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An experimental study on the quantitative interpretation of local convective heat transfer for a plate fin and tube heat exchanger using the lumped capacitance method

Ye Yong Kim [a,*], Kui Soon Kim [b], Gi Ho Jeong [c], Sooin Jeong [b]

[a] *LGE PC Division 19-1, Chengho-Ri Jinwuy-Myun Pyungtaik Gyunggi-Do 451-713, Republic of Korea*
[b] *Department of Aerospace Engineering, Pusan National University, Busan 609-735, Republic of Korea*
[c] *SAMSUNG Electro-Mechanics, 314 Metan3-Dong Yeongtong-Gu Suwon Gyunggi-Do 443-743, Republic of Korea*

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Abstract

An experimental study has been performed to investigate the heat transfer characteristics of a plate fin and tube heat exchanger. Existing transient and steady methods are inappropriate for the measurement of heat transfer coefficients of the thin heat transfer model. In this study, the lumped capacitance method based on liquid crystal thermography was adopted. The method is validated through impinging jet and plate flow experiments. The two experiments showed very good agreements with those of the well-known transient method with the thick acryl model. And the lumped capacitance method showed similar results regardless of the thickness of the polycarbonate model if the *Bi* of the fin is small enough. The method was also applied for the heat transfer coefficient measurements of a fin and tube heat exchanger. Quantitative heat transfer coefficients of the plate fin were successfully obtained.

Keywords: Fin and tube heat exchanger; Lumped capacitance method; *Bi* number; Liquid crystal; Convection heat transfer

1. Introduction

Heat transfer enhancement of heat exchangers is a very important research topic. So far, many studies have been performed and various fin shapes have been developed for heat transfer enhancement of the heat exchangers. But almost all the fin shapes have been developed without sufficient information of the local heat transfer characteristics of the fin, and yet the local heat transfer characteristics of the fin are not clearly understood. Therefore, understanding of the local heat transfer characteristics of the fin is essential for the development of the superior heat exchanger.

For the purpose of investigating the local heat transfer characteristics of fins, Guannan et al. [1] investigated the phenomena with two and three dimensional numerical models for the fin and tube heat exchangers. And Atkinson et al. [2] investigated flow and heat transfer characteristics over louvered fin arrays in compact heat

* Corresponding author. Tel.: +82 31 610 6257; fax: +82 31 610 6358.
E-mail address: yeyongkim@hotmail.com (Y.Y. Kim).

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Nomenclature

Bi	Biot number, $\frac{hL}{k}$	*t*	time, s
c	specific heat, J/kg K	*T*	temperature, °C
CCD	charge coupled device		
D	diameter of circular impinging jet, *m*	*Greek symbol*	
h	convection heat transfer coefficient, W/m^2K	ρ	density, kg/m^3
k	thermal conductivity, W/m K		
L	1/2 of the fin thickness, m	*Subscripts*	
NTSC	National Television System Committee	c	characteristic value
$\ddot{q}_w$	heat flux, W/m^2	i	initial condition
Re	Reynolds number, $\frac{UD}{\nu}$	ref	reference value (free stream value)
RGB	Red, Green, Blue	w	wall condition

exchangers. Tsai et al. [3] tried the heat transfer analysis for the 2 row plate fin and tube heat exchanger by the three dimensional finite volume method and Jang et al. [4] also tried numerical heat transfer and fluid flow analysis of a three dimensional wavy fin and tube heat exchanger. All these local heat transfer analysis of the fin and tube heat exchangers of complicated shapes have been performed numerically.

In the field of experimental study, Wang et al. [5] investigated the heat transfer and friction correlation for compact louvered fin and tube heat exchangers, and Critoph et al. [6] compared steady and transient methods for the measurement of local heat transfer in plate fin and tube heat exchangers using liquid crystal thermography with radiant heating. Yun et al. [7] investigated the heat transfer characteristics on various kinds of fin and tube heat exchangers with interrupted surfaces. There are many other experimental studies, but all the reported information reveals average heat transfer coefficients. These are due to the existing heat transfer measurement methods which are inappropriate for application to the complicated and very thin heat transfer model such as the heat exchanger fin.

In this study, the lumped capacitance method (LCM) [8] is presented as a heat transfer measurement method for the very thin heat transfer model. At first, to validate whether this method is applicable or not for the heat transfer measurement, experiments are performed for both the impinging jet and the flat plate flow. The results are compared with those of the transient method which is known as a reliable method. And to investigate the possibility of application to the heat exchanger fin, the local heat transfer characteristics of the fin are investigated in the experiment for the fin and tube heat exchanger model.

