THERMAL CALCULATION OF A MOLD FOR CONTINUOUS CASTING OF METAL STRIP

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The article describes the design of a roll mold consisting of a block of centrifugal heat pipes, a radial and an axial one. It presents the method of calculating the structural elements, the capillary structure, and the temperature gradient of the system, proceeding from the conditions of ensuring normal processes of heat and mass transfer in the zones of transport and of phase transition.

Widely used at present are molds operating on the principle of centrifugal heat pipes (CHP). The high rotational frequency of the roll makes it possible to remove heat to the air fairly efficiently. This is the operating principle of the installation [1] in which the sloping walls, around which air flows, act simultaneously as condensers and heat exchangers. To intensify heat removal, these walls are ribbed on the outside. However, their small area limits the discarded heat flow, and consequently also the productivity of the installation. To increase it, a mold for continuous casting of metal strip was suggested; it consists of a block of CHP, a radial and an axial one, and the condenser of the radial pipe acts as evaporator of the axial CHP while the condenser of the latter is simultaneously an air-cooled heat exchanger [2]. This design makes it possible to increase the area of heat removal, and consequently also productivity.

The mold (Fig. 1) contains the rotating roll 1 in the form of a radial CHP with evaporator 2 and condenser 3 which acts simultaneously as evaporator 4 of the axial CHP 5. Its condenser 6 has ribs 7 on the outside. The mold is provided with the pouring gate system 8 for feed of the melt. In operation roll 1 seizes the strip of liquid metal which, cooling down, transfers its heat to the mold. On account of the closed evaporation—condensation cycles in both CHP this heat is transferred to the wall of condenser 6 and is removed into the air by the rotating ribbed surfaces 7.

For optimal organization of the process of phase transitions and transport, the elements of the CHP are provided with capillary structures.

In designing a mold, the main problem of thermal calculation was to determine the dimensions of its elements and the characteristics of the capillary structures which ensure heat transfer of the nominal heat flux removed into the air at the specified rotational frequency of the cylindrical walls which have certain overall dimensions and a system of ribbing.

The initial data for the calculation were: outer diameter of the roll and of the axial CHP, maximally permissible length of heat-removing surface, operating rotational frequency.

The method of calculation includes the following stages: determination of heat removal of the rotating ribbed walls into the air; calculation of the productivity of the mold; choice of the capillary structure and calculation of the dimensions of the elements of the axial CHP; choice of the capillary structure and calculation of the elements of the axial CHP; calculation of the temperature gradient of the mold.

In consequence of the variety of dimensions and shapes of ribbing, and the ambiguity of the effect of geometric factors on the aerodynamics and heat exchange, there does not exist at present a sufficiently correct method of calculating the intensity of heat transfer between the rotating ribbed walls and the air. For each actual case it is usually deter-
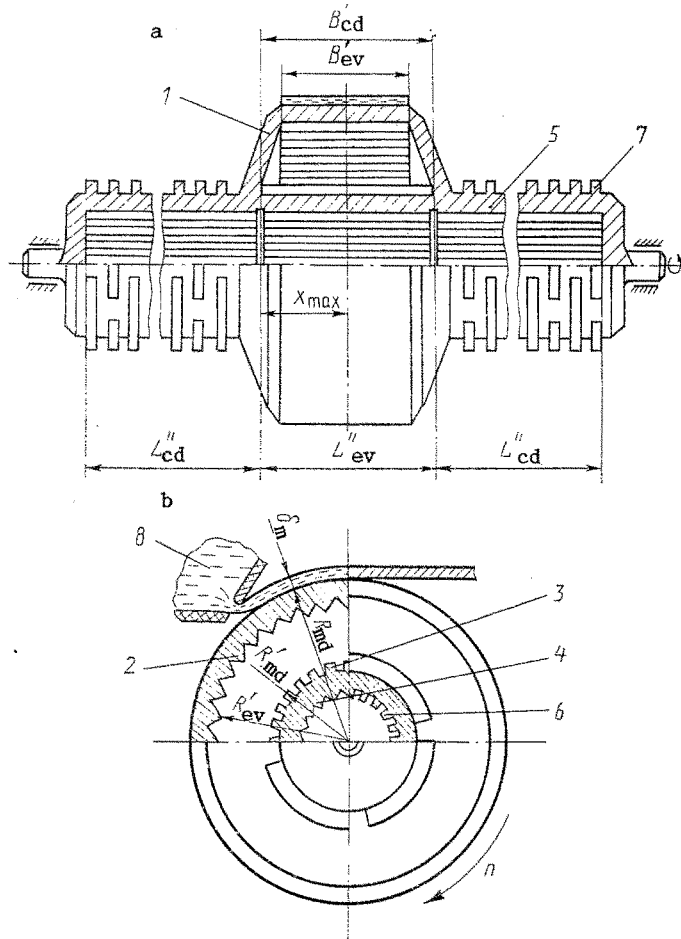
Fig. 1.   Mold for the continuous casting of metal strip
[a) front view; b) side view].

