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INFLUENCE OF THE SAMPLE GEOMETRY ON THE INVERSE DETERMINATION OF THE HEAT TRANSFER COEFFICIENT DISTRIBUTION ON THE AXIALLY SYMMETRICAL SAMPLE COOLED BY THE WATER SPRAY

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Abstract

The paper presents the results of the heat transfer coefficient determination while the water spray cooling process. To determine the boundary condition over the metal surface cooled by water spray the inverse heat conduction problem has been used. In the investigations the axially symmetrical sample has been used as a cooled object. Because of the specific setup of the sensor used in investigations, two finite element models have been tested in the inverse determination of the heat transfer coefficient. The first one, which simplifies the sensor geometry to a cylinder and the second one, that describes the real shape of the sensor. Also, the comparison between two different models employed to determine the heat transfer coefficient over the cooled sample surface have been presented. The boundary condition models differ in description of the function that has been employed to approximate the heat transfer coefficient distribution over the cooled surface in the time of cooling.

Key words: water spray cooling, heat transfer coefficient, boundary inverse problem, finite element method

1. INTRODUCTION

In the metal industry the water cooling is widely used to control the product temperature variation in the production process. Continuous casting lines are equipped with the water spray secondary cooling zones. The main goal in these case is to ensure sufficient heat transfer from the ingot surface to achieve a proper solidification structure. The industrial hot rolling mills are equipped with systems for controlled cooling of hot steel products. In the case of strip rolling mills the main cooling system is situated at run-out table to ensure the required strip temperature before coiling (Tacke et al., 1985; Malinowski et al., 2012). The proper cooling rate affects the final mechanical properties of products which strongly dependent on microstructure evolution processes. Numerical simulations can be used to determine the water flux which should be applied in order to ensure desired product temperature. The heat transfer boundary condition in case of water cooling is defined by the heat transfer coefficient (HTC). Due to complex nature of the cooling process the existing heat transfer models are not accurate enough in the case of high temperature processes common in metal industry. Also, the direct measurements of the HTC by such methods as mass transfer or transient method that uses liquid crystals to measure the surface temperature cannot be used in the case of steel industry processes (Mascarenhas & Mudawar, 2010;

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Liu et al., 2012). For such processes the best way to determine the HTC is to formulate the boundary inverse heat conduction problem (IHCP). There, HTC can be determined as a function of cooling parameters and product surface temperature. In inverse algorithm various heat conduction models and boundary condition models can be implemented. In the paper the results of the inverse calculation of HTC have been presented. The calculations have been performed on the basis of temperature measurements inside selected points of axially symmetrical sample cooled by water spray. The experimental investigations have been conducted for two materials: inconel and brass.

2. BOUNDARY INVERSE MODEL

The HTC on the cooled surface of the cylinder can be determined from the inverse solution to heat transfer problem by minimizing the objective function defined as:

$$E(p_i) = \sum_{m=1}^{Nt} \sum_{n=1}^{Np} (Te_n^m - T_n^m)^2 \qquad (1)$$

where: p_i is the vector of the unknown parameters to be determine by minimizing the objective function, Nt – number of the temperature sensors, Np –number of the temperature measurements performed by one sensor in the time of cooling, Te_n^m – the sample temperature measured by the sensor m at the time τ_n, T_n^m – the sample temperature at the location of the sensor m at the time τ_n calculated from the finite element solution to the heat conduction equation:

$$\frac{1}{r}\frac{\partial}{\partial r}\left[\lambda(\tau) r \frac{\partial T(r,z,\tau)}{\partial r}\right] + \frac{\partial}{\partial z}\left[\lambda(\tau)\frac{\partial T(r,z,\tau)}{\partial z}\right] + q_v - \rho(T) c_p(T)\frac{\partial T(r,z,\tau)}{\partial \tau} = 0 \qquad (2)$$

where: T – temperature, τ – time, r, z – cylindrical coordinates, q_v – internal heat source, λ – thermal conductivity, c – specific heat, ρ – density.

In the finite element model employed to solve equation (2) linear shape functions have been used. Descriptions of the model with linear shape functions has been presented in the paper of Gołdasz et al. (2009).

The heat transfer boundary condition on the cooled surface of the metal cylinder has been expressed as a function of surface radius and time:

$$\dot{q}(r,\tau) = h(r,\tau)(T_s(r,\tau) - T_a) \qquad (3)$$

where: T_s – cooled sample surface temperature, T_a – cooling water temperature, $\dot{q}$ – heat flux, h – heat transfer coefficient.

