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DETERMINATION OF THE HEAT TRANSFER COEFFICIENT BY THE INVERSION METHOD

STANOVENÍ SOUČINITELE PŘESTUPU TEPLA INVERZNÍ METODOU

Abstract

The surface of a blank, cast on the continuous casting equipment, is cooled by water nozzles. The intensity of heat dissipation for individual nozzles cannot be practically determined directly on the casting machine therefore it is investigated on a laboratory model of the Department of Thermal Engineering. A physical quantity that characterizes transport of heat from a surface is the heat transfer coefficient whose size depends on several operational parameters. This contribution describes determination of the heat transfer coefficient on a hot physical model. An inversion method, based on the measuring of heat gradient between two thermocouples placed on the body of a hot probe, has been used for the calculation.

Abstrakt

Povrch předlitku, odlévaného na zařízení plynulého lití oceli, je chlazen vodními tryskami. Intenzitu odvodu tepla pro jednotlivé trysky prakticky nelze stanovit přímo na licím stroji, proto je zkoumána na laboratorním fyzikálním modelu katedry tepelné techniky. Fyzikální veličina, charakterizující transport tepla z povrchu, je součinitel přestupu tepla, jehož velikost závisí na několika provozních parametrech. V příspěvku je popsáno stanovení součinitele přestupu tepla na teplém fyzikálním modelu. Pro výpočet je použita inverzní metoda, založená na měření teplotního spádu mezi dvěma termočlánky umístěnými v tělese žhavené sondy.

1 INTRODUCTION

A basic metallurgical process of continuous casting is the transfer of heat from liquid steel by direct or indirect cooling. In the primary area of cooling the main goal is achieving of sufficient thickness and rigidity of solidified casting shell, while in the area of secondary cooling the heat transfer affects origination of external and internal defects and cracks of the blank. Knowledge of the heat transfer coefficient is important for optimum setting of intensity and uniformity of cooling, elimination of cracks and increasing of the quality of cooled blanks. The information obtained about individual types of nozzles will allow the modification of existing or design of new systems of secondary cooling with higher efficiency and lower cooling water consumption.

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2 HEAT TRANSFER IN THE SECONDARY COOLING AREA

The secondary blank cooling area (Fig. 1) is located between the primary area that is represented by a crystallizer and the tertiary cooling area where the blank is freely cooled on conveyors. From the design view this is several steel sections (Fig. 2), on which there are installed cooling nozzles with required spacing.

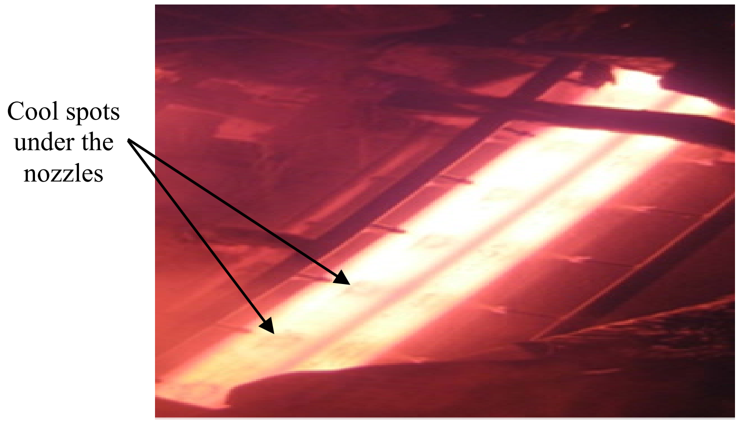

Fig. 1 Secondary cooling area (source J. Kubena)

A blank moves on support rollers. The heat transfer from the blank should be uniform and sufficiently intensive for the casting shell not to break. On the contrary high intensity of the heat transfer causes creation of internal and external defects and cracks [1].

