International Journal of Recent Technology and Engineering (IJRTE)
ISSN: 2277-3878 (Online), Volume-8, Issue-1, May 2019

Heat transfer Enhancement against Void Fraction in Flow Boiling of R407C using Twisted Tapes

M. D. Hambarde, Ramakant Shrivastava

Abstract: ***Heat transfer enhancement with respect to void fraction (i.e. vapor quality) is investigated in a flow boiling of R407C using twisted tapes in a tubular evaporator of 13.40 mm inner diameter and 2m length. Following operating conditions are considered for investigations: (i) heat flux – 1.5 to 9.5 kWm^{-2}; (ii) mass flux - 150 to 300 kg s^{-1} m^{-2}; (iii) evaporating temperature ranging from -15^{0}C to +15^{0}C; (iv) average void fraction(vapor quality) -0.05 to 0.90. Heat Transfer enhancement due to twisted tapes is found to be varied with respect to void fraction (vapor quality) and operating conditions. 10 to 40% void fraction is found to be most suitable range for heat transfer enhancement for all operating conditions. Twisted tapes of twist ratio 12, 10 & 8 are compared for their heat transfer performance. Twisted tape of twist ratio 12 has shown maximum heat transfer enhancement.***

Index Terms: ***flow boiling, heat transfer enhancement, refrigerants, R407C, twisted tape, turbulent promoters.***

I. INTRODUCTION

Heat transfer augmentation techniques mostly involved turbulent promoters such as twisted tapes, coiled wire inserts, vortex generators *etc.* for heat transfer enhancement. The analyses of heat transfer coefficient variations with different geometries of turbulent promoters will helps to select a particular geometry for maximum heat transfer. In last six decades many investigators have carried out research on the heat transfer enhancement using turbulent promoters, such as twisted tapes, coiled wire inserts, vortex generators *etc.* Mostly investigations on heat transfer enhancement is carried out with air or water. Current work presents investigation on heat transfer enhancement in flow boiling of refrigerant, R407C, which is rare one.

K.N. Agrawal et al. [1], [2], [3] have carried out investigation on heat transfer enhancement during evaporation of R12, using twisted tapes. Investigation highlighted that heat transfer enhancement depends on test conditions and twist ratio. In the investigation it has been found that the ratio of heat transfer coefficient with twisted tape to heat transfer coefficient with plain tube per unit of pumping power was more than one, for most of the operating conditions of twisted tapes. Similar type of investigation is carried out by **M Cumo et al. [4]** on evaporation of R12 study, with effect of twisted tape on the evaporation of Freon 12 and found that the heat transfer rate was significantly improved due to swirl flow, up to 200% at dry out point. **Mark.A. Kedzierski and Min Soo KIM [5]** in investigation of effect of twisted tape inserts in condensation and boiling of pure and mix refrigerants mentioned that partial dry out of the inner tube surface was due twisted tape and such partial dry out was responsible for decrease of Nusselt number with increase in vapor quality. In 1993, **R. M. Manglik, A. E. Bergles** have published two studies, **[6],[7]** on heat transfer and pressure drop for laminar flow and transition and turbulent flows, using twisted tapes inserts in the flow of ethylene and water glycol and highlighted that heat transfer enhancement depends on flow rates and twist ratio. **Ramakant Shrivastava et al. [8]** investigated heat transfer enhancement in condensation of R-22 using three twisted tapes of twist ratio 6, 9 and 15. Investigation had shown, twisted tape with twist ratio 6 as a best heat transfer performer with 25% increase in heat transfer coefficient over plain tube values and maximum improvement in heat transfer coefficient was seen with full length twisted tape inserts. **H. Gül and D. Evin[9]** has used surface geometry in the form of helical tape, milled on the outer surface of the tube for heat transfer enhancement in a flow of water and found that large improvement in heat transfer coefficients in the downstream side of the tape. **[10[& [11]** have studied the effect of coiled wire inserts, helical screw-tape inserts on heat transfer augmentation and pressure drop. **P. Promvonge and S. Eiamsa-ard [12]** have tested combination of conical-ring and twisted-tape insert for heat transfer enhancement with air as working fluid and mentioned that with the combination of twisted tape of twist ratio 3.75 and conical ring, maximum 367 % heat transfer enhancement with 1.96 enhancement efficiency was achieved. **M.A. Akhavan-Behabadi et al. [13],[14]** in their investigation of heat transfer enhancement and pressure drop in evaporation of R134a, using twisted tapes and coiled wire inserts, highlighted that enhancement in heat transfer occurred at the higher penalty of pressure drop. Performance of inserts depends on geometry and test conditions. **Fabio Toshio Kanizawa and Gherhardt Ribatski [15]** have carried out the investigation on pressure drop and flow pattern during flow boiling of R134 with twisted tapes and found that frictional pressure drop decreases with increasing twist ratio and temperature.

Revised Manuscript Received on 22 May 2019.

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It was also mentioned that pressure drop increases with mass flux and void fraction (vapor quality). **M. Saeedinia, M.A. Akhavan-Behabadi and M. Nasr [16]** worked on coiled wire inserts with Nano Fluid flow and shown that coiled wire inserts with specific nanoparticle concentration increases pressure drop and heat transfer. **Taye Stephen Mogaji et al. [17]** have carried out experimental study on heat transfer enhancement and penalty of pressure drop, in flow boiling of R134a. Investigation highlighted that twisted tapes are profitable in high vapor quality region with mass flux more than 150 kg $m^{-2}s^{-1}$. **Mao-Yu Wen et al. [18]** have used dispersed copper porous inserts in a circular tube to investigate the heat transfer enhancement and corresponding pressure drop during flow boiling of R-600a. Investigation mainly highlighted that substantial improvement in heat transfer coefficient was seen with porous inserts as compared to plain tube, because porous material offered maximum nucleation sites and more heat transfer area as compared to plain tube.

