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Experimental Investigation on Heat Transfer Analysis of R407C Evaporator Using Twisted Tape

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***Abstract* - An experimental study on heat transfer enhancement and pumping loss is done during evaporation of R407C in horizontal 2 m long copper tube Evaporator. Plain tube and plain tube with turbulent promoters are considered for investigation. Experimentation is carried on the plane tube and plane tube with twisted tapes. Operating conditions: refrigerant mass flux, G-130 – 250 kg $m^{-2}s^{-1}$, heat Flux, q 2– 9 kW m^{-2}, temperature range: -14 to 10.9 ^{0}C, pressure range (absolute): 4 -8 bar and vapor quality, x- 0.05 - 0.95.Three twisted tapes are used as turbulent promoters of twist ratios 8, 10 and 12. Study reveal that heat transfer enhancement depends on geometry of twisted tape and operating conditions. Twisted tape with twist ratio, y = 12 shows maximum heat transfer at all operating conditions with 67.88% maximum average increase in heat transfer coefficient. With increasing operating conditions of heat flux, mass flux and pressure, pumping loss increases. Twisted tape with twist ratio, y = 12 is giving minimum percentage increase of pumping loss from 3.3% to 6.09% over plain tube.**

Keywords : *Heat transfer enhancement; investigation; pumping power; R407C; twisted tapes.*

NOMENCLATURE		**Subscripts**	
Diameter, mm Dittus –	d DB	Enthalpy of liquidInner	efi
Boelter	e	Enthalpy of evaporation	efg
Enthalpy			
Heater	H	Inlet	in
Full- scale deflection	FSD	Left side	ls
Full Scale	FS	Right side	rs
Heat Loss, kW	L	Saturation	sat
Mass flux, kgs^{-1} m^{-2}	G	Outer	o
Heat transfer coefficient, W m^{-2}k^{-1}	h	Test section	ts
Proportional–integral–derivative	PID	Inside wall	wi
Heat flux, Wm^{-2}	q	Outside wall	wo
Two phase	tp	Outside wall location	woz
Heat, kW	Q		
Temperature	t		
Upper Range Limit	URL		
Vapor quality	x		

INTRODUCTION

Swirl The rate of heat performance is accelerated by the swirl flow created by the turbulent promoter. Due to the turbulence promoter, M. Cumo et al. [1] discovered a 200% increase in heat transfer at the dry out point in the Evaporator of Feron 12. [2,3,4] have conducted an experimental investigation on the use of twisted tapes to improve heat transfer during the evaporation of Feron 12. According to their research, the twist ratio and test conditions have an impact on heat transfer enrichment. Mark.A.Kedzierski et al.[5] attribute the drop in Nusselt number with increasing vapour quality to twisted tape produced a partial dry out of the inner tube surface.

In the condensation of R22, the research of Ramakant Shrivastav et al [6] discovered when twisted tape with a twist ratio of 6 was compared to twisted tape with a twist ratio of 15 and 9, the heat transmission performance improved by 25%. However, M.A. Akhavan et al. [7-8] discovered that using twisted tapes with twist ratios of 15, 12, 9, 6, and with coiled wire inserts of 13, 10, 8, and 5 mm resulted in a larger pumping power penalty at the charge of improvement in heat transmission. Nano fluid with Coiled wire inserts flow were employed by M. Saeedinia et al. [9] to demonstrate similar types of increases in Nano particle concentration with increasing pressure drop and heat transfer.

In areas where the vapour quality is high, twisted tapes are profitable. Taye Steven Mogaji and others [10]. Because they provide a vast heat transfer area, spongey inserts significantly improve heat transfer. Mao-Yu Wen and others [11] For improving heat transfer, annular and intermittent flow patterns are more efficient. [12] Maziar Shafaee et al.

Coiled tubes have been employed by [13,14,15] to improve heat transfer. In areas with high vapour quality, the curvature radius or coiled tube diameter has a larger effect on the heat transfer performance Ahmad Reza Salimpour and others [13].

