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Investigation on heat transfer and pressure drop during swirl flow boiling of R-134a in a horizontal tube

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ABSTRACT

An experimental investigation has been carried out to study the effect of twisted tape inserts on heat transfer enhancement and pressure drop in a horizontal tube during swirl flow boiling of R-134a. The test-evaporator was an electrically heated horizontal copper tube and twisted tapes with different twist ratios of 6, 9, 12 and 15 were inserted one by one. The data were acquired at the refrigerant mass velocities of 54, 86, 114 and 136 kg/s m^2. The twisted tape inserts increases the boiling heat transfer coefficients and the pressure drop across the test-evaporator. An empirical correlation has also been developed to predict the swirl flow pressure drop in the test-evaporator.

1. Introduction

The limited availability of conventional energy resources and the ever increasing cost of energy over the last several years have accelerated research on energy conservation with the focus on the possible reduction in energy consumption. In the area of heat transfer, extensive efforts have been made to develop compact and efficient heat exchangers. There is wide application of heat exchangers in the form of evaporators and condensers in Refrigeration and Air-conditioning industries.

The flow boiling of refrigerants inside horizontal tubes has been studied for quite some time [9,16,17]. The use of twisted tape inserts to enhance the heat transfer rate during evaporation of refrigerants while flowing inside a horizontal tube has also been studied [2,7]. It is noted that, the use of twisted tape enhances the heat transfer coefficient only at the pressure drop penalty. Though, the main objective of all the research work in this field remains to attain the highest increase in heat transfer coefficient with the least increase in pressure drop.

The tube inserts such as twisted tape are relatively low in cost, easy to insert and remove for maintenance [1]. Further, recently the application of R-134a has increased as it is ozone-safe and has less impact on global warming. Therefore, in the present investigation, the heat transfer and pressure drop have been determined during flow boiling of R-134a inside a horizontal copper tube with twisted tape inserts.

2. Experimental set-up

An experimental set-up was designed, fabricated and installed in the laboratory to study the flow boiling of R-134a inside a horizontal tube. The schematic diagram of the experimental set-up has been shown in Fig. 1. In fact, the test apparatus was a well instrumented vapor compression refrigeration system.

The test set-up consisted of a test-evaporator (2) having a test-section length of 1200 mm. The test-section was a horizontal copper tube of 7.50 mm inside diameter and 9.54 mm outside diameter. The heating arrangement of this tube was made by wrapping a 2 kW capacity electrical heating tape around the tube surface. In order to obtain desired vapor quality at the inlet of test-evaporator, a preheater (1) was installed upstream of the test-evaporator. This arrangement facilitated to study the entire range of vapor quality in the test-evaporator. An after-evaporator (3) has also been installed at the downstream of the test-evaporator to ensure that the dry vapor enters the compressor. All the three evaporators were thermally insulated. An accumulator (4) was also provided before the suction of compressor (5). The vapor emerging from the compressor (5) was condensed in a condenser (6) and the condensate was subsequently sent to the receiver (8). A site glass (7) was also provided to view the flow of refrigerant in the pipeline and the flow rate was measured by a flow meter (9).

The outside tube wall temperature of test-section was measured at six axial locations. At each location the temperature of tube wall was measured at top, side and bottom positions. The refrigerant temperature at the inlet and outlet of the test-evaporator was also measured. All the temperature measurements were

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Nomenclature

d	diameter, m
H	180° twist pitch, m
Fr	Froude number, $\frac{G^2}{\rho^2 g d_i}$
g	Gravitational acceleration, m/s^2
h	heat transfer coefficient, W/m^2 K
$\bar{h}$	average heat transfer coefficient, W/m^2 K
hf	Enthalpy saturated liquid, kJ/kg
hi	Enthalpy at inlet, kJ/kg
L	length of test-evaporator, m
G	mass velocity of refrigerant, kg/s m^2
Q	Electrical work on evaporator, W
Re	Reynolds number, $\frac{G \cdot d_i}{\mu}$
t	temperature, K
k	thermal conductivity, W/m K
y	twist ratio of the tape, H/d_i
x	vapor quality
We	Weber number, $\frac{G^2 d_i}{\rho\sigma}$
ΔP	pressure drop, kPa
$\bar{\Delta}P$	pressure drop, kPa
φ	two-phase multiplier

Greek symbols

α	void fraction
λ	latent heat of vaporization, kJ/kg
σ	surface tension, Pa m
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3

Subscripts

b	bottom
fri	friction
fr	Friedel
h	homogeneous
in	inlet
i	inside
f	liquid
mom	momentum
out	outlet
o	outside
wo	outside wall
p	plain flow boiling
s	saturated, swirl flow boiling, side
sta	static
t	top
tot	total
g	vapor
w	wall

made by using T-type thermocouples. For the measurement of pressure drop across the test-evaporator a pressure drop measuring apparatus was employed.

In order to collect the experimental data, first of all, the cooling water supply in the main condenser (6) was started. Later, the heating of test-evaporator and the compressor were switched on. The refrigerant mass flow rate was held constant and the desired vapor quality at the test-section inlet was attained with the help of preheater (1). The vapor quality in evaporator was determined by the heat balance in the test-evaporator (2). For a particular mass flow rate of refrigerant the system was reached in steady state in nearly 40 min. The electrical heating of evaporator was controlled by a variac. Initially, the data were acquired for the plain flow boiling of R-134a, i.e., without any insert. The data for plain flow were acquired to establish the integrity of the experimental apparatus and to have a reference data for comparing the performance of different twisted tape inserts. The data for test-section tubes with different twisted tape inserts were acquired at refrigerant mass velocities of 54, 85, 114 and 136 kg/s m^2. Different types of full length twisted tapes were inserted in the tube, one by one, having different twist ratios of 6, 9, 12 and 15. Table 1 represents the specifications of the twisted tape inserted tubes and Fig. 2 illustrates the characteristic parameters of a typical twisted tape inserts. All together 160 test-runs were performed and the range of operating parameters is given in Table 1. The thermophysical properties of R-134a were taken from Refprop 7.0. The local vapor quality in the test-evaporator was attained through energy balance in the test-evaporator. Using the following Eq. (1):

$$x = \frac{hi + 4\frac{Q}{\pi G d_i^2} - hf}{\lambda} \quad (1)$$

The thermocouples and flowmeters were calibrated prior to installation. The temperature was measured with an accuracy of 0.1 °C. The uncertainly analysis of experimental data was done by the method proposed by Schultz and Cole [14]. The uncertainty in the determination of flow boiling heat transfer coefficient was found to be within 10 percent.

