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# HEAT TRANSFER AUGMENTATION BY COILED WIRE INSERTS DURING FORCED CONVECTION CONDENSATION OF R-22 INSIDE HORIZONTAL TUBES

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**Abstract**—In the present work, heat transfer augmentation by coiled wire inserts during forced convection condensation of R-22 inside horizontal tubes has been studied. The test-condenser consisted of four separate coaxial double pipe test sections assembled in series. It was a counterflow heat exchanger where R-22 condensed inside the inner tube by rejecting heat to the coolant water flowing in the annulus. Coiled wires with three different wire diameters of 0.65, 1.0 and 1.5 mm and three different coil pitches of 6.5, 10.0 and 13 mm were made and used in full length of test-condenser. The use of helically coiled wires were found to increase the condensing heat transfer coefficients by as much as 100% above the plain tube values on a nominal area basis. 

*Key Words:* heat transfer, condensation, augmentation, coiled wire, inserts, R22

## 1. INTRODUCTION

Condenser is an important and widely used equipment of refrigeration and air-conditioning industry and it also forms an integral part of many other engineering systems such as the steam power plants and chemical apparatus. The design of efficient heat exchange equipment is conducive to the current drive towards energy conservation. The desire to improve the performance of condensers has resulted in the development of method such as the use of heat transfer augmentation devices in the condensing equipment.

The use of augmentative techniques, either active or passive, to increase the convective heat transfer rates on tube side has been studied for quite some time. Seigel (1946), Bergles (1976, 1978), Bergles *et al.* (1983), Lazarek (1980), and Schlager *et al.* (1987). One of the passive techniques commonly considered is the use of various types of insertions, such as twisted tape, wire mesh or brush and coil or spiral spring. The tube inserts are relatively low in cost, easy to install inside the tube and to take out for cleaning operations. This is the reason why the bulk of the research in the area of augmentation of heat transfer during condensation pertains to this class.

A review of the existing literature revealed that numerous investigations have been carried out to study the effect of different passive techniques on augmentation of heat transfer coefficient during condensation of vapours inside horizontal tubes. However, frugal work has been published on using helically coiled wires inside condenser tubes. Although the use of coiled wire inserts in single phase flow and boiling liquids has been studied extensively and has been found to increase the heat transfer coefficients significantly, Chiou (1987) and Larson *et al.* (1949), the published work on its application for condensation of refrigerant vapours is scantily available. Moreover, it is also realised that the use of coiled wire inserts for condensation augmentation will be more effective than the inserts, like twisted tape, which generate rotating flow.

In the condensers which are used for refrigeration and air-conditioning applications, due to the high wettability of the refrigerants, only filmwise condensation is observed. Since the liquid thermal conductivity is low, heat transfer coefficients on the refrigerant side of the condenser are modest. So, augmentation of these coefficients in condensers used in refrigeration and air-conditioning systems is advantageous and more important than the steam condensers.

The present investigation was, therefore, undertaken to generate more heat transfer data on the R-22 condensing heat transfer rates inside horizontal copper tubes with helically coiled wire inserted inside it and to study the consequent heat transfer augmentation effects.

## 2. LITERATURE REVIEW AND INFORMATION GAPS

Table 1, summarises some important augmentation of in-tube condensation heat transfer studies using different passive techniques that have been published till this date. After the Montreal Protocol 1987, the scientists have focused their attention on testing the efficiency of heat transfer augmentation techniques with ecologically less hazardous refrigerants like R-22. Even though refrigerant-22 is also to be phased out in the long-term, by the year 2030 (Miro and Cox 1993), its easy availability and already successful use in ice plants and cold storages has not abated research in this direction. Most of the papers on condensation augmentation pertains to non-refrigerants and refrigerants other than R-22. Some of the recent papers on condensation augmentation which have come up with this refrigerant are Schlager *et al.* (1989, 1990), Koyama *et al.* (1990), Lal (1992) and Chiang (1993).

Schlager *et al.* (1989, 1990) studied the heat transfer and pressure drop performance during evaporation and condensation of R-22 inside horizontal micro-finned tubes. The average condensing heat transfer coefficients for such tubes were found to be 1.5–2.0 times greater than those for the plain smooth tubes on a nominal equivalent tube area basis. Pressure drop also increased, but by a smaller margin in comparison to the heat transfer coefficient. No correlation was proposed by them.

Koyama *et al.* (1990) investigated the condensation of pure and mixed refrigerant of R-22 and R-114 inside a spirally grooved horizontal tube. The local Nusselt number of pure refrigerants for a grooved tube was 60% higher than that for smooth tube; however, the pressure drop also increased by about 30%. To correlate their experimental data, they modified Fujii and Nagata (1973) empirical correlation, which was obtained for the condensation of pure organic substances inside a horizontal smooth tube. This showed that the internal spiral grooves were effective in enhancing condensation.

Lal (1992) investigated the effect of twisted tape inserts on augmentation of in-tube condensation of R-22 inside 12.7 mm I.D. horizontal tubes. The insertion of full length tapes inside the test sections increased the heat transfer coefficients from 9–26% above those of the plain tube on a nominal area basis. The maximum enhancement was for the lowest twist ratio of $y = 5.9$, where the twist ratio $y$ of the tape was defined as the ratio of half of the pitch of the helix to the inside tube diameter. For tubes with twisted tape inserts, Lal modified the smooth tube correlation of Akers and Rosson (1960) to develop a correlation equation.

Chiang (1993) investigated the heat transfer performance of R-22 flowing in axial and helical micro-fin tubes. Six horizontal test sections, each having a length of 1.83 m, were connected in series so the total heat transfer test length was about 11 m. Axial and helical micro-fin tubes of 7.5 and 10 mm O.D. with different fin apex angle, fin helix angle and number of fins were used. The experimental results indicated that for both axial and helical micro-fin tubes, the convective condensing heat transfer coefficient increased fairly linearly with the vapour quality. The axial micro-fin tubes yielded 10–20% higher condensing heat transfer coefficients over those of helic micro-fin tubes for all the conditions tested. No comparison of the experimental results with any smooth tube data was made and also no correlation was proposed.

## 3. EXPERIMENTAL PROGRAMME

The schematic diagram of experimental set-up is shown in figure 1. It was basically a well instrumented vapour compression refrigeration system. The test-condenser consisted of four separate coaxial double pipe condensers assembled in series and labelled A, B, C and D, as shown in figure 1. For each test section the inner tube was a hard drawn copper tube of 12.7 mm I.D., 15.4 mm O.D. and 85 cm length, and the outer tube was also a hard drawn copper tube of

