1996

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T. H. Lee
Hanyang University

K. S. Lee
Hanyang University

W. S. Kim
Hanyang University

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Tae-Hee Lee[1], Kwan-Soo Lee[2], Woo-Seung Kim[2]

1 : Graduate Research Assistant, Hanyang University, Seoul, 133-791, Korea
2 : Professor, Dept. of Mechanical Eng., Hanyang University, Seoul, 133-791, Korea

ABSTRACT

An experimental study on the effects of various factors(fin spacing, fin arrangement, air temperature, air humidity, and air velocity) on the frost growth and thermal performance of a finned-tube heat exchanger has been conducted under the frosting condition. It is found that the thermal performance of a heat exchanger is closely related to the blockage ratio of air flow passage due to the frost growth. The maximum allowable blockage ratio is used to determine the criteria for the optimal operation conditions of a finned-tube heat exchanger.

1. INTRODUCTION

The frost layer formed on finned-tube heat exchanger such as an evaporator of refrigerator not only blocks the heat flow between moist air and refrigerant, but also decreases the thermal performance of the heat exchanger by interfering the air flow. The results of the previous studies on the heat transfer and air side pressure drop in the finned-tube heat exchanger under the frosting condition can be summarized as follows. Stoecker[1], Hosoda et al.[2], and Gatchilov et al.[3] reported that the heat transfer rate increases at the initial stage of frosting and then decreases with time, whereas Aoki et al.[4] showed the opposite trend. Rite et al.[5] concluded that the heat transfer rate increases continuously with a frost growth. Schneider[6] maintained that the frost growth rate is not a function of Reynolds number. O'Neal[7], Kamei et al.[8], and Yamakawa et al.[9] reported that the growth of frost layer is affected by Reynolds number only below certain value of Reynolds number. Reid et al.[10] and Trammel et al[11] presented different results on the effect of air humidity on the density of frost layer. Many researchers indicated that the decrease of air flow rate caused by pressure drop is the index which represents the bad influence of the frost formation on the heat transfer performance[1,12~14].

In the present study, using 2-row, 2-column finned-tube heat exchanger, the effects of design factors (fin spacing and fin arrangement) and operating conditions (temperature, humidity, and velocity of air) on the growth of frost layer and the performance of the heat exchanger under frosting condition are investigated.

2. EXPERIMENT

2.1 Experimental apparatus

Figure 1 shows the experimental apparatus used in the present work. The apparatus is composed of four sections, such as test section, circulation section, climate chamber and cooling section. To see the frosting phenomena with various types of heat exchanger, the test section was constructed of acryl plate with 0.02m thickness and was insulated by detachable styrofoam with 40mm thickness. By altering the number of revolutions of the fan with 0.5HP, the circulation section could control air flow rate induced into the test section. The volume of the climate chamber was 252 l and it controlled the temperature and humidity of air. The cooling section was composed of a pump with 0.5HP and a refrigerator with 5HP to control the flow rate and temperature of the refrigerant circulated in the test heat exchanger. Figure 2 shows the finned-tube heat exchanger used in the present work.

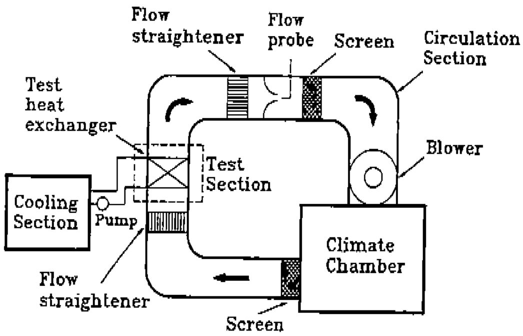

Fig. 1 Schematic diagram of experimental apparatus

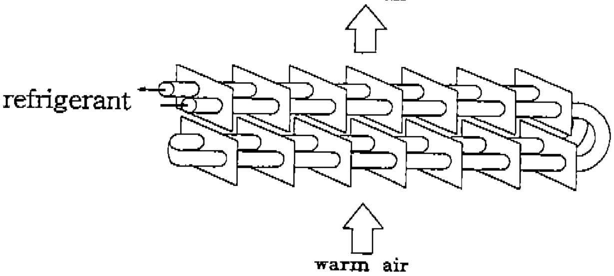

Fig. 2 Finned-tube heat exchanger used in this study

2.2 Data analysis

Heat transfer rate

Total heat transfer rate of air through the heat exchanger can be expressed as the sum of sensible heat transfer due to the variation of air temperature and latent heat transfer due to the phase change of water vapor. Since the mass of water vapor contained in humid air has the order of 10^{-3}, the sensible heat transfer of remaining water vapor could be neglected as follows:

$$q_T = q_l + q_s \simeq \dot{m}_a(i_{a,i} - i_{a,o}) \tag{1}$$

where,

$$i_{a,i} = c_{p,a} \cdot T_{a,i} + L_H \cdot w_{a,i} \tag{2a}$$

$$i_{a,o} = c_{p,a} \cdot T_{a,o} + L_H \cdot w_{a,o} \tag{2b}$$

Energy transfer coefficient

To calculate the energy transfer coefficient on the surface of frost layer, the enthalpy on the surface of frost layer must be evaluated. To obtain this enthalpy, the surface temperature of frost layer is needed. The temperature can be calculated by using the analogy of heat and mass transfer as follows:

$$\frac{h_m \cdot c_{p,a}}{h_a} = \frac{q_l \cdot c_{p,a} \cdot (T_a - T_{fs,av})}{q_s \cdot [\ w_a - w_s(T_{fs,av})\ ] \cdot L_H} = 1 \tag{3}$$

where $w_s(T_{fs,av})$ represents the saturation absolute humidity at mean temperature, $T_{fs,av}$, on the surface of frost layer. Based on mean temperature, $T_{fs,av}$, calculated from equation (3), the mean enthalpy on the surface of frost layer is expressed as follows:

$$i_{fs,av} = c_{p,a} \cdot T_{fs,av} + L_H \cdot w_s(T_{fs,av}) \tag{4}$$

Hence, the energy transfer coefficient is given by:

$$E_a = \frac{q_T}{A_T \cdot (i_a - i_{fs,av})/c_{p,a}} \tag{5}$$

Energy transfer resistance

The energy transfer resistance between air and coolant is given by equation (6), based on the enthalpy difference of air and coolant. The average temperature of coolant is used to obtain the average enthalpy of coolant.

$$R_e = \frac{(i_a - i_c)/C_{p,a}}{Q_T} \tag{6}$$

3. EXPERIMENTAL RESULTS AND DISCUSSIONS

In this section, the effects of design factors(fin spacing and fin arrangement) and operating conditions(temperature, humidity, and velocity of air stream) on the frost growth and performance of finned-tube heat exchanger are discussed. The specifications of test heat exchanger are shown in Table 1 and baseline operation condition is shown in Table 2.

3.1 Frost growth

Thickness of frost formed on the fin and tube for in-line type finned-tube heat exchanger with

Table 1 Coil geometry and design factors of finned-tube heat exchanger

	Component	Spec.	Component	Spec.
Design Conditions	Number of columns	2	Number of rows	2
	Transverse tube spacing	27 (mm)	Longitudinal tube spacing	30 (mm)
	Tube ID	6 (mm)	Tube OD	8 (mm)
	Tube length	370 (mm)	Fin pitch	20 (mm)
	Fin type	Flat type	Fin thickness	0.2 (mm)

Symbol	Fin spacing	Operation condition	Fin type	Symbol	Fin spacing	Operation condition	Fin type
●*	20.0	Baseline	In-line type	○	20.0	Baseline	Staggered type
☆	10.0			△	10.0		
★	7.5			□	7.5		
✡	5.0			◇	5.0		

* Baseline design condition

Table 2 Baseline operation condition

Air inlet temperature	6 °C	Air inlet humidity	70 %
Air inlet velocity	1.0 m/s	Coolant mean temperature	-30 °C

fin spacing of 20 mm for baseline operation condition is illustrated in Fig. 3. In this figure, average thickness of frost given by eq.(7) is also presented.

$$X_{f,av} = \frac{X_{f,F} \cdot A_{F,0} + X_{f,t} \cdot A_{t,0}}{A_{T,0}} \tag{7}$$

The thickness of frost formed on the fin and tube is average value of thickness of frost formed on the first and sencond row. Frost layer grows rapidly at early stage of frost formation, and then the growth rate decreases with time. Fast growth of frost layer at early stage is due to large flux of water vapor from air to heat exchanger surface. The large vapor flux results from low temperature of heat exchanger surface. The frost thickness on tube surface is thicker than the one on fin. Since the surface temperature of tube is lower than that of fin, the water vapor flux onto the tube surface has a larger value than that onto the fin.

