Estimation of Heat Transfer Characteristics on Hot Surface of Fireproof Curtain with Down-flowing Water Film

Qurrotin A'yunina MOA[1], Shin-Ku Lee[2], Ming-Chin Ho[3], Ming-Ju Tsai[3], Ta-Hui Lin[1, 2*]
[1]Department of Mechanical Engineering, National Cheng Kung University
[2]Research Center for Energy Technology and Strategy, National Cheng Kung University
[3]Architecture and Building Research Institute, Ministry of the Interior

[*]Corresponding Author, *Email: thlin@mail.ncku.edu.tw*

NSC Project No: NSC102-2622-E-006-023-CC2

Abstract

In investigating the effectiveness of down-flowing water films in improving the performance of fireproof curtains, it is desirable to determine the heat transfer characteristic on both sides of the curtain. However, due to the extreme heat, the heat transfer characteristics on the hot surface of the curtain are not easily obtained using direct experimental methods. Accordingly, in the present study, a hybrid inverse heat conduction scheme is used to derive the unknown transient total heat flux, overall heat transfer coefficient and surface temperature on the hot surface of the curtain based on a small number of experimental temperature measurements obtained on the cold side. The validity of the proposed approach is investigated for both small-scale ($1.2 \times 1.2 \times 0.0005$ m^3) and full-scale ($3.0 \times 3.0 \times 0.0005$ m^3) curtains. In general, the results show that for both curtains, the down-flowing water film provides an effective means of enhancing the latent and sensible heat transfer, and therefore improve the fire resistant properties of the curtain.

Keywords: hybrid inverse scheme, down-flowing water film, fireproof curtain, heat transfer

1. Introduction

The most crucial aspect of a building's safety in the face of fire is the possibility of safe escape [1]. However, many houses, offices, hotels, and shopping malls use curtains as a means of blocking light or as decoration. Interior decorative fabric has a major impact on the development and spread of indoor fires because of its easy ignition, fast burning speed and rapid spread rate [2]. Therefore, suitable methods for enhancing the fire resistant properties of curtains is an important concern.

The literature contains various investigations into the burning behavior of common curtain materials. For example, Bei et al. [2] evaluated the burning behavior of cotton, jeans, wool, linen rope and sponge using a cone calorimeter. It was shown that a higher heat flux condition increased the average heat release rate and peak mass loss rate; whereas a lower heat flux condition increased the smoke production rate and the CO yield. Li et al. [3] investigated the fire hazards posed by three typical curtain materials, namely 35%-cotton, 40%-cotton and 45%-cotton. The experimental results showed that the average flame spread rate first increased and then decreased with an increasing cotton content, but continuously decreased with an increasing pleat rate.

Large heated solid surfaces can be effectively cooled by flowing a thin liquid film over it under the effects of gravity together with an external counter current air stream [4]. Wu et al. [5] performed an experimental investigation into the effect of water spray on the heat resistance property of a glass pane. Wu and Lin [6] further examined the fire insulation and fire integrity properties of glass panes covered with a down-flowing water film in a standard full-scale $3m \times 3m$ door/wall refractory furnace in accordance with the ISO 834-1 standard code [7]. The results showed that the application of the water film enabled the glass pane to satisfy the permitted integrity and insulation requirements for as much as 100 minutes. Lee et al. [8] applied a water film system on a steel roller shutter. It was shown that the water film system combined with the steel roller shutter could effectively improve the heat resistance and temperature of the shutter slat surface.

In general, using down-flowing water system has many advantages compared to traditional flame retardants. For example, water is cheaper than flame retardant and its installation more straightforward. Furthermore, water is readily obtained, whereas flame retardant materials need to be chemically produced. In developing down-flowing water systems for fire retardant applications such as fire retardant curtains, it is necessary to understand the temperature conditions not only on the cold side of the curtain, but also on the hot surface. However, due to the extreme temperature on the fire-exposed surface, the temperature conditions are not easily measured using direct experimental methods.

