EXPERIMENTAL STUDY OF HEAT TRANSFER IN HOT ROLLING

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INTRODUCTION

The research is aimed at creating an appropriate methodology and its practical utilization at significant transfer feature and boundary condition settings during roll cooling, product cooling and descaling at hot rolling.

![Figure 1 Schema of hot rolling with significant heat transfer area](image)

The output of experiment is expressed mathematically through functions which define the boundary conditions of the above-mentioned states in a form suitable for numerical models. All data necessary for calculations are obtained from laboratory experiments and sometimes directly from the industry. The stands were developed to simulate the industrial processes and to enable the analysis of different factor impacts on these processes. All data obtained from measurements are processed in several steps. A very exact inverse heat transfer calculation is necessary to set boundary conditions. In numerical models of cooling a boundary conditions are specified either in the form of data files or as a set of interpolation function coefficients.

**Key words:** Heat transfer, rolling, descaling, cooling, nozzle.

**MATHEMATICAL MODELING OF COOLING**

Mathematical models are designed to provide the temperature history of the studied object. The methodology of solving internal heat conduction problem is well understood. The problems can occur in boundary conditions. In the area of cooling the heat flux removed from the surface can be expressed by

\[ q = h(T_S - T_W) + q_R, \]  

(1)

![Figure 2. Variation of HTC with surface temperature](image)

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where $q$ is total heat flux, $h$ is heat transfer coefficient (HTC), $T_s$ is surface temperature, $T_w$ is cooling water temperature and $q_r$ is radiation heat flux. Heat transfer coefficient of spray cooling depends on the surface temperature and characteristics of the spray. In many publications the most important factor, affecting HTC, is the mean quantity of water impinging in the surface per unit area and time – impact density [1]. HTC re-computed from the impact density can be typically used within limited surface temperature range only. Figure 2 shows a typical relationship between HTC and surface temperature. The rapid change in HTC in dependence on surface temperature demonstrates why simple formula cannot be used for all cooling situations.

**DETERMINATION OF HEAT TRANSFER COEFFICIENT**

There are two major methods of HTC determination: a steady state and a transient. In the steady state method the specimen is heated at a prescribed temperature and at the same time is cooled by spray. HTC is computed from heat balance established at a particular temperature. The process must be repeated to cover the whole temperature range. This system needs to be controlled carefully. For the smaller, electrically heated specimen, there are dangers of burnout when the character of heat transfer is rapidly changed.

The method used here is a transient one. Specimen is first heated to a required temperature and then cooled. The whole studied temperature range is covered during one experiment and a realistic space distribution of HTC can thus be evaluated.

**INVERSE HEAT CONDUCTION TASK**

Temperature distribution inside test specimen is described by a differential equation (2).

$$\frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial T}{\partial z} \right) = \rho c \frac{\partial T}{\partial t}$$  \hspace{1cm} (2)

Direct solution of this equation can be done relatively easily when using a numerical technique, with the knowledge of thermo physical material properties and boundary conditions.

Inverse task means, in this case, finding the heat transfer coefficient on the surface body. One or more temperature records at points (sensor positions) inside the body are obtained from the experiment. The method used here is based on a minimization principle [2]. Basic part of this procedure is scheme of heat transfer h computation at time M. An example of this scheme is is shown in Figure 3. The direct computation is branched-out at time M-1. Two different values of HTC are applied for a certain number of forward steps $r$. The HTC h minimizes mean square root deviation between computed $T_{h}^i$ and measured $T_{\exp}^i$ temperature. The derivative of this expression is equated to zero.

$$\sum_{i=1}^{r} (T_{h}^i - T_{\exp}^i) D' = 0$$

Where

$$D' = \frac{T_{h}^i - T_{h}^i}{h_1 - h_2}$$ \hspace{1cm} (4)

Figure 3. Inverse computation scheme
is so called sensitivity coefficient. Then the optimal value of HTC $h^*$ can be expressed as:

$$h^* = \frac{\sum_{i=1}^{r} h_i D_i^2 + \sum_{i=1}^{r} (T^{\text{exp}}_i - T^{\text{h}_i}) D_i}{\sum_{i=1}^{r} D_i^2}$$  \hspace{1cm} (5)$$

HYDRAULIC DESCALING

Hydraulic descaling is the process of removing oxides layer from a hot (typically steel) surface using high-pressure water jet. This process is essential in the hot rolling. The quality of descaling can strongly influence final quality of the rolled surface. The scale is formed during heating in a furnace (primary scale) and during rolling process (secondary scale). Both types of scale must be removed before entering rolling gap. High-energy water beam impacts a scale layer in two ways. The first one is a relatively intensive thermal shock depending on a set of parameters (water pressure, nozzle type, distance from the surface, inclination angle, speed of product moving). The second effect is mechanical and caused by impact pressure. Realistically working numerical model needs realistic boundary conditions [2]. The knowledge of heat transfer coefficient distribution and impact pressure distribution is necessary.

