INVERSE ANALYSIS CALCULATION OF HEAT TRANSFER COEFFICIENT FOR FEM SIMULATION OF RAILS HARDENING

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INVERZNÍ VÝPOČET SOUČINITELE PŘESTUPU TEPLA PRO SIMULACI KALENÍ KOLEJNIC

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Abstrakt

Předložený příspěvek je věnován problematice stanovení okrajových podmínek při matematickém modelování přestupu tepla. Konkrétně je příspěvek věnován stanovení součinitele přestupu tepla α při volném ochlazovaní ocelových kolejnic na chladícím loži a při zrychleném ochlazovaní části hlavy kolejnice ponořením do klidné vody. První podkapitola je věnována měření ochlazovacích křivek pro oba režimy tepelného zpracování. K měření bylo využito celkem jedenáct zavrtaných termočlánků. V další části příspěvku je proveden výpočet součinitele přestupu tepla na základě znalosti tepelného toku získaného z analýzy časové závislosti teplotního gradientu mezi dvěmi termočlánky v blízkosti povrchu kolejnice. Matematicko - fyzikálnímu pozadí výpočtu je věnovaná zvláštní podkapitola. Získané výsledky jsou dále porovnány s výsledky pocházející z inverzní analýzy, kdy je na základě známých ochlazovacích křivek inverzním výpočtem s využitím programu na bázi metody konečných prvků (program pro simulaci tvářecích procesů FormFEM) stanovena nová závislost součinitele přestupu tepla α jako funkce teploty ochlazovaného povrchu kolejnice. V textu jsou rovněž diskutovány příčiny rozdílu mezi výsledky (zvláště pak při zrychleném kalení ve vodě) obou metod vzájemně mezi sebou jako i vzhledem k teoretickým hodnotám. Shrnut je zde vliv vznikajícího parního polštáře na povrchu hlavy kolejnice na množství odváděného tepla. Výsledné hodnoty získané pomocí inverzní analýzy byly po další korektuře, vzhledem k teplu vznikajícímu během fázové přeměny, použity pro matematické modelování původního laboratorního experimentu. Výsledné ochlazovací křivky prokázaly výbornou shodu s naměřenými hodnotami.

Abstract

The present paper concerns determination of boundary conditions for mathematical modelling of heat transfer. In particular, it covers determination of heat transfer coefficient α during ambient air cooling of steel rails on cooling bed and during accelerated cooling of a part of rail web by submerging in still water. First sub-section covers measurement of cooling curves for both heat treatment processes. Eleven thermocouples placed in drilled holes have been used for the measurements. The following chapter contains calculation of the heat transfer coefficient based on heat flux obtained by analysis of the time dependence of gradient between two thermocouples placed closed to the rail surface. The mathematical-physical background of the calculation is given in a special sub-section of the study. The obtained results were compared with inverse analysis results. The inverse analysis involved inverse calculation by finite element method (with the aid of FormFEM software for forming process simulation) and processing of measured cooling curves to obtain a new relationship between heat transfer coefficient α and temperature of the rail surface during cooling. The paper discusses the causes of the discrepancies between results (particularly during accelerated cooling in water) of both methods and between the theoretical values. There is a summary of the impact of formation of rail head steam cushion upon the amount of extracted heat. Inversion analysis results corrected with respect to generated phase transformation heat were used for mathematical modelling of the laboratory experiment in the initial setting. Resulting cooling curves were in excellent agreement with measured values.

Keywords: cooling curves, inverse analysis, FEM, heat transfer coefficient

1. Introduction

The paper presents a description of application of measured cooling curves for rails cooled in ambient air and water. The curves were used for determination of boundary conditions (heat transfer coefficient α) in FEM simulation of cooling under various conditions (e.g. non-homogeneous temperature field at the start of cooling, alternating water-air cycles, etc.). The α–coefficient has been determined with the aid of two methods. First one was a direct method based on known thermal gradients near the rail surface. Second method was derived from inverse analysis principles, involving computation of $\tilde{\alpha}$values by an inverse calculation.

2. Material and Experimental Methods

Experimental measurements have been conducted on a 900A-steel rail. Chemical composition of the steel is shown in table 1. The FormFEM program uses chemical composition for computation of thermal properties of the steel.

