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Thermal Analysis Of Cooling Process Of A High-Temperature Vertical Hollow Cylinder Using A Water-Air Spray

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***Abstract*—**In this paper, a methodology for solution of transient thermal problem is presented using finite-element method. A vertical hollow circular cylinder is heated up to a specific temperature using moving induction heating, and the heated parts then quenched by moving water-air spray. The effects of natural convection with air on the both inner and outer surfaces of cylinder, and also radiation of outer surface of cylinder with ambient, on cooling process are taken into account.

The transient thermal conduction equation is solved to obtain temperature distribution produced by the moving heat source in work-piece over time with considering of moving free and forced (due to spray) convection boundary conditions. This procedure includes temperature-dependent properties.

For quenching of work-piece, a specific kind of atomized spray cooling is used. Spray cooling using a mixture of water and air with different mass fractions; hence, with spray cooling, one can raise or lower the cooling rate by increasing or decreasing the amount of liquid in the mixture.

***Keywords*—** water-air spray, cooling process, finite-element method, vertical hollow cylinder, temperature distribution

I. Introduction

Quenching from high temperatures is usually performed on steel to produce high strength levels. However, in medium carbon, high carbon, and alloy steels, these rapid cooling rates may lead to formation of cracks. Spray cooling offers an attractive alternative to uncontrolled rapid quenching for thermal processing of steels, and widely used in today's industrial operations including surface hardening, melting, brazing, welding, forging and other similar applications. It is used in a variety of engineering and materials processing applications in the automotive, aerospace and some others. With spray cooling, one can raise or lower the cooling rate by increasing or decreasing the amount of liquid in the mixture.

When work-piece is heated to a specified temperature, it must be quenched. The quality of cooling is a specified problem by itself, and is a complex process. Because, as said before, hardness of steel after heat treatment depends on the time of cooling. So, cooling of heated body is an important and sensitive part in heat treatment, and should be studied carefully. Spray cooling using a mixture of water and air was found useful in controlling the cooling rate of hot medium-carbon steel bars. With spray cooling, one can raise or lower the cooling rate by increasing or decreasing the amount of liquid in the mixture. The atomized spray consists of small liquid droplets in a conical jet of air. In this application, water is the liquid of choice because of its low cost. However, many literatures have been made for single-phase cooling [1-3], papers for spray-cooled surface at a high temperature are so limited. Because, at the high temperatures flow becomes multi-phase, and the heat-transfer relations are not well established as they are in single-phase flow. In the other words, Spray cooling is a new discussion in high-heat flux cooling[4]. Buckingham and Haji-Sheikh [5], described the heat transfer characteristics of a spray-cooled surface of cylinder at a high temperature before the onset of surface wetting phenomena. Experimental heat flux data are presented for different liquid mass fractions, and at surface temperatures up to 1000^{o}C. Thomas and Haji-Sheikh [6], presented finite-element modeling and experimental verification of spray-cooling process in a steel cylinder from an initial temperature of 1273K. They also demonstrated the prediction of quench cracks with commercial finite-element analysis (FEA) using available information on a complex heat-transfer phenomenon like spray cooling. The temperature fields predicted by the model are used as an input for the thermal-stress model to predict the occurrence of quench cracks.

In this publication, the analysis of cooling process of a heated cylinder which is heated up using moving heat induction is investigated. It means, the magnetic field is first simulated by means of solution of maxwell's electromagnetic field equations, and the moving heat source is obtained from this magnetic field.

The review of the previous works showed that some studies were conducted on analysis of moving induction heating problem to investigate temperature distribution during the heating process and they didn't engage in subsequent quenching process sufficiently. However, according to the authors' knowledge, there is no study of thermal analysis of moving induction heating with prefect investigation of cooling process. The coupled magnetic and thermal problem in a vertical hollow circular cylinder must be solved, because the material properties in the induction heating depend on the temperature.

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II. GOVERNING EQUATIONS

The basic arrangement of moving induction heating of a vertical hollow cylinder is depicted in Fig.1.

