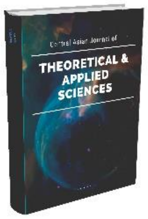

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Calculation of Thermal State of Sleeves and Cylinder Covers

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Abstract: *This article provides an overview of the engineering choice and improvement of urban areas, taking into account emergency situations.*

Keywords: *city; general plan; release; groundwater level, climate zone.*

The bushing and cover are thermally stressed body parts, heated mainly by hot gases and having the same type of cooling. Therefore, despite the difference in geometric shape and some significant features in the work, it is advisable to consider the calculation of the temperature of these parts together.

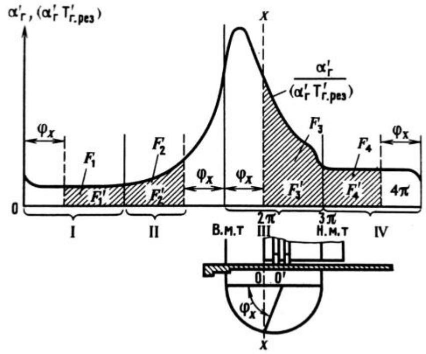
As in the calculation of the piston, the temperature field in this case is practically stationary. The heat transfer boundary conditions for the bushing and cylinder head are selected in the same way as for the piston.

The calculation is performed primarily for the steady-state operation of the engine. It consists in determining the stationary temperature field at various points of the structure at a given operating mode of the engine.

The boundary conditions of heat transfer from the gas side on the upper belt of the sleeve (up to the edge of the piston at its position in the CMT) and on the fire bottom of the cylinder cover are expressed by the dependencies given in the textbook "Internal combustion engines. Theory of working processes of reciprocating and combined engines". Heat transfer coefficient $\alpha_{Г.З}$ and the average resulting temperature $T_{Г.З}$ in the gap between the sleeve and the piston top land are approximately equal $\alpha_{Г.З} = 0,25\alpha_1$; $T_{Г.З} = T_{Г.РЕЗ}$

Heat is transferred to the working surface of the sleeve directly from hot gases, as well as from the piston (mainly through the rings). A significant part of the heat is the heat released during the friction of the rings, as well as during the friction of the piston body.

Surface points below the position of the first compression ring at v. m. t., are exposed to hot gases only during a part of the individual strokes of the engine operation, when these points are not covered by the piston (Fig. 313). On coal φ_x turning shaft belt $x-x$ The bushing is insulated from heat transfer from the gas side. Then the heat transfer coefficient from the gas side $\alpha_T^{'}$ and the average resulting temperature $T^{'}_{Г.РЕЗ}$



$$\alpha_T^{'} = \sum_{i=1}^{4} F_i / (4a); \qquad T_{Г.РЕЗ}^{'} = \sum_{i=1}^{4} F_i^{'} / \alpha_T,$$

where F_i, $F_i^{'}$ - area, respectively, under the curves α_T and α_T $T^{'}_{Г.РЕЗ}$ (see fig. 313); α - length on the abscissa axis in the drawing scale corresponding to a similar time step.

For two-stroke engines, only the areas are determined F_2, $F_2^{'}$, F_3, $F_3^{'}$. Changing angles φ_x from 0 to π with 10-20°, define $\alpha_T^{'}$ and $T_{Г.РЕЗ}{}^{'}$ along the working length of the sleeve. The values obtained should be corrected for each surface area, taking into account the additional supply of heat through the rings, as well as due to heat exchange between the bushing and charge air in the sub-piston cavity in the case of crosshead engines. The frictional heat removed through the bushing into the coolant is 20-40% of the total heat received by the bushing. Correction is carried out in accordance with the diagram shown in Fig. 313, change and $T^{'}_{Г.РЕЗ}$ for every angle $x-x$.

Valves with a high temperature and a significant heat transfer coefficient between the chamfer and the seating surface transfer a significant heat flux to the cover.

