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ENHANCEMENT AND PREDICTION OF FLOW BOILING HEAT TRANSFER INSIDE HORIZONTAL TUBES CONTAINING TWISTED-TAPE INSERTS

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Abstract. *This paper presents an experimental study on heat transfer augmentation during convective boiling of R134a inside a 15.9 mm ID horizontal tube containing twisted-tape inserts. The use of twisted-tape inserts is a heat transfer enhancement technique applied on heat exchangers over more than one century ago. However, the heat transfer augmentation comes together with pressure drop increment, impacting the pumping power and, consequently, the system efficiency. Moreover, until now it is not clear when the effects on the heat exchanger efficiency of the heat transfer enhancement overcomes pressure drop penalties. In the present study, pressure drop and heat transfer coefficient experiments were performed. Experimental results were obtained during flow boiling of R134a for twisted-tape ratios of 3, 4, 9 and 14, mass velocities ranging from 75 to 200 kg / m^2 s, heat flux of 10 kW / m^2 and saturation temperatures of 5 and 15 °C. The results obtained lead to the conclusion that higher heat transfer rates can be achieved using twisted-tape inserts at the expense of a reasonable pressure drop. In addition, a new correlation to predict the heat transfer coefficients during two-phase flow inside tubes containing twisted-tape inserts is proposed. The performance of the correlation can be deemed adequate, considering that it compares well with experimental results of different authors.*

Keywords: Flow boiling, twisted tape, heat transfer enhancement, pressure drop, swirl flow.

NOMENCLATURE

d_i	Internal diameter [m]
E	Electrical power [kW]
e	Tape thickness [m]
f	Friction factor [dimensionless]
f_w	Surface material parameter
F	Forced convective heat transfer enhancement factor
G	Mass velocity [kg / m^2 s]
h	Heat transfer coefficient [kW / m^2 K]
H	Length of 180° tape turn [m]
k	Thermal conductivity [kW / m K]
L	Length [m]
M	Molecular mass [kg / kmol]
m	Exponent of ϕ
$\dot{m}$	Mass flow rate [kg / s]
Re	Reynolds number [dimensionless]
Ra	Surface roughness of the material [μm]
V	Fluid velocity [m / s]
T	Temperature [K]
x	Vapor quality
y	Twist ratio
z	Distance from inlet [m]
v	Kinematic viscosity [m^{-2} s]
Δp	Pressure drop [kPa]
ϕ	Heat flux [kW / m^2]
ε	Enhancement/penalty factors [dimensionless]
ζ	Absolute mean deviation

Subscripts

acc	Accelerational
a	Axial component
$fric$	Frictional
$grav$	Gravitational
l	Liquid
v	Vapor
p_r	Reduced pressure [dimensionless]
Pr	Prandtl number [dimensionless]
s	Swirl flow boiling
sat	Saturation condition
TS	Test section
TT	With twisted tape
W	Wall

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1. INTRODUCTION

The use of heat transfer enhancement techniques are observed in numerous applications in the refrigeration and air-conditioning sectors. In general, enhancement techniques can be divided into two groups: namely active and passive techniques. A detailed description of these techniques is given by Webb (1994). One of the passive techniques to enhance the heat transfer is the insertion of devices in tubes such as twisted tapes. According to Bergles (1999) several investigations carried out earlier in this direction have established that twisted-tape insert are promising devices for augmenting heat transfer rate in heat exchangers. Moreover as pointed out by Wongcharee and Eiamsa-Ard (2011), among the techniques used in the passive method, the use of twisted tape in the enhanced tube as the swirl generator is apparently prominent, due to its low cost and high efficiency for improving heat transfer rate. In addition, as pointed out by Ribatski and Thome (2005) and Akhavan-Behabadi et al. (2009), twisted-tapes are wide spreading used due to the possibility of being installed in heat exchangers already in use. Jensen and Bensler (1986) have examined saturated forced-convective boiling heat transfer with twisted-tape in a vertical tube. According to them, the heat transfer coefficient increases with decreasing twist-ratio. These authors have obtained heat transfer enhancement up to 2 times compared to the plain tube counterparts. In the study of Hejazi et al. (2010), heat transfer enhancements up to 40% were observed. Cumo et al. (1974) and Agrawal et al. (1986) have also observed that the flow boiling heat transfer coefficient is higher for tubes containing twisted-tapes. Recently, Mogaji, et al. (2013) studied the effect of twisted-tape inserts on flow boiling heat transfer enhancement and pressure drop penalty. The authors observed heat transfer coefficient enhancements up to 36 %, in comparison to experimental values for plain tubes.

In general, the heat transfer enhancement obtained through this technique is accompanied by a drastic increase in pressure drop. This increment in pressure drop increases the pumping cost which consequently impacts the system efficiency. However, Shatto and Peterson (1996) highlighted the fact that the negative result of increasing the pressure drop gradient by the use of inserts can be overcome by a reduction in the heat exchanger size due to the heat transfer enhancement. Additionally, Bandarra-Filho and Saiz-Jabardo (2006) have pointed out that the use of inserting devices especially in dry expansion evaporator coils are in fact efficient in upgrading the heat transfer coefficient but at the cost of significant pressure drop. Therefore the use of this heat transfer enhancement technique should be optimized taken into account the benefits due to the heat transfer coefficient augmentation and the cost of efficiency because of pressure drop increase.

For this purpose, the present study was developed to evaluate the performance of twisted tape inserts on augmentation of the heat transfer coefficient and pressure drop during flow boiling of R134a inside horizontal tubes. This study covered a broad range of experimental conditions. The overall performance was evaluated in terms of enhancement parameter which is based on the tradeoff between both heat transfer augmentation and pressure drop penalty. Finally, a new correlation was developed to predicts two phase flow heat transfer coefficient inside tubes containing twisted-tape inserts. The correlation captures most of the experimental data obtained in the present study.

