Condensation heat transfer and pressure drop characteristics of Isobutane in horizontal channels with twisted tape inserts

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Abstract

The enhancement of heat transfer rate in heat exchangers has been considered a major concern. In this research, the impacts of inserting twisted tapes in horizontal two-phase flow heat exchangers are discussed, and different values of vapor qualities (in the range of 0.1 to 0.7) and mass velocities (in the range of 119 to 367 $kgm^{-2}s^{-1}$) are considered during forced convective condensation of R600a (Isobutane). The test case is a pipe made from copper with an inner diameter of 8.1 mm and a length of 1000 mm. Furthermore, three twisted tape inserts with various twist ratios (defined as the ratio of the twisted tape pitch to the test pipe inner diameter) of 4, 10, and 15 are used. The results illustrated that installing twisted tapes results in the increment of pressure drops and the rate of heat transfer in comparison to the smooth case. Furthermore, the pressure drops and heat transfer rates augment as the refrigerant mass velocity and vapor quality increase. Depending on the inserts type and operating conditions, the performance factor (a criterion to assess the performance modification compared to the primary test case) between 0.39 to 1.05 was obtained. It was also observed that there exists an optimum amount of the refrigerant mass velocity at which the performance factor is higher. Results showed that generally using twisted tapes in heat exchangers is not instrumental unless when the main concern is to improve the heat transfer rate or when the augmented power consumed by pump can be justified.

Keywords:

Condensation; Heat transfer coefficient; Pressure drop; R600a; Twisted tape insert

Nomenclature:

d	Tube diameter(mm)
G	Mass velocity ($kgm^{-2}s^{-1}$)
h	Enthalpy
PDR	Pressure drop ratio
PF	Performance factor
p	Pressure (kPa)
x	Vapor quality

Greek symbols

ε	Zivi void fraction
μ	Viscosity
ρ	Density

Subscripts

eva	Evaporator
f	Fluid
fg	Vaporization
fric	Frictional
g	Vapor phase
i	Inner
mom	Momentum
o	Outer
r	Rough
s	Smooth
tot	Total
wat	Water

1. Introduction

The tendency toward utilizing compact and efficient heat exchangers in different industrial applications such as HVAC and refrigeration systems has increased considerably. While there exist numerous techniques for heat transfer rate enhancement in heat exchangers [1], using cost-effective methods is notable. Installing twisted tape inserts, as a passive method of heat transfer improvement, inside heat exchangers is an effective and beneficial technique. Producing twisted tapes is easy and cost-effective. These instruments do not require high maintenance considerations and contribute to a considerable improvement in the value of transferred heat within heat exchangers by inducing a swirling flow and promoting flow turbulence. Garg et al. [2] have reviewed using twisted tapes for the promotion of single-phase and two-phase heat transfer in different applications.

Two-phase heat transfer mechanism (i.e. flow boiling or condensation) is able to deliver a higher rate of heat transfer in comparison to the single-phase mechanism due to the significant amount of exchanged heat during phase change from liquid to vapor or vice versa. Despite the numerous studies conducted on single-phase flows using twisted-tapes as turbulators [3-11], the number of studies on two-phase flows is limited.

Within an empirical investigation, Akhavan-Behabadi et al. [12] installed several twisted tapes inside horizontal heat exchanger pipes under different operating conditions. For this purpose, they used inserts with twist ratios in the range of 6 to 15. Their results illustrated that implementing inserts raised both evaporative heat transfer rate (up to 57%) and pressure drop (up to 180%) of R134a. Similarly, Kanizawa et al. [13] empirically evaluated the impacts of using twisted tapes on the performance of pipes located horizontally using R134a as working refrigerant. It was observed that twisted tapes augmented the pressure drops and heat transfer coefficients. However, the amounts of their growth were significantly dependent on the operating conditions and flow patterns. It was reported that by the transition of flow regimes from intermittent pattern to annular pattern, the heat transfer rate enhanced considerably. Also, by raising the vapor quality, the value of pressure drops was reduced. Hejazi et al. [14] assessed the influences of twisted tapes with twist ratios between 6 to 15 on the performance of pipes with a length of 1004 mm and an internal diameter of 10.7 mm during condensation of R134a. Among the instruments employed, the insert with the twist ratio of 9 demonstrated a better performance by augmenting the value of transferred heat with a lower pressure drop.

However, it is noticeable that the above-mentioned studies were conducted using R134a which is a traditional refrigerant with a considerable global warming potential (GWP) of 1300 [15, 16], although it has good thermodynamic properties. According to the Kyoto Protocol under the United Nations Framework Convention on Climate Change (UNFCCC), it is required to reduce the emissions of six groups of greenhouse gases, R134a included [17, 18]. Therefore, it is necessary to introduce and evaluate the performance of more environment-friendly refrigerants.

During recent years many studies have focused on evaluating hydrocarbon refrigerants due to their excellent characteristics. According to Domanski [19], Granryd [20], Thome et al. [21], Mohanraj et al. [22], and Copetti et al. [23] hydrocarbon refrigerants are attractive for use in various applications because these fluids have comparatively small molecular weight, are compatible with various lubricants and materials used commonly in refrigeration industries, and have very good transport and thermodynamic properties.

Isobutane (R600a) with ozone depletion potential (ODP) of zero and GWP of about 3, is an appropriate hydrocarbon refrigerant; and many investigations have been carried out using this fluid. Copetti et al. [23] experimentally assessed and compared the evaporative heat transfer rates and pressure drops of refrigerant R600a with those obtained by R134a. It was observed that employing R600a yields in higher rates of heat transfer and pressure drops. Their observations also showed that values of the pressure drop increased by augmenting the mass velocity and vapor quality. Shafaee et al. [24, 25] in empirical researches on condensation and evaporation of R600a, evaluated the thermal performance of smooth and rough pipes. Their study illuminated that using rough pipes increased both heat transfer rate and frictional pressure drops of R600a. Also, it was seen that under higher mass velocities and vapor qualities, higher values of the heat transfer coefficients were obtained, though the frictional pressure drops were also increased. Their following research on the flow patterns of R600a in smooth and rough pipes also showed that rough surfaces influenced the flow patterns and their transitions from annular regime to intermittent regime or vice versa as compared to the smooth tube [26]. Within another experimental study on evaporation of R600a in horizontally placed pipes, Yang et al. [27] reported that the refrigerant mass velocity and applied heat flux have significant impacts on the transitions of flow regimes from intermittent to annular; and the transition was experienced at a lower vapor quality with the increment of the mass velocity and applied heat flux. Within two other studies, Alimardani et al. [28, 29] explored flow boiling R600a in heat exchangers with coiled wire inserts under different refrigerant mass velocities ranged between 109.2 to 505 $kgm^{-2}s^{-1}$. Their studies showed that using wire coil inserts contributes to the enhancement of heat transfer, but a considerable growth in frictional pressure drop was also observed. It was concluded through their experiments that using coiled wire inserts was useful at higher values of mass velocities as the obtained performance factor was above one for higher mass velocities.

The present experimental research is carried out so as to illuminate the influences of utilizing twisted tape inserts on the values of heat transfer coefficients and frictional pressure drops of the environment-friendly refrigerant R600a during condensation in horizontal test tubes under various vapor qualities and mass velocities. This research aims to show the effectiveness of utilizing twisted tape inserts in two-phase flow tube heat exchangers by determining the system performance factor. It is worth noting that there is no research performed already to:

- Explore the heat transfer characteristics of R600a in horizontal pipes with twisted tapes.
- Explore the frictional pressure drops of R600a in horizontal pipes with twisted tapes.

In the following sections, firstly, the experimental procedure will be explained. At the next step, the obtained data for the smooth pipe will be compared with previously developed correlations. Then, the impacts of using inserts will be explored and the obtained data using inserts will be compared to those obtained by the smooth pipe. Finally, the two-phase flow regimes will be discussed.

2. Experimental procedure

To perform the tests, an empirical setup (as shown in Fig. 1) is constructed by assembling the components including condenser test section, heater, gear pumps, flow meters, sensors for measuring temperature and pressures, differential pressure drop transducer, and reservoir. The test section is a horizontal pipe constructed from copper with the inner diameter of 8.1 mm, thickness of 0.71mm, and length of 1000 mm. The examined test part is a counter-flow coaxial double-pipe heat exchanger. The cycle is comprised of two loops. Refrigerant R600a flows in the main loop, while the coolant water flows in the second loop. In each loop, a variable frequency gear pump is installed for circulation of water and R600a. Similarly, a flow meter is installed in each loop to measure the mass flow of the refrigerant and coolant water. Temperature and pressure sensors are attached to the cycle in different positions to capture the data. Two electrical heaters are used to attain the desired vapor quality at the inlet of the examined condenser. Furthermore, insulation is used to lower the amount of heat loss to the surrounding environment.