Table 1. Augmentation of condensation heat transfer inside horizontal tube (except coiled wires)

| S. No. | Author(s) and Date | Augmentation technique | Turbulator twist ratio, No. of fins | Fluid | Test section Dimn (cm) Outside Dia. | Inside Dia. | Equi. Dia. | Leng. |
|---|---|---|---|---|---|---|---|---|
| 1 | Royal and Bergles (1976, 1978) | Twisted tape | 3.3, 7 | Steam | 1.5875 | 1.3843 | 0.825 | 91 |
| 2 | Luu and Bergles (1980) | Twisted tapes | 5.56, 9.2 | R-113 | 1.5875 | 1.3386 | 0.8202 | 91.4 |
| 3 | Said and Azer (1983a) | Twisted tape | 5.5, 9.2 | R-113 | 1.5875 | 1.3843 | — | 61 |
| 4 | Lal (1992) | Twisted tape | 5.5, 9.15 | R-22 | 1.580 | 1.270 | — | 95 |
| 5 | Royal and Bergles (1976, 1978) | Internal fin | 6, 6, 16, 32 | Steam | 1.2776–1.5900 | 1.1532–1.4707 | 0.6767–0.8260 | 91 |
| 6 | Luu and Bergles (1980) | Internal fin | 6, 16, 32 | R-113 | 1.2784–1.5900 | 1.1811–1.4770 | 0.6767–0.8260 | 91.4 |
| 7 | Said and Azer (1983b) | Straight and spiral fins | 10, 16, 16, 16 | R-113 | 1.5875–2.6568 | 1.4199–2.5375 | 1.360–2.457 | 61 |
| 8 | Venkatesh and Azer (1985) | Straight and spiral fins | 10, 16, 16, 16, | R-11 | 1.5875–2.6568 | 1.4199–2.5375 | 1.360–2.457 | 61 |
| 9 | Schlager *et al.* (1990) | Micro-fins | 60, 70 | R-22 | 0.592–1.270 | 0.892–1.170 | — | 367 |
| 10 | Chiang (1993) | Micro-fins | | R-22 | 0.75 | 0.706–0.936 | — | 183 |
| 11 | Lin *et al.* (1981) | Static mixer | | R-113 | — | 0.635–1.90 | — | 101<br>140 |
| 12 | Luu and Bergles (1981) | Repeated rib | | R-113 | 1.5875 | 1.4605 | — | 91 |
| 13 | Luu and Bergles (1981) | Spirally flutd | | R-113 | 1.5875<br>1.7856 | 1.0693<br>1.6586 | — | 91 |
| 14 | Koyama *et al.* (1990) | Spiral groove | | R-22<br>R-114 | 0.952 | 0.832 | — | 40 |

Range of operating parameters

| S. No. | Author(s) and date | Pres. (bar) | mass flux (kg/s m$^2$ | heat flux (kW/m$^2$) | Outlet quality | Heat transfers coefficient (W/m$^2$ K) | ER = $h_s/h_p$† |
|---|---|---|---|---|---|---|---|
| 1 | Royal and Bergles (1976, 1978) | 2.8–5.6 | 150–583 | 220–1400 | 0.019–0.041 | 19 000–110 000 | 30% |
| 2 | Luu and Bergles (1980) | 2.41–6.55 | 86–760 | 11–88 | 0.052–0.060 | 1056–7382 | 30% |
| 3 | Said and Azer (1983a) | 1.32–3.05 | 44.8–281.5 | — | 0.03–0.20 | 645–2091 | 23% |
| 4 | Lal (1992) | 14.8–20.4 | 209–372 | 4.9–2.9 | sub. | 1440–2290 | 26% |
| 5 | Royal and Bergles (1976, 1978) | 2.8–5.6 | 150–583 | 220–1400 | 0.019–0.041 | 19 000–110 000 | 150% |
| 6 | Luu and Bergles (1980) | 2.41–6.55 | 86–760 | 11–88 | 0.052–0.060 | 1056–7382 | 120% |
| 7 | Said and Azer (1983b) | 1.32–3.05 | 14.14–305.9 | — | 0.233–0.001 | 343–2716 | 51% |
| 8 | Venkatesh and Azer (1985) | 1.27–1.73 | 17.14–85.55 | — | 0.158–0.000 | 1044–3456 | 55% |
| 9 | Schlager *et al.* (1990) | 15–16 | 75–400 | — | 0.10–0.20 | 2500–5200 | 200% |
| 10 | Chiang (1993) | 15.53 | 270–1100 | — | sub/ | 5500–13 000 | — |
| 11 | Lin *et al.* (1981) | 1.37–2.61 | 10.55–33.05 | 31.7–98.1 | — | Nu = 200–600 | 85% |
| 12 | Luu and Bergles (1981) | 2.41–6.55 | 90–530 | — | — | 1700–4750 | 80% |
| 13 | Luu and Bergles (1981) | 2.41–6.55 | 140–715 | — | — | 1650–5200 | 50% |
| 14 | Koyama *et al.* (1990) | 3–21 | 133–358 | — | — | Nu = 700–1000 | 60% |

†ER = enhancement ratio, $h_s$ = heat transfer coefficient when coiled wire fitted inside the tubes, $h_p$ = heat transfer coefficient in plain tube (no coil inside the tube)

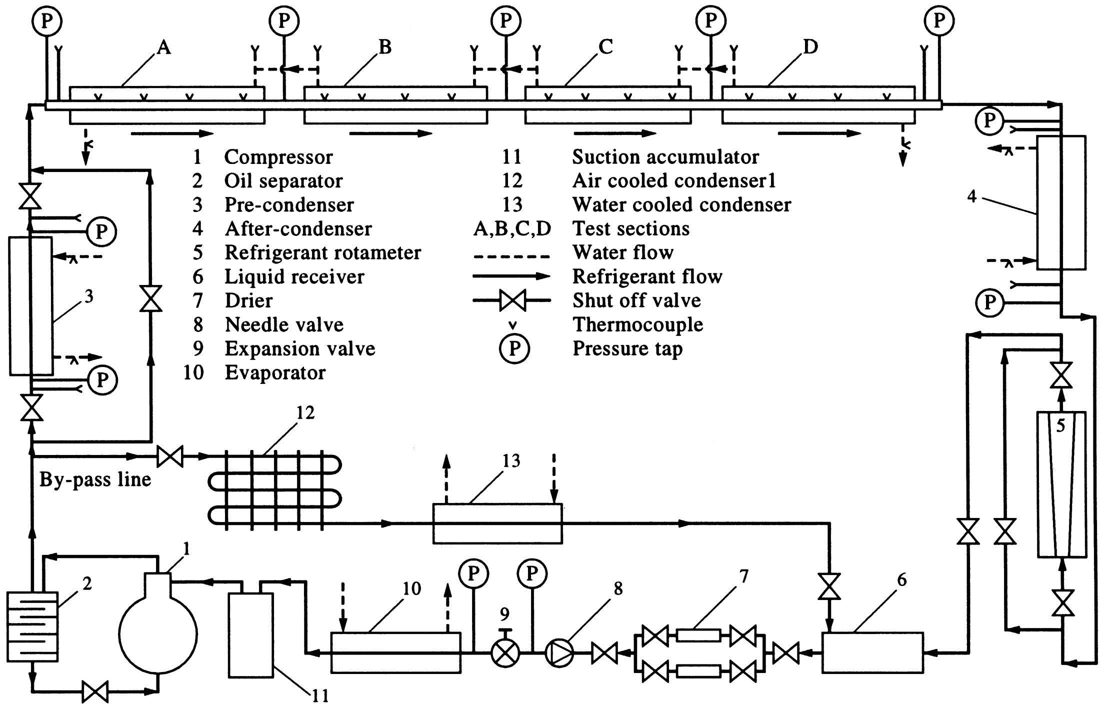


Figure 1. Schematic diagram of experimental set-up.

equal length having an I.D. of 50 mm. It was a counter-flow heat exchanger where refrigerant-22 condensed inside the inner tube by rejecting heat to the cooling water flowing in the annulus. The four tubes were instrumented to obtain the outside wall temperatures of the inner tubes. To estimate a better average value, the outside wall temperatures of each tube were measured at four axial locations. At each of these four stations, three thermocouples were each located at the top, side and bottom of the tube to take care of any circumferential temperature variation. The refrigerant temperatures at the inlet and outlet of the test-condenser and the coolant water temperatures at the inlet and outlet of each test section were measured by thermocouples. Arrangements were also made to measure refrigerant inlet and outlet static pressures of the test-condenser. The whole test-condenser was well insulated with glass wool to effectively prevent heat leakage.