2. Temperature measurement by using liquid crystal

Liquid crystal is not a material but a state of a material. Organic materials having a liquid crystal state have two distinct melting points. The milky-white liquid state between the first and the second melting point is called the liquid crystal state. The liquid crystal state exhibits the fluid nature of a liquid while maintaining a degree of anisotropic and ordered structure like a crystalline solid. This crystal structure scatters light of different wavelengths depending on the sensed temperature. This selective light reflection usually gives rise to a spectrum of colors. The selective light reflection is the property caused by the temperature dependent distance between molecular layers of liquid crystal. The color response of a thermochromic liquid crystal depends mainly on the temperature, so various color changes of the liquid crystal can be used as a temperature indicator [9]. The temperature of the point of interest on the liquid crystal sprayed heat transfer model is directly related to the color displayed at that point. Liquid crystals have a color response time of 5–50 ms which is fast enough to be used in transient heat transfer analysis [10].

Three characteristics, hue, saturation and intensity represent the total information necessary to define or recreate a specific color stimulus. Conceptually, this definition of color is highly convenient and appropriate for an image processing system to be used in the determination of convective heat transfer parameters from a liquid crystal sprayed surface. Since the orientation of the liquid crystal is the main controlling parameter for the color (hue, wavelength), a direct relationship between the local temperature and the locally measured hue value can be established.

To investigate the relationship between hue and temperature, the calibration experiment is performed. The image of heat transfer model is captured as a image including RGB information of the liquid crystal surface by using a camera (Panasonic, WV-CL320) which has the high-sensitive color CCD sensor. The camera is located on the opposite side of liquid crystal sprayed surface. The illumination system was also located on the camera side of the plate. According to Kim et al. [11], this approach provided a viewing angle which was

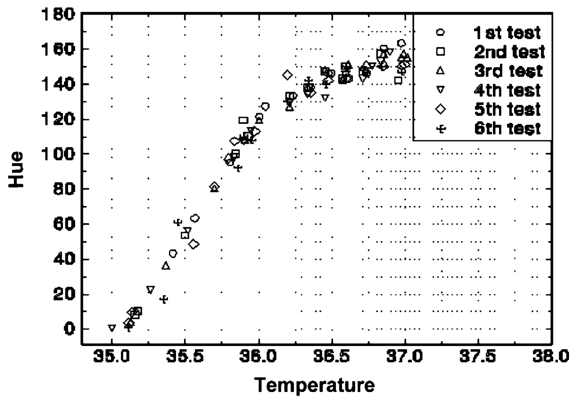

Fig. 1. Hue versus temperature relation.

normal to the heat transfer surface resulting in a linear relationship between the hue and temperature. NTSC signals obtained from the CCD camera are converted to RGB/HSI information via an image processor(Data Translation, DT2871).

A fast response K-type thin foil thermocouple (Omega, CO2-K) was flush mounted on the liquid crystal sprayed surface for calibration purposes. The temperature information is displayed through temperature indicator(Omega, DP41-TC). The indicator readout was located on the camera side of the heat transfer surface. The video recordings of the liquid crystal image also carried the instantaneous temperature information at a local point. In this way, it is possible to compare the hue at thermocouple position and measured temperature. The calibration result for the Chiral–Nematic liquid crystal (Hallcrest, BM/-R35C1W) is shown at Fig. 1. Fig. 1 shows that the hue and temperature relationship is nearly linear between 35.3 °C and 36.3 °C and hue can successfully be used as a temperature indicator.

3. Validation of the lumped capacitance method

Currently available heat transfer measurement techniques based on liquid crystals are the transient and the steady method. And the convective heat transfer coefficient is defined as follows:

$$h = \frac{\ddot{q}_{\mathrm{w}}}{T_{\mathrm{w}} - T_{\mathrm{ref}}} \tag{3.1}$$

3.1. Transient method

The transient method measures the heat transfer coefficient by establishing an unsteady heat transfer model and measuring the wall temperature and time instead of measuring heat flux. Many researchers including Schultz and Jones [12] tried this method. When the model has a low thermal conductivity, the wall temperature response is confined to a thin layer near the wall surface when subjected to a step change in the temperature of a convective heating or cooling. With this abrupt temperature change at the wall surface, the normal temperature gradient becomes very large compared to other directions. Therefore, the heat conduction may be assumed to be one-dimensional into a semi-infinite medium. The convective heat transfer coefficient is deduced from the time history of temperature by solving the one dimensional transient conduction equation analytically. To use this technique, it is required that the heat transfer model should have low thermal conductivity and thick enough that the thermal energy during the transient test does not penetrate into the other side of the wall. The heat exchanger fin to be used in this study has relatively low conductivity but the thickness is so thin that the transient method is inappropriate to be adopted.

