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International Journal of Heat and Mass Transfer 50 (2007) 4864–4871

International Journal of HEAT and MASS TRANSFER

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Condensation heat transfer of R-134a inside a microfin tube with different tube inclinations

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Received 2 February 2006; received in revised form 13 February 2007
Available online 2 May 2007

Abstract

Experimental heat transfer studies during condensation of pure R-134a vapor inside a single microfin tube have been carried out. The microfin tube has been provided with different tube inclination angles of the direction of fluid flow from horizontal, α. The data are acquired for seven different tube inclinations, α, in a range of -90 to $+90°$ and three mass velocities of 54, 81, and 107 kg/m^2-s for each inclination angle during condensation of R-134a vapor. The experimental results indicate that the tube inclination angle of, α, affects the condensation heat transfer coefficient in a significant manner. The highest heat transfer coefficient is attained at inclination angle of $\alpha = +30°$. The effect of inclination angle, α, on heat transfer coefficient, h, is more prominent at low vapor quality and mass velocity. A correlation has also been developed to predict the condensing side heat transfer coefficient for different vapor qualities and mass velocities.

Keywords: Microfin tube; Condensation; Heat transfer; Inclination; Two-phase flow

1. Introduction

The condensers have a very important role to play in different industries such as in refrigeration, air-conditioning, power plants and chemical industries. As the energy sources are limited and for their conservation, appropriate design and optimization of condensers is very important. Therefore, different methods have been used by different investigators to increase heat transfer rate in these condensers [1]. In refrigeration and air-conditioning industries, due to the high wettability of the refrigerants, only film-wise condensation is observed and since the thermal conductivity of refrigerants is low, it is desired to augment the heat transfer coefficient in refrigerant side of condensers. There are number of techniques to enhance the heat transfer coefficient [2,3]. One of the passive techniques to enhance heat transfer coefficient is the use of microfin tubes.

Numerous researchers have carried out experiments to study the effect of fin geometry [4], presence of lubricating oil in refrigerant [5] and, different refrigerants flows [6] on the performance of microfin tubes. A review of the existing literature reveals that, although vast studies have been done on heat transfer enhancement in these tubes, yet the focus of almost all of the studies is to study the condensation during refrigerant flow in a horizontal tube. In fact, the mechanism of heat transfer augmentation in microfin tubes is dependent on the flow regime of two-phase flow. The flow regime is also influenced by interfacial shear stress, surface tension and gravitational force. Thus, there is a great necessity to consider and study the effect of gravitational force on heat transfer rate during condensation of refrigerants inside a tube. Therefore, an experimental investigation has been carried out to study the condensation of R-134a vapor inside a microfin tube with different inclinations of the tube.

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doi:10.1016/j.ijheatmasstransfer.2007.02.030

Nomenclature

c_{pw}	specific heat of cooling water, J kg^{-1} K^{-1}
c_{pf}	specific heat of liquid refrigerant, J kg^{-1} K^{-1}
D	internal diameter of test-section, m
D_o	outside diameter of test-section, m
e	fin height, m
G	mass velocity, refrigerant mass flow rate per unit cross-sectional area, kg m^{-2} s^{-1}
h	heat transfer coefficient, W m^{-2} K^{-1}
$\bar{h}$	average heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity of tube material, W m^{-1} K^{-1}
k_f	thermal conductivity of liquid phase, W m^{-1} K^{-1}
L	length of test-section, m
m_w	mass flow rate of cooling water, kg s^{-1}
Nu	Nusselt's number, defined in Eq. (3)
p	fin pitch, m
Pr_f	Prandtl number of liquid refrigerant, defined in Eq. (3)
Re_f	Reynold's number of liquid refrigerant, defined in Eq. (3)
t	test-section tube wall thickness, m
T_{ci}	inlet temperature of cooling water, K
T_{co}	outlet temperature of cooling water, K
T_s	average saturation temperature of vapor, K
T_{wi}	average inside test-section wall temperature, K
T_{wo}	average outside test-section wall temperature, K
x	vapor quality or dryness fraction of vapor
X_{tt}	Lockhart–Martinelli parameter, defined in Eq. (3)
α	test-section tube inclination angle, °
θ	fin tip angle, °
γ	helix angle, °
μ_f	liquid phase dynamic viscosity, Pa s
μ_v	vapor phase dynamic viscosity, Pa s
ρ_f	liquid phase density, kg m^{-3}
ρ_v	vapor phase density, kg m^{-3}