Motivation for scrutiny of R407C

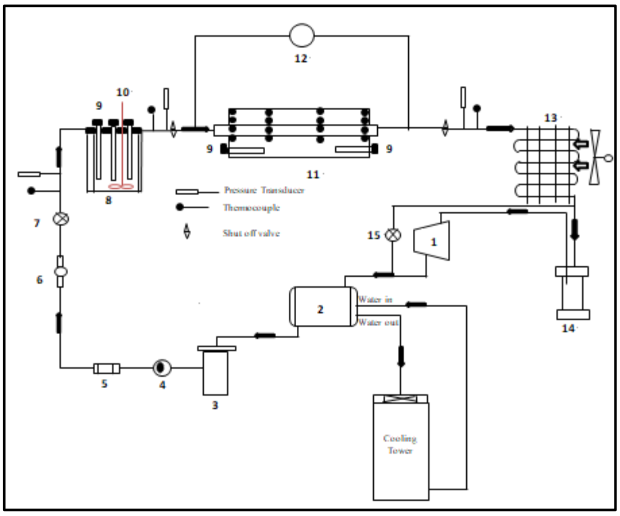
Excellent thermal properties have elevated R22 to the position of a leading refrigerant in the refrigeration industry. However, it has been prohibited from usage beyond 2020 due to its great potential for ozone depletion and global warming. The refrigeration industry is looking for novel refrigerants to replace R22 that have a low global warming potential and no ODP. R22 replacement refrigerants include R407C, R410A, R-290, R-134a, and many more. However, as R134a breaks down due to sunlight in the troposphere, it produces acid and toxic byproducts, which can be even worse than ozone depletion and global warming. et al. Shailendra Kasera [16]. On the other hand, R-290 having a safety concern in handling . R407C has thermal performance that is similar to R22 because its thermo physical characteristics are more similar to R22 than those of other substitutes. As the finest R22 conversion, in current refrigeration systems handling R-22 can be used. According to C. Aprea et al. [17], the possible replacement of R22 could be R407C. Experimental comparisons between the thermal performance of R407C and R22 by [18,19,20] revealed a modest thermal performance gap between R407C and R22. According to assessments from the United Nations Environment Programme (UNEP) [21] and [22], R-407C is the best transitional refrigerant and may be used as a replacement for R-22 in current systems with minimum modification. Juan Garcia et al. [23] examined the impact of diameter, quality, heat flux, mass flow, and pressure on the two-phase pressure drop of R407C. Recent research by P HorakM et al. [24] on the evaporation of R407C in a vertical smooth evaporation tube found that, at low fluxes of 9 $kgm^{-2}s^{-1}$, nucleate boiling predominated over convective boiling in terms of heat transfer.

Therefore, it may be inferred from many studies on R407C that R407C is a promising alternative to R22. Heat transfer improvement strategies should be used to improve R407 C's thermal performance. To examine heat transfer enhancement during the flow boiling of R407C in a horizontal evaporator, twisted tapes are therefore used in the current work. On the other hand, heat transfer improvement must be compared to pumping loss brought on by twisted tapes. This will make it easier to comprehend the advantages of the heat transfer improvement method. Therefore, twisted tape is used as a turbulent promoter in the current investigation's experimental examination of heat transfer augmentation and pumping loss in the R407C evaporator.

Global warming has become an issue as a result of population increase and the industrialisation of nations. Therefore, it is essential to use eco-friendly refrigerants in all types of HVAC and refrigeration systems. The effects of HVAC and refrigeration systems on global warming are significant. In 2014, these systems contributed 7.8% of all greenhouse gas emissions. Employing natural refrigerants is desirable because they have a negligibly low GWP and an ozone depletion potential (ODP) when compared to other refrigerants, which are the two main environmental concerns with refrigerants.

EXPERIMENTATION

Figure 1 shows layout diagram of experimental test facility. 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.



1-Compressor, 2-Condenser, 3-Receiver, 4-Flow meter, 5-Drier, 6-Sight Glass, 7-Manual Expansion Valve, 8- Pre- Evaporator, 9-Heater, 10-Stirrer, 11-TestEvaporator, 12-Differential Pressure Transducer, 13-By Pass Valve,
14-Accumulator, 15- By Pass Valve

Figure1. Experimental setup.