2. EXPERIMENTAL APPARATUS AND METHODS

The experimental set-up comprises the refrigerant (also named test circuit) and ethylene–glycol circuits. Figure1 illustrates a simplified schematic of the refrigerant test loop where the heat transfer and pressure drop measurements were performed. The test fluid is driven through the refrigerant circuit by a gear micropump (self-lubricating). The refrigerant mass flow rate is measured by a Coriolis-type flow meter positioned just downstream the micropump. Before entering the pre-heater, the refrigerant passes through a sub-cooler that consists of a tube-in-tube type heat exchanger. So, it can be assured that the fluid is free of vapor bubbles and its thermodynamic state at the pre-heater inlet can be determined from the measurements of its temperature and pressure. The cooling effect in the sub-cooler is obtained through the anti-freezing solution from the ethylene–glycol circuit. The test section is a 2 m long copper tube of 15.9 mm ID (5/8 in.) and 3.2 mm thickness (1/8 in.), located downstream the calming section between two visualization sections. Two visualization sections were used with the aim to verify flow pattern transitions along the test section during the experiments. The test section surface temperature was measured at four equally spaced cross sections along the tube length by using K-type thermocouples of 120 μm wire diameter and with accuracy of 0.13^{o}C. These thermocouples were nested in longitudinal grooves 0.5 mm distant from the internal surface, filled with high-conductive epoxy and distributed in four sections 460 mm distant from each other. At each measuring cross section, the surface temperature is read at four locations 90^{o} spaced from the bottom to the top of the tube. The pre-heater and test section are heated by tape electrical resistors (624 W/240 V each tape) uniformly wrapped on the external surface of the tubes, guaranteeing a uniform heat flux. The electrical powers supplied to the pre-heater and test sections are adjusted through three manually controlled voltage converters and measured using power transducers. The refrigerant vapor quality at the test section inlet is set by adequately adjusting the electrical power supplied to the pre-heater. The refrigerant bulk temperature is measured at the pre-heater inlet, and test section outlet, through thermocouples installed in bulbs positioned in the center of the cross section of the tube. Local pressure measurements were taken at the pre-heater, stabilization section, and test section inlet using absolute pressure transducers with measurement span of 120 kPa. Three differential pressure transducers were installed in the refrigerant circuit with pressure probes positioned at the test

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section inlet and outlet, each one with accuracy of 0.075% of the set span, and measurement ranges of 3, 10 and 300 kPa. The refrigerant vapor generated in the pre-heater and test section is condensed in a shell and tube heat exchanger cooled by a 60% solution of ethylene–glycol /water. The solution is cooled by a chiller located in the ethylene–glycol circuit specially built for operation in the experimental set-up. The refrigerant reservoir is used to adjust the refrigerant inventory without adding any external refrigerant charge to the overall refrigerant circuit (including the reservoir itself).

The implementation of this technique occurs through the twisting of a metal tape which is then inserted into the plain tube. The twisted-tapes are characterized by the twist ratio given by the ratio between 180^o turn length (H) and tube internal diameter (d_i). The twisted-tapes were manufactured as suggested by Lopina and Bergles (1969). The twisted tapes were made of an aluminum foil of 1 mm thick and 15.3 mm width, consequently the tapes are installed in the tube, since the test section internal diameter is 15.9 mm. Each insert was composed of a unique and continuous piece of 3.8 m total length, therefore their lengths covered the visualizations, test and calming sections. The uniformities of the twist ratio along the tube were checked before introducing the insert into the tube and after its removal.

Initially, data for plain tube without inserts were acquired to establish the accuracy of the experimental apparatus and to have a reference data for comparing with the performance of the tube with twisted tape inserts. Experiments were performed for mass velocities of 75, 100, 150, and 200 kg / m^2 s, saturation temperature of 5 and 15 °C and twist ratios (defined according to Fig. 2) of 3, 4, 9 and 14. The work of Mogaji, et al. (2013) is suggested for further details about the experimental equipment and methods.

The temperature and mass flow rate measurements were calibrated before the experimental campaign. The accuracy of the experimental data was determined according to the procedure suggested by Abernethy and Thompson (1973). Accounting for all instrument errors, uncertainties for the calculated parameter were estimated using EES (V9.103-3D), Engineering Equation Solver, whose procedure is based on the methodology suggested by Taylor and Kuyatt (1994).

