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Ramakant Shrivastva
Indian Institute of Technology

Ravi Kumar
Indian Institute of Technology

Akhilesh Gupta
Indian Institute of Technology

Sachida Nand Lal
Bhagalpur College of Engineering

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Ramakant Shrivastava[1] *Ravi Kumar[2] Akhilesh Gupta[3] Sachida Nand Lal[4]

[1]Department of Mechanical & Industrial Engineering, Indian Institute of Technology Roorkee
Roorkee, Uttaranchal (INDIA)
Tel: +91 9412911941, e -mail: rksridme@iitr.ernet.in

[2]Department of Mechanical & Industrial Engineering, Indian Institute of Technology Roorkee
Roorkee, Uttaranchal (INDIA)
Tel: +91-1332-285740, Fax: +91-1332-273560, e -mail: ravikfme@iitr.ernet.in

[3]Department of Mechanical & Industrial Engineering, Indian Institute of Technology Roorkee
Roorkee, Uttaranchal (INDIA)
Tel: +91-1332-285613, Fax: +91-1332-273560, e-mail: akhilfme@iitr.ernet.in

[4]Department of Mechanical Engineering, Bhagalpur College of Engineering Bhagalpur
Bhagalpur, Bihar (INDIA)

* Corresponding Author

ABSTRACT

In this paper the enhancement in heat transfer has been studied for the forced convection condensation of R-22 saturated vapor inside a tube in presence of twisted tape inserts. The test-condenser is constituted by four test-sections connected in series. Each test-section has inside diameter of 12.7 mm and the length of 950 mm. Three twisted tape inserts with the twist ratio, y, of 15, 9 and 6 were put, one by one, in the test-condenser. The insert with twist ratio, y, of 6 gave the best performance and enhanced the average heat transfer coefficient, $\bar{h}$ by 25 percent as compared to the plain flow. The segmented tapes of best performing twist ratio, y, of 6 were inserted one at a time in the test condenser. The length of segmented tape inserts was taken as 1/4, 1/2, and 3/4 of the total length of test-condenser. The refrigerant flow rate, G, has been taken as 237 kg/s-m^2.

1. INTRODUCTION

Heat exchangers are the vital part of an energy efficient refrigeration system. Most of the refrigerants offer phase change resistance as dominant resistance in the process of heat exchange particularly with water cooled system, where refrigerants flow inside the tube. Current drive towards energy conservation necessitates the augmentation of condensation coefficient in refrigerant condensers. This will lead to operate the system at low temperature difference at same heat flux and reduced physical size. The exergetic efficiency as per second law of thermodynamics will improve. An attempt to reduce entropy generation is most welcome feature of present day technology. The use of R-22 as an refrigerant will continue in refrigeration and airconditioning industry till the year 2030 as R-22 has a very low ozone depleting potential (5% of R-11) and till date there is no reliable drop in substitute of R-22. The studies on the condensation of R-22 inside a horizontal tube have been carried out by a number of investigators (Boissieux et al. 2000, Jung et al. 2003). Research on the performance improvement of R-22 charged refrigeration systems is still relevant with the goal of complex technologies. In order to enhance the condensing side heat transfer coefficient, turbulence promoters viz. twisted tapes, coiled wires have been reported (Bergles 1985) to be a good heat transfer augmentation devices with easy to remove the insert devices for maintenance and cleaning purpose.

Twisted tape inserts develops the complex flow pattern inside the tube. Turbulent flow develops ahead of time at much low mass velocities of refrigerant flow, thus higher heat transfer coefficient during condensation offers less resistance to heat transfer. However, the enhancement in heat transfer depends upon the range of experimental parameters. In refrigerant condensers the upstream portion of the tube is flooded with saturated vapor and the quality of vapor keeps on reducing along the tube and at the down stream all the vapor is converted into liquid. It will be interesting to find the portion of the condenser where the turbulence promoter is most effective

Keeping the above features in a view, an experimental investigation has been carried out to study the enhancement in heat transfer coefficient, during condensation of R-22 inside a single horizontal tube with the help of twisted tapes. In fact, the twisted tapes of different twist ratio are being used to investigate the enhancement in heat transfer coefficient. The present investigation also involves the study of the performance of shortened promoter its comparison with the full length one, as a first step work on segmented tape inserts in R-22 condensers.

