Heat Transfer Coefficient in Flow Convection of Pipe-Cooling System in Massive Concrete

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Abstract

Pipe-cooling has been widely used for reducing hydration heat and controlling cracking in massive concrete structures. Therefore, the heat transfer coefficients in flow convections, which represent the thermal transfer between the inner stream of the pipe and the concrete, must be estimated accurately. In this paper, a device measuring the heat transfer coefficient is developed based on the concept of internal forced convection. The main influencing factors on the heat transfer coefficient in the flow convection are the flow velocity, pipe diameter and thickness, and the pipe material. Using experimental results obtained from the developed device, a general prediction model for heat transfer coefficients is suggested. The proposed prediction model was found to estimate the heat transfer coefficient correctly with respect to the properties of the flow and pipe in comparisons of measured data and the numerical results of a heat transfer analysis conducted on an actual massive concrete structure.

1. Introduction

Recently, many researchers have focused on analyzing the thermal distribution in concrete structures using the Finite Element Method (FEM) and verifying the results by applying them to actual structures. However, the largest source of uncertainty in this type of analysis is the dependence of model parameters on the thermal transfer. The pipe cooling method, which has been widely used for the reduction of hydration heat and the control of cracking, incorporates these parameters. Therefore, an investigation of the model parameters should be performed prior to the installation of a pipe cooling system.

The heat transfer coefficient in the flow convection is the most important thermal characteristic as it defines the thermal transfer between the inner stream within the pipe and the concrete. In the early 1980s, several researchers in Japan worked to evaluate the heat transfer coefficient. (Kawaraba \textit{et al.} 1986; Tanabe \textit{et al.} 1985) Their studies show that the heat transfer coefficient is dependent on the velocity of the flow. Based on these results, Japan Concrete Institute (JCI) proposed a prediction model of the heat transfer coefficient considering the velocity of the flow. According to this model (Kawaraba \textit{et al.} 1986; Tanabe \textit{et al.} 1985), the heat transfer coefficient is equal to 138-328 kcal/m\textsuperscript{2} h\textdegree C when the flow velocity ranges from 0.2 to 0.6 m/s. In addition, prediction methods which account for the properties of the flow and the pipe have been developed. (Gauthier \textit{et al.} 1982; Tanabe \textit{et al.} 1984) Although there have been numerous studies of pipe-cooling systems in concrete structures, most have focused mainly on systematic methods or on the efficiency of cooling systems while only contending with the heat transfer coefficient theoretically. (Sodha \textit{et al.} 1993; John \textit{et al.} 2003; Mihalakakou \textit{et al.} 1995) Therefore, studies of the heat transfer coefficient in civil engineering in which the prediction of a general heat transfer coefficient achieves a moderate level of success, as in the aforementioned Japanese studies, are rare.

To propose a prediction model for the heat transfer coefficient, it is necessary to understand the thermal characteristics of the flow convection and to provide an experimental method. In this study, parameters for the analysis were chosen based on the concept of the heat transfer coefficient, and a prediction model for the heat transfer coefficient was proposed from the experimental results using a new testing device.

2. Theoretical background

2.1 Heat flux and flow convection heat transfer coefficient

As described in the introduction, a parameter study must be conducted prior to the installation of a pipe-cooling system to evaluate the thermal distributions caused by cooling water. The temperature reduction of concrete by cooling water is due to thermal convection. This is expressed as

\[ q_w = -\rho c_p \frac{\partial T}{\partial t} = h_c (T_0 - T_s) \quad (1) \]

where \( q_w \) is the heat flux applied to the convection sur-
face of concrete (kcal/m² h), $\lambda_n$ is the thermal conductivity of concrete, $\frac{\partial T}{\partial n}$ is the temperature gradient at the concrete surface adjacent to the cooling pipe, $h_w$ is the heat transfer coefficient (kcal/m² h°C), $T_s$ is the nodal temperature of the location at which convection occurs (°C), and $T_w$ is the temperature of the cooling water (°C). It is important to note that this equation applies to the convection between the cooling water and the concrete.

In Eq. (1), the thermal transfer due to flow convection is proportional to the heat transfer coefficient. Therefore, the heat transfer coefficient is an important parameter that represents the heat transfer between the inner stream within the pipe and the concrete.