mined empirically. We therefore take it that at the operating rotational frequency pipes with specified dimensions and structure of the ribbing remove the nominal heat flux Q_{nom} into the air, and its value is the basis for the further calculation of the mold.

The productivity of the mold is determined in the following way. The thickness of the strip of melt forming on the roll [3] is:

$$\delta_m = \frac{1}{2\pi n} \sqrt{\frac{\sigma_m \cos \theta_m}{\rho_m R_{md}}};$$

its width is

$$B_m = \frac{Q_{nom}}{2\pi R_{md} n \delta_m \rho_m (C_m \Delta T_{s,m} + k r^*)}.$$

The width of the roll is chosen equal to the greatest width of the strip: $B_r = B_m$. The productivity of the mold is therefore calculated by the formula

$$M = B_m \sqrt{R_{md} \rho_m} \; \overline{\sigma_m \cos \theta_m}.$$

The capillary structure is chosen by proceeding from the criterion of the technological suitability of its formation. That concerns differently shaped grooves cut directly into the walls of the working zones of the CHP. In the evaporator grooves with triangular profile are used because they ensure that their pitch is minimal, in the condenser their profile is rectangular because the process of condensation proceeds on the front part of the necks between the grooves (Fig. 2a).

The volume of working liquid that has to be evaporated for removing the nominal thermal load is:
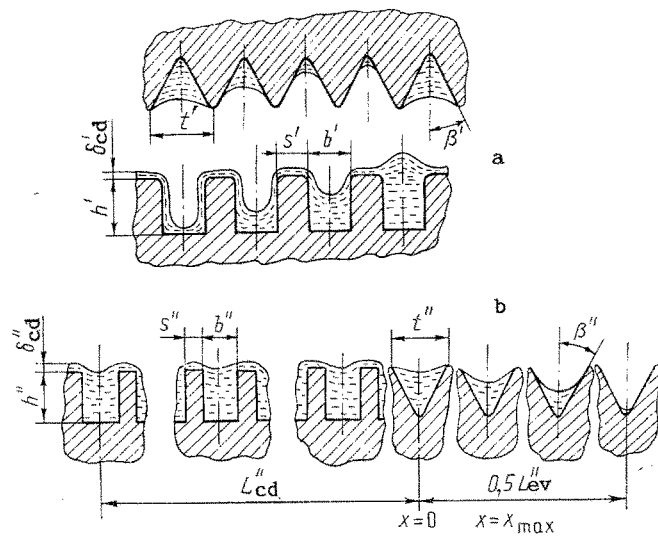
Fig. 2. Capillary structures of the mold: a) axial CHP; b) radial CHP.

$$W = \frac{Q_{,\text{nom}}}{r' \rho_{\ell} n}.$$

It is recommended to make the angle at the apex of the triangular grooves within the limits [t] $2\beta' = 38\text{-}40°$. The width of the grooves and their number in the evaporator should be

$$t' = \frac{2W}{\pi R'_{\text{ev}} B'_{\text{ev}} \text{ctg}\, \beta'}, \quad m'_{\text{ev}} \approx \frac{2\pi R'_{\text{ev}}}{t'}, \quad B'_{\text{ev}} = B_{\text{m}}$$

The vapor tension in the grooves is determined from the pressure with the effect of the centrifugal forces taken into account:

$$P'_{\text{v}} = \frac{(t')^2}{4} \text{ctg}\, \beta' \rho'_{\ell} a_{\text{c}} + P_{\text{sat}}.$$

The working density of the heat flux in the evaporator is

$$q' = \frac{Q_{\text{nom}}}{2\pi R'_{\text{ev}} B'_{\text{ev}}}.$$

The maximal heat flux density in boiling is [5]

$$q'_{\text{max}} = \frac{\pi}{24} r' \rho'_{\text{v}} \left[\frac{\sigma'_{\ell} (\rho'_{\ell} - \rho'_{\text{v}}) a'_{\text{d}}}{(\rho'_{\text{v}})^2} \right]^{0,25} \left[\frac{\rho'_{\ell}}{\rho'_{\ell} - \rho'_{\text{v}}} \right]^{0,5}.$$