2. EXPERIMENTAL SET-UP

The schematic diagram of the experimental set-up is shown in Figure 1. The experimental set-up is a well instrumented 5 Ton capacity vapor compression refrigeration system. The unit is designed to yield the experimental parameters shown in Table-1. The main constituents of the test facility are the test-condenser, pre-condenser, after-condenser, evaporator, a water cooled condenser, instrumentation and the accessories.

Figure 2 shows the sectional arrangement of one test-section. The test-section is made of hard drawn copper tube with 12.7 mm inside diameter, d_i, 15.8 mm outside diameter, d_o and 950 mm length. This tube is located concentrically inside a copper tube of 50 mm inside diameter. Four such test-sections were installed in series to form the complete test-condenser to have wide range of vapor quality during condensation. In the test-condenser, the cooling water flowed in the annular space, whereas, the refrigerant vapor flowed inside the inner tube. The outer wall temperatures of the inner tube were measured at four axial locations in each test-section. At each of these four locations, T-type thermocouples were fixed on the top, side and bottom positions. The T- type thermocouples with teflon sleeves were brazed over the tube surface, so as to avoid thermocouple bit contact with flowing cooling water over the test section.

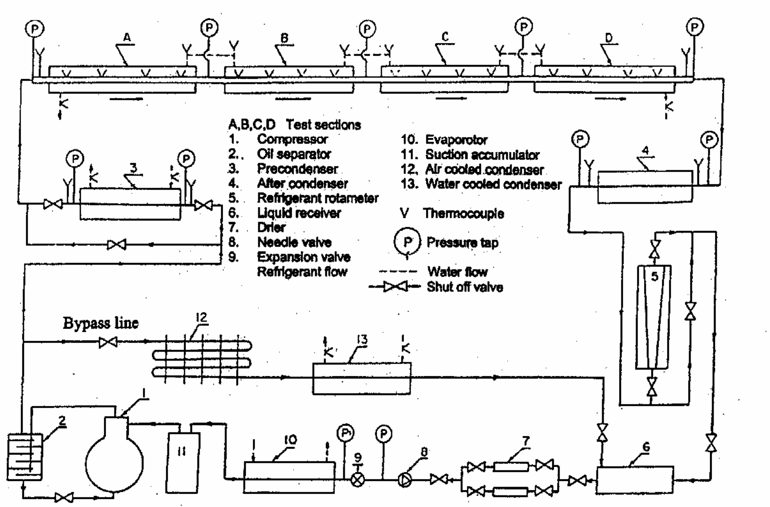

Figure 1 Schematic diagram of experimental set-up

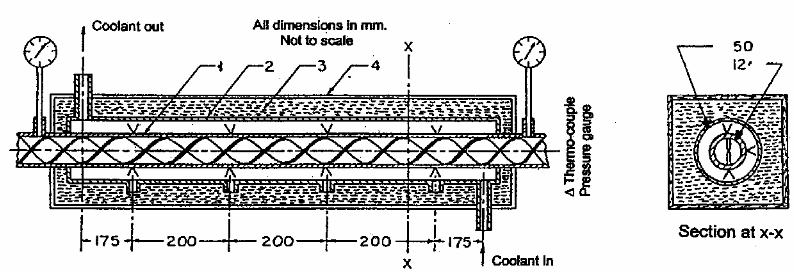

Figure 2 Detail of test-section

The entire test-condenser was thermally insulated. The heat loss was up to 1.0 percent of the heat transfer during condensation inside the test-section ensuring the effectiveness of insulation. The inlet and outlet temperatures of the cooling water and R-22 were also measured using thermopiles. Pressure taps were provided to measure the pressure at inlet and outlet of each test-section and also for the whole test-condenser. The operating parameters of the present investigation are given in Table 1.