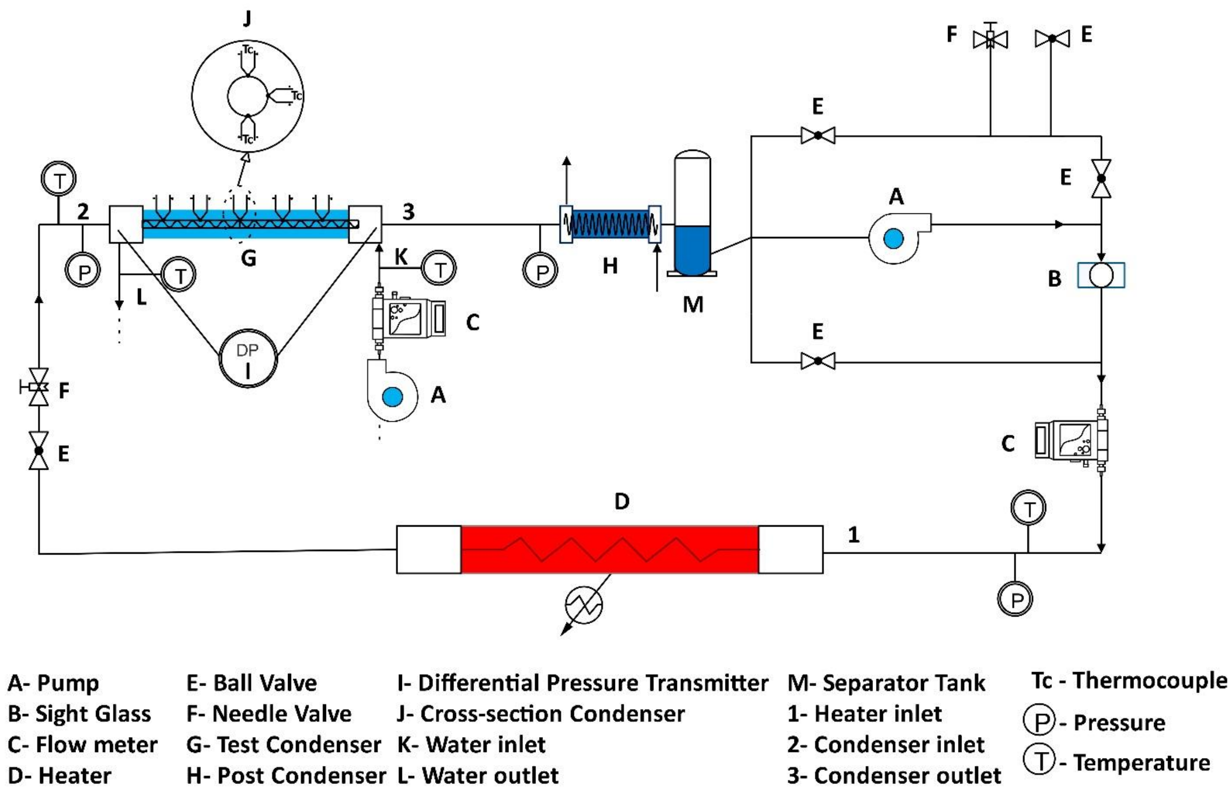


Fig. 1. Schematic view of the present experimental rest rig.

To record temperature values along the test section, thermocouples (T-type) are attached in five different axial positions with a distance of 200 mm from each other. In each position, three sensors are attached to the bottom, top, and side of the examined tube. Also, a data logger is used for recording data. A post condenser and a reservoir are installed after the test section to ensure that liquid refrigerant enters the pump. Further information regarding the instruments used in the setup is presented in Table 1.

Table 1. The employed instruments in the current experimental setup.

Instrument	Model	Precision
Differential pressure drop transducer	PDM 75	0.075% of full scale
Temperature sensor (attached to test tube)	T-type	0.1 ^{o}C
Temperature sensor (inlet and outlet)	RTD PT 100	0.1 ^{o}C

Gear pump	ZDF, Czech	-
Data logger	Lutron 4208 SD	-
Pressure gauge	EN 837-1 Wika	1kPa
Coriolis mass flow meter	Mass/2100/6000 Danfoss, Denmark	0.1% of full scale

To assess the influences of using twisted tapes on the performance of tube heat exchangers, three inserts with various twist ratios of 4, 10, and 15 are installed. The twist ratio is a dimensionless parameter which is defined as the ratio of insert pitch to pipe internal diameter. Table 2 illustrates the physical dimensions of the utilized inserts. Furthermore, Fig. 2 depicts the schematic view of twisted tape inserts.

Table 2. Physical dimensions of the examined pipes and inserts.

Tube set	Twist ratio	Tube inner diameter (mm)
Smooth	-	8.1
TT1	15	8.1
TT2	10	8.1
TT3	4	8.1

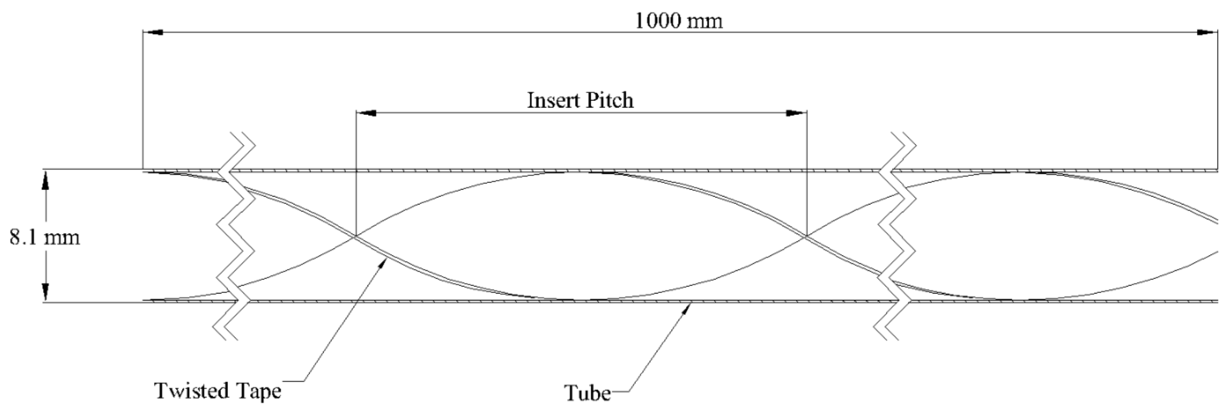


Fig. 2. Schematic view of the examined pipe with twisted tape insert.

To determine the uncertainty of the obtained heat transfer coefficients and frictional pressure drops, a methodology suggested by Schultz and Cole [30] is used. Their proposed relation for determining the uncertainty of the data is:

$$U_R = \left[\sum_{i=1}^{k} \left(\frac{\partial R}{\partial V_i} U_{V_i} \right)^2 \right]^{0.5} \tag{1}$$

Where U_R shows the estimated uncertainty in the obtained amount of the desired variable R, due to the independent uncertainty U_{V_i} in the primary measurement of n number of variables, V_i [24].

Table 3 presents the obtained uncertainties for the current data.

Table 3. The uncertainties of the computed parameters.

Parameter	Uncertainty (%)
Vapor quality	±6.4
Heat transfer coefficient	±8
Frictional pressure drop	±7.1
Refrigerant mass velocity	±16.9
Performance factor	±14

The experiments have been conducted under four mass velocities and different vapor qualities. The range of operating conditions of the current experiments is presented in Table 4.

Table 4. The operating parameters of the present experimental investigation.

Parameter	Type or value	Unit
Refrigerant	Isobutane (R600a)	-
Vapor quality	0.1-0.7	-
Refrigerant mass velocity	119, 162, 264, and 367	$Kgm^{-2}s^{-1}$
Average pressure	5.5	bar
Saturation temperature	41.4	^{o}C

3. Data reduction

3.1. Calculation of vapor quality

The vapor quality of the evaluated condenser is computed based on the mean of vapor qualities at its inlet and outlet. Considering the view of the cycle in Fig. 1, the vapor quality at the inlet of the condenser is computed as the following:

$$\dot{Q}_{eva} = \dot{m}_{ref}(h_2 - h_1) = \dot{m}_{ref}(h_{f2} + x_2 h_{fg2} - h_1) \tag{2}$$

$$x_2 = \frac{1}{h_{fg2}}[h_1 - h_{f2} + \frac{\dot{Q}_{eva}}{\dot{m}_{ref}}] \quad (3)$$

Where $\dot{Q}_{eva}$, $\dot{m}_{ref}$, h and x_2 show the provided heat rate by heaters, refrigerant mass flow rate, refrigerant enthalpy, and the vapor quality at the inlet of the condenser, respectively. Furthermore, $\dot{Q}_{eva}$ is calculated as follows:

$$\dot{Q}_{eva} = VI \quad (4)$$

Where I and V illustrate the electric current and voltage, respectively.