2. Experimental set-up and procedure

The test set-up was a well instrumented vapor compression refrigeration system. The schematic diagram of experimental set-up is shown in Fig. 1. The set-up included a test-condenser (3), a pre-condenser (2), an after-condenser (4), and a by-pass line. The test-condenser was a 1040 mm long double pipe counter-flow heat exchanger as shown in Fig. 1. The cooling water flowed in the annulus and the refrigerant flowed inside the internal microfin tube. In order to change the inclination angle of the test-condenser the connections to this condenser were made by flexible pressure hoses (14). The refrigerant vapor was circulated inside the test-condenser with the help of a reciprocating compressor (1). In order to cover the whole domain of vapor quality a pre-condenser (2) was used. The flow rate of water in the pre-condenser was regulated in order to control the quality of refrigerant vapor entering the test-condenser. The pre-condenser (2) was connected to the test-condenser with the help of a flexible pipe. The fluid emerging from test-condenser was passed through the post-condenser (4) for the complete condensation of R-134a vapor. The condensed R-134a enters the rotameter (5), expands in expansion valve (9) and enters the evaporator (10). The flow of R-134a vapor in the test-condenser was manipulated by controlling the flow of R-134a in the by-pass condenser (7). The test-section consisted of a microfin tube. The microfin tube was a copper tube having internal microfins with triangular fin cross-section. The geometrical parameters of microfin tube are shown in Fig. 2. The average outside wall temperatures of the inner tube was measured at four axial locations. At each location four thermocouples were fixed at top, two sides in the middle of tube and bottom positions (when microfin tube is in horizontal position). The refrigerant temperature at the inlet and outlet of the test-condenser was also measured. All the above temperature measurements were done by J-type (iron-constantan) thermocouples with a calibrated accuracy of 0.1 °C. The thermocouples were carefully soldered on the outer surface of microfin tube. For the measurement of cooling water temperatures the mercury in glass thermometers were used. The arrangements were also made for the measurement of refrigerant pressure at inlet and outlet of the test-condenser, pre-condenser and post-condenser. The refrigerant mass flow rate was measured by a rotameter (5) installed down stream of after-condenser. It was ensured that the complete liquid refrigerant enters the rotameter by providing an after-condenser (4). The whole of test-condenser, pre-condenser and after-condenser were insulated by glass wool to prevent any heat loss to the surroundings. A total of 84 test runs with three different refrigerant mass velocities of 54, 81 and 107 kg/m^2-s were performed for seven different tube inclinations from $\alpha = -90°$ to $\alpha = +90°$ (with intervals of 30°). The range of operating parameters is given in Table 1.

For each test run, the refrigerant side heat transfer coefficient, h, was calculated by using the Eq. (1) developed by [7].

$$h = \left[\frac{\pi DL(T_s - T_{wo})}{m_w c_{pw}(T_{co} - T_{ci})} - \frac{D}{2k}\ln\left(\frac{D_o}{D}\right)\right]^{-1} \quad (1)$$

The outside tube wall temperature, T_{wo}, was taken as the arithmetic mean of outside tube wall temperatures at top,