Table 1 Chemical composition of the examined 900A-grade steel.

Element	C	Mn	Si	Cr	Ni	Mo	W	V
Percentage (wt. %)	0.711	0.89	0.363	0.07	0.04	0.014	0.01	0.004

Experimental measurements of kinetics of the temperature field of the rail during ambient air-cooling were performed in a natural gas-fired chamber furnace. Eleven Ni-NiCr K-type thermocouples have been attached to the face of the rail sample (see Fig. 1). They were inserted into 3 mm diameter holes to depth of 50 mm beneath the surface.

Upon reaching the average temperature of 1,000°C, the rail has been withdrawn from the furnace and placed on a cooling bed where it cooled in air. For first measurement, the rail was standing upright, while for the second measurement it was laid side down on the bed.

Water bath was used as the second cooling medium in subsequent measurements. To avoid water sprinkling of the rail web from the boiling bath, the head of the rail was submerged in the bath to about two thirds of its height. Since the amount of evaporated water during cooling was large, water was continuously pumped into the bath to maintain steady water level.

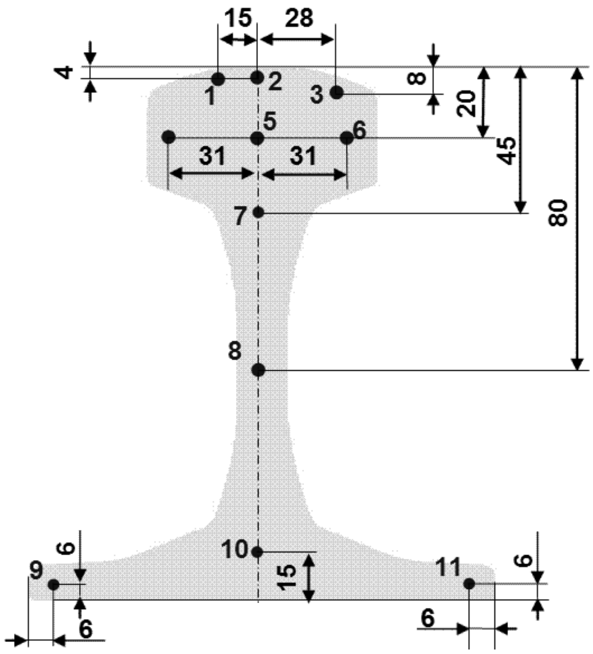

Fig.1 Positions of thermocouples in rail cross-section

3. Measurement of cooling curves of rails

The values measured in the upright rail are shown in Fig. 2 Values measured in the water-cooled rail are shown in Fig. 3. T12 thermocouple measured the ambient air temperature or the temperature of the cooling water).

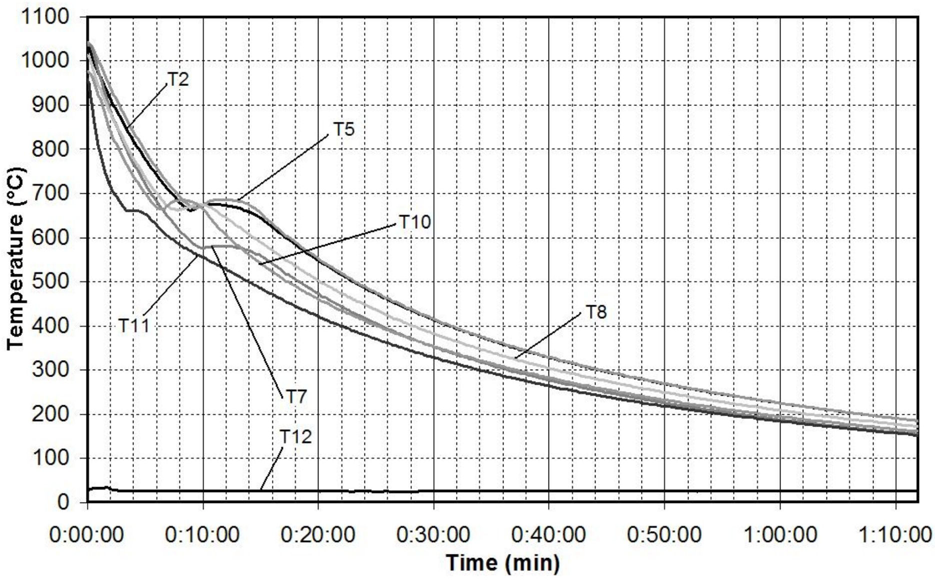

Fig.2 Cooling curves of upright rail in air.

