REVIEW

COOLING SYSTEMS OF ROTATING MOLDS

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In the last 20 years the method of high-speed (as a rule, higher than 10^3 K/sec) metal solidification from the liquid state has found wide application in manufacture of materials. It offers the possibility to fix metastable phases, to extend the range of solid solutions, to form an amorphous state, and thereby to obtain materals whose physicomechanical properties are superior to those of traditional alloys [1]. Methods of high-speed solidification are also employed in the manufacture of either disperse particles (powders) or continuous products (strips, sheets, wires).

At present two main methods of high-speed melt solidification are being used for the manufacture of continuous products: due to contact with a solid medium (a substrate) having a heat removal system; in a liquid medium [2]. The present review is largely concerned with the former method. For this case a melt metal is brought to one or two rotating roll-molds, where it hardens and transforms into strip, wire, short fibers or threads. When being poured onto one roll, a liquid metal may be fed by one of three methods: top casting from a gating system due to a metal-static pressure or an excess pressure (spinning); thin melt layer entrainment by a roll immersed in a bath placed beneath it (extraction); liquid-metal strip entrainment by a roll from a gate box placed on the side (one-roll continuous casting).

1. Heat Load of a Mold. A stable process of obtaining continuous pieces may be accomplished if an effective system of heat removal from a roll (a surface to be cooled) is provided. To design such heat removal systems, one must know the heat load of a mold, which is determined by the dimensions of the finished products and the mold capacity. In the literature there are works on determination of the thickness of a forming melt layer in withdrawal of it from a bath by a rotating roll [3, 4] and in casting onto a single rotating roll [5, 6]. The calculated relations obtained are based on analysis of the eddy flow of a melt extracted by a roll [4] or on processing of experimental data [5]. The main drawback of the cited works is neglect of the mechanism that holds a melt layer on a roll and essentially affects the melt-to-roll heat transfer. Therefore the relations proposed are applicable only for some particular cases or are complicated for design purposes.

In [7], it is established that thin metal strips may be obtained with the aid of a rotating mold only in the case when the melt wets the roll; therefore the wetting angle must be within $0 \le \theta < 90^{\circ}$. This prerequisite has been used as a basis for a model of strip formation on a roll adopted to determine the strip thickness as well as to analyze the forces acting on an elemental portion of the melt under spinning conditions [8, 9]. Forces of friction, gravitation, and surface tension hold a melt on a roll, while centrifugal and inertia forces tear it off. An expression for the strip thickness in final form is:

$$\delta_s = \sqrt{\frac{\sigma_m \cos \theta_{mo}}{4\rho_m \pi^2 n^2 R_{mo} - \rho_m g \cos \gamma}}\ .$$

When the gravitational acceleration is negligible as compared to the centripetal one, this formula may be simplified:

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$$\delta_s = \frac{1}{2\pi n_{mo}} \sqrt{\frac{\sigma_m \cos\theta_{mo}}{\rho_m R_{mo}}}. \quad (1)$$

The width of the melt strip is determined by the equation of mass balance of liquid metal discharge via the slot of the gate box [10] with a height H of the melt in it

$$b_s = \frac{A_{sl}\sqrt{2gH}}{2\pi R_{mo}\delta_s n_{mo}}.$$

The difference between the maximum and minimum widths of the strip with a melt height change of Δh (due to melt feed fluctuation) is

$$\Delta b_s = \frac{A_{g.b} 2g(\sqrt{H} - \sqrt{H - \Delta h})}{2\pi R_{mo}\delta_s n_{mo}}.$$

The surface tension forces developing during contact of the melt with the roll-mold influence the liquid metal discharge via the slot of the gate box; therefore the maximum strip thickness at a fixed slot width is expressed as

$$\delta_s \leqslant \frac{b_{sl}\cos\theta_{mo}}{2\cos\theta_{sl}}.$$

In one-roll continuous casting, the roll withdraws the melt from the gate box by the forces of surface tension and friction. The thickness of the strip formed on the roll is determined as follows:

$$\delta_s = \sqrt{\frac{f\sigma_m \cos\theta_{mo}}{\rho_m(a + g\cos\gamma + 4f\pi^2 n_{mo}^2 R_{mo} - fg\sin\gamma)}}, \quad (2)$$

where

$$a = \frac{2\pi^2 n_{mo}^2 R_{mo}^2}{\Delta l}.$$

The strip width during continous casting is equal to the width of the gate box.

In extraction the strip thickness is calculated just as in one-roll continuous casting. The strip width is equal to the mold width.

To calculate the strip thickness, one must substitute the values of the melt surface tension σ_m and the angle of mold wetting θ_{mo} by the melt into formulas (1) and (2). Examples of comparison of experimental and calculated results are given in [8, 9]. In [11-13], the surface tension for commercial-purity aluminium is found to be $\sigma_m \approx 0.86$ N/m. The angle of wetting of a copper surface by liquid aluminium is experimentally determined from the rise in a capillary as $\theta_{mo} \approx 52^\circ$.