Jeng et al. [9] performed STAR-CD simulations to investigate the heat transfer performance of a high-efficiency air-cooled condenser system for power plants under various frontal speeds and axis ratios of the elliptic tubes. However, it was shown that the simulation results for the overall heat transfer coefficient varied by as much as 45% from the experimental data. The deviation may arise due to the mathematical model, boundary condition or assumption are not completely agree with the actual physical model. Thus, a good estimation of the transient heat transfer characteristics for the present problem is difficult to be obtained using the commercial software. To address the limitations of simulation methods, the inverse heat conduction scheme can be introduced to investigate the present problem. Importantly, the inverse scheme enables the unknown heat transfer characteristics to be estimated using the experimental data obtained at only a limited number of measurement locations. However, the solutions are sensitive to changes in the input data resulting from measurement errors [10,11]. Accordingly, Chen and Chang [12] introduced a hybrid scheme comprising the Laplace transform method and the finite-difference method to estimate the unknown surface temperature based a small number of temperature measurements obtained within the test sample. It was shown that while the proposed method reduced the sensitivity of the estimated results to the measurement error, the results were still sensitive to the measurement location. Consequently, Chen et al. [13-15] combined the hybrid scheme with a sequential-in-time concept and the least squares method to predict the unknown surface conditions. A good agreement was obtained between the estimated values of the unknown surface conditions and the experimental results. Importantly, the results obtained using the inverse heat conduction scheme were found to be insensitive to both the measurement time-step and the measurement location.

Accordingly, in the present study, the proposed inverse heat conduction scheme is used in a numerical investigation to estimate the unknown conditions on the hot surface of a curtain with a down-flowing water system.

2. Methodology

To obtain the condition of the unexposed surface, some thermocouples were attached on the surface. The surface temperature was measured by K-type thermocouple at different locations. It can be seen from Figure 1. The data used in the present simulations were obtained from the small-scale and full-scale experiments involving a fireproof curtain with a down-flowing water system. The dimensions of the small-scale test were as follows: 1.2 m in length, 1.2 m in width, and 0.0005 m in thickness. Meanwhile, the dimensions for the full-scale test were 3.0 m in length, 3.0 m in width, and 0.0005 m in thickness. The furnace used in the small-scale test had a length of 1.5 m and a width of 1.5 m, while that used in the full-scale test had a length and width of 4.0 m. In the small-scale test, the operating pressure was 0.4 kgf/cm^2 (corresponding to 65.92 L/min), while in the full-scale test, the operating pressure was 0.9 kgf/cm^2 (corresponding to 111.39 L/min).

In both tests, the surface temperature was measured by K-type thermocouples positioned at the locations shown in Fig. 1.

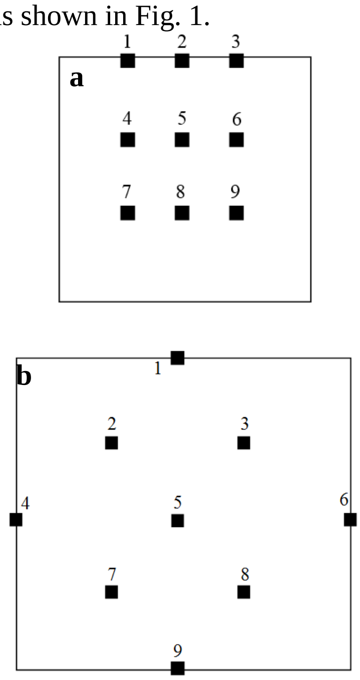


Fig. 1 Thermocouple positions on the (a) small-scale test and (b) full-scale test.

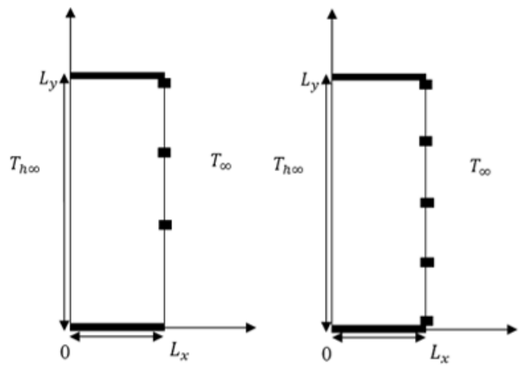


Fig. 2 Schematic geometry considered in the (a) small-scale test and (b) full-scale test.