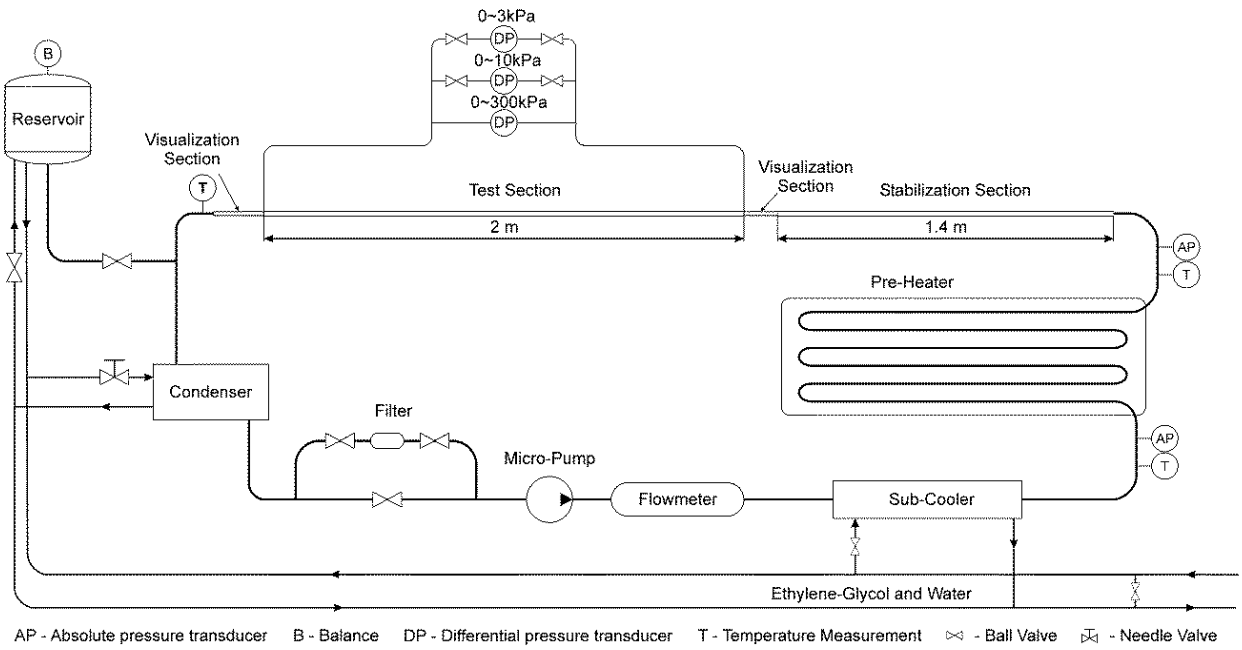

Figure 1 Schematic diagram of the refrigerant circuit.

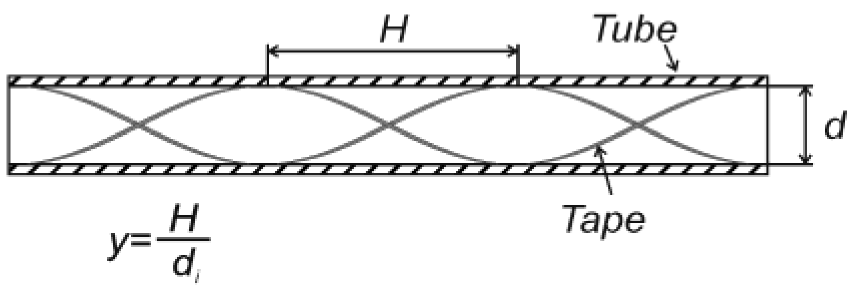

Figure 2 Schematic view of twisted-tape insert inside a tube.

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2.1 Data reduction

In order to analyze the experimental data, it is necessary to reduce them into desire parameters as follows: The local vapor quality was determined based on energy balance through pre-heater and test section, considering the electrical power supplied by electrical resistances and the heat losses (or gains) to the environment.
The mass velocity is given by the ratio between the mass flow rate, $\dot{m}$, measured by the Coriolis type mass flow meter and the tube internal area, neglecting the tape thickness given as follows:

$$G = \frac{4\dot{m}}{\pi d_i^2} \tag{1}$$

The total pressure drop over the test section is the sum of the frictional, accelerational, and gravitational pressure drop components as follows:

$$\Delta p_{Total} = \Delta p_{fric} + \Delta p_{acc} + \Delta p_{grav} \tag{2}$$

Since the pressure drop experiments were performed for a horizontal tube and under almost adiabatic conditions, the gravitational and accelerational pressure drop contributions were negligible, therefore the frictional pressure drop was given by the pressure drop measured by the differential pressure transducers. The frictional pressure drop gradients were given by the ratio between the frictional pressure drop and the test section length (L) of 2.0 m.

The radial heat flux, ϕ_{TS}, for the test section was calculated as follows:

$$\phi_{TS} = \frac{E_{TS}}{\pi d_i L} \tag{3}$$

where E_{TS} is the electrical power supplied to the test section taken into account the heat exchanges with the external environment.

Finally, the heat transfer coefficient was calculated according to the Newton´s cooling law as follows:

$$\bar{h} = \frac{\phi_{TS}}{\left(\bar{T}_W - T_{sat(z)}\right)} \tag{4}$$

where $\bar{T}_W$ is the arithmetic average surface temperature of the inner tube wall estimated according to the Fourier's law assuming one-dimensional conduction and based on the four wall temperature measurements at each cross section and $T_{sat}(z)$ is the local saturation temperature of the refrigerant evaluated from the local saturation pressure assuming a constant pressure gradient along the test section.

2.2 Validation of the Experimental Bench and Procedures

Single-phase frictional pressure drop experiments for plain tube without inserts were performed in order to assure the accuracy of the pressure drop measurements. The single-phase results were compared against the well-established correlations of Blasius (1913), Colebrook (1939), apud White (1998), and Churchill (1977) respectively. Figure 3a, shows that the measured single-phase frictional pressure drop are in good agreement with those predicted by the correlations. In case of Colebrook (1939), and Churchill (1977), the friction factor was calculated considering a surface roughness of 9.5 μm. This value was obtained from surface roughness measurements along the test section.

A similar analysis was performed for heat transfer coefficient data. Single-phase experimental results were compared against the predictions of Gnielisnki (1976) and Dittus-Boelter (1930) correlations. Figure 3b, shows a comparison between experimental and estimated heat transfer coefficients covering Reynolds numbers from 5000 to 32,500. These comparisons show that the experimental data agree quite well with the predicted values. Based on the above analyses, it can be concluded that the apparatus is reasonably accurate for measuring single and two-phase flows for tubes with and without twisted-tape inserts.