3. Results and discussion

For each test run, the heat transfer coefficient was calculated by using the following Eq. (1).

$$\bar{h} = \left[\frac{\pi d_i L(t_{wo} - t_s)}{Q} - \frac{d_i}{2k_w}\ell n\left(\frac{d_o}{d_i}\right)\right]^{-1} \quad (2)$$

where, $t_{wo} = \frac{t_t + 2t_s + t_b}{4}$

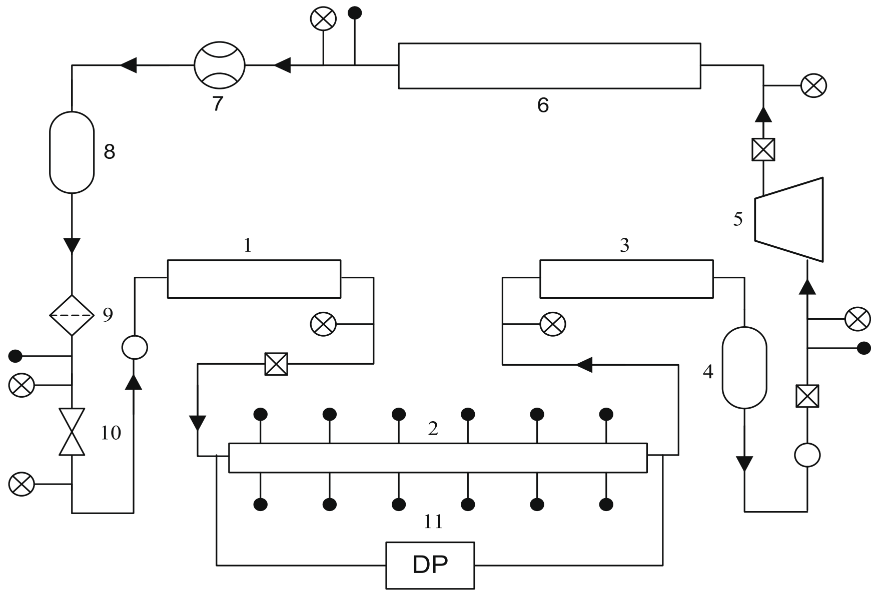
The heat transfer coefficient was calculated on the basis of heat gained by R-134a in test-evaporator and the temperature difference between inside wall of test-evaporator and the boiling refrigerant.

3.1. Effect of mass velocity on heat transfer coefficient

The variation of heat transfer coefficient with vapor quality for plain flow and that for the tubes with twisted tube inserts has been shown in Figs. 3–6 for the refrigerant mass velocity of 54, 85, 114 and 136 kg/s m^2, respectively.

In Figs. 3–6, it has been observed that the heat transfer coefficient increases with the rise of vapor quality down the length of test-evaporator. Because at high vapor quality, the condensate liquid film on the inner tube wall is thinner, offering lower thermal resistance. By the comparison of Figs. 3–6, it has been found that, in general, the heat transfer coefficient increases with the increase of mass velocity. This is in agreement with the fact that at high mass velocities of refrigerant, the Reynolds number is also high, which in turn, increases the refrigerant side heat transfer coefficient.

Figs. 3–6 also show that the insertion of a twisted tape inside the test-evaporator tube has produced higher heat transfer coefficient compared to plain flow. The highest heat transfer coefficient has been attained for the insert with lowest twist ratio of 6. In fact, as the twist ratio reduces, the forces due to centrifugal acceleration near the wall increases and as a result the heat transfer coefficient also increases. In the high vapor quality region the insert enhances the heat transfer coefficient by as much as 57 percent above the plain flow. At vapor quality, $x = 0.2$ the enhancement in heat transfer coefficient is in a range of 35 to 45 percent with the insert of



1- pre evaporator	7- sight glass
2-test evaporator	8-receiver
3-after evaporator	9- flow meter
4-accumulator	10-expansion valve
5-compressor	11-differential pressure transducer
6-condenser	
thermocouple -	shut off valve -
flow - direction	pressure gauge

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

Table 1
Range of operating parameters.

Inlet vapor quality	0.2–0.9
Exit vapor quality	0.3–1.0
Twist ratio	6–15
Refrigerant mass velocity	54–136 kg/m^2 s
Boiling temperature	−3 to −19 °C
Liquid Reynolds number	1250–3500
Vapor Reynolds number	41,000–96,000
Heat flux	1.8–5.3 kW/m^2

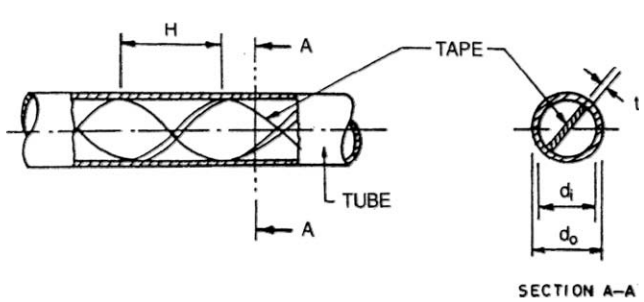


Fig. 2. Characteristic parameters of a typical twisted tape.

twist ration 6. For the same insert at the vapor quality, $x = 0.8$ the enhancement in heat transfer coefficient is in a range of 47 to 57 percent.