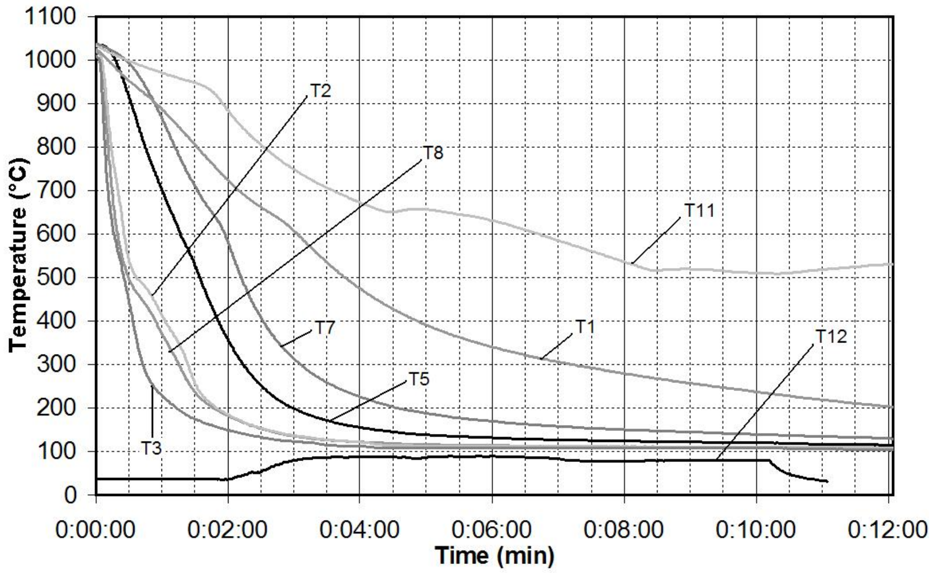

Fig.3 Cooling curves of rail with head submerged in water by 2/3 of the head height.

All air-cooled thermocouples recorded an increase in temperature between 580°C and 670°C due to $\gamma \rightarrow$ pearlite transformation. In third measurement (water), the transformation manifested itself in records of the T7 and T8 thermocouples only through slight shift of cooling curves to the right.

4. Computed Values of Heat Transfer Coefficient

From measured cooling curves the values of heat transfer coefficient have been computed. Determination of α for air was performed with this relationship [1]:

$$\alpha_c = \alpha_k + \alpha_s \quad [\mathrm{W.m^{-2}.K^{-1}}] \tag{1}$$

Where α_k is convective heat transfer coefficient [W.m^{-2}.K^{-1}],
α_s is heat radiation transfer coefficient [W.m^{-2}.K^{-1}].
Determination of heat convection transfer coefficient:

$$\alpha_k = \frac{Nu \cdot \lambda}{l} \quad [\mathrm{W.m^{-2}.K^{-1}}] \tag{2}$$

where $Nu = c \cdot (Gr \cdot Pr)^n$ is the Nusselt number [-], (3)

$Gr = \frac{g \cdot \gamma \cdot \Delta T \cdot l^3}{v^2}$ is Grasshoff number [-], (4)

$Pr = \frac{v}{a}$ is the Prandtl number [-], (5)

λ thermal conductivity coefficient [W.m^{-1}.K^{-1}],
l characteristic dimension [m],
g gravity acceleration [m.s^{-2}],

γ thermal volume expansion [K^{-1}],
ΔT temperature difference [K],
v kinematic viscosity of fluids [$m^2.s^{-1}$],
a thermal diffusivity [$m^2.s^{-1}$].