Variation of the heat transfer coefficient h at the metal surface in time of cooling has been approximated by two HTC models. In the first model (model A) average HTC over the cooled surface as a function of the time of cooling and average sample surface temperature has been determined. In the second model (model B) HTC distribution over the cooled surface has been approximated by the witch of Agnesi type function with the expansion in time of the HTC parameters Cebo-Rudnicka et al. (2012).

3. PROBLEM FORMULATION

In the present study the boundary condition over the surface of the metal sample cooled by water spray has been sought. The sample has a form of a cylinder 20 mm in height and 20 mm in diameter. The top surface of the cylinder has been cooled by the water sprays. The cylinder has been completed with a flange 30 mm in diameter and 1 mm thickness and it has been placed in cylindrical housing. The space between the cylinder and the housing has been filled with air, that allows to reduce the heat losses to the surrounding. However, the flange which allows to join the cylinder and housing, causes that the sample temperature field is not perfectly one dimensional. In figure 1 the schematic illustration of the experimental setup which consists of the cylindrical sensor, flange and housing has been presented. The cylinder with flange, as well as housing have been made from the same material. To measure the temperature inside the cooled sample three fast response, NiCr - NiAl thermocouples have been used. Thermocouples have been placed in the symmetry axis of the cylinder in the distance of: 2, 4 and 6 mm from the cooled surface.

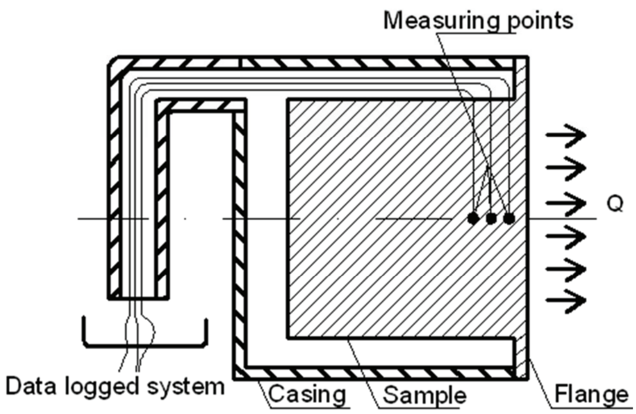

Fig. 1. *Schematic illustration of the experimental setup employed for the determination of the heat transfer boundary condition.*

The experimental tests have been performed for two materials, which differ substantially in thermal conductivity. Inconel and brass samples have been selected for the study. The initial temperature to which the materials have been heated up was 730°C for inconel and 517°C for brass. The water spray pressure in both testes was 1 MPa and water temperature was equal to 20°C. The water flux was 38.6 $kg/(m^2 \cdot s)$ while cooling inconel sample and 1 $kg/(m^2 \cdot s)$ for cooling the brass sample. The temperature measurements logged while experimental tests have been assumed as an input data in inverse calculation of HTC. Because of the sensor construction two finite element models have been tested in the inverse determination of the heat transfer coefficient. The first finite element model simplifies the sample geometry to the perfect cylinder (the simplified model). In the case of the second model the cylindrical sample and the adapter ring (flange) have been described by the finite element mesh (the exact model). Further, two boundary condition models have been employed in equation (3) in order to determine the heat transfer coefficient on the sample surface.

4. RESULTS OF INVESTIGATIONS

The results of the inverse calculations have allowed to determine the influence of the sample geometry description on the heat transfer coefficient identification. In figures 2 to 5 the comparison between HTC variations in the cooling process calculated for simplified and exact description of the sample geometry in the finite element model have been presented. The figures present variations in the average values of HTC (boundary condition model A) versus the time of cooling (figures 2 and 4) and versus the average sample surface temperature (figures 3 and 5).

In case of water spray cooling of inconel sample, the simplification in sample geometry description to the perfect cylinder does not effects the average HTC for the mean sample surface temperature from 730°C to about 250°C (figure 3). This range of temperature corresponds to the film and transition boiling regimes that take place on the sample surface while water spray cooling process. During that processes the vapor film is formed on the cooled surface and it limits the heat transfer between the cooled surface and the cooling water. Additionally, low heat conductivity of inconel causes that heat transfer to the flange is low and does not influence the average HTC in these two boiling regimes. The heat transfer in radial direction to the flange increases while the surface temperature decreases. Below 250°C (figure 3) the heat transfer process changes to the nucleate boiling. That results in significant increase in the HTC values. Simultaneously heat conduction in radial direction is more significant. These to processes affect the inverse determination of HTC. Therefore the sample geometry simplification in the finite element model to the perfect cylinder results in the HTC values about 10 percent higher if compared to those obtained with the real sample geometry description (with flange) in the finite element model of heat transfer (figures 2 and 3). The average difference between the calculated and measured temperatures has been equal to 7.95°C and has not decreased for the better definition of sample geometry (table 1).