To obtain desired vapour quality at the inlet of the test-condenser, so as to cover the entire range of vapour quality, a pre-condenser was installed upstream of the test-condenser. The flow rate of water through the pre-condenser could be regulated in order to control the state of refrigerant vapour entering the test condenser. It was ensured that the refrigerant coming from the test condenser is completely condensed before it entered the rotameter by installing an after-condenser downstream of the test-condenser. Provisions were made to measure temperatures and static pressures at the inlet and outlet of both the pre-condenser and after-condenser. They were also well insulated with glass wool to prevent any appreciable heat loss.

Refrigerant mass flow rate was measured by a pre-calibrated rotameter installed down stream of the after-condenser. Water flow rates through the test-condenser, pre-condenser and after-condenser were also measured by pre-calibrated rotameters. All temperatures were measured by copper–constantan thermocouples. A data logging system of 100 channels was used for display and printing out of temperatures directly in degree Celsius with a least count of 0.01°C.

For the augmentation of heat transfer, coiled wire inserts were employed. Coiled wires with three different wire diameters of 0.65, 1.0 and 1.5 mm and three different coil pitches of 6.5, 10 and 13 mm were made and used in full length of the test-condenser.

## 4. DATA COLLECTION AND DATA REDUCTION

The condensation of compressed vapour is established in the pre-condenser, test sections and after-condenser. The liquid then flowed into the receiver, through the drier and manually operated expansion valves, to the evaporator and suction accumulator. The vapour was then sucked by the compressor. An over-sized oil separator was used in the discharge line just after compressor. Though the lubricating oil content in the refrigerant was not measured, it is expected that the bigger size oil separator would keep it relatively free from oil.

During each test run, three sets of data were taken at intervals of 10 min each after the system reached steady-state. An average of three readings was used for further analysis. Altogether, 140 test runs with five different refrigerant mass velocities of 206, 236, 280, 327 and 372 kg/s m$^2$ were performed for plain flow and for flows in roughed tubes (tubes with coiled wire inserts). The range of operating parameters is as follows:

| | |
|---|---|
| Working fluid | R-22 |
| Refrigerant mass velocity | 200–372 kg/s-m$^2$ |
| Inlet degree of superheat | 0–7°C |
| Average condensing temperature | 34–48°C |
| Average cooling heat flux | 9000–28000 W/m$^2$ |
| Coolant water mass flow rate | 30–1500 kg/h |
| Coolant water temperature | 15–30°C |
| Inlet vapour quality | 0.09–1.0 |
| Twist ratio ($p/D$) of the coils = pitch of the coil/inside tube diameter | 0.50–1.02 |
| Wire thickness ratio ($e/D$) = wire diameter/inside tube diameter | 0.052–0.118 |

Saturation temperature and superheated and saturation enthalpies of R-22 were calculated from equations given by Cleland (1986). Surface tension was evaluated from the "Thermophysical properties of R-22, Japanese Association of Refrigeration" (1975). Other thermophysical properties of R-22 were computed from equations given by Charters and Sadafi (1987).

For each test tun, four sectional heat transfer coefficients were calculated by [1]. It was done on the basis of heat gained by water in that test section and the temperature difference between inside wall and condensing refrigerant.

$$h = \left[\frac{\pi D L(t_s - t_{wo})}{m_w C_{p_w}(T_{wo} - T_{wi})} - \frac{D}{2k_w}\ln(D_o/D)\right]^{-1} \quad [1]$$

where $D$ and $D_o$ are the inside the outside tube diameters, $L$ is the length of one test-section and $k_w$ is the thermal conductivity of the tube material.

The overall inside heat transfer coefficient for the entire test-condenser was also calculated by using [1], while $L$ in these equations was replaced by $4L$, the combined length of the four test sections. $t_{wo}$ was taken as the arithmetic mean of outside tube wall temperatures of four test sections. $t_s$ was taken as the saturation temperature corresponding to the average pressure of R-22 in the entire test-condenser. ($T_{wo}$–$T_{wi}$) represents the temperature rise of coolant water for the entire tests-condenser while $m_w$ and $C_{pw}$ are its mass flow rate and the specific heat.

The local vapour quality at the inlet and outlet of each test section was calculated by making energy balance in that section (heat gained by coolant = heat rejected by R-22). The sectional vapour quality of each test section was obtained by taking the average of the vapour qualities at its inlet and outlet.

## 5. RESULTS AND DISCUSSION

Experimental data for tubes with coiled wire inserts (roughed tubes) were obtained for three different wire diameters of 0.65, 1.0 and 1.5 mm with a pitch of 10.0 mm and three different coil pitches of 6.5, 10 and 13 mm for the wire of 1.0 mm diameter. Helix angles for all coiled wires

Table 2. Characteristic dimensions of coiled wire inserted tubes

| Tube set | D mm | Do mm | De mm | L mm | e mm | P mm | $Y = ((e^2/pD) \times 1000)$ | α deg |
|---|---|---|---|---|---|---|---|---|
| A | 12.7 | 15.4 | 10.51 | 850 | 0.65 | 10.0 | 3.327 | 79 |
| B | 12.7 | 15.4 | 10.13 | 850 | 1.0 | 13.0 | 6.057 | 77 |
| C | 12.7 | 15.4 | 9.560 | 850 | 1.0 | 10.0 | 7.784 | 77 |
| D | 12.7 | 15.4 | 8.460 | 850 | 1.0 | 6.50 | 12.11 | 80 |
| E | 12.7 | 15.4 | 8.610 | 850 | 1.5 | 10.0 | 17.72 | 78 |
| F | 12.7 | 15.4 | 15.40 | 850 | J—— | | Smooth tube→ | |

were nearly same. The characteristic parameters, which define the roughness geometry of coiled wire inserted tubes are given in table 2. It may be noted that experimental tests have been conducted and described on the basis of tube sets, named 'A', 'B', 'C', 'D', 'E' and 'F', each corresponding to a particular tube geometry.

The effects of various parameters viz. mass velocity, vapour quality and coiled wire specifications on heat transfer coefficients have been studied and are presented in the following sections.

First, the experimental heat transfer coefficients of the present investigation were correlated with the help of several existing correlations. The best agreement could be obtained by using [2], which is a modification of Boyko and Kruzhilin (1967) correlation.

$$\overline{Nu} = 0.81 Pr^{1/3} (Re)^{0.66} \left\{ \frac{(\rho_L/\rho_m)_{in}^{0.5} + (\rho_L/\rho_m)_{out}^{0.5}}{2} \right\}^{0.90} \quad (\Delta X \cdot D/L)^{0.29} \qquad [2]$$

where

$Nu =$ average Nusselt number $= h\, D/k$

$Pr =$ liquid Prandtl number $= \mu C_p/k$

$Re =$ liquid Reynolds number $= GD/\mu$

$G =$ liquid mass velocity considering total refrigerant is flowing inside the tube as liquid

$\mu, C_p, k =$ absolute viscosity, specific heat and thermal conductivity of liquid refrigerant calculated at the saturation temperature corresponding to the average pressure existing in the test section length

$\Delta X/L =$ change in vapour quality inside the test section per unit of its length.

$$\rho_L/\rho_m = 1 + X(\rho_L - \rho_G)/\rho_G \qquad [3]$$

in which $\rho_L$, $\rho_G$, $\rho_m$ are the densities of liquid, vapour and mixture (condensate + vapour) phases as defined by Boyko and Kruzhilin (1967).