Experiments

The experimental conditions were prepared in such way, which resembles as close as possible to the real mill conditions. There are two basic parameters, which should be kept. The first is the initial temperature of tested sample and the second is the speed of sample motion. A special experimental stand was developed for these tests.

Experimental stand

The experimental stand was built to study the cooling of linearly moving objects. A six-meter-long girder carrying a movable trolley and a driving mechanism (see Figure 4) forms the basic part of the experimental device. An electronic device measuring the instant position of the trolley is embeded in the trolley. The driving mechanism consists of an electric motor controlled by a programmable unit, a gearbox, two rollers and a hauling rope. The girder is divided into three sections. The marginal sections are used for the trolley's acceleration or deceleration. The velocity of the trolley is constant in the mid-section and it is here where the spray nozzles quench the measured sample.
PROCEDURE OF THE EXPERIMENT

- An electric furnace (heater) heats the test plate to an initial temperature of the experiment.
- The plunger water pump is switched on and the pressure is adjusted.
- A driving mechanism moves the test plate under the spray. After recovering the temperature field in the plate, the movement of the plate under the spray is repeated.
- The sensor measures the temperature in nozzle axis position at a certain depth from the cooled surface and the temperature is recorded into datalogger memory.
- The positions of the test plate and the sensors (in the direction of movement) are recorded together with the temperature values. The record of instant positions is used for computation of instant velocities while moving under the spray.

Experiment evaluation

The pass under the nozzle causes temperature drop in the material sample that is indicated by a temperature sensor. An example of temperature record is plotted in Figure 5. This information together with material properties and calibration characteristics of temperature sensor is used as input of inverse heat conduction task. The results of computation are surface temperature, heat flux and heat transfer coefficient. An example of inverse computation output is plotted in Figure 5 (surface temperature) and Figure 6 (HTC).

![Figure 5. Measured and computed surface temperature drop – one path under the nozzle](image1)

![Figure 6. Computed HTC](image2)

The surface temperature drop $\Delta T$ under the jet was used as one of the parameters for comparing particular experiments. This was obtained as a difference between the temperature at the beginning of contact with spray and the lowest temperature after the spray (see Figure 5).

PRODUCT COOLING

Product cooling can by applied between two roll-mill stand (interstand cooling) or at the end of rolling (run-out table cooling). The reason for usage of this type of cooling is decreasing of product surface temperature or heat treatment of final product. The methodology of research is similar to hydraulic descaling. Results of this investigation are typically presented in a form of HTC distribution. Documentation of running experiment is presented at Photo 1. Full-cone nozzles are used in this case. An example of HTC distribution for different spraying parameter is plotted in Figure 7.
ROLL COOLING

Cooling of working rolls during rolling is an important factor which can strongly influence the quality of the process. Properly designed cooling system should guarantee optimal working conditions of the rolls [5]. Heat is transferred from product into rolls in rolling gap. The temperature of the rolls must be held at a reasonable level. Thermal load and operating temperature are in proportion to the roll live. The second important factor is thermal balicity of rolls which influences the shape of the rolls and finally dimensional sharpness of the product.

Water nozzles are typically used in this case. There are many factors which can influence efficiency of the nozzle cooling system.

- Type of a nozzle
- Geometrical configuration (nozzle pitch, distance from the roll, orientation, number of manifolds)
- Coolant pressure and temperature

Cooling intensity is mostly specified through heat transfer coefficient or heat flux distribution. Coolant flow on the roll surface is very complex. No analytical or numerical solution of heat transfer and fluid flow for this case is known. The task can be successfully solved experimentally. An experimental bench which enables realistic simulation of the roll cooling was developed.