The mold capacity under spinning conditions is

$$M = b_s\sqrt{R_{mo}\rho_m\sigma_m\cos\theta_{mo}},$$

and in one-roll continuous casting

$$M = 2\pi R_{mo} n_{mo} b_s \rho_m \sqrt{\frac{f\sigma_m\cos\theta_{mo}}{\rho_m(a + g\cos\gamma + 4f\pi^2 n_{mo}^2 R_{mo} - fg\sin\gamma)}}.$$

The heat load is directly related with the mold capacity. In spinning

$$Q = M(C_m\Delta T_{ms} + kr).$$

The coefficient k is the ratio of roll-to-air solidification time. In continuous casting, when the hardened strip is retained against the roller,

$$Q = M(C_m\Delta T_{ms} + r + C_s\Delta T_{co}).$$

The specific heat flux of the mold is calculated based on the premise that under steady-state casting conditions heat is removed from the entire area of the inner periphery of the rim owing to roll rotation and its high heat conductivity

$$q = \frac{Q}{2\pi R_{mo} B_{mo}}.$$

2. Heat Removal Systems of Roll-Molds. The design solution of heat removal systems applied in rotating molds is specified by the presence of a cooling liquid and its mass velocity, the pressure head, the overall dimensions of the roll, and its operation regimes. Since the heat transfer rate is affected by the field of centrifugal forces, the type of heat removal system depends to a great extent on the casting rate of the mold. The centripetal-to-gravitational ratio may be used as a criterion allowing rotating molds to be classified into two groups [14]. At $a_c/g < 5$, rolls are considered slow-rotating, and at $a_c/g > 5$, fast-rotating. In fast-rotating rolls the developing centrifugal forces favor stretching of the cooling liquid over the inner rim of the roll. As a rule, fast-rotating rolls are used in spinning, and slow-rotating rolls in continuous casting.

At present the following methods of cooling a mold are used:

1) cooling with a running liquid;

2) cooling involving phase transitions (evaporation, boiling) of a working liquid in the inner part of a casting mold;

3) design of a mold as a system with a closed evaporation-condensation cycle;

4) design of a mold as a vacuum system in which heat and mass transfer processes are determined by centrifugal forces (utilization of centrifugal heat pipes).

2.1. Molds Cooled by a Running Liquid. Such molds are in most common use. They may be of two types: with a single cooling cavity, i.e., with a single-channel liquid heat exchanger [15-18] (Fig. 1a); with a multichannel heat exchanger [19-27] (Fig. 1b). A multichannel heat exchanger provides better cooling due to an increase of the cooling-liquid velocity.

To design a cooling system, the following initial data are usually used: the mass flow rate of the cooling liquid $\dot{m}$, the permissible pressure loss ΔP_2, the initial T_{ini} and final T_{max} temperatures of the cooling liquid. The first two quantities are specified by the pumping system capacity, while the remaining parameters depend on environmental conditions of the heat load and prevention of liquid boiling. The design is aimed at determination of the optimal dimensions of the heat exchanger channels on the basis of the given conditions.

The maximal heat flux released by the cooling system [26] is

$$Q = \dot{m} C_l (T_{max} - T_{ini}), \quad T_{max} < T_{sa}.$$

The heat transfer rate in the channels of the cooling system is determined for two liquid flow modes: laminar and turbulent.

L a m i n a r F l o w (Re < 2300). The dimensions of the channel elements are found by solving the Hagen-Poiseuille equation [27]:

$$\Delta P_h = \frac{\xi \dot{m} \mu_l}{A_{ch} D_h^2 \rho_l} L.$$

Substituting the values of the quantities entering this expression, we arrive at

$$\Delta P_h = \frac{\dot{m} \mu_l \pi D_{in} B_{mo} (h_{ch} + b_{ch})^2}{4 h_{ch}^3}.$$

If the channel height h_{ch} is given from design considerations, their width is determined by solving the following equation:

$$b^2 - b_{ch} \sqrt{\frac{K}{\Delta P_h h_{ch}^3}} - \sqrt{\frac{K}{\Delta P_h h_{ch}}} = 0,$$

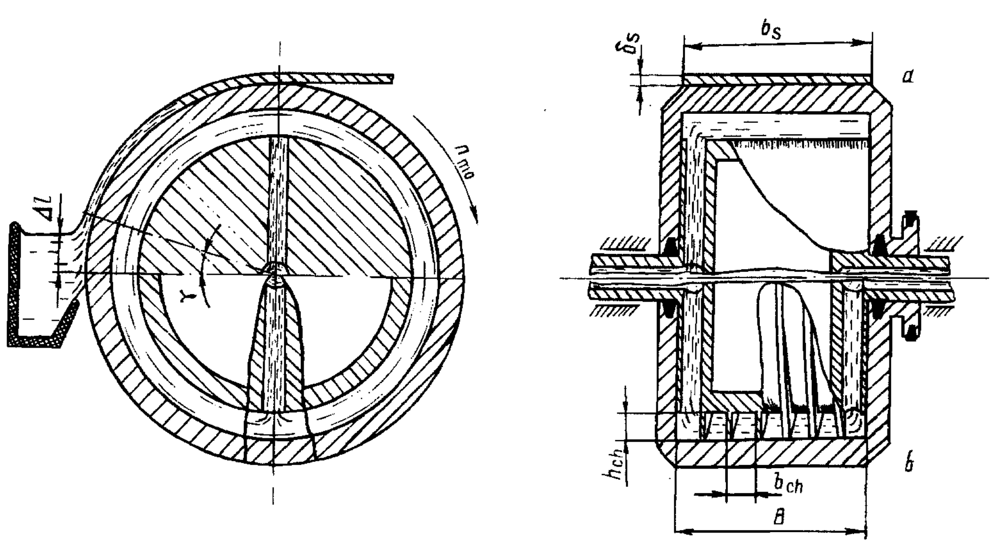


Fig. 1. Molds with liquid heat exchangers: a) single-channel; b) multichannel heat exchanger.

where

$$K = \frac{\xi \dot{m} \mu_l \pi D_{in} B_{mo}}{4\rho_l}.$$

Recommendations on determination of the resistance coefficient are given in [28].