Figure 2 is presented as a typical graph to show the random variation of sectional heat transfer coefficient with sectional vapour quality. It is plotted for tube sets 'A' and 'C' for the same mass velocity of 206 kg/s m$^2$. The points are joined by straight lines. The plain flow curve given by our developed correlation, i.e. [2], is also drawn on this figure. From this figure, random variation of heat transfer coefficients is clearly discerned. The reason is that the forced convection condensation is a highly unstabilized phenomenon because the two phases are flowing together accompanied by phase transformation. Further, the presence of wire at the tube periphery will also have an influence on the instability of condensate film formed over the wall.

The variation of heat transfer coefficient with vapour quality for roughed tubes is shown in figures 3–6. Each of the figures is for one mass velocity. The sectional heat transfer coefficients and sectional vapour qualities were used to draw these graphs with the tube geometry as a parameter. These graphs will be evaluated in the following lines.

### 5.1. *Heat transfer augmentation*

On the basis of closer examination of figures 3–6 and the entire data collected for roughed tubes, it is noted in general that the insertion of helically coiled wires inside the tubes significantly increases the heat transfer coefficients during the forced convection condensation of R-22. The gain in heat transfer coefficient is not very consistent for a particular coiled wire and it is a

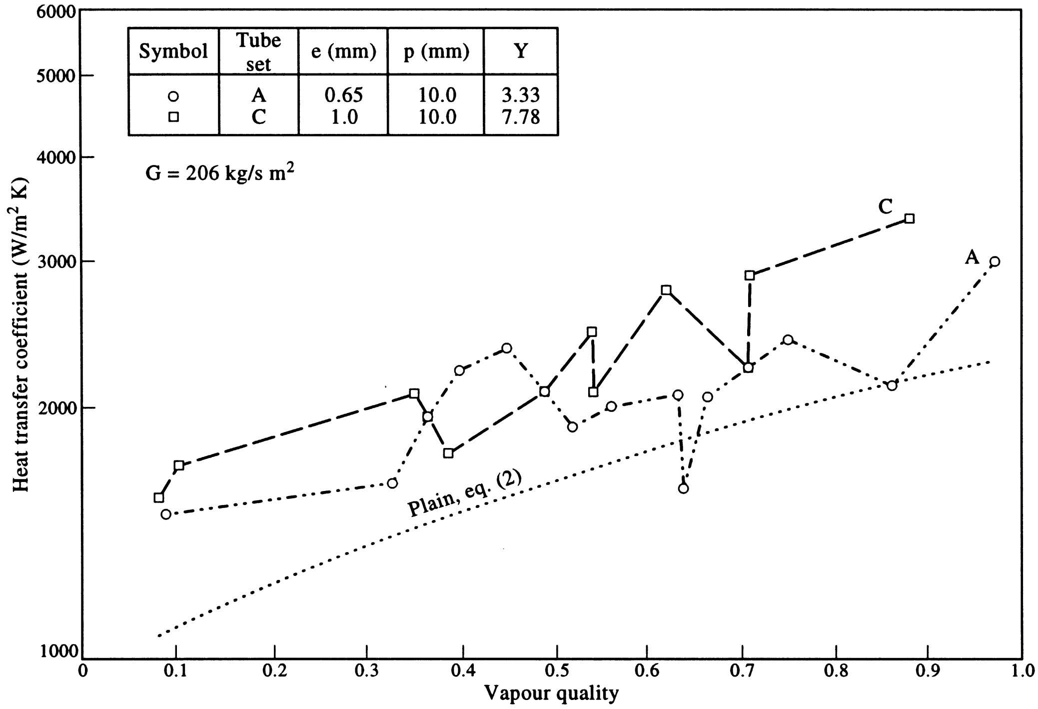


Figure 2. Comparison of heat transfer coefficients for tube sets ‘A’ and ‘C’ at 206 kg/s m$^2$ refrigerant mass velocity.

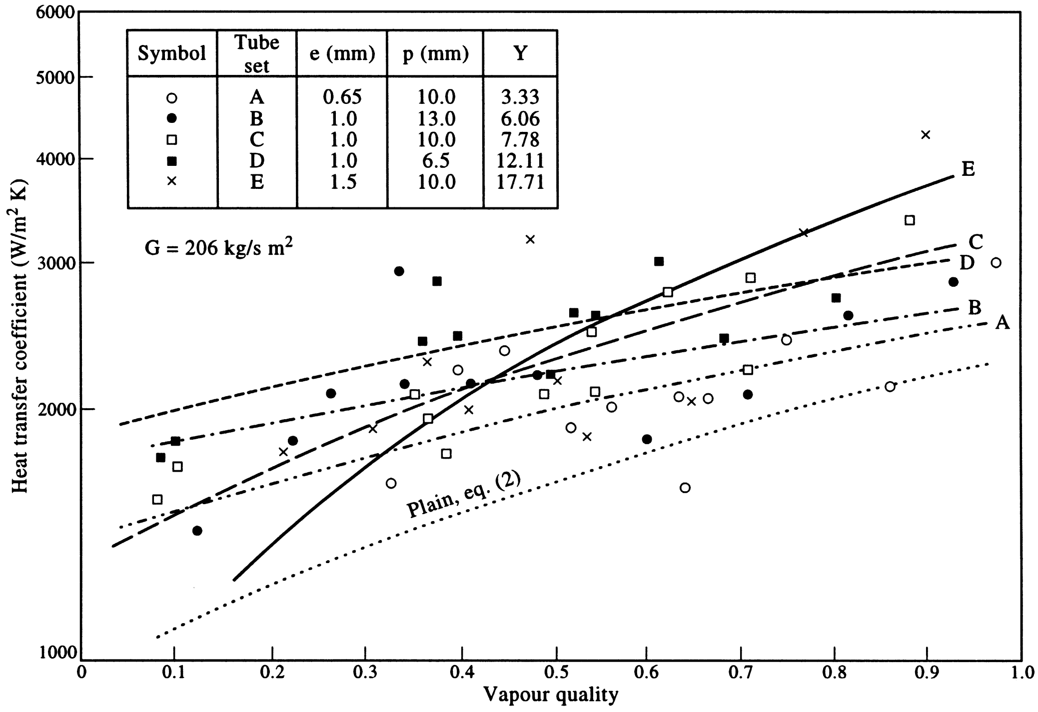


Figure 3. Comparison of heat transfer coefficients for tubes with coiled wire inserts at 206 kg/s m$^2$ refrigerant mass velocity.

