which have insufficient antideposit effectiveness are used, intense deposit formation occurs in the oil-cooling channels of railway diesel pistons. When diesel fuel with a sulfur content of 0.3-0.5% (by weight) and lubricating oils M-12B and M-14B with 8% of the additive VNII NP-360 are used, the change in thermal resistance of the piston due to deposit formation in the oil-cooling channels of pistons in 2D100, 10D100, and 11D45 diesel engines is characterized by the data of Table 2.

The causes of variation in the amount of deposit formation in the piston oil-cooling zone have been studied by the author, with results published in [6, 7].

The screening effect of the deposit depends not only on its thickness, but also on its thermal conductivity. The thermal conductivity of deposits found in laboratory, teststand, and in-use experiments [6] was $\lambda = 0.46$ W/m·deg. It is evident from Table 2 that after 2000 h of operation the thermal resistance of 2D100, 10D100, and 11D45 pistons increases by factors of 1.9, 4.8, and 6.3, respectively. Such a sharp increase in thermal resistance leads to a significant change in all heat-transfer characteristics through the piston.
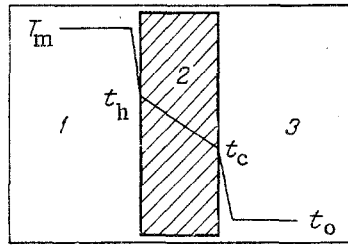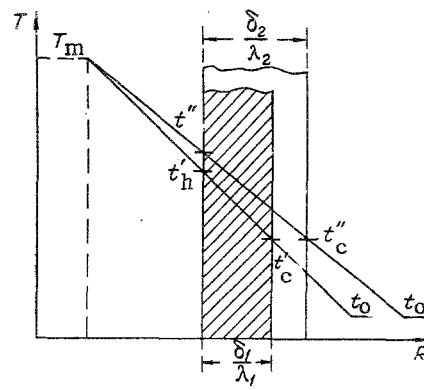
Fig. 1.   Fig. 2.

Fig. 1. Physical model of heat transfer through piston: 1) com-
bustion chamber; 2) screen; 3) cooling chamber.

Fig. 2. Diagram of piston screening effect.

TABLE 1. Specific Thermal Flux through Piston and Fraction
of Total Heat Removed through Piston into Cooling Oil

Engine type	Piston pow-er, kW	Specific fuel consumption, kg/J	Piston area, m²	Specific thermal flux q, kW/m²	Heat lost through piston, %
2D100	73,6	$6,43 \cdot 10^{-8}$	0,034	476	7,9
10D100	110	$6,41 \cdot 10^{-8}$	0,034	540	6,1
11D45	138	$6,41 \cdot 10^{-8}$	0,042	413	4,6
5D49	138	$5,85 \cdot 10^{-8}$	0,053	233	3,4
16ChN26/26	184	$5,85 \cdot 10^{-8}$	0,053	255	2,9
20ChN26/26	220	$6,04 \cdot 10^{-8}$	0,053	293	2,8
8 DKRN 74/160	1100	$5,92 \cdot 10^{-8}$	0,429	177	2,7

TABLE 2. Change in Thermal Resistance of Piston Bottom due
to Deposit Formation in Oil-Cooling Channels in Operation
with M-12B and M-14B Motor Oils

Elapsed operating time, h	2D100		10D100		11D45	
	deposit for-mation, mm	$\frac{\delta}{\lambda} \cdot 10^4$	deposit for-mation, mm	$\frac{\delta}{\lambda} \cdot 10^4$	deposit for-mation, mm	$\frac{\delta}{\lambda} \cdot 10^4$
5	0	3,6	0	2,2	0	4,9
50	0	3,6	0	2,2	0,10	7,1
200	0,02	4,0	0,10	4,3	0,34	12,3
500	0,04	4,5	0,20	6,5	0,68	19,1
1000	0,07	5,2	0,30	8,7	1,00	26,6
2000	0,13	6,7	0,39	10,7	1,20	31,0
5000	0,27	10,0	0,78	19,2	1,26	32,3

The screening effect of deposits on the piston surface is especially strong in 11D45
engines using M-14B oil. The effect of deposit formation on heat transfer through the piston
in this engine is shown in Fig. 3. The piston bottom is shown here as a planar cross-hatched
wall, the thickness of which is proportional to the thermal resistance of the clean piston
(Table 2).