Prior to entering the test evaporator, the pre-evaporator modifies the refrigerant quality to meet the requirements. Pre-evaporator heat input is accurately controlled using a PID controller and solid state relay. A copper test tube runs through a long, cylindrical stainless-steel pipe known as a test-evaporator. A water-glycol solution is placed within the test evaporator. Figure 1 shows the installation of two 0.9 m-long heating rods into the test evaporator parallel to the copper test tube. Heat input of the test evaporator is managed by a solid state relay controller. As shown in figure 2, a group of four thermocouples are brazed together on the test tube's exterior edge, each thermocouple 900 degrees apart from the others. Six of these sets of thermocouples are evenly spaced apart and brazed onto the test tube's exterior. After-evaporator is installed in the refrigerant line on the bottom side of the test-evaporator to control the superheat of the refrigerant at the compressor's inlet.

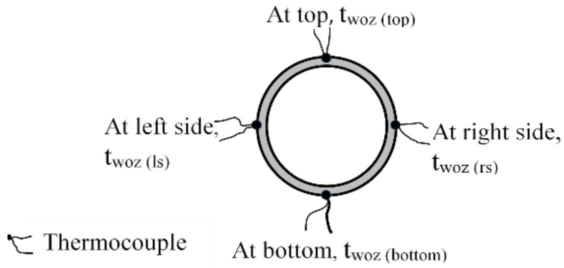


Figure 2: Test section tube with thermocouples

The temperature and pressure of the refrigerant are measured using absolute pressure transducer (Piezo-resistive and T-type thermocouples. The pressure drop across the test evaporator tracked by a differential pressure transducer to determine pumping loss. (Piezo-resistive type). The use of oval gear type flow meters for monitoring refrigerant flow. Table I displays the accuracy and range of the instrument.

Table I: Accuracy with range of instruments

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

Table II: Specification of Twisted Tapes

	Material	Thickness (t) mm	Pitch(H) mm	Width of twisted tape ≈ inner diameter of tube inmm (d)	Twistratio,y = H/d
TwistedTape I	Copper	0.25	105	13	8
TwistedTape II	Copper	0.25	132	13	10
TwistedTape III	Copper	0.25	156	13	12

Figure 3 shows twisted tapes used in the present experimentation. Table II shows geometrical specifications of twisted tape. Table III shows operating parameters and their range.

Table III: Operating Parameters with operatingrange.

Sr.No.	Operating parameters	Range
1.	Mass flux, G (kgm^{-2}s^{-1})	130- 250
2.	Heat Flux, q (kWm^{-2})	2 - 8
3.	Evaporating pressure in testsection (bar)	4 - 8
4.	Evaporating temperature in testsection- T_s (^{0}C)	-14 to 10.9
5.	Average vapor quality in testsection, *x*	0.05 - 0.95

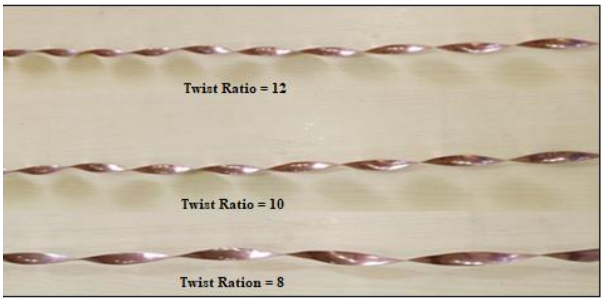


Figure.3. Twisted tapes used for experimentation

Data Reduction

Experimental heat transfer coefficient in test -evaporator is calculated, using equation 1.

$$\mathbf{h} = \frac{\mathbf{qts}}{\mathbf{twi - tsat}} \qquad (1)$$

qts - heat flux (kWm^{-2}) on the test section tube, calculated as,

$$\mathbf{qts} = \frac{\mathbf{Qts}}{\mathbf{As}} \qquad (2)$$

Where,

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

Q_{ts} is net heat (kW) given to test section tube, Q_h - total heat from test-evaporator heaters, Q_L - heat loss to surrounding, estimated from heat leakage calibration curve or heat loss equation .

t_{wi} in equation (1) is calculated using equation (4).

$$\boldsymbol{twi = two - \left(\frac{Q}{2\prod LK}\right) * ln\left(\frac{do}{di}\right)} \qquad (4)$$