Table 1. *The average difference between measured and calculated temperatures at the thermocouples locations.*

Case of study	Inconel sample	Brass sample
	Average difference in temperatures, °C	Average difference in temperatures, °C
Average HTC over the cooled surface calculated for simplified definition of the sample geometry in the finite element model	7.953	4.418
Average HTC over the cooled surface calculated for exact definition of the sample geometry in the finite element model	7.953	3.795
Radial distribution of HTC over the cooled surface calculated for simplified definition of the sample geometry in the finite element model	7.735	5.924
Radial distribution of HTC over the cooled surface calculated for exact definition of the sample geometry in the finite element model	7.735	3.752

The inverse calculations performed on the basis of temperature measurements obtained for the spray cooling of brass sample have indicated a significant influence of the sample geometry description on the average HTC values (figures 4 and 5). The thermal conductivity of the brass is much higher than the inconel one. In such a case heat transfer to the flange is much more important and the exact description of the cooled sample geometry plays an important role in the HTC identification. Neglecting the flange in the definition of the sample geometry has caused that the calculated values of HTC in the whole spray cooling process have been greater for about 25 percent than the ones calculated by using the model with the flange (the exact geometry model) (figure 4 and 5). Moreover, in the case of brass sample the exact definition of the sample geometry (with flange) has lead to the lower difference between measured and calculated temperatures at the thermocouple locations (table 1).

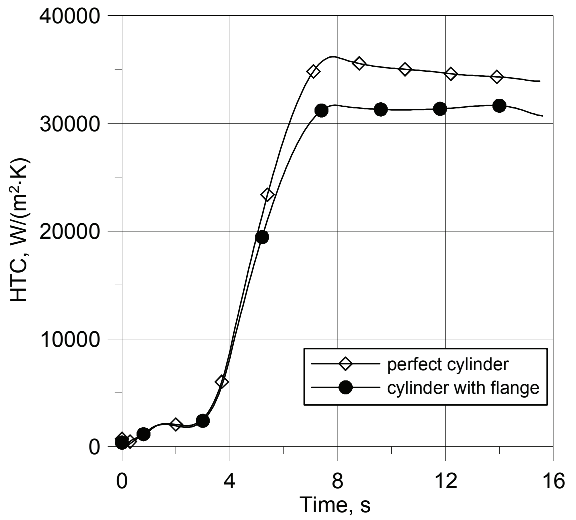

Fig. 2. *The comparison of the average HTC variations in the time of cooling obtained for the simplified and exact definition of the sample geometry in the finite element model. Inconel sample.*

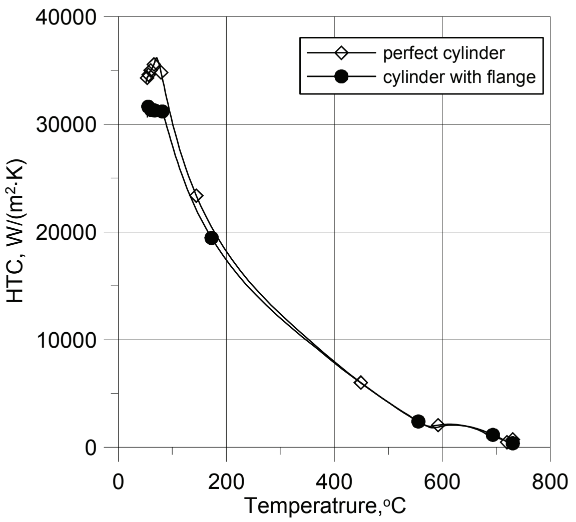

Fig. 3. *The average HTC variations as a functions of sample surface temperature obtained for the simplified and exact definition of the sample geometry in the finite element model. Inconel sample.*

The discussed above boundary condition model gives only average HTC over the cooled sample surface. In practice it is expected HTC distribution over the cooled surface. Such a possibility gives the second boundary condition model. Due to axially symmetrical problem only radial variation of HTC in the time of cooling has been modeled. The analysis has also been performed for two materials: inconel and brass. Further, simplified and exact description of sample geometry in the finite element model has been considered. The results of the inverse calculation of HTC distributions as functions of the sample radius and the time of cooling have been presented in figures to 9 for inconel sample and in figures 10 to 11 for the brass sample.