The pistons for the 11D45 engine are constructed of a high-temperature steel alloy 2Kh13I
with a thermal conductivity of λ = 24.3 W/m·deg and mean thickness δ = 12 mm. The forcing
coefficient, i.e., the rated operating load coefficient for the 11D45 engine at nominal power
(N_e = 3000 hp, n = 750 rpm), is equal to

$$\Pi_{1c} = \frac{2.45 \cdot 10^7 P_e C_m T_i g_e}{Z P_i} = 9210\,\text{kg}^{0.5} \cdot \text{deg}^{0.5}/\text{sec}^{0.5}.$$

The specific thermal flux through the piston will be

$$q = \Pi_{1c}\sqrt{\lambda/\delta} = 9210\sqrt{24.3 : 0.012} = 413\ \text{kW/m}^2.$$

Thermometry has shown that the temperature of the inner (oil-cooled) piston surface at nominal engine power is $t_c = 265°C$ [7]. Then the temperature of the hot side of the piston is

$$t_h = t_c + q\frac{\delta}{\lambda} = 265 + 413 \cdot 10^3\,\frac{0.012}{24.3} = 467\,°C.$$

The temperature values thus found are entered on the cross-hatched rectangle in the proper scale. The temperature t_h is marked on the left side, t_c on the right. The two points are joined by a straight line.

After 40-50 h of operation of an 11D45 diesel engine with M-14B lubricating oil and fuel with a 0.3-0.5% sulfur content the thermal resistance of the piston increases due to deposit formation from $4.9 \cdot 10^{-4}$ to $7.1 \cdot 10^{-4}$ m²•deg/W (Table 2), while the thermal flux decreases to q = $9210\sqrt{10^4 : 7.1}$ = 345 kW/m².

It was shown in [6, 7] that with increase in deposit thickness on the inner surface of the piston, the metal temperature rises, and the temperature of the surface of the deposit facing the cooling oil falls. With deposit thicknesses of 0.1 and 0.3 mm on the piston of an 11D45 engine, the surface temperature of the deposit on the cool side is 260 and 250°, respectively. With a deposit thickness of 0.1 mm the temperature of the hot piston surface increases to t_h = 260 + 345•10³•7.1•10⁻⁴ = 503°C. These temperature values may be plotted on the rectangle like the previous ones, with the rectangle width now corresponding to the increased thermal resistance ($\delta : \lambda = 7.1 \cdot 10^{-4}$). The points t_h and t_c are again connected by a straight line. Since the slope of the first line is greater than that of the second ($q_1 >$ q_2), when the lines are extended they intersect at the point T_m^{min} (Fig. 3). This point characterizes a mean temperature in the 11D45 cylinder, equivalent with respect to heat transfer, in the presence of insignificant deposit buildup on the outer surface of the piston.

With further increase in deposit thickness the piston screening effect becomes more and more intense. After 2000 h of engine operation the deposit thickness in the oil-cooling channels reaches 1.2 mm in thickness (Table 2), the temperature of the hot side of the piston increases to 713°C, and the deposit temperature on the side facing the cooling oil decreases to 210°C [6-8]. The specific thermal flux through the piston decreases to 163 kW/m².

With each increase in thermal resistance of the piston, the point of intersection of the T_m lines moves upward and to the left to a higher value. At a deposit thickness of $\delta = 1.2$ mm the mean resultant gas temperature reaches $T_m = T_m^{max}$. It is evident from Fig. 3 that for a clean piston $T_m^{min} = 700°C$, while for a piston contaminated by 1.2-mm deposits $T_m^{min} = 880°C$.

Tests of diesel engines with high-sulfur fuels have revealed intense deposit formation on cylinder parts, leading to an increase in specific fuel consumption of 2-3 g/hp•h and increase in exhaust gas temperature by 60-70°C. Substitution of numerical values for the specific thermal flux and T_m, t_h, t_c, and t_o = 65°C in Eq. (1) gives α_w' = 1910 W/m²•deg and α_o' = 2050 W/m²•deg for the clean piston, while for a 1.2-mm-thick deposit, α_w'' = 1040 W/m²•deg and α_o'' = 1120 W/m²•deg. The overall heat-transfer coefficients for the clean piston is 670 W/m²• deg, and for the dirty piston, 200 W/m²•deg.

Thus, for intense contamination of the oil-cooling channels by deposits the thermal flux decreases by a factor of 2.5, the heat-transfer coefficients α_w and α_o decrease by a factor of 1.8, and the overall heat-transfer coefficient decreases by a factor of 3. The mean resultant gas temperature in the 11D45 cylinder increases by 170-180°C, and the mean temperature of the piston metal t_m increases from 366 to 675°C. It follows from these data that α_w and α_o are not constant, but variable quantities, which can only be calculated accurately with Eq. (1). Moreover, the quantities α_w and α_o for diesels with oil-cooled pistons are of the same order of magnitude, and their roles in heat transfer are practically equal in importance.