The Nusselt number for laminar flow in pipes is

$$\mathrm{Nu} = 1{,}55 \left(\mathrm{Pe}\frac{D_h}{L}\right)^{1/3} \left(\frac{\mu_w}{\mu_l}\right)^{0,14} \varepsilon,$$

where

$$\varepsilon = \frac{0{,}1\left(\frac{1}{\mathrm{Re}}\frac{L}{D_h}\right)^{-1,7}}{\left(1 + 2{,}5\frac{1}{\mathrm{Re}}\frac{L}{D_h}\right)}.$$

This formula is valid when the channel length is considerably larger than its diameter ($L/D_h > 200$). The physical properties of the liquid are determined from the mean temperature in the channels.

Turbulent Flow (Re > 10,000). In this case the pressure drop is calculated by the Weissbach-Darcy equation [29]:

$$\Delta P = \xi \frac{l}{2D} \frac{\dot{m}}{A_h^2 \rho_l}.$$

The resistance coefficient is determined by the Blasius relation [30]:

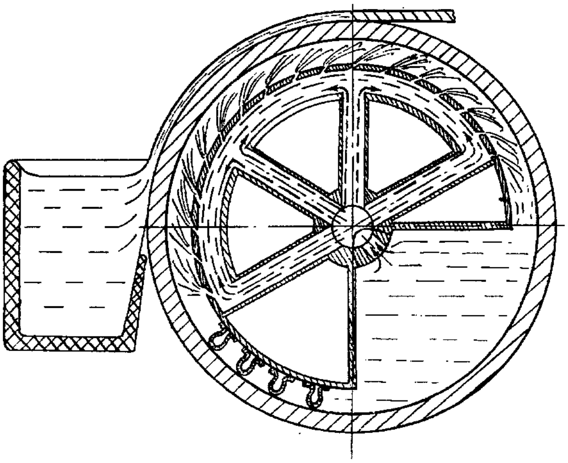
$$\xi = 0{,}316/\mathrm{Re}^{0,25} \quad (\mathrm{Re} < 10^5).$$

The groove dimensions of the heat exchanger may be calculated using the relation below, which is obtained by substituting all the quantities into the Weissbach-Darcy equation:

$$\frac{b_{ch} + h_{ch}}{b_{ch}(b_{ch}h_{ch})} = 15 \frac{\Delta P_h \rho_l^{0,75}}{\dot{m}^{1,75} \nu_l^{0,25} (\pi D_{in} B_{mo})}.$$

The Nusselt number, according to [31], is as follows:

$$\mathrm{Nu} = 0{,}017\,\mathrm{Re}^{0,8}\,\mathrm{Pr}_l^{0,4} \left(\frac{\mathrm{Pr}_l}{\mathrm{Pr}_w}\right)^{0,25} \left(\frac{D_{in}}{D}\right)^{0,18}, \quad D = D_{in} + 2h_{ch}.$$



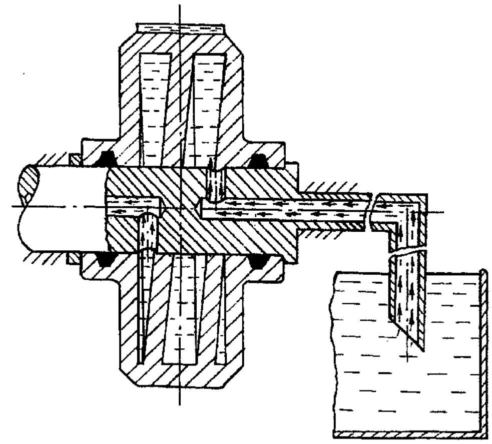


Fig. 2. A mold with cleaners to descale the inner roll surface.

Fig. 3. A mold with its inner cavity in the form of a spiral channel.

A turbulent flow regime is preferable to a laminar one since it ensures a larger heat transfer coefficient and a lower boiling probability. However, in a viscous sublayer local boiling accompanied by scale formation on the inner surface of the roll develops. To descale the roll, it is recommended to use special cleaners consisting of an assembly of stainless grids [17, 32] (Fig. 2). The cleaners are tightly pressed against the inner part of the rotating rim, thus removing the scale therefrom.

It should be noted that cooling by a running liquid is used in continuous casting. For spinning, casting molds whose inner cavity is manufactured as a spiral channel are used (Fig. 3) [33]. They need no pumping system; the cooling liquid, forced through the spiral cavity, is intensely turbulized, thus decreasing the probability of onset of boiling at the heating surface. The mass flow rate of the pumped liquid is calculated analogously to the case of screw pumps.

2.2. Molds Cooled by Evaporation and Boiling on an Open Inner Part. Such molds may have a roll with a smooth inner surface (Fig. 4) [34-37] or with a capillary-porous coating [38, 39]. The cooling liquid may be supplied to the heat removal surface either by wetting (for both types) or via an elastic capillary-porous stack (only for casting molds with a capillary coating).