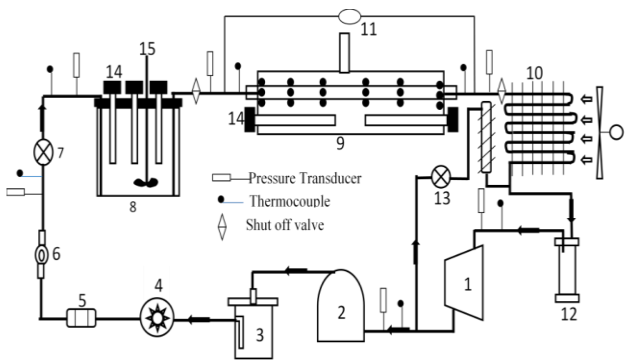
Maziar Shafaee et al. [19] have worked on study of heat transfer enhancement and flow pattern during evaporation of R-600 using coiled wire inserts of pitches 30, 20 and 10 mm. The investigation has mentioned that coiled wire inserts have increased heat transfer coefficient 1.24 times of plain tube values and coiled wire of pitch 10 mm and wire diameter 0.5 mm has shown maximum change in transition of vapor quality. **Mohammad Reza Salimpour et al. [20]** have studied heat transfer enhancement in condensation of R-404A, using coiled tubes in helical form with various coil pitches and coiled diameters or curvature radii. **Kanit Aroonrat et al. [21]** have used dimple tube to study the heat transfer enhancement and pressure drop in condensation of R-134a and found heat transfer enhancement with dimple tube at the cost higher pressure drop. **Zahid H. Ayub et al. [22]** have worked on the heat transfer enhancement with dimpled tube in flow boiling of R134a, with solid round rod as insert and found heat transfer rate three times of that plain tube without inserts, but at the higher penalty of pressure drop. Recently in 2018, **Anand Kumar Solanki and Ravi Kumar [23]** have experimentally compared heat transfer enhancement by helically coiled tube, helical coiled tube with dimple and a plain straight tube during condensation of R-134a.

A. Why R407C?

Refrigeration, Air conditioning and Heat pumps sectors are demanding new refrigerants having minimum ozone depletion and global warming potential. Due to excellent thermal performance, HCFC-22 or R-22 was the favorite refrigerant of HVAC sectors. But due to its high ozone depletion and global warming potential issues, use of R-22 is discouraged. Hence it becomes necessary to develop a substitute to R22 which will have thermal performance to the level of R-22 and which will be best retrofit to the existing refrigeration systems handling R-22 to avoid immediate obsoleting of them. Many refrigerants such as R-290, R-134a, R407C, R 410A *etc.* are developed as an alternatives to R22. According **Shailendra Kasera et al. [24]**, reactive analysis of R134a suggested that R134a would decompose in the presence of sunlight in the troposphere and would produce acid and poisonous products which would be worst situation than global warming and ozone depletion. High flammability of R-290 creates a safety issues in using it as alternative to R22. Comparing R407C with R410A, thermo-physical properties of R407C are close to R22 and hence shows thermal performance close to R22 and can be used in the existing refrigeration systems handling R-22 without making more changes in the design. Some research has been carried out to compare performance of R-407C with R-22. **C. Aprea al. [25]**, recommended R407C as possible substitute to R22. According to **Wang et al. [26]** investigation, heat transfer performance of R407C was considerably less than that of R22. **T.Y. Choi et al. [27]** mention that refrigerant mixtures considered in their investigation show poor thermal performance than its pure components. In the investigation of **C. Aprea et al. [28]**, R407C has shown performance 8 to 14% less than R22.**S. Devotta et al. [29]** have found COP of R22 as 2.57 and COP of R407C as 2.36 with outdoor conditions of 35^0C, T_{db} and 30^0C of T_{wb} was 2.57 **Devotta et al. [30]** have carried out experimentations for comparison of COP of R-22 with R-134a, R-290, R-407C and R-410 A and found that at evaporating temperature of 55^0C, R407C has shown COP 1.76% less than R-22. R Reports of United Nations Environment Programme (UNEP) **[31] & [32]** have mentioned that R-407C has a wide scope in refrigeration and air conditioning sector and as a best transitional refrigerant with little modification in the existing system using R-22 to use R-407C as a substitute.

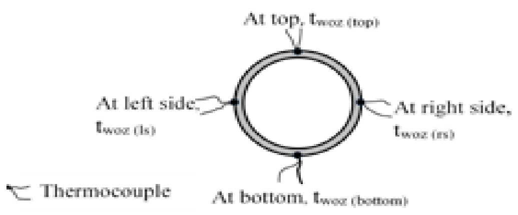
Thus, different researches on R407C indicate that R-407C is a promising substitute to R-22, but it shows an average thermal performance less than R-22 by 7 to 12 %. However, the thermal performance of R407 C can be improved through some enhancement techniques. Indeed, detail investigation on heat transfer enhancement in flow boiling of R 407C will bridge the gap of poor thermal performance of R 407 C as compared to R22. Hence in the present research work R-407C is selected as working substance for heat transfer enhancement during its evaporation using twisted tapes.