Figure 2 shows the physical geometry of the problem. Figure 3 shows the heat transfer mechanism with the down-flowing water on the cold surface of the curtain. The boundary condition for this problem is specified as thermal insulation at $0<x<L_x$ when $y=0$ and $y=L_y$. L_x and L_y are the lengths of the curtain in the x and y directions. Based on this condition, the governing equation can be formulated as

$$\rho c \frac{\partial T}{\partial t} = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \text{in } 0<x<L_x,\ 0<y<L_y \text{ and } t>0 \quad (1)$$

with boundary condition

$$-k\frac{\partial T}{\partial x} = h_h(y,t)(T_h - T_{h\infty}) \quad \text{at } x=0 \quad (2)$$

$$-k\frac{\partial T}{\partial x} = Q_{\text{convection}} + Q_{\text{radiation}} + Q_{\text{evaporated water}} + Q_{\text{unevaporated water}} \quad \text{at } x=L_x \quad (3)$$

where

$$Q_{\text{convection}} = h_c(y,t)(T_c - T_\infty) \quad (4)$$

$$Q_{\text{radiation}} = \sigma\varepsilon\left(T_c^4 - T_\infty^4\right) \quad (5)$$

$$Q_{\text{evaporated water}} = \dot{m}_{wc}c_p(373.15 - T_c) + \dot{m}_{wc}h_{fg} \quad (6)$$

$$Q_{\text{unevaporated water}} = \dot{m}_{wd}c_p(T_{wd} - T_\infty) \quad (7)$$

$$\frac{\partial T}{\partial y} = 0 \text{ at } y=0 \text{ and } y=L_y \quad (8)$$

and the initial condition

$$T = T_{in} \text{ for } t=0 \quad (9)$$

where x and y are the spatial coordinates; $\dot{m}_{wc}$ and $\dot{m}_{wd}$ are the water consumption and water drain mass flow rates, respectively; c_p and h_{fg} are the specific heat and heat of vaporization. T_{in} is the initial temperature; and T_c and T_h are the cold and hot surface temperatures. In addition, T_∞ and $T_{h\infty}$ are the ambient temperatures in the low and high temperature environments; T_{wd} is the water drain temperature; k and ρc are the thermal conductivity and heat capacity per unit volume of the curtain, respectively. Finally, σ and ε are the Stefan-Boltzmann constant and emissivity of the curtain; respectively, and $h_h(y,t)$ is the unknown overall heat transfer coefficient. The local Nusselt number Nu_y and local Rayleigh number Ra_y are related as [16].

$$Nu_y = 0.103 \quad a_y^{1/4} \quad (10)$$

where

$$Nu_y = \frac{h_c(y,t)\,y}{k_{air}} \quad (11)$$

and

$$Ra_y = \frac{g\beta y^3\left[T_c(y,t) - T_\infty\right]}{v_{air}\alpha_{air}} \quad (12)$$

In Eqs. (11) and (12), k_{air} , v_{air} and α_{air} are the thermal conductivity, kinematic viscosity and thermal diffusivity, respectively, of the ambient air. In addition, g is the gravitational acceleration and β is the volumetric thermal expansion coefficient. All of the physical properties are evaluated as the average of $T_c(y,t)$ and T_∞.

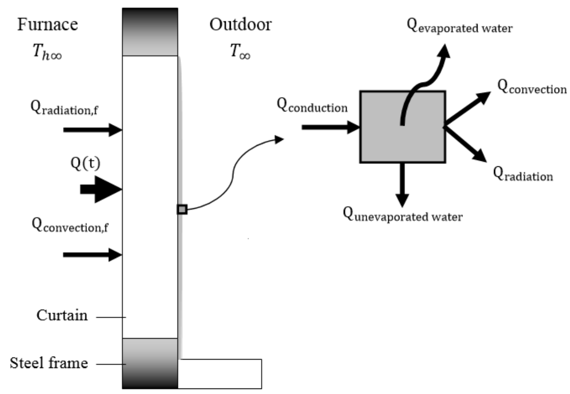


Fig. 3 Heat transfer mechanism on the fireproof curtain with down-flowing water film on the cold surface.