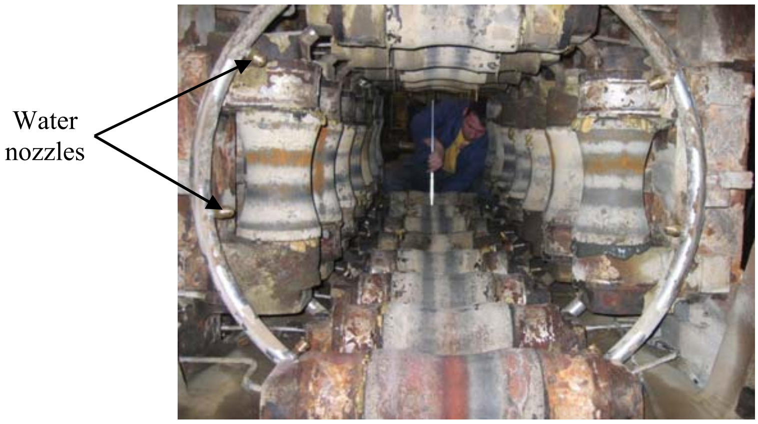

Fig. 2 Cooling section with nozzles and support rollers

Cooling of a blank in the continuous casting equipment (CCE) is done by spraying of cooling water that partially evaporates after falling on the hot surface. Another part of the heat is transferred by conduction to support rollers. Natural convection and radiation occurs in places where the blank does not contact water or the support rollers.

Generally, heat transfer with concurrent phase change is physically more complicated than convection without the water phase change. The amount of heat transfer in this type of cooling depends on the type of boiling of water that falls on the hot surface. If there is a small difference ΔT between the surface and the temperature of cooling medium, than the surface is cooled by convection only without boiling of the liquid. During gradual increase of ΔT boiling centers are created on the surface and so called bubble boiling takes place that transfers a large amount of heat from the cooled surface. A transition boiling regime occurs after reaching of critical density of the heat flow where steam bubbles gradually join together and create an unstable steam membrane. Due to the occurring steam membrane the heat transfer is lower than in the case of bubble boil, since the amount of water on the surface is getting smaller. As soon as a stable steam membrane is created heat transfer starts increasing slowly again. In the area of membrane boiling heat transfer from the surface occurs due to radiation across the created steam layer.

Cooling of the blank in the secondary CCE depends on physical properties of material, type of the cooling medium, pressure and the velocity of the media flow, character and position of the surface in relation to cooling nozzles and the amount of steam in the steam-water mixture.

This convection heat transfer can be expressed using the Newton's equation.

$$P = \alpha_{\mathrm{k}} \cdot \Delta t \cdot S \qquad (1)$$

where:

P – heat flow $[\mathrm{W}]$,

α_{k} – convection heat transfer coefficient $\left[\dfrac{\mathrm{W}}{\mathrm{m}^2 \cdot \mathrm{K}}\right]$,

Δt – temperature difference between blank surface and cooling water $[\mathrm{K}]$,

S – area of the cooled blank $[\mathrm{m}^2]$.

In order to calculate heat flow we need to determine the size of convection heat transfer coefficient that depends on the difference of temperatures between the hot surface and cooling water, water pressure, spray intensity, and diameter and kinetic energy of water droplets. The size of heat transfer coefficient ranges within limits for the given convection heat transfer type. The values of convection heat transfer coefficient are shown in the following table.

Tab. 1 The values of convection heat transfer coefficient α_{k} [2]

Convection type	Coefficient α_{k} ($\mathrm{W.m^{-2}.K^{-1}}$)
Gases in natural convection	5 - 120
Water in natural convection	120 - 1200
Water during change of phase	2300 - 45000

The more intensive flow of liquid around the body surface, the larger the convection heat transfer coefficient. Determination of the coefficient α for a certain case can be performed analytically or experimentally. The analytical solution is complicated and valid for a specific simplified case only. We need to base it on differential equations that describe convection heat transfer (Fourier – Kirchhoff equation, Navier – Stokes motion equations, and continuity equations). These equations are only solvable using limit conditions different for each particular case. The heat flow density and the heat transfer coefficient can be experimentally determined in several ways. The Department of Thermal Engineering uses two methods. The stationary method that lies in the measuring of electric input to a probe needed for maintaining its constant temperature. Knowing the input and heat losses of the system we can calculate the coefficient α. An advantage of this method is the possibility to neglect thermo-physical parameters of the probe material. This contribution describes using a non-stationary method based on knowledge of the heat gradient between two thermocouples placed in the probe body [3].