Table 2 Values of c, n coefficient in equation (3)

Range of validity	c	n
$1 \cdot 10^{-3} < Gr \cdot Pr < 5 \cdot 10^2$	1,180	0,125
$5 \cdot 10^2 < Gr \cdot Pr < 2 \cdot 10^7$	0,540	0,250
$2 \cdot 10^7 < Gr \cdot Pr < 1 \cdot 10^{13}$	0,135	0,330

Determination of heat radiation transfer coefficient:

$$\alpha_s = \frac{C_0 \cdot \varepsilon_n \cdot \left[\left(\frac{T_p}{100}\right)^4 - \left(\frac{T_{ok}}{100}\right)^4\right]}{(T_p - T_{ok})} \quad [W.m^{-2}.K^{-1}], \qquad (6)$$

where ε_n is a relative absorptive index, which is determined according to the general equation:

$$\varepsilon_n = \frac{1}{\left(\frac{1}{\varepsilon_1} - 1\right).\varphi_{12} + \left(\frac{1}{\varepsilon_2} - 1\right).\varphi_{21} + 1} \quad [\text{ - }], \qquad (7)$$

where ε_1, ε_2 are relative absorptive indices for surfaces 1; 2 ,
φ_{12}, φ_{21} are angular heat radiation transfer coefficients between the surface 1 and target surface 2 and between the surface 2 and the target surface 1.

Surface 1 comprises the rail surface: head, flange and sides of the rail. Surface 2 comprises the surrounding environment area or cooling bed area.

The angular coefficient for upper surface of rail head and bottom surfaces of flange of the side-down rail is as follows $\varphi_{12} = 1$. For vertical sides of the rail it has the value of

$$\varphi_{12} = \frac{S_3}{S_1} \quad [\text{ - }] \qquad (8)$$

where S_3 is a fictitious surface defined by length of the rail and distance between the upper edge of flange and bottom edge of head in meters,
S_2 is a rail inner surface area in meters.

Heat radiation of the bottom surface of rail to the cooling bed may be assumed as radiation between two parallel surfaces ($\varphi_{12} = 1$ and $\varphi_{21} = 1$).

The α coefficient for water may be obtained from the balance of heat:

$$\Delta Q_{chl} = \alpha \cdot (T_p - T_v) \cdot S \cdot \tau_{chl} \quad [J], \qquad (9)$$

$$\Delta Q_{chl} = m \cdot c \cdot \Delta t \quad [J], \qquad (10)$$

where T_p surface temperature prior to cooling [°C],

T_v water temperature [°C],
S surface area [m^2],
τ_{chl} cooling time [s].
m mass [kg],
C specific heat of the rolled product [J.kg^{-1}.K^{-1}],
Δt decrease of surface temperature [°C].

Computed values of heat transfer coefficient for ambient air cooling of upright rail and for cooling of the rail head in water are shown in Figs. 4. and 5.

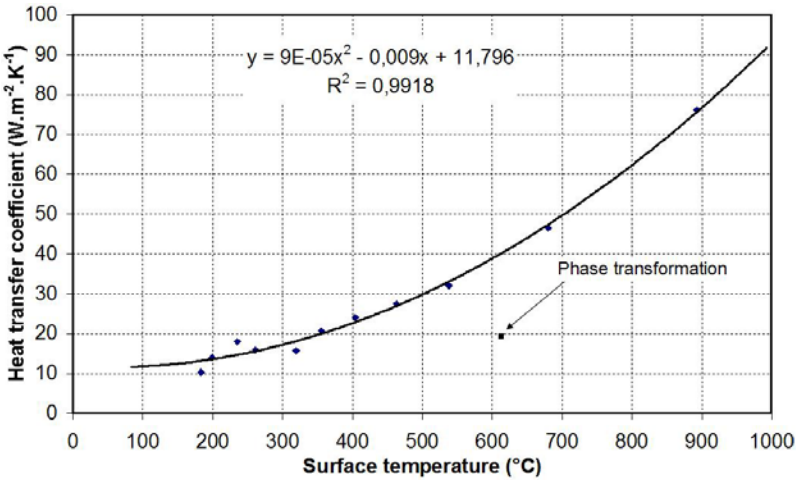

Fig.4 Heat transfer coefficient for upright rail in ambient air.

Resulting relationship $\alpha = f(T)$ for cooling of rail by submerging the rail head in water proves the theoretical assumption that cooling efficiency of water exponentially decreases with increasing temperature of the cooled surface. This is due to bubbles of steam, which are generated on the surface and prevent the contact between water and rail.