3.2 Effects of design factors

Effects of fin spacing

Figure 4 represents the variation of frost thickness formed on fin with respect to fin spacing. It is observed that decreasing in fin spacing reduces the frost thickness. This is due to the fact that the heat transfer coefficient decreases as described in Ref[15], and hence the mass transfer rate per unit area decreases by the analogy between heat and mass transfer.

Figure 5 shows that frost density increases with decreasing fin spacing. This results from the fact that the surface temperature of frost layer becomes high, the thickness of frost layer becomes thinner with fin spacing getting closer, and hence the amount of water vapor diffused into frost layer increases.

Figure 6 depicts the variation of heat transfer rate with fin spacing. There is little change in the amount of heat transfer with time in the case of a large fin spacing, but the significant change of it with time is noticed in a small fin spacing. The heat transfer rate increases at the initial stage of frost formation. This results from the fact that as Hayashi et al.[16] asserted, the ice-like frost nuclei generated during the crystal growth period acts as a small fin. During frost layer growth period, frost nuclei grows to porous frost layer, and the thermal insulation effect of the frost layer increases because porous frost layer acts like a thermal insulator. Hence, heat transfer rate decreases. During the latter term of the frost layer growth period, however, the heat transfer rate increases and reaches maximum value due to the increase of the air velocity and frost surface roughness. However, as thermal resistance of frost layer increases and heat transfer area decreases, heat transfer rate decreases again. Since the period during which nuclei of frost acts like a small fin is very short compared with the total operating time of heat exchanger, it partially agrees with the results of Aoki et al.[4], who reported that heat transfer rate decreases at initial stage of frosting and then increases. The previous works[1~3] also stated that heat transfer rate increases to reach a maximum value and then decreases. Therefore, the present results also partially agree with the previous results[1~3]. As the fin spacing