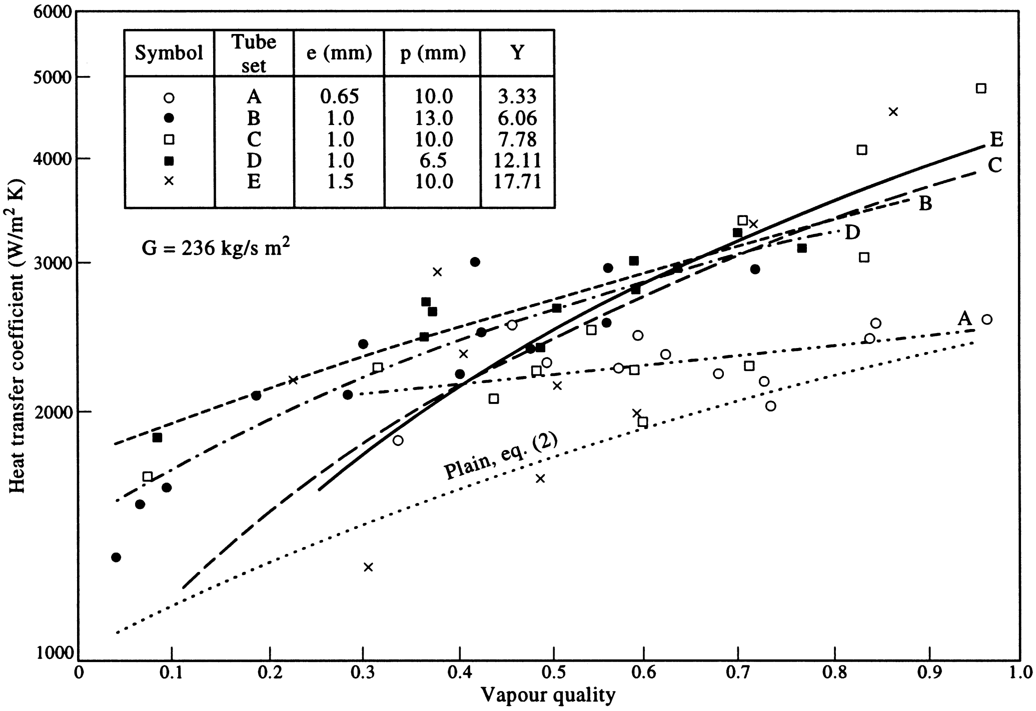


Figure 4. Comparison of heat transfer coefficients for tubes with coiled wire inserts at 236 kg/s m$^2$ refrigerant mass velocity.

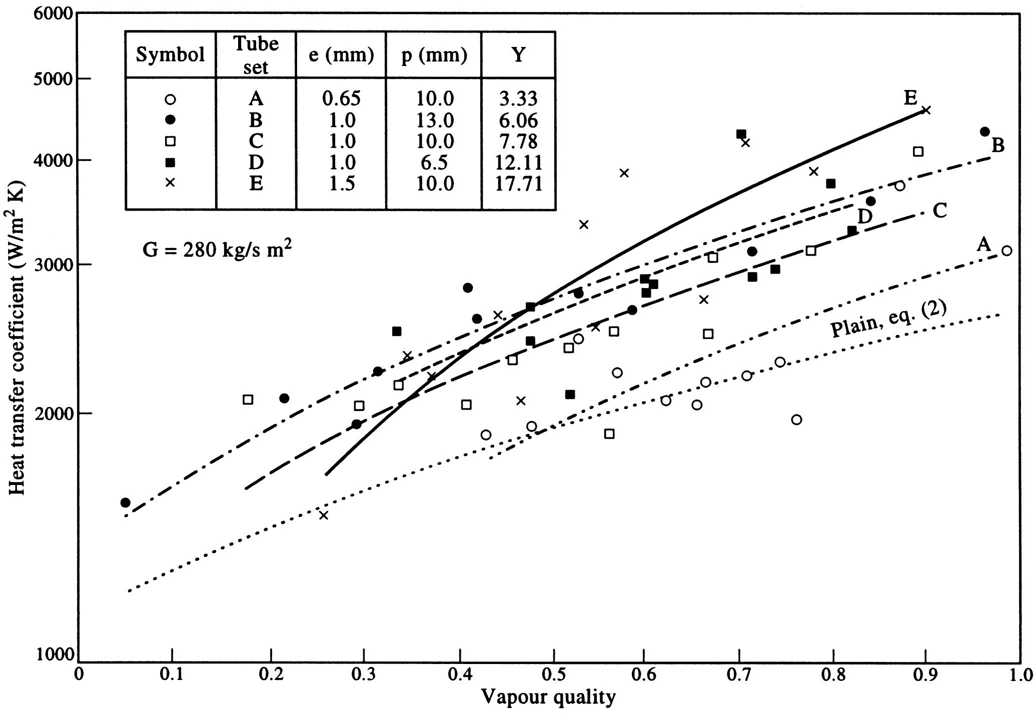


Figure 5. Comparison of heat transfer coefficients for tubes with coiled wire inserts at 280 kg/s m$^2$ refrigerant mass velocity.

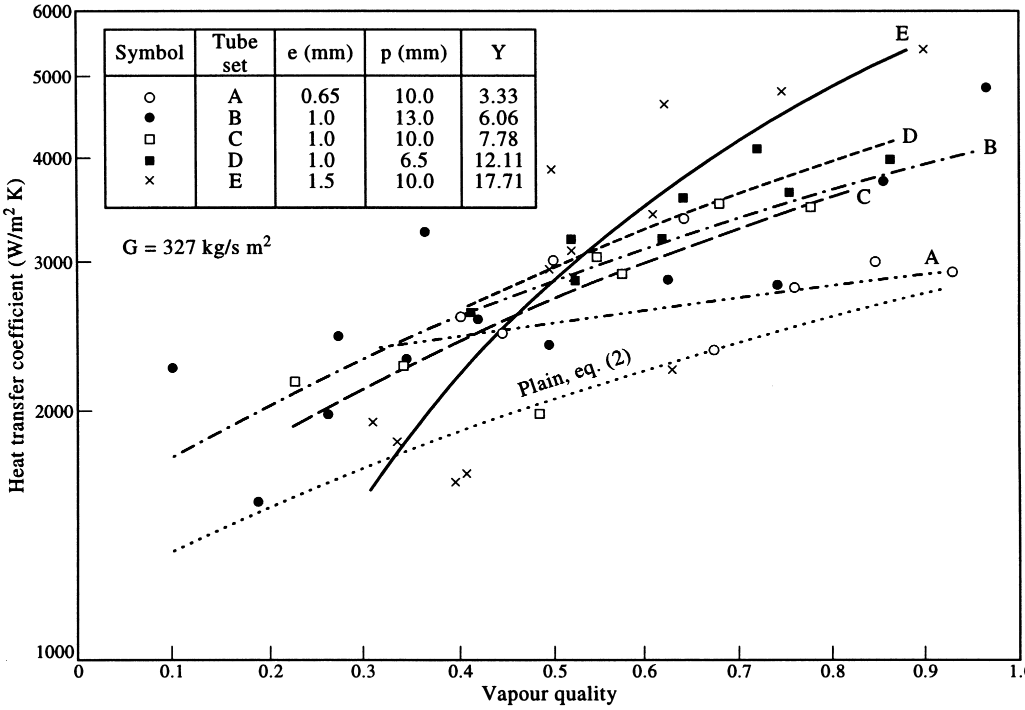


Figure 6. Comparison of heat transfer coefficients for tubes coiled wire inserts at 327 kg/s m$^2$ refrigerant mass velocity.

complex function of mass velocity, vapour quality and physical dimensions of the coil. The best performing roughed tube in the high vapour quality region (tube set 'E') increased the sectional heat transfer coefficient by as much as 100% above the plain tube values on a nominal area basis.