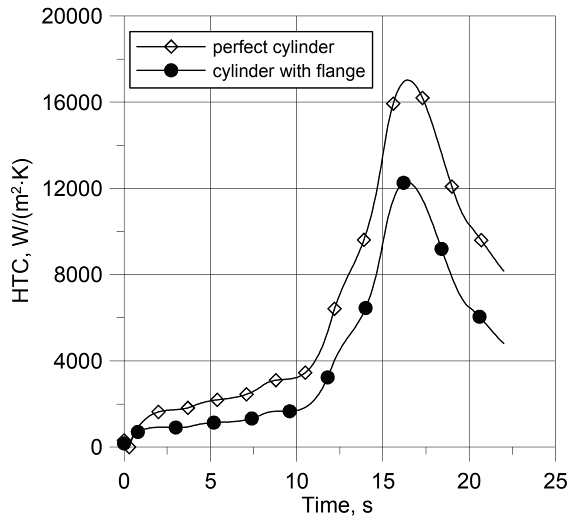

Fig. 4. *The comparison of the average HTC variations in the time of cooling obtained for the simplified and exact definition of the sample geometry in the finite element model. Brass sample.*

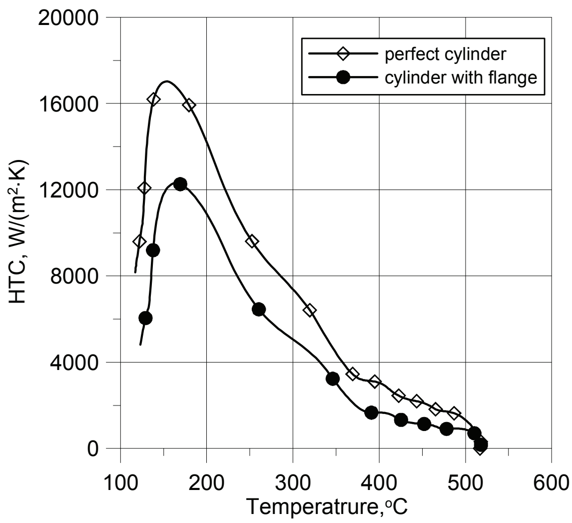

Fig. 5. *The average HTC variations as a functions of sample surface temperature obtained for the simplified and exact definition of the sample geometry in the finite element model. Brass sample.*

The inverse solution to HTC distribution along the cooled sample radius performed for the inconel with simplified definition of the sample geometry in finite element model has not indicated a visible differences in HTC along the radius of the cooled sample surface (figure 6 and 7). Exact description of the sample geometry (with flange) in the finite element model has resulted in lower of about 10% values of HTC (figure 8).

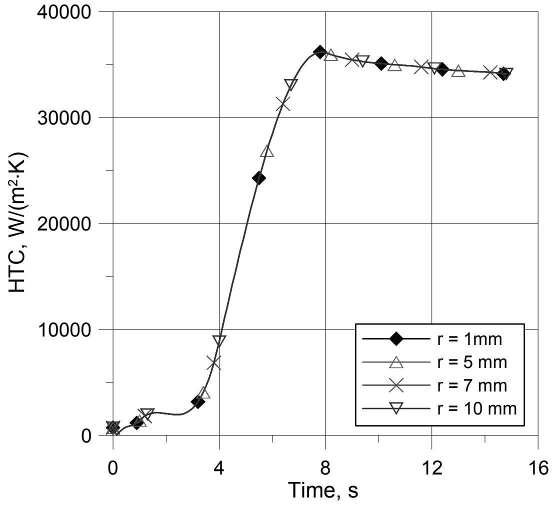

Fig. 6. *HTC variation versus time of cooling for selected locations along the cooled surface radius calculated for simplified definition of the sample geometry in the finite element model. Inconel sample, HTC model B.*