Kawaraba et al. (1986) and Tanabe et al. (1985) respectively proposed the following prediction models, which are subject to the velocity of the flow for the heat transfer coefficient:

$$h_w = 43.0 + 4.75u_w \quad \text{for} \quad 20 \text{ cm/s} \leq u_w \leq 60 \text{ cm/s} \quad (2)$$

$$h_w = 28.5 + 5.15u_w \quad \text{for} \quad 15 \text{ cm/s} \leq u_w \leq 100 \text{ cm/s} \quad (3)$$

where $u_w$ is the velocity of the flow (cm/s).

However, Eq. (2) and Eq. (3) show that the models only consider the flow velocity as the affecting factor. Therefore, a new model that works as a function of various factors and a test method that can easily measure the heat transfer coefficient are needed.

### 2.2 Heat transfer coefficient using the concept of internal forced convection

In a pipe-cooling system, the heat transfer coefficient is expressed in terms of the internal forced flow. In this section, the concept of internal forced convection is discussed in detail to propose a model and formulate equations. Fig. 1 shows the thermal boundary that develops within a pipe when a fluid stream with a uniform temperature enters the pipe. If the surface condition of pipe is fixed at a constant temperature or heat flux, the condition of inner stream becomes a fully developed state. The shape of the temperature distribution $T(r,x)$ depends on the constant surface temperature or heat flux condition while the fluid temperature increases for both states.

#### 2.2.1 Mean temperature

To estimate the internal forced flow, the concept of the mean temperature is used. The mean temperature in a given section is defined as the transferred heat energy when the fluid bypasses the section. The rate of this energy transfer $E'_c$ is acquired by integrating the mass velocity $\rho u$ and the internal energy $c_v T$, as

$$E'_c = \int \rho u c_v T dA \quad (4)$$

where $E'_c$ is the rate of energy transfer (kcal/s), $\rho$ is the unit mass (kg/m³), $u$ is the flow velocity (m/s), $c_v$ is the specific heat (kcal/kg °C), and $T$ is the temperature (°C). (Bejan 1993; Lee et al. 1992)

Eq. (4) can be rearranged using mean temperature, as

$$E'_c = m' c'_v T_m \quad (5)$$

with

$$T_m = \frac{\int \rho u c_v T dA}{m' c'_v} \quad (6)$$

where $m'$ is the mass flux (kg/s) and $T_m$ is the mean temperature (°C).

For an incompressible flow within a pipe, the mean temperature is expressed as Eq. (7), which uses the concept of the mean velocity.

$$T_m = \frac{2}{u_m r_0} \int u T r dr \quad (7)$$

where $u_m$ is the mean velocity (m/s) and $r_0$ is the radius of the pipe (m).

#### 2.2.2 Energy Equilibrium

As the internal forced flow is entirely surrounded by the pipe, the variation of the mean temperature $T_m(x)$ along the cross-section and the dependency of the inlet and outlet temperature on the total convection heat transfer $q_{conv}$ is determined by the energy equilibrium. In Fig. 2, the fluid moves with constant flux $m'$, and the convection as quantified by the local surface heat flux rate $q'_c$ occurs at the inner surface of the pipe. Energy transfer by axial conduction and the kinetic and potential ener-
gies of the fluid can be ignored generally. Therefore, the change in the thermal energy and the flow work are major factors in energy equilibrium. The flow work is used in the movement of the flow through a verification surface and is expressed as a function of the pressure of the flow $p$ and the specific volume $\rho$.

By applying a conservation equation to the verification element in Fig. 2 and using the definition of the mean temperature, the equilibrium equation is expressed as

$$dq_{conv} + m'(c_v T_v + pv) - \left[ m'(c_v T_v + pv) + m' \frac{\partial(c_v T_v + pv)}{\partial x} \right] dx = 0$$

$$dq_{conv} = m' \frac{\partial(c_v T_v + pv)}{\partial x} dx$$ (8)

where $q_{conv}$ is the rate of convective heat transfer through the entire pipe and $p$ is the pressure.