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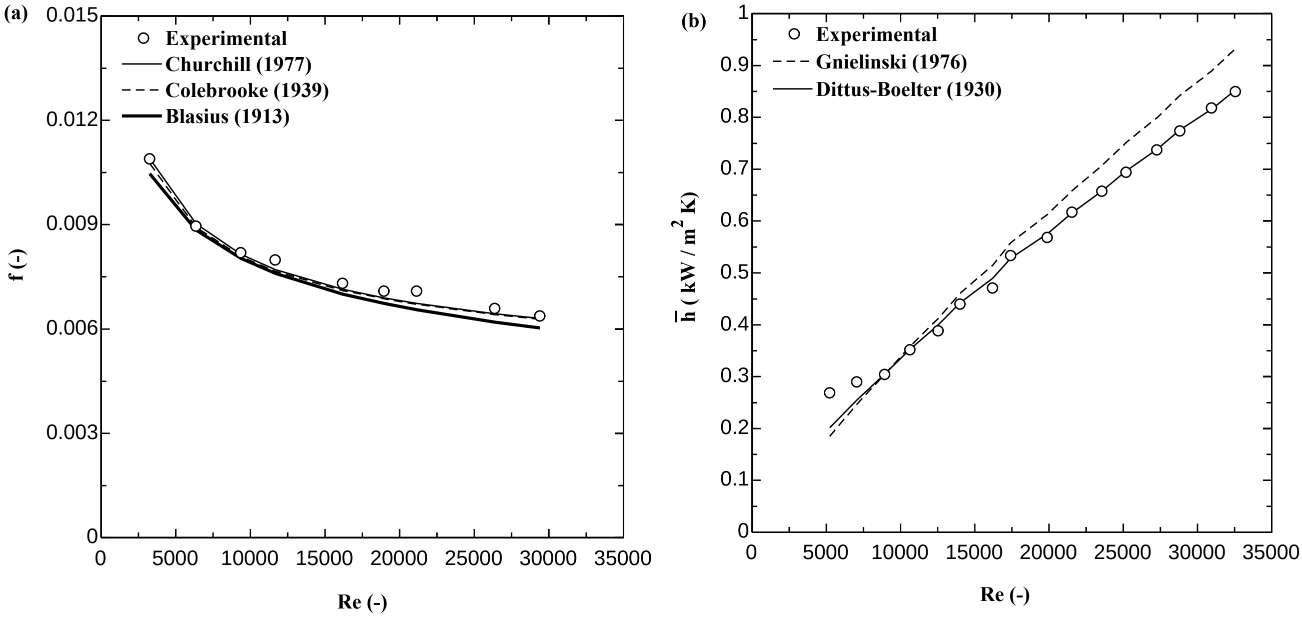

Figure 3 Comparison between the experimental and predicted values for single-phase flow (a) Friction factor; (b) heat transfer coefficient.

3. RESULTS AND DISCUSSIONS

In this section, pressure drop and heat transfer coefficient results for the tube with twisted-tape inserts are comparatively reported with those of the tube without insert. Adiabatic pressure drop and flow boiling heat transfer experiments were conducted for mass velocities of 75, 100, 150 and 200 kg / m^2 s, saturation temperature of 5 and 15 °C for the plain tube and the tube with twisted-tape inserts. The pressure drop and heat transfer coefficient results observed in this study for a tube without twisted tapes follows the main trends pointed out in the literature as reported in Mogaji, et al. (2013).

3.1 Pressure drop gradient for tube with twisted tape inserts

Similar to the trends observed in the previous studies of Akhavan-Behabadi et al. (2009) and Kanizawa and Ribatski (2012), in the present study the tube with twisted-tape presented higher pressure drop than the tube without insert. Additionally the pressure drop in tube with twisted tape increases with decreasing twist-ratio and increasing mass velocity. The pressure drop augmentation by the twisted tape is related to the fact that, additionally to the reduction of the cross sectional area, the insert induces turbulence and swirl effects on the liquid film and vapor core.

3.2 Heat transfer coefficient results for tube with twisted tape inserts

Figure 4 shows the variations of the heat transfer coefficient with vapor quality for a heat flux of 10 kW / m^2 for different mass velocities values and a saturation temperature of 15 °C. According to this figure, the use of twisted-tape causes a significant heat transfer coefficient increase. The higher turbulence intensity of fluid in the vicinity of the tube wall generated by the twisted tape compared to that induced in the plain tube counterpart is referred as the main reason for this behavior. As shown in Fig. 4, the heat transfer coefficient for the tube with inserts and mass velocities of 75 and 100 kg / m^2 s remained almost constant over nearly the whole range of vapor qualities. Such a behaviour is typical of stratified flows as reported by Bandarra-Filho and Saiz-Jabardo (2006) and Akhavan-Behabadi et al. (2009). Similarly, according to the results for mass velocities of 150 and 200 kg / m^2 s displayed in Fig. 4, the twisted tapes are effective for the entire range of vapor quality and the increment of the heat transfer coefficient seems to be independent on the value of twist ratio.