Table- 1 Range of operating parameters

working fluid	R-22
inlet degree of superheat	5 to 10, K
refrigerant mass velocity, G	237 kg/s-m^2
cooling water flow rates, m_c	200-1200 kg/hr
coolant water temperature	20 to 30°C
condensing temperature of R-22, T_s	37 - 54 ^{o}C
condensing pressure of R-22, P	14 to 22 bar
vapor quality range, x	1.0- 0.10
average cooling heat flux, q	21 to 4 kW /m^2
segmented tape length	l/4, 1/2 & 3/4 length
twist ratio for tapes, y	15, 9, 6

The twisted tapes were used for creating swirl in the fluid flow. These tapes were made from 0.5 mm thick stainless steel flats. The width of the strips was 5 percent more than the inside tube diameter of the test-section to allow for contraction in the twisting. A full-length tape fitted inside the tube has been shown in Figure 2. For the 3/4 segmental length, the tape was fitted in section A, B and C, for 1/2 segmental length the tape was fitted in A and B, and for 1/4 length it was fitted only in section A, Figure 1. The data were acquired for the condensation of R-22 at the mass velocity of 237 kg/s-m^2 inside the plain tube and in the presence of different twisted tapes having the twist ratios of 15, 9 and 6. The performance of segmented tape inserts in 1/4, 1/2 and 3/4 of the full-length of the test-condenser with best performing twist ratio was also investigated.

The outside average tube wall temperature of test-section at one station was calculated by Equation (1)

$$T_{ws} = \frac{TT + 2TS + TB}{4} \tag{1}$$

The average temperature of outside wall of one test-section, T_{wa}, was calculated from the average of four axial stations. In order to calculate the heat flux the heat carried away by the cooling water was divided by the surface area, A.

With the help of radial conduction equation the drop across the test-section tube wall, ΔT_w, was evaluated by the Equation (2).

$$\Delta T_w = \frac{d_o \left\{ \ln\left({}^{d_o}\!/_{d_i} \right) \right\} q}{2k} \tag{2}$$

The average inside tube wall temperature, T_{wi}, was evaluated by adding the temperature drop across the wall to the measured average outside wall temperature already evaluated by Equation (2).

The average static pressure in the test-section was determined by taking the mean of inlet and exit pressure for each test-section. The average heat transfer coefficient, h, was computed with the help of the radial heat flux of the tube wall, q, the average tube inside wall temperature, T_{wi}, and the average temperature of condensing vapor, T_s, using Equation (3).

$$h = \frac{q}{\left(T_s - T_{wi}\right)} \tag{3}$$

As shown in Figure 1, a pre-condenser was installed before the test-condenser to control the vapor quality at the test-condenser inlet. The data were collected for a vapor quality range of 5 to 10 K superheat at inlet of test-condenser to about 0.10 at the outlet of test-condenser. The after condenser was installed after the test sections to condense the vapor before entering the flow measuring device rotameter.

The test-section tube wall temperature and cooling water temperature were measured with an accuracy of 0.1 ^{o}C and 0.05 ^{o}C respectively. The mass flow rate of cooling water was measured within an error of ± 2 percent. The uncertainty in heat transfer coefficient, h, was found to be within 7 percent (Lal 1993).

3. RESULTS AND DISCUSSION

First of all experimental data were collected for the condensation inside a plain horizontal tube in order to have reference data to evaluate the performance of twisted tape inserts.

Figure 3 is drawn taking condensing side heat transfer coefficient, h, as abscissa and the vapor quality, x, of R-22 as ordinate during condensation of vapor inside the test-condenser tube for plain flow i.e. without any insert. The mass velocity of refrigerant remains constant as 237 kg/s-m^2. During condensation of vapor inside the horizontal tubes the vapor quality is 0.93 at the entrance of test-condenser and it approaches to 0.13 at the exit of the test-condenser. It is clear from Figure 3 that the heat transfer coefficient, h, and the vapor quality, x, decrease down the length of test-section tubes. The heat transfer coefficient at the inlet of tube is 1725 W/m^2-K, which is reduced by 42 percent at the outlet of the test condenser. As the quality of vapor goes down the length of test-condenser the thickness of condensate layer increases. The thicker layer of condensate inside test-section tube surface offers greater thermal resistance resulting in a lower value of heat transfer coefficient near the exit of test-section.