Furthermore, through an energy balance along the investigated tube and considering the removed heat from the refrigerant by coolant water, the vapor quality at the outlet of the condenser is determined according to Eq. (5):

$$x_3 = \frac{1}{h_{fg3}}[h_{f2} + x_2 h_{fg2} - h_{f3} - \frac{\dot{m}_{wat} C_{P,wat} \Delta T_{wat}}{\dot{m}_{ref}}] \quad (5)$$

In above-mentioned equation, $\dot{m}_{wat}$, $C_{P,wat}$, and ΔT_{wat} are coolant water mass flow rate, coolant water specific heat, and the difference in the temperatures of coolant water at inlet and outlet. Subscripts 1, 2, and 3 are related to the inlet of the evaporator, inlet of the condenser, and outlet of the condenser, respectively.

Finally, the vapor quality of the investigated pipe is obtained as follows:

$$x = \frac{x_3 + x_2}{2} \quad (6)$$

3.2. Calculation of frictional pressure drops

The total pressure drop along the test pipe is determined using the differential pressure drop transducer which is comprised of three parts including momentum, frictional, and static pressure drop. The value of the latter part is zero as the test pipe is installed horizontally. Therefore:

$$\Delta P_{tot} = \Delta P_{mom} + \Delta P_{fri} \quad (7)$$

The momentum pressure drop is computed based on the suggested relation by Collier and Thome [31]:

$$\Delta P_{mom} = G_{tot}^2 \left\{ \left[\frac{(1-x)^2}{\rho_l(1-\varepsilon)} + \frac{x^2}{\rho_g \varepsilon} \right]_{out} - \left[\frac{(1-x)^2}{\rho_l(1-\varepsilon)} + \frac{x^2}{\rho_g \varepsilon} \right]_{in} \right\} \quad (8)$$

In Eq. (8), G, ε, x, and ρ are mass velocity, void fraction, vapor quality, and density, respectively.

The void fraction is computed employing the proposed formula by Zivi [32]:

$$\varepsilon = \frac{1}{1 + (\frac{1-x_{tc}}{x_{tc}})(\frac{\rho_g}{\rho_l})^{2/3}} \quad (9)$$

Therefore, having read the overall pressure drop from differential pressure drop transducer and by computing the momentum pressure drop from Eq. (8), the frictional pressure drop is determined based on Eq. (7).

3.3. Calculation of heat transfer coefficient (HTC)

To calculate the quasi-local heat transfer coefficients of the test condenser, the following relation is used [33]:

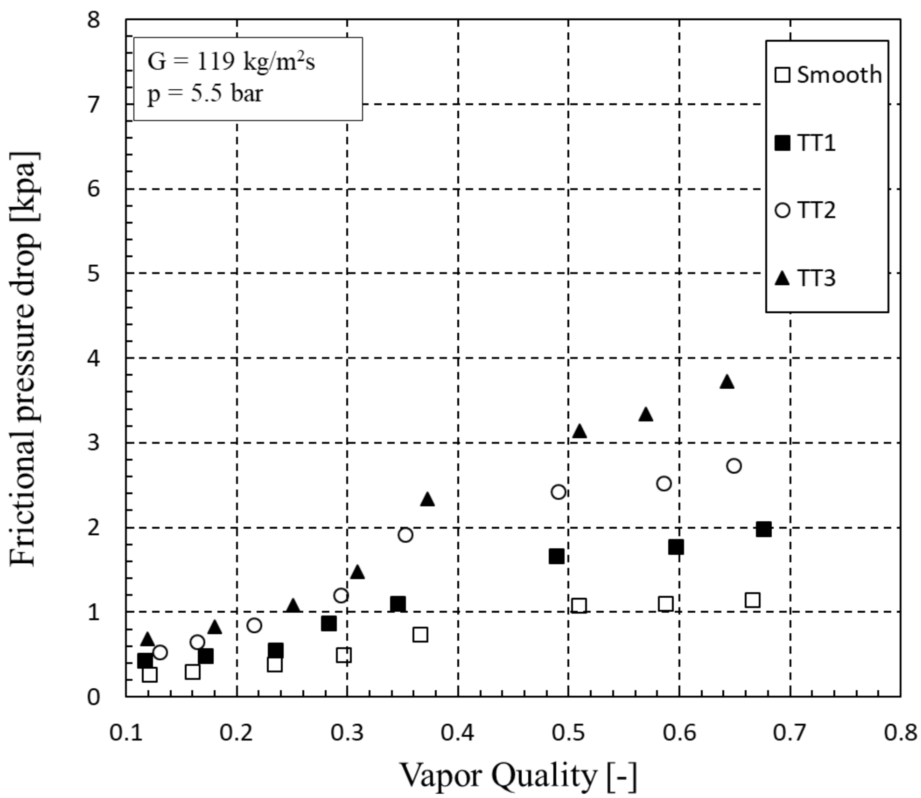
$$HTC = \left[\frac{\pi D_i L(T_{sat} - T_{wall})}{\dot{m}_w C_{p,w}(T_{w,out} - T_{w,in})} - \frac{D_i}{2k} ln\left(\frac{D_o}{D_i}\right) \right]^{-1} \quad (10)$$

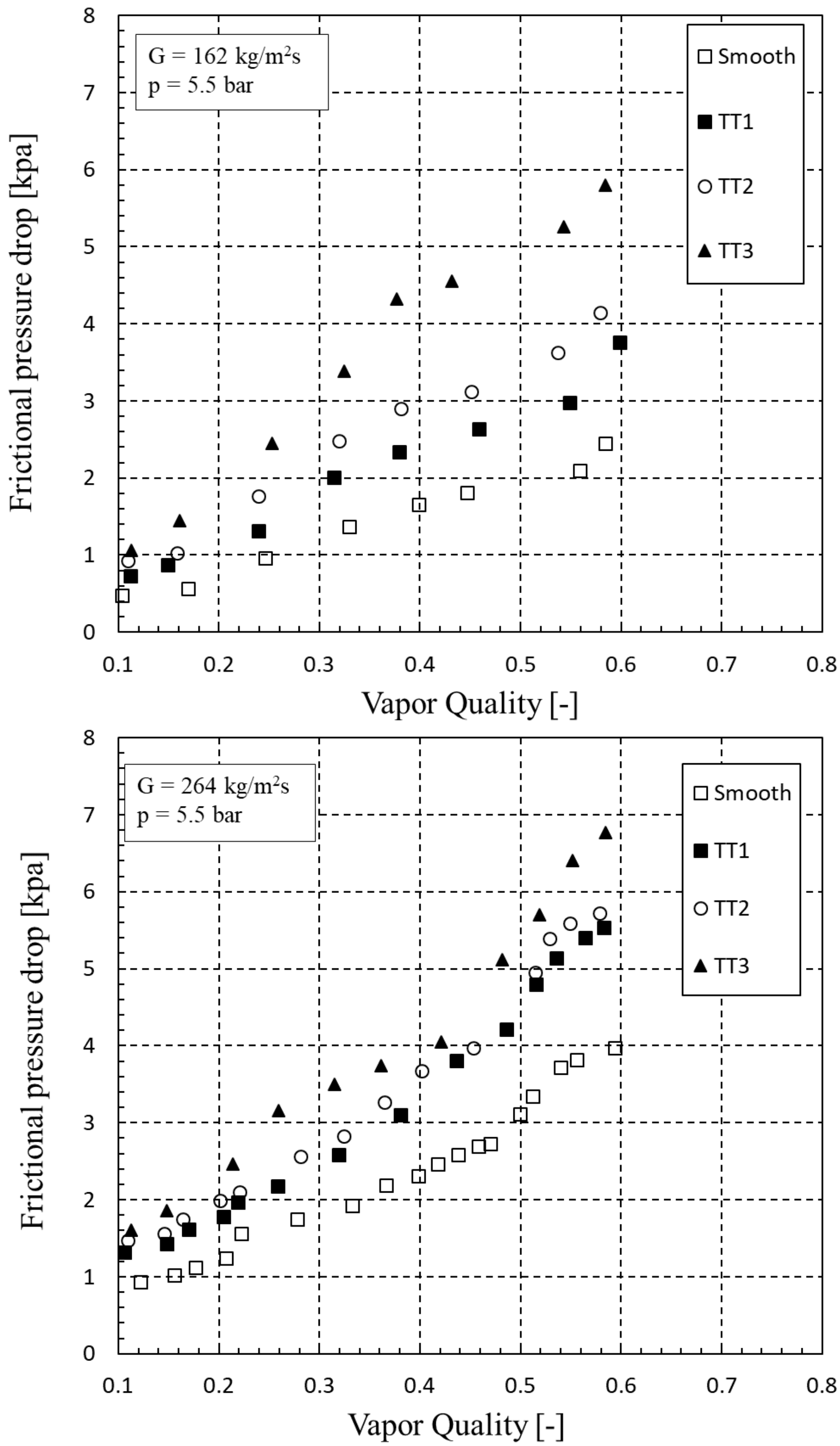
Where L, D_o, and D_i represent the test tube length, tube outer diameter, and tube inner diameter, respectively. T_{sat} shows the saturation temperature of R600a associated with its pressure. T_{wall} shows the tube wall temperature. T_w illustrates the water temperature. Finally, k represents the thermal conductivity of the investigated pipe.