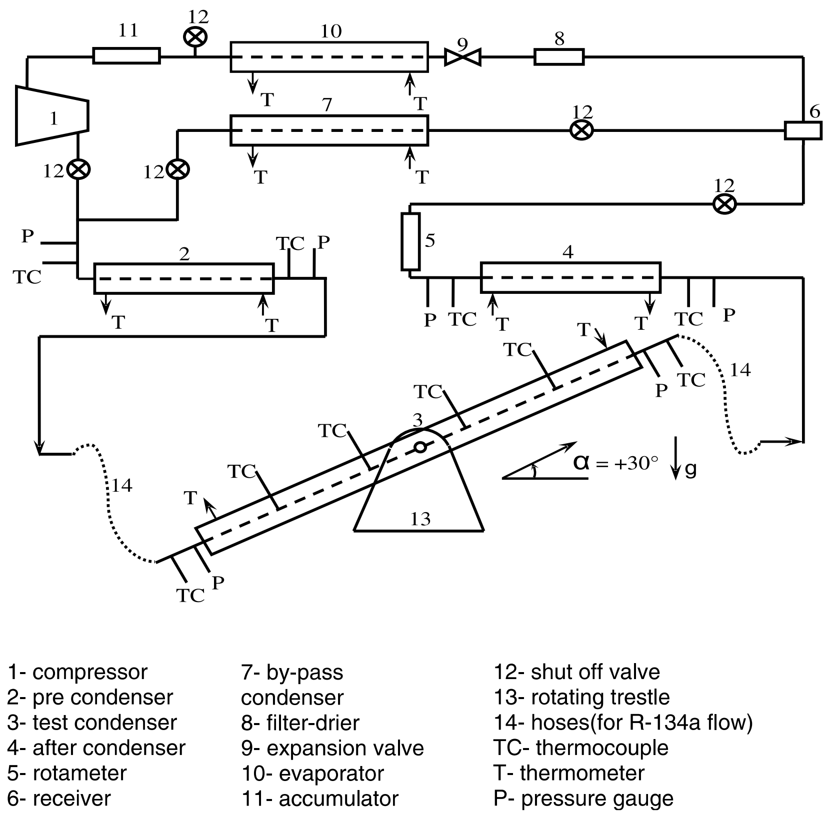

Fig. 1. Schematic diagram of experimental set-up.

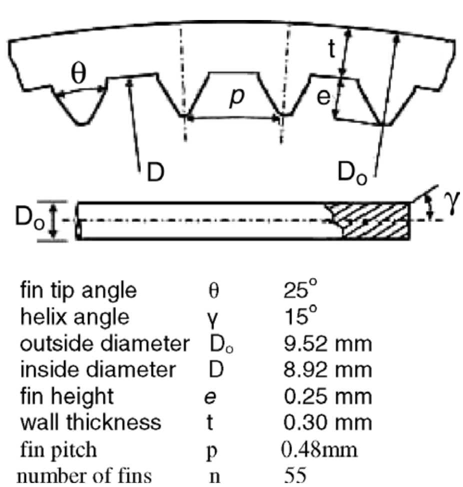

Fig. 2. Geometry of microfin tube.

Table 1
List of operating parameters

Working fluid	R-134a
Refrigerant mass velocity	54–107 kg/m^2-s
Average condensing temperature	26–32 °C
average cooling heat flux	8.7–20.3 kW/m^2
Coolant water mass flow rate	90–112 kg/h
Average vapor quality	0.2–0.8

side and bottom positions of a particular station. The local vapor quality at the inlet and outlet of each section was calculated by energy balance in that test-condenser (heat gained by cooling water is equal to the heat rejected by R-134a). The vapor quality of test-condenser was obtained by taking the average of vapor qualities at the inlet and outlet. The thermophysical properties of R-134a were taken from [8]. The uncertainty analysis of experimental results has been carried out by the method proposed in [9] and it was found that the expected experimental uncertainty was within a band of ±8.5% for all the test runs.

3. Results and discussion

First of all, the integrity of experimental set-up has been established by collecting data for the condensation of R-134a vapor inside a horizontal microfin tube. The experimental heat transfer coefficient has been compared with that predicted by four different correlations [10–13] for the condensation of vapor inside a horizontal microfin tube. In Fig. 3 such a comparison has been made taking experimental heat transfer coefficient as abscissa and predicted heat transfer coefficient as ordinate. It is observed in Fig. 3 that the experimental heat transfer coefficients, h, are in an error band of ±20% from those predicted by the four models [10–13]. This agreement of experimental

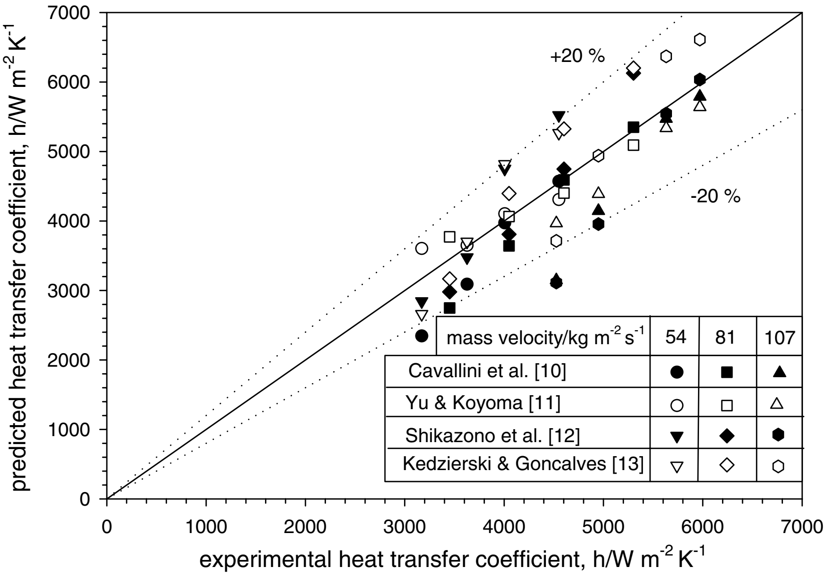

Fig. 3. Comparison of experimental heat transfer coefficient with that predicted by different models.