5.2. *Effect of vapour quality*

In figures 3–6, it is also observed that the heat transfer coefficient decreases with the decrease of vapour quality for all mass velocities. The sectional heat transfer coefficients of the first test section was the highest among all and usually decreased as the condensation progressed in the direction of flow for the three following test sections. The reason is that at high vapour quality, the liquid film on the inner tube wall is thinner offering lower thermal resistance. Similar observations were reported for internally finned tubes and those with twisted tapes insert by Said and Azer (1983a, 1983b).

5.3. *Effect of mass velocity*

Figure 7 represents the variation of heat transfer coefficient with vapour quality for tube set 'C' using mass velocity as a parameter. By comparison of figures 3–6 and also from figure 7 it is found that, in general, the heat transfer coefficient increases with the increase of mass velocity when other parameters remain unchanged. However, in certain regions of low vapour quality, a reverse trend is seen. Presumably these points lie near flow transition zone from annular to wavy. The effect of mass velocity is well understood and the observed trend is in conformity with the usual behaviour for plain flow.

5.4. *Effect of coiled wire characteristics*

The effect of helically coiled wire inserts on the heat transfer coefficient can be studied again with the help of figures 3–6. These figures show that the insertion of a coiled wire inside the condenser tube has produced higher heat transfer coefficients compared to plain flow. It was also seen that in the high vapour quality region, the best performing roughed tube (tube set 'E')

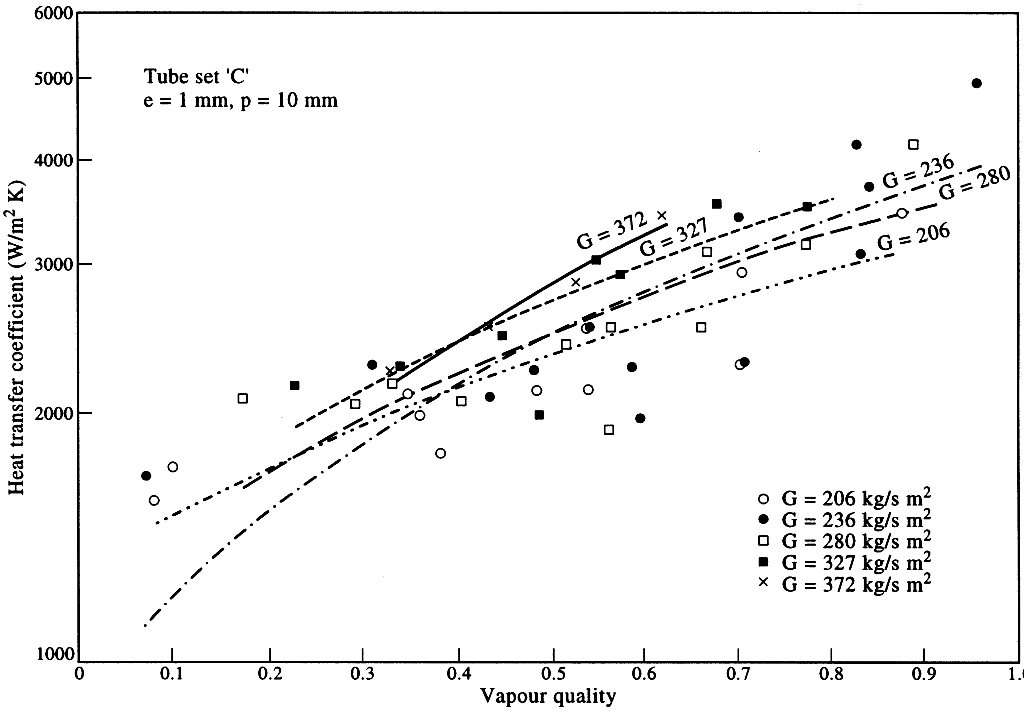


Figure 7. Variation of heat transfer coeffficients and vapour quality for different refrigerant mass velocities (Tube Set 'C').

increased the sectional heat transfer coefficient by as much as 100% above the plain flow values. Characteristic features of condensation inside roughed tubes are discussed by considering figure 3 which is for the lowest mass velocity of 206 kg/s $m^2$. The graph shows the comparison between heat transfer coefficients of flow inside tube sets 'A', 'B', 'C', 'D' and 'E' (roughed tubes), and plain tube in a wide range of vapour quality.

For the same mass velocity and same coil pitch of 10 mm (tube sets 'A', 'C' & 'E'), at high vapour qualities, the maximum heat transfer enhancement was achieved for tube set 'E' with thickest wire of 1.5 mm diameter, while for low vapour qualities the trend reverses and the maximum augmentation was gained generally in the case of tube set 'A' with thinnest coiled wire of 0.65 mm diameter. For the same mass velocity and same wire diameter of 1 mm (tube sets 'B', 'C' and 'D'), for high vapour qualities, the heat transfer enhancement was observed to be a complex function of coil pitch and no particular pitch produced maximum enhancement. However, at low vapour qualities, it exhibited a general trend to give maximum enhancement for tube set 'D' with lowest coil pitch of 6.5 mm.

With the help of equations given by Soliman (1982) and by using the present roughed tubes experimental data, in the case of annular flow the thickness of liquid film on the tube inside wall was calculated. The results show that the liquid film thickness in annular flow reaches around 0.7 mm in the case of lowest mass velocity and about 1.1 mm at highest mass velocity before the flow regime changes from annular to wavy. The transition usually happens at vapour qualities between 0.4 and 0.5. In the case of roughed tubes, it is expected that due to the presence of coiled wire at the tube periphery, decrease of liquid film velocity results. Because of this as well as due to increased surface tension effects, the liquid film thickness in roughed tube is expected to be more than for plain flow before the flow regime changes from annular to wavy. It means that at very low vapour qualities also the flow pattern may be annular with a film thickness of even more than the thickest coiled wire, i.e. 1.5 mm.

The reason for achievement of maximum heat transfer coefficients at high vapour quality is that a portion of coiled wire is out of liquid film adjacent to the tube wall and so in vapour core; therefore, the coiled wire not only acts as partitions and interrupts the development of laminar sub-layer, but it will also highly increase the liquid film turbulence. As the condensation progresses, the liquid film thickness increases and at low vapour quality, the thick coiled wire is completely submerged inside the liquid film. In this condition, it will act as a thick liquid wall and will retard the turbulence induced in the liquid film. At low vapour qualities, the thin coiled wire performance is better, since it will interrupt the development of laminar sub-layer and allow more turbulence of liquid film.

### 5.5. *Comparison of coiled wire inserts with twisted tapes*

Chiou (1987) investigated the effect of coiled wire inserts during cooling of oil inside tubes. In describing the reasons for using coiled wire inserts as turbulator in cooling, he has stated that the principal mechanism for heat transfer enhancement in tubes with coiled wire inserts is due to the disruption of the laminar sub-layer of liquid film and increasing the degree of flow turbulence; coiled wires usually do not generate rotating flow. Chiou has also noted that in the heating mode operation, the rotating flow has favourable centrifugal convection effect, which can substantially increase the heat transfer coefficient between the flow and the tube wall. In the cooling mode operation, however, the rotating flow may have an adverse centrifugal convection effect which may even reduce the convection effect. Finally, Chiou concluded that due to the above reason, the tube inserts which can generate rotating flow (such as twisted tapes) are generally not used in oil coolers; instead, coil or spiral springs are used.