Applying Newton's law of cooling, Eq. (2) can be written as

$$-k\frac{\partial T}{\partial x} = q_h(y,t) \quad \text{at } x=0 \quad (13)$$

where $q_h(y,t)$ is the unknown surface heat flux on the hot surface of the curtain, and is defined as $h_h(y,t)[T_h(0,y,t)-T_{h\infty}]$. It can also be considered as the sum of the convective heat flux and radiative heat flux. Thus, $h_h(y,t)$ can be regarded as the sum of the convection and radiation heat transfer coefficients. Due to the surface heat flux acting on the cold surface of the curtain $q_c(y,t)$, boundary condition (3) can be written as

$$-k\frac{\partial T}{\partial x}=q_c\left(y,t\right) \quad \text{at } x=L_x \tag{14}$$

The time-dependent terms can be removed from the governing differential and boundary conditions using the Laplace transform method. The governing equation can then be discretised using the central-difference approximation and rearranged in the following matrix form [17]

$$\left[A\right]\left[\tilde{T}\right]=\left[B\right] \tag{15}$$

where [A] is a coefficient matrix, $\left[\tilde{T}\right]$ is a matrix representing the nodal temperatures in the Laplace transform parameter *s* domain and [B] is a matrix representing the *s* forcing term. Finally, the temperatures in the *s* domain are inverted into physical quantity *t*, using the Gaussian elimination algorithm and the numerical inversion of the Laplace transform. Importantly, the hybrid inverse scheme yields an effective reduction in the computational time since the transient heat flux at a specific time t_s can be estimated directly without the need for step-by-step computation from initial time t_0.

Once the unknown surface heat flux $q_h(y,t_s)$ at a specific time t_s can be determined, the overall heat transfer coefficient $h(y,t_s)$ can also be obtained from the Newton's law of cooling. However, it can be difficult to obtain an approximate function that can completely fit the distribution of the unknown surface heat flux $q_h(y,t)$ at a specific time. Under this circumstance, $q_h(y,t)$ can be approximated using a cubic polynomial function at a specific time. Thus, $q_h(y,t)$ can be assumed to be a cubic polynomial function in space and a linear function in time during a specific time interval. On the other hand, the surface heat flux $q_h(y,t)$ can be expressed as

$$q_{h,r}\left(y,t\right)=\sum_{i=1}^{p}\left(C_{2i-1}+C_{2i}t\right)y^{i-1} \quad \text{for} \quad t_s\le t\le t_s+\Delta t \tag{16}$$

where C_{2i} and C_{2i-1} are the unknown coefficients which can be determined using the least-squares method in conjunction with experimental measured temperatures. The number of the measurement locations or thermocouples, *J*, is equal to $2p$. Δt indicates the measured time-step.

The least-squares minimization technique is applied to minimize the sum of the squares of the deviations between the calculated and experimental measured temperatures at several selected measurement locations and two specific times t_s and $t_s+\Delta t$. In order to avoid repetition, the computational procedures for determining the unknown coefficient C_j are not shown in this paper. Its detailed computational procedures can be found in Ref. [13-15]. Finally, the above numerical procedures are repeated until the values of residual any specific time are all less than 10^{-3} through all the inverse calculations.

Tab. 1 The comparison of heat transfer mechanism on the small-scale test and full-scale test.