Average outside surface temperature of test–evaporator tube, $t_{wo,}$ is obtained through equation (5).

$$\boldsymbol{two = \frac{twoz1 + twoz2 + \cdots twoz16}{16}} \qquad (5)$$

Where, $t_{woz1}, t_{woz2}, \ldots \ldots \ldots \ldots t_{woz6}$ are the average outside surface temperatures of test-section tube,

$Q_{loss} = 1.4167[T_{water\text{-}glycol} - T_{ambient}] - 5.7177$ (11)

measured at six locations z1, z2,….z6 on the copper tube and each is estimated as:

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

Vapor quality is considered as an average of entry and exist vapor qualities of test- evaporator, using equation (7).

$$\boldsymbol{Xavg = \frac{Xin + Xout}{2}} \qquad (7)$$

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

Uncertainty in measurement of heat input, heat flux, mass flux and heat transfer coefficient is calculated using **Robert J Moffat [25]** method. If 'R' is the result and is a function of independent variables $x1,\ x2,\ x3$…….then uncertainty in result, u_R is calculated according to **Robert J Moffat [25]** as,

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

Where, $u1,\ u2 \ldots\ un$ are the uncertainties in independent variables. Thus, uncertainties calculated are shown in table IV.

Heat Loss Calibration of Test-Evaporator

Heat loss calibration of test-evaporator is carried out using calorimeter test procedure. In heat loss test of Test – Evaporator, heat loss data for different heat input is collected. Using this data with the help of regression methods, heat loss calibration curve and heat loss equation as shown by equation (11) is obtained.

Table V: Operating sets

OperatingSet	Operating Conditions		
	Pressure, P (bar)	Heat flux, q (kWm^{-2})	Mass Flux, G ($kgm^{-2}s^{-1}$)
1	4	2	130
2	6	4	170
3	8	9	250

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 ^{0}C	Mass Flux	± 0.251 % - 0.491 %
Pressure	± 0.24%	Mass Flow Rate	±0.15% - 0.49%

RESULTS AND DISCUSSIONS

A. *Heat transfer enhancement using twisted tapes*

Figure 5, 6 and 7 show how twisted tapes I, II, and III improve heat transfer coefficients compared to a plain tube. It is noted that the heat transfer improvement caused by twisted tapes over a plain tube can be noticed up to 60% vapour quality for all the twisted tapes in the graphs After 70% vapour quality region twisted tapes have a slightly better heat transfer coefficient than plain tube. The component R32 which is highly volatile of the zeotropic mixture R407C evaporates extremely quickly in comparison to other components in the lower vapour quality area of 0.1 to 0.6. As a result, the zeotropic mixture R407C's vapour to liquid density ratio rises and two-phase flow quickens. This acceleration causes swirl in the two-phase flow in addition to flow.

Respectively, the heat transfer coefficients within the vapour quality range of 10 to 60% are improved by all these favourable environmental factors. After 70% vapour quality, two phase flow transforms into mist flow due to ongoing improvements in the vapour to liquid density ratio (i.e. flow acceleration) and swirl. Once more, after 70% vapour quality, local surface dry out happens as a result of twisted tapes. Therefore, after 60% vapour quality, the improvement in coefficient of heat transfer over plain tube caused by twisted tapes can be found to be small.

In comparison to other twisted tapes, it appears that twisted tape-III having twist ratio 12 exhibits the more heat transfer performance in Figure 5, 6 and 7. Table VI presents quantitative analysis to comprehend the extent of the increase in heat transfer performance brought on by the use of twisted tapes opposed to a plain tube.