Akhavan-Behabadi et al. [12] investigated the effect of twisted tape inserts during forced convective condensation of R-134a inside a 10.7 mm inside diameter horizontal copper tube. They reported a maximum of 43 percent heat transfer enhancement through twisted tape having twist ratio, $y = 6$. Since, the present investigation has been carried out for the evaporation of R-134a, the highest enhancement in heat transfer coefficient is 57 percent. It can be concluded that the twisted tape perform better in evaporators in comparison to condensers.

3.2. Pressure drop in plain flow

In fact, during flow boiling, the total pressure drop in the test-evaporator is the sum of friction pressure drop, the momentum pressure drop, and the static pressure drop:

$$\Delta P_{tot} = \Delta P_{fri} + \Delta P_{mom} + \Delta P_{sta} \tag{3}$$

For horizontal tube, $\Delta P_{sta} = 0$, therefore,

$$\Delta P_{tot} = \Delta P_{fri} + \Delta P_{mom} \tag{4}$$

The momentum pressure drop is calculated by:

$$\Delta P_{mom} = G^2 \left\{ \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{out} - \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{in} \right\} \tag{5}$$

In which, 'α' is the void fraction and is computed by Eq. (6) by Steiner [4].

$$\alpha = \frac{x}{\rho_g} \left[(1 + 0.12(1-x)) \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_f} \right) + \frac{1.18(1-x)[g\sigma(\rho_f - \rho_g)]^{0.25}}{G^2 \rho_f^{0.5}} \right]^{-1} \tag{6}$$

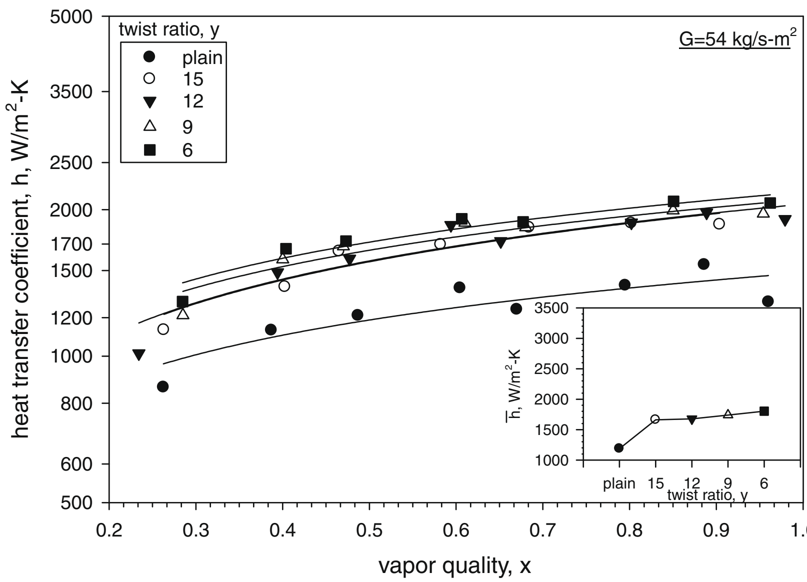


Fig. 3. h *v/s* x at 54 kg/s m^2.

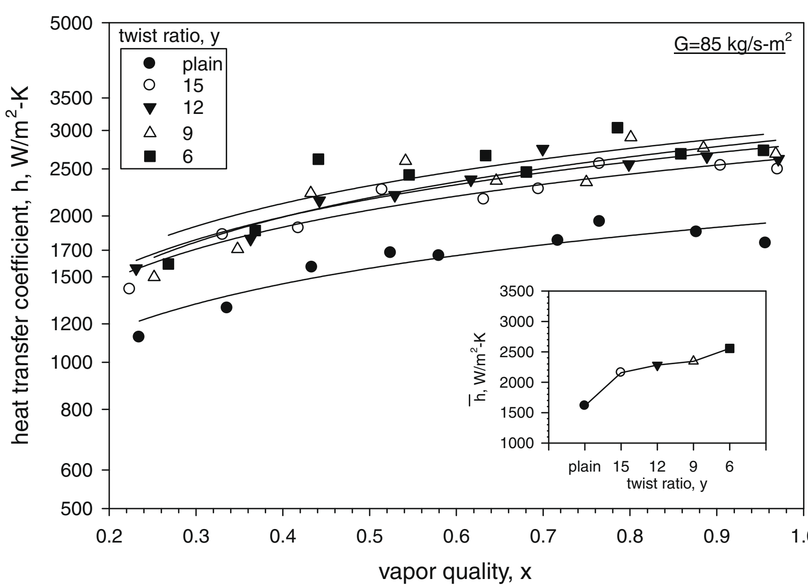


Fig. 4. h *v/s* x at 85 kg/s m^2.

The total pressure drop in the test-evaporator, Δp_{tot}, was measured by a pressure drop measuring apparatus. The momentum pressure drop is calculated by Eq. (4) and the total pressure drop has been computed by Eq. (3).

The following Friedel correlation [11] has been used to calculate friction pressure drop.

$$\Delta P_{\text{fri}} = \Delta P_f \varphi_{fr}^2 \tag{7}$$

where,

$$\Delta P_f = \frac{2 f_f G^2 L \upsilon_f}{d_i} \tag{8}$$

The liquid friction factor f_f and the liquid Reynolds number, Re_f, have been obtained from the following Eq. (9).