3 EXPERIMENTAL MEASUREMENT OF HEAT TRANSFER COEFFICIENT

The Department of Thermal Energy uses so called hot physical model to experimentally measure the heat transfer coefficient during cooling of the blank hot surface. The main design element is an indirectly heated probe (Fig. 3) that uses the Kanthal winding to resistance heat up to the temperature of 1100 °C. The probe is placed in a refractory concrete casing, and temperature insulated from its surroundings by cotton-wool and metal covering. Cooling water falls on a round profile surface. The K-type thermocouples are located in the probe 2.5 mm apart. A nozzle is located in a moving mechanism that enables it to move along horizontal and vertical axes. The distance of the nozzle from the probe surface is set on the z axis that is oriented perpendicularly in relation to the measured surface.

The nozzle motion is secured by two motors controlled by a control program on a PC. A nozzle with 65° spray angle and the water pressure of 0.4 MPa was used for the experimental measurement. The nozzle distance from the probe surface was set to 102 mm. The probe heat field was stabilized before spraying and then at the time τ=0 the cooling of the hot probe surface was started [4].

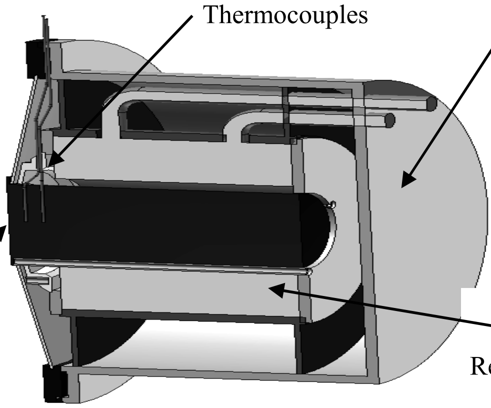

Fig. 3 Schematic of indirectly heated probe

We used the explicit network method and the inverse calculation version to determine the dependency of heat transfer coefficient on surface temperature and time. This calculation includes dependency of the thermo-physical parameters of the probe material on its surface temperature calculated in the previous time step. The calculation time step was set to 0.005 s with regard to numerical stability of the method. The temperature measured in the distance of 4.5 mm from the probe surface has been used for the calculation as a surface condition of the first order, and on the cooling surface side a size of the heat transfer coefficient was selected as the surface condition. The calculation was performed for 20 time steps at a time, and the coefficient α size for the first time interval was estimated. The calculated temperature at 2 mm was compared with the temperature measured by a thermocouple at each time interval. If these temperatures were different the calculation program would modify the size of heat transfer coefficient depending on the temperature difference. As soon as the minimum set difference between the two temperatures was reached, the calculation moved to the next time interval. This procedure was repeated until the final measurement time. A result of this algorithm is the probe temperature field and the heat transfer coefficient values depending on the time of measurement. The average deviation between measured and calculated values at 2 and 4.5 mm does not exceed 0.008 K.

The graphs show the calculated probe temperature field and the dependency of heat transfer coefficient on the temperature of cooled surface during spraying. Temperatures at 2 and 4.5 mm are the measured temperatures that were used to calculate the remaining temperature field and the size of heat transfer coefficient.

During cooling of higher surface temperatures (Fig. 4) lower values of the heat transfer coefficient were achieved in the temperature range from 800 °C to 400 °C, which is caused by the presence of membrane boiling. During the temperature decrease from 400 °C to 300 °C the type of boiling changes from the membrane to the bubble one which increases the heat transfer coefficients from 3 kW.m^{-2}.K^{-1} to 13 kW.m^{-2}.K^{-1}. The increase in heat transfer causes faster temperature decrease in the probe body.

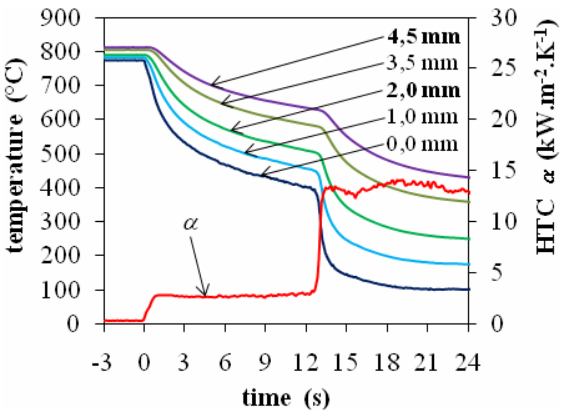

Fig. 4 Temperature field and heat transfer coefficient during surface cooling at the temperature of approx. 770 °C

This phenomenon did not show itself in the cooling of lower surface temperatures (Fig. 5, 6), since the boiling type was considered the bubble one from the beginning of cooling. The shown graphs show stabilization of values of the heat transfer coefficient to approx. 13 kW.m^{-2}.K^{-1}.