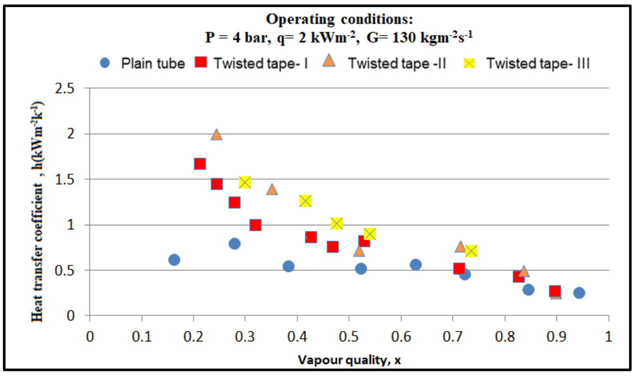


Figure 5: Heat transfer coefficient enhancement due to twisted tape inserts I, II &III over plain tube at P = 5 bar, q = 2 kWm^2, G = 130 kgm^{-2}s^{-1}

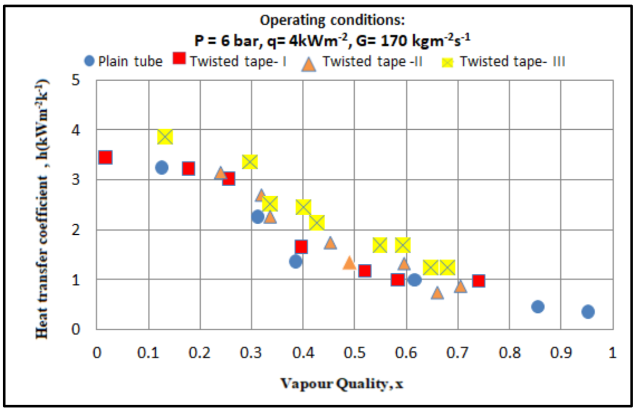


Figure 6. Heat transfer coefficient enhancement due to twisted tape inserts I, II &III over plain tube at P = 6 bar, q = 4 kWm^2, G = 180 kgm^{-2}s^{-1}

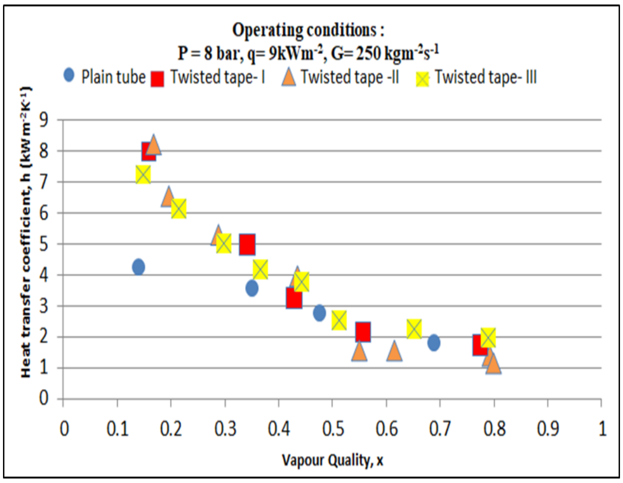


Figure 7. Heat transfer coefficient enhancement due to twisted tape inserts I, II & III over plain tube at P = 8 bar, q = 9kWm^2, G = 250 kgm^{-2}s^{-1}

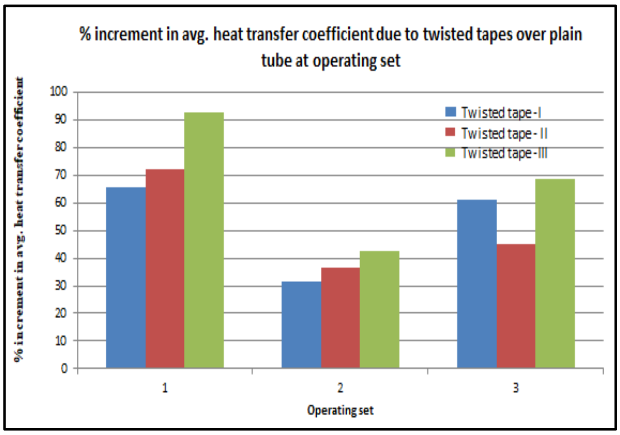


Figure 8. % increment in average heat transfer coefficient due to twisted tapes over plain tube at different operating conditions.