The working density must not exceed the maximal one: $q' \leq q'_{\text{max}}$. The dimensions of the rectangular grooves of the condenser are determined from the condition that drops of working liquid maintained by the forces of surface tension are ejected from them by the centrifugal forces, i.e., it has to be ensured that

$$P'_{\text{cd}} \leqslant P'_{\text{ef}}$$

or

$$\frac{2\sigma_{\ell}}{b} \leqslant 4\pi^2 n^2 R'_{\text{cd}} h' \rho'_{\ell} + 0.5\pi^3 b R'_{\text{cd}} n^2 \rho'_{\ell}.$$

Solving this equation, we obtain

$$b \geqslant \frac{-4h' \pm \sqrt{16 (h')^2 + \pi K}}{\pi},$$

where

$$K = \frac{4\sigma'_\ell}{\pi^2 n^2 R'_{cd} \rho'_\ell} .$$

The width of the groove b cannot exceed 10^{-3} m, its minimal value is determined by the technological possibilities. The maximal depth h is limited by the thickness of the wall dividing the pipe. On the basis of these considerations the dimensions of the grooves are calculated.

The thickness of the film of condensate maintained by the forces of surface tension on the front part of the neck between grooves is determined as

$$\delta'_{cd} = \frac{1}{2\pi n} \sqrt{\frac{\sigma'_\ell \cos\theta'_\ell}{\rho'_\ell R'_{cd}}} .$$

The heat transfer coefficient in condensation in the field of centrifugal forces is calculated by Nusselt's equation [6].

The indispensable area of the condenser and the width of the neck between grooves are

$$s'_{cd} m'_{cd} B'_{cd} = \frac{Q_{nom}}{\alpha'_{cd} \Delta T_4} ,$$

$$s'_{cd} = \frac{4.25 K Q_{nom}}{W \alpha'_{cd} \Delta T_4} .$$

The number of grooves is

$$m'_{cd} = \frac{2\pi R'_{nom}}{b' + s'} .$$

In the calculation of the elements of the axial CHP we choose, like in the preceding case, triangular capillary grooves for the evaporator, and rectangular ones for the condenser (Fig. 2b). The length of the evaporator of the axial CHP is chosen equal to the width of the condenser of the radial CHP:

$$L''_{ev} = B'_{cd}.$$

The apex angle is also chosen within the lmits 38-40°. The heat flux density of the evaporator is

$$q'' = \frac{Q_{nom}}{2\pi R'' L''_{ev}} .$$

The width of the triangular groove of the evaporator is determined on the basis of the following considerations. It was established that with the method of feeding the working liquid that is used in the design under consideration, heat removal along the triangular groove of the evaporator is of a hyperbolic nature [7]. Since the axial CHP is symmetric, we deal with the section of the evaporator $x_{max} = 0.5 \, L_{ev}''$. Consequently, maximal heat removal occurs at the point $x = 0$, and zero heat removal at the point $x = x_{max}$.

The balance of pressures at any point of the section of the groove is:

$$P_{cd} + P_{cf} = F_h,$$

$$P_{cp} = \int_0^{t''} \frac{2\sigma_\ell \cos\theta_\ell (1 - \sin\beta'')}{t^2(x)\cos\beta''} dt,$$

$$P_{cf} = \int_0^{t''} \frac{t^2(x)\,\text{ctg}\,\beta'' \rho''_\ell a''_c}{t'' x_{max}} dt.$$

(1)

The hydraulic resistance to the flow of liquid in the groove is

$$P_h = \int_0^{x_{max}} \frac{q'' x_{max} t'' f \mu''_\ell \ln \frac{0.01 + x_{max}}{0.01 + x}}{r'' \rho''_\ell t^4(x) C_1(\beta) \ln(1 + 100 x_{max})} \, dx,$$

$$C_1(\beta) = \frac{\operatorname{ctg}\beta'' \cos^2\beta''}{4(1 + \sin\beta'')^2}.$$