$$f_f = 0.079 Re_f^{-0.25} \tag{9}$$

$$Re_f = \frac{G d_i}{\mu_f} \tag{10}$$

The Friedel two-phase multiplier has been determined as follows:

$$\varphi_{fr}^2 = E + \frac{3.24 FH}{Fr_h^{0.045} We_f^{0.035}} \tag{11}$$

The dimension less factors E, F, H and Fr_h are as follows:

$$E = (1-x)^2 + x^2 \left(\frac{\rho_f f_g}{\rho_g f_f}\right) \tag{12}$$

$$F = x^{0.78}(1-x)^{0.224} \tag{13}$$

$$H = \left(\frac{\rho_f}{\rho_g}\right)^{0.91} \left(\frac{\mu_g}{\mu_f}\right)^{0.19} \left(1 - \frac{\mu_g}{\mu_f}\right)^{0.7} \tag{14}$$

$$Fr_h = \frac{G^2}{g d_i \rho_h^2} \tag{15}$$

The liquid Weber number, We_f, is defined as

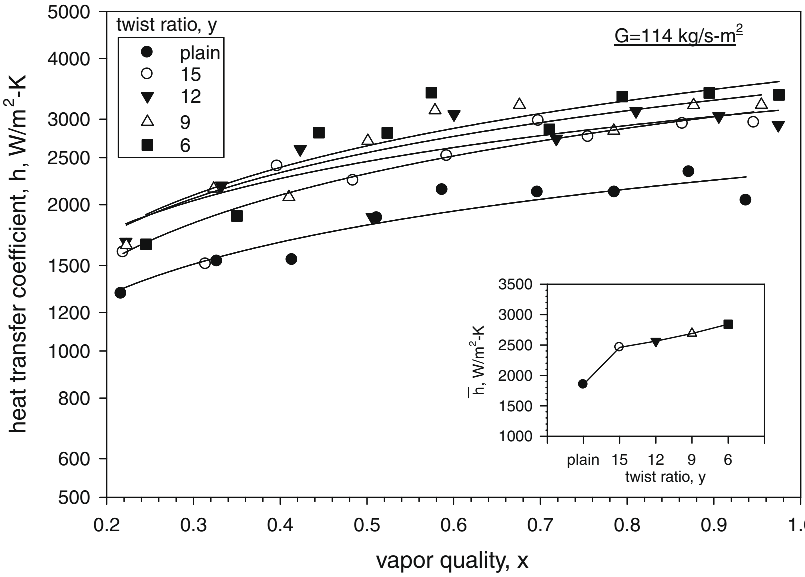


Fig. 5. *h v/s x* at 114 kg/s m^2.

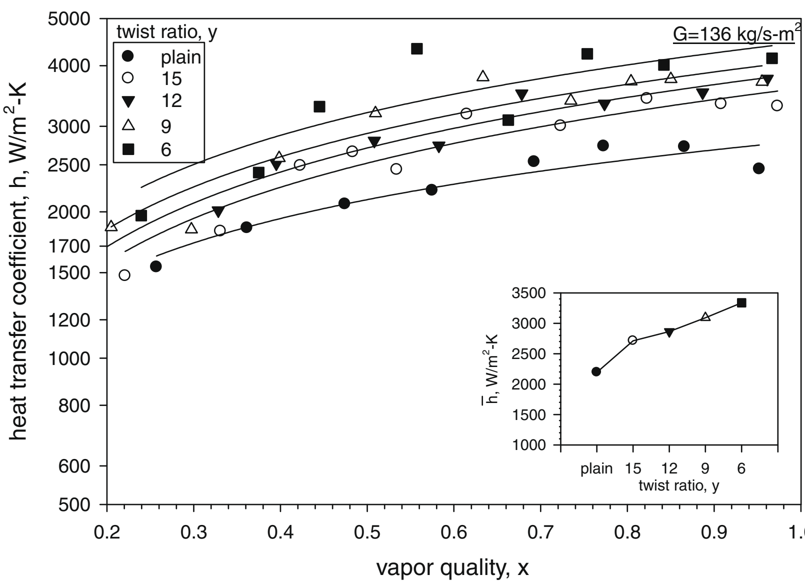


Fig. 6. *h v/s x* at 136 kg/s m^2.

$$We_f = \frac{G^2 d_i}{\rho_h \sigma} \tag{16}$$

In which, the homogeneous density, ρ_h, is obtained from:

$$\rho_h = \left[\frac{x}{\rho_g} + \frac{(1-x)}{\rho_f}\right]^{-1} \tag{17}$$

In Fig. 7, the friction pressure drops obtained from experimental data for the mass flow rate of 114 kg/s m^2 are compared with the values predicted by different friction pressure drop methods [3,6,11,13,15]. From this figure, it is revealed that the Friedel correlation [11], and the Müller-Steinhagen correlation [6] have the best agreement with the present experimental data. In addition, the pressure drop has also been determined using the correlations of Bankoff [18] and Chawla [8]. These correlations have been developed for completely different operating conditions, therefore, the estimations by these correlations have a large deviation with the experimental data and, hence, the results of these correlations have not been shown in Fig. 7. The deviation of the present data with the predicted values from Chisholm [3] and Lockhart and Martinelli correlations [15] was expected, because both of them are applicable for $\mu_f/\mu_g < 1000$ which is not in consistence with the properties of the present working fluid. Regarding the inconformity between the experiment results and Muller [6], it is worth to be noted that the correlation proposed by [5] is for air–oil, air–water, water–steam, and several refrigerants, therefore its accuracy domain is comparatively low. The friction pressure drop obtained from experimental data and that predicted by Friedel correlation [11] has been shown in Fig. 8. It is observed that the Friedel correlation underpredicts the present experimental data within an error band of +10 to −30 percent. The Friedel estimation is recommended for the fluids with $\mu_f/\mu_g < 1000$ and for the operating parameters ranges of $0 \leqslant x \leqslant 1$ and $G < 2000$ kg/s m^2.

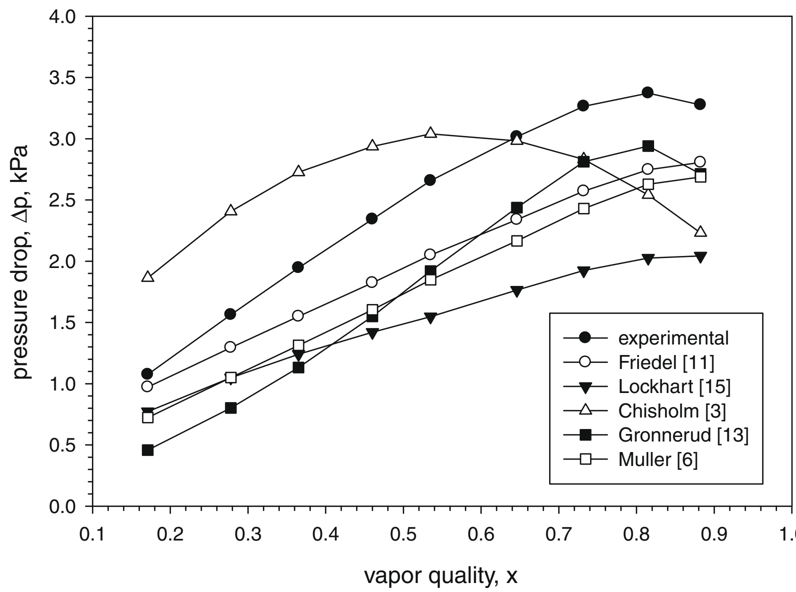


Fig. 7. Experimental Δp *v/s* predicted Δp at 114 kg/s m^2.