As mentioned previously, the rate of the convective heat transfer of a fluid is equal to the sum of the rate of the thermal energy increase and the work of the fluid movement. If the fluid is an ideal gas ($pv=RT_m$, $cp=cv+R$ where $R$ is gas constant) and $cp$ is constant, Eq. (8) can be converted to

$$dq_{conv} = m' c_p \frac{\partial T_v}{\partial x} dx$$ (9)

A special form of Eq. (9) representing pipe conditions is obtained by integrating Eq. (9) from the inlet, $i$, to the outlet, $o$, of the cooling pipe. This is expressed as

$$q_{conv} = m' c_p (T_{in,avg} - T_{in})$$ (10)

where $T_{in,avg}$ and $T_{in}$ are the mean inlet and outlet temperatures of the fluid ($^\circ\text{C}$), respectively.

Using this simple energy equilibrium, the relationship between the three major thermal variables $q_{conv}$, $T_{in,avg}$, and $T_{in}$ is expressed in the form of a general formula which can be used regardless of the temperature and flow conditions of the pipe.

Substituting the rate of convective heat transfer for the derivative element $dq_{conv} = q''P dx$, Eq. (9) is simplified using Newton's cooling law. This is expressed as

$$\frac{dT_v}{dx} = \frac{q''P}{m'c_p} - \frac{P}{m'c_p}h(T_v - T_{in})$$ (11)

where $h$ is the heat transfer coefficient (kcal/m² h $^\circ\text{C}$) and $P$ is the perimeter (m).

### 3. Experimental procedure

#### 3.1 Test variables

From the conceptual reviews on internal forced convection in Chapter 2, it is known that the property of the pipe material and convective condition as well as the property of the flow affect the flow convection heat transfer coefficient. In this study, the pipe geometry and materials, the surface temperature, and the flow velocity are selected as the test parameters, as shown in Table 1.

#### 3.2 Test method

The set-up of a newly developed device measuring the heat transfer coefficient and its thermo-sensors are shown in Fig. 3. The new device is composed of two parts: a constant surface temperature controller and a measurement system. The controller includes a circulator and acryl cells. The circulator has a capacity of 30 liters and a temperature range of -25 $\sim$ 110 $^\circ\text{C}$. To keep the pipe at a constant surface temperature, the acryl cell is connected to the circulator. To monitor the surface temperature of the pipe and the inlet and outlet temperatures of the inner stream, K-type thermocouples were used.

#### 3.3 Analytical method

The thermal transfer procedure of the proposed device consists of three parts: (1) Convection between a circulating fluid within the cell and the outer surface of the pipe, (2) Conduction through the pipe, and (3) Convection between the inner surface of the pipe and the cooling water of the pipe (Fig. 4). In this study, the convective heat transfer coefficient is related to the third step in the procedure.

The total thermal resistance of this procedure as shown in Fig. 4 can be expressed as

$$R_t = \frac{1}{h_o2\pi r_oL} + \frac{\ln(r_o/r_i)}{2\pi k_oL} + \frac{1}{h_i2\pi r_iL}$$ (12)

where $R_{tot}$ is the total thermal resistance ($h^-\text{C}/\text{kcal}$), $r_o$ is the outer radius of the pipe (m), $r_i$ is the inner radius of the pipe (m), $L$ is the length of the pipe (m), $k_o$ is the conductivity of the pipe (kcal/m h $^\circ\text{C}$), and $h_o$ and $h_i$ are correspondingly the heat transfer coefficients at the out-

### Table 1 Test parameters.

<table>
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<tr>
<th>Parameters</th>
<th>Variables for the test</th>
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<td>Properties of the flow</td>
<td>Velocity</td>
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<td>Surface condition</td>
<td>Temperature</td>
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<td>40 $\sim$ 60 $^\circ\text{C}$</td>
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<td>Properties of the pipe</td>
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<td>Thickness</td>
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<td>2 mm, 3 mm</td>
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</table>


Given that the outer surface temperature $T_{w,o}$ is directly measured using surface thermal sensors, the total thermal resistance between the outer surface of the pipe and the inner stream, of which the temperatures are denoted as $T_{w,o}$ and $T_{m}$, respectively, in Fig. 4, can be expressed as

$$R'_{tot} = \ln\left(\frac{r_i}{r_o}\right) + \frac{1}{h_{L,2\pi r_i}L}$$  \hspace{1cm} (13)

where $R'_{tot}$ is the total thermal resistance between the outer surface of the pipe and the inner stream (h/℃).  