Experimental bench

A laboratory experimental device was developed to allow conducting full-scale measurements on roll cooling (see Figure 8). The full-scale tests utilize a complete configuration of rows of nozzles as it is used in plant conditions or prepared by a designer. The basic part of the bench is formed by a roll of 650 mm in diameter and 600 mm in width. There is a segment of the roll equipped by thermal sensors connected to the datalogger. Sensors are individually calibrated. The result of calibration is used in the numerical model of the test segment with sensors. The roll is powered by electrical motor. The speed of rotation can be adjusted in the range from 0.1 to 12 m/s.

Preparation of an experiment starts by heating of the test segment by external electric heater. The roll is stationary during heating. The experiment starts as soon as the temperature of the test segment reaches a uniform starting temperature. The heater is removed, rotation starts and the pump is switched on. Signals from temperature and position sensors are stored into internal datalogger memory. After finishing of the experiment, rotation is stopped and data from datalogger are transferred into computer for further evaluation.
EXPERIMENT EVALUATION

The evaluation procedure is similar to the one used for a linear stand. Temperature records from individual sensors pass through inverse computation and then results are amended into a format convenient for usage as a boundary conditions in numerical models.

Effect of Surface Temperature on cooling of rolls

The surface temperature determines the mechanism of heat transfer. Spray cooling is influenced by the presence of boiling. It could be expected that boiling would support heat transfer, but experiments showed that the reality is quite the opposite.

Significant points on the boiling curve known from pool boiling can be found in spray cooling. The most important point is the Leidenfrost temperature. Spray cooling efficiency strongly depends on surface temperature. A stable vapour layer can be formed at the cooled surface. The stable vapour layer protects the surface from direct contact with the coolant and the cooling becomes of lower intensity. Stability of the vapour layer is coupled to the surface temperature. When temperature decreases and the vapour layer collapses, the cooling instantly grows. The cooling intensity can be ten times higher in a low temperature region as compared to intensity in a high temperature region. The border between these two temperature areas is the Leidenfrost temperature.

Spray cooling experiments starting at surface temperatures of about 300°C are in the area below the Leidenfrost temperature. Initial surface temperatures found at the exit point from the rolling gap fall rapidly. Numerical simulation provides a surface temperature at the position of the first point of contact of a cooling jet at a maximum level of 300°C.
Experiment using two rows of nozzles shows the influence of surface temperature. Figure 9 shows heat transfer coefficient distribution on the cylinder surface. The diameter of the cylinder is 600 mm and circumferential velocity 1 m/s. Horizontal axis in Figure 9 is for a position on the cylinder circumference where zero represents the surface point on the level of roll axis. Position of the first row of nozzles is 100 mm and position of the second row is 500 mm (measured at the roll circumference). Figure 9 shows that the highest intensity of heat transfer was obtained at low surface temperatures, below 100°C. Temperature intervals of 100-200°C provide lower HTC values but the difference is not significant. Increasing surface temperature makes heat transfer less intensive.

The above explains the negative role of vapour formation. Vapour formed when the water makes contact with the hot surface prevents the surface from making direct contact with coolant. A second effect is that impacting water sticks well to a cold surface thereby providing support cooling outside direct impact of droplets. Water mixed with vapour is easily sprayed-out due to centrifugal forces acting on the rotating surface. The conclusions presented here were obtained for water pressure of 5 bar. Figure 9 shows that the decrease in cooling intensity can be observed both in the direct impingement area and in the area influenced only by water flowing at the surface. The function of water pressure on temperature dependent heat transfer should be the subject of further study.

**Intensification of cooling by the use of a two-pressure cooling system**

Cooling systems at hot rolling plants frequently use high flow rates for roll cooling purposes, with maximum values reaching about 10,000 litres per minute on one meter of roll length (l/(min m)). On observing roll usage on site, it can be seen that the cooling water floods the rolls. Sprayed water flows down the roll surface and forms a relatively thick layer. Water jets cooling the upper roll surface at the exit side and near the rolling gap are essential for complete thermal balance of the roll. These jets can be inefficient when spraying through a thick water layer. Improvements in cooling practice cannot be achieved by increasing the flow rate. Increasing total pressure of the cooling system can be difficult therefore a two-pressure cooling system is an option for solution to this problem. Research into the application of the two-pressure system was motivated after the comparison of two measurements. Heat transfer coefficient (HTC) on the circumference of the roll was investigated. The configuration of the nozzles for the experiment is plotted in Fig. 10. The cooling system uses four spray bars in its full configuration. The configuration is designed for upper roll and cooling at the exit side of the rolling mill. The hot roll surface is cooled first by spray bar A and then by spray bar B, spray bar C and finally spray bar D.

