ISME

Iranian Journal of Mechanical Engineering
The Iranian Society of Mechanical Engineers
Research Paper
DOI: 10.30506/jmee.2021.118453.1217

Influence of Coiled Wire Presence on the Convective Heat Transfer Coefficients in R-600 Condensers

A. Chakerizade*
M.Sc.

S. A. A. Mirjalily†
Assistant Professor

A. J. Ahrar‡
Assistant Professor

An experimental investigation has been performed to examine the heat transfer enhancement achieved by helically coiled wire inserts during forced convective condensation of R-600 vapor inside horizontal tubes. The experimental set-up is a vapor-compression refrigeration cycle. This system includes a main condenser which is a coaxial double-pipe counter-flow heat exchanger with water as a coolant. To reach the favorable vapor qualities, pre-condenser and after-condenser systems were invoked. Experiments were carried out for a plain tube and five different coiled wire inserted tubes. At each test, various parameters were measured like refrigerant mass flow rate, inlet and outlet water temperature of the condenser, refrigerant pressure, etc. Using the acquired data, heat transfer coefficients were calculated for different tubes. Investigation of the results showed that increasing the thickness of the wires can augment the heat transfer rate. Also, it is seen that reducing the coil pitch can enhance the heat transfer rate as high as 31.6%. Also, it was seen that employing coiled wire inserts inside the tube, increases the average heat transfer coefficient up to approximately 80 % regarding the plain tube results.

Keywords: Coiled wire, Enhanced heat transfer, R-600 refrigerant, Condenser

1 Introduction

Many novel and fruitful methods have been used during the last decades to enhance the heat transfer rate of common heat exchangers. These progressing techniques can lead to better heat transfer efficiency and consequently, lower costs and smaller sizes of the heat exchangers are achieved [1-6]. Due to the expensive costs of energy and lower living area limits, the research on the topic is accelerated in recent years. Especially, in lots of industries like solar concentrators, reservoir productions, refrigeration, and cooling process of electric devices, which suffer the lack of free spaces and are in need of strong cooling systems, keeping the cooling procedure costs in an affordable and reasonable range is a must [7-12]. Condensers are considered as one of the most applicable types of heat exchanging devices. In the industry usually one can find condensers in steam power plants, HVAC systems, and chemical production systems.

* M.Sc., Department of Mechanical Engineering, Yazd Branch, Islamic Azad University, Yazd, Iran Chakerizade@gmail.com
† Corresponding Author, Assistant Professor, Department of Mechanical Engineering, Yazd Branch, Islamic Azad University, Yazd, Iran SAA_mirjalily@iauyazd.ac.ir
‡ Assistant Professor, Department of Mechanical Engineering, Faculty of Sadooghi, Yazd Branch, Technical and Vocational University (TVU), Yazd, Iran amirahrar@yahoo.com

As an example, in HVAC systems due to the wetting characteristics of the coolants, only the layer condensation is observed, and since the thermal conductivity of the coolants is usually very low, the total entropy generation of the system is increased. Therefore, these heat transfer enhancement methods are even more essential in HVAC systems. Akers et al. [13] investigated the influence of the vapor velocity, temperature difference, liquid load, and fluid properties on the mean heat transfer coefficient of condensing vapor in a horizontal tube. They replaced vapor with a liquid of the same surface tension on the vapor-liquid contact line in a theoretical analysis. Boyko and Kruzhilin [14] proposed a correlation to predict the mean heat transfer coefficient during the condensation process of water vapor. Also, Soliman [15] studied the influence of the mist-annular transition on the heat transfer process of condensation of vapor. He indicated that using annular correlations for the mist region is not fully accurate, and to calculate the deviation of flow characteristics from the annular region, one can apply for the Weber number. He also proposed an experimental correlation to estimate the heat transfer rate in mist regions. This equation was formed under the assumption of a homogeneous vapor-liquid mixture.

Also, Fard et al. [16] in an experimental study investigated the effect of carbon nanotubes on the heat transfer rate and pressure drop of water in a helically coiled tube. Their findings indicated that the critical Reynolds number of the spiral tubes is higher than that of the straight tubes. Mozaffari et al. [17] studied the effect of the twisted tape inserts on the fluid flow and heat transfer rate of Al_2O_3/Water nanofluid. They stated that higher Nusselt numbers can be achieved in lower twist ratios of the inserts.

Royal and Bergles [18] investigated the effects of tube inclination angle, internal fins, and twisted tape on the condensation of vapor in tubes. They discovered that the tube inclination angle does not necessarily affect the heat transfer rate, but on the other hand, the twisted tapes can enhance the heat transfer coefficient of the horizontal condenser by 30% with respect to a plain tube. Moreover, the straight and spiral fins increased the heat transfer rate up to 150%. Akhavan Bahabadi et al. [19] studied the influence of applying twisted tape in R-134a condensers. They reported significant heat transfer enhancement in the system.