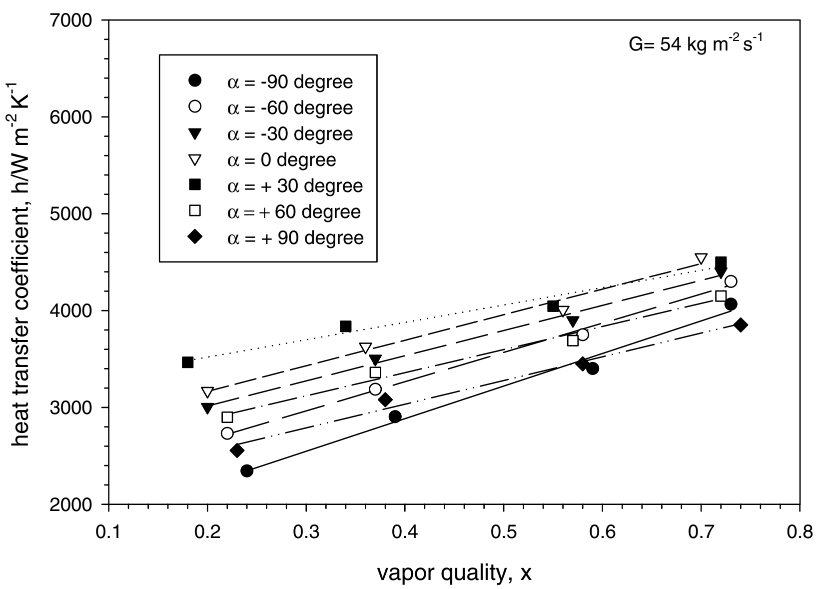

Fig. 4. Variation of heat transfer coefficient with vapor quality at vapor mass velocity of 54 kg m^{-2} s^{-1}.