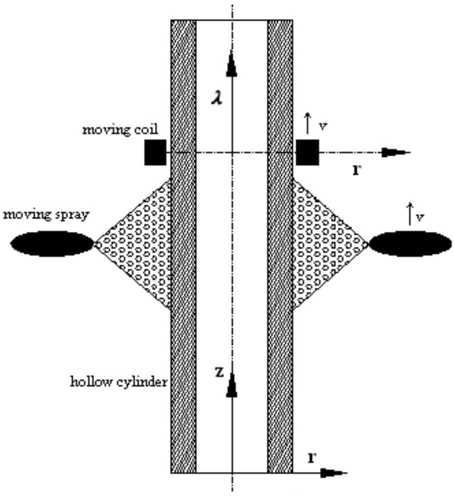

Fig. 1, Physical model of problem

The inductor moves at a velocity of v along the cylinder. The moving spray moves with a specific distance to the inductor at at the same velocity with it to assure consequent fast cooling of the heated part.
Since this investigated arrangement may be considered axi-symmetric, the domain of solution can be modeled as shown in fig.2.

A. Analysis of electro-magnetic field:

To evaluate the heat sources generated within the material during induction heating, it is first necessary to obtain the magnetic vector potential in the work-piece relevant to a current flowing through the induction coil. For this purpose, the maxwell's electro-magnetic field equations must be solved with the appropriate boundary conditions. It for our problem in cylindrical co-ordinates becomes:

$$\frac{1}{\mu}\left(\frac{\partial^2 A}{\partial r^2}+\frac{1}{r}\frac{\partial A}{\partial r}+\frac{\partial^2 A}{\partial z^2}-\frac{A}{r^2}\right)-i\omega\sigma A+J_S=0 \qquad (1)$$

Where A is the magnetic vector potential, J_S is the source current density, μ is the permeability, σ is the electric conductivity and for a sinusoidal source current with a angular frequency of $\omega = 2\pi\ f$.

The boundary condition along the artificial boundary EFGHIJE sufficiently distant from the cylinder and inductor shown in fig.2 [7-8], holds:

$$A = 0 \qquad (2)$$

For the arrangement shown in Fig. 2 and for a delta-function coil of strength J_S , the other boundary conditions are

$$A_1(R_i,z)=A_2(R_i,z) \qquad (3)$$

$$\frac{1}{\mu_0}\frac{\partial}{\partial r}A_1(R_i,z)=\frac{1}{\mu}\frac{\partial}{\partial r}A_2(R_i,z) \qquad (4)$$

$$A_2(R_O,z)=A_3(R_O,z) \qquad (5)$$

$$\frac{1}{\mu}\frac{\partial}{\partial r}A_2(R_O,z)=\frac{1}{\mu_0}\frac{\partial}{\partial r}A_3(R_O,z) \qquad (6)$$

Where the subscripts represent the regions shown in fig.2; and μ_0 is the permeability of air.

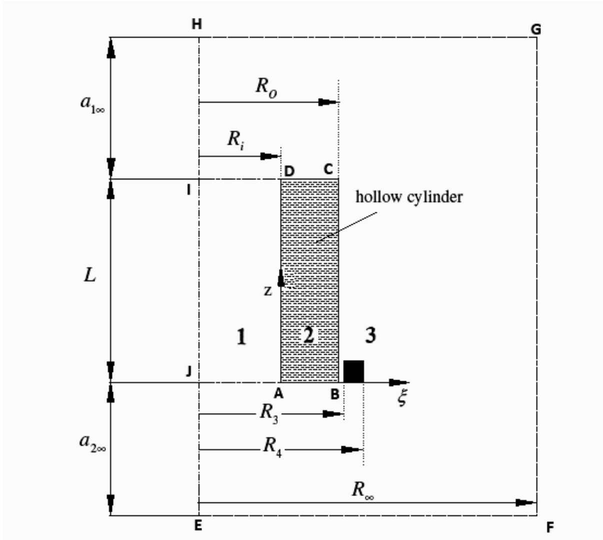

Fig. 2, Domain of solution of problem

Therefore heat source can be expressed as:

$$\dot{g}=\frac{1}{2}\omega^2\sigma AA^* \qquad (7)$$

Where A^* is the complex conjugate of the magnetic vector potential A .