Table VI: % increase of average heat transfer coefficient with twisted tape over plain tube

Set	Operating Conditions			Average Heat Transfercoefficient (havg), kWm^{-2}K^{-1}				% increase of havg over plain tube dueto twisted tape inserts.		
	Psat bar	q kWm^{-2}	G Kgm^{-2}s^{-1}	Plain Tube	Twisted Tape I	Twisted Tape II	Twisted TapeIII	Twisted Tape I	Twisted Tape II	Twisted Tape III
1	4	2	130	0.5230	0.8648	0.900282	1.00738	65.35	72.13	92.61
2	6	4	170	1.546	2.03257	2.1099	2.204881	31.47	36.47	42.61
3	8	8	250	2.47681	3.98404	3.593355	4.131831	60.85	45.08	68.47
Percentage of increase in heat transfer performance (average) with twisted tape over plain tube								**52.55**	**51.22**	**67.88**

According to Table VI, all of the twisted tapes used in this experiment have the best heat transfer performance, with average coefficient of heat transfer increases over plain tubes of 65.35%, 72.13%, and 92.61% under low working conditions (set1). The table VI shows how the performance of twisted tapes degrades when operating conditions get worse. The average heat transfer coefficient on twisted tape III indicates a maximum increase of 67.88%. At the lower operating set "1," Figure 8. illustrates the maximum% increment in the average coefficient of heat transfer.

The performance of Twisted tape with a twist ratio of 12 is better than the other two twisted tapes having twist ratios 8,10 at operating set 3 (highest operating condition) operating set 1 (lowest operating condition). Pressure, mass flux, and heat flux all rise when operating circumstances do. The heat transfer enhancement is sustained by these conditions of increased heat flux and mass flux, but it is inhibited by situations of higher operating pressure. With increasing operating set "2" and "3," it is possible to see a decline in the percentage increment of the average heat transfer coefficient.

B. *Pumping power due to twisted tape –*

In this section, effect of mass flux on pumping power is investigated. Pumping power values within test evaporator tube with twisted tapes and without twisted tapes are calculated. From these values, % increase in pumping power over a plain tube is estimated.

Effect of Mass Flux on Pumping Power

Turbulent promoters like twisted tapes and coiled wire inserts increases frictional pressure drop. As a consequences pumping power required to maintain the mass flow rate of refrigerant increases. Pumping power is calculated as a product of volume flow rate of refrigerant (m^3/sec) and pressure drop across test section tube (N/m^2).

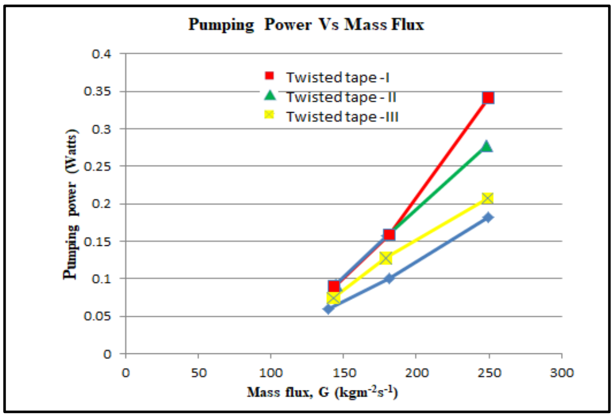


Figure 9. Pumping power versus mass flux for plain tube and plain tube with twisted tapes.

Figure 9 depicts how the pumping power for twisted tape-III, II, and I varies with mass flux. Figure 8 shows that for all twisted tapes, pumping power increases linearly with increase in mass flux. Twisted tape-III (twist ratio = 12) exhibits the least increase in pumping power compared to plain tube. Reduced free area across the flow is caused by twisted tapes. Therefore, high pumping power is needed to maintain the refrigerant mass flow rate because greater resistance will be offered for refrigerant flow as a result of the decrease in cross sectional area of flow. Once more, the swirl that the twisted tape creates helps to raise the frictional pressure drop. However, the cross sectional area of the flow increases with rising pitch or lowering twist ratio of the twisted tape. This will aid in lowering the frictional pressure drop and flow resistance, which will decrease pumping power when pitch is raised. Along with the mass flux and pitch of the twisted tape, the heat flux and saturation pressure also have a significant impact on the pumping power.

Quantitative investigation of pumping loss caused by turbulent promoters in heat transfer enhancement is required before choosing the best turbulent promoters. The information for a quantitative examination of the pumping loss caused by the twisted tapes I, II, and III is presented in tables VII and VIII.