When determining the stationary temperature field of the bottom of the lid on the inner contours, corresponding to the holes for the valves, you should set the average values characterizing the heat exchange as a whole for the entire cycle - the average heat flux q_{Lj}, the corresponding reduced heat transfer coefficient and the resulting temperature of the valve $T_{клLj}$ by heat transfer:

$$q_{Lj} = \frac{1}{\theta}\left[\int_{\varphi o} \alpha_{Г.С}(T_C - T_{Г.С})\partial\varphi + \overline{F}\int_{\varphi 3} \alpha_K (T_C - T_{ф.кл}) d\varphi\right];$$

$$\alpha_{Lj} = \frac{1}{\theta}\left(\int_{\varphi o} \alpha_{Г.С} d\varphi + \overline{F}\int_{\varphi 3} \alpha_K d\varphi\right)$$

$$T_{клLj} = \left(\int_{\varphi 0} \alpha_{Т.С} d\varphi + \overline{F}\int_{\varphi 3} \alpha_K T_{ф.кл} d\varphi\right) / \left(\int_{\varphi 0} \alpha_{Т.С} d\varphi + \overline{F}\int_{\varphi 3} \alpha_K d\varphi\right),$$

where $\alpha_{Т.С,}$ α_K - heat transfer coefficients respectively for the seat with an open valve and with a closed valve in the case of contact heat exchange (different for inlet and outlet valves, see § 5 Ch. 8); φ_0, $\varphi_{3,}$ θ - respectively, the duration of the opening and closing of the valve and the duration of the cycle in the angles of rotation of the crankshaft; T_C, $T_{Г.С}$, $T_{ф.кл}$, - temperatures, respectively, of the seat, the washing medium (gas or air) and the chamfer of the valve; $\overline{F}$ - the ratio of the contact areas of the valve (support band) and the side surface of the bottom on the contour $L_{j.}$

The corresponding reduction can be performed on the average temperature of the washing circuit $L_{j.}$ sides.

Along with the heat transfer coefficient and the average resulting gas temperature, the specific heat flux is often used as the boundary conditions q_0. The average heat flux within the part can be determined in the first approximation by the formula $q = K_T K (c_m p_к)^{0,57} / \alpha$,

where $K_T = 4,27$ for four-stroke and $K_T = 7,6$ for two-stroke diesel engines; $K = 132 \cdot 10^3$ for heads and $K = 39 \cdot 10^3$ for cylinder liners; C_m - average piston speed; $p_к$ - boost pressure, MPa; α - excess air ratio.

Having calculated q the distribution of the flux over the surface should be estimated, for example, depending on the radius for the bottom of the lid, i.e., the local values of the heat flux should be determined q_0, which are used in the calculations.

A well-grounded choice of boundary conditions on the cooling side is also important. At moderate levels of boost, when the temperature of the surface of the bushing and cover is washed by the liquid is less than the saturation temperature T_s coolant, to calculate the heat transfer coefficient, the formula was used

$$\alpha_{ж} = 1,163(300 + 1800\sqrt{W_{ж}}),$$

where $\alpha_{ж} - в \ Bm/(м^2 \cdot {}^oC)$; $W_{ж}$ - the speed of movement of the coolant.

Cooling systems of modern engines are characterized by higher values $\alpha_{ж}$, than those obtained by formula (435). Cooling of bushings and cylinder heads can take place in forced convection and surface boiling modes.