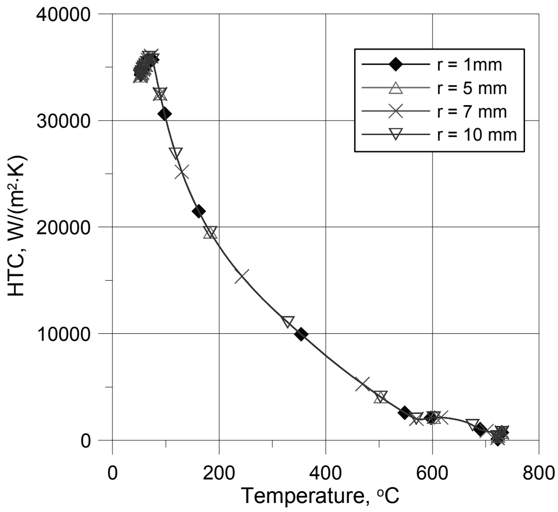

Fig. 7. *HTC variation versus surface temperature for selected locations along the cooled surface radius calculated for simplified definition of the sample geometry in the finite element model. Inconel sample, HTC model B.*

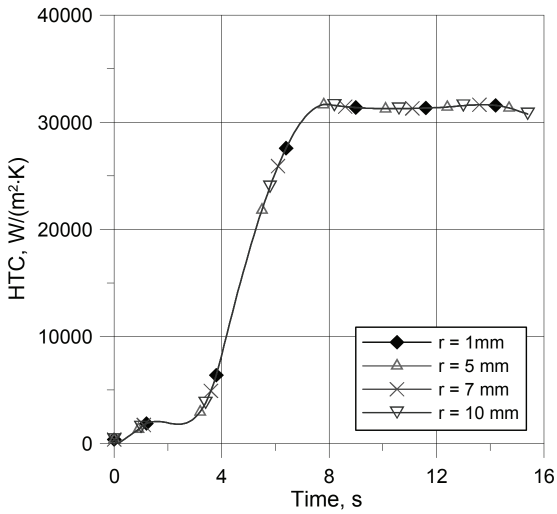

Fig. 8. *HTC variation versus time of cooling for selected locations along the cooled surface radius calculated for exact definition of the sample geometry in the finite element model. Inconel sample, HTC model B.*

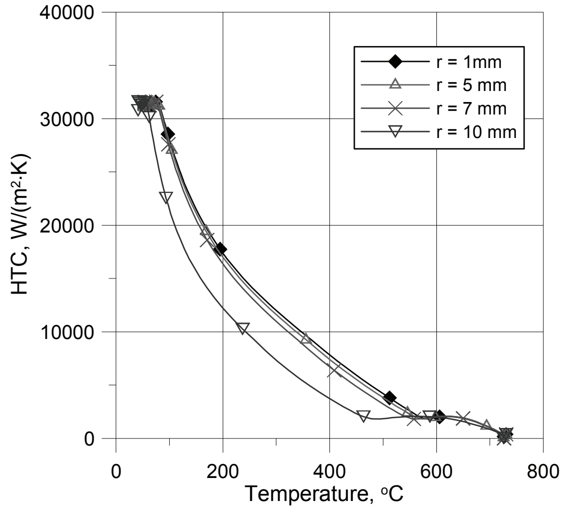

Fig. 9. *HTC variation versus surface temperature for selected locations along the cooled surface radius calculated for exact definition of the sample geometry in the finite element model. Inconel sample, HTC model B.*

Same difference of HTC distribution versus sample surface temperature for the HTC model B has been observed only at the sample and flange connection (r = 10 mm in figure 9). It can be explained by the better description of the sample temperature near the flange by the exact geometry model. Implementation of the HTC model which allows for the distribution of heat transfer coefficient resulted in very similar solutions to the average HTC model. It can be explained by the high water flux applied in the cooling of inconel sample. In such a case sample surface is cooled uniformly. In the case of cooling brass sample the diversification of HTC along the cooled surface radius has been observed both for simplified and exact description of the sample geometry in the finite element model (figures 10 and 11). Implementation of the exact definition of the sample geometry and the HTC variation over the sample surface in the finite element model has allowed to illustrate both the influence of thermal conductivity of sample material as well as the influence of the cylinder flange on the heat transfer between cooled sample and water spray. In the case of simplified definition of the sample geometry to the perfect cylinder in the finite element model the HTC values along the radius of the sample decrease. The greatest difference between the maximum HTC value in the cylinder axis and at the distance of 10 mm from the symmetry axis is equal to about 9 kW/(m^2·K) (figure 10). Implementation of the exact definition of the sample geometry results in much higher diversification of HTC values. In these case the difference between the maximum values of HTC is equal to about 17 kW/(m^2·K) (figure 11). In both considered cases of cooling the metal samples exact definition of the sample geometry (with the flange) in the finite element model have resulted in lower average differences between measured and calculated temperatures (table 1).