Apparently the better performance of coiled wire promoters than twisted tape turbulators during condensation is governed by the above reason. Several investigators have used twisted tapes as turbulence promoters during condensation of vapours inside horizontal tubes. The specifications of these investigations and corresponding heat transfer coefficient enhancements have been listed in table 1. From this table it is evident that the heat transfer coefficient enhancement of all these researches are less than heat transfer enhancement of the present investigation conducted with coiled wire inserts instead of twisted tape.

Lal (1992) investigated the effect of twisted tape inserts during condensation of R-22 inside a 12.7 mm I.D. horizontal copper tube in the same laboratory where present investigation has

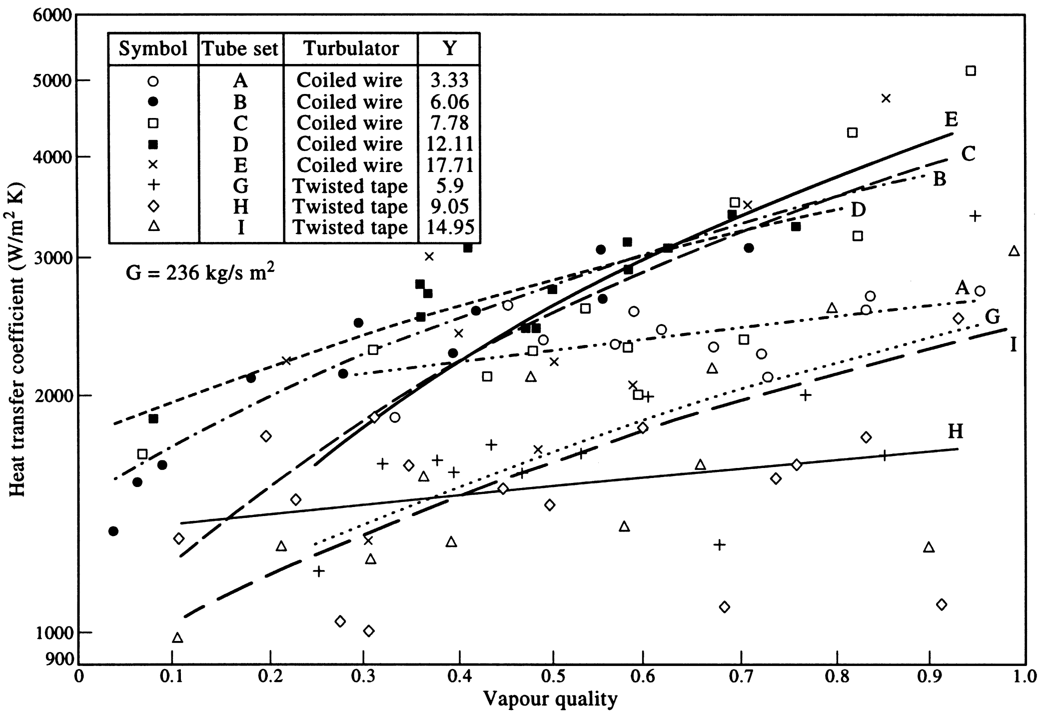


Figure 8. Comparison of heat transfer coefficients for different coiled wire inserted tubes of the present investigation with data of Lal (1992) for twisted tapes.

been conducted. Figure 8 shows the data for coiled wires of figure 4, along with the experimental data of Lal (1992), to bring out the relative performance of these two types of inserts. The data of heat transfer coefficients for twisted tapes used in this graph corresponds to twist ratios of 5.9, 9.05, and 14.95, designated by tube sets 'G', 'H' and 'I' in the figure. Both data of the present investigation and that of Lal used in figure 8 were collected at the same mass velocity of 236 kg/s $m^2$ and condensing pressure range of about 15 to 18 bar. From this figure it is evident that the heat transfer coefficients for all tubes fitted with coiled wire inserts, i.e. tube sets 'A', 'B', 'C', 'D' and 'E' are greater than the corresponding heat coefficients of tube sets 'G', 'H', and 'I' with twisted tape inserts.

## 6. CONCLUSIONS

On the basis of the study of the entire data, the following inferences are drawn regarding the effect of coiled wire inserts on the heat transfer coefficients.

1. The insertion of helically coiled wires inside horizontal tubes was found to increase the condensing heat transfer coefficients by as much as 100% above the plain tube values on a nominal area basis.
2. For the same coil pitch while other conditions remained unchanged and at high vapour qualities, the maximum heat transfer coefficient enhancement was achieved for tube with thickest wire but, for low vapour qualities, the maximum enhancement was gained generally in tube with thinnest coiled wire. The transition occurs at vapour quality of around 0.4.
3. For the same wire diameter while other conditions remained unchanged and for high vapour qualities, the heat transfer enhancement was observed to be a complex function of coil pitch and no particular pitch produced maximum augmentation. However, at low vapour qualities it exhibited a general trend to give maximum enhancement for tube with lowest coil pitch.

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## APPENDIX A

*Uncertainty Analysis*

In this investigation many parameters were measured by instruments having various degree of accuracy. The measurement errors of these parameters affect the accuracy of the heat transfer coefficients. Therefore, it is of considerable importance to examine the combined effect of uncertainty in each variable on the heat transfer coefficient values. The experimental uncertainty is defined as the absolute value of the maximum expected deviation from the reported experimental value. Schultz and Cole (1979) suggested the following method of analyzing the effect of uncertainty in each variable on the uncertainty of the result.

$$U_{\mathrm{R}} = \left[ \sum_{i=1}^{n} \left( \frac{\partial R}{\partial Vi} U_{vi} \right)^2 \right]^{1/2} \quad \text{[A1]}$$

where, $U_R$ is the estimate of the uncertainty in the calculated value of the desired variable, $R$, due to the independent uncertainty, $U_{vi}$, in the primary measurements of $n$ numbers of variables, $Vi$, affecting the results. Using [A1], the experimental uncertainty of heat transfer coefficient is given by

$$U_{\mathrm{h}} = \left[ \sum_{i=1}^{n} \left( \frac{\partial h}{\partial Vi} U_{vi} \right)^2 \right]^{1/2} \quad \text{[A2]}$$

where, $n$ is the number of variables whose errors of measurement affect the uncertainty of heat transfer coefficient.