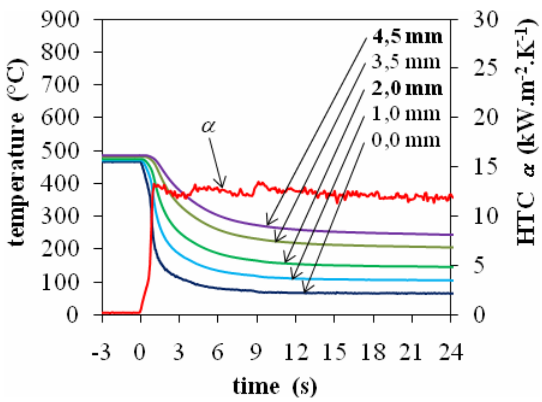

Fig. 5 Temperature field and heat transfer coefficient during surface cooling at the temperature of approx. 770 °C

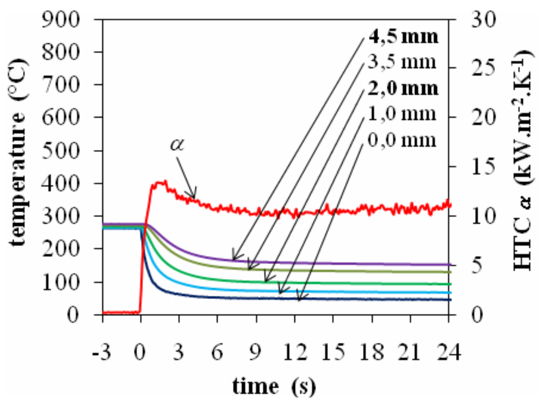

Fig. 6 Temperature field and heat transfer coefficient during surface cooling at the temperature of approx. 260 °C

The Fig. 7 graph describes the dependency of the heat transfer coefficient size on the temperature of cooled surface. A series of experimental measurements was performed for various starting temperatures of cooled surface. For clarity we show the selected curves only. According to measured and calculated results we can show independence of the coefficient α size on the original probe temperature in the temperature interval 800 – 400 °C (the area of membrane boiling). The idealized dependency only shows the step change of boiling from the membrane to bubble one.

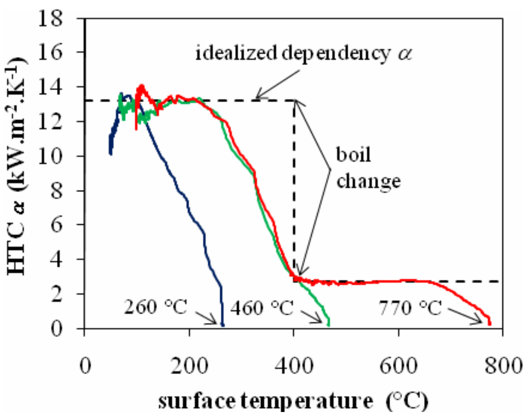

Fig. 7 The relation of the size of heat transfer coefficient and the surface temperature

4 CONCLUSIONS

The research of heat transfer in the secondary area of continuous casting equipment is important from the point of view of high quality of blanks. The cooling of hot surface of a blank is a physically complicated phenomenon that is hard to investigate directly on the casting machine. The Department of Thermal Engineering uses two methods of physical modeling to solve questions related to heat transfer; the one described here – non-stationary method – confirms the influence of water boiling type on the temperature of cooled surface and the value of heat transfer coefficient. A single-component water nozzle with spray angle of 65 ° was used for the experimental measurement. The indirectly heated probe was equipped by two thermocouples that measured temperatures at the locations 2 and 4.5 mm. By using the proposed algorithm we calculated the probe heat field and the size of heat transfer coefficient α.

The maximum difference between the measured and calculated values was only 0.008 K. The influence of Leidenfrost‘s phenomenon on the intensity of cooling was proven at the cooling of high surface temperatures. Transition between membrane and bubble boiling caused the increase of heat transfer coefficient by approx. 10 kW.m^{-2}.K^{-1}. Knowledge of these critical physical phenomena can decrease the occurrence of surface and internal defects and fractures of finished blanks.

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