3.2. Steady method

The steady method measures the heat transfer coefficient by supplying the constant heat flux on the heat transfer surface and was tried by Baughn et al. [13]. To provide constant heat flux, a gold-coated film powered by a DC power supply was used. As long as the shape of the gold-coated film is square, this film produces a uniform heat flux. The hue of the image produced by liquid crystal sprayed on the gold-coated film can be converted into the surface temperature information by the hue capturing method [9]. In this way, as we know all the variables on the right side of Eq. (3.1), the heat transfer coefficient can be calculated. The success of the steady method is largely dependent on the uniformity of the heat flux on the gold-coated film. But the shape of the heat exchanger fin to be used in this study is thin and complex. It is very difficult to attach the gold-coated film to the fin surface and the film may interfere with the flow itself. Also, uniform heat flux condition is not guaranteed. For these reasons, the steady method using the gold-coated film is inappropriate to be applied to the heat exchanger fin.

3.3. Lumped capacitance method

It is not easy to measure the heat transfer coefficient of the thin and complicated model such as the heat exchanger fin by existing methods. It can be assumed that the temperature is nearly uniform along normal direction to the fin surface if the fin is thin enough. This assumption leads us to use the lumped capacitance method during the transient test. On the basis of the LCM, the equation for the heat transfer coefficient is as follows [8]:

$$h = \frac{\rho L_c c}{t} \ln \frac{T_i - T_{ref}}{T_w - T_{ref}}. \quad (3.2)$$

According to this equation, the heat transfer coefficient can be obtained by measuring the time variant wall temperature. But this method has not been validated as a heat transfer coefficient measurement method. So, in this study, two experiments were performed for validation purposes. One experiment is the impinging jet experiment of which results are well known physically and quantitatively, and the other is the flat plate experiment which is similar to a single fin in a heat exchanger. For the two experiments, polycarbonate is used as heat transfer model for the LCM, and the results were compared with those of existing transient method with the thick acryl model. The LCM experiments were performed with four different thicknesses of the polycarbonate. The hue capturing method using liquid crystal (R35C1W, Fig. 1) was adopted for the measurement of the surface temperature.

The properties of polycarbonate as a heat transfer model material are as follows. These properties are for the polycarbonate plate coated with liquid crystal and black paint.

- Thermal conductivity: $k = 0.1925927$ W/m K,
- Density: $\rho = 1300.0$ kg/m^3,
- Specific heat: $c = 1350.0$ J/kg K.

Under the above condition, Bi for the thickness of 0.5 mm is about 0.058.

The first experiment to obtain the validity of LCM is an impinging jet flow. The experimental setup for the transient method is shown in Fig. 2 and for LCM in Fig. 3. The same devices were used for both the transient method and the LCM, except for the heat transfer model, to provide the same flow condition. A single jet is divided into two jets, and the distance between the jet exit and the heat transfer surface is four times the jet diameter (70 mm). For heat transfer models, an acryl plate thickness of 10 mm is used for the transient method, and polycarbonate plate thicknesses of 0.178, 0.25, 0.375 and 0.5 mm are used for the LCM.

Initially the plate temperature was kept constant at room temperature and the heated jet was diverted by flow diverters consisting of removable plates. The flow diverters were used to guarantee no influence of the heated jets on the uniform initial temperature distribution of the impingement plate. With the removal of the flow diverters, the heated jets suddenly impinged on the plate. As the model surface temperature changed, a band of color moved across the liquid crystal sprayed surface. Then the measurements of required time to reach the known color display temperature would allow the solution of Eq. (3.2) for the heat transfer coefficients. Time information can be obtained by a timer(LG, SZL-WL), and temperature information by the image processing of the liquid crystal image.

The results are shown in Fig. 4, and represent the typical heat transfer characteristics of impinging jets. The heat transfer coefficient is the maximum at the stagnation region and decreases along the radial direction. And the LCM result agrees well with all the radial locations with the transient method which is known as a reliable method. And the lumped capacitance method showed nearly the same results regardless of the thickness of the polycarbonate model when the Bi of the fin is small. This means that LCM is the appropriate method for the heat transfer coefficient measurement when the Bi is less than 0.058 like this experiment.

The second experiment has been performed for a flat plate flow. The characteristic length of the plate is 76 mm, which is similar to the actual heat exchanger fin. The experimental setup for the transient method is shown in Fig. 5 and for LCM in Fig. 6. The thermal and flow conditions are identical for both methods. The heat transfer model is located at ten times of the jet diameter downstream from the jet exit, and jet exit velocity is 10.7 m/s.