The variation of heat transfer coefficient, h, with different twist ratios, y, at 237 kg/s-m^2 vapor mass velocity, G, is shown from Figure 4, which is drawn taking vapor quality, x, as abscissa and heat transfer coefficient, h, as ordinate. The twisted tape insets with twist ratio, y, of 6 is the best performing insert and has yielded the average heat transfer coefficient, $\overline{h}$, nearly twenty five percent more than that the plain tube flow. The enhancement in heat transfer coefficient due to twisted tape inserts is more prominent and effective at low refrigerant mass velocities in comparison to that at high refrigerant mass velocities, Kumar et al. (2005).

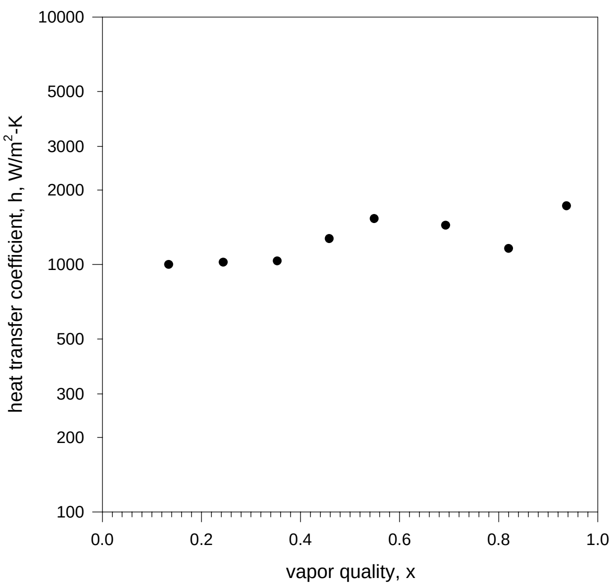

Figure 3 Variation of heat transfer coefficient with vapor quality for the plain flow