As mentioned previously, the thermal release rate by the fluid stream within the pipe can be expressed as

$$q = m'c_p(T_{m,out} - T_{w,o}) = m'c_p[(T_{w,o} - T_{m,in}) - (T_{w,o} - T_{m,out})]$$ \hspace{1cm} (14)

where $q$ is the thermal release rate (kcal/s), $c_p$ is the specific heat of the fluid (kcal/kg·℃), $T_{m,out}$ is the outlet mean temperature of the fluid (℃), $T_{m,in}$ is the inlet mean temperature of the fluid (℃), $T_{w,o}$ is the outer surface temperature of the pipe (℃), and $m'$ is the mass velocity (kg/s).

The mean temperature of the fluid stream is determined by the idealized condition, as shown in Fig. 5. As the outer surface temperature of the pipe is held constant in this device, the fluid temperature of the stream within the pipe varies along longitudinal the direction of the pipe. The log-mean temperature difference between the pipe wall and the stream can be expressed as follows:

$$\Delta T_{ml} = \ln\left(\frac{\Delta T_{w,o}}{\Delta T_{m}}\right)$$ \hspace{1cm} (15)

The corresponding thermal transfer rate is expressed as

$$q = \frac{\Delta T_{ml}}{R'_{tot}}.$$ \hspace{1cm} (16)

As the thermal transfer in Eq. (16) occurs due to the thermal release in Eq. (14), the heat transfer coefficients at the inlet can be derived using Eq. (14) and Eq. (16). This is expressed as

$$h = \frac{1}{2\pi r_i L} \cdot \frac{m'c_p}{m'c_p} \cdot \frac{\Delta T_{tot}}{\ln\left(\frac{r_i}{r_o}\right)} \cdot \frac{1}{2\pi k_i L}.$$ \hspace{1cm} (17)
4. Experimental results and analysis

4.1 Experimental results

Table 2 shows the test results of the heat transfer coefficient using the developed device. In Table 2, specimen names of test cases are defined as the first letter as the material type of the pipe (S denotes a steel pipe and P is a PVC pipe), the second letter D as the number being the pipe diameter in mm, and the last number after the letter T as the pipe thickness in mm. For example, SD34.3T3.3 represents a specimen with its pipe material being steel and its pipe diameter and thickness being 34.3mm and 3.3mm, respectively. In Table 2, A, B, C, and D represent the outer surface temperatures of the pipe. Inlet and outlet temperatures were measured at the center of pipe. The heat transfer coefficient was calculated from Eq. (17).

When the heat transfer coefficient is calculated from Eq. (17), it is assumed that the surface temperature of the pipe is constant. Therefore, the heat transfer coefficient is constant for the given constant temperature condition. To investigate this assumption, experiments were performed for three different pipe surface temperatures (40, 50 and 60 °C). Fig. 6 is the graphed results of specimen SD34.3T3.3 with the three constant surface temperature conditions. As shown in Fig. 6, the heat transfer coefficient is constant as the surface temperature varies. Therefore, the heat transfer coefficient obtained using the developed device with a constant surface temperature condition and Eq. (17) is valid and acceptable. Additionally, heat transfer coefficients proposed by Kawaraba et al. (1986) and Tanabe et al. (1985) are also shown as Eq. (2) and Eq. (3), respectively in Fig. 6. As shown in Fig. 6, the existing models underestimate the heat transfer coefficient with respect to flow velocity, and the relevant differences in heat transfer coefficient can cause some differences in heat transfer analysis, which will be discussed in section 5.2.

4.2 Prediction model of the flow convection coefficient

4.2.1 Derivation of a prediction model

In this section, the goal is to obtain a general prediction model of the heat transfer coefficient by theoretical procedures. Generally, there are several influencing factors of the heat transfer coefficient; these include not only the aforementioned factors but also the flow conditions represented by laminar or turbulent flow conditions, along with the surface toughness of the pipe. In this study, a limited condition is addressed in which a round pipe with a smooth surface and a turbulent flow condition is considered.

Figure 7 shows the conceptual flow chart used to develop the prediction model. In order to develop a general prediction model for the heat transfer coefficient, heat flux by conduction through the pipe as well as the convection between the pipe and the cooling water should be reflected. In this concept, several states which simulate the heat transfer in a pipe-cooling system were assumed. From these assumed states, the general prediction model was derived, and correcting coefficients were determined through the experimental results given in Table 2.