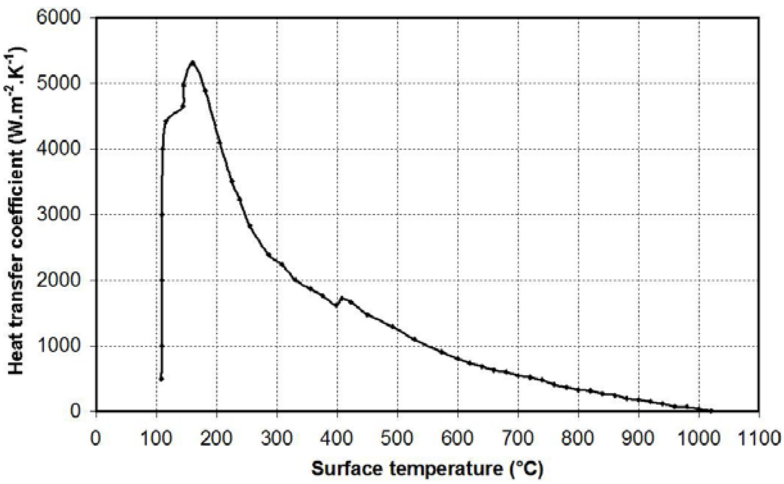

Fig.5 Heat transfer coefficient for still water cooling.

5. Determination of α by means of inverse analysis

Calculation has been conducted with the FormFEM program for simulation of forming processes. Its temperature module is using the boundary condition of 3rd kind. The temperature of environment is known at every time instant and the mathematical relationship for heat transfer between the environment and the surface of the body is given as:

$$-\lambda \frac{\partial t}{\partial x}|x=0 = \alpha \cdot (tp - to) \tag{11}$$

where α is the heat transfer coefficient [W.m^{-2}.K^{-1}].

5.1 Air Cooling

The Form FEM program performs automatic computation of temperature dependence of the heat transfer coefficient according to the equations (1-6). Initial calculation was conducted with environment temperature of T = 22°C and emissivity of 0.8. The program generates resulting temperature fields only as isosurfaces. For further mathematical processing these must be converted from FormFEM-generated text files into cooling curves for pertinent nodes. The mesh is prepared in such way that positions of selected nodes correspond with those of thermocouples (T1 to T11) (cp. Fig. 6 and Fig. 1). Results of temperature simulation are shown in Fig. 7. This figure also shows comparison between the actual cooling curve (recorded by T2 thermocouple for upright rail cooled in ambient air) and the curve computed by FormFEM.

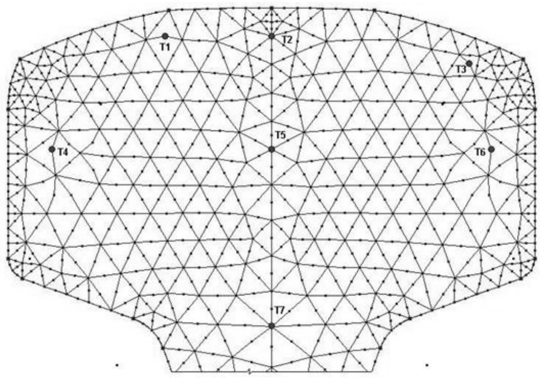

Fig.6 FEM mesh in the rail head. Nodes corresponding with locations of thermocouples are marked.

The $\alpha = f(T_P)$ relationship was corrected according to analysis of computed cooling curves. To maintain its character, the resulting relationship must be of the quadratic type with its minimum corresponding with ambient air temperature. The correction consisted in multiplication with a constant. Ultimately, the influence of angular coefficients from the equation (7) are sought in this way.

In order to correctly describe the cooling curves in the phase transformation region (where the phase transformation heat changes the shape of the curve), complicated correction had to be done. The FormFEM program does not feature computation of phase transformation heat. For this reason, the only way to include transformation effects in the calculation was another modification of heat transfer coefficient. Clearly, this is an irregular modification. It has been implemented as a step change of α at the phase transformation temperature.