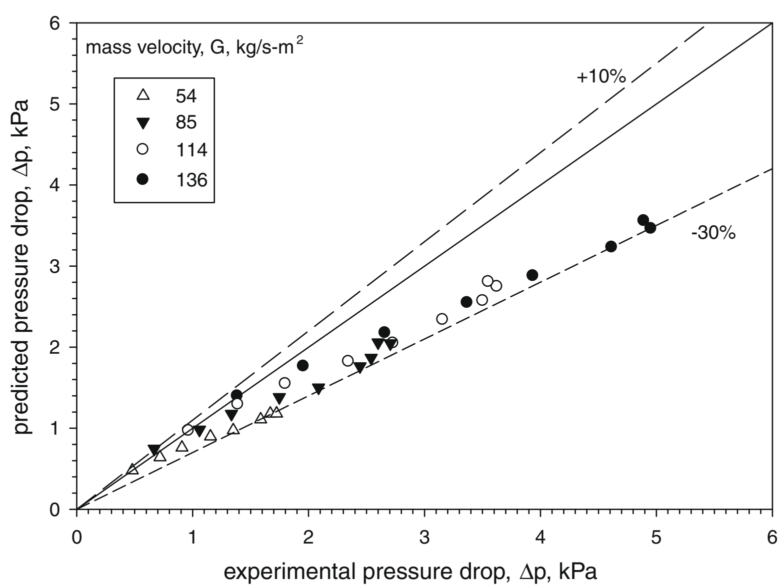


Fig. 8. Experimental Δp *v/s* Δp by Friedel [11] for plain flow.

3.3. Pressure drop with twisted tape inserts

The variation of pressure drop with vapor quality for the plain flow and the tubes with twisted tape inserts at different mass velocities has been shown in Figs. 9–12. Also the comparison between the average pressure drop of plain flow and tubes with twisted tape inserts with twist ratio has been done in the same figures. It has been found that, in general, by insertion of twisted tape inside horizontal tubes, the pressure drop is increased in comparison to that plain flow. It is also observed that, by lowering twist ratio, the pressure drop has been increased only at low vapor quality at the mass velocity of 54 kg/s m^2. As the mass velocity has increased the pressure drop is also increased, however, the effect of twist ratio, *y*, on the pressure drop has reduced. In fact, for the high vapor quality the effect of twist ratio, *y*, is not significant.

The increase in pressure drop across the test-section is due to the fact that with the reduction of twist ratio, *y*, more turbulence and swirl is induced in liquid film and vapor core. One of the reasons to use the twisted tape for the enhancement of flow boiling heat transfer coefficient is that the presence of twisted tape in the tubes causes the postponement of partial dryness and as a result the heat transfer is augmented. This is followed by increase in pressure drop due to twisted tapes as it is apparent from Figs. 9–12. At low mass velocities, the gravity force is greater than the centrifugal force and above mentioned effect does not take place. However, at low mass velocity and high vapor quality the partial dryness is postponed in the presence of twisted tape and as a result the heat transfer coefficient and pressure drop increases. The boiling pressure drop increases by as much as 180 percent by twisted tapes in the low vapor quality region at mass velocity of 54 kg/s m^2 with the tape of twist ratio of 6. This may be due to the fact that,

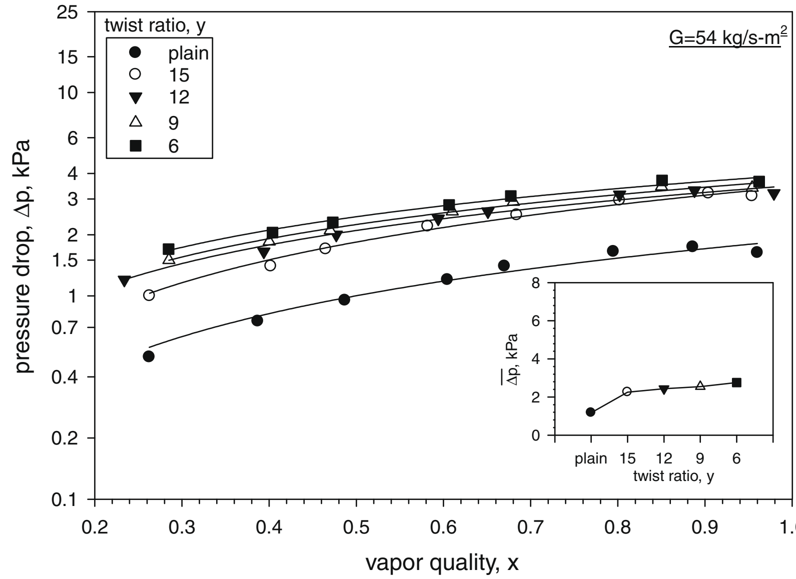


Fig. 9. Δp *v*/*s* x at 54 kg/s m^2.