The results are shown in Fig. 7, and show that as the thermal boundary layer grows, the heat transfer coefficient decreases gradually. The transient method

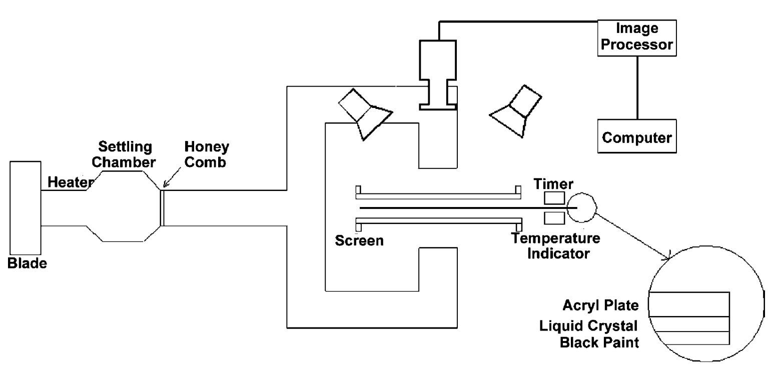

Fig. 2. Experimental setup 1 with acryl for transient method.

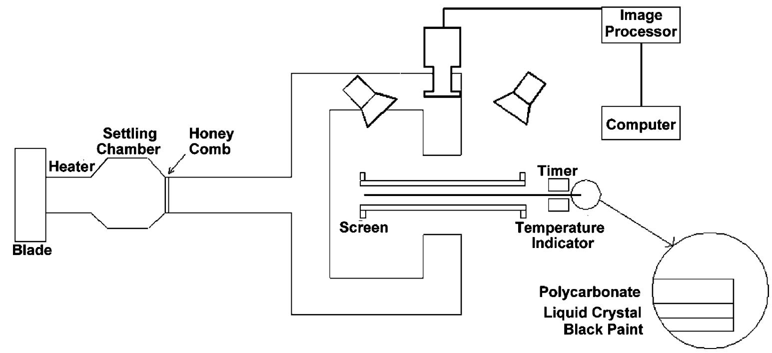

Fig. 3. Experimental setup 1 with polycarbonate for verification of LCM.

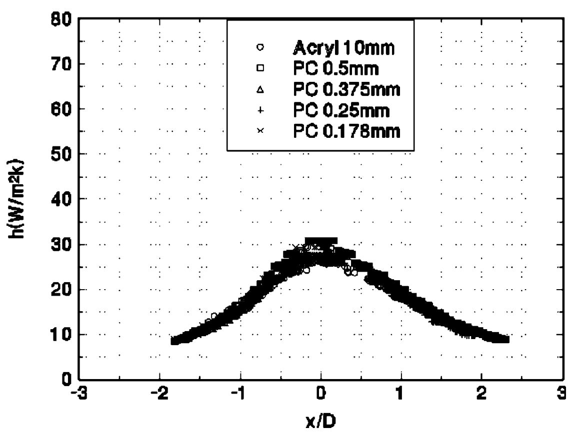

Fig. 4. Results of experiment 1.

and the LCM show similar results, and the LCM shows nearly same results regardless of thickness.

The fact that the results of the LCM shows very good agreements with those of the transient method leads us to conclude that the LCM is also a reliable heat transfer measuring tool. The method can be successfully applied to a thin foil type heat transfer model or a heat transfer model of which the *Bi* is small.

4. Fin-tube heat exchanger experiments using the LCM

4.1. Experimental apparatus and procedure

In this chapter, the LCM was applied for the measurements of the local heat transfer coefficient of a plate fin model of a home air conditioner. For convenience and accurate measurements, the plate fin model was made to be triple size of the prototype fin. The operating velocity range of the prototype was 1.0–1.5 m/s. The velocity during the experiment was kept about 0.3–0.5 m/s to have same Reynolds number with the prototype.

The experiments are carried out in a small wind tunnel. The wind tunnel consists of 2 settling chambers, a contraction body, a test section and a fan. Fig. 8 shows the schematic of the wind tunnel which is an open/suction type. Room air enters through the first settling chamber and passes through the test section and the nozzle for the flow velocity measurement in the second chamber. Finally, the fan acts as an exhaust for the air.

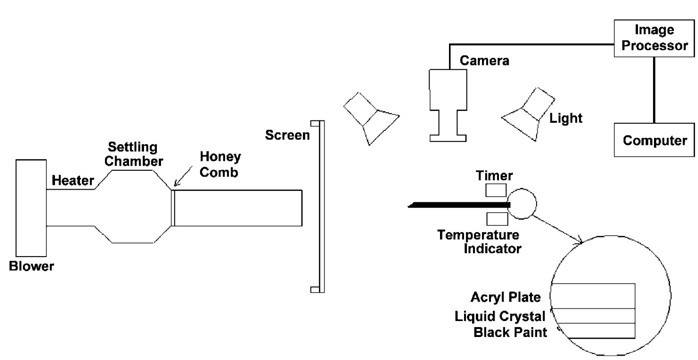

Fig. 5. Experimental setup 2 with acryl for transient method.