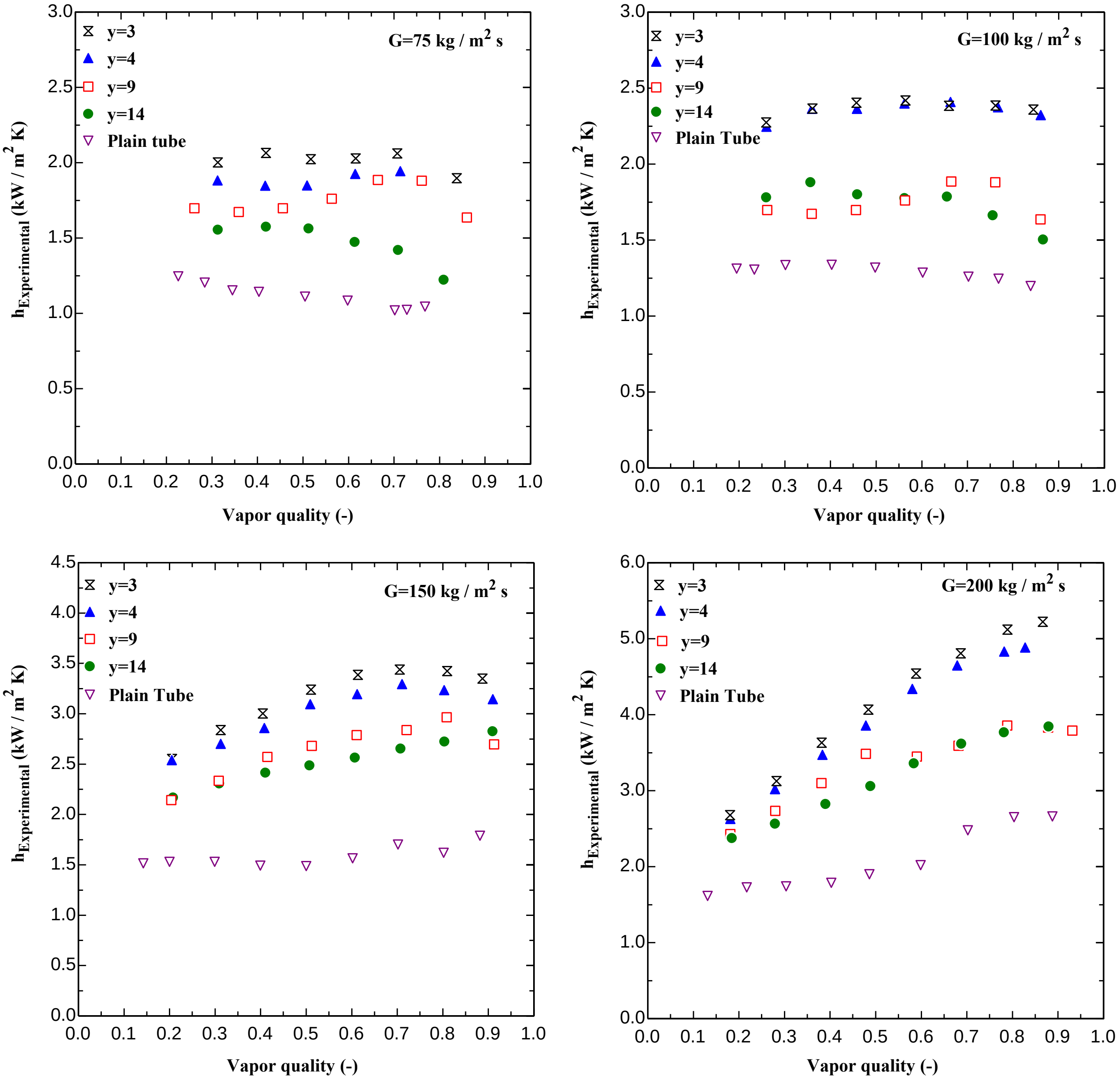

Figure 4 Heat transfer coefficient variation with vapor quality for various mass velocities, heat flux of10 kW / m^2 and T_{sat}=15°C

4. OVERALL PERFORMANCE OF THE HEAT TRANSFER ENHANCEMENT TECHNIQUE

In order to determine the overall performance of the heat transfer enhancement technique, enhancement parameter as suggested by Webb (1994) was evaluated. The enhancement parameter, ε , is defined as the ratio between the heat transfer enhancement and pressure drop penalty factors. The heat transfer enhancement, pressure drop penalty factors and enhancement parameter are given as follows:

$$\varepsilon_h = \frac{h_{TT}}{h_{PT}} \tag{5}$$

$$\varepsilon_p = \frac{\Delta p_{TT}}{\Delta p_{PT}} \tag{6}$$

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$$\varepsilon = \frac{\varepsilon_h}{\varepsilon_p} \tag{7}$$

According to Agrawal and Varma (1990), a value of ε higher than one suggests a condition under which the use of the twisted tape is profitable, since the heat transfer gain is higher than the pressure drop penalty. Using the twisted tape data obtained in the present study, for a mass velocity of 200 kg / m^2 s under high vapor quality conditions, enhancement parameter values of 1.15, 1.026, 0.983 and 1.012 were achieved for the twist ratios of 3, 4, 9 and 14 respectively at saturation temperature of 15 ^{o}C. This implies that the application of enhancement techniques using twisted-tape is suitable and would be an interesting substitute for the tubes without inserts

5. CORRELATION OF HEAT TRANSFER DATA

In the present study, twisted-tape insert heat transfer coefficients data is correlated by adopting the following steps as suggested by Shatto and Peterson (1996). (1) Obtain a reasonable prediction of the present plain tube flow heat transfer coefficients using well-established correlation from the literature; (2) modify the plain tube correlation to predict the twisted-tape heat transfer data.

The correlation of Liu and Winterton (1991) was selected since it provided the best predictions of the flow boiling heat transfer coefficient inside a tube without twisted-tape inserts data gathered in the present study reported by Mogaji et al. (2013).This correlation was developed from the method of combining nucleate boiling and single phase forced convection heat transfer mechanisms, in a similar manner

The proposed heat transfer coefficients correlation to predict the heat transfer coefficient inside tubes with twisted tape inserts is given as follows:

$$h_{TTproposed}{}^2 = \left(F_{TT} h_{lTT}\right)^2 + \left(S_{TT} h_{poolNew}\right)^2 \tag{8}$$

where the single-phase twisted tape insert heat transfer coefficient, h_{lTT}, is calculated from a correlation developed for tape-generated swirl flow by Naphon (2006). The Naphon (2006) single-phase swirl flow correlation is given by the following equation:

$$h_{l\text{Naphon (2006)}} = 0,648\,\text{Re}^{0.36}\left[1+\frac{d_i}{H}\right]^{2.475}\text{Pr}^{1/3}\,\frac{k_l}{d_i} \tag{9}$$

As pointed out by Lopina and Bergles (1969), the swirl effect of the twisted-tape insert can be accounted by substituting the Re in Eq. (9) by the term called swirl Reynolds number, Re_s. The swirl effect is referred to as spiral convection heat transfer effect which consists of the convectional turbulent forced convection mechanism with special considerations to account for the increased velocity at the tube wall caused by the tangential component of the fluid motion. The swirl Reynolds number, Re_s is given as follows:

$$\text{Re}_s = \text{Re}\left[\frac{\sqrt{\pi^2+4y^2}}{2y}\right] \tag{10}$$

where Re is calculated by:

$$\text{Re} = \frac{V_a d_i}{v} \tag{11}$$

The axial velocity V_a is given by:

$$V_a = \frac{\dot{m}}{\rho\left[\frac{\pi}{4}d_i^2 - ed_i\right]} \tag{12}$$

where e is the tape thickness.