Salimpour and Yarmohammadi [20, 21] studied the same phenomenon using R-404a refrigerant. In another experimental research, Salimpour and Golami [22] investigated the pressure drop due to the condensation of R-404a in the presence of coiled wire and reported a significant increment in heat transfer rate. Agrawal et al. [23] also, used the coiled wire in a tube filled with R-22 refrigerant and studied its condensation process. Mostly, their achievements indicated spiral and semi-spiral flow patterns in the tube. In addition, they observed oscillation in the temperature of the tube wall. Moreover, they reported an increment of up to 100 % in the heat transfer regarding the plain tube. In this study, it is shown that the heat transfer coefficient is a complicated function of the refrigerant vapor, Re number, and the coiled wire pattern. They also observed better heat transfer enhancement in the presence of coiled wire with respect to the tape twisted tubes.

Akhavan Behabadi et al. [24] investigated R-134a condensation in a plain tube and a tube equipped with coiled wire, experimentally. They declared that the heat transfer coefficient is a function of vapor, Re number, and the coiled wire pattern again. In their investigations, an 80% increase in the heat transfer rate is reported as well. Although fine researches are done on various refrigerants and coil patterns [25-27] to the best of our knowledge, the influence of the coiled wire presence on the condensation process of R-600 vapor has not been considered as a case study by researchers yet. Consequently in this study, for the first time, the condensation process of R-600 is investigated experimentally, and the influence of the coiled wire patterns on the forced convection and condensation of R-600 is studied.

2 Experimental setup

The designed experimental setup is a vapor-condensing refrigerator system consisting of all necessary sensors and test devices. A schematic view of the test setup is illustrated in figure (1). It is obvious from this figure that the main condenser is consisting of three subdivisions: the primary condenser, the test condenser, and the secondary condenser. The test condenser is a double pipe counter-flow heat exchanger. R-600 refrigerant is condensed in the inner tube and exerts heat to the water flowing in the outer tube. The inner tube is made of copper with 14.1 mm internal diameter, 0.9 mm thickness, and 1 m length, while the outer tube is made of steel with an 80 mm internal diameter. The flow temperature is measured at four individual middle sections of the tube on both top and bottom sides. The sensors for temperature are chosen to be the J-type thermocouples and four additional sensors are placed at the entrance and exit locations of the tube. Also, the temperature of the coolant water is recorded at the inlet and outlet sections. Moreover, the absolute pressure of the refrigerant flow is observed at the inlet and outlet sections of the test condenser via pressure gauges. Also, the whole test condenser is isolated by glass wool to prevent any undesired heat loss from the system. In order to have a suitable and adjustable vapor quality at the entrance of the test condenser, a primary condenser is used. This primary condenser provides us the ability to regulate and adjust the vapor quality by altering the coolant (which in this case is water) flow rate, and consequently one can monitor the influence of altering refrigerant quality on the heat transfer rate and pressure drop of the system. This condenser is a double tube heat exchanger in which the cold water flows in the outer tube and the refrigerant flows through the inner pipe. The primary condenser size is the same as the test condenser, with only one difference that the copper tube length is 9 m and is swirled 9 times. The outer and inner diameters of this condenser are 9.525 mm and 7.525 mm, respectively. To increase the heat transfer capacity of the system a coiled wire is used in the test section. This additional coiled wire may have 3 various diameters of 0.5, 1, and 1.5 mm and 3 different pitches of 8, 10, and 13 mm.