The above described steps led to new separate $\alpha = f\,(T_P)$ relationships for head and flange of rail (Fig. 8 shows comparisons with original values). Differences between theoretical and actual values of α are caused by radiation between surfaces of rail. (FormFEM is not capable of including this in the automatic computation). Naturally, there are also effects of radiation between the rail and the cooling bed.

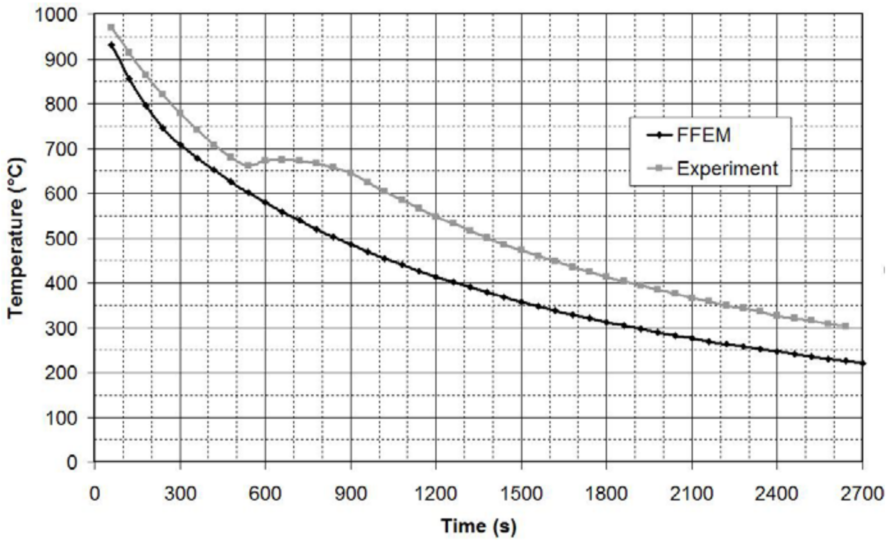

Fig.7 Comparison between results of simulation and actual cooling curves.

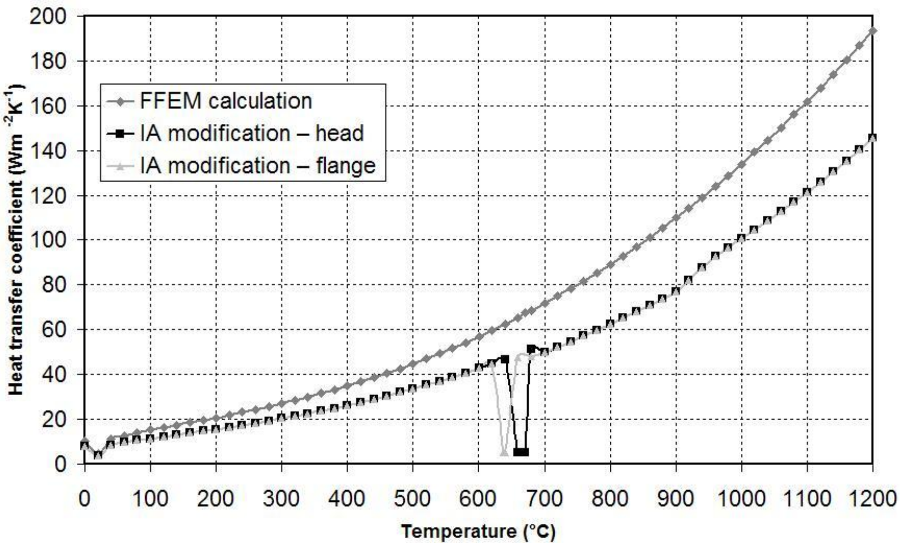

Fig.8 Comparison between the original and inverse-analysis-processed $\alpha = f(T_P)$ relationships.

Results of simulation with the new α and the comparison with the experimental values are shown in Fig. 9. The very good match is evident.