Case	Heat transfer	Heat flux (W/m^2)
Small-scale test	$Q_{convection}$	92.00
	$Q_{radiation}$	65.98
	$Q_{evaporated,\ w}$	84249.68
	$Q_{unevaporated,\ w}$	103567.09
	Q_{total}	187974.76
Full-scale test	$Q_{convection}$	328.95
	$Q_{radiation}$	211.00
	$Q_{evaporated,\ w}$	76028.47
	$Q_{unevaporated,\ w}$	39641.32
	Q_{total}	120446.94

3. Results and discussion

In general, the experimental results showed that for both the small-scale test and the full-scale test, the temperature on the cold side was less than 100°C. In other words, the temperature was consistent with CNS 14803 [18] which specified that the maximum temperature on the cold side of a fire door should be no more than 210°C. Table 1 shows the heat fluxes calculated from the experimental temperature measurements in the two test cases. As shown in the small-scale test, the heat transfer is dominated by the unevaporated water heat transfer. By contrast, in the full-scale test, heat transfer is dominated by the evaporated water heat transfer mechanism. The difference between the two cases arises because in the full-scale test, the curtain has a larger heated surface and thus the amount of evaporated water is increased.

In performing the inverse computations, the thermal conductivity of the curtain was taken as k=0.2629 W/mK. For the small-scale case, the number of nodes in the $x-$ and $y-$ directions was specified as n_x=5 and n_y=12, while in the full-scale test, the number of nodes was specified as n_x=5 and n_y=24. In evaluating the heat transfer characteristics on the hot and cold sides of the curtain, the average overall heat transfer coefficient, surface temperature, and average surface heat flux were calculated as

$$\bar{h}(t) = \int_0^{L_y} h(y,t)dy / L_y \tag{17}$$

$$\bar{T}(t) = \int_0^{L_y} T(y,t)dy / L_y \tag{18}$$

$$\bar{q}(t) = \int_0^{L_y} \mathrm{q}(y,t)dy / L_y \tag{19}$$

Figure 4 shows the variation over time of the total incident heat flux on the hot and cold surfaces of the curtain in the small-scale and full-scale cases. In both cases, the total heat flux increases over time. Furthermore, at a given time, the heat flux on the hot surface is greater than on cold surface. This result is to be since the heat on the cold surface of the curtain is transported away by convection, radiation, evaporated water and unevaporated water heat transfer. The existence of the water film on the cold surface of the curtain has a significant effect on the total incident heat flux. For example, in the full-scale test, the maximum difference between $\bar{q}_h(t_s)$ and $\bar{q}_c(t_s)$ is about 70158 W/m^2. Similarly, in the small-scale test, the maximum difference is around 87628.57 W/m^2. It is noted that the difference between the heat flux on the hot and cold surface is greater in the small-scale test than in the full-scale test. This result arises since in the small-scale test, the water supply is approximately 54.9 L/min per unit length, whereas in the full-scale test, the water supply is only around 37.13 L/min per unit length.

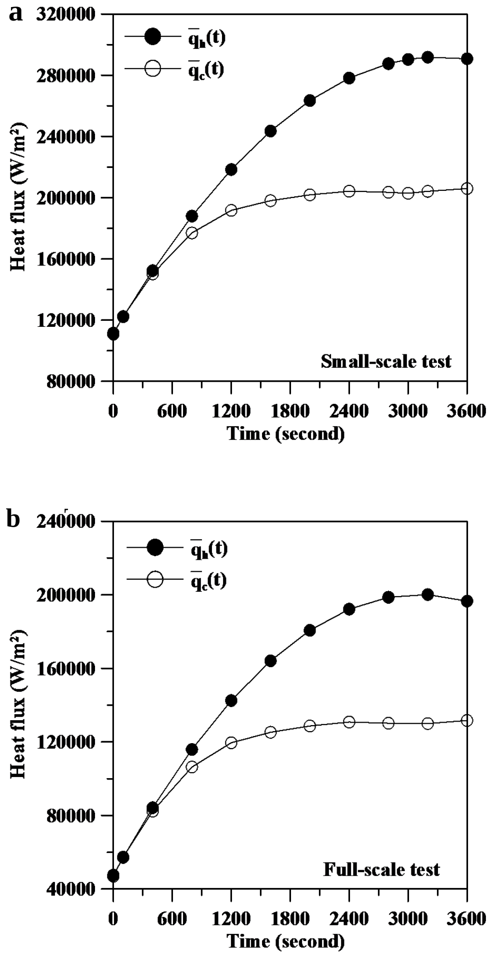
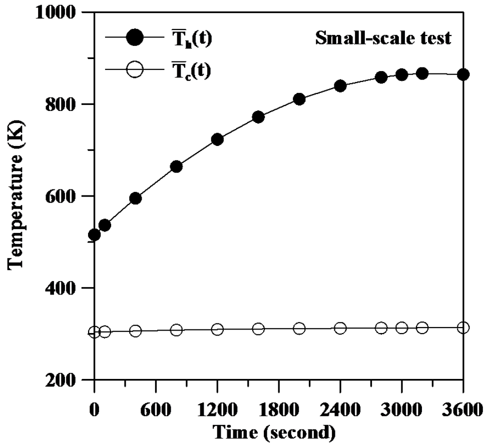