Molds with a smooth surface must be fast-rotating to provide uniform distribution of the cooling liquid over the inner part of the rim by centrifugal forces. Molds with liquid supply via a capillary-porous stack must be slow-rotating, since the liquid flows over the a porous body with a low velocity.

The maximum heat-removing capacity of a mold with a smooth surface is limited by the specific heat flux causing film boiling. Taking account of centrifugal forces, we may write

$$q_{max} = q_{cr} \left(\frac{a_c}{g}\right)^m .$$

The critical heat flux determined from analysis of the Helmholt instability [40] is expressed by the following relation [41]:

$$q_{cr} = \frac{\pi}{24} r^* \rho_v \left[\frac{\sigma_l (\rho_l - \rho_v) g}{\rho_v^2}\right]^{0,25} \left[\frac{\rho_l}{\rho_l + \rho_v}\right]^{0,5} .$$

According to [42-44], m ≈ 0.25.

Calculations performed [45] have revealed that the exponent depends both on the ratio a_c/g and on the liquid layer height on the inner part of the rim of the casting mold. The minimum height is

$$h_{min} > \frac{q_{max}}{\rho_l n_{mo} r_l} .$$

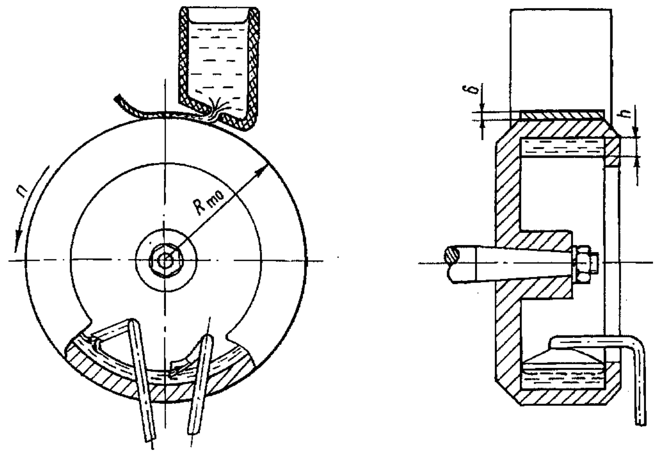


Fig. 4. A mold cooled by liquid evaporation and boiling on the smooth inner surface of a roll.

Varying this quantity, one may increase the maximum heat-removing capacity of the mold.

An example of a fast-rotating mold with a capillary-porous inner surface is shown in Fig. 5 [39]. The mould has porous corks with a certain permeability in a hub, through which the cooling liquid is discharged due to centrifugal forces at a fixed flow rate.

In a slow-rotating mold (Fig. 6) [46, 47] the cooling liquid is supplied to the surface of roll 1, which is coated inside with capillary-porous layer 2, by surface tension forces from branch pipe 3 via elastic porous element 4. The main goal of the design is determination of the dimensions and characteristics of the porous elements of the mold on the basis of the possibility of providing the necessary rate of cooling-liquid feed to the phase transition surface.

Molds with a porous inner layer have some advantages over smooth-walled molds since the boiling process is enhanced because it proceeds on developed surfaces. However, on long operation in the atmosphere, the working liquid is contaminated and clogs up the pore space with suspended particles. Therefore it is more expedient to use phase transition cooling with a closed evaporation-condensation cycle.

2.3. Molds with a Closed Evaporation-Condensation Cycle. In slow-rotating moulds ($a_c/g < 5$) centrifugal forces do not exert a crucial influence on heat and mass transfer processes. Therefore when closed evaporation-condensation systems are used, the working liquid may be returned mainly by gravitation and surface tension forces.

Use of such heat removal systems makes it possible:

1) to enhance heat and mass transfer processes because of phase transitions proceeding in equilibrium with the vapor in the absence of noncondensing gases;

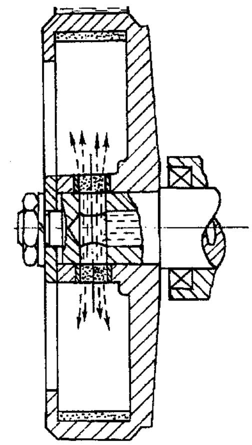
2) to avoid contamination of the porous surface with scale and suspended particles by evacuating the system;

3) to develop phase transition surfaces in connection with use of capillary-porous bodies;

4) because of a temperature gradient in the system to avoid cooling-liquid boiling and sedimentation in the heat exchanger.

A slow-rotating mold in which heat is transferred from a roll in a closed evaporation-condensation cycle is schematically shown in Fig. 7 [48]. The mold has roll 1 with capillary-porous layer 2 on its inner surface, condenser 3 which houses liquid heat exchanger 4, storage unit 5, porous stack 6, and heater 7. When the roll is rotating its outer surface "freezes" a layer of metal taken from the superheated melt. The heat released on cooling, solidification, and crystallizationm of the liquid metal is absorbed on evaporation and boiling of the working liquid in the capillary-porous layer. The vapors formed are transferred to the minimum-pressure zone, viz., to condenser 3 cooled with water running in liquid heat exchanger 4. The condensate film flows by gravity to storage unit 5, from where it is admitted to porous stack 6 and then returned by surface tension forces to porous layer 2. So, a closed evaporation-condensation cycle results. Heater 7 mounted on the stack serves to heat up the mold in the prestart-up period and to eliminate defective strips in evaporation of moisture adsorbed on the roll.