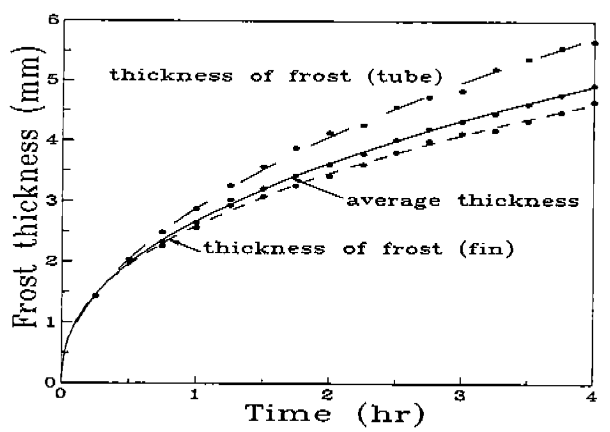

Fig. 3 The variation of frost thickness with time

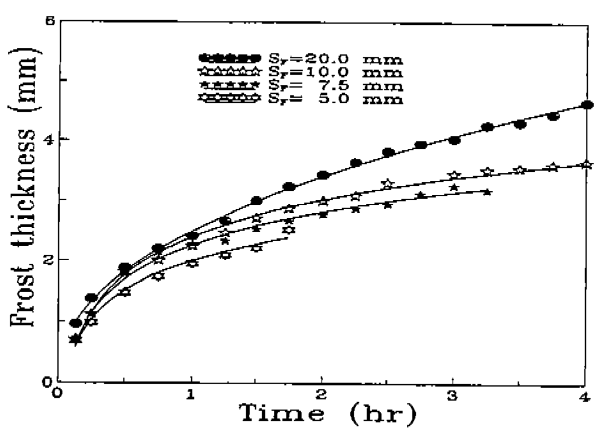

Fig. 4 The variation of frost thickness with respect to fin spacing

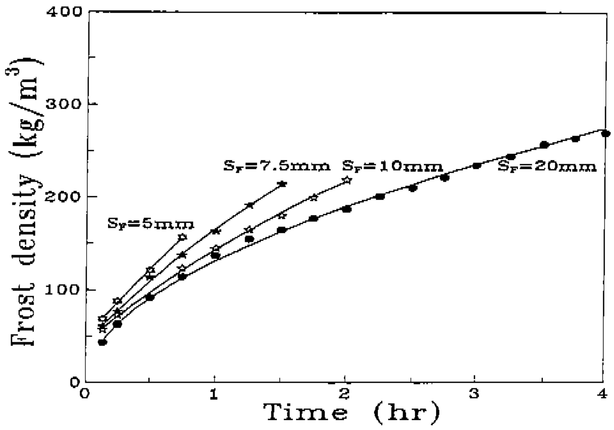

Fig. 5 The variation of frost density with respect to fin spacing

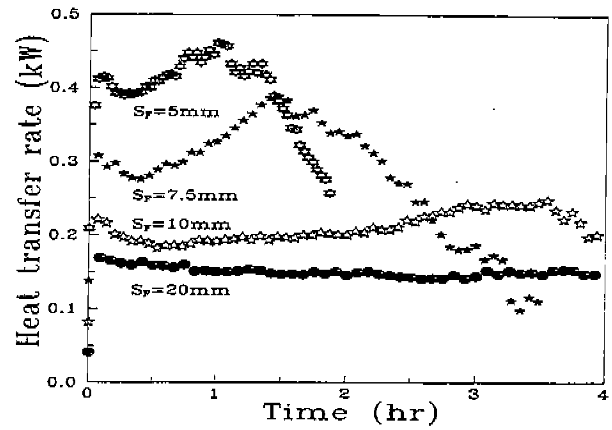

Fig. 6 The variation of heat transfer rate as a function of fin spacing

getting closer, the heat transfer rate increases, and the time required to reach a maximum value is getting shorter.

To set up the criteria for optimal operation condition of a finned-tube heat exchanger, blockage ratio(BR) is introduced and defined as

$$BR = 2\times\frac{X_{f,F}}{S_F}\times 100 \ [\ \%\ ] \tag{8}$$

It represents the ratio of air flow passage blocked by frost layer. The blockage ratio increases with decreasing fin spacing. Especially, in the case of 5 mm fin spacing, the flow passage is almost fully blocked after 2 hours, and in the case of 7.5mm fin spacing, more than 80 percents of the flow passage is blocked after 3 hours.

Figure 7 represents the variation of heat transfer rate with blockage ratio. It shows that when the blockage ratio is less than 10%, the heat transfer rate increases at initial stage of frosting due to the influence of frost nuclei. And then, the heat transfer rate decreases due to the influence of thermal resistance of frost layer until the blockage ratio reaches about 35~45%. After that it increases to a maximum value and then it decreases with a relatively constant gradient irrespective of fin spacing. Therefore, it is desirable to operate the finned-tube heat exchanger under the certain value of blockage ratio at which heat transfer rate has a maximum value. The maximum allowable thickness of frost layer can be represented as follows:

$$X_{f,F,max} = \frac{S_F}{2}\times BR_{max}(S_F) \tag{9}$$

where the maximum allowable blockage ratio, $BR_{max}(S_F)$, is derived from the least square method and is expressed as follows:

$$BR_{max}(S_F) = 89.1 + 2.24\times S_F \tag{10}$$

where applicable range of eq. (10) is $5 \le S_F \le 20$ mm.

Effect of fin arrangement

Figure 8 depicts the variation of the ratio of total heat transfer rate of staggered finned-tube heat exchanger to that of in-line type heat exchanger. It shows that heat transfer rate of heat exchanger with staggered fin array increases about 17% compared with in-line type heat exchanger. This is due to the fact that heat transfer coefficient of air side is enhanced with staggered fin array. The cases with fin spacing of 7.5 mm and 5 mm show less inhancement in heat transfer. This is attributed to the fact that the blockage effect of flow passage increases due to frost layer formed on the fins.

3.3 The effects of the operation conditions

In this section, the effects of temperature, humidity and velocity of air stream on the frost growth and energy transfer resistance are examined to investigate the performance of a finned-tube heat exchanger under the experimental conditions shown in Table 3.