B. Analysis of thermal field

The calculation of the temperature distribution in cylinder during the moving induction heating and subsequent quenching requires a solution of the heat equation with convection and radiation boundary conditions. The transient heat conduction equation for a solid with a heat source of strength $\dot{g}$ is expressed as [9]:

$$\rho C_p\frac{\partial T}{\partial t}=\nabla.(k\nabla T)+\dot{g} \qquad (8)$$

Since our problem is axi-symmetric, Eq.(8) in cylindrical co-ordinates can be written as:

$$\rho C_p\frac{\partial T}{\partial t}=\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{g} \qquad (9)$$

Where k is thermal conductivity, ρ is density, C_p is specific heat.
Ceramic rings were attached to both ends of the cylinder to eliminate heat loss and they works as insulators. Therefore, there is no heat transfer from lines AB and CD.

C. Boundary conditions for internal surface

$$r=R_i\ :\quad -k\frac{\partial T(R_i,z,t)}{\partial r}=(h_c)_1[T(R_i,z,t)-T_\infty] \qquad (10)$$

Where T_∞ is ambient temperature and $(h_c)_1$ is natural convection heat transfer coefficient with air as shown in fig.3.

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The convection coefficient for air cooling was assumed as $(h_c)_1 = 20\ W/m^2.K$.

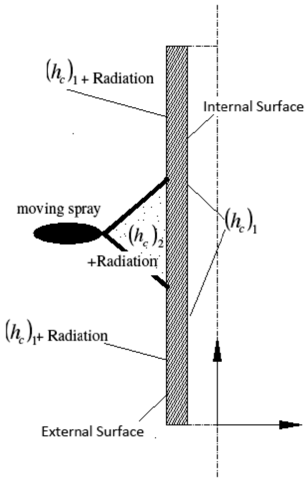

Fig. 3, Presentation of Boundary conditions in problem

D. Boundary conditions for external surface

The cooling process on outer surface of cylinder is established by three different forms. The most important of them that has main parts in cooling process is forced convection produced by moving cooling ring. Some other mechanisms are natural convection with adjacent air and radiation that have less effect on reduction of temperature of work-piece. Hence, outer surface of cylinder can be divided into two different parts in each time step: the region exposed to cooling ring, and the dry region that isn't influenced by cooling ring. Forced convection heat transfer coefficient due to cooling ring is much greater than natural convection heat transfer coefficient with air. As a result in for the region exposed to cooling the effect of natural convection can be neglected. Therefore, the boundary conditions for this region are:

$$r = R_O \quad : \quad -k\frac{\partial T(R_O,z,t)}{\partial r} = (h_c)_2[T(R_O,z,t) - T_j] + h_r[T(R_O,z,t) - T_\infty] \tag{11}$$

$$h_r = \sigma\varepsilon(T^2(R_O,z,t) + T_\infty{}^2)(T(R_O,z,t) + T_\infty) \tag{12}$$

Where $(h_c)_2$ is forced convection heat transfer coefficient for outer surface due to spray cooling that is discussed later with the details. T_j is the jet stream temperature, h_r is radiation heat transfer coefficient that is defined by Eq.(12). σ is the Stefan–Boltzmann constant that is $5.67\times10^{-8}\ W.m^{-2}.K^{-4}$ and ε is the material emissivity coefficient.