II. EXPERIMENTATION



Experimental setup as shown in figure 1, is a vapor compression refrigeration test-rig with an arrangement for variation of mass velocity, heat flux and evaporating pressure.

Experimental test facilities consist of refrigerant flow loop with flow meter, pre-evaporator, test- evaporator, after-evaporator, accumulator, compressor, bypass valve, condenser, receiver, sight glass and manual expansion valve as shown in figure 1. Pre-evaporator adjusts the quality of refrigerant as per requirement at entry of test-evaporator. Pre-Evaporator is fabricated in the form of stainless steel drum filled with water-glycol solution and cooper tube coil immersed in the solution. Three heating coils are immersed in the water –glycol solution of Pre-Evaporator to maintain the required void fraction (vapor quality) at entry to Test-Evaporator. Stirrer helps to maintain uniform heat flux condition in Pre-Evaporator. Solid state relay with PID controller is used to control heat input precisely in the pre-evaporator. Test-evaporator is a long cylindrical stainless-steel pipe, filled with water-glycol solution and with copper test tube passing through it. Copper test tube that passes through the test evaporator is of 2 m length and 13.40 mm of inner diameter. Two heating rods, 0.9 m each in length are fitted into the test evaporator, parallel to copper test tube as shown in figure 1. As two heating rods cover the entire length of test evaporator, uniform heat flux condition can be assured on copper test tube. Heat input in test-evaporator is



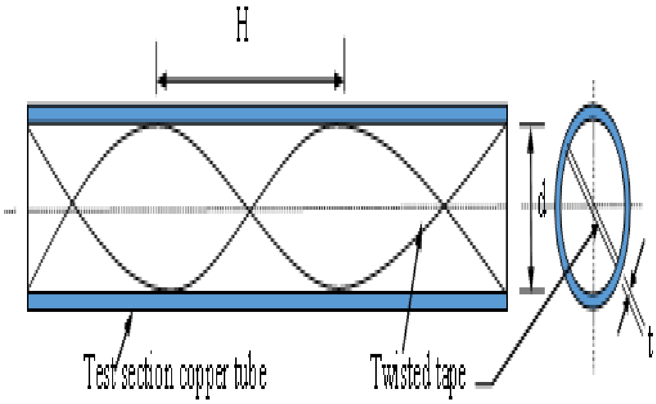
controlled through solid state relay controller. Voltmeter and ammeter are used to measure voltage and current corresponding to heat input in pre-evaporator and test-evaporator.

Thermocouples, separated by 90^0 are brazed circumferentially on the outer surface of test tube as shown in figure 2.There are six such locations of thermocouples, brazed on the test tube, equidistant from each other. Superheat of refrigerant at the entry to the compressor is adjusted by After –Evaporated, fitted on the downstream side of the Test-Evaporator. Flexible elastomeric nitrite foam rubber insulation is provided on outer surface of pre-evaporator and test-evaporator to minimize the heat leakage to atmosphere. The same insulation is provided on low pressure copper tube line to prevent frosting on its surface. The refrigerant leaving the pre-evaporator attains saturation condition with certain vapor quality according to heat input given in pre-evaporator. This will be the quality of refrigerant entering the test-evaporator and can vary according to heat input in pre-evaporator. Piezo-resistive absolute pressure transducers and thermocouples (T-type) are used to measure pressure and temperature of refrigerant at different locations as shown in figure 1. Piezo-resistive differential pressure transducer is used to measures pressure drop in the test-evaporator. Oval gear type flow meter measures mass flow rate of refrigerant. Table I shows accuracy and range of instruments used in the experimental set-up.

Table I: Measuring instruments with accuracy and range

Variable	Instrument	Accuracy	Range
Temperature	T-type thermocouple	± 0.375^0C	-40 ^{0}C to 150 ^{0}C
Pressure	Piezo Resistive	± 0.25 % FS	0 to 20 bar (Abs)
Differential Pressure	Piezo Resistive	± 0.1 % of URL	0 to 0.3 Bar
Mass flow rate	Oval gear-Positive displacement	± 0.15 % FSD	20 to 300 LPH

Figure 3 shows geometry of twisted tapes with pitch ‘H’, width ‘d’ and thickness ‘t’. Geometrical configuration values of twisted tapes, used in the present investigation are shown in the table II. During experimentation twisted tapes of twist ratio 8, 10 & 12 as shown in figure 4 are inserted in to the test



–evaporator copper tube to acquire data for investigations.

Fig.3: Twisted with its geometrical configuration.

Table II: Specification of Twisted Tapes

	Material	Thickness (t) mm	Pitch (H) mm	Width of twisted tape ≈ inner diameter of tube in mm (d)	Twist ratio, y = H/d
Twisted Tape I	Copper	0.25	105	13	8

To carry out the analysis of heat transfer enhancement due to twisted tapes, experimentation is conducted with operating parameters and specified range as shown in table III.

Table III: Operating Parameters with operating range.