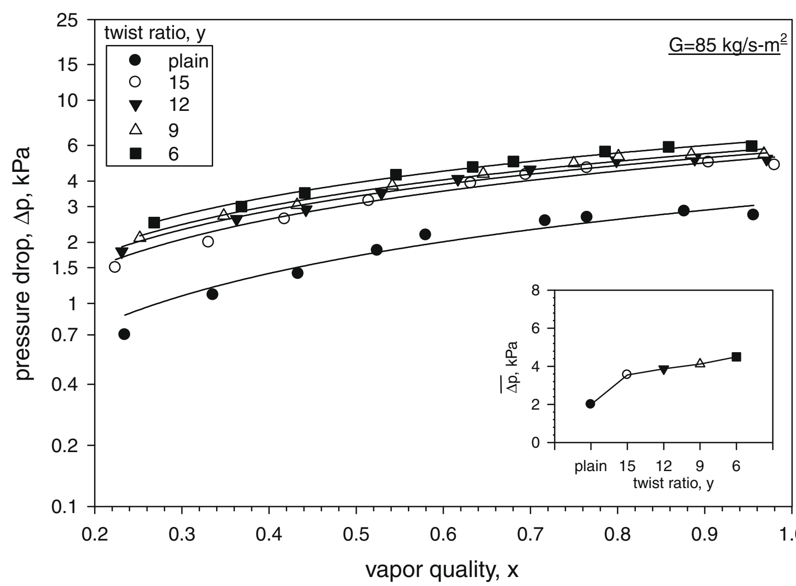


Fig. 10. Δp *v*/*s* x at 85 kg/s m^2.

with the presence of twisted tape, the flow regime changed to annular and as a result, the pressure drop has increased relative to the plain flow in which the flow regime is stratified-wavy.

3.4. Development of pressure drop correlation for the flow with twisted tape inserts

The variation of the total pressure drop in the test-section tube with twisted tape insert to the plain flow pressure drop, by Friedel correlation ratio [11], with the twist ratio has been shown in Fig. 13. In fact, in the plain flow the total pressure drop is the sum of the friction pressure drop computed by the Friedel correlation [11] and the momentum pressure drop calculated by Eq. (5). The variation of the swirl flow pressure drop to plain flow experimental pressure drop ratio with twist ratio has also been shown in Fig. 13. This pressure drop ratio varies between 1.9 and 2.8 for twist ratio of 6 and between 1.4 and 2.3 for twist ratio of 15. The lowest pressure drop increase of the order of 40 percent has been observed in the high vapor quality region at mass velocity of 136 kg/s m^2 with the tape of twist ratio of 15. This is because of the fact that at high refrigerant mass velocities, the flow pattern in plain flow is annular and insertion of twisted tape with twist ratio of 15 does not alter the flow pattern so the pressure drop does not increase in a significant manner. The pressure drop ratio also differs for different twist ratios. Agrawal and Varma [10] and Blatt and Adt [14] presented a correlation for the estimation of swirl flow boiling pressure drop of R-11 and water, and R-12, respectively, in the following form:

$$\left(\frac{\Delta P_s}{\Delta P_P}\right) = \frac{C}{Y^n} \tag{18}$$

In which, ΔP_p is the total pressure drop of plain flow and ΔP_s is the total pressure drop of the tube with twisted tape insert. The values of constants 'C' and 'n' are as follows. Agrawal correlation [10]: $C = 5.12$, $n = 0.509$ Blatt correlation [19]: $C = 7.36$, $n = 0.6$

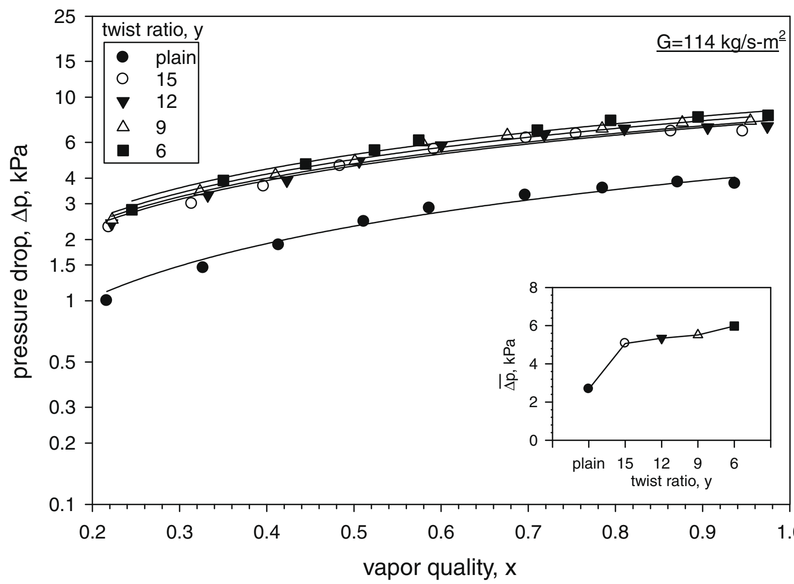


Fig. 11. Δp *v/s* x at 114 kg/s m^2.

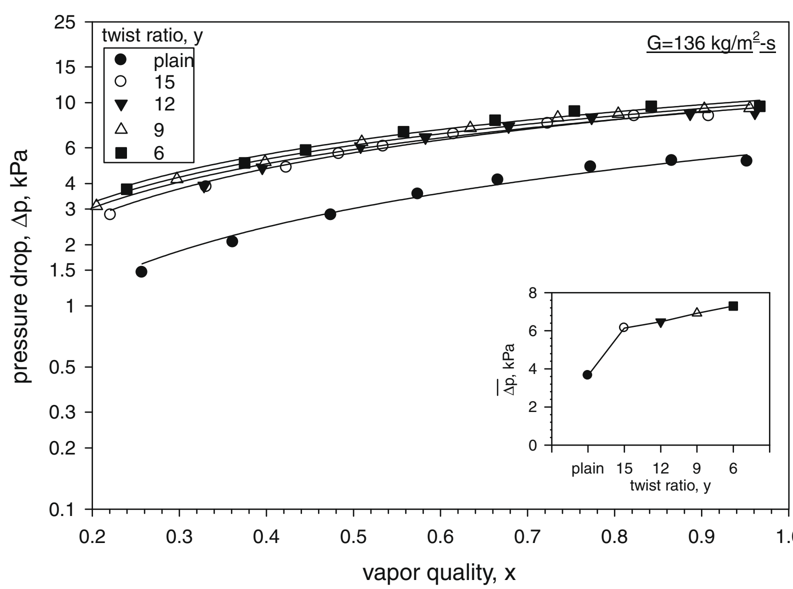


Fig. 12. Δp *v/s* x at 136 kg/s m^2.