The procedure of design of the mold described above [49] includes determination of the heat load, the dimensions and characteristics of its parts, and the evaporator-to-condenser mass transfer rate.



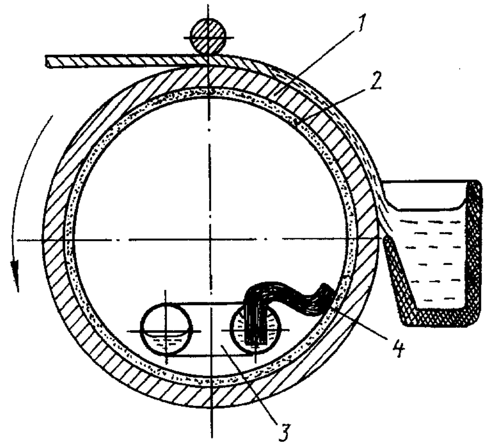


Fig. 5. A mold with cooling-water feed via porous plugs.

Fig. 6. A mold with cooling-water feed via a capillary-porous stack.

When choosing the working liquid, one must take into consideration the degree of wetting the capillary-porous body by it and its general characteristic, which is defined as follows [50]:

$$N_l = \frac{r^* \rho_l \sigma_l}{\mu_l}, \quad \mathrm{W/m^2}$$

It conventionally characterizes the specific heat flux transferred at phase transitions. Of a wide class of liquids used in cryogenic heat pipes (Freons, acetone, Dowtherm, etc.) water has the best characteristic ($4.6 \cdot 10^{11}$ $\mathrm{W/m^2}$ at 100 °C).

The flow rate of the working liquid to be evaporated for removal of the working heat load is

$$m = \frac{Q}{r^*} K.$$

The coefficient K accounts for ejection of the liquid out of pores in boiling.

The contacting length of the elastic porous stack is determined from the condition of filling the capillary-porous layer with the working liquid in rotation of a mold at a prescribed frequency:

$$l = \frac{2\pi R_{\mathrm{in}} n_{\mathrm{mo}} \mu_l h^2 \Pi}{\Pi p_{\mathrm{cp}} \Delta P}.$$

The dimensions of the elastic porous stack are calculated on the basis that the required amount of the working liquid is transferred:

$$\frac{h}{L} = \frac{Q \mu_l K}{\Pi p_{\mathrm{cp}} \Delta P r^* b_{\mathrm{cp}}}, \quad \Delta P = \frac{2\sigma}{R_{\mathrm{cp}}}.$$

To determine the condenser dimensions, we calculate the temperature gradient between the vapor and the condenser wall. From the gradient value and the size of the housing the rate of heat transfer in condensation is evaluated using the Nusselt equation. In the case of a finned condenser, the reduced heat transfer coefficient for such a surface is calculated and the following relation is checked:

$$\frac{Q}{A} \leqslant \alpha_{\mathrm{re}} \Delta T_{\mathrm{cn}}.$$

Use of porous molds with a closed evaporation-condensation cycle allows one to maintain a constant thermal regime of strip formation because of the steady-state of the cooling processes. As a result, this ensures the constancy of the geometric dimensions and the mechanical properties of the items produced.

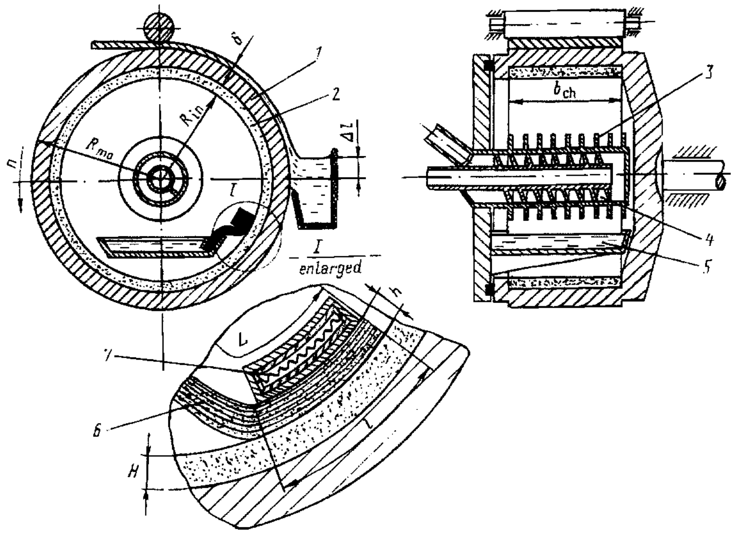


Fig. 7. A mold cooled in a closed evaporation-condensation cycle.

2.4. CHP Molds. If in operation of a mold the centripetal acceleration is much higher than the gravitational one, then in order to enhance heat transfer, it is beneficial to design the roll in the form of a centrifugal heat pipe (CHP). In that case the working liquid is transferred from the condenser to the evaporator by centrifugal forces. An ordinary variant of such a mold is depicted in Fig. 8 [51, 52]. It consists of rotating roll 1 and rim 2, which make up a leakproof cavity filled with the working liquid, liquid heat exchanger 3, and gating system 4. An outer rim serves as the mold surface, on which the strip hardens, and the CHP evaporator. An inner rim serves as the CHP condenser and the housing of the liquid heat exchanger that cools the mold.