Fig. 4 History of the total incident heat flux on the hot $\bar{q}_h(t)$ and cold surface $\bar{q}_c(t)$ of the (a) small-scale test, (b) full-scale test.

Figure 5 shows the variation over time of the surface temperature on the hot $\bar{T}_h(t)$ and cold surfaces $\bar{T}_c(t)$ of the curtain in the small-scale and full-scale cases. The average surface temperature on the hot surface of the curtain increases from 516 K to 864 K for the small-scale test, while for full-scale test, the surface temperature increases from 402 K to 729 K. On the other side, the average surface temperature increases from 304 K to 314 K for the small-scale test and 308 K to 343 K for full-scale test. The cold surface temperature $\bar{T}_c(t)$ of the small-scale test is lower than the full-scale test due to larger amount of the down-flowing water flow rate per length along the curtain. In addition, the maximum difference between $\bar{T}_h(t)$ and $\bar{T}_c(t)$ for the small-scale test is 553 K. However, the maximum difference for full-scale test is 392 K.

This difference can be explained by the heat conduction equation ($q''=-k[\partial T/\partial x]$), both cases have the same thickness, but the small-scale has higher heat flux than full-scale case.

a



b

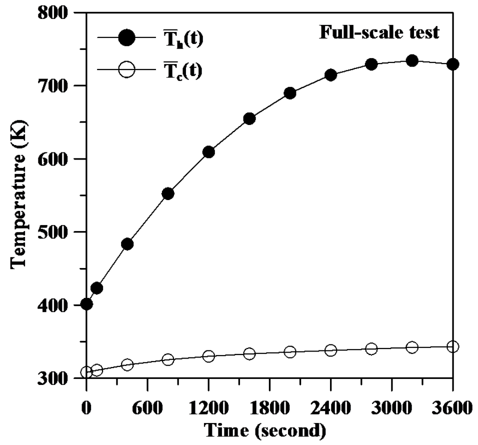
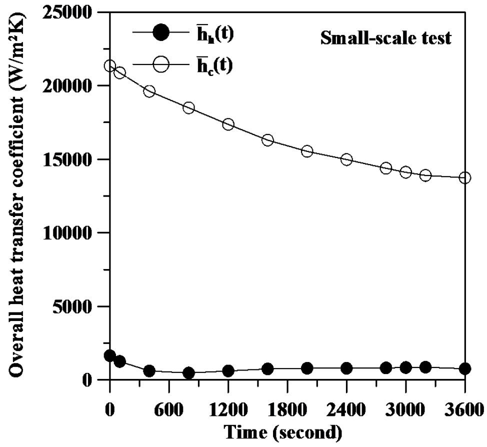


Fig. 5 History of the surface temperature on the hot $\overline{T}_h(t)$ and cold surface $\overline{T}_c(t)$ of the (a) small-scale test, (b) full-scale test.