As a result of the solution we obtain

$$q'' = \frac{(t'')^2 r'' \rho''_\ell \sigma''_\ell \cos\theta''_\ell C_2(\beta) K(\beta)}{\mu''_\ell x_{max}\left[\frac{0.01 + x_{max}}{\ln(1 + 100 x_{max})} + 0.01\right]} + \frac{(t'')^5 a''_c \rho''_\ell C_1(\beta) K(\beta)}{2.3 x_{max}^2 \mu''_\ell \left[\frac{0.01 + x_{max}}{\ln(1 + 100 x_{max})} + 0.01\right]},$$

where

$$C_2(\beta) = \frac{\cos\beta'' \operatorname{ctg}\beta''(1 - \sin\beta'')}{2(1 + \sin\beta'')^2};$$

$$K(\beta) = 0.0535 (2\beta'')^{1.55},$$

where the value of $K(\beta)$ is given in radians [8]. From the known heat flux density the width of the triangular groove of the evaporator is determined.

The number of grooves in the axial CHP is

$$m'' \approx \frac{2\pi R''}{t''}.$$

The dimensions of the capillary structure in the condenser are chosen from the condition of ensuring the transport of such an amount of working liquid which is essential for the evaporator. The evaporator and the condenser have the same number of grooves. Since the moving meniscus is the curvature of the liquid in the last element of the structure, the triangular groove, the section of the flow of liquid in the rectangular groove remains unchanged.

The balance of pressures in any section of the groove is determined analogously to expression (1):

$$P_{cp} = \frac{2\sigma''_\ell \cos\theta''_\ell(1 - \sin\beta'')}{t'' \cos\beta''},$$

$$P_{cf} = h'' \rho''_\ell a''_d, \quad P_h = \int_0^{L''_{cd}} \frac{f \dot{m}'' \mu''_\ell}{A'' D''_h \rho''_\ell} \, dx, \quad A = b'' h'', \quad D_h = \frac{2 b'' h''}{b'' + h''}.$$

After transformation we obtain

$$\frac{b'' h''}{(b'' + h'')^2} = \frac{f \dot{m} \mu''_\ell L''_{cd}}{\rho''_\ell \left[\frac{2\sigma_\ell \cos\theta''_\ell(1 - \sin\beta'')}{t'' \cos\beta''} + h'' \rho''_\ell a''_d\right]}. \tag{2}$$

To solve this equation we adopt additionally:

$$f \approx 36, \quad \dot{m}'' = \frac{Q_{nom}}{2r''}, \quad h'' = \frac{t''}{2} \operatorname{ctg}\beta''.$$

The width of the neck between grooves is

$$s'' = \frac{2\pi R''}{m''} - b''.$$

The indispensable value of the coefficient of condensation is

$$\alpha''_{cd} = \frac{Q_{nom}}{2\Delta T_7 s'' m'' L''_{cd}}.$$

In accordance with the Nusselt equation the thickness of the film of condensate is determined from α_{cd}''. This is attained by choosing the volume of working liquid poured into the axial CHP.

It has to be pointed out that in the solution of Eq. (2) the depth of the triangular and the rectangular grooves were taken to be equal. However, if necessary, the depth of the rectangular groove may be changed. To reduce the resistance to the transition, a circular bore was made on the border of the grooves.

To ensure the required intensity of heat exchange during cooling of the strip, it is necessary to calculate the temperature gradient arising in different parts of the system melt—roll—environment:

$$\Sigma\Delta T = \Delta T_1 + \Delta T_2 + \Delta T_3 + \Delta T_4 + \Delta T_5 + \Delta T_6 + \Delta T_7 + \Delta T_8 + \Delta T_9,$$

where ΔT_1 is the temperature gradient of contact of the liquid metal strip with the rim of the mold obtained by solving the equation of contact heat exchange; ΔT_2 and ΔT_3 are the temperature gradients along the rim of the roll and the layer of liquid in the groove, calculated by the numerical solution of the equation of heat conduction; ΔT_4 is the temperature difference between the vapor and the condenser wall in the radial CHP. Its minimal temperature is determined by the difference of the saturation pressure at which the vapor moves from the evaporator to the condenser against the centrifugal forces:

$$\Delta P = \frac{t'}{4}\operatorname{ctg}\beta' \rho_v' a_c';$$

ΔT_5 and ΔT_6 are the temperature gradients on the wall dividing the pipes, and along the layer of liquid in the groove of the evaporator of the axial CHP, calculated by the numerical solution of the equation of heat conduction; ΔT_7 is the temperature difference between the vapor and condenser wall of the axial CHP; ΔT_8 is the temperature gradient on the condenser wall of the axial CHP calculated by the numerical solution of the equation of heat conduction; ΔT_9 is the temperature gradient between the ribbed wall of the mold and the air, determined experimentally together with the intensity of heat removal from the rotating mold.