1.	Mass flux ,G ($kgm^{-2}s^{-1}$)	150 - 300
2.	Heat Flux, q (kWm^{-2})	1.5 - 9.5
3.	Evaporating pressure in test section (bar)	3 - 8
4.	Evaporating temperature in test section- T_s (^0C)	-15 to 15
5.	Average vapor quality in test section, *x*	0.05 - 0.9

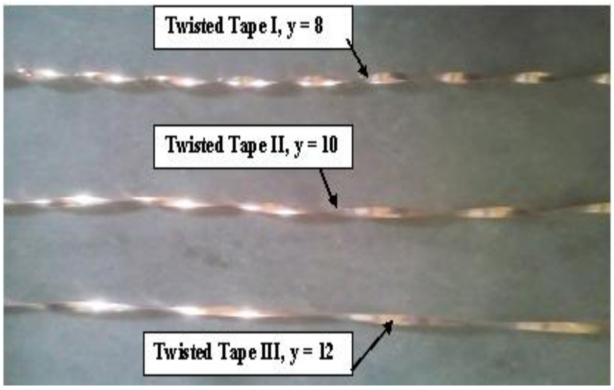


Fig.4: Twisted tapes used for experimentation

A. Data Reduction

Experimental heat transfer coefficient in the test-section tube is calculated, using equation 1.

$$h = \frac{q_{ts}}{t_{wi} - t_{sat}} \quad (1)$$

q_{ts} - applied heat flux (kWm^{-2}) on the test section tube and is estimated as,

$$q_{ts} = \frac{Q_{ts}}{A_s} \quad (2)$$

where

$$Q_{ts} = Q_h - Q_L \quad \text{(kW)} \quad (3)$$

Q_{ts} is net heat (kW) transferred to test section tube and Q_h is the total heat from heaters of test-evaporator. Q_L is the heat loss from the test-evaporator to surrounding and is obtained from heat leakage calibration curve or heat loss equation.

The inner surface temperature of test section tube, t_{wi} is estimated from equation 4.

$$t_{wi} = t_{wo} - \frac{q_{ts}}{2\pi L_{ts} K} \ln \frac{d_o}{d_i} \quad (4)$$

Average outside surface temperature of test–section tube, t_{wo}, is estimated from equation (5).

$$t_{wo} = \frac{t_{woz1} + t_{woz2} + \cdots \cdots t_{woz6}}{6} \quad (5)$$

Where, $t_{woz1}, t_{woz2}, \dots \dots \dots t_{woz6}$ are the average outside surface temperatures of test-section tube, measured at six locations z_1, z_2,….z_6 on the copper tube and each is estimated as:

$$t_{woz} = \frac{t_{woz(top)} + t_{woz(bottom)} + t_{woz(ls)} + t_{woz(rs)}}{4} \quad (6)$$

Saturation temperature, t_{sat}, corresponding to evaporating pressure at inlet to test section tube, in equation 1, is obtained from Refprop 7.0

Void fraction (vapor quality) is calculated as an average value within test-section tube, is calculated using equation 7.

$$x_{avg} = \frac{x_{in} + x_{out}}{2} \quad (7)$$

x_{in} & x_{out} are void fractions (vapor qualities) at entry and exist of test section and are calculated by equations 8 and 9.

$$x_{in} = \frac{e_{in} - e_{f(in)}}{e_{fg(in)}} \quad (8)$$

$$x_{out} = \frac{e_{out} - e_{f(out)}}{e_{fg(out)}} \quad (9)$$

Enthalpy values, $e_{in}, e_{out}, e_{f(in)}, e_{fg(in)}, e_{f(out)}$ and $e_{fg(out)}$ are obtained from REPROP 7.0, corresponding to measured values of pressure and temperature.

Uncertainty in measurement of heat input, heat flux, mass flux and heat transfer coefficient is calculated using **Robert J Moffat [33]** method. If 'R' is the result and is a function of independent variables x_1, x_2, x_3…….then uncertainty in result, u_R is calculated according to **Robert J Moffat [33]** as,

$$u_R = \sqrt{\left(\frac{\partial R}{\partial x_1} u_1\right)^2 + \left(\frac{\partial R}{\partial x_2} u_2\right)^2 + \cdots \left(\frac{\partial R}{\partial x_n} u_n\right)^2} \quad (10)$$

Where, u_1, u_2… u_n are the uncertainties in independent variables.

Thus, through uncertainty calculations, uncertainties in primary measurements and derived quantities are found out, as shown in table IV.

Table IV: Uncertainty of Variables

Primary Measurements		**Derived quantities**	
Parameter	**Uncertainty**	**Parameter**	**Uncertainty**
Voltage	± 1.16%	Heat flux	± 1.16 %
Current	± 0.059%	Heat transfer coefficient	± 1.665 % - 15.37 %
Temperature	± 0.375 ^0C	Mass Flux	± 0.251 % - 0.491 %
Pressure	± 0.24%		
Mass Flow Rate	±0.15% - 0.49%		

International Journal of Recent Technology and Engineering (IJRTE)
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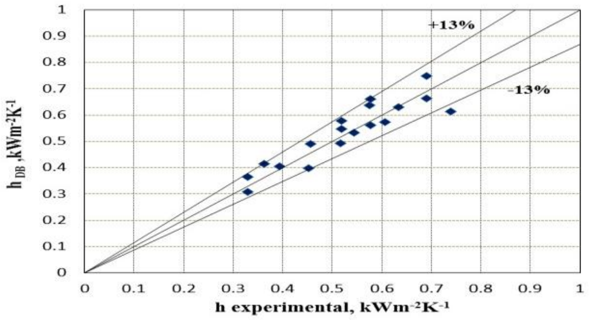
B. Heat Loss Calibration of Test-Evaporator