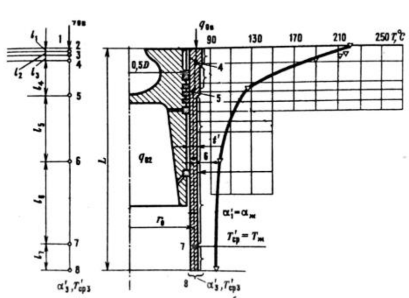
Intensification of heat transfer due to vibration of the sleeve during piston shifting is taken into account by introducing an amplitude velocity $W_{виб}$ vibration bushing:

$$\alpha_{ж} = A W_{ж}^{0,41} W_{виб}^{0,23} / (\rho^{0,32} d_э^{0,36}),$$

where $A = 39,31$ for press-on bushings; $A = 71,5$ for bushings pressed into the cylinder block; ρ — fluid density; $d_э$ - equivalent diameter, $d_э = (d_2 - d_1)$; $d_{1,2}$ - respectively, the outer diameter of the sleeve and the inner diameter of the cooling jacket; $W_{виб} = 8,34 \cdot 10^{-10}(2n/i)^3 + 7,25 \cdot 10^{-6}(2n/i)^2 - 1,385 \cdot 10^{-3}(2n/i) + 54 \cdot 10^{-2}$, $2n/i$ - the number of working strokes per unit of time.

Empirical dependence to determine $W_{виб}$ is valid for bushings with a diameter to wall thickness ratio of 14-17. The influence of the flow around the sleeve on the heat transfer associated with the coolant supply scheme is taken into account by the following relationship:

$$Nu_{ж} = C \, \mathrm{Re}_{ж}^{0,6} \, \mathrm{Pr}_{ж}^{0.43} \left(\frac{\mathrm{Pr}_{Ж}}{\mathrm{Pr}_{CT}} \right)^{0,25} \left(\frac{d_2}{d_1} - 1\right)\left(1 + 1.8\frac{1}{d_1}\right),$$



Where $Nu_{ж}$ Nusselt number $Nu_{ж} = \alpha_{ж} d_{экв} / \lambda_{ж}$; $\mathrm{Pr}_{ж}$ - Prandtl number, $\mathrm{Pr}_{ж} = \nu_{ж} / a_{ж}$; $\in \lambda_{ж}$, $a_{ж}$, $\nu_{ж}$ - coefficients, respectively, of thermal conductivity, thermal diffusivity and kinematic viscosity of the coolant; l - cooling channel length; $C = 1,03$ with longitudinal flow, $C = 0,72$ at diagonal, $C = 1,34$ with uniform flow; the subscripts w and st refer to the flow parameters calculated at the average temperature and at the wall temperature, respectively. Equation (437) is valid for

$1500 \le \mathrm{Re}_{ж} \le 15000;\ \ 1,5Pr_{ж} \le 20;\ 1,15 \le (d_2 / d_1) \prec 2;\ 1,0 \prec l / d_1 \le 10.$

At surface boiling, approximately for the sleeve

$$Nu_{ж} = 1,2 \cdot 10^{-2} K_{ф}^{0,4} (p_e^{"})^{0,7} (d_1 / d_э)^{-0,35} (p_{ж} / p_o)^{0.5}\, \mathrm{Pr}_{ж};$$

for cylinder cover

$$Nu_{ж} = 0,45 \cdot 10^{-2} (p_e^{"})^{0,7} K_{ф}^{0,4}\, \mathrm{Pr}_{ж}^{0,3} (p_{ж} / p_o)^{0.5},$$

Where $p_e^{"} = q l_o / (r p^{"} \alpha_{ж})$; $K_{ф}$ - phase transformation criterion, $K_{ф} = r / (c \Delta T_s)$; l_o - characteristic size, $l_o = \sqrt{\sigma / (\rho^{/} - \rho^{"})}$; -коэффициент поверхностного натяжения; r -скрытая теплота парообразования; ΔT_s - the difference between the saturation and coolant temperatures; $p_{ж,}\ p_o$ - pressure, respectively, in the cooling circuit and atmospheric; $p^{/}$ and $p^{"}$ - density of liquid and vapor, respectively

The values $a_{ж}$, obtained from the given dependencies should be considered as indicative. They can serve as a first approximation for specifying local boundary conditions when calculating the temperature field. The latter are prescribed taking into account the results of numerous calculations, and most importantly, according to experimental data obtained on running engines. Due to the limited nature of these data, it is important to solve inverse problems of heat conduction, which allows, based on the temperature field of the part, to obtain an adequate distribution of heat transfer parameters over its surface.