Figure 6 shows the variation over time of the overall heat transfer coefficient on the hot and cold surfaces of the curtain in the small-scale and full-scale cases. The overall heat transfer coefficient on the cold surface $\overline{h}_c(t)$ for both cases decrease from 0 s to 3600 s. This result can be clarified that most heat taken away by water, some water will be evaporated and the rest (unevaporated water) will be flowed down to the water drain. The amount of unevaporated water will be reduced due to the evaporation process. Meanwhile, the overall heat transfer coefficient on the cold surface $\overline{h}_c(t)$ is higher than on the hot surface $\overline{h}_h(t)$. This phenomenon is due to the latent and sensible heat of down-flowing water.

Therefore, the flow rate of the down-flowing water film along the surface of the curtain can be regarded as an important factor for the heat-resistance performance of the curtain. Sufficient water has to be sprayed over the curtain surface uniformly in order to avoid the large temperature gradient, the occurrence of dry spots, further it will perforate the curtain surface.

a



b

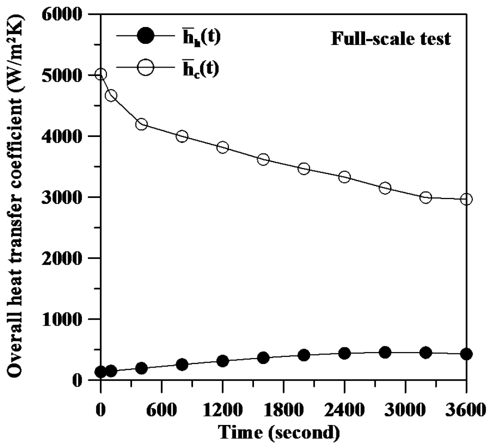


Fig. 6 History of the overall heat transfer coefficient on the hot $\overline{h}_h(t)$ and cold surface $\overline{h}_c(t)$ of the (a) small-scale test, (b) full-scale test.

4. Conclusions

The present study has applied a hybrid inverse method of the Laplace transform technique and the finite-difference methods with a sequential-in-time procedure, and the least squares method to estimate the unknown transient total heat flux, surface temperature and overall heat transfer coefficient on the hot surface of the curtain. The results show that the heat on the cold surface of the curtain is transported away by convection, radiation, evaporated water and unevaporated water heat-transfer. The heat transfer mechanisms cause the overall heat transfer coefficient on the cold surface $\bar{h}_c(t)$ is higher than on the hot surface $\bar{h}_h(t)$. Moreover, the small-scale test has larger maximum heat flux difference between the cold and hot surface than the full-scale test. This is due to the larger amount of water supply of the small-scale test. In addition, water film on the cold surface of the curtain has a significant improvement in the thermal resistance. The sensible and latent heat of water could protect the curtain from the damage as fire occurs.

Acknowledgments

This work was supported by the National Science Council, Taiwan, ROC, under contract. NSC102-2622-E-006-023-CC2.

References

[1] Kobes, M., Helsloot, I., Bauke, De V., and Pos, J. G., Building Safety and Human Behaviour in fire: A Literature Review, *Fire Safety Journal*, Vol. 45, pp. 1-11, 2000.

[2] Bei, P., , Chen, L. and Chang, L., An Experimental Study on the Burning Behavior of Fabric Used Indoor, *Procedia Engineering*, Vol. 43, pp. 257-261, 2012.

[3] Li, L., Zhang, H. Xie, Q., Chen, L., and Xu, C., Experimental Study on Fire Hazard of Typical Curtain Materials in ISO 9705 Fire Test Room, *Fire and Materials*, Vol. 36, pp. 85-96, 2012.

[4] Sha, W. T., and Soo, S. L., Instability of Flow of Liquid Film Over a Heated Surface, Third International Symposium on Multiphase Flow, Xian,China, September 19-21, 1994.

[5] Wu, C. W., Lin, T. H., Lei, M. Y., Chung, T. H., Huang, C. H. and Chiang, W. T., Fire Test on A Non-Heat-Resistant Fireproof Glass with Down-flowing Water Film, In: Proceedings of the Eight International Symposium on Fire Safety Sciences, IAFSS, Beijing, China, pp. 327-339, 2005.

[6] Wu, C. W., and Lin, T. H., Fire Resistance Tests of Glass Pane with Down-Flowing Water Film, Technical Report, Taiwan, ROC: Safety Technology Co., Ltd.; 2005.