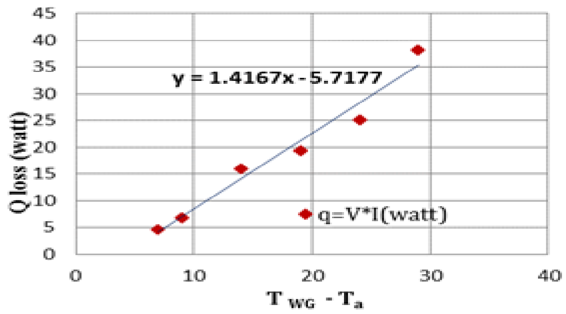
Major portion of the heat from heater of Test-evaporator will transfer to copper test-section tube by convection and rest of heat will transfer from Test-Evaporator to surrounding as a heat loss by conduction. Hence net heat input to test section, Q_{ts} can be estimated as a total heat input by heater, 'Q_h' minus heat loss, 'Q_L' to surrounding.

Thus, $Q_{TS} = Q_H - Q_L$. (11)

Hence heat loss calibration of test evaporator is carried out to estimate heat loss to surrounding. Figure 5 shows heat leakage calibration curve for Test-Evaporator obtained through heat loss calibration procedure adopted for the Test-Evaporator. Through regression analysis heat loss equation for Test-Evaporator is obtained as shown by equation 12

$$\mathbf{Q_{loss} = 1.4167[T_{water} - T_{ambient}] - 5.7177} \qquad (12)$$





During experimentation, it has been found that heat loss to surrounding from Test-Evaporator at various heat applied was negligible as compared to heat applied in test evaporator (less than 0.1% of heat applied in test evaporator).

C. Reliability Check of Experimentation

Before proceeding for the experimental procedure on setup, reliability of experimental setup is to be validated. Hence, single phase test on refrigerant R407C is to be carried out, during its flow through test-evaporator. At different heat flux and mass flux conditions in the test–evaporator; experimental single phase heat transfer coefficient values are calculated through data reduction methodology (A), using experimental data. These experimental single phase flow boiling heat transfer coefficients values within the test-evaporator tube are compared with heat transfer coefficients values calculated by Dittus-Boelter equation, using same experimental operating conditions in test-evaporator.

Experimental heat transfer coefficient values are compared with projected values by Dittus-Boelter correlation as shown in figure 6 and found that Dittus–Boelter correlation projects the experimental values within an error band of ± 13%. Hence it confirms that experimental test facility is reliable.

III. RESULTS AND DISCUSSIONS

A. Heat Transfer Enhancement against Void Fraction (vapor quality)

Experimentations are carried out with different operating conditions or sets, out of which three operating conditions are taken for discussion in the present work, as shown in table V.

Table V: Operating sets

Operating Set	Operating Conditions		
	Pressure , P (bar)	Heat flux, q (kWm^{-2})	Mass Flux, G ($kgm^{-2}s^{-1}$)
1	5	2.185	145.70
2	6	4.170	183.32
3	7	9.100	251.70

Heat transfer enhancement analysis using turbulent promoters is carried out in three ways: a) Graphical analysis, b) Quantitative analysis, c) Quantitative analysis with vapor quality

a) Graphical Analysis-

Figure 7, 8 & 9 exhibits heat transfer enhancement due to twisted tapes over a plain tube at three different operating conditions, with respect to void fraction(vapor quality).From figure 7,8 and 9,it can be observed that with increasing void fraction (vapor quality), the heat transfer enhancement over a plain tube decreases. Major improvement in heat transfer coefficient over a plain tube is observed between void fraction (vapor quality) range of 10 to 40%.Negligible improvement in heat transfer coefficient over a plain tube can be seen after 70% vapor quality. After 70% vapor quality or void fraction, the flow becomes mist flow and surface dry out occurs due to twisted tape, in such situation. Hence negligible heat transfer enhancement can be seen after 70% void fraction.

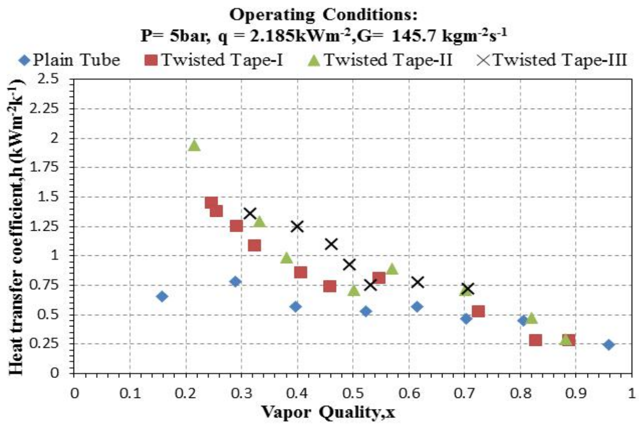


Fig.7: Heat transfer coefficient enhancement due to twisted tape inserts I, II &III over plain tube at P = 5 bar, q = 2.185 kWm^{-2} , G = 145.70 $kgm^{-2}s^{-1}$.

However, another observation from figure 7, 8 and 9, implies that enhancement in heat transfer coefficients as an effect of twisted tapes over a plain tube, decreases with increasing operating conditions.

Major enhancement in heat transfer coefficient over a plain tube can be observed for operating conditions: P = 5 bar, q = 2.185kWm^{-2}, G = 145.7kgm^{-2}s^{-1}.