The cylinder surface region not exposed to cooling were subjected to natural convection with air during the heating and quenching process:

$$r = R_O \quad : \quad -k\frac{\partial T(R_O,z,t)}{\partial r} = (h_c)_1[T(R_O,z,t) - T_j] + h_r[T(R_O,z,t) - T_\infty] \tag{13}$$

The initial condition for the temperature field within the whole regions of cylinder is:

$$T(r,z,0) = T_0 \tag{14}$$

The calculation of the temperature field within the work-piece can be carried out in a moving co-ordinate system (r,λ) that is attached to the moving coil, where $\lambda = z - vt$. Therefore, the same mesh can be used at each time step. So, in each time step, the position of coil, position of moving spray and regions exposed to natural convection must be updated.

E. Evaluation of forced convection heat transfer coefficient due to spray cooling

The atomized spray consists of small liquid droplets in a conical jet. Since spray cooling process for heated part occurs at high temperatures, the heat transfer establishes in multi-phase flow. In addition, numerical investigation of multi-phase flow heat transfer is very complicated. Therefore, available experimental relations for spray cooling process of steel are used for obtaining forced convection heat transfer coefficient for outer surface due to spray cooling. With considering $BETE\ \ 1/4''\,XA\ PR\ 150B$ manufactured in Bete Fog Nozzle Inc.[6], which internally mixes compressed air and water in an annular region within the nozzle to form a fine conical spray. Schematic view of nozzle is shown in fig.4.

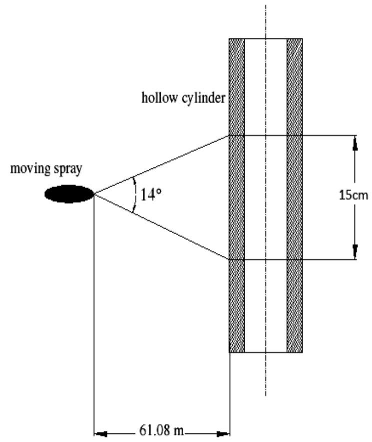

Fig. 4, Schematic view of cooling spray

By increasing or decreasing the amount of liquid in the spray, one can increase or decrease the cooling rate at the surface. With presenting X, that is defined as the ratio of the water-mass-flow rate to the air-mass-flow rate. The heat-transfer process during spray cooling can be divided into three specific regions:

1) Radiation-dominated region:

when surface temperature becomes high enough that radiant energy evaporates all of the water droplets before reaching the boundary layer and the mixture of air and vaporized water will cool the surface by convection. In this region, heat transfer from the cylinder is by radiation and convection to a mixture of air and water vapor as said before in Eq.(23). Hence, a single-phase model can predict the convection heat transfer. The heat-transfer coefficient at the stagnation point can be evaluated using equation for single-phase flow over a cylinder, [9]:

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$$h_{R,0} = \frac{k_m}{2R_O}[82.81 + 0.003014\left(\mathrm{Re}_m \times \mathrm{Pr}_m{}^{0.33}\right) - 3.256\times10^{-9}\left(\mathrm{Re}_m \times \mathrm{Pr}_m{}^{0.33}\right)^2] \tag{15}$$

Where Reynolds number Re_m is defined by:

$$\mathrm{Re}_m = \frac{\rho_a u_0 (1+X)(2R_0)}{\mu_m}\frac{T(R_O, z, t)}{T_{sat}} \tag{16}$$

In above equations k_m , μ_m and Pr_m are thermal conductivity, the viscosity and prandtl number of mixture, respectively. ρ_a is density of air, u_0 is maximum velocity of nozzle at the distance of cylinder, T_{sat} is the saturation temperature.

Bird et al. [10] presented a method for obtaining thermophysical properties of mixture. Also, all properties of air and water considered at mean film temperature that equals $(T(R_O, z, t) + T_j)/2$, where T_j is the jet stream temperature, and can be obtained by the following equation, [11]:

$$T_j = T_\infty + 5°C \tag{17}$$

This condition occurs when $T(R_O, z, t) - T_{\max}$ is large, where $T_{\max}$ is calculated by[5]:

$$T_{\max} = 125 + 0.33X^3 \tag{18}$$

2) Convection-dominated region:

In the convection dominated region, the evaporation of droplets mainly takes place in the boundary layer and heat transfer to the droplets is by convection and water droplets reach the surface of the cylinder and boil away. This region begins when $T(R_O, z, t) < T_{\max}$. The heat-transfer coefficient at the stagnation point in this region can be calculated by [6]:

$$h_{C,0} = \frac{1.9\times10^5 k_a (T(R_o, z, t) - T_j)^{-0.84} X^{0.75}}{2R_0} \tag{19}$$

Where k_a is thermal conductivity of air.