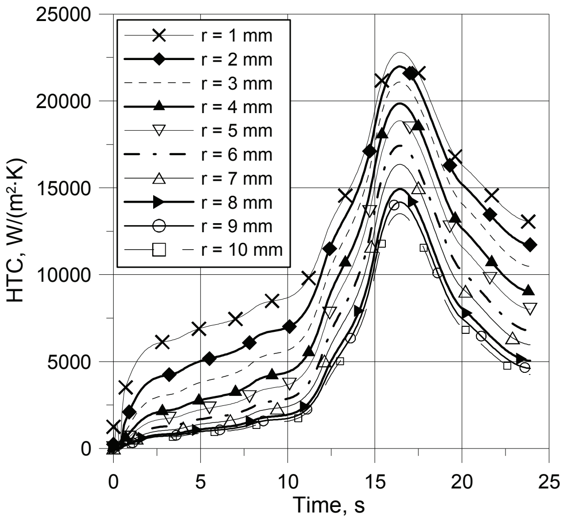

***Fig. 10.** HTC variation versus time of cooling for selected locations along the cooled surface radius calculated for simplified definition of the sample geometry in the finite element model. Brass sample, HTC model B.*

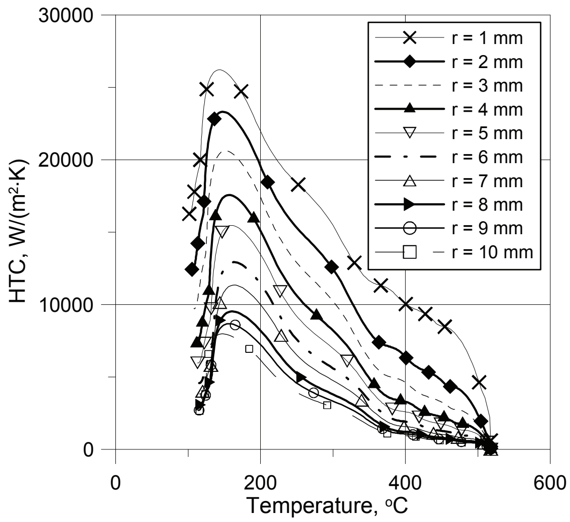

***Fig. 11.** HTC variation versus surface temperature for selected locations along the cooled surface radius calculated for exact definition of the sample geometry in the finite element model. Brass sample, HTC model B.*

5. CONCLUSIONS

The conducted analysis has allowed to determine the influence of the exact definition of the cooled sample geometry in the finite element model on the solution to the heat transfer process. It has been shown that simplification of the sample geometry to the perfect cylinder in the finite element model results in about 10 percent growth in the heat transfer coefficient determined by the inverse method in case of material which is characterized by low heat conductivity (inconel) and about 25 percent growth in HTC value in the case of high conductivity materials (brass). Identification of HTC performed for two boundary condition models has shown that allowing for the HTC distribution over the cooled surface results in more accurate determination of the heat transfer boundary condition. The developed definition of the boundary condition is capable of identification both constant and variable heat transfer coefficient over the cooled surface of the cylindrical sample.

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WPŁYW GEOMETRII PRÓBKI OSIOWOSYMETRYCZNEJ NA WYZNACZANIE ROZKŁADU WSPÓŁCZYNNIKA WYMIANY CIEPŁA PODCZAS CHŁODZENIA NATRYSKIEM WODNYM

Streszczenie

W pracy przedstawiono wyniki obliczeń współczynnika wymiany ciepła wyznaczonego na podstawie badań eksperymentalnych. Do wyznaczenia warunku brzegowego na powierzchni metalu chłodzonego natryskiem wodnym wykorzystano rozwiązanie brzegowego odwrotnego zagadnienia przewodzenia ciepła. Badania eksperymentalne przeprowadzono dla próbki osiowosymetrycznej. Ze względu na specyficzną budowę czujnika wykorzystanego w badaniach, w algorytmie metody odwrotnej przetestowano dwa modele elementów skończonych opisujące geometrię próbki. Pierwszy model upraszczał geometrię próbki do postaci „zwykłego” walca, drugi model opisywał rzeczywisty kształt próbki. W pracy testowano również dwa modele aproksymacji warunku brzegowego.

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