When the mold is operating, the rotating roll entrains the melt and cools it. Particles of the working liquid in the leakproof cavity are transferred by centrifugal forces to the roll wall, where they are evaporated due to heat drawn from the roll. The formed vapor is directed to the minimum-pressure zone, viz., to the rim, where it condenses, thus heating the cooling liquid running in the liquid heat exchanger. To enhance heat and mass transfer involving phase transitions, capillary-porous layers are deposited onto the evaporator and the condenser.

Such a system will function properly provided [53]: a) the centrifugal forces exceed the gravitational and surface tension forces (the liquid is transferred to the outer rim); b) the pressure gradient between the evaporator and the condenser determined by the temperature gradient is higher than the pressure caused by the centrifugal force (the vapor is returned to the condenser). The latter condition is expressed by the following relation:

$$\frac{\pi D_b^2}{4}\,\Delta P_{sa} > \frac{\pi D_b^3\, v_{ev}\,\rho_v}{6R_{in}} + \frac{4\sigma_l}{D_b}\,.$$

A CHP mold has one more merit: the high frequency of rotation allows its effective cooling by air. If we fin the heat-releasing side walls, then such a mold does not need liquid cooling. An engineering realization of such a system is shown in Fig. 9 [54]. As a capillary structure, it uses triangular grooves. A special feature of the design consists in the fact that the walls serve simultaneously as a CHP condenser and as finned heat exchangers blown on the outside by air in rotation. When a liquid metal strip is formed on the roll rim, the working liquid, capillary-closed in grooves, evaporates, and because of the pressure difference between the hot rim and the cold side walls the vapor moves to the latter, where it condenses and is transferred by centrifugal forces to the evaporator along the grooves formed on the inner side of the walls. Heat received by the side walls is released into the air in rotation.

The design procedure consists of determination of the dimensions of the triangular grooves in the condenser and the evaporator and of the maximum width of the metal strip on the basis of the transport capacity of the grooves in the condenser and the intensity of heat release from the rotating finned side walls in interaction with the air [55, 56].

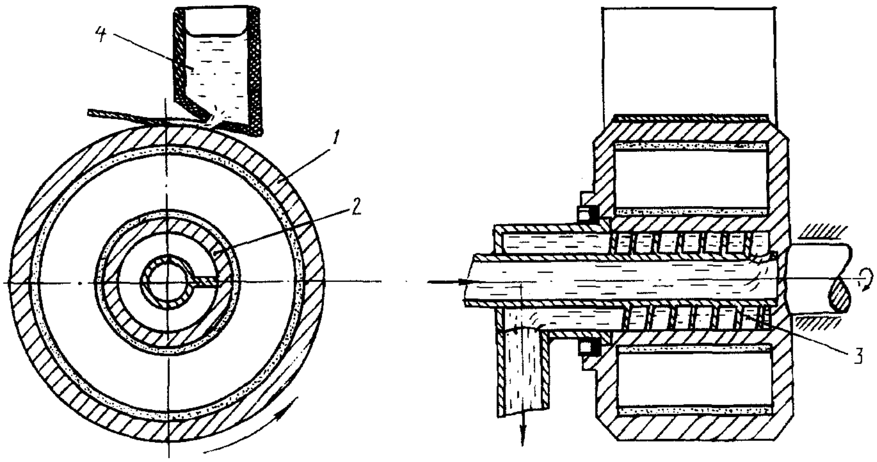


Fig. 8. CHP mold.

The dimensions of the grooves in the condenser are found from the condition of their total draining due to centrifugal forces in rotation. The groove width is found from the equality of the pressure due to the centrifugal force and the hydraulic resistance determined by the Hagen-Poiseuille equation:

$$t_{cn} = \frac{(1 + \sin \beta_{cn})}{2\pi \cos \beta_{cn}} \times$$

$$\times \sqrt{\frac{f\mu_l}{\rho_l n_{mo}(1 - n_{mo}\tau_{ms} - n_{mo}\tau_{mo}) \sin \varphi} \ln \frac{R_{in}}{R_{hu}}}\,.$$

The optimal value of β_{cn} is about 20° [57].

The number of grooves cut on the side wall of the mold is:

$$m_{cn} = \frac{60 R_{hu}}{t_{cn}}\,.$$

The width of the melt strip is found from the transport capacity of the system of grooves in the mold:

$$b_1 = \frac{t_{cn}^2 L \rho_l r^* m_{cn} \operatorname{ctg} \beta_{cn}}{4\pi R_{mo} \delta_{\text{ш}} \rho_m (C_m \Delta T_{ms} + kr)}\,.$$

On the basis of the number of grooves in the evaporator and the condenser being the same, the groove width in the evaporator is

$$t_{ev} = \frac{60 R_{in}}{m_{cn}}\,.$$

The half-angle at the vertex is determined from the condition of equal volumes of a groove in the evaporator and two grooves in the condenser:

$$\beta_{ev} = \operatorname{arcctg} \frac{2 t_{cn}^2 L \operatorname{ctg} \beta_{cn}}{t_{ev}^2 b_1}\,.$$