3) Transition region:

A transition region is between two above regions. In this region, the droplets evaporate partially before they enter the boundary layer. The heat-transfer coefficient at the stagnation point in this region is approximated by [6]:

$$h_0 = h_{R,0} + (h_{\max} - h_0)e^{-\frac{h_{\max}}{h_{\max}-h_{R,0}}\left(\frac{T(R_O,z,t)-T_{\max}}{T_{\max}-T_a}\right)} \tag{20}$$

Where $h_{\max}$ is the value of $h_{C,0}$ in Eq.(19) when $T(R_o, z, t)$ equals $T_{\max}$.

Consequently, forced convection heat transfer coefficient for outer surface due to spray cooling can be evaluated from heat-transfer coefficient at the stagnation point for each region as follows, [6]:

$$h = h_0[1 + 0.5(1-B)(\frac{1-4C}{4C} + \cos\theta - \frac{\cos2\theta}{4C})](1 - 0.18Z^2) \tag{21}$$

In Eq.(21):

$$B = 0.385 - 0.0426X_0 \tag{22}$$

$$C = \cos\theta_0 \tag{23}$$

Where,

$$\theta_0 = 1.995 - 0.016X_0 \tag{24}$$

$$\begin{cases} X_0 = X & X < 4 \\ X_0 = 4 & X > 4 \end{cases} \tag{25}$$

In Eq.(21), Z is distance from stagnation point along cylinder. Since domain of spray is $15cm$, we have $-7.5cm \le Z \le 7.5cm$.

Since this problem is axi-symmetric in geometry and all boundary conditions, θ must be eliminated so that the problem remains in axi-symmetric case. In this problem two similar sprayer are placed symmetrically two different sides of the cylinder. As a result, For this reason the average of the heat-transfer coefficients over the range of $0 < \theta < \pi/2$ is calculated by integrating Eq.(21) from $\theta = 0$ to $\theta = \pi/2$ and dividing by $\pi/2$, and this average value is specified over the circumference of cylinder.

The governing equations with appropriate boundary conditions are solved by employing the finite-element method [12]. After discretization, the finite-element formulation obtained from the governing equation with applying appropriate boundary conditions for each element, whereas the entire space is modeled by isoparametric Eight-node elements.

III. RESULT AND DISCUSSION

The numerical solution is validated by a comparison performed for the spray cooling of point “A” between the obtained results and those presented experimentally by Thomas et al. [6], As observed in Fig.5 , they are in good agreement.

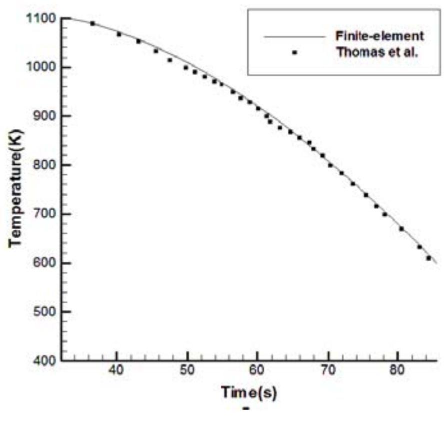

Fig. 5, Comparison between numerical and experimental works

The finite-element procedure was applied to simulate the complete heat treatment cycle including both heating stage and subsequent quenching in an AISI 4140 steel hollow cylinder with the following geometry specifications according to fig.2.