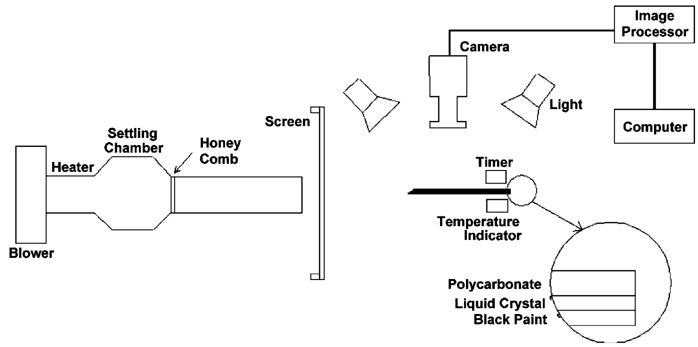

Fig. 6. Experimental setup 2 with polycarbonate for verification of LCM.

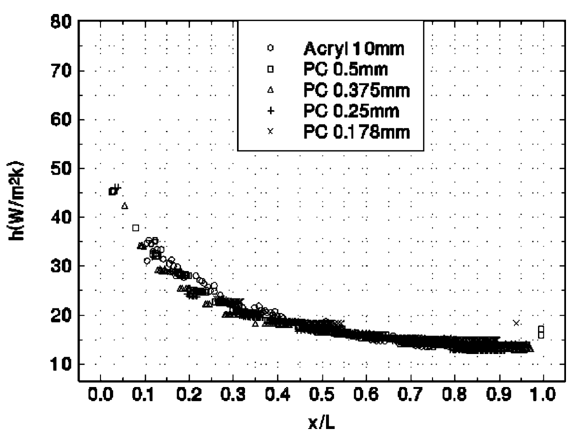

Fig. 7. Results of experiment 2.

The velocity range in the test section is 0.2–0.8 m/s. But it is very difficult to measure this velocity range with the general velocity measurement device. So an elliptic nozzle is installed in the second chamber to indirectly measure the velocity. First, the amplified nozzle exit velocity is measured, and then the velocity of the test section is converted to test section velocity by using the calibration curve. The maximum exit velocity is about 27 m/s and measured by a Pitot-tube and a digital manometer (FCO12).

The success of the LCM experiment is highly dependent on the unsteady temperature development. A step change in the convective cooling is created by rapidly injecting a hot model into a relatively cold flow. To do this, the test section for the LCM experiment is designed and fabricated as shown in Fig. 9. Fig. 10 shows the detailed view of the heat transfer model installed with fins. As shown in Figs. 9 and 10, a virtual flow passage which has the same cross section area as the part installed with the heat exchanger fin is made. The heat transfer model is kept in the constant temperature reservoir for the uniform temperature for five hours. And then the heat transfer model is moved and installed in the upper part of the test section as soon as possible. During this period, the model is covered with Styrofoam to keep the temperature of the model uniform constant. Of course the test section is also insulated. During a very short time period, the Styrofoam cover is removed and the heat transfer model is dropped down rapidly. In this way, the heat transfer model is exposed to the relatively cold flow. This method makes the disturbance of flow minimum when the heat transfer model is exposed to

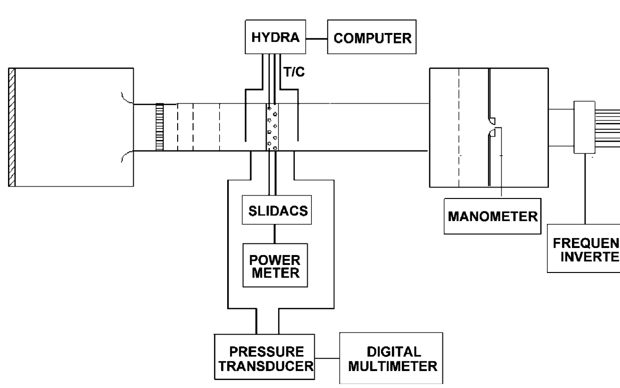

Fig. 8. Detail view of wind tunnel.

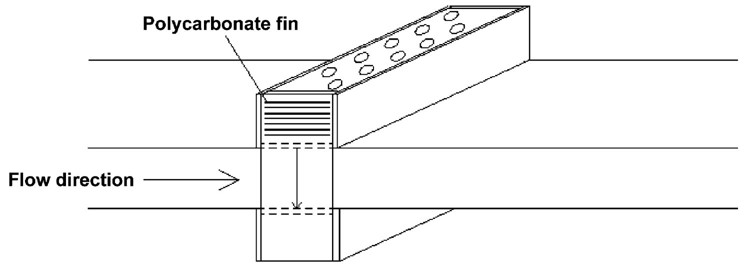

Fig. 9. Schematic of experimental setup.