heat transfer coefficient with the predicted values establishes the integrity of experimental set-up. Fig. 4 has been drawn to show the variation of condensation heat transfer coefficient, h, with vapor quality at seven different inclination angles ($\alpha = -90°$ to $\alpha = +90°$) of a microfin tube. The mass velocity of R-134a inside the tube has remained constant as 54 kg/m^2-s. It has been found that the heat transfer coefficient, h, for the condensation of R-134a vapor inside a horizontal finned tube reduces for all tube inclination angles as the condensation of vapor progresses. In fact, the vapor side heat transfer coefficient, h, has reduced for all inclinations in a range of 22–42% with an average of 32.5% at the exit of test-section as compared to that at the inlet of test-section. The reason for such a phenomenon is the fact that, with the progress of condensation along the tube in the direction of flow the average thickness of condensate film around the tube wall increases. The increased condensate thickness offers more thermal resistance to heat flow from vapor to the cooling water. Further, as the condensation progresses the vapor phase velocity decreases resulting in the lowering of the interfacial shear stresses. The cumulative effect of the above two factors contributes towards the decrease in the heat transfer coefficient, h. The random variation in heat transfer coefficient, h, is also visible in Fig. 4. This is due to the instability of heat transfer, in which, both the phases are flowing together during phase change. In addition, the presence of microfins in the passage and the different flow regimes inside the tube with different tube inclinations cause these random variations. The tube inclination angle has influenced the heat transfer coefficient, h, in a significant manner. The tube with +30° inclination angle turns out to be the best performing tube at low vapor quality, near exit of the test-section tube. The tube having inclination angle of −90° has the lowest heat transfer coefficient, h. The performance of the tube with +30° inclination is much superior to that of tube with −90° inclination in the low vapor quality region where the heat transfer coefficient, h, for +30° inclination tube is 48% more than that of the −90° inclined tube. However, the average heat transfer coefficient, $\bar{h}$, for the tube with +30° inclination is 25% more than that for the tube with −90° inclination for the entire range of vapor quality. In comparison to the performance of the horizontal tube the heat transfer coefficient, h, for the tube having inclination angle of +30° is 9.3% more at lower vapor quality, however, the average heat transfer coefficient, h, is merely 5.8% more than that for a plain tube, which lies within the uncertainty of 8.5% in experimental data. The variation of heat transfer coefficient with vapor quality for a microfin horizontal tube with different inclinations is shown in Figs. 5 and 6 for the mass velocity of 81 kg/m^2-s and 107 kg/m^2-s respectively. In these figures it is observed that the change in tube inclination has considerable effect on condensation heat transfer coefficient. The tube with the inclination angle of +30° is still the best performing tube at both the refrigerant flow rates. The enhancement in heat transfer coefficient, h, at lower vapor quality i.e. near exit of test-condenser is 40% and 32% at respectively at the refrigerant mass velocities of 81 kg/m^2-s and 107 kg/m^2-s. However, the average heat transfer coefficient, $\bar{h}$, is 20% and 14% more than that for the tube having −90° inclination. In fact, at high vapor phase velocity and vapor quality the interfacial shear stress is predominant and at low vapor quality the vapor phase velocity is reduced resulting in decrease of interfacial shear stresses and inertia force as well. Therefore, at low vapor quality the gravitational force has considerable effect on condensate flow rates and this is the major reason for different rates of heat transfer coefficient for different inclinations of finned tube. From Figs. 4–6, it is also noted that the vertical tube with inclination angle of −90° has the lowest heat transfer coefficient. In fact, during downward vertical flow, the interfacial shear stress and gravitational force are unidirectional and the only probable flow pattern is annular flow. Wang et al. [14] also reported the same observations in their study of different flow patterns in inclined plain tubes. The phenomenon of annular flow causes to form a thick layer of condensate around the periphery of tube as a result the heat transfer coefficient, h, is reduced. Since, the gravitational force and vapor shear stress are in same direction, the interfacial turbulence is the lowest. For other inclination angles, especially tubes with inclination close to horizontal, the condensate liquid tends to flow in lower side of the tube and as a result the heat transfer in the unflooded part of the tube takes place. During upward vertical flow ($\alpha = +90°$) of vapor, due to high interfacial turbulence, higher heat transfer coefficient is observed in comparison to that for downward vertical flow in low vapor qualities. The same observation has also been made by Wang et al. [14] for plain tubes. However, at high vapor quality, the case is vice-versa and i.e. upward vertical condensation flow has the lower heat transfer coefficient than that for the tube with inclination angle, α, of −90°. With regard to high probability of annular flow existence for both upward and downward vertical flow cases in high vapor qualities, the reason for the superior performance of tube having inclination angle, α, as −90° can be related to better condensate liquid film flow inside grooves and tube surface

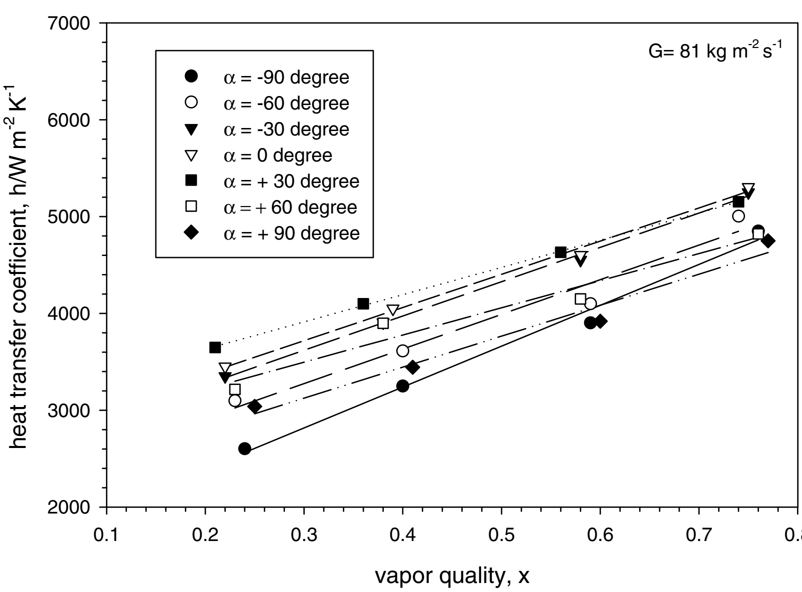

Fig. 5. Variation of heat transfer coefficient with vapor quality at vapor mass velocity of 81 kg m^{-2} s^{-1}.

