Modeling the thermal state of a strip during laminar cooling

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Abstract. A local physical and mathematical model of the thermal state of the strip during laminar cooling has been developed. The mathematical model is implemented as a software product. The software product was tested under real production conditions based on a comparison of actual and calculated strip temperatures when passing through the first group of coilers after laminar cooling. The standard deviation (1.15 °C), average relative error (1.6%), relative errors (do not exceed 3.7% or 28 °C) were determined. Calculation criteria confirm the adequacy of the developed mathematical model.

1 Introduction

One of the most effective and cheapest ways to increase the strength and toughness characteristics of a metal is strengthening heat treatment using accelerated (controlled) cooling.

Accelerated cooling of a metal sheet by liquid jets is widely used in industry, in rolling production. At the moment, there are many studies of the process [1-8], which use data on heat transfer in the process of accelerated cooling obtained in laboratory experiments and numerical modeling [9-11].

A significant groundwork on this topic in world science allows the use of theoretical and empirical knowledge about heat transfer in real production conditions.

This paper presents a local physical and mathematical model of laminar cooling and checks the adequacy of its operation on a 2000 hot rolling mill.

2 Model of the thermal state of the strip during laminar cooling

or laminar cooling of the strip, the following main heat transfer mechanisms are identified: convective cooling in the zone of collision of the jet with the strip, heat transfer in the zone of film boiling of the liquid, and heat transfer in the zone of convective and radiative cooling in the air. The diagram of the interaction of a freely falling liquid jet with a hot strip surface is presented in Figure 1.

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On the lower side of the strip there is no film boiling zone, since the liquid from the lower nozzle, colliding with the strip, flows along the strip, and then, due to the loss of the vertical component of the velocity, breaks up into droplets and falls down.

The calculation of the parameters of the impinging jet in the model is carried out by equation (1) to determine the pressure in the area of impingement of a round jet, obtained on the basis of experimental data [1,2].

$$P^*(r) = \frac{P-P_0}{P_s-P_0} = 1 - 0{,}61\left(\frac{r}{D}\right)^2, \quad (1)$$

where P_s is the pressure at the critical point; P_0 – atmospheric pressure; r is the distance from the jet axis; D is the diameter of the incoming jet.

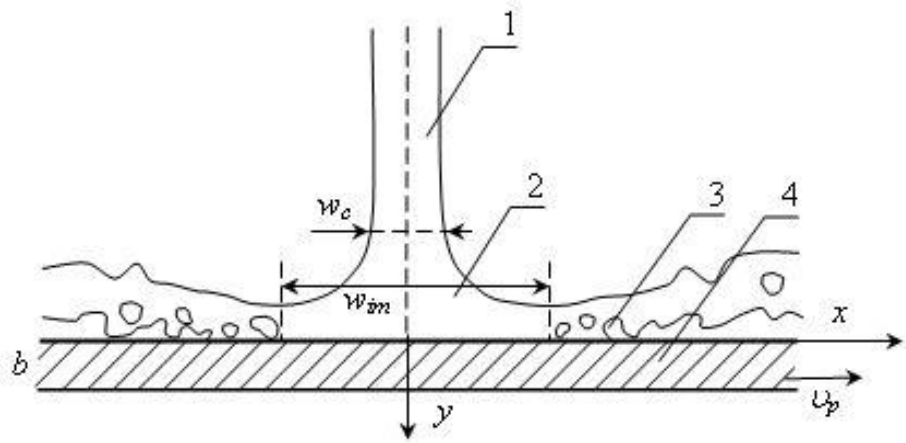

Fig. 1. Diagram of interaction of a jet with a horizontal sheet: 1 – liquid jet, 2 – collision zone, 3 – film boiling zone, 4 – stripe, w_{im} – width of the collision zone, mm

Heat transfer in the collision zone is determined using relation (2), obtained based on the model proposed by Miyasaka [3] and a number of experimental works by other authors:

$$q_{im} = A \cdot 8{,}67 \cdot 10^7 \Delta T_{sat}^{-0{,}385}\left[1 + 0{,}4\left(\frac{v_{im}}{100 w_c}\right)\right], \quad (2)$$

where A = $q_c(\Delta T_{sub})$ / q_c (85°C); $T_{sat} = T_p - T_s$, T_p is strip surface temperature, °C; T_s – water saturation temperature, °C; $\upsilon_{im} = \upsilon_{im}(x)$ – vertical component of the jet velocity in the collision zone, m/s; q_c – ritical heat flux density during boiling of a supercooled liquid in a free volume, W/m^2; ΔT_{sub} – hypothermia of water, °C.

To calculate heat transfer during film boiling, we used the analytical model of film boiling of a liquid on a moving horizontal surface, presented in [3].

The resulting heat equation (3) used in the model is as follows:

$$\frac{\delta(\rho v_p c T)}{\partial x} = \frac{\partial}{\partial y}\left(\lambda \frac{\partial T}{\partial y}\right). \quad (3)$$

Equation (3) is stationary in the reporting system associated with liquid jets. It represents a non-stationary equation of thermal conductivity in the reference frame associated with the strip under the condition of a constant speed of movement of the strip and dx= $\upsilon_p$dt.

Boundary conditions (7) for the heat conduction equation are as follows:

$$-\lambda\left(\frac{\partial T}{\partial y}\right)_{y=0,b} = q(x,z), \qquad (4)$$

where $q(x,z)$ is the local value of heat flow on the surface of the strip, determined by equations (1-2), and also depending on the geometric parameters of the cooling system.

3 Verification of the adequacy of the mathematical model

The mathematical model of the thermal state of the strip during laminar cooling was numerically implemented in the form of a software product (Figure 2).