$R_i = 12mm;\quad R_O = 20mm;\quad R_3 = 22mm;\quad R_4 = 26mm;$

$R_\infty = 45mm;\quad L = 40mm;\quad a_{1\infty} = a_{2\infty} = 20mm$

The current density and other field current applied to 4×4 coil of finite cross-sectional area hold:

$J_S = 1.5\times10^{10} A/m^2;\quad f = 50Hz$

Since material properties of work-piece are temperature-dependent, the material properties of AISI 4140 steel are obtained from [13].

Fig. 6 shows the cylinder and the coil and their relative initial positions. The coil and cooling ring move along the cylinder at the same velocity of $v = 2.5mm/s$. The width of the spray cooling is $15cm$ as said before, with the upper boundary of the spray cooling band located at a distance of $2cm$ below the coil center. The cylinder is assumed to be at initial temperature 25°C that is the ambient temperature.

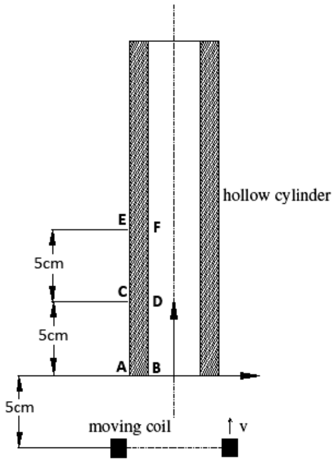

Fig. 6, Geometry of cylinder with initial position of coil

Fig. 7 shows the calculated temperature-time curves relevant to cooling process of six points of cylinder which are specified in fig.6. In this figure the cooling process relevant to spray cooling, natural convection and radiation is investigated. The temperature of each point after reaching at specified temperature is depicted. In the other words, temperature analysis of each point begins while the moving spray approach that point.

Fig. 8 shows distribution of the Local Nusselt number due to spray cooling along the external surface of cylinder at 180s after beginning of the process.

Fig. 9, 10 and 11 shows distribution of the temperature within the axial cut through the cylinder at the time of 100s, 150s and 200s respectively.

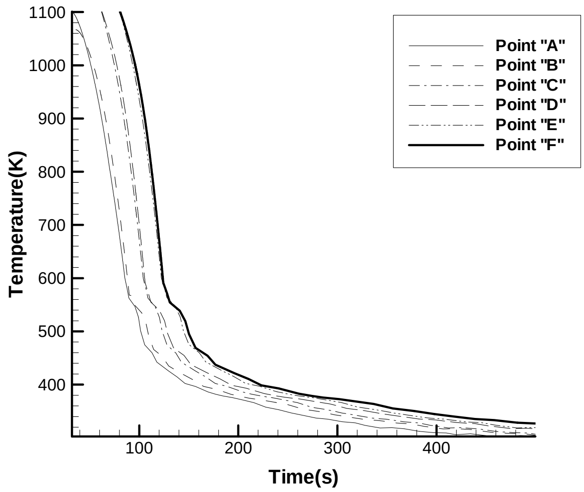

Fig. 7, Temperature-time histories at six points depicted in previous fig.

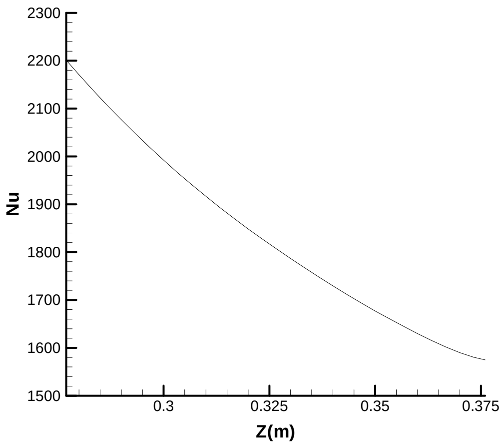

Fig. 8 Distribution Nu due to spray cooling along the cylinder outer surface at 180s after beginning of the process