The minimum temperature drop between the vapor and the condenser wall is found from the condition of return of the working-liquid vapors from the evaporator to the condenser against the action of the centrifugal forces:

$$\Delta P = \frac{2}{3} D_{se} \rho_v a_c\,.$$

The coefficient of heat transfer between the air and the fins on the rotating inclined side surface is determined [58] as

$$\alpha_{1a} = 0{,}078 \rho_a (\pi n_{mo} L \sin \varphi)^{0,75},$$

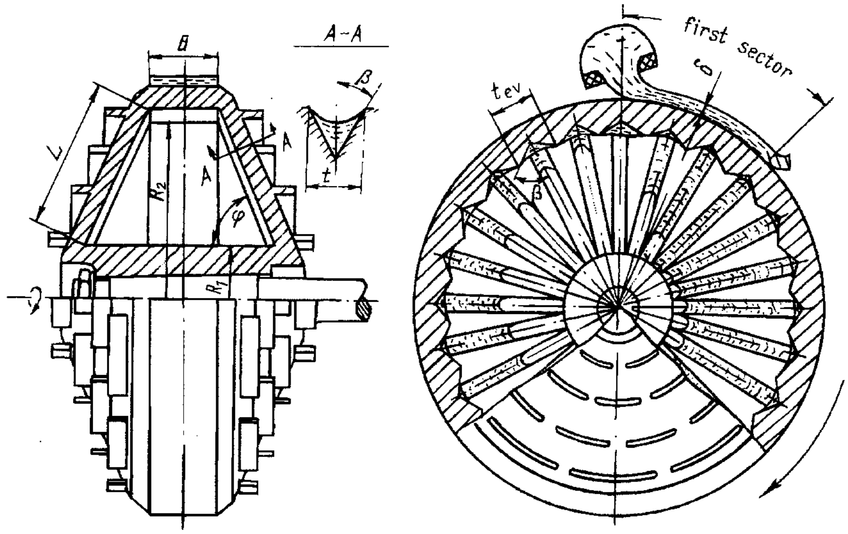


Fig. 9. CHP mold cooled by air.

and the coefficient of heat transfer between the rotating inclined side surface and the air [59] is

$$\alpha_{2a} = 0{,}33\lambda_a \left(\frac{2\pi n_{mo} \sin\varphi}{\nu_a}\right)^{0,5}.$$

The maximum permissible specific heat flux in the evaporator [60] is

$$q_{max} = \frac{t_{ev}^2 r^* \rho_l \cos\theta_l \, C(\beta)\, K(\beta)}{\mu_l \dfrac{b}{2}\left\{\dfrac{(0{,}01 + b/2)}{\ln\dfrac{0{,}01 + b/2}{0{,}01} - 0{,}01}\right\}}.$$

Calculations and experiments have shown that an air-cooled mold allows molding of wire of size 0.5 to 1 mm, which may be used for metal cord. Steel wire needs heat treatment, viz., patenting, usually done in a lead-plating bath. However a simpler decision is the patenting of wire obtained on molds with two rotating rolls having a closed evaporation-condensation cycle proceeds [61].

As pointed out above, a mold with finned side walls allows one to obtain workpieces with a small useful cross section. To increase the efficiency, a mold is recommended to use a mould (Fig. 10) [62] which consists of a single unit of centrifugal radial and axial heat pipes. A roll on which a liquid metal strip is cooled pertains to a radial CHP. To enhance phase transition processes, capillary structures are provided on its rims. The radial CHP is interlinked with the axial CHP, whose outer wall serves simultaneously as a rim of the radial CHP. On the inner part of the axial CHP, transport and phase transitions of the working liquid take place in the capillary structure, and the outer walls are provided with fins to release heat into the air in rotation.

With a melt being solidified on the roll, heat is transferred in the radial CHP to the inner rim, thus heating it, due to phase transitions. In the axial CHP, this heat is transferred to the wall finned on the outside and is released into the air.

Since the mold consists of a CHP unit, viz., radial and axial heat pipes, the latter may be filled with working liquids having different freezing points. Thus, the temperature range in which the mold may be used is extended.

The procedure of design of the mold is based on the choice of the capillary structure (usually grooves of different profiles) and calculation of its dimensions [63].

In the evaporators, triangular grooves, as those providing the minimum grooving pitch, are used, and in the condensers rectangular grooves are used, since the condensation process proceeds on the front part of the connecting

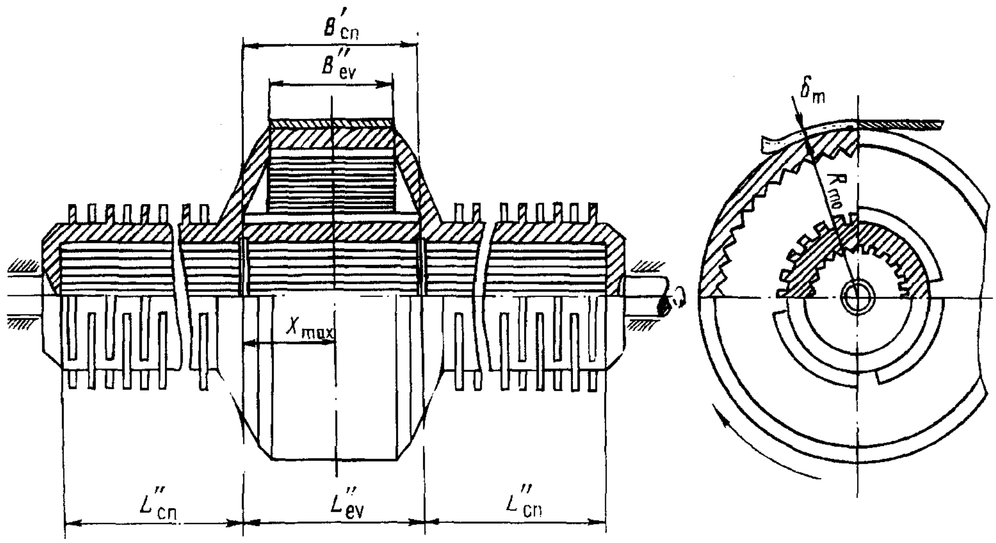


Fig. 10. An air-cooled mold consisting of a unit of centrifugal radial and axial heat pipes.