The calculation of heat transfer through the piston may be done in a similar manner for 2D100, 10D100, 5D49, and other engines with oil-cooled pistons.
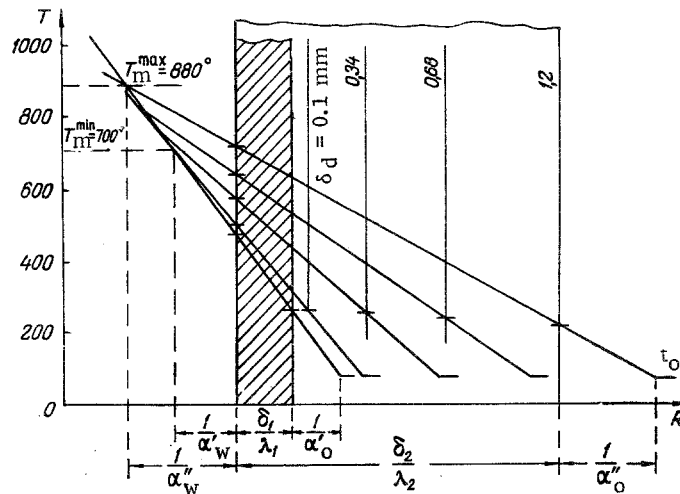
Fig. 3. Effect of deposits in oil-cooling channels on heat transfer through piston of 11D45 diesel engine. δ_d, deposit thickness.

A quantitative estimate of the piston screening effect shows that increase in thermal resistance of the piston leads to an increase in the overall thermal resistance to heat transfer R,

$$R = \frac{1}{\alpha_w} + \frac{\delta}{\lambda} + \frac{1}{\alpha_o}. \tag{3}$$

Calculations with Eqs. (1)-(3) and the graphical construction for railway and marine diesels (as in Fig. 3) have shown that the total thermal resistance R is a linear function of the thermal resistance of the piston bottom:

$$R = a + b\frac{\delta}{\lambda}. \tag{4}$$

For two-cycle engines $a = 8.6 \cdot 10^{-4}$ m$^2 \cdot$deg/W and b + 1.16. For four-cycle engines $a = 12.9 \cdot 10^{-4}$ m$^2 \cdot$deg/W and b = 1.25.

Equation (4) permits calculation of the total thermal resistance to heat transfer through the piston for real or virtual change in thermal resistance of the piston.

It follows from Eqs. (1) and (4) that

$$T_m = t_o + q\left(a + b\frac{\delta}{\lambda}\right). \tag{5}$$

The values of α_w and α_o are calculated with Eq. (1).

To calculate T_m and t_h with Eqs. (1) and (5) a single boundary condition is needed. This is the temperature of the cooled surface, t_c.

Operational tests of railway diesel engines have shown that reliable long-term piston operation is possible only in the absence of deposit formation in the piston oil-cooling channels [6, 8]. On the basis of this deposit-free operation, the temperature of the cooled piston surface must be no higher than 200°C with type B lubricating oils, and not more than 220°C with type V oils (M-14V₂) [6].

Thus, in contrast to all existing methods of heat-transfer calculation for internal combustion engines, which are based on analysis of indicator diagrams and determination of the gas-to-wall heat-transfer coefficient, we have proposed a new method of calculation, based on the functional dependence of specific thermal flux through the piston upon engine ratings and the time-varying thermal resistance of the piston. The graphical method of calculating heat transfer through the piston permits determination of all heat-transfer characteristics even in the design stage: specific thermal flux, mean gas temperature in the cylinder, temperature of hot piston surface, etc.

NOTATION

T_m, mean resultant gas temperature in engine cylinder; t_h, temperature of heated side of piston; t_c, temperature of cooled piston surface; t_o, temperature of cooling oil; q, specific thermal flux through piston; P_e, mean effective engine pressure; C_m, mean piston speed; T_i, injected air temperature, °K; P_i, injected air pressure; g_e, effective fuel flow rate; δ, mean thickness of piston bottom; λ, thermal-conductivity coefficient of piston bottom; Z, number of strokes in engine cycle; α_w, gas-to-wall heat-transfer coefficient; α_o, wall-to-oil heat-transfer coefficient; R, total thermal resistance.

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