If heat transfer occurred due to the internal flow in a pipe-cooling system, the thermal gradient denoted as the continuous line in Fig. 7 should be induced. In this case, the net thermal resistance can be expressed as

\[ R_i = R_k + R_h = \frac{\ln(r_i/r_o)}{2\pi k_p} + \frac{1}{2\pi r_i} \]

where \( R_i \) is the net thermal resistance; \( R_k \) is the thermal resistance by conduction of the pipe; \( R_h \) is the thermal resistance due to convection between the pipe and the cooling water; \( r_o \) and \( r_i \) are the outer and inner radii of
pipe, respectively; \( k_p \) is the thermal conductivity of the pipe material; and \( h_i \) is the heat transfer coefficient.

If an ideal convection condition were applied to the above state, the thermal gradient could be assumed to take the form of the dotted line shown in Fig. 7. The net thermal resistance could then be expressed as

\[
R_{h, \text{ideal}} = R_{\text{con}} + \frac{1}{h_{\text{ideal}}} \quad \pi \times k \frac{d}{r} \]

(19)

where \( R_{\text{con}} \) is the thermal resistance due to the conduction of the pipe corresponding to the ideal convection condition; \( R_{h, \text{ideal}} \) is the thermal resistance in an ideal convection condition; \( k \) is the thermal conductivity of the pipe material corresponding to the ideal convection condition; and \( h_{\text{ideal}} \) is the heat transfer coefficient in an ideal condition, which is expressed for a turbulent flow condition based on theoretical and empirical backgrounds (Sodha et al. 1993), as follows:

\[
Nu_d = h_{\text{ideal}} D = 0.023 Re_d^{1/3} Pr^{1/3} \]

(20)

where \( Nu_d \) is the Nusselt number; \( D \) is the diameter of the pipe, \( k_w \) is the conductivity of the inner stream, \( Re_d \) is the Reynolds number based on the diameter of the pipe, and \( Pr \) is the Prandtl number.

Assuming an equivalent convection condition reflecting the heat flux due to conduction through the pipe and convection between the pipe and the cooling water, the thermal transfer can be presented as the thermal gradient of the third state in Fig. 7. The net thermal resistance of this equivalent condition can then be expressed as

\[
R_{h} = \frac{1}{h} \pi \times k \frac{d}{r} \]

(21)

where \( R_h \) is thermal resistance in the equivalent convection condition and \( h \) is the equivalent heat transfer coefficient.

As the net thermal resistance of Eq. (19) is identical to those in Eq. (21), Eq. (22) can be induced; the equivalent heat transfer coefficient is finally obtained as Eq. (23),

\[
R_{h, \text{ideal}} = R_{h} + R_{h, \text{ideal}} \]

(22)

### Table 2 Experimental results.

<table>
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<th>Case</th>
<th>Temperature in the cell (°C)</th>
<th>Inlet temp. (°C)</th>
<th>Outlet temp. (°C)</th>
<th>Flow velocity (m/s)</th>
<th>Heat transfer coefficient (kcal/m² h °C)</th>
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\[ h_i = \frac{1}{r_i \ln \frac{\beta}{k_i}} \]  
(23)

where \( k' = \alpha k_p \); \( \bar{h}_{\text{ideal}} = 0.023 \left( \frac{k_p}{2r_p} \right) \text{Re}^{0.5} \text{Pr}^{0.5} \); and \( \beta = (r_o / r_i) \).

Given that the equivalent heat transfer coefficient in Eq. (23) is composed of the flow velocity of the cooling water, the thermal conductivity of the pipe material, and the geometry of the pipe, as in the outer and inner radii \( (r_o, r_i, k_p, u) \), this equation can be proposed as a general model for the flow convection heat transfer coefficient considering the aforementioned factors. When the thermal conductivity of the pipe material becomes an infinite value \( (k' \to \infty) \) and the thickness of the pipe goes becomes an infinitesimal value \( (\beta \to 1) \), the heat transfer coefficient of Eq. (23) is identical to those in Eq. (20), which represents the ideal convection condition. Hence, this verifies that the basic assumption of this procedure is valid.