[7] ISO 834-1, Fire Resistance Tests-Elements of Building Construction: Part 1-General Requirements, Int. Org. Standardization; 1999.

[8] Lee, S. K., Ho, M. C., Chen, J. J., Lin, C. Y., and Lin, T. H., Fire Resistant Evaluation of A Steel Roller Shutter With Water-film Cooling System, *Applied Thermal Engineering*, Vol.58, pp.465-478, 2013.

[9] Jeng, J. Y., Chen, H. T., and Ay, T., The Development of High Efficiency Air-Cooled System Condenser for The Power Plant, Technical Report, National Science Council of the Republic of China, under Grant No. NSC 92-2622-E-006-103, 2005.

[10] Ozisik, M. N., *Heat Conduction*, 2nd Ed., Wiley, 1993.

[11] Kurpisz, K., and Nowak., A. J., *Inverse Thermal Problem*, WIT Press, 1995.

[12] Chen, H. T., and Chang, S. M., Application of The Hybrid Method to Inverse Heat Conduction Problems, *International Journal of Heat and Mass Transfer*, Vol. 33, pp. 621-628, 1990.

[13] Chen, H. T., Lin, S. Y., and Fang, L. C., Estimation of Surface Temperature in Two-Dimensional Inverse Heat Conduction Problems, *International Journal of Heat and Mass Transfer*, Vol. 44, pp. 1455-1463, 2001.

[14] Chen, H. T., Lin, S. Y., and Fang, L. C., Estimation of Two-Sided Boundary Conditions for Two-Dimensional Inverse Heat Conduction Problems, *International Journal of Heat and Mass Transfer*, Vol. 45, pp. 15-23, 2002.

[15] Chen, H. T., and Wu, X. Y., Investigation of Heat Transfer Coefficient in Two-Dimensional Transient Inverse Heat Conduction Problems Using The Hybrid

Inverse Scheme, *Numerical Methods Eng*, Vol. 73, pp. 107-122, 2008.

[16] Chen, H. T., and Chang, S. M., Numerical Simulation for Conjugate Problem of Numerical Convection on Both Sides of A Vertical Wall, *International Journal of Heat and Mass Transfer*, Vol. 39, pp. 383-390, 1996.

[17] Chen, H. T., and Lee, S. K., Estimation of heat-transfer characteristics on the hot surface of glass pane with down-flowing water film, *Building and Environment*, Vol. 45, pp. 2089-2099, 2010.

[18] CNS 14803, Method of Fire Resistance Tests for Rolling Shutter of Buildings, Bureau of Standards, Metrology & Inspection, M.C.E.A., ROC, 2002.

水膜防火布幕之熱傳特性預測

莫柔婷 [1]、李訓谷 [2]、何明錦 [3]、蔡銘儒 [3]、林大惠 [1, 2*]

[1] 國立成功大學機械工程學系
[2] 國立成功大學能源科技與策略研究中心
[3] 內政部建築研究所
[*] 通訊作者，Email: thlin@mail.ncku.edu.tw

摘　要

過去針對水膜系統提升防火布幕阻熱性能的研究上，一般透過解析布幕非曝火側與曝火側表面的熱傳特性進行探討。然而，防火布幕曝火側表面的熱傳遞過程極為複雜，因此曝火面表面的熱傳現象，往往難以透過實驗方法實現。本研究藉由混合逆向熱傳法，以小尺度($1.2\times1.2\times0.0005$ m^3)以及全尺度($3.0\times3.0\times0.0005$ m^3)之防火布幕阻熱實驗所量測的非曝火側表面溫度，並考量非曝火側表面水膜的升溫以及蒸發熱傳，以預測防火布幕曝火側表面的熱傳特性。研究結果顯示，水膜系統於不同實尺寸實驗尺度下，皆得以有效提升布幕表面熱傳現象，以及強化防火布幕防火阻熱性能。

關鍵字：混合逆算法、水膜系統、防火布幕、熱傳現象