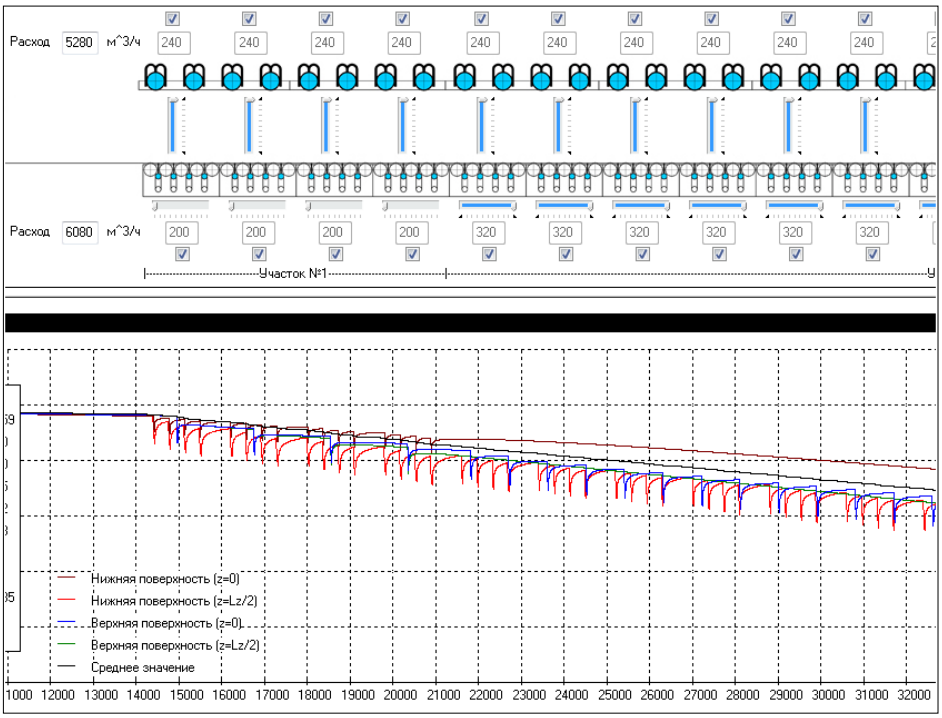

Fig. 2. Window of the program for calculating the thermal state of the strip during laminar cooling of hot rolling mill 2000

To check the adequacy of the model, temperature calculations were carried out in the first group of winders for 50 strips.

Calculations were carried out for the following steel grades: 09G2S, 10ps, 08ps, 20, SPCC, 10ps.

Information on the values of the thermophysical properties of steel grades was obtained from open sources and databases [12-23].

The Figure 3 shows a graph of the correspondence between the calculated and actual temperatures of the strips on the first group of coilers of the 2000 hot rolling mill.

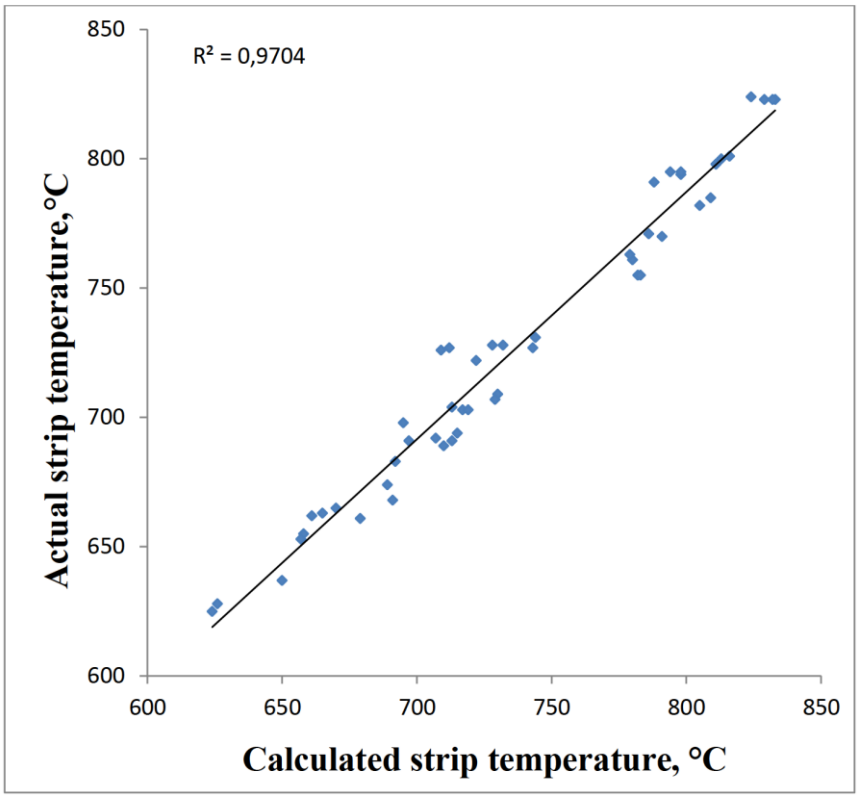

Fig. 3. Graph of the correspondence between the calculated and actual temperatures of the strips when passing the first group of coilers after laminar cooling

The standard deviation of the calculated data from the actual data was 1.15 °C. The average relative calculation error was 1.6%, relative errors did not exceed 3.7% (or 28 °C), R^2 value=0.97. Calculation criteria confirm the adequacy of the developed mathematical model.

4 Conclusion

Thus, a mathematical model has been developed, which is numerically implemented in the form of a software product. A comparison of actual and calculated values showed that the resulting mathematical model adequately reflects the processes occurring in real production conditions.

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