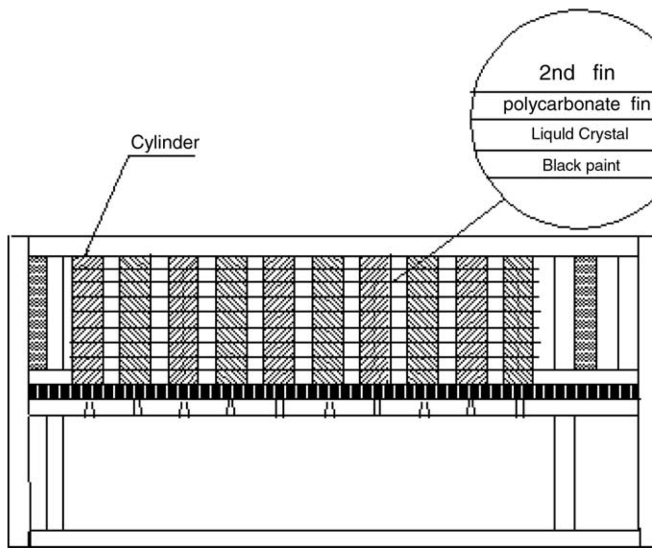

Fig. 10. Detail view of heat transfer model.

flow passage. Just before the heat transfer model is exposed to the flow, the initial temperature of the heat transfer model is measured. The initial temperature is measured by the thermocouple which is installed in a very small hole on the upper surface of the heat transfer model. Since keeping the heat transfer model in the constant temperature reservoir for a long time means that the heat transfer model keeps a uniform temperature, it is assumed that the temperature measured from one point represents the initial temperature of the heat transfer model. From the moment when the heat transfer model is exposed to the flow, the heat transfer model of the uniform temperature begins to be cooled. At this time, the timer is on and the color image of the heat transfer model and time information are stored in the computer simultaneously by the 3CCD camera. By replaying the stored image frames, the heat transfer coefficient can be measured using the time information and the surface temperature which is based on the hue and temperature relationship. The heat transfer coefficient for the whole surface of the heat transfer model is obtained by superposition of the heat transfer coefficient from each frame obtained at different times.

The experiments have been performed for the plate fin heat exchanger model. Seven fins are combined with ten cylinder-shaped tubes, and the spacing between the fins is kept constant using annular rings. To investigate the vertical position effect of the fin, the experiments have been performed with three different locations of liquid crystal coated fins. The result shows that the color image of the first layer from the wall appears late compared to the other layers, while the other layer shows after a long time period, which means that the heat transfer coefficient is much lower than those of other cases. After a long time interval, there is no change for the liquid crystal image, so it is implied that there is difficulty in measuring the heat transfer coefficient when the fin is located on the first layer. But the second and the third layer cases show that the same liquid crystal image appears at the same time, and therefore the heat transfer coefficients for the two cases are nearly the same. Since there is no difference for the image between the second and the third layer cases, and at the second layer more vivid images can be captured, all the experiments are performed under the condition that the fin coated with liquid crystal is located at the second layer.

The liquid crystal used for the heat transfer coefficient measurement is R35C2W, which means red starts at 35 °C and violet ends at 37 °C. The hue and temperature relationship is shown in Fig. 11, from captured images black is bound with blue, so this band (yellow) is used for the temperature information. As shown in Fig. 11, the hue of this color-band represents 35.5 °C, so this temperature of the color-band is substituted for the wall temperature of the Eq. (3.2), and therefore the heat transfer coefficients are obtained. Since this experiment is a cooling process, the part which shows the color band earlier has the higher heat transfer coefficient.

4.2. Results and discussion

The experiment is performed under the condition that the frontal velocity is 0.3 m/s, the temperature of the free stream is 17.5 °C and the initial temperature is 60.1 °C. The seven consecutive images recorded during the passage of the colors from the liquid crystal are processed to obtain the convection heat transfer coefficient of the fin. Three representative images out of the seven images are shown in Fig. 12 with their associated time, measured from the beginning of the experiment. When the section with fins was inserted into the test section, the images from the fin was completely dominated by black as an indication of a reasonably uniform initial temperature of 60.1 °C. The blue color begins to appear on the front side of the fin in one second. As time goes on and the fin is cooled, the blue band moves into the downstream and green, yellow and red color continuously appear. During the experiment, this color-band moves into the rear side of the fin. These time variant color changes represent the heat transfer characteristic,

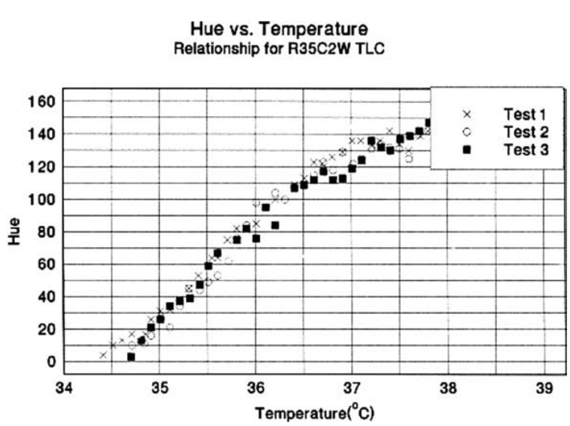

Fig. 11. Hue versus temperature relation for R35C2W.