Effect of inlet air temperature

Table 4 shows the effects of inlet air temperature on the frost growth and energy transfer

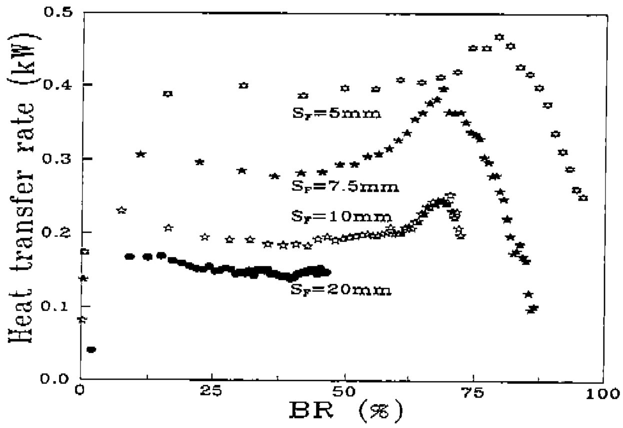

Fig. 7 The variations of heat transfer rate with blockage ratio as a function of fin spacing

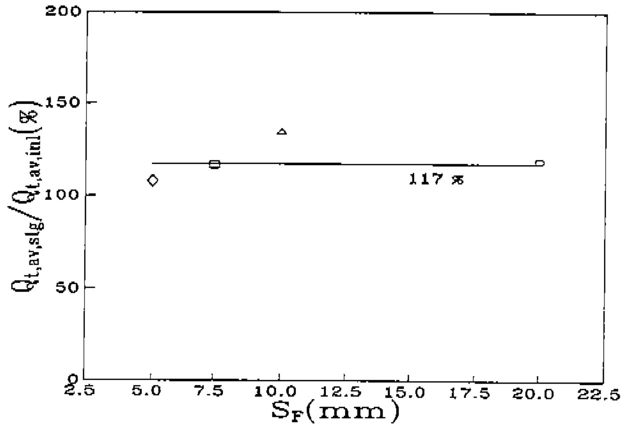

Fig. 8 The variation of the ratio of heat trasfer rate of staggered finned-tube heat exchanger to that of in-line type heat exchanger as a function of fin spacing

Table 3 Operation conditions

Experiment number	Air inlet temp.	Air inlet humid.	Air inlet velocity	Coolant mean temp.	Experiment number	Air inlet temp.	Air inlet humid.	Air inlet velocity	Coolant mean temp.
1*	6 ℃	70.0 %	1.00 m/s	-30 ℃	6	6 ℃	63.0 %	1.00 m/s	-30 ℃
2	10 ℃	53.3 %	1.00 m/s	-30 ℃	7	6 ℃	77.0 %	1.00 m/s	-30 ℃
3	8 ℃	61.0 %	1.00 m/s	-30 ℃	8	6 ℃	70.0 %	0.50 m/s	-30 ℃
4	4 ℃	80.5 %	1.00 m/s	-30 ℃	9	6 ℃	70.0 %	0.75 m/s	-30 ℃
5	6 ℃	55.0 %	1.00 m/s	-30 ℃	10	6 ℃	70.0 %	1.50 m/s	-30 ℃

* : baseline operation condition

Table 4 Effect of inlet air temperature

Experiment number	Average frost thickness (at time=200min)	Frost density (at time=200min)	Energy transfer resistance (average value)
2	3.98 mm	330.64 kg/m^3	0.242 °C/W
3	4.48 mm	307.01 kg/m^3	0.250 °C/W
1	4.90 mm	265.02 kg/m^3	0.274 °C/W
4	5.67 mm	245.28 kg/m^3	0.303 °C/W

resistance. The frost layer with large density and thin thickness is formed with increasing air temperature. This is due to the fact that the amount of vapor transferred into frost layer increases with increasing temperature gradient and increasing vapor pressure gradient in frost layer, resulting from the increase of energy flux into the frost layer. The energy transfer resistance has a small value with increasing inlet air temperature. This is due to the fact that the frost thickness is thin and frost density is high with increasing inlet air temperature.

Effect of inlet air humidity

Table 5 shows the effects of inlet air humidity on the frost growth and energy transfer resistance. The frost thickness becomes thick and the frost density becomes low because of a larger potential of mass transfer by more water vapor in air. The energy transfer resistance has a large value

Table 5 Effect of inlet air humidity

Experiment number	Average frost thickness (at time=200min)	Frost density (at time=200min)	Energy transfer resistance (average value)
5	3.78 mm	294.19 kg/m^3	0.242 °C/W
6	4.31 mm	279.45 kg/m^3	0.252 °C/W
1	4.90 mm	265.02 kg/m^3	0.274 °C/W
7	5.76 mm	243.42 kg/m^3	0.294 °C/W

with increasing inlet air humidity. This results from the fact that the frost thickness is thick and frost density is low with increasing inlet air humidity.