necks of the grooves. The dimensions of the triangular grooves in the evaporator of the radial CHP are determined from the condition of placing the required amount of the working liquid. Their width is

$$t' = \frac{2W'}{\pi R'_{ev} B'_{ev} \operatorname{ctg} \beta'_{ev}},$$

and the number of them is

$$m_{ev} = \frac{2\pi R_{ev}}{t'} .$$

The dimensions of the rectangular grooves in the condenser are determined from the condition of ejection from them by centrifugal forces of working-liquid drops retained by surface tension forces:

$$b'_g \geqslant \frac{-h'_g \pm \sqrt{(h'_g)^2 - 4\pi K}}{2\pi}, \quad K = \frac{2\sigma_l}{\pi^2 n^2_{mo} R'_{cn} \rho'_l} .$$

The width of the connecting necks between the grooves and the number of grooves are

$$s'_g = \frac{4,25 K Q_r}{W' \alpha'_{cn} \Delta T_{4\min}}, \quad m_g = \frac{2\pi R'_{cn}}{b'_g + s'_g} .$$

In the axial CHP, the width of a triangular groove [64] may be determined from the following expression:

$$q_r = \frac{(t'')^2 (r^*)'' \rho''_l \sigma''_l \cos\theta''_l C(\beta) K(\beta)}{\mu_l x_{max} \left[\dfrac{0,01 + x_{max}}{\ln(1 + 100x_{max})} + 0,01 \right]} + \frac{(t'')^5 a''_c C(\beta) K(\beta)}{2,3 x^2_{max} \mu_l \left[\dfrac{0,01 + x_{max}}{\ln(1 + 100x_{max})} \right]},$$

where

$$q_r = \frac{Q_r}{2\pi R_{mo} B'_{ev}} .$$

The number of grooves in the axial CHP is

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

On the basis of the condition of providing the required amount of working liquid transferred to the evaporator, the dimensions of the rectangular grooves in the axial CHP are calculated as

$$\frac{b''h''}{(b''+h'')}=\frac{fm''\mu''_l L''_{cn}}{\rho_l\left[\frac{2\sigma''_l\cos\theta''_l(1-\sin\beta'')}{t''\cos\beta''}+h''\rho''_l a''_c\right]}.$$

Then the temperature gradient over the mould is calculated; the temperature difference between the evaporator and condensation zones must ensure return of the vapor to the condenser.

CONCLUSION

The present review will be of help to engineers in the choice and development of cooling systems of molds and evaluation of the possibility of their application with regard not only for the cost of the mold but also for its operation regime, the variety and dimensions of workpieces, the program and type of production , the presence of the cooling liquid, and the material and technical basis. Only with all these factors correctly evaluated, may one choose a cooling system which enables the mold to provide high efficiency, low cost, and the required quality of the items produced.

NOTATION

A, area, m^2; a, acceleration, m/sec^2; B, b, width, m; C, heat capacity, J/(kg·K); f, resistance coefficient; D, diameter, m; g, gravitational acceleration, m/sec^2; H, h, height, m; K, k, coefficients; L, l, length, m; Δl, free height of the melt, m; M, output, kg/sec; m, number of grooves; $\dot{m}$, mass flow rate of the liquid, kg/sec; n, rotation frequency, sec^{-1}; P, pressure, N/m^2; ΔP, pressure difference, N/m^2; Q, heat flux, W; q, specific heat flux, W/m^2; R, radius, m; r, latent heat of crystallization, J/kg; r^*, latent heat of vaporization, J/kg; s, width of the connecting neck of the grooves, m; T, temperature, K; ΔT, temperature gradient, K; t, groove width, m; t(x), width of the liquid layer in any cross section of a groove, m; W, volume, m^3; x, coordinate, m; Π, porosity; Πp, permeability, m^2; α, heat transfer coefficient, $W/(m^2 \cdot K)$; β, half-angle at the vertex of a triangular groove, deg; γ, angle of onset of gripping of the liquid-metal strip, deg; δ, thickness, m; θ, wetting angle, deg; λ, thermal conductivity, W/(m·K); μ, dynamic viscosity, $N/(sec m^2)$; ν, kinematic viscosity, m^2/sec; ξ, resistance coefficient; ρ, density, kg/m^3; σ, surface tension, N/m; τ, time, sec; φ, angle of inclination of the side walls, deg; Nu, Pe, Pr, Re, Nusselt, Peclet, Prandtl, and Reynolds numbers. Indices: superscripts: ′, radial CHP; ″, axial CHP; subscripts: a, air; in, inner; h, hydraulic; gr, gravitational; l, liquid; ev, evaporator; g, groove; ch, channel; cn, condenser; cp, capillary-porous; mo, mold; cr, critical; s, strip; gb, gate box; sa, saturated (saturation); r, rated; se, separated; co, cooling; v, vapor; b, bubble; s.t, surface tension; re, reduced; ms, melt superheating; m, melt; hu, hub; w, wall; c, centripetal; s1, slot; max, maximum; min, minimum.

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