With an increase in forcing, the temperature of the cooled surface of the bushing and cover can exceed the saturation temperature of the coolant. In this case, surface boiling begins in the liquid layers adjacent to the surface and the heat transfer rate increases sharply.

Calculation of the thermal state of the cylinder liners of four-stroke engines, as well as two-stroke engines with a valve-slotted gas exchange scheme, is carried out in an axisymmetric setting *(Fig. 314)*. Taking the temperature distribution over the bushing thickness as a quadratic, i.e. $T = T_o^{/} + T_1^{/} \vec{r} + T_1^{/} \vec{r}^{-2}$, (where $\vec{r} = r - r_o$) the problem of calculating the temperature field of the sleeve can be reduced to determining the temperature $T_o^{/}$ the middle surface of the sleeve at the radius r_o, corresponding to the line 1-2-3-4-5-6 of the wall. In this case, the main differential equation of the problem, obtained using a relation of the type (16), coincides with the differential equation of heat conduction (179). When determining the coefficients $f_1^{/}, f_2^{/}$ use the same formulas as when calculating the piston body, changing the designation in accordance with Fig. 314 and 100. The x coordinate along the length of the sleeve is measured from the point of the top end of the sleeve. When solving equation (179), boundary conditions are added at the ends of the sleeve. For example, $q_{ов} = \lambda \partial / \partial x$ at $x = 0$; $\lambda \partial T / \partial x = \alpha_3^{/} (T_o^{/} - T_{ср3}^{/})$ at $x = L$.

With the length-averaged heat transfer parameters and, accordingly, the coefficients $f_{1ср}^{/}\ u\ f_{2ср}^{/}$ the solution to equation (179) has the form

$$T_0^{/} = C_{1e}^{\ x\sqrt{-f_{2ср}^{/}/\lambda}} + C_{2e}^{\ x\sqrt{-f_{2ср}^{/}/\lambda}} + \vartheta,$$

Where $\vartheta = -f_{1ср}^{/} / f_{2ср}^{/}$.

However, such a solution can be considered only as an approximate one, since the conditions of heat transfer along the length of the working surface of the sleeve can vary significantly.

As a result, it is advisable to solve the problem by numerical methods, in particular, using the FEM, using two-node one-dimensional elements (see Fig. 19, a). The functional to be minimized coincides with functional (180) with the notation corresponding to Fig. 314 and 100.

As an example, consider the problem of determining the temperatures in the cylinder liner of a diesel engine at $D = 120\,mm$. A finite element model of the bushing is shown in Fig. 314.6. Seven one-dimensional finite elements were selected at eight anchor points. When performing design calculations, the number of nodal points and, accordingly, elements must be increased. The heat flux is set as the heat transfer conditions on the inner side of the sleeve д02, and on the outer, cooled side, the heat transfer coefficient $\alpha_1^{/} = \alpha_{ж}$ and temperature $T_{cp1} = T_{ж}$. Average specific total heat flux into the bushing $q_{o\Sigma}$ includes — $0{,}583 \cdot 10^5 B_T / m^2$. At the top, it is accepted at the bottom. The calculation was carried out for a bushing made of gray cast iron.