4. Results

4.1. Pressure drops

Fig. 3 depicts the distribution of frictional pressure drops of refrigerant R600a during condensation in the plain pipe and rough pipes for different operating conditions. Based on the obtained results, it is deduced that the values of pressure drop augment as the vapor quality and mass velocity grow. As it was discussed in previous investigations [14, 34] by increasing the value of vapor quality, the volume of vapor within the channel increases leading to a higher vapor velocity. By increasing the vapor velocity, a larger difference between the velocity of vapor refrigerant and liquid refrigerant is generated which augments the shear stress between the two phases. As a consequence, the pressure drops rise. Besides, by growing the mass velocity, the velocity of flow inside the channel increases. Therefore, a larger shear stress is produced between the flow and tube inner wall which results in higher pressure drops.




G = 162 kg/m²s
p = 5.5 bar
Smooth
TT1
TT2
TT3
Frictional pressure drop [kpa]
Vapor Quality [-]
G = 264 kg/m²s
p = 5.5 bar
Smooth
TT1
TT2
TT3
Frictional pressure drop [kpa]
Vapor Quality [-]

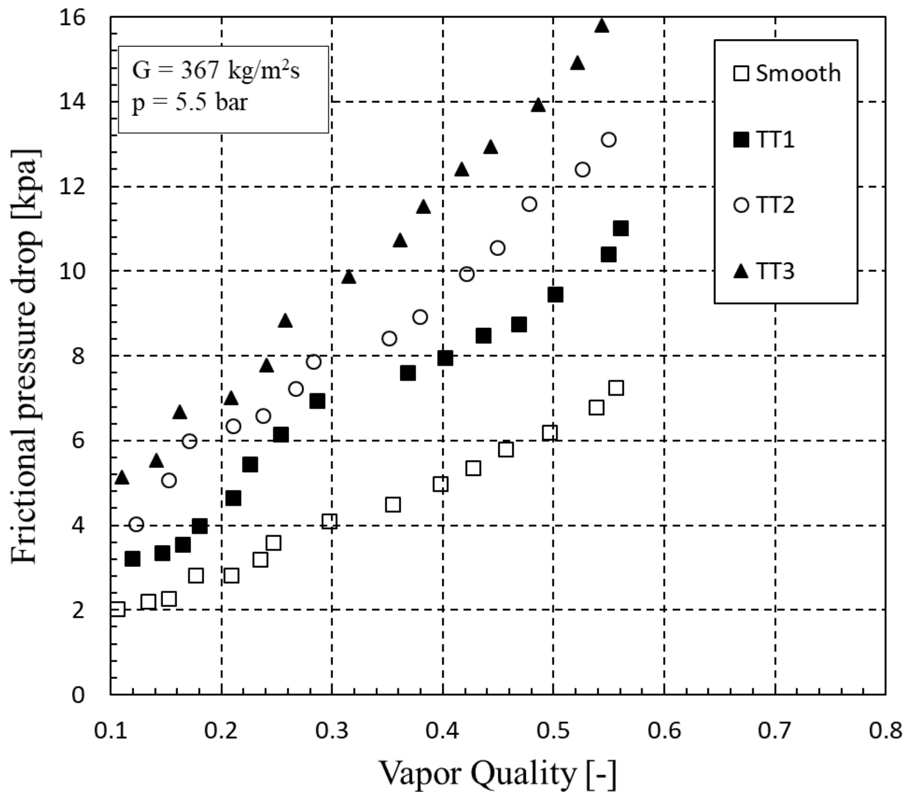


Fig. 3. Variations in the values of the obtained frictional pressure drops under different operating conditions for different test tubes (TT1, TT2, and TT3 are associated with twist ratios of 15, 10, and 4, respectively).

To verify the consistency of the pressure drop results obtained by the smooth pipe, the empirical results are compared to those attained by previously developed correlations. For this purpose, the correlations introduced by Friedel [35], Chisholm [36], and Muller-Steinhagen and Heck [37] are assessed. As can be observed in Fig. 4, the relation of Muller-Steinhagen and Heck [37] is able to predict the pressure drop data with a higher accuracy in comparison to the other correlations. It is worthy of attention that the average deviations (AD) and the average absolute deviations (AAD) of current pressure drop data from the relation of Muller-Steinhagen and Heck [37] are -3.23% and 11.57%, respectively.

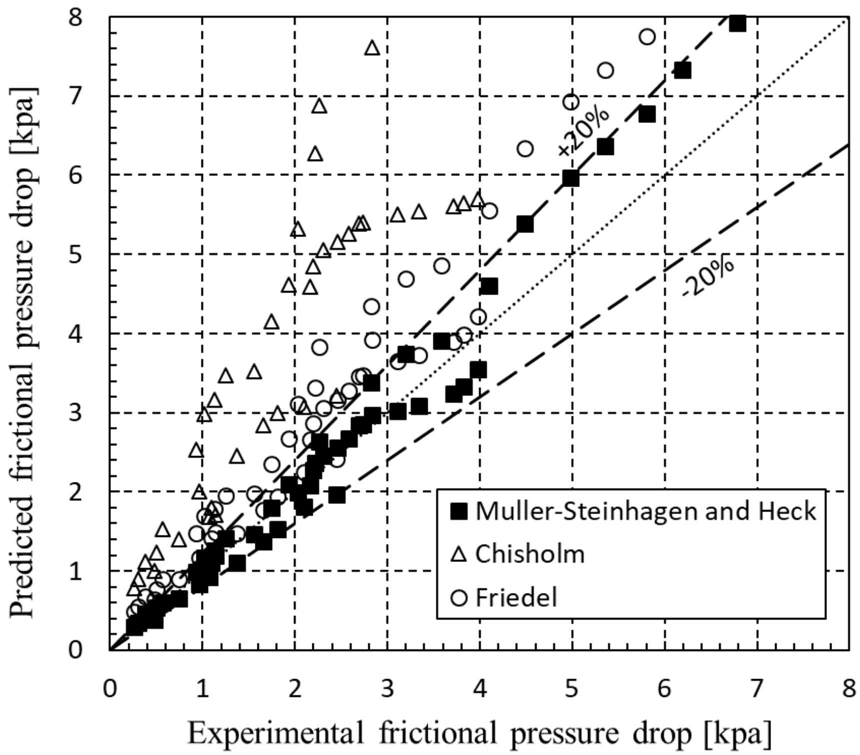


Fig. 4. Deviations of the current empirical pressure drop data from the correlations of Muller-Steinhagen and Heck [37], Chisholm [36], and Friedel [35].

As observed in Fig. 3, inserting twisted tapes in the test section results in the increment of the pressure drops. The first reason for this trend is that by utilizing the inserts, frictional surface inside the tube increases. Therefore, the pressure drop augments. The second reason is that using twisted tapes reduces the cross-sectional area for the fluid flow. Consequently, the flow velocity increases which in turn augments the losses. Furthermore, utilizing twisted tapes promotes the flow turbulence in both vapor and liquid phases. Hence, the shear stress between the refrigerant flow and the inner surface of tube and surface of inserts increases. The value of frictional pressure drop is proportional to the shear stress between the flow and surfaces. Therefore, the pressure drops increase as the shear stress increases.

However, it is considerable that the value of pressure drops growth is a function of geometrical dimensions of the inserts. According to Fig 3, for any given mass velocity and vapor quality larger pressure drops are imposed when an instrument with a smaller twist ratio is used. Indeed, when a twisted tape with a smaller twist ratio or pitch is used, the frictional surface per length of the test section augments which results in larger losses. Furthermore, using inserts with a smaller pitch promotes the turbulence level of the flow more as comparted to an insert with a larger pitch. Therefore, the pressure drop increases.