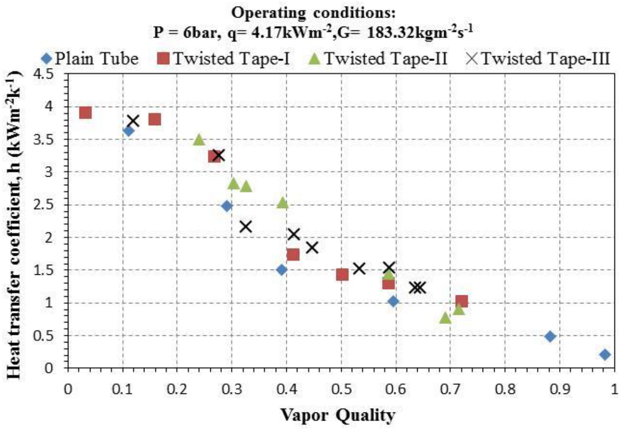


Fig.8: Heat transfer coefficient enhancement due to twisted tape inserts I, II &III over plain tube at P = 6 bar, q = 4.17 kWm^{-2}, G = 183.32 kgm^{-2}s^{-1}

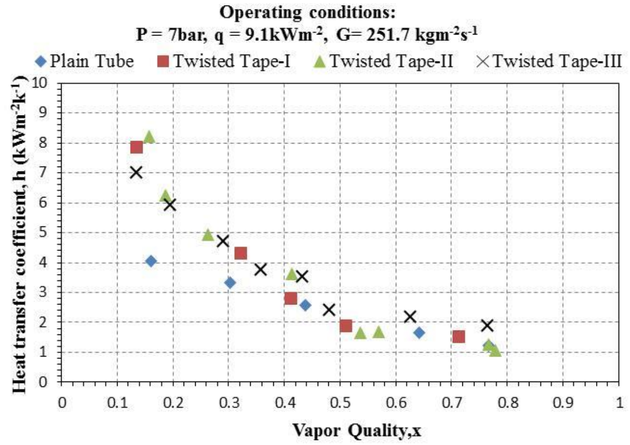


Fig.9: Heat transfer coefficient enhancement due to twisted tape inserts I,II&III over plain tube at P = 7 bar, q = 9.1 kWm^{-2} , G = 251.7 kgm^{-2}s^{-1}

From graphical analysis of figure 7, 8 and 9, it seems that twisted tape-III(y=12) is showing major heat transfer enhancement compare to other twisted tapes, for all operating conditions and over entire void fraction (vapor quality) range.

b) Quantitative Analysis-

Figure 7, 8, & 9 depicts only heat transfer enhancement due to twisted tapes over a plain tube, but not magnitude of enhancement. Hence average values of heat transfer coefficient are calculated for each operating conditions and twisted tapes and plain tube by taking the average of all heat transfer coefficient values at various void fractions (vapor qualities). Thus, percentage increment in average heat transfer coefficient due to twisted tapes over an average values of plain tube heat transfer coefficients are calculated.
Table VI shows average heat transfer coefficient values for plain tube and plain tube with twisted tape inserts and % increase in average heat transfer coefficient due to twisted tape inserts. As per table VI, all the twisted tapes in present works are showing highest values of 63.20%, 71.33% and 84.25% as a percentage increase in average heat transfer coefficient over plain tube at low operating set 1 with operating conditions of 5 bar pressure, 2.185 kWm^{-2} of heat flux and 145.7 kgm^{-2}s^{-1} of mass flux. With increasing operating conditions, degradation in heat transfer performance of each twisted tape can be observed. At operating set 2, twisted tape I, II & III are showing lower % increase in average heat transfer coefficient of 37.20%, 35.82% & 32.78 % respectively as compared to other two operating set.
In general it has been observed that twisted tape-III (y = 12) is showing better heat transfer performance than other two twisted tapes at all operating conditions. At all operating conditions, twisted tape-III exhibits maximum increase in average heat transfer coefficient of 56.51%. Twisted tape II & I shows maximum increase of 48.88% & 47.87% in average heat transfer coefficient respectively.

Set	Operating Conditions			Average Heat Transfer coefficient (h_{avg}), kWm^{-2}K^{-1}				% increase of h_{avg} over plain tube due to twisted tape inserts.		
	P_{sat} bar	q kWm^{-2}	G Kgm^{-2}s^{-1}	Plain Tube	Twisted Tape I	Twisted Tape II	Twisted Tape III	Twisted Tape I	Twisted Tape II	Twisted Tape III
1	5	2.185	145.7	0.5320	0.8682	0.9114	0.9802	63.20	71.33	84.25
2	6	4.17	183.3	1.5556	2.1343	2.1129	2.0656	37.20	35.82	32.78
3	7	9.1	251.7	2.5681	3.6783	3.5826	3.9166	43.23	39.50	52.50
Average % increase in heat transfer coefficient over plain tube due to twisted tapes								**47.87**	**48.88**	**56.51**

As operating conditions increases to pressure 6 and 7 bar, heat flux to 4.17 and 9.1kWm^{-2}, mass flux to 183.32 and 251.7 kgm^{-2}s^{-1}, the gradual reduction of heat transfer coefficients over plain tube can be observed. All the values of heat transfer coefficients due to twisted tapes seems to come close to the values of plain tube with increasing operating conditions.