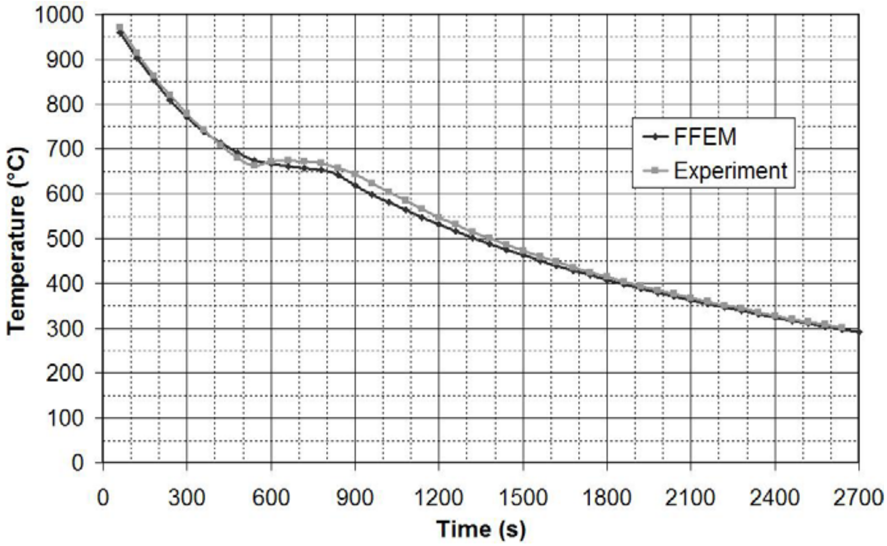

Fig.9 Comparison between the measured and computed cooling curves for T2 thermocouple.

5.2 Water Cooling

The procedure is identical with the one used for air cooling. New $\alpha = f(T_P)$ relationship was thus obtained (see Fig. 10 with comparison of α tabulated values and those derived from measured data). Enormous difference between the tabulated and actual values obtained via inverse analysis (IA) can be explained by the cooling history. The vapour blanket formed on the rail surface at higher temperatures (about 1,000°C) prevents the expected intense cooling.

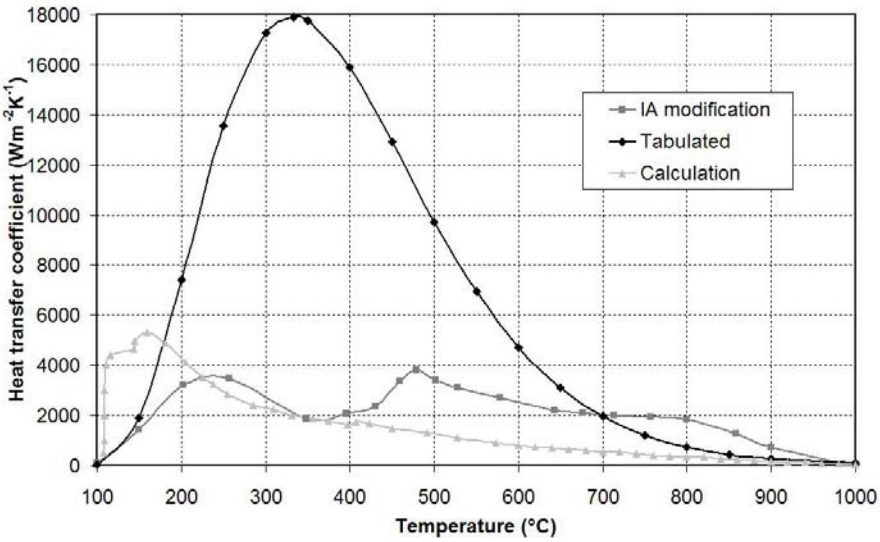

Fig.10 Comparison of values of heat transfer coefficient obtained by inverse analysis and those computed with temperature gradients and tabulated.

6. Conclusion

The above results lead to conclusion that inverse analysis may be used in all cases where determination of $\alpha = f\,(T_P)$ from measured set of cooling curves is needed. In future it may be possible to incorporate the inverse analysis module into the FormFEM software if agreed with the ITA Ostrava company. It would greatly simplify the work of the program‘s operation.

The values of heat transfer coefficient obtained by inverse analysis may be further used either for computation of temperature fields during cooling (further applications may be found in [2]) or for computer simulation of heat treatment containing several cooling stages with different cooling media [3, 4]. This system of computer simulation offers an inexpensive and powerful alternative to the traditional trial and error approach in industry.

Acknowledgments

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