The Blatt correlation [19] underpredicts the present experimental data in a range of 50 to 10 percent and the Agrawal [10] correlation underpredicts the experimental data in a range of 55 to 25 percent. Thus, for the present investigation the following correlation has been developed to predict the pressure drop in tubes with twisted tape inserts.

$$\frac{\Delta P_s}{(\Delta P_P)_{friedel}} = \frac{5.1}{y^{0.28}} \tag{19}$$

The mean deviation and standard deviation of swirl flow pressure drop calculated by Eq. (19) from the experimental data are 2.1 percent and 9.7 percent, respectively. The comparison between the computed pressure drop from developed correlation and the present experimental data has been shown in Fig. 14. As it is observed, most of the estimated values are within an error band of ±15 percent of the experimental data. So, the above developed correlation is in good agreement with the present experimental data.

It is a known fact that the augmentation of heat transfer with least increase in pressure drop causes saving in energy consumption, reduction in material used to produce heat exchangers, minimizing the heat exchanger size and eventually reduction in primary costs. In fact, to have an evaporator with appropriate performance, i.e., to gain the highest heat transfer rates with the lowest pressure drop is one of the primarily design goals.

3.5. Enhancement in heat transfer coefficient and pressure drop penalty

The previous data analysis has shown that the insertion of twisted tapes inside a plain tube increases the heat transfer coefficient in a range of 35 to 57 percent above the plain flow. On the other hand, the flow boiling pressure drop is also increased in a range of 40 to 180 percent above the plain flow. The variation of heat transfer coefficient to pressure drop ratio $(h/\Delta p)$ with vapor

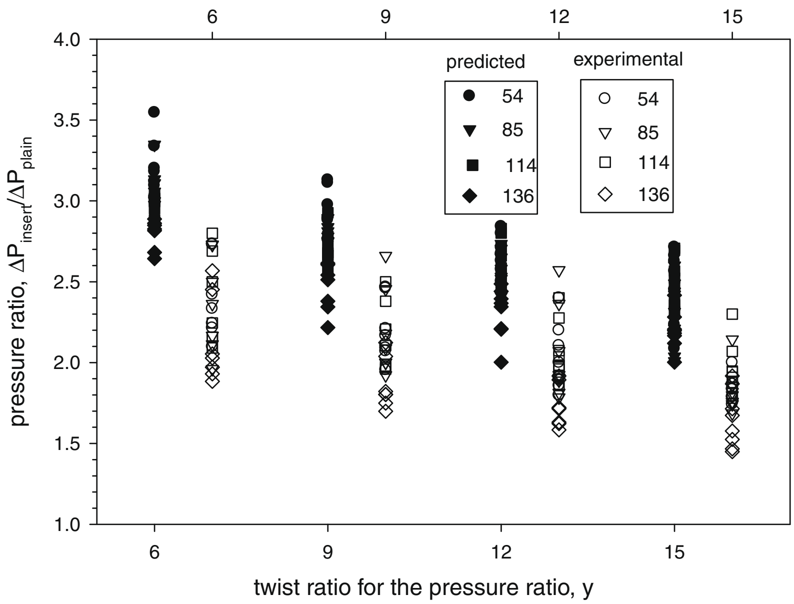


Fig. 13. $\Delta p_{insert}/\Delta p_{plain}$ *v/s* y.

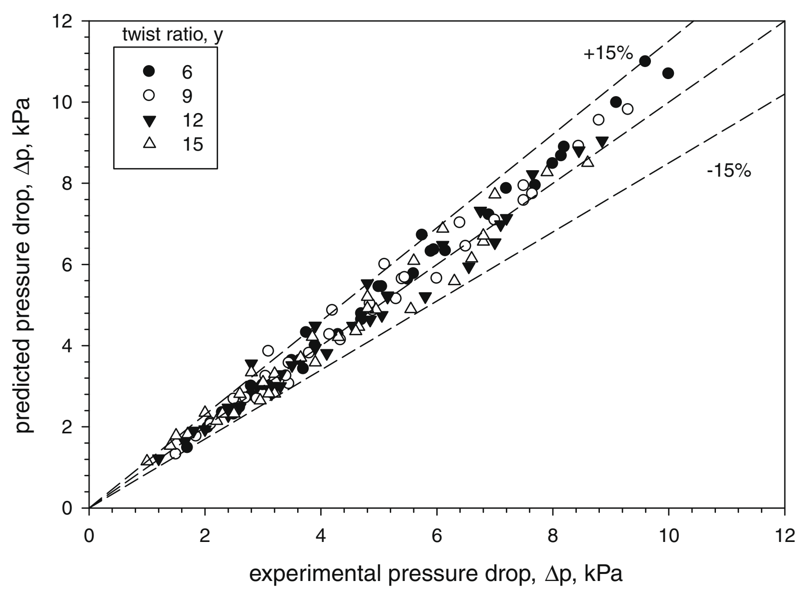


Fig. 14. Experimental Δp *v/s* Δp by Eq. (19).

quality has been shown in Fig. 15. From Fig. 15 it is also revealed that for plain flow and the tubes with twisted tape inserts, the heat transfer coefficient to pressure drop $(h/\Delta p)$ is reduced by increasing the vapor quality as slope of Δp–x curves is more than the slope of h–x curves. The high slope of the plain flow curve is due to the fact that in plain tube at low vapor qualities, the flow pattern is stratified-wavy and it changes to annular flow by increasing the vapor quality. This fact causes the pressure drop to increase faster than the heat transfer enhancement. But, for the tubes with twisted tape inserts, the flow pattern is annular and no change shall happen by increasing the vapor quality. From Fig. 15 it can also be concluded that the twisted tapes have a weak performance and it is always better to use them in the end part of the tube to prevent the formation of partial dryness and as a result to increase the heat transfer rates.