The parameters necessary for the calculation, related to the selected elements of the discrete model of the bushing, are given in table. 25. Expressions for the contributions of individual elements to the overall functional, as well as the dependence of partial derivatives on the functionals of individual elements in terms of temperatures $T_{0i}^{/}$ their nodes are obtained by adding to dependencies (181) and (182) a term with a factor $f_2^{/}$ which in the problem under consideration is nonzero. Let us denote by $\Delta\Phi_e(T_o^{/})$ terms to be added to the expression for $\Phi_e(T_o^{/})$, determined by formula (181). For individual elements (from 1st to 7th), these additional terms will have the form

$$\Delta\Phi_1(T_o^{/}) = -\int_{l_1} 0{,}5 f_{(2)1} F_1 (N_1 T_{01} + N_2 T_{02})^2\, dx;$$

$$\Delta\Phi_2(T_o^{/}) = -\int_{l_1} 0{,}5 f_{(2)2} F_2 (N_2 T_{02} + N_3 T_{03})^2\, dx;$$

$$\Delta\Phi_3(T_o^{/}) = -\int_{l_3} 0{,}5 f_{(2)3} F_3 (N_3 T_{03} + N_4 T_{04})^2\, dx;$$

$$\Delta\Phi_4(T_o^{/}) = -\int_{l_4} 0{,}5 f_{(2)4} F_4 (N_4 T_{04} + N_5 T_{05})^2\, dx;$$

$$\Delta\Phi_5(T_o^{/}) = -\int_{l_5} 0{,}5 f_{(2)5} F_5 (N_5 T_{05} + N_6 T_{06})^2\, dx;$$

$$\Delta\Phi_6(T_o^{/}) = -\int_{l_6} 0{,}5 f_{(2)6} F_6 (N_6 T_{06} + N_7 T_{07})^2\, dx;$$

$$\Delta\Phi_7(T_o^{/}) = -\int_{l_7} 0{,}5 f_{(2)7} F_7 (N_7 T_{07} + N_8 T_{08})^2\, dx,$$

where the area of element 4 is constant along its length. The contribution of individual elements to the general system of equations is determined by formulas (182) with the addition of the terms $\partial(\Delta\Phi_e)/\partial T_{0i}^{/}$, , equal to partial derivatives of $\Delta\Phi_e$ by temperature $T_{0i}^{/}$ nodes of the corresponding elements:

element 1

$\partial(\Delta\Phi_1)/\partial T_{01}^{/} = -\Delta C_1(T_{01}^{/}/3 + T_{02}^{/}/6);$

$\partial(\Delta\Phi_1)/\partial T_{02}^{/} = -\Delta C_1(T_{01}^{/}/3 + T_{02}^{/}/3);$

element 2

$\partial(\Delta\Phi_2)/\partial T_{02}^{/} = -\Delta C_2(T_{02}^{/}/3 + T_{03}^{/}/6);$

$\partial(\Delta\Phi_2)/\partial T_{03}^{/} = -\Delta C_2(T_{02}^{/}/6 + T_{03}^{/}/3);$

element 3

$\partial(\Delta\Phi_3)/\partial T_{03}^{/} = -\Delta C_3(T_{03}^{/}/3 + T_{04}^{/}/6);$

$\partial(\Delta\Phi_3)/\partial T_{04}^{/} = -\Delta C_3(T_{03}^{/}/6 + T_{04}^{/}/3);$

element 4

$\partial(\Delta\Phi_4)/\partial T_{04}^{/} = -\Delta C_4(T_{04}^{/}/3 + T_{05}^{/}/6);$

$\partial(\Delta\Phi_4)/\partial T_{05}^{/} = -\Delta C_4(T_{04}^{/}/6 + T_{05}^{/}/3);$

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element 7

$\partial(\Delta\Phi_7)/\partial T_{07}^{/} = -\Delta C_7(T_{07}^{/}/3 + T_{08}^{/}/6);$

$\partial(\Delta\Phi_7)/\partial T_{08}^{/} = -\Delta C_7(T_{07}^{/}/6 + T_{08}^{/}/3),$

Having the terms of formula (182), taking into account the addition of expressions (442), we form the global matrix of thermal conductivity (H) and the vector of the right-hand side $\{f\}$ general system of equations (183):

Vectors $\{T_0^{/}\}$ and $\{f\}$ have the same form as in expression (183), but for the values $q_{0в}$, $f_{(1)e}$, l_e, F_e, corresponding to the case under consideration (see Fig. 314).