By inserting the parameters of \( \text{Re} = 2 \pi r_p \rho_w u / \mu_w \); \( \rho_w = 10^6 \text{ g/m}^3 \); \( \mu_w = 4680 \text{ g/m h} \); \( k_w = 0.5 \text{ kcal/m h} \degree \text{C} \); and \( \text{Pr} = 9.5 \) into Eq. (23), the general prediction model for the heat transfer coefficient can be obtained as

\[ h_i = \frac{1}{r_i \ln \frac{\beta}{k_i}} \frac{1}{\alpha k_p} \cdot \frac{1}{1258 \pi^{0.5} \alpha^{0.5}} . \]  
(24)

4.2.2 Determination of the model parameter \( \alpha \)

In Eq. (24), the only unknown parameter is \( \alpha \), which serves to improve the accuracy of the proposed model. In this study, the unknown parameter \( (\alpha) \) was calculated using the experimental results given in Table 2. Table 3 shows calculated results for \( \alpha \). As shown in Table 3, \( \alpha \) varies with the type of the pipe material and with the geometry of the pipe. From these results, the factors influencing the value of \( \alpha \) are clear; however, it is possible to assume that the geometry and the material of the pipe influence \( \alpha \) from Eq. (24).

In this study, the \( \alpha \) value for a steel pipe was obtained through a function of \( \beta \), representing the geometry of the pipe, as shown in Fig. 8 and Eq. (25). For a PVC pipe, the \( \alpha \) value was assumed to be constant because the only experimental parameter was the flow velocity.

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\[ \alpha = 0.2909 \beta - 0.2848 \quad \text{For a steel pipe} \]  \hspace{1cm} (25)
\[ \alpha = 1.094 \quad \text{For PVC pipe} \]  \hspace{1cm} (26)

Inserting Eq. (25) and Eq. (26) to Eq. (24), the general prediction model for the flow convection coefficient can be proposed. To verify accuracy of the proposed model, the heat transfer of the flow convection coefficient calculated by Eq. (24) was compared with the experimental results, as shown in Fig. 9 shows that the proposed model of the flow convection heat transfer coefficient is in good agreement with the experimental results.

5. Verification example of the suggested model

5.1 Modeling of spread footing

In this section, a heat transfer analysis of a concrete structure is performed to verify the applicability of the heat transfer coefficient model suggested in Eq. (24) to the heat transfer phenomenon in an actual concrete structure. The type of structure selected as a numerical example for the heat transfer analysis is a part of the Seohae Grand Bridge, which was constructed in Korea in the year 2000, the spread footing of which was constructed as a massive concrete structure. The spread footing, in which the temperature history due to hydration heat has been measured, consists of reinforced concrete structures placed onto rock. (Choi et al. 1994) The footing was cast using two lifts, the heights of which are both 2m. The geometrical layout selected is illustrated in Fig. 10. In this figure, the dimension of the rock is considered to be 20m×32m×6m to simulate the effect of the heat transfer from the placing of concrete onto the rock. The layout of the pipe loop located in the center of the first lift is shown in Fig. 10(b), where the horizontal spacing of the pipe is 1.2m.

![Fig. 10 Layout of the footing and the pipe (Choi et al. 1994).](image-url)
The footing and pipe loop were modeled as shown in Fig. 11 using eight-node solid element and two-node line element, respectively. The placed concrete was divided more densely than the rock. Each line element was numbered along the flow direction of the cooling water to consider the effects of the internal flow. Hence, the total numbers of solid and line elements of this finite element mesh were 71,160 and 790, respectively, and the total number of nodes was 80,405.

The measured temperatures were compared with the predicted results at section A. As shown in Fig. 12, the nodal points, where the numerical results were displayed, were precisely in accordance with the locations of the temperature sensors. The initial temperature of the placed concrete and rock was assumed to be 31°C.

The adiabatic temperature rise curve of the used concrete, which was obtained using a calorimeter, is shown in Fig. 13. The maximum temperature was 47.0°C, the reaction rate was 1.3, and the delayed time was 0.1 day.