Among the instruments used, the tube set TT3 with the twist ratio of 4 results in the highest pressure drop growth of 325% over the smooth test section at the mass velocity of 119 $kgm^{-2}s^{-1}$. On the contrary, the lowest pressure drop growth over the smooth case is 125% which is related to the pipe TT1 at the mass velocity of 264 $kgm^{-2}s^{-1}$.

It is noteworthy that the rate of pressure drop increase for any given insert is a function of mass velocity. Fig. 5 depicts the ratio of pressure drop (PDR) of the twisted tape installed condenser to the plain condenser. It is seen that the influences of twisted tapes on the growth of pressure drop is much considerable under lower mass velocities. By augmenting the refrigerant mass velocity from 119 to 264 $kgm^{-2}s^{-1}$, the impact of twisted tapes fades as the impacts of the flow velocity and the momentum pressure drops are much considerable. These results approve the studies of Naulboonrueng et al. [38] and Alimardani et al. [28] who stated that effects of using modified surfaces and inserts at lower mass velocities on the pressure drop increment is much considerable. However, the pressure drops increase by increment of mass velocity beyond 264 $kgm^{-2}s^{-1}$. Indeed, at much higher mass velocities because of the augmented fluid velocity the shear stress between the working fluid and surfaces (i.e. tube inner surface and insert surface) augment resulting in higher pressure drops.

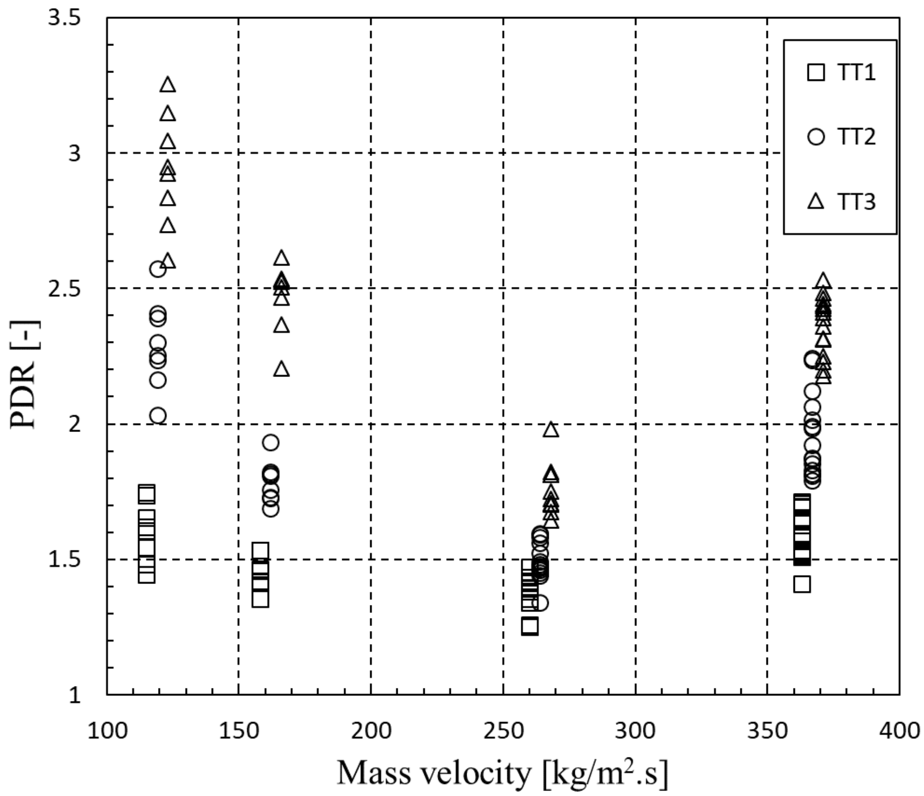
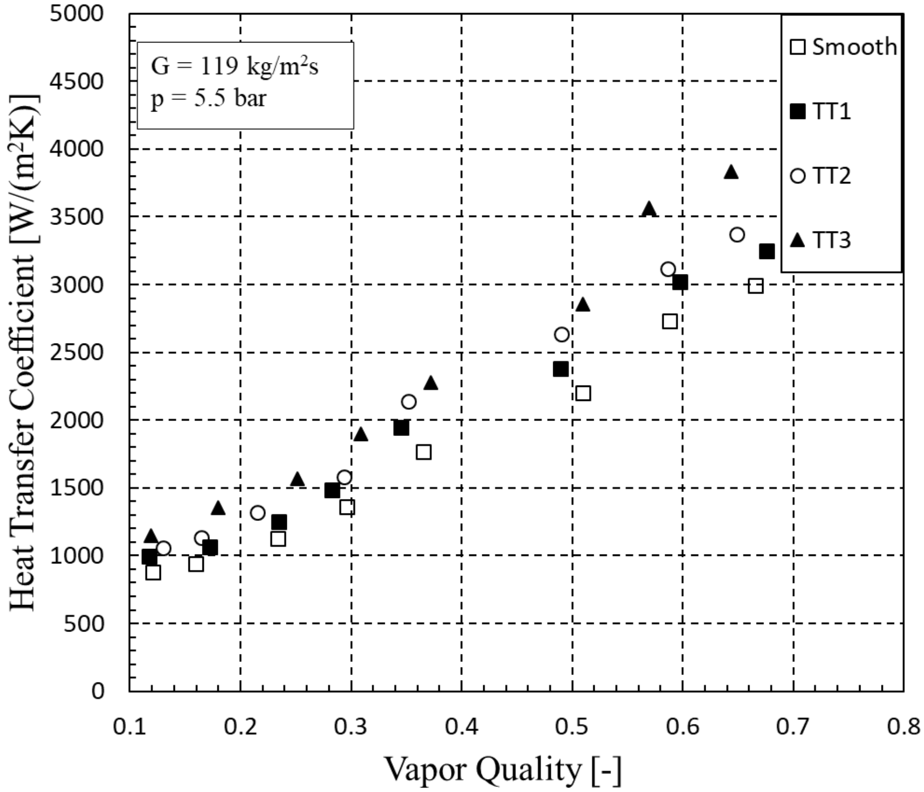
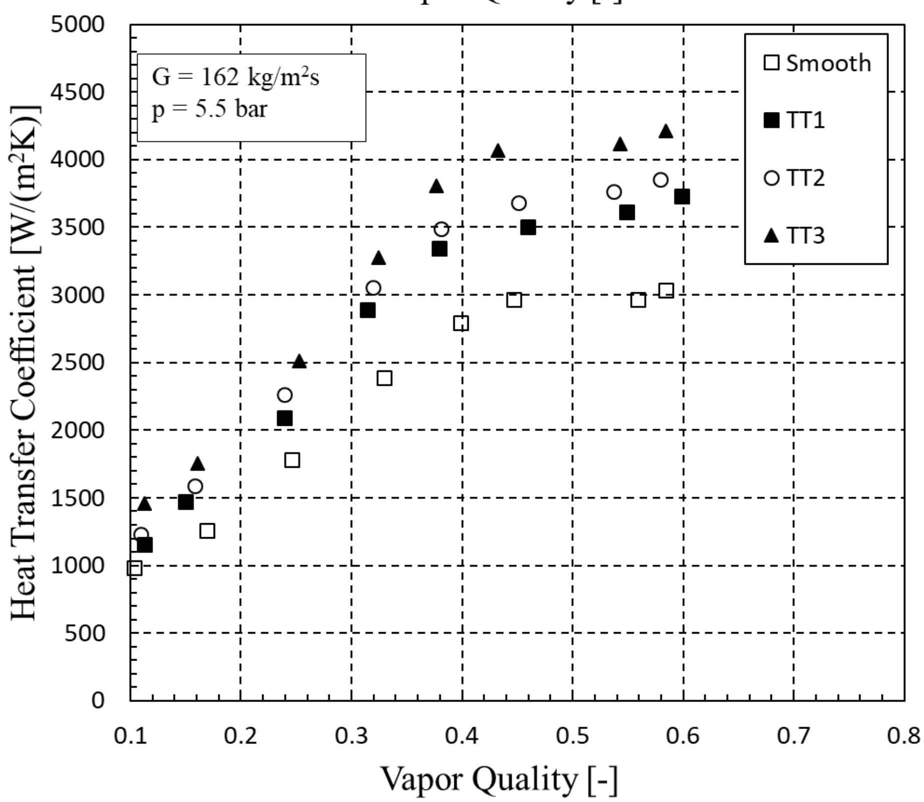


Fig. 5. Pressure drop ratios of twisted tape installed condensers to the plain condenser at mass velocities of 119, 162, 264, and 367 kg/m^{2}s, (please note that the horizontal position of data for TT1 and TT3 is changed for each mass velocity to avoid the overlapping of symbols).