In the present investigation the experimental value of heat transfer coefficient was calculated by using the following equation:

$$h = \frac{q}{(t_{\mathrm{s}} - t_{\mathrm{wi}})}. \quad \text{[A3]}$$

Applying [A2], the uncertainty in the heat transfer coefficient calculation can be written as,

$$U_{\mathrm{h}} = \left\{ \left( \frac{U_{\mathrm{q}}}{(t_{\mathrm{s}} - t_{\mathrm{wi}})} \right)^2 \left( \frac{-q \cdot U_{\mathrm{ts}}}{(ts - t_{\mathrm{wi}})^2} \right)^2 \left( \frac{q \cdot U_{\mathrm{twi}}}{(t_{\mathrm{s}} - t_{\mathrm{wi}})^2} \right)^2 \right\}^{1/2} \quad \text{[A4]}$$

where, $U_{\mathrm{q}}$, $U_{\mathrm{ts}}$ and $U_{\mathrm{twi}}$ are the uncertainties for quantities $\boldsymbol{q}$, $\boldsymbol{t}_{\mathrm{s}}$, and $\boldsymbol{t}_{\mathrm{wi}}$, respectively. These can be estimated as follows:

(a) Evaluation of $U_{\mathrm{q}}$

The radial heat flux, $q$, for a test section was obtained from

$$q = Q/A_{\mathrm{i}}. \quad \text{[A5]}$$

Hence,

$$U_{\mathrm{q}} = \left[ \left( \frac{1}{A_{\mathrm{i}}} \cdot U_{\mathrm{Q}} \right)^2 + \left( \frac{-Q}{A_{\mathrm{i}}^2} \cdot U_{\mathrm{Ai}} \right)^2 \right]^{1/2}. \quad \text{[A6]}$$

The heat flow is calculated as the heat gained by coolant water

$$Q = m_{\mathrm{w}} \cdot C_{\mathrm{pw}} \cdot \Delta T_{\mathrm{w}}. \quad \text{[A7]}$$

Therefore,

$$U_{\mathrm{Q}} = [(C_{\mathrm{pw}} \cdot \Delta T_{\mathrm{w}} \cdot Um_{\mathrm{w}})^2 + (m_{\mathrm{w}} \cdot \Delta T_{\mathrm{w}} \cdot U_{CP_{\mathrm{w}}})^2 + (m_{\mathrm{w}} \cdot C_{\mathrm{pw}} \cdot U_{\Delta T_{\mathrm{w}}})^2]^{1/2} \quad \text{[A8]}$$

where,

$$U_{\Delta T_{\mathrm{w}}} = [(U_{T_{\mathrm{wi}}})^2 + (U_{T_{\mathrm{wo}}})^2]^{1/2}.$$

Since, $U_{T_{\mathrm{wi}}} = U_{T_{\mathrm{wo}}} = U_{T_{\mathrm{w}}}$ (uncertainty in water temperature measurements) therefore,

$$U_{\Delta T_{\mathrm{w}}} = [2(U_{T_{\mathrm{w}}})^2]^{1/2}. \quad \text{[A9]}$$

For the working range, the coolant specific heat does not change appreciably. Therefore, it was assumed, $U_{CP_{\mathrm{w}}} = 0$.

The heat transfer area was calculated as

$$A_{\mathrm{i}} = \pi DL. \quad \text{[A10]}$$

Therefore,

$$U_{\mathrm{Ai}} = \left[(\pi D U_{\mathrm{L}})^2 + (\pi L U_{\mathrm{D}})^2\right]^{1/2} \quad \text{[A11]}$$

(b) Evaluation of $U_{\mathrm{ts}}$

The saturation temperature, $t_{\mathrm{s}}$, was calculated corresponding to the pressure of the condensing fluid. Therefore, the uncertainty of saturation temperature is given by

$$U_{\mathrm{ts}} = \left[\left(\frac{\partial t_{\mathrm{s}}}{\partial P}\right) U_{\mathrm{P}}\right]. \quad \text{[A12]}$$

The value of $(\partial t_{\mathrm{s}}/\partial P)$ was calculated from the refrigeration tables for the range of experimental parameter. It was found to be within 0.4°C/Psi. The measurement uncertainty of pressure was 0.5 Psi (0.034473 bar).

Therefore,

$$U_{\mathrm{ts}} = 0.4 \times 0.5 = 0.2\mathrm{C}.$$

(c) Evaluation of $U_{\mathrm{two}}$

The average outside tube wall temperature at a section, $t_{\mathrm{wo}}$ was calculated by

$$t_{\mathrm{wo}} = \frac{t_{\mathrm{T}} + t_{\mathrm{B}} + 2t_{\mathrm{si}}}{4}. \quad \text{[A13]}$$

Following the procedure of (b),

$$U_{\mathrm{two}} = [4(U_{\mathrm{t}}/4)^2]^{1/2} \quad \text{[A14]}$$

where, $U_{\mathrm{t}}$ is uncertainty in outside tube wall temperature measurements.

The average outside wall temperature of a test section, $t_{\mathrm{wo}}$, was calculated from the average of four axial stations.

$$U_{\mathrm{wo}} = \frac{\sum_i^4 t_{\mathrm{wo}}}{4}. \quad \text{[A15]}$$

Hence,

$$U_{\mathrm{two}} = [4(U_{\mathrm{two}}/4)^2]^{1/2}. \quad \text{[A16]}$$

The average tube inside temperature, $t_{\mathrm{wi}}$, was obtained by

$$t_{\mathrm{wi}} = t_{\mathrm{wo}} + \Delta t_{\mathrm{w}}. \quad \text{[A17]}$$

Hence,

$$U_{\mathrm{twi}} = [(U_{\mathrm{two}})^2 + (U_{\Delta\mathrm{tw}})^2]^{1/2}. \quad \text{[A18]}$$

Temperature drop across the tube wall, $\Delta t_{\mathrm{w}}$, was calculated as

$$\Delta t_{\mathrm{w}} = \frac{q \cdot D}{2k_{\mathrm{w}}} \ln(D_{\mathrm{o}}/D). \quad \text{[A19]}$$

Therefore,

$$U_{\Delta\mathrm{tw}} = \left\{\left[\frac{D}{2k_{\mathrm{w}}}\ln\left(\frac{D_{\mathrm{o}}}{D}\right)U_{\mathrm{q}}\right]^2 + \left[\left(\frac{q}{2k_{\mathrm{w}}}\ln\left(\frac{D_{\mathrm{o}}}{D}\right) - \frac{q}{2k_{\mathrm{w}}}\right)U_{\mathrm{D}}\right]^2 + \left[\frac{q \cdot D}{2k_{\mathrm{w}}}\frac{1}{D_{\mathrm{o}}}U_{\mathrm{Do}}\right]^2 + \left[\frac{q \cdot D}{2k_{\mathrm{w}}^2}\ln\left(\frac{D_{\mathrm{o}}}{D}\right)U_{\mathrm{kw}}\right]^2\right\}^{1/2}. \quad \text{[A20]}$$

Assuming,

$$B = \frac{q}{2k_w} \ln(D_o/D) \qquad [A21]$$

and,

$$C = q/2k_w. \qquad [A22]$$

Equation [A20] will become,

$$U_{\Delta tw} = \left\{ \left[ \frac{B \cdot D}{q} U_q \right]^2 + [(B - C)U_D]^2 + \left[ C \frac{D}{D_o} U_{Do} \right]^2 + \left[ \frac{B \cdot D}{q} U_{kw} \right]^2 \right\}^{1/2}$$

The uncertainty intervals for individual measurements are as follows:

| SI. No. | Measurement | Uncertainty | Remarks |
|---|---|---|---|
| (1) | Tube diameter | $\pm 0.1$ mm | Least count of vernier |
| (2) | Tube length | $\pm 0.5$ mm | L.C. of measuring tape |
| (3) | Coolant flow rate | $\pm$20f flow rate | |
| (4) | Tube wall temperature | $\pm 0.2^\circ$C | |
| (5) | Water temperature | $\pm 0.02^\circ$C | |
| (6) | Pressure | $\pm 0.0345$ bar | |
| (7) | Saturation temperature | $\pm 0.2^\circ$C | |

The uncertainty errors were calculated for all test runs with the help of the above procedure. It was found that the expected experimental error for reported data were within 7%.