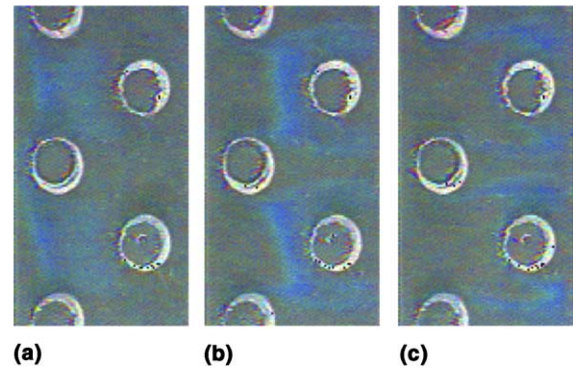

Fig. 12. Real images for plate fin (frontal velocity = 0.3 m/s). (a) After 16.6 s, (b) after 27.4 s and (c) after 39.3 s.

which means that the frontal area with the earlier liquid crystal color appearance is well cooled.

In this study, the captured seven images and time information are restored and the heat transfer coefficients are investigated. Fig. 12(a) is the second liquid crystal image after 16.6 s from the beginning of the experiment. The color boundary of the image means the same temperature and same heat transfer coefficient. Substituting the corresponding time 16.6 s and boundary temperature 35.5 °C into the Eq. (3.2), the heat transfer coefficient of 13.75 W/m^2K is obtained. The area in front of this color boundary shows the earlier color band, so it has the higher heat transfer coefficient than 13.75 W/m^2K. The color boundary meaning the same heat transfer coefficient was formed perpendicularly to the flow direction on the front side of the fin. The color band between the tubes droops downward. This means that the middle area between the tubes has the higher heat transfer coefficient resulting from the fast flow velocity of the middle area.

Fig. 12(b) is the fourth liquid crystal images at 27.4 s. The color boundary shown in Fig. 12(b) is located at the more rear area, but around the tube there is little difference for the color boundary location compared with Fig. 12(a). By substituting time information 27.4 s and boundary temperature 35.5 °C for the Eq. (3.2), the heat transfer coefficient of 8.33 W/m^2K is obtained.

Fig. 12(c) is the sixth liquid crystal image at 39.3 s. Fig. 12(c) shows that the color boundary moves into the further downstream area and extends between tubes compared with Fig. 12(b). The color boundary starts in front of the front tube, turns around the tube and droops backward. When the color boundary crosses over the rear tube, it moves around the rear tube again. By substituting time information 39.3 s and boundary temperature 35.5 °C for the Eq. (3.2), the heat transfer coefficient 5.81 W/m^2K is obtained.

Fig. 13 shows the distribution of heat transfer coefficient for the plate fin from the seven images. The results

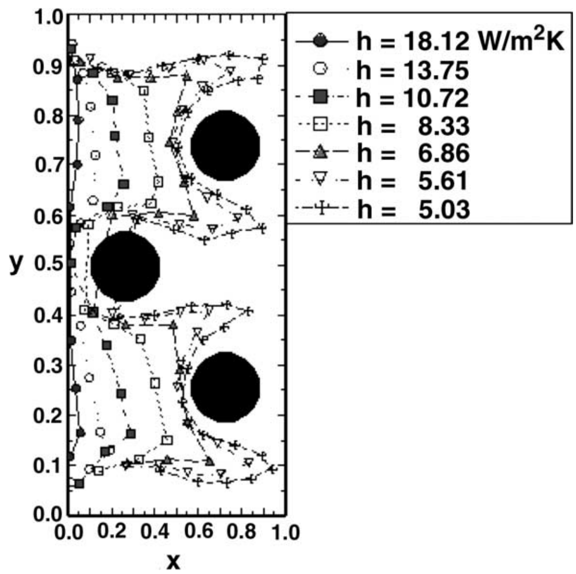

Fig. 13. Heat transfer coefficient distribution map for plate fin (frontal velocity = 0.3 m/s).

show that the higher heat transfer coefficient appears in the front area of the fin and the lower heat transfer coefficient appears in the rear area owing to slow cooling. Also results show that the higher heat transfer coefficient appears in the flow passage which has fast flow velocity rather than the area around the tube. While there is an area that has higher heat transfer coefficient than 18.12 W/m^2K, the rear area of the rear tube has a lower heat transfer coefficient than 5.03 W/m^2K.