4.2. Heat transfer coefficient

Fig. 6 is provided to illustrate the variations of HTC of R600a with changing vapor quality and mass velocity within the smooth and rough test cases. As can be seen, HTC grows by augmenting the vapor quality and mass velocity. When the amount of vapor quality is large, thinner liquid film is distributed on the inner surface of the test pipe. Therefore, the thermal resistance decreases at larger qualities which contributes to the higher rate of heat transfer. Furthermore, as the refrigerant mass velocity rises, the turbulence intensity of the refrigerant flow in both phases augments which in turn results in enhancement of HTC. Similar observations are reported in previous studies [14, 39].


G = 119 kg/m²s
p = 5.5 bar
Smooth
TT1
TT2
TT3
Heat Transfer Coefficient [W/(m²K)]
Vapor Quality [-]
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8


G = 162 kg/m²s
p = 5.5 bar
Smooth
TT1
TT2
TT3
Heat Transfer Coefficient [W/(m²K)]
Vapor Quality [-]
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8

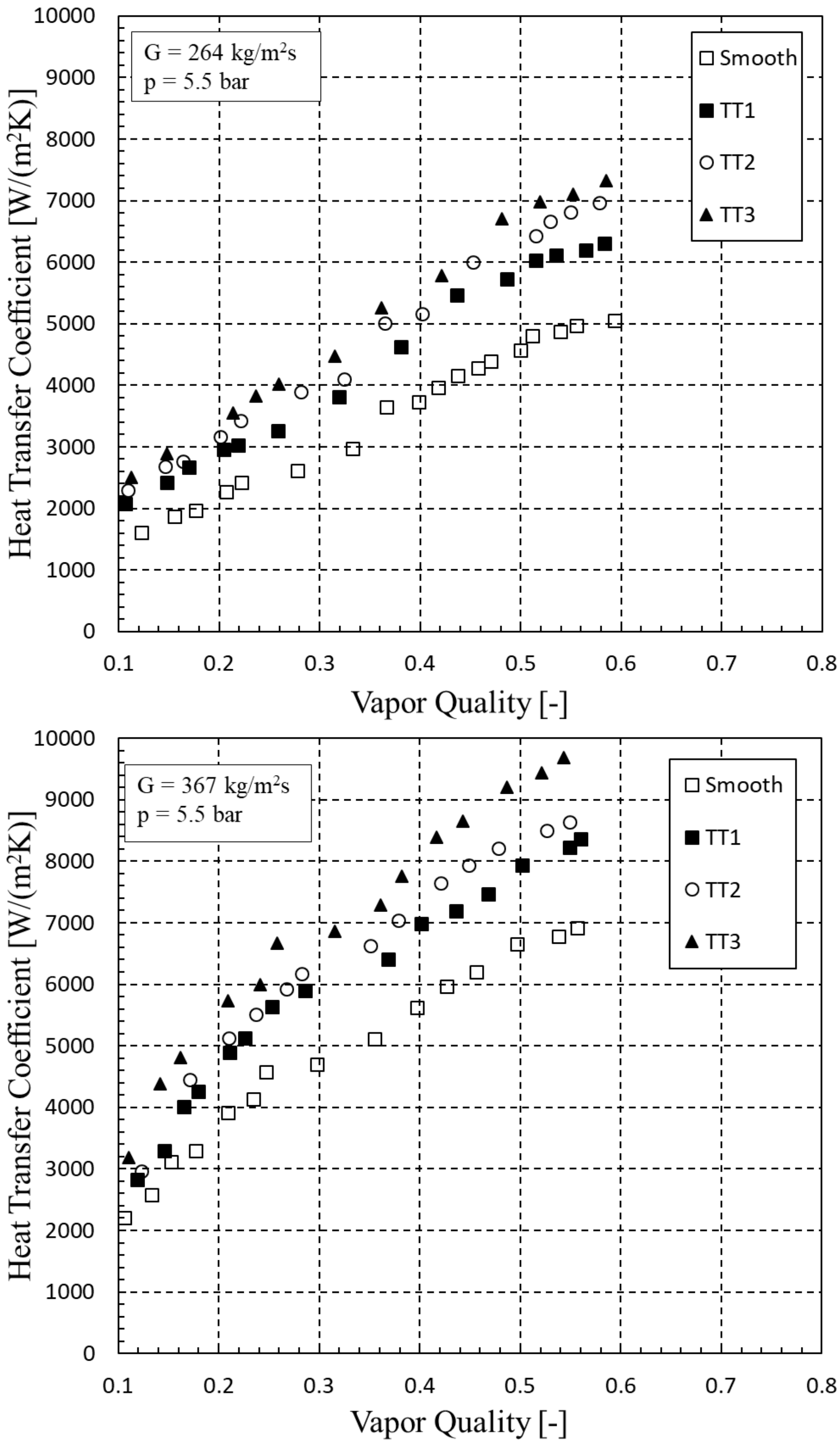


Fig. 6. Variations of heat transfer coefficients for different test tubes under different operating conditions.

It is worthy of attention that the refrigerant mass velocity plays a significant role in the thermal performance of system. Indeed, the two-phase flow patterns (which significantly influence the HTC) and their transitions are a complex function of various parameters including the refrigerant mass velocity and vapor quality. The flow regime transition from annular to stratified-wavy or intermittent is related to the interaction of body force and interfacial shear forces. As the refrigerant mass velocity grows, the role of forced convection becomes more evident and interfacial shear forces dominate the body force which contributes to the formation of annular flow regime which is able to deliver higher rates of heat transfer [34]. More discussions regarding the flow regimes will be provided later in the current research.

To assess to the consistency of the obtained heat transfer results, the plain condenser data are compared to those achieved by correlations of Cavallini et al. [40] and Thome et al. [41]. Fig. 7 shows that these relations are able to predict most of the HTC data in an error range of -20% to +30%.

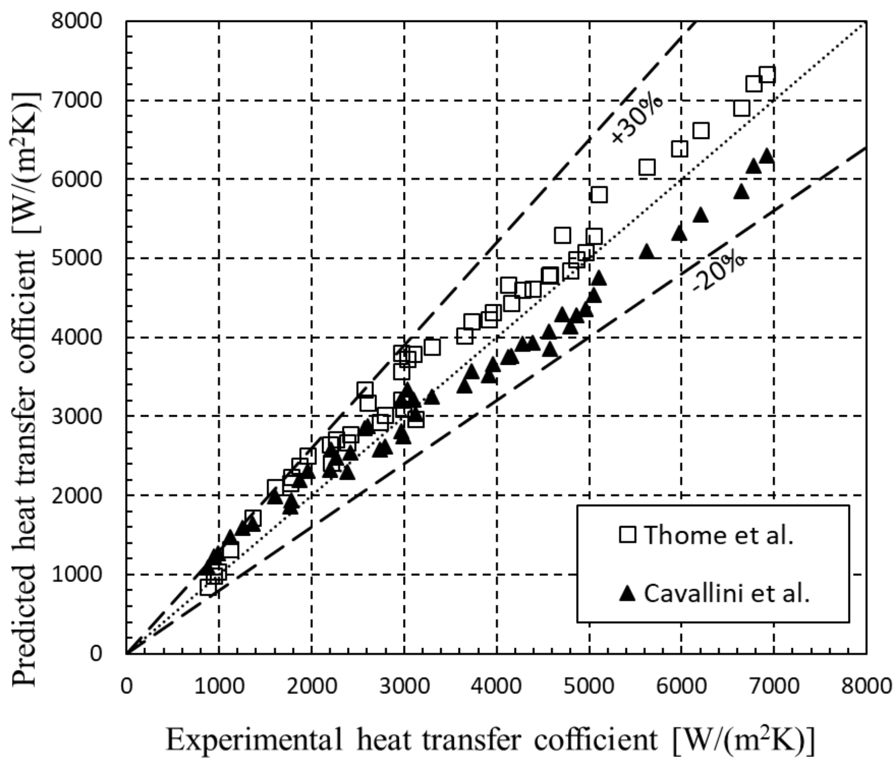


Fig. 7. Deviations of current empirical heat transfer data from the correlations of Thome et al. [41] and Cavallini et al. [40].

The impacts of installing twisted tapes on HTC is also illustrated in Fig. 6. That is seen that for all operating conditions using twisted tapes improves the heat transfer rate. The reason for this behavior is that the insertion of twisted tapes reduces the cross-sectional area of the examined pipe. Therefore, it is deducible that the flow velocity rises which contributes to the higher rate of convection. Besides, at lower mass velocities (which gravitational force dominates the interfacial forces) or lower vapor qualities (which the portion of liquid is larger), the liquid refrigerant tends to accumulate bottom of the tube, but twisted tapes prevent from accumulation of liquid refrigerant by providing a swirling flow and entraining it into the main stream.