Hence Eq. (9) is modified as follows:

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$$h_{lTT} = 0.648\,\mathrm{Re}_s^{\;0.31}\left[1+y^{-1.6}\right]^{2.475}\mathrm{Pr}^{1/3}\left(\frac{k_l}{d_i}\right)^{0.02} \tag{13}$$

The convective heat transfer enhancement factor is given as follows:

$$F_{TT} = 22.9\left[1+x\,\mathrm{Pr}\left(\frac{\rho_l}{\rho_v}-1\right)\right]^{0.3515} \tag{14}$$

The nucleate boiling heat transfer coefficient, $hpoolNew$, is calculated according to Ribatski and Saiz Jarbado (2003) correlation given as follow:

$$hpool_{New} = \phi^{m}\left(f_W\,p_r^{\;0.45}\left[-\log\left(p_r\right)\right]^{-0.8}Ra^{0.2}M^{-0.5}\right) \tag{15}$$

where

$$m = 0.9 - 0.3\,p_r^{\;0.2} \tag{16}$$

and the surface material parameter for copper tube, $f_W = 100$ (17)

The nucleate boiling suppression factor, S_{TT} is calculated as:

$$S_{TT} = 0.03\left(1+2.5F_{TT}^{\;(-0.05)}\,\mathrm{Re}_s^{\;0.25}\right)^{0.15} \tag{18}$$

where the constants and exponents given above were determined based on the data obtained in the present study by using least square regression analysis.

5.1 Evaluation of the proposed correlation

The absolute mean deviation ζ and the parcel of the data predicted within an error band of ±30 % of correlations with respect to experimental results were considered as the reference parameter in the evaluation of the proposed correlation. Table 1 presents the statistical comparisons between the proposed correlation, predictive methods of Agrawal et al (1986), Jensen and Bensler (1986), Akhavan-Behabadi et al. (2009) and the data obtained in the present study. According to Tab. 1, the correlation proposed in the present study predicts satisfactorily well the database with 82% of the predictions within an error band less than ±30%. An overview of the correlation performance with respect to the present study experimental twisted-tape heat transfer data can be seen in the plot of the proposed model versus the experimental heat transfer coefficients of Fig. 5. As presented in Tab. 1, both methods of Jensen and Bensler (1986) and Agrawal et al. (1986) fail to predict most of the experimental trends. Such a result was expected, given the relatively wide range of operational conditions and the different refrigerant involved in the present database. Additionally, the method by Jensen and Bensler (1986) is based on data for upwards flow in a vertical tube, so, it is not surprising that the method fails to predict the experimental trends for horizontal tubes of the present study. Better than the other two correlations and the proposed model, the method by Akhavan-Behabadi et al. (2009) agrees quite well with the present experimental database, predicting 88.5% of the data within an error band of ± 30%. Here, it must be emphasized that a good predictive method should not only be statistically accurate, but also able of capturing the main trend of the experimental results. Taking this into account, Figs. 6 and 7 display the evolution of the heat transfer coefficient versus vapor quality in comparison to the proposed model and Akhavan-Behabadi et al. (2009) correlation for different twist ratios. Though the statistical evaluated results shown in Tab. 1 reveal that the method by Akhavan-Behabadi et al. (2009) correlation is more accurate in average than the correlation proposed in the present study. As showed in Figs 6 and 7, the proposed method captured the main trend of the data obtained in the present study satisfactorily well. Additionally, the proposed method accurately predicted the increase in the heat transfer coefficient with increase in vapor qualities and decreasing twist ratios compared to the poor performance of Akhavan-Behabadi et al. (2009) correlation using the same experimental operational conditions. Moreover, the method proposed in the present study has been developed by considering the physical picture of the swirl flow phenomenon and takes into account the swirl flow effects of the twisted-tape inserts whereas the predictive method of Akhavan-Behabadi et al. (2009) is a purely empirical type method.

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The performance of the proposed correlation has also been evaluated through comparisons with experimental results obtained elsewhere. According to Fig. 8, the proposed model compares satisfactorily well with data from Agrawal et al. (1986) despite the fact that their data include refrigerants different from the one used in the present study.

Data from Akhavan-Behabadi et al. (2009) for the same refrigerants considered in the present investigation have also been considered for comparison. Figure 9 shows the comparison between the estimated versus the experimental heat transfer coefficient. According to this figure, the method proposed in present study accurately predicted the database of Akhavan-Behabadi et al. (2009) predicting 96% of the data points within an error band of ±30% and absolute mean deviation of 11.2%. The results seem to be quite reasonable, considering that neither the constant nor the coefficients of the proposed correlation have been adjusted to the experimental database of Akhavan-Behabadi et al. (2009).

Table 1. Statistical comparisons of correlations from literature and the proposed correlation with the data obtained in the present study.

Authors	ζ (%)	λ_{30} (%)
Proposed correlation	20.0	82.0
Jensen and Bensler (1986)	23.4	36.3
Akhavan-Behabadi et al (2009)	17.0	88.5
Agrawal et al. (1986)	41.3	28.0