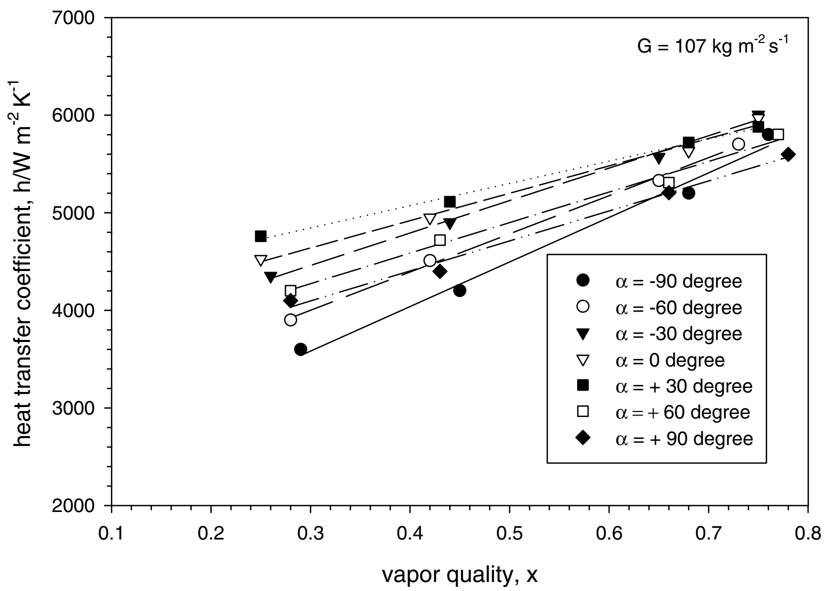

Fig. 6. Variation of heat transfer coefficient with vapor quality at vapor mass velocity of 107 kg m^{-2} s^{-1}.

in downward flow in comparison with upward flow. Finally it is also revealed from Figs. 4–6 that the highest heat transfer coefficient for all vapor qualities occurs when the microfin tube is horizontal at high vapor quality or with an inclination of +30° at low vapor quality. In fact, the flow pattern existing for these inclinations is the most befitting two-phase flow pattern to obtain highest condensation heat transfer at low vapor quality with inclination of +30° and at high vapor qualities in horizontal tube. Wang et al. [14] noted that the flow regimes in horizontal plain tube and plain tube with inclination of +30° are too close to each other. They found that the only difference between these two cases is the increase of interfacial turbulences in tube with inclination of +30° in comparison with the horizontal tube. At high vapor quality the increase in condensation heat transfer coefficient in tube with inclination of $\alpha = -30°$ in comparison with the tube with inclination of $\alpha = +30°$ for mass velocities of 81 kg/m^2-s and 107 kg/m^2-s is due to better flow of condensate inside grooves and over tube inside wall because of favorable direction of gravitational force. But with decrease in vapor quality and as a result of increase in condensate, the interfacial turbulences would become the effective parameter and cause increasing of heat transfer in tube with $\alpha = +30°$ in comparison with horizontal tube. Royal and Bergles [15] have also indicated that positive inclination of tube causes increasing of interfacial turbulences in comparison with horizontal tube and could be considered as a suitable

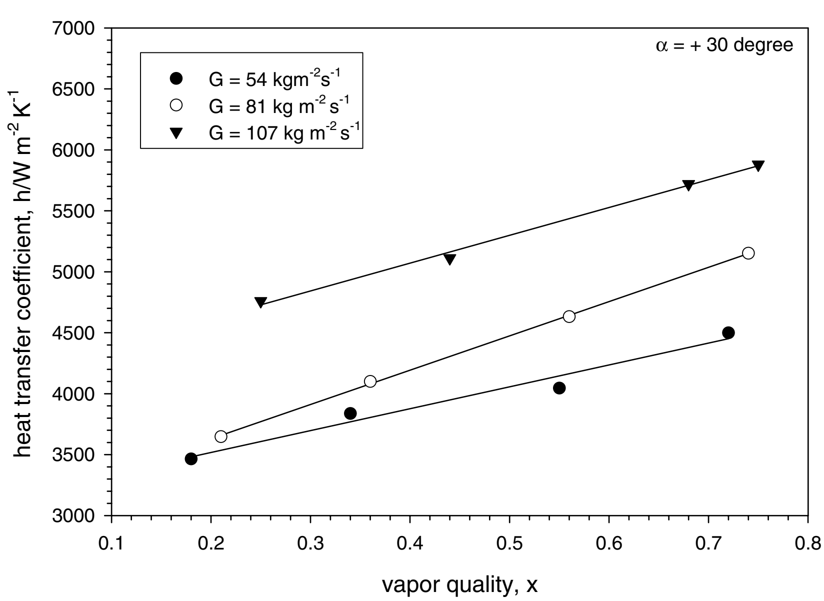

Fig. 7. Effect of vapor mass velocity on heat transfer coefficient at inclination angle of +30°.