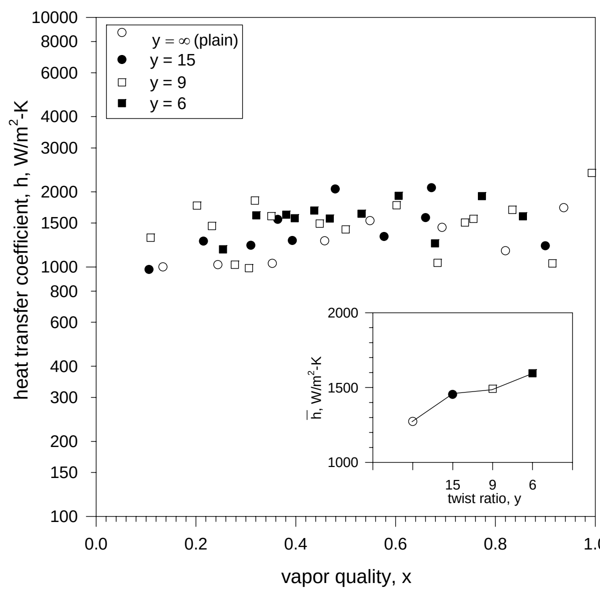

Figure 4 Comparison of heat transfer coefficients for different twist ratios

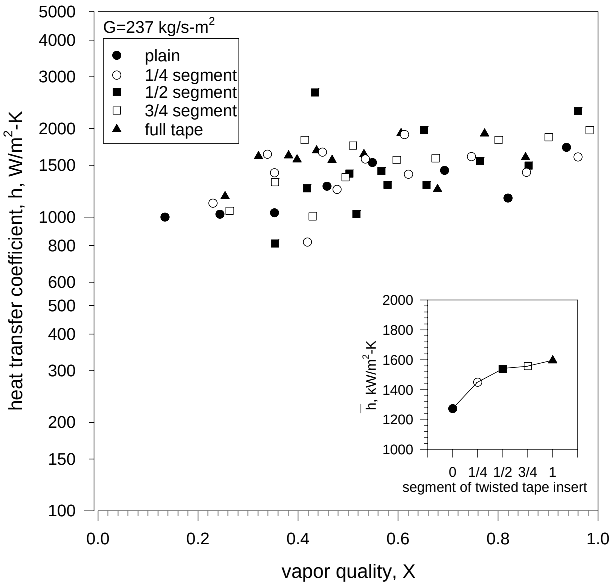

Figure 5 Comparison of heat transfer coefficients
with segmented tape inserts

At low refrigerant mass velocities the presence of tape generates the vortex motion, Kreith (1959), in the flow field. This results in the condensate entrainment in the flow and the vapor shear effect, causing augmentation in the heat transfer coefficient. The Figure 5 shows the variation of condensing side heat transfer coefficient, h, with the vapor quality, x, for different segmented tape insert lengths. The refrigerant mass flow rate has remained constant at 237 kg/s-m^2, for all the test-runs in this figure. The results are scattered as the condensation inside the condenser-tube is considered to be a highly unstablised phenomenon. The two phases are flowing together inside the condenser-tube and at the same time the phase transformation is also taking place.

The full-length tape insert has improved the average heat transfer coefficient by 25 percent, in comparison to the plain flow tube. The 3/4 segmented and 1/2 segmented inserts have improved the average heat transfer coefficient by 22 percent, 21 percent respectively. As both of them lie in the uncertainty range of each other, no rational conclusion can be drawn regarding the best performing insert. However, the 1/4 segmented insert has improved the average heat transfer coefficient, $\bar{h}$, by only 14 percent.

In Table 2 a comparison has also been made between the augmentation of heat transfer coefficient, h, in the present investigation and that by the other techniques of augmentation of heat transfer during condensation of R-22 inside a horizontal tube.

Table- 2 Different augmentation techniques for the heat transfer during condensation of R-22

Author(S)	Augmentation technique	Inside Tube Diameter, d_i, mm	Mass flux Kg/s-m^2	Outlet quality	Enhancement in heat transfer coefficient
Schlager et al. (1990)	micro fins	8.9-11.7	75-400	0.1-0.2	50-100%
Behabadi et al. (1999)	coiled wire	12.7	206-372	sub	up to 100%
Chiang (1993)	helical grooves	7.06	270-1100	sub	10-20%
Present	twisted tapes	12.7	210-372	sub	9-26%

6. CONCLUSIONS

On the basis of the entire experimental results on the twisted tape inserts of this investigation, the following conclusions can be made:

1. During condensation inside a plain tube the heat transfer coefficient decreases with the decrease in vapor quality from inlet to exit of the condenser tube.
2. The twisted tape insert with the twist ratio of 6 is the best performing insert which enhances the heat transfer coefficient approximately 25 percent.
3. The segmented twisted tape increases the average heat transfer coefficients, however, the increase is relatively small. In general, the average heat transfer coefficients are found to be more when the length of the tape is more and it is the maximum for the full-length tape inserts.

NOMENCLATURE

A	outside area of test-section, m^2
d_i	inside diameter of the test-section, m
d_o	outside diameter of test section, m
G	refrigerant mass velocity, kg/s-m^2
h	heat transfer coefficient, W/m-K
$\overline{h}$	average heat transfer coefficient, W/m^2-K
H	helix pitch length of tape, m
k	thermal conductivity of the test-section, W/m-K
m_c	Coolant flow rate, Kg/hr
P	condensing pressure of R-22, bar
q	heat flux, W/m^2
sub	subcooled liquid
TB	temperature at bottom position, K
TS	temperature in side position, K
Ts	saturation temperature of vapor, K
TT	temperature at the top position, K
T_{wi}	inside test-section wall temperature, K
T_{wo}	outside average tube wall temperature of one test-section, K
T_{ws}	outside average tube wall temperature at one station, K
ΔT_w	temperature difference across the tube wall, K
x	vapor quality of condensing vapor
y	twist ratio, ratio of half pitch of the helix and the inside tube diameter (H/d_i)

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