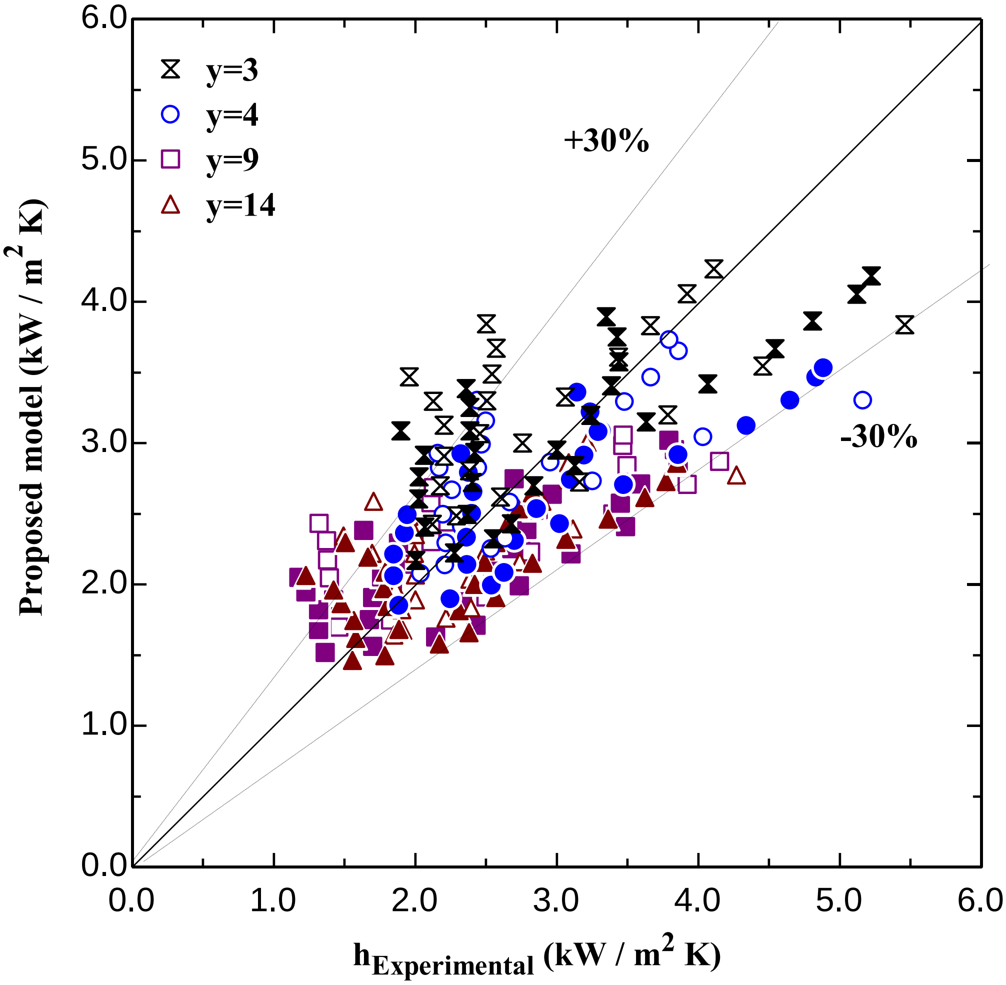

Figure 5 Comparison between experimental heat transfer coefficients data obtained in the present study and proposed model T_{sat} = 5°C (empty symbol) and 15°C (filled symbol)

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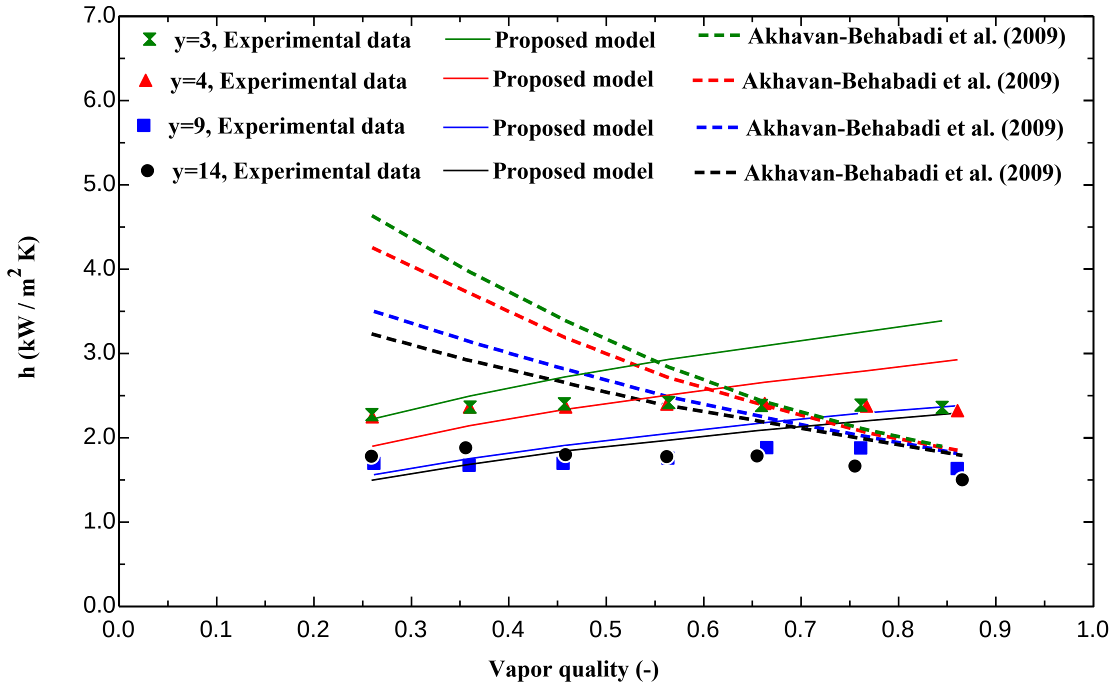

Figure 6 – Heat transfer coefficient variation with vapor quality, G=100 kg / m^2 s , T_{sat}=5^{o}C