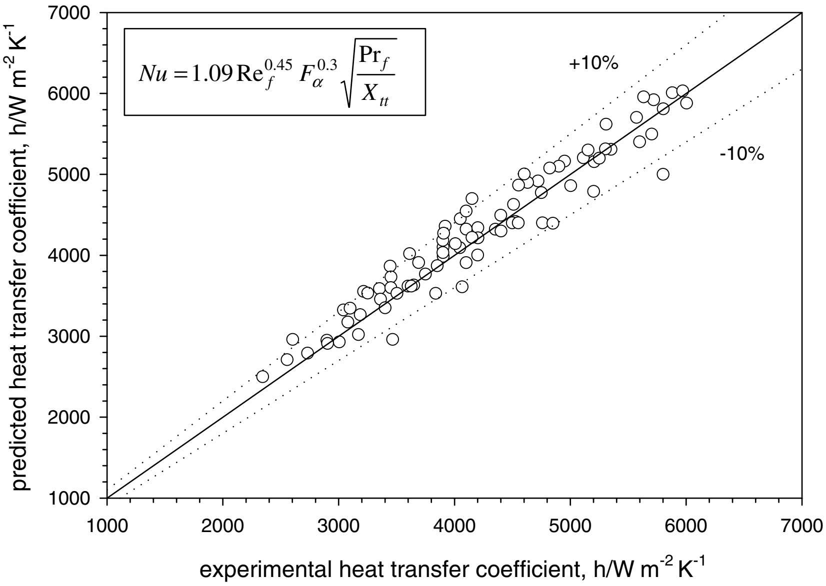

Fig. 8. Comparison of experimental heat transfer coefficient with that predicted by proposed correlation.

way for heat transfer enhancement in plain tubes. Here, it should be noted that the vapor quality of the cross point of the curves relating to inclinations of ($\alpha = +30°$ and $\alpha = 0°$), ($\alpha = +30°$ and $\alpha = -30°$), ($\alpha = -60°$ and $\alpha = +60°$) and ($\alpha = +90°$ and $\alpha = -90°$) is mostly in a range of 0.2 to 0.8 vapor quality for all mass velocities. In fact, at high vapor qualities the flow of condensate liquid inside grooves and the vapor flow over fin surface is the most effective arrangement for heat transfer, however, at low vapor qualities, due to increasing the thickness of condensate film, parameters such as interfacial turbulences affects the rate of heat transfer.

Fig. 7 has been drawn to show the variation of heat transfer coefficient, h, with vapor quality at different vapor mass velocities for the condensation of R-134a inside a tube of $+30°$ inclination. The highest heat transfer coefficient, h, is for the mass velocity of 107 kg/m^2-s. The average heat transfer coefficient, h, at the mass velocity of 107 kg/m^2-s is nearly 36% more than that for the mass velocity of 54 kg/m^2-s. The following correlation, Eq. (2), has been developed to predict the heat transfer coefficient at different vapor qualities, mass velocities and tube inclinations.

$$Nu = 1.09 Re_{\mathrm{f}}^{0.45} F_{\alpha}^{0.3} \sqrt{\frac{Pr_{\mathrm{f}}}{X_{\mathrm{tt}}}} \tag{2}$$

where,

$$Nu = \frac{\bar{h}D}{k_{\mathrm{f}}}; \quad Re_{\mathrm{f}} = \frac{\mathrm{GD}(1-x)}{\mu_{\mathrm{f}}}; \quad Pr_{\mathrm{f}} = \frac{\mu_{\mathrm{f}} c_{\mathrm{pf}}}{k_{\mathrm{f}}}$$
$$X_{\mathrm{tt}} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{f}}}\right)^{0.5}\left(\frac{\mu_{\mathrm{f}}}{\mu_{\mathrm{v}}}\right)^{0.1}, \text{ Jung et al. [16]} \tag{3}$$

$$F_{\alpha} = (1 + (1-x)^{0.2}\mathrm{Cos}(\alpha - 10°))/x^{0.4} \tag{4}$$

The above correlation given in Eq. (2) predicts the experimental data for all angles of inclination of finned tune in an error band of $\pm 10\%$ as shown in Fig. 8.