It is observable from Fig. 6 that employing inserts with a smaller twist ratio gives more rise to the HTC. The highest increment of the HTC is obtained using tube set TT3 with the lowest pitch. Inserts with a smaller pitch are able to generate turbulent flow with more intensity which means higher heat transfer rate. The largest HTC of 9682 $Wm^{-2}K^{-1}$ is obtained using TT3 at the mass velocity of 367 $kgm^{2}s^{-1}$ and vapor quality of 0.54.

4.3. Performance factor

According to the results in previous sections, it is observed that implementing twisted tapes results in the augmentation of heat transfer rate and pressure drops. Therefore, to reach a conclusion regarding the usefulness of twisted tape inserts in two-phase flow heat exchangers, a criterion is required. In this regard, a parameter entitled performance factor (PF) is introduced which is as follows [24, 25]:

$$PF = \frac{({}^{W}/_{HTC})_s}{({}^{W}/_{HTC})_r} = \frac{{}^{HTC_r}/_{HTC_s}}{{}^{(\Delta P)_r}/_{(\Delta P)_s}} = {}^{R_{HTC}}/_{R_{\Delta P}} \tag{11}$$

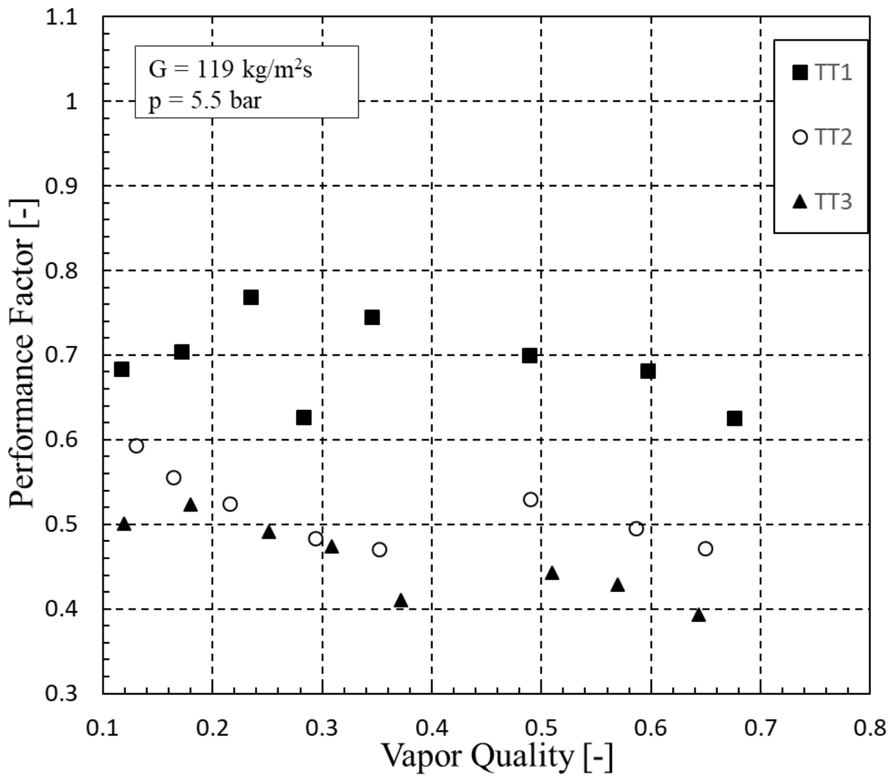
In Eq. (11), the ratio of power used by pump to the HTC is considered for the cases of smooth pipe, $({}^{W}/_{HTC})_s$, and rough or twisted tape inserted pipes, $({}^{W}/_{HTC})_r$.

This criterion was first suggested by Agrawal and Varma [42]. To determine the power that pump consumes, the following relation is used:

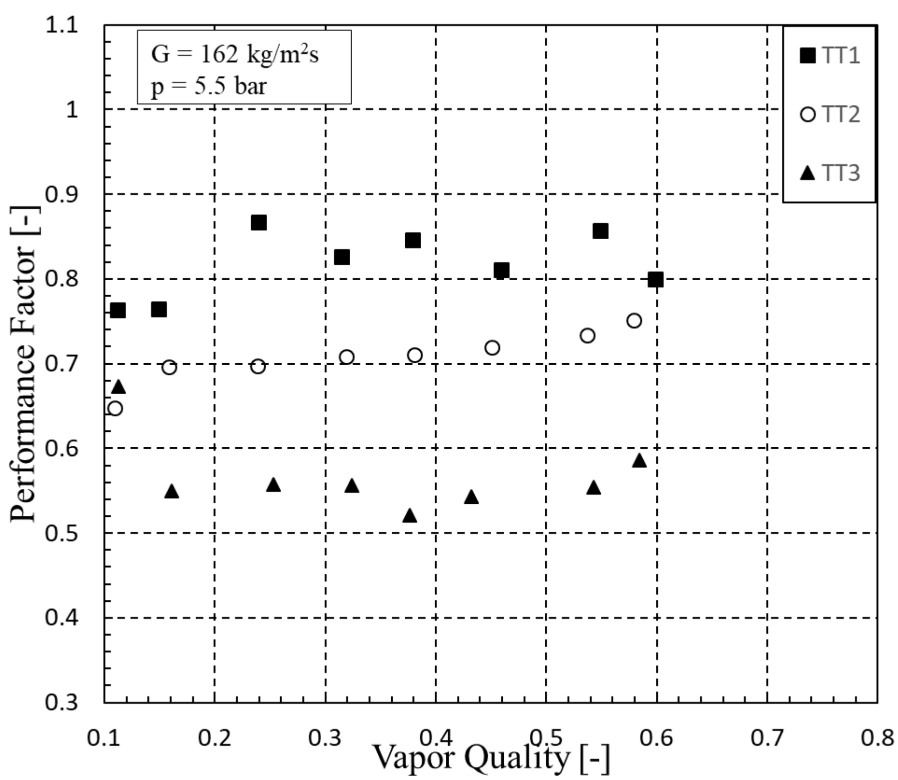
$$W = \dot{v}\Delta P \tag{12}$$

Where $\dot{v}$ demonstrates the volumetric flow rate. Also, ΔP represents the pressure drop within the pump. To calculate the performance factor using Eq. (11), the ratio of pump consumed power to HTC is computed for both smooth and rough test cases. Then, their ratio is calculated. If their ratio (PF) is larger than one, it is assumed that utilizing twisted tapes is beneficial. Otherwise, using these instruments is not recommended.

Fig. 8 shows the changes in the values of performance factor for all inserts under various mass velocities and vapor qualities. The results offer that generally utilizing twisted tapes at larger mass velocities is useful as PF augments. Based on the obtained data, there exists an optimum value of the mass velocity (264 kgm^2s^{-1}) at which the PF is higher. The reason is that for the larger values of the mass velocities, the influences of flow velocity and momentum pressure drops are dominant. As it was discussed earlier regarding Fig. 9, the pressure drop ratios of the twisted tape installed pipes to the smooth pipe at lower mass velocities are higher than that of at higher mass velocities. On the other hand, the inserts still give a considerable rise to HTC at higher mass velocities. Therefore, it can be concluded that at higher mass velocities, the HTC enhancement can overweight the pressure drop increment resulting in performance factors higher than one. Previous studies also approve the current conclusions [28, 38]. On the other hand, by augmenting the mass velocity beyond the optimum value the pressure drops and PF slightly decrease because the impacts of shear stresses between the fluid and frictional surfaces become stronger.