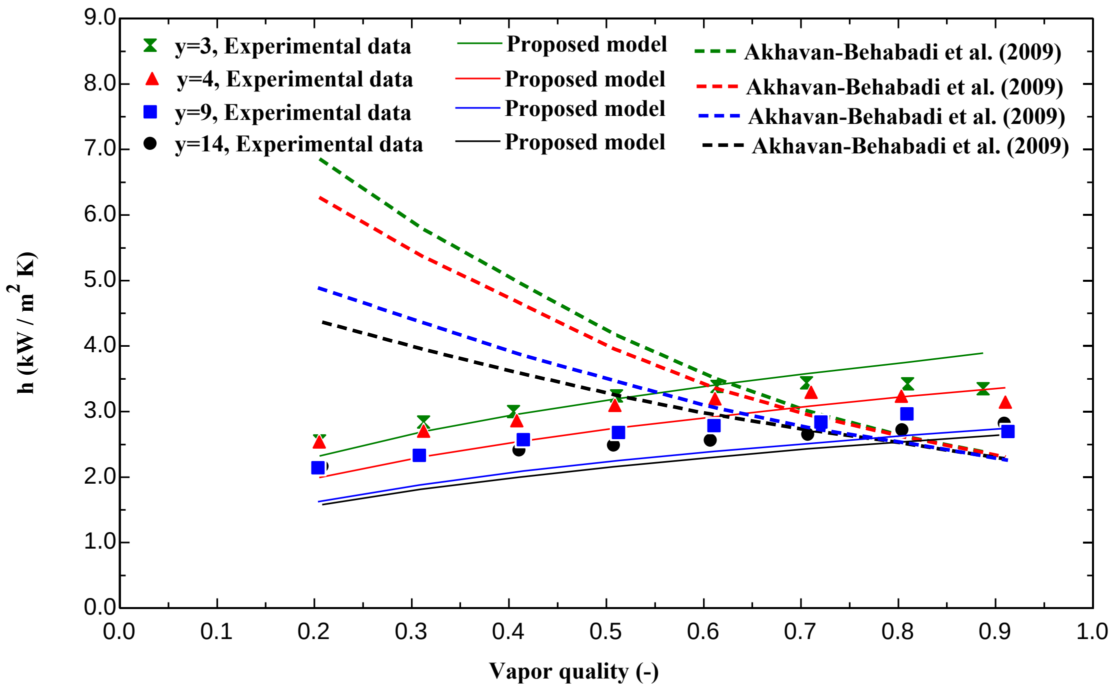

Figure 7 – Heat transfer coefficient variation with vapor quality, G=150 kg / m^2 s , T_{sat}=15^{o}C

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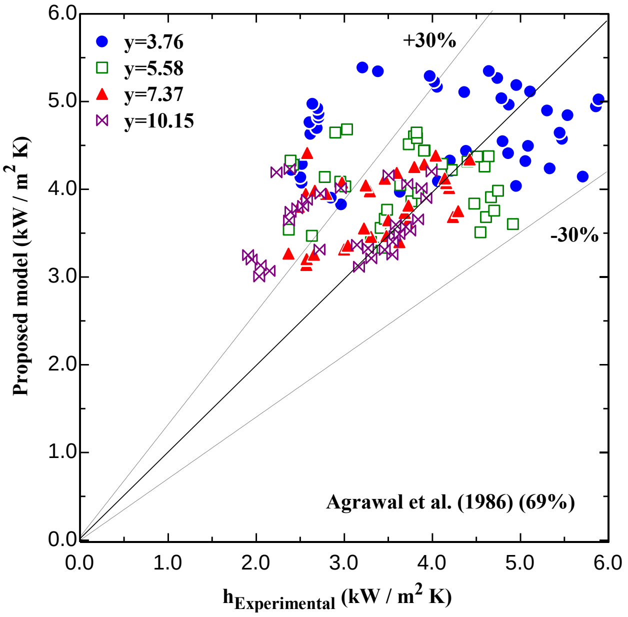

Figure 8 Comparism between experimental heat tarnsfer data from Agrawal et al. (1986) and the present study model

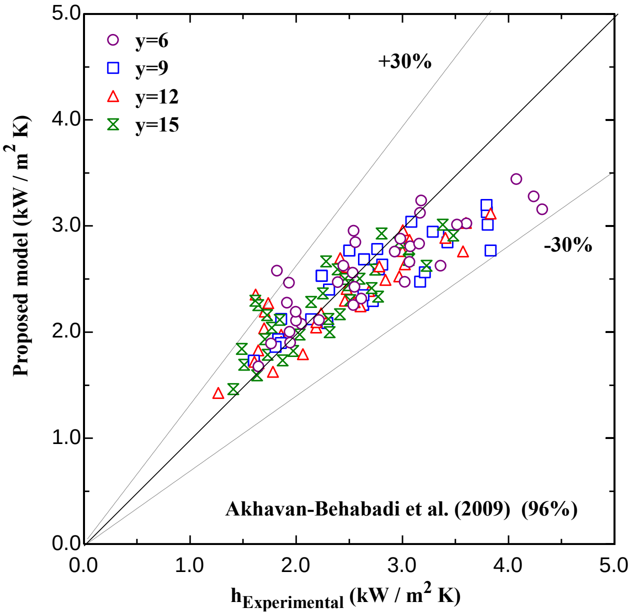

Figure 9 Comparism between experimental heat tarnsfer data from Akhavan-Behabadi et al. (2009) and the present study model

6. CONCLUSIONS

- Twisted tape insert has been found to increase the heat transfer coefficient compared to the plain tube counterparts. The magnitude of the increment depends upon the flow conditions and the twisted tape geometry.
- The heat transfer correlation of Liu and Winterton (1991) provided the best prediction of the flow boiling data gathered in the present study for a plain tube without inserts, predicting 89% of the data within an error band of ± 30%.
- The heat transfer correlation of Akhavan-Behabadi et al. (2009) provided the best prediction of the heat transfer data obtained in the present study for the tube with twisted tape, predicting 88.5% of the data within an error band of ± 30%.
- Among the inserts tested, the enhanced tube with twist ratio 3 gave the highest heat transfer enhancement for the same flow condition and is around 2 times the value of the plain tube counterparts.
- The analysis of the enhancement parameter allowed concluding that the use of twisted tape is suitable when it is applied to the high vapor quality region of the evaporator and under mass velocities of 200 kg / m^2 s.
- New correlation has been developed which satisfactorily predicted the flow boiling twisted tape heat transfer coefficient data obtained in the present investigation.
- Performance evaluation of the proposed correlations can be deemed adequate, considering that it compares well with independent experimental results of different authors.

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