4. Conclusions

The following conclusions have been drawn from the present investigation:

1. The heat transfer coefficient depends on the tube inclination angle and it decreases with the decrease in vapor quality and mass velocity.
2. The highest heat transfer coefficient occurs when the microfin tube is horizontal at high vapor qualities and is inclined at $+30°$ at low vapor qualities. The lowest heat transfer coefficient has been attained for the vertical tube. Further, for high vapor quality, upward flow has the lowest heat transfer coefficient and for low vapor quality, heat transfer coefficient is the lowest for downward flow. The microfin tube having inclination angle of $+30°$ outperforms the tube with $-90°$ inclination in a range of 32–48%.
3. The inclination angle of $+30°$ provides marginal rise of 5.8% in heat transfer coefficient, h, in comparison to that for a plain tube. Other inclination angles adversely affect the heat transfer coefficient, h.
4. The following correlation has been developed to predict the heat transfer coefficient

$$Nu = 1.09 Re_{\mathrm{f}}^{0.45} F_{\alpha}^{0.3} \sqrt{\frac{Pr_{\mathrm{f}}}{X_{\mathrm{tt}}}}.$$

References

[1] K.N. Agrawal, R. Kumar, S.N. Lal, H.K. Varma, Heat transfer augmentation by segmented tape inserts during condensation of R-22 inside a horizontal tube, ASHRAE Transact. 110 (2004) 143–150.
[2] R. Kumar, K.N. Agrawal, S.N. Lal, H.K. Varma, An experimental study on condensation enhancement of R-22 by the turbulence promoter, ASHRAE Transact. 111 (2005) 18–25.

[3] M.A.A. Behabadi, M.R. Salimpore, R. Kumar, K.N. Agrawal, Augmentation of forced convection condensation heat transfer inside a horizontal tube using spiral spring inserts, J. Enhanc. Heat Transfer 12 (4) (2005) 373–384.
[4] L. Chamra, R. Webb, M. Randlett, Advanced microfin tubes for condensation, Int. J. Heat Mass Transfer 39 (9) (1996) 1839–1846.
[5] L.M. Schlager, M.B. Pate, A.E. Bergles, Heat transfer and pressure drop performance of smooth and internally finned tubes with oil and refrigerant 22 mixtures, ASHRAE Transact. 95 (2) (1989) 375–385.
[6] D. Jung, Y. Cho, K. Park, Flow condensation heat transfer coefficients of R22, R134a, R407C, and R410A inside plain and microfin tubes, Int. J. Refrig. 27 (2004) 25–32.
[7] S.A. Said, N.Z. Azer, Heat transfer and pressure drop during condensation inside horizontal finned tubes, ASHRAE Transact. 89 (1) (1983) 114–134.
[8] American Society of Heating Refrigerating and Air-conditioning Engineers Handbook, Fundamentals, 2005, p. 20.17.
[9] R.R. Schultz, R. Cole, Uncertainty analysis in boiling nucleation, Am. Inst. Chem. Eng. Sympos. Ser. 75 (189) (1979) 32–38.
[10] A. Cavallini, L. Doretti, N. Klammsteiner, LG. Longo, L. Rossetto, Condensation of new refrigerants inside smooth and enhanced tubes, in: Proceedings 19th International Congress of Refrigeration, Hague The Netherlands, 1995, pp. 105–114.
[11] J. Yu, S. Koyama, Condensation heat transfer of pure refrigerants in microfin tubes, in: Proceedings of International Refrigeration Conference, Purdue, West Lafayette, USA, 1998, pp. 325–330.
[12] N. Shikazono, M. Itoh, M. Uchida, T. Fukushima, T. Hatada, Predictive equation proposal for condensation heat transfer coefficient of pure refrigerants in horizontal microfin tubes, Transact. Jpn. Soc. Mech. Eng. 64 (1998) 196–203.
[13] M.A. Kedzierski, J.M. Goncalves, Horizontal convective condensation of alternative refrigerants within a microfin tube, J. Enhanc. Heat Transfer 6 (1999) 161–178.
[14] W.C. Wang, X.H. Ma, Z.D. Wei, P. Yu, Two-phase flow patterns and transition characteristics for in-tube condensation with different surface inclinations, Int. J. Heat Mass Transfer 41 (1998) 4341–4349.
[15] J.H. Royal, A.E. Bergles, Experimental study of the augmentation of horizontal in-tube condensation, ASHRAE Transact. 82 (1) (1976) 919–931.
[16] D. Jung, K. Song, Y. Cho, S. Kim, Flow condensation heat transfer coefficients of pure refrigerants, Int. J. Refrig. 26 (2003) 4–11.