Substituting the numerical values of the coefficients C_i and calculating the components of the vector $\{f\}$, we find, by solving system (443), the temperatures of the nodal points of the elements: $T_{01}^{/} = 220\,^oC$, $T_{02}^{/} = 219{,}6\,^oC$, $T_{03}^{/} = 215\,^oC$, $T_{04}^{/} = 190{,}8\,^oC$, $T_{05}^{/} = 126\,^oC$, $T_{06}^{/} = 98\,^oC$, $T_{07}^{/} = 106{,}5\,^oC$, $T_{08}^{/} = 97\,^oC.$. In fig. 314 shows the temperature distribution of the sleeve over the middle radius. 8600130924410865

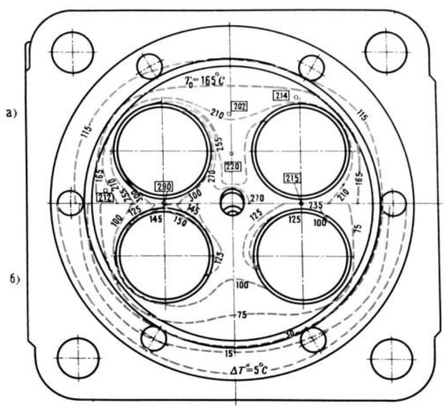
When calculating the thermal state of the covers (heads) of cylinders, the determination of the temperature field of the bottom is of paramount importance. The calculation of the thermal state of the bottom is reduced to the calculation of a plate of an arbitrary contour with a system of holes for given (see Fig. 18) heat transfer conditions on the plate surfaces. A reasonable choice of heat transfer conditions is a difficult task. So, when calculating the thermal state of the cover, knowledge of the temperature state of the valves and the sleeve, which is in contact with the cover directly or through a gasket, is required.

When calculating valves and bushings, in turn, you need to know the thermal state of the cover.

This problem is solved by creating a design model (for example, a complete finite element model) of the entire node.

However, the cumbersomeness of the model and the excessive amount of initial information do not allow until now to widely use this approach in solving the problem of heat conduction in relation to the parts of the cylinder-piston group, therefore, an isolated model of the cylinder cover is considered. The complexity of the design and the high tension of the cylinder heads of diesel engines contributed to the widespread use of flat fire bottom designs in engines of various classes and purposes.

Taking the temperature distribution over the bottom thickness to be quadratic, the problem of calculating its temperature field is reduced to determining the temperature T_o the middle surface. In this case, the main differential equation of the problem, obtained using relation (16), and the corresponding functional coincide with equation (18) for the stationary case and with functional (20).



α - temperature distribution T_o in the median plane; δ - drop $\Delta T''$ by bottom thickness (within the framework of the experiment results are given)

If we represent the bottom of the lid as a set of elements, for example, of a triangular type (Fig. 315), then the task of calculating the thermal state of the bottom will be to find the temperatures of the nodal points T_{oi} the middle surface. Taking into account the locality of heat transfer conditions on the bottom surfaces is carried out by setting the boundary conditions for heat transfer in the zones into which its surfaces are divided, including the side surfaces of the holes for the valves. For example, when calculating the temperature field of the cylinder head of an engine of the CP 26/26 type, using the symmetry assumptions (Fig. 315), 24 zones are distinguished, within which the heat transfer conditions are taken unchanged (Table 26).

In fig. 316 presents the results of calculating the stationary temperature field in the median plane and the temperature difference $\Delta T''$ on the thickness of the bottom, many of the options for the engine cover of the ChP 26/26 type, corresponding to the operating mode $p_e = 1{,}5$ MPa, $n = 700$ rpm The number of elements into which half of the bottom was split was 808 at 496 nodes. The calculation was carried out using a program for solving modified plane problems.

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