The following amount of coolant is applied by the spray bars at a pressure of 4.5 bar: Spray bar A - 150 l/min/m, spray bar B - 870 l/min/m, spray bar C - 870 l/min/m and spray bar D - 1670 l/min/m.

It was expected that part of the water sprayed by bars B, C and D would flow down the roll surface.
The spray jet at position A has to penetrate the layer of water. The first experiment used all four spray bars. The second experiment used only spray bar A.

The resulting cooling intensity measurements are shown in Fig. 11. The horizontal axis in Fig. 11 is for the angle on the roll circumference. Zero is for the position in rolling gap and half of the roll surface is shown. The differences in cooling are significant. The most important factor for our study purposes is the difference in impact position of spray bar A. The impact position is at 70° from the rolling gap. Fig. 11 shows that cooling intensity of bar A is 43% higher when the single spray bar is used. The decrease in the angular position of spray bar A in the experiment using a complete spray configuration is a result of the studied effect; spray bar A lacks sufficient kinetic energy to enable it to cool the roll surface through the water already flowing from the upper bars.

The upward direction of rotation shown in figs. 10 and 12 is to demonstrate the situation at the exit from the rolling gap. The roll surface is first impacted by the spray from bar A and last by spray from bar D. This direction of rotation supplies less water to the impact area of spray bar A, causing lower HTC values here. The HTC distribution obtained for the experiments using water pressure in bar A of 7.5, 80 and 200 bar is shown in Fig. 12.
The curves in Fig. 12 represent the average HTC values for 15 revolutions of the test roll. The surface temperature falls down from 300°C to 80°C during the experiment. Maximum and average HTC values are given in Tab. 1. Table 1 shows an increase in the average HTC values by 38% when only 5.5% of flow rate is used at higher pressure of 200 bar.

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Table 1. Heat transfer coefficients for upward-direction experiments

Numerical models show how this change in cooling affects thermal balance and thermal deformation of roll. Numerical modelling of the thermal behaviour of rolls during rolling campaign can be created using results of the experiments. The HTC distribution on the roll surface shown in Fig. 12 can be used as a thermal boundary condition for the numerical model. The numerical model simulates several hours of roll work and provides information regarding temperature field evolution and changes in roll shape due thermal expansion.

Heat transfer intensity depends on both the magnitude of heat transfer coefficient and the difference in temperatures between the surface and coolant. The surface temperature is highly transient and the temperature at the point the spray hits the surface [4], [5] is of importance.

The following example was selected for the purpose of judging efficiency of the two-pressure system. The heat transfer coefficients in Fig. 12 assume best use of the advantages gained when increasing pressure from spray bar A.

Conditions for the simulation are as follows:

Five hours of roll work is simulated, the rolling campaign is formed by repeating 60 seconds of rolling alternated by a 15 second pause between rolled plates. The roll diameter is 850 mm, roll length 1700 mm, the width of the rolled strip is 1285 mm and the width of the homogeneously cooled area is 1530 mm. The temperature of the rolled strip is 980°C. The thickness reduction in the rolling gap is 6.7 mm and the initial shape of the roll is cylindrical.

The roll crown is most important for the strip profile and the two cases in Fig. 13 are for the roll crown after five hours of rolling. The initial state is “zero” deformation for the initial roll temperature of 20°C.

Full-scale experiments on roll cooling in combination with numerical simulation of rolling provide a tool for cooling design and optimisation. Older experiments confirmed that usage of a superposition for complex sprays composed from simple units (single nozzle or a single bar of nozzles) is not possible. Only the experiments with complete cooling configuration and spraying of a test roll rotating at proper speed, could sufficiently extract a precise description of heat transfer.

Numerical simulations of rolling show a high sensitivity of the roll crown to the distribution of cooling intensity on the circumference of the roll. The main advantage of the two-pressure system can be used when intensive cooling is applied near the rolling gap where spray must penetrate through a layer of water produced by the upper sprays.
CONCLUSION

The experimentally based research enables to specify relatively complicated heat transfer and fluid flow phenomena on hot surface cooled by a nozzle spray. In this case the situation during descaling and roll cooling is studied. Laboratory stands were developed to simulate realistically industrial conditions. Mathematical procedures and precise inverse computation are used for evaluation of experimental results. Final output format of data are boundary conditions which can be used in numerical models of these processes. Numerical simulation can be used for detail understanding, optimization and controlling functions.

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LITERATURE