Effect of initial inlet air velocity

Table 6 shows the effect of inlet air velocity on frost growth and energy transfer resistance. Frost thickness and density increase with increasing inlet air velocity. This phenomena can be attributed to the fact that the increase of mass transfer rate and temperature gradient in frost layer results from a large amount of mass and energy transfer due to high velocity of air . Energy transfer resistance has a small value with a high inlet air velocity because convective resistance decreases with increasing air velocity.

Table 6 Effect of inlet air velocity

Experiment number	Average frost thickness (at time=200min)	Frost density (at time=200min)	Energy transfer resistance (average value)
8	4.10 mm	256.58 kg/m^3	0.351 °C/W
9	4.49 mm	260.36 kg/m^3	0.298 °C/W
1	4.90 mm	265.02 kg/m^3	0.274 °C/W
10	5.52 mm	281.16 kg/m^3	0.249 °C/W

4. CONCLUSIONS

In this study, the effects of various factors(fin spacing, fin arrangement, air temperature, air humidity, and air velocity) on the frost growth and thermal performance in a flat plate finned-tube heat exchanger has been conducted under the frosting condition. The conclusions obtained from present work are as follows :

1. The thicker frost layer is formed on the tube surface compared with the one on the fin surface.
2. With decreasing fin spacing, the frost growth rate is reduced and the frost density increases.
3. There is little change in the amount of heat transfer with time in the case of a large fin spacing, but the significant change of it with time is noticed in a small fin spacing.
4. The thermal performance of heat exchanger is closely related to the blockage ratio of air flow passage due to frost growth, and maximum allowable blockage ratio is obtained.
5. Staggered finned-tube heat exchanger shows an improvement of about 17 % in heat transfer rate.
6. An increasing inlet air temperature results in the reduction of frost thickness, the increase of frost density, and the decrease of thermal resistance.
7. An increasing air humidity leads to a thicker frost layer, the decrease of frost density, and the increase of thermal resistance.
8. An increasing air velocity results in the reduction of thermal resistance, the increase of the frost thickness and frost density.

REFERENCE

[1] Stoecker, W. F., 1957, Refrigeration Engineering, Vol. 65, No. 2, pp. 42-46
[2] Hosoda, T. and Uzuhashi, H., 1967, Hitachi Review, Vol. 16, No. 6, pp. 254-259
[3] Gatchilov, T. S. and Ivanova, V. S., 1979, Int. J. Refrig., Vol. 2, No. 4, pp. 233-236
[4] Aoki, K. and Hattori, M. and Hayashi, Y., 1985, JSME Trans., Ser.B, Vol.51, No.469, pp.3048-3054
[5] Rite, R. W., and Crawford, R. R., 1991, ASHRAE Transactions, Vol. 97, Pt. 2, pp. 428-437
[6] Schneider, H. W., 1978, Int. J. of Heat and Mass Transfer, Vol. 21, No. 3, pp. 1019 - 1024.
[7] O'Neal, D. L., 1983, Ph. D. Thesis, Purdue University.
[8] Kamei, S., Mizushima, T., Kifune, S. and Koto, T., 1952, Jap. Science Review, Vol.2, No.3, pp. 317-326.
[9] Yamakawa, N., Takahashi, N., and Ohtani, S., 1972, Heat Transfer Jap.-Res., Vol.1, No.2, pp.1-10.
[10] Reid, R. C., Brian, P. L. T., and Weher, M. E., 1966, A.I.Ch.E.-J., Vol. 11., pp. 1190 - 1195.
[11] Trammel, G. J., Little, D. C., and Killgore, E. M., 1968, ASHRAE-J., Vol. 10, No. 7, pp. 42 - 47.
[12] Barrow, H., 1985, Journal of Heat Recovery Systems, Vol. 5, No. 3, pp. 195-201.
[13] Gates,R.R. and Sepsy,C.F. and Huffman,G.D., 1967, ASHRAE Trans., Vol.73, Pt.2, pp.I.2. 1-I. 2. 13.
[14] Niederer, D. H., 1976, ASHRAE Transactions, Vol. 82, Pt. 1, pp. 467-473.
[15] Gray, D. L., and Webb, R. L., 1986, Proc. of 8th Int. Heat Transfer Conference, San Francisco, ASME, Vol. 6, pp. 2745-2750.
[16] Hayashi, Y., Aoki, A., and Yuhana, H., 1977, Heat Transfer, Jap. Res., Vol. 6, No. 3, pp. 79-94.