At various operational parameters, including saturation pressure, heat flux, and mass flux, Table VII displays the pumping power for plain tubes and plain tubes with twisted tape inserts. The percentage increase in pumping power caused by the twisted tape inserts in the plain tube is shown in Table VII. Table VII's general finding suggests that pumping power rises as operational conditions rise. The increase in pumping power from the twisted tape group ranges from 0.0778 watt to 0.202 watt for twisted tape-III and from 0.095 watt to 0.354 watt for twisted tape-I.

The twisted tape III with a twist ratio of y = 12 exhibits the least increase in pumping power as operating conditions increase. From table VII, it can be observed that twisted tape-III is giving minimum percentage increase of pumping power from 3.3% to 6.09% over plain tube.tube.

Table VII: Pumping power in Plain tube and Plain tube with inserts at differnt operating conditions.

Operating set	Operating conditions			Pumping Power in Test-Evaporator (watts)			
	Psat, (bar)	q, (kW/m^2)	G (Kg/m^{2}s^1)	Plain tube	Twisted tape-I	Twisted tape-II	Twisted tape- III
1	4	2	130	0.0753	0.095	0.0938	0.0778
2	6	4	170	0.1063	0.1691	0.1642	0.111
3	8	8	250	0.1904	0.354	0.289	0.202

Table VIII: Percentage increase of Pumping power due to inserts in plain tube over pumping power in plain tube.

Mass flux, G_{kgm}-2_s-1	Percentage increase of pumping power for plain tube with inserts with respect toplain tube (%)		
	Twisted tape-I	Twisted tape-II	Twisted tape-III
130	26.16	24.56	3.30
170	59.07	54.46	4.42
250	85.92	51.78	6.09

C. *Heat transfer enhancement against pumping power*

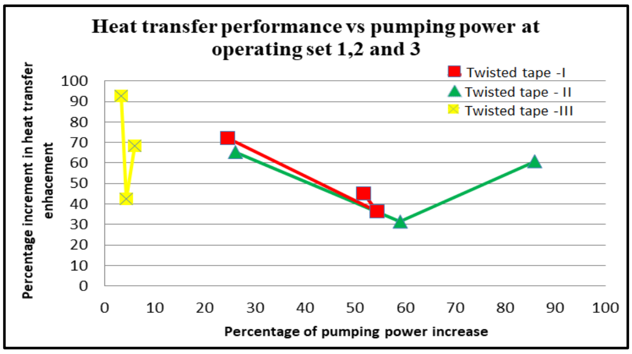


Figure 10. Heat transfer enhancement against pumpingpower with respect to operating sets or conditions.

As per data of table VI, VII and VIII, Figure 10 shows percentage of heat transfer performance due to twisted tape vs their percentage of increase in pumping power at operating set 1, 2 and 3. In the figure 10 we can observe twisted tape – III having twist ratio 12 shows maximum heat transfer performance means vertical growth in graph and minimum increase in pumping power means less horizontal growth. Twisted tape – I shows highest pumping power (horizontal growth) and lower heat transfer performance (vertical growth) in the graph heat transfer performance vs pumping power.

Thus, twisted tape – III having twist ratio shows maximum percentage average of heat transfer performance 67.88 % over a plain tube with minimum increase in percentage of pumping power ranges from 3.30 to 6.09 % over a plain tube between all the twisted tapes used in current experimentations. From current experimentation shows that twisted tape – III having twist ratio observe the most useful turbulent promoter for performance of heat transfer at different operating conditions. With increase in operating conditions it is observed that pumping power also increases for tube having twisted tape and plain tube.

CONCLUSIONS

Through results and discussion, following conclusions can be drawn.

- All twisted tapes are effective in lower vapor quality region of 10 to 60% only.
- Effectiveness of twisted tapes depends on operating conditions, vapor quality region, geometry of twisted tapes and pumping loss.
- In the current study, the better heat transfer performance exhibited by twisted tape-III having a twist ratio of 12, with an average increase in percentage of heat transfer coefficient 67.88% and the lowest pumping loss of 3.30% to 6.09%.
- The stronger vertical performance growth and smaller horizontal performance growth of any turbulent promoter on the heat transfer-pumping power graph indicate that it is a turbulent promoter for the enhancement of heat transfer within the defined operating range.

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