To investigate the heat transfer characteristics with the frontal velocity, the images captured from the experiment for the frontal velocity 0.6 m/s are shown in Fig. 14 and the distribution of the heat transfer coefficient is shown in Fig. 15. The free stream temperature is 25.8 °C, and the initial temperature is 55 °C. As shown in Figs. 12 and 13, even though the frontal velocity changes, there is little difference for the color band pattern.

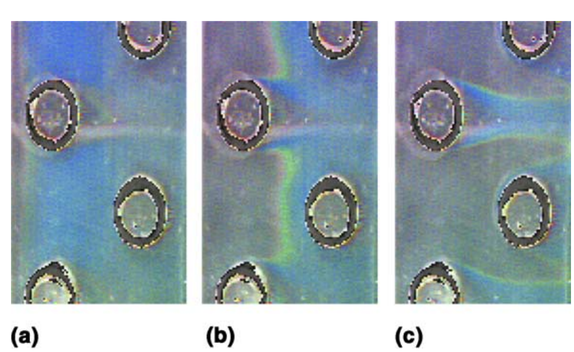

Fig. 14. Real images for plate fin (frontal velocity = 0.6 m/s). (a) After 7.3 s, (b) after 15.7 s and (c) after 23.8 s.

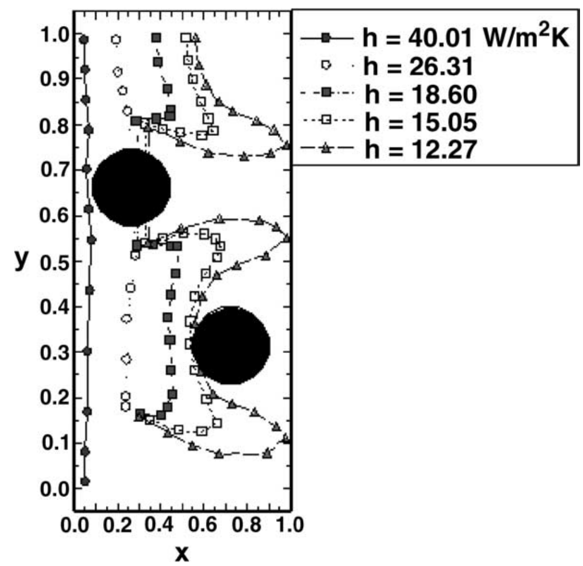

Fig. 15. Heat transfer coefficient distribution map for plate fin (frontal velocity = 0.6 m/s).

As shown in Fig. 15, there is a difference just for the absolute value of the heat transfer coefficient. Since the frontal velocity increases, the heat transfer coefficient becomes high, but the proceeding of the color band appears similarly to that of the case of 0.3 m/s.

The maximum errors in the measurements at better than 95% confidence level are estimated as Table 1. The uncertainty of heat transfer coefficient (h) is the combination of the above listed errors and is obtained by using the error combination principle suggested by Kline and McClintock [14].

5. Conclusions

In the complex and thin plate type heat transfer model like the heat exchanger fin, it is not easy to measure the local heat transfer coefficient. In this study, the local heat transfer characteristics of the heat exchanger fin were investigated experimentally by using the LCM. For the purpose of confirming whether the LCM is suitable to heat transfer coefficient measurement, the LCM was compared with the result of the well-known transient test. To investigate the possibility for the heat exchanger fin, experiments were performed for the fin and tube model. With these experiments, the results are as follows:

(1) Since both impinging jet and plain plate (parallel to jet direction) experiments shows good agreement with the LCM, the LCM is suitable to heat transfer coefficient measurement.

Table 1
Uncertainty analysis

$\frac{\delta\rho}{\rho}$	0.7%
$\frac{\delta L_c}{L_c}$	0.7%
$\frac{\delta c}{c}$	2.0%
$\frac{\delta t}{t}$	0.2%
$\frac{\delta(T_i - T_\infty)}{(T_i - T_{ref})}$	0.4%
$\frac{\delta(T - T_\infty)}{(T - T_\infty)}$	1.4%
$\frac{8h}{h} - \left[\left(\frac{8\rho}{\rho}\right)^2 + \left(\frac{8L_c}{L_c}\right)^2 + \left(\frac{8c}{c}\right)^2 + \left(\frac{8t}{t}\right)^2 + \left(\frac{8(T_i - T_\infty)}{(T_i - T_\infty)}\right)^2 + \left(\frac{8(T - T_\infty)}{(T - T_\infty)}\right)^2\right]$	2.6%

(2) Since the LCM using Polycarbonate shows the same results irrespective of thickness, the LCM is suitable to a heat transfer coefficient measurement for *Bi* that is less than 0.058.
(3) The possibility for a heat transfer characteristic investigation of a plain fin was confirmed, and it is known that it is possible to analyze heat transfer characteristics of fins with more complicated slits.

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