In the calculation of ΔT_1 the coefficient of contact heat exchange between the strip and the mold is found as the ratio of the thermal conductivity of air to the height of microroughnesses of the outer surface of the roll. Gradients in the vapor are so small that they are not taken into account. The vapor temperatures in the radial and axial CHP are, respectively, equal to:

$$T_v' = T_m - (\Delta T_1 + \Delta T_2 + \Delta T_3), \quad T_v'' = T_v' - (\Delta T_4 + \Delta T_5 + \Delta T_6).$$

The saturation pressure in the pipes is determined from the temperature of the vapor.

The temperature difference between the vapor and the condenser wall of the axial CHP is $\Delta T_7 = T_v'' - (\Delta T_8 + \Delta T_9)$.

The newly devised method of calculation was used in designing and producing an installation making metallic strip up to 10^{-3} mm thick and up to 0.05 m wide, of complex alloys with improved physicomechanical properties at elevated temperatures.

NOTATION

A) area, m^2; a) acceleration, m/sec^2; B) width, m; b) width of the rectangular groove, m; C) heat capacity, $J/(kg \cdot K)$; D) diameter, m; f) coefficient of resistance; h) height of the rectangular groove, m; k) coefficient taking into account the ratio of crystallization time on the roll and in the air; L) length, m; M) productivity, kg/sec; m) number of grooves; $\dot{m}$, mass flow rate of liquid, kg/sec; n) rotational frequency, sec^{-1}; P) pressure, N/m^2; ΔP) pressure difference, N/m^2; Q) heat flux, W; q) heat flux density, W/m^2; R) radius, m; r) latent heat of vaporization, J/kg; r^*) latent heat of crystallization, J/kg; s) width of the neck between rectangular grooves, m; T) temperature, K; ΔT) temperature gradient, k; t) width of the triangular groove, m; t(x)) width of the layer of liquid in any section of the triangular groove, m; W) volume, m^3; x) coordinate, m; α) heat transfer coefficient, $W/(m^2 \cdot K)$; β) half-angle at the apex of the triangular groove, deg; δ) thickness, m; θ) wetting angle, deg; μ) viscosity, $N/(sec \cdot m^2)$; ρ) density, kg/m^3; σ) surface tension, N/m. Subscripts: ') radial CHP; ") axial CHP; r) roll; h) hydraulic; ℓ) liquid; ev) evaporator; cd) condenser;

cp) capillary; md) mold; sat) saturation; nom) nominal; v) vapor; s.m) superheating of the melt; m) melt; c) centripetal; cf) centrifugal; max) maximal.

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NONISOTHERMAL RHEODYNAMICS IN SHS PRESSING OF POWDER MATERIALS

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The article deals with thermal and rheodynamic processes of SHS pressing (SHS stands for self-propagating high-temperature synthesis) of viscous compressible materials. It presents numerical calculations of the nonisothermal kinetics of compaction with different thermal and technological parameters. Conditions are found for the realization of qualitatively different regimes of compacting SHS materials.

One of the topical problems of the theory and practice of SHS pressing (the same as in hot pressing) of powdered high melting materials is the study of the state of stress of the products of synthesis under the effect of applied external forces. It is usual to apply the macrorheological approach to the description of the behavior of porous materials suggested in [1-3] which makes use of the model of a viscous compressible liquid. Buchatskii et al. [3] and Stolin et al. [4] found an analytical solution of the problem of one-sided compression of such systems for the case when there is no temperature distribution in the material. Buchatskii et al. [5] made a qualitative analysis of different thermal regimes of compaction on the assumption that in the process a thermal gradient is not involved. The obtained analytical solution of the problem made it possible to evaluate the conditions of realization of a quasiisothermal regime of compaction where the process of pressing is not accompanied by a noticeable change of temperatures. However, in practice the nonuniformity of the temperature regime in the material and the conditions of heat exchange have a substantial effect on the distribution of densities, speeds, and stresses, and consequently also on the quality of the finished products. The aim of the present work is a numerical analysis of the temperatures, densities, speeds, and stresses within the bulk of viscous porous material in the process of its one-sided compression in dependence on the initial distribution of temperature and density throughout its volume.