Table 4 shows the thermal properties of the rock, placed concrete, and pipe. In order to investigate the pipe cooling effect, the thermal properties of the coolant shown in Table 5 were used. While pipe cooling was carried out, the inlet temperatures were in the range of

<table>
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<th>Material</th>
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<th>Specific heat (kcal/kg·°C)</th>
<th>Density (kg/m³)</th>
<th>Atmosphere convectivity (kcal/m²·hr·°C)</th>
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<tr>
<td>placing concrete</td>
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<tr>
<td>pipe</td>
<td>60.0</td>
<td>0.16</td>
<td>7800</td>
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</table>

Fig. 11 Mesh modeling for the concrete and the cooling pipe.

Fig. 12 Comparison of the analyzed and measured points (section A).

Fig. 13 Adiabatic temperature rise curve (Choi et al. 1994).
21°C to 28°C, whereas the volumes of the cooling water varied in the range of 0.9 to 1.2 m³/hr. Thus, the mean values were used in this numerical analysis; i.e., the inlet temperature of the cooling water was 25°C, and the volume of the cooling water was 1.08 m³/hr. These values corresponded to a cooling water velocity of 60 cm/sec. For verification of the prediction model in Eq. (24), two different heat transfer coefficient values, \( h_u \) and \( \overline{h} \) as predicted by Eq. (2) and (24), respectively, were implemented in a heat transfer analysis and the relevant results were compared.
5.2 Comparison between measured and analyzed results

Figure 14 shows the temperature distributions obtained from the heat transfer analysis with the heat transfer coefficient predicted by Eq. (24). At the initial stage, e.g., 2 days in Fig. 14(a), when the internal temperature increases abruptly due to the rapid hydration reaction of the concrete, the temperature at each pipe location shows a rather low value in contrast with the surrounding area. This occurs because the hydration heat of the concrete is transferred to the cooling pipe and to the water due to the energy balancing process. The temperature histories of the concrete at section A due to the hydration heat and the heat transfer process are presented from Fig. 15(a) to Fig. 15(d). These figures show that the temperature of the concrete with the heat transfer
coefficient predicted by Eq. (24) decreases more rapidly than that predicted by Eq. (2). This phenomenon is clearer at point b, which is closest to the cooling pipe among the four points displayed in Fig. 12. The more rapid reduction of the concrete temperature with Eq. (24) is attributed to the fact that Eq. (2), which considers only the flow velocity, underestimates the rate of heat transfer from the concrete to the cooling water, in contrast with Eq. (24), which considers not only the flow velocity but also the pipe geometry and the pipe material properties. Therefore, the temperature result with Eq. (24) is in better agreement with the measured data as compared with Eq. (2).

The highest peak temperature occurred at point d. Point d is located on the centerline of the section, which is far from the cooling pipe. However, it was found that the temperature histories with the heat transfer coefficient predicted by Eq. (24) successfully simulate the measured temperature data, whereas the temperature result from Eq. (2) considerably overestimates the measured temperature.

The temperature change of the cooling water is shown in Fig. 15(e). In the numerical analysis, the temperature of the cooling water at the inlet was assumed to be 25°C; however, the measured inlet temperatures range from 21°C to 28°C. In spite of the fluctuations in the actual temperatures, it was observed that the numerical result of Eq. (24) suitably predicts the temperature variation of the cooling water, in contrast with Eq. (2). Fig. 15(e) indicates that the temperature increase of the cooling water can be appropriately estimated reasonably well with Eq. (24); hence, the flow convection theory adopted in this paper simulates actual situations well.

6. Conclusions

From the results of these investigations of the flow convection heat transfer coefficient, the following conclusions can be drawn.

(1) The heat transfer coefficient represents the thermal transfer between the pipe and the cooling water in a pipe cooling system and varies with the velocity of the flow, the type of pipe material, and the geometry of the pipe.

(2) In this study, an experimental device was developed to investigate the heat transfer coefficient, and experiments for selected the parameters using this developed device were performed. From the experimental results, the heat transfer coefficient was calculated.

(3) Based on the theoretical background of the flow convection, a general prediction model including the influencing factors for the heat transfer coefficient was proposed. The flow convection heat transfer coefficient obtained by the proposed model was in good agreement with those from the experiments.

(4) From a heat transfer analysis conducted on an actual massive concrete structure, it was found that the numerical results of the proposed model are in excellent agreement with the measured data from a real structure compared to an existing model. Thus, the applicability of the proposed model is verified.

Acknowledgments

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References