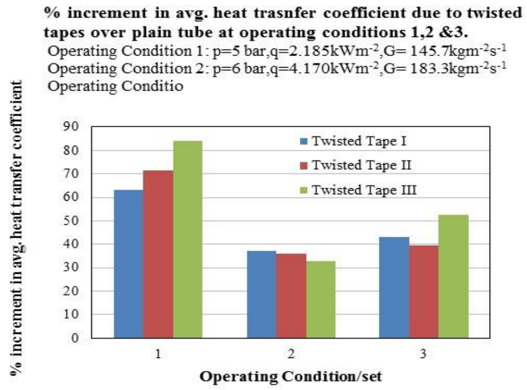


Fig. 10: % increment in average heat transfer coefficient due to twisted tapes over plain tube at different operating conditions.

From figure 10 it can be observed that all twisted tapes shows maximum % increment in average heat transfer coefficient at lower operating condition '1' of P = 5 bar, q = 2.185 kWm^{-2} , G = 145.70 kgm^{-2}s^{-1} as compared to other operating conditions. With increasing operating condition, heat transfer performance all twisted tapes is found to be degraded compared to lowest operating condition '1'. At operating condition '1' (lowest operating) and '3' (highest operating), twisted tape III (twist ratio 12) is performing best compared to other two twisted tapes. With increasing operating conditions, operating pressure, mass flux and heat flux are increased. Increased heat flux and mass flux conditions support the heat transfer enhancement, but increased operating pressure condition suppressed the heat transfer enhancement. Hence degradation in % increment of average heat transfer coefficient due to twisted tapes over plain tube can be seen with increased operating conditions '2' and '3' as compared to lowest operating condition '1'.

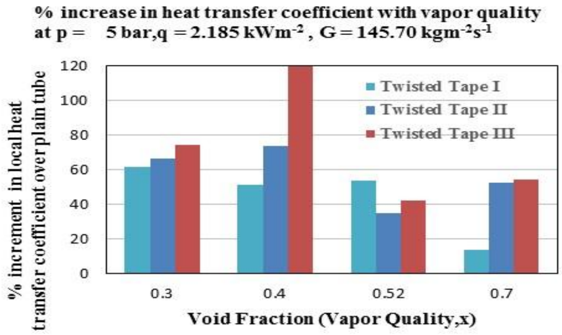
c) Quantitative Analysis with Void Fraction (vapor quality)-

The range of void fraction (vapor quality) within which improvement in heat transfer coefficient takes place is not fixed and it depends upon the refrigerant, operating parameters and geometry of turbulent promoters. Hence it is necessary to investigate the range of void fraction within which maximum improvement in heat transfer coefficient takes place. Thus heat transfer enhancement analysis of twisted tape at various vapor qualities helps us to select such void fraction range. Percentage increase in local heat transfer coefficient around some intermediates vapor qualities is shown in table VII, at three selected operating sets for twisted tape I, II, III.

i) Operating set 1: P = 5 bar, q = 2.185 kWm^{-2} , G = 145.70 kgm^{-2}s^{-1}

From figure 11, it is observed that at operating condition of 5 bar, 2.185 kWm^{-2} and mass flux of 145.70 kg m^{-2}s^{-1}, % increase of local heat transfer coefficient over plain tube take place from 30 to 40% void fraction (vapor quality), for twisted tapes II &III. From table VII, it is seen that for twisted tape II &III, between 30 to 40% void fraction, % increase in heat transfer coefficient over plain tube, improves from 66.42% to &73.47% and from 74.47% to 119.75% respectively, however for twisted tape-I, decrement from 61.37% to 51.32% is observed. Between 40% to 52% void fraction (vapor qualities), drop in % increase in local heat transfer coefficient over plain tube can be seen for twisted tapes II and III while for twisted tape- I, small increase can be seen. In overall twisted tape-III is showing best performance compared to other twisted tapes at 30%, 40% and 70% void fraction. From figure 11 and table VII, twisted tape-II is showing better heat transfer performance after twisted tape-III with respect to void fraction.

Fig.11: % heat transfer enhancement against vapor quality due to twisted tape inserts at P = 5 bar, q = 2.185 kWm^{-2} , G = 145.70 kgm^{-2}s^{-1}



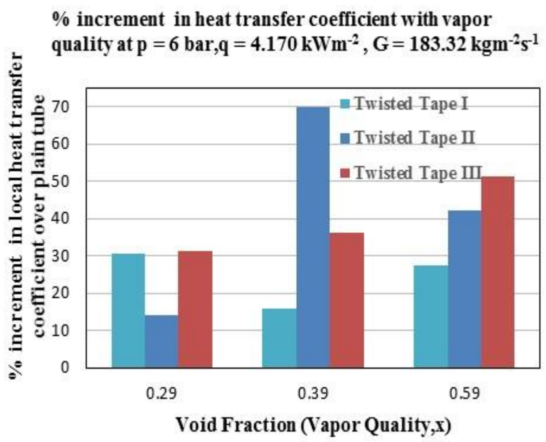


Fig.12: % heat transfer enhancement against vapor quality due to twisted tape inserts I, II &III at P = 6 bar, q = 4.17 kWm^{-2}, G = 183.32 kgm^{-2}s^{-1}

ii) Operating set 2: P = 6 bar, q = 4.170 kWm^{-2} , G = 183.32 kgm^{-2}s^{-1}

From figure 12 and table VII, it can be observed that with operating condition of 6 bar pressure, 4.170 kWm^{-2} of heat flux and 183.32 kg m^{-2}s^{-1}of mass flux, twisted tape-III is showing continuous improvement in % increase of local heat transfer coefficient of 31.43%, 36.21% & 51.17 % over plain tube at 29%, 39% & 59% vapor quality.