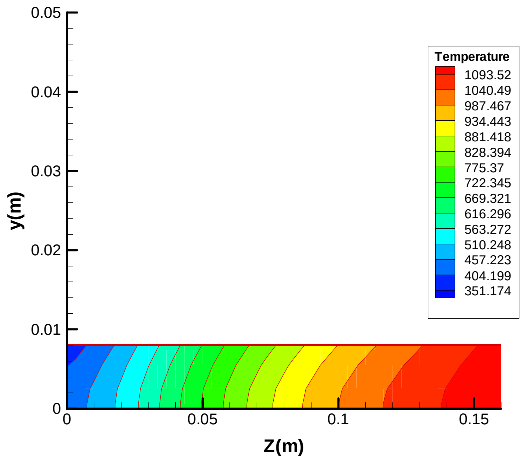

Fig. 9, Distribution of temperature within the axial cut through the cylinder at the time of 100s

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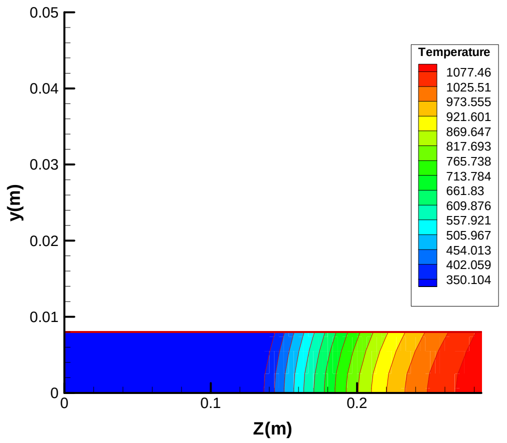

Fig. 10, Distribution of temperature within the axial cut through the cylinder at the time of 150s

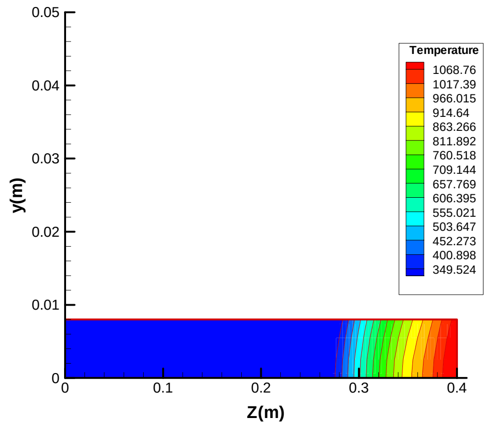

Fig. 11, Distribution of temperature within the axial cut through the cylinder at the time of 200s

At the beginning of cooling stage, since the temperature is high, the heat transfer coefficient due to spray cooling is relevant to radiation-dominated region, and gradually with decreasing of temperature the heat transfer coefficient is evaluated by transient region and convection-dominated region equations. Therefore the heat transfer coefficient due to spray cooling increases with passing the time. Consequently, the slope of the temperature-time curve increases with time until the moving spray completely traverses the specified point. Then cooling of cylinder happens only by natural convection with air. And since the heat transfer coefficient of natural convection is so lower than the heat transfer coefficient due to spray cooling, cooling of cylinder occurs at a much lower rate. This process finishes at the moment when the temperature of the any points of cylinder decreases below $T_{finish} = 100°C$.

IV. Conclusion

The spray cooling process of vertical hollow circular cylinder which is heated at specified temperature using moving induction heating is investigated. . Available experimental relations for spray cooling process of steel are used for obtaining forced convection heat transfer coefficient for outer surface due to spray cooling. Solution to this problem described by the coupled Maxwell's equation (for magnetic analysis) and Fourier's equation (for thermal analysis) with temperature-dependent properties and time-variable boundary conditions. The effects of natural convection with air and radiation on the both internal and external surfaces of cylinder have been taken into account.
The simulation procedure can be used as a useful tool in induction coil design and in the selection of process parameters. The temperature distribution within work-piece at different times and temperature-time curve for different points are presented and discussed in this paper.

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