G = 119 kg/m²s
p = 5.5 bar
TT1
TT2
TT3
Performance Factor [-]
Vapor Quality [-]
1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8



G = 162 kg/m²s
p = 5.5 bar
TT1
TT2
TT3
Performance Factor [-]
Vapor Quality [-]
1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8

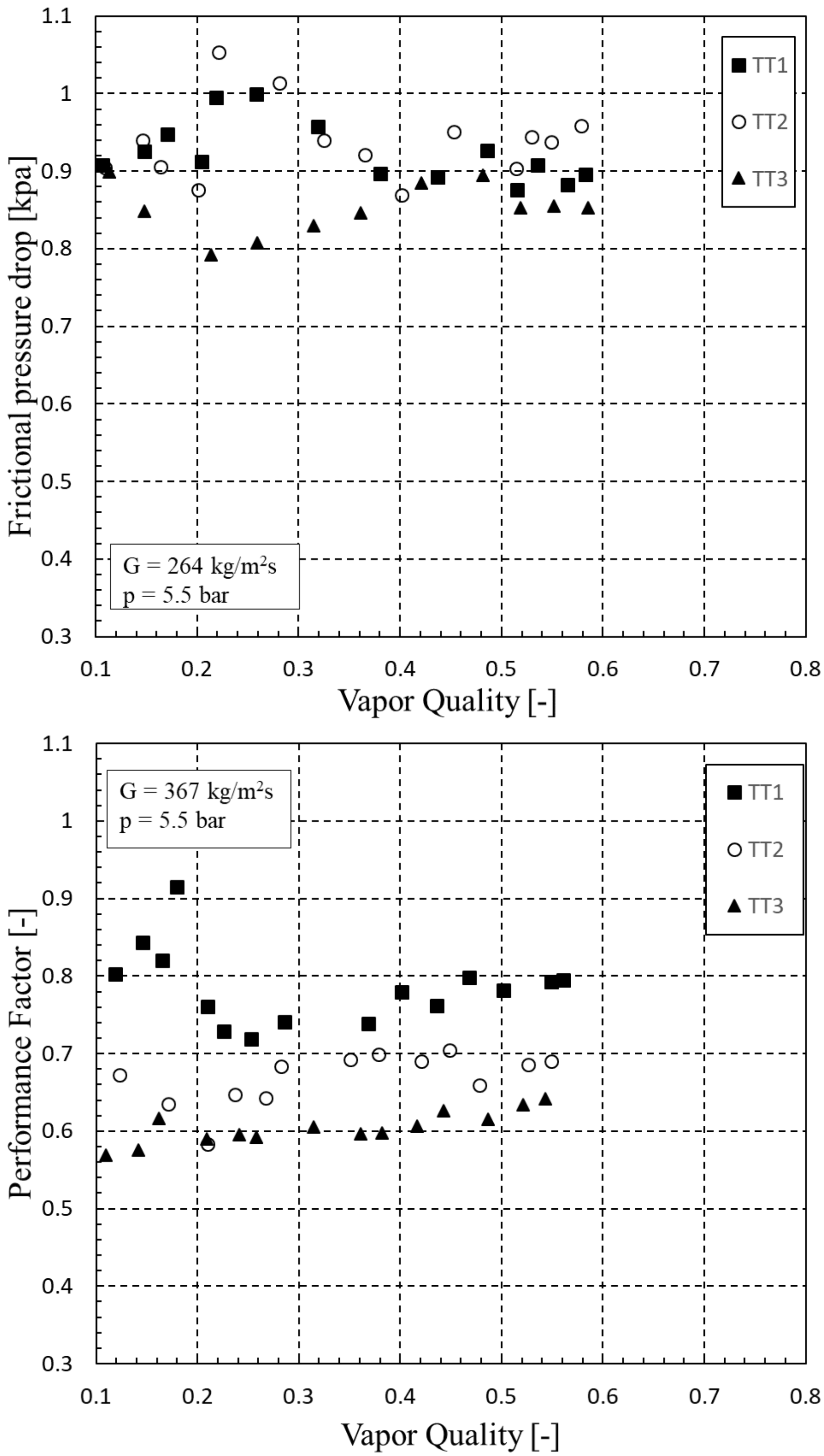


Fig. 8. Variations in the values of the performance factors with vapor quality for different inserts and mass velocities.

According to Fig. 8, it is observed that generally using twisted tapes degrade the performance of the system. Among the insert employed, generally the tube set TT1 results in higher performance factors.

According to the obtained performance factors, it is deduced that generally utilizing twisted tapes is not recommended unless the enhancement of heat transfer rate is the main concern, the increased pump consumed power is justifiable, or a compact heat exchanger is required.

4.4. Flow regime

To determine the flow pattern regimes of R600a condensation within the smooth channel, the presented map by El Hajal et al. [43] is used. According to Fig. 9 it is perceivable that the governing flow regimes of the current smooth test case are stratified wavy, intermittent, and annular. Generally, the refrigerant mass velocity and its interaction with gravitational force have a prominent role on the type of flow regime. At lower mass velocities, the gravitational force overcomes the interfacial shear forces between the liquid and vapor refrigerant. Therefore, the intermittent or stratified wavy flow regimes are observable. It should be noticed that under these conditions, i.e. at lower mass velocities, the refrigerant vapor quality also influences the flow pattern. At lower mass velocities, the flow regime changes from intermittent to stratified wavy by decrease of the vapor quality. On the other hand, the influence of gravity fades at higher mass velocities due to augmented interfacial shear forces. Furthermore, at higher mass velocities the impacts of vapor quality are less significant on the flow regime type.

Fig. 9 predicts that the governing flow regimes in the current research are stratified-wavy, intermittent, and annular which would justify the reason for comparatively good consistency between the current heat transfer data and correlation of Thome et al. [41]. As discussed by Thome et al. [41], their heat transfer correlation is suitable for annular, intermittent, stratified-wavy, fully stratified, and mist flow regimes. Therefore, that correlation can successfully predict the current heat transfer data with comparatively acceptable accuracy.

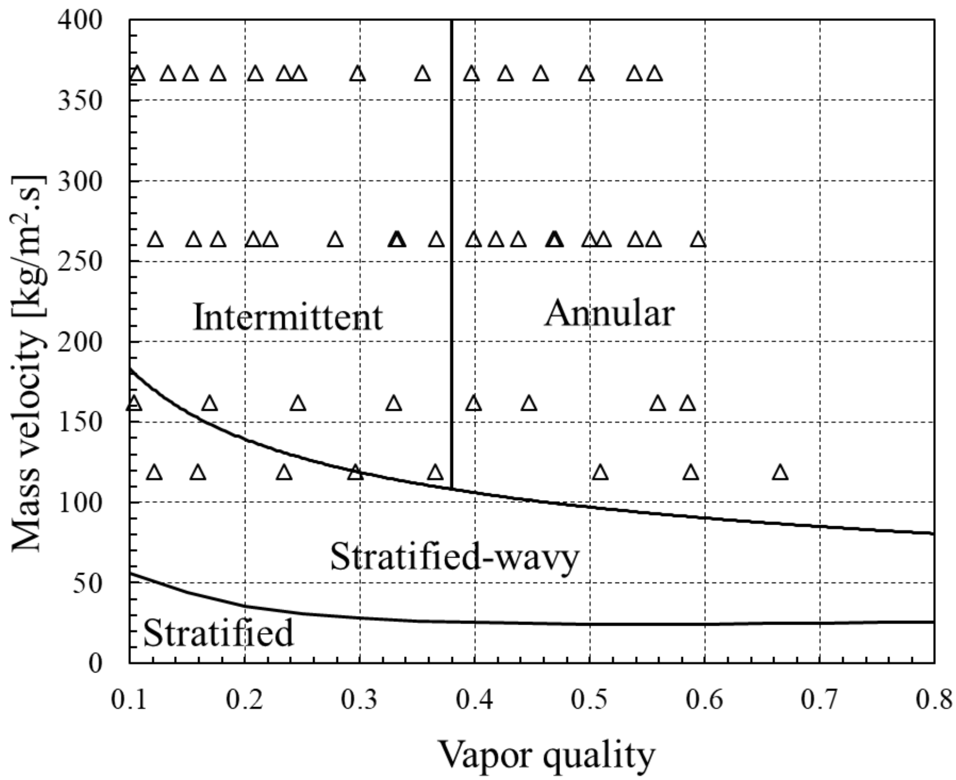


Fig. 9. Flow regime map of the current empirical data based on the proposed map by El Hajal et al. [43].

5. Conclusion

The current experimental research presents a discussion regarding the influences of implementing twisted tape inserts in two-phase flow tube heat exchangers. For this purpose, natural refrigerant R600a is used and vapor qualities within the range of 0.1 to 0.7 and mass velocities between 119 to 367 $kgm^{-2}s^{-1}$ are considered. Furthermore, three twisted tapes with twist ratios between 4 to15 are used. The main results are as the following:

- The correlation of Muller-Steinhagen and Heck [37] showed a better performance in predicting the pressure drop data.
- The correlations of Thome et al. [41] and Cavallini et al. [40] both showed a good performance in predicting the heat transfer data.
- Using inserts with lower twist ratios gives more rise to the values of pressure drops and heat transfer coefficients.

- Depending the operating conditions and the type of inserts, the values of performance factor varied between 0.39-1.05. The best performance was obtained using tube set TT2 with the twist ratio of 10.
- Obtained data demonstrated that there exists an optimum value of the mass velocity at which the performance factor is higher.

It is concluded through the results that using twisted tapes in two-phase flow heat exchangers results in different performances depending on the system operating conditions. Generally, considering the obtained values for the performance factor and its uncertainty amount, it can be concluded that employing twisted tapes does not improve the system performance as compared to the original smooth test case. Therefore, when the heat transfer augmentation is a major concern or when the employed power by pump can be justified, using twisted tapes is suggested.

6. Declaration of competing interests:

None.

7. References

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