4. Conclusions

The following conclusions have been drawn from the present study:

1- The use of twisted tape inside horizontal tubes increases the flow boiling heat transfer coefficient. The insert with twist ratio of 6 has produced the highest increase in heat transfer coefficient in a range of 35 to 57 percent.
2- The twisted tape inserts also increase the pressure drop. The flow boiling pressure drop increases by as much as 180 percent above that for plain flow at mass velocity of 54 kg/s m^2.
3- Following correlation has been developed to predict the pressure drop during swirl flow boiling of R-134a inside a horizontal tube.

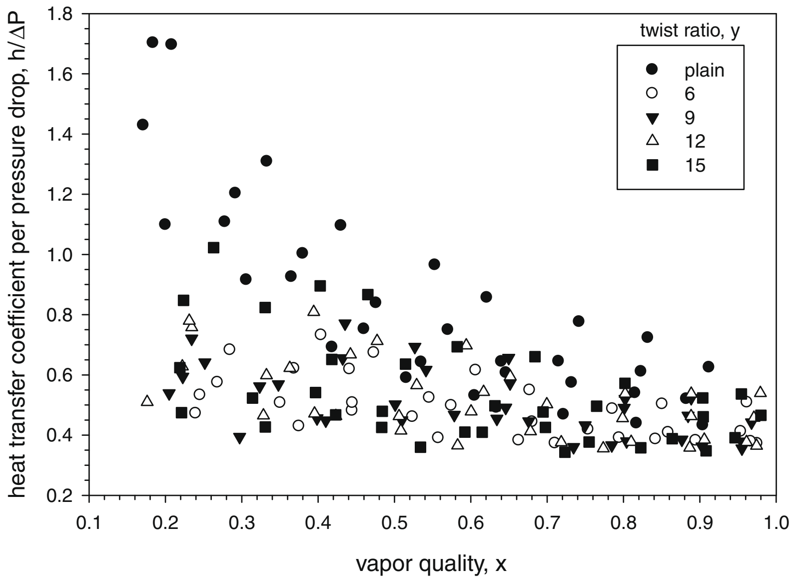


Fig. 15. $h/\Delta p$ *v/s* x at different y.

$$\frac{\Delta P_s}{(\Delta P_P)_{fr}} = \frac{5.1}{y^{0.28}}$$

This correlation predicts the experimental data in an error band of ±15 percent.

4- By considering the heat transfer and pressure drop, the plain tube has the best performance. The twisted tape inserts are recommended under special conditions by considering the appropriate consistence between heat transfer performance and the rate of pumping power increasing.

References

[1] A.E. Bergles, R.S. Reid, M.B. Pete, A comparison of augmentation techniques during in-tube evaporation of R-113, J. Heat Transfer 113 (1991) 451–458.
[2] A.E. Bergles, Techniques to augment heat transfer, in: Handbook of Heat Transfer Applications, McGraw-Hill, New York, NY, 1985, pp. 3.1–3.80.
[3] D. Chisholm, Pressure gradients due to friction during the flow of evaporating two-phase mixtures in smooth tubes and channels, Int. J. Heat Mass Transfer 16 (1973) 347–355.
[4] D. Steiner, VDI-Wärmeatlas (VDI Heat Atlas) verein deutscher ingenieure VDI-Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (GCV), Düsseldorf Chapter Hbb (1993).
[5] Didi Ould, M.B. Kattan, J.R. Thome, Prediction of two-phase pressure gradients of refrigerants in horizontal tubes, Int. J. Refrigeration 25 (2002) 935–947.
[6] H. Müller-Steinhagen, K. Heck, A simple friction pressure drop correlation for two-phase flow in pipes, Chem. Eng. Process. 20 (1986) 297–308.
[7] J. Collier, J. Thome, Convective Boiling and Condensation, Oxford University Press, 1994.
[8] J.M. Chawla, Laminar condensation inside horizontal and inclined tubes, ASHRAE J. 4 (1967) 52–60.
[9] J.R. Thome, Engineering Data Book III, Swiss Federal Institute of Technology, Lausanne, Switzerland, 2006.
[10] K.N. Agrawal, H.K. Varma, Pressure drop during forced convection boiling of R-12 under swirl flow, ASME 104 (1982) 758–762.
[11] L. Friedel, Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow, European Two-Phase Flow Group Meeting, Paper E2, Ispra, Italy, 1979.
[12] M.A. Akhavan-Behabadi, R. Kumar, A. Rajabi Najar, Augmentation of heat transfer by twisted tape inserts during condensation of R-134a inside a horizontal tube, Heat Mass Transfer 44 (2008) 651–657.
[13] R. Gronnerud, Investigation of liquid hold-up flow-resistance and heat transfer in circulation type evaporators part IV: two-phase flow resistance in boiling refrigerants, Annexe 1972-1 Bull. De l'Inst, du Froid (1979).
[14] R.R. Schultz, R. Cole, Uncertainty analysis in boiling nucleation, AIChE Symp. Ser. 75 (189) (1979) 32–38.
[15] R.W. Lockhart, R.C. Martinelli, Proposed correlation of data for isothermal two-phase two-component flow in pipes, Chem. Eng. Progr. 45 (1949) 39–45.
[16] S.G. Kandlikar (Ed.), Handbook of Phase Change: Boiling and Condensation, Taylor and Francis, 1999, pp. 367–402 (Chapter 15).
[17] S.G. Kandlikar, Critical heat flux in sub cooled flow boiling: an assessment of current understanding and future directions for research, Multiphase Sci. Technol. 13 (3&4) (2001) 2003–2010.
[18] S.G. Bankoff, A variable density single-fluid model for two-phase flow with particular reference to steam–water, J. Heat Transfer 2(B) (1960) 265–272.
[19] T.A. Blatt, R. Adt Jr., The effects of twisted tape swirl generators on the heat transfer rate and pressure drop of boiling Freon 11 and water, ASME 63 (1963) 63–71.