For operating set 2 and twisted tape-II, figure 12 and table VII shows % increase in local heat transfer coefficient over twisted tape, high volatile component, R32 of R407C evaporates at a faster rate, compared to its other components

Table VII: Percentage increment in local heat transfer coefficient around some intermediates vapor qualities with twisted tape I, II, III.

set	Operating Condition	Vapor Quality	% increase in local heat transfer coefficient		
			Twisted tape-I	Twisted tape-II	Twisted tape-III
1	P = 5 bar, q = 2.185 kWm^{-2} G = 145.70 $kgm^{-2}s^{-1}$	Around 0.30	61.37	66.42	74.47
		Around 0.40	51.32	73.47	119.75
		Around 0.52	53.83	34.58	42.01
		Around 0.70	13.48	52.13	53.95
2	P = 6 bar, q = 4.170 kWm^{-2} , G =183.32 $kgm^{-2}s^{-1}$	Around 0.29	30.63	14.24	31.43
		Around 0.39	15.97	69.79	36.21
		Around 0.59	27.48	42.18	51.17
3	P = 7 bar, q = 9.100 kWm^{-2} , G = 251.70 $kgm^{-2}s^{-1}$	Around 0.16	94.88	103.28	73.19
		Around 0.30	29.69	47.81	40.99
		Around 0.43	9.91	41.29	37.70
		Around 0.76	21.15	1.86	50.96

plain tube between 29% to 39% void fraction and then shows decrement around 59% void fraction. Over all maximum improvement from14.24 to 42.18% is shown by twisted tape -II from void fraction 29 to 59%, comparing to other twisted tapes.

iii) Operating set 3: P = 7 bar, q = 9.100 kWm^{-2}, G = 251.70 $kgm^{-2}s^{-1}$

Operating set 3 represents extreme operating conditions of in the current investigation. As per figure 13 and table VII, at operating set 3, twisted tape-II is showing best performance as compared to other two twisted tapes, with increase in local heat transfer coefficients as: 103.28%, 47.81% and 41.29%, around 16%, 30% and 43% Void Fractions(vapor qualities). But at 76% void fraction (vapor quality) twisted tape-II perform negligible with increase of 1.86% in local heat transfer coefficient.

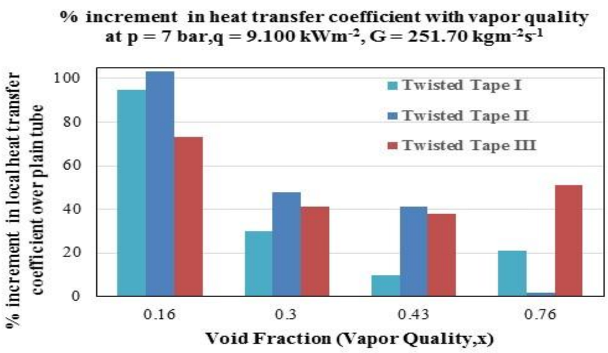


Fig. 13: % heat transfer enhancement over plain tube with respect to vapor quality due to twisted tape inserts I,II&III at P = 7 bar, q = 9.1 kWm^{-2} , G = 251.7 $kgm^{-2}s^{-1}$

From figures 11 & 12, it can be observed that up to 40% vapor quality, all twisted tapes shows improvement in local heat transfer coefficient over a plain tube, but after 40% vapor quality, % increment in local heat transfer coefficient is found to be in decreasing order. Due to turbulence created by and also large heat capacity of R32, heat transfer coefficient starts to increase within low vapor quality region of 10 to 40%. But due to large mass transfer of highly volatile R32, from liquid phase to vapor within lower vapor quality region, mass transfer resistance is developed after 40% vapor quality. As a consequence corresponding decrease in heat transfer coefficient can be seen after 40% vapor quality.

In general, from graphical and quantitate analysis, in all operating conditions, twisted tape-III is performing best for heat transfer coefficients enhancement at various vapor qualities. Second best performance is shown by twisted tape-II at various operating conditions.

IV.CONCLUSIONS

To carry out the investigation in the present works experimental set up is fabricated and installed in the Refrigeration lab of Government college of Engineering, Aurangabad, (Maharashtra) India.

Investigation and discussions on results outlined following conclusions.

- Void fraction region of 10 to 50% is the best for the effective heat transfer enhancement for all the operating conditions and twisted tape geometries in the current work.
- After 70% void fraction use of twisted tapes is not so effective.
- Effectiveness of twisted tapes depends on their geometries and operating conditions.
- Twisted tape-III with twist ratio 12 shows prominent heat transfer performance for all operating condition within entire void fraction (vapor quality) range.

- Due to large mass transfer of highly volatile R32, from liquid phase to vapor within lower void fraction(vapor quality) region up to 40%, mass transfer resistance is developed. As a consequence corresponding decrease in heat transfer coefficient can be seen after 40% vapor quality.
- Degradation in the increment of heat transfer performance due to twisted tape is seen with increasing operating conditions.
- Twisted tape –III of twist ratio 12 shows best heat transfer performance of **56.51%,** increase in average heat transfer coefficient among other two twisted tapes.

Effectiveness of any turbulent promoters in heat transfer enhancement depends on their geometries and operating conditions.

.ACKNOWLEDGMENT

The financial aid for the fabrication of experimental setup under TQIP Phase-II (Government College of Engineering, Aurangabad, India) is politely acknowledged.

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