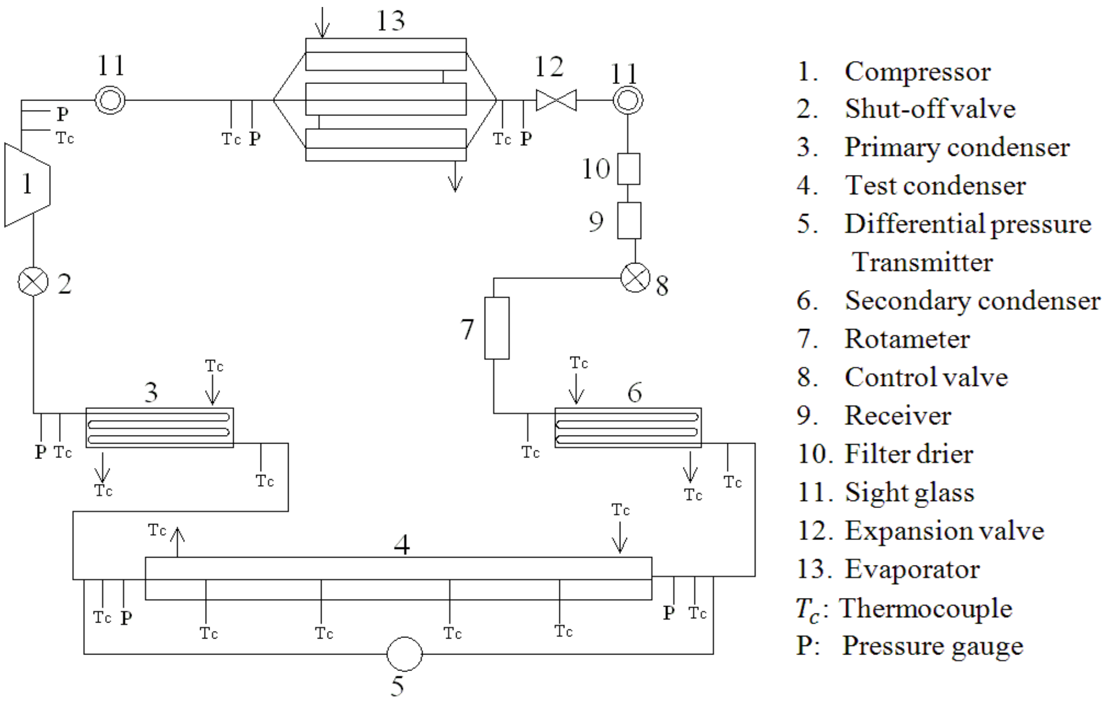

Figure 1 A schematic diagram of the experimental setup

3 Experimental discussion

The test instruments were calibrated with the predefined format, and the system went through several validation tests. After correctness and exactness assurance, real tests and data refinements were carried out. During test runs, three datasets in 10 min intervals were recorded after the system reached the steady-state condition. The average value of these three records was applied for the rest of the calculations. The recorded data are as follows:
*. The mass flow rate of the refrigerant
*. The mass flow rate of the coolant water in the test condenser, primary condenser, and secondary condenser
*. The refrigerant temperature at the inlet and outlet of the test, primary and secondary condensers
*. The temperature of the outer walls of the inner tube of the test condenser
*. The temperature of the coolant water at the entrance and exit sections of the test, primary and secondary condensers
*. The static pressure of the refrigerant at the inlet and outlet of the test, primary and secondary condensers
A total number of 180 individual tests with five different mass flow rates and six various vapor qualities for plain and coiled wire equipped tubes were carried out. At first, the experimental data for the plain tube were achieved, then the system was reset with the mentioned coiled wire and the experiments were carried out again for the new setup.

4 Calculations and formulations

In each test case, the heat transfer coefficient is obtained based on the absorbed heat by the coolant water in the test section and the temperature difference between the condensing refrigerant and the tube walls. In order to calculate the heat transfer coefficient, firstly the condenser outer wall average temperature T_{ws} is estimated as:

$$T_{ws} = \frac{T_T + T_B}{2} \tag{1}$$

In which T_T and T_B refer to the temperature of the top and bottom sides of the tube. Then the mean temperature of test condenser walls T_{wo} is obtained from the 4 estimated average temperatures:

$$T_{wo} = \frac{\sum_{1}^{4} T_{ws}}{4} \tag{2}$$

The exerting heat by the coolant water is calculated from:

$$Q = \dot{m}_w C_{pw} \left(T_{wo} - T_{wi}\right) \tag{3}$$

Here, $\dot{m}_w$ is the coolant water mass flow rate and C_{pw} is the thermal capacity of water. The radial heat flux can be computed then as:

$$q = \frac{Q}{\pi DL} \tag{4}$$

The temperature drop in the tube wall ΔT_w is also related to the radial heat flux and conduction heat transfer coefficient of the copper as:

$$\Delta T_w = \frac{qD \ln\left(\frac{D_o}{D}\right)}{2K_w} \tag{5}$$

Thus the inner wall mean temperature is obtained from:

$$T_{wi} = T_{wo} + \Delta T_w \tag{6}$$

The average static pressure in the test section is estimated as the average of inlet and outlet pressure gauges, and since the vapor is in the saturated region the mean temperature of the vapor is assumed to equal the temperature related to this static pressure. Finally, the heat transfer coefficient of the test condenser is calculated from:

$$h = \frac{q}{\left(T_s - T_{wi}\right)} \tag{7}$$

In which, T_s is the saturated temperature of the vapor.

5 Results and discussions

The condensing heat transfer coefficient for R-600 flow in both coiled wires equipped tube and plain tube in different refrigerant situations is investigated experimentally. The vapor quality is estimated from the average of vapor qualities at the inlet and outlet sections of the test condenser. Then the influences of certain parameters like refrigerant quality and mass velocity, coolant mass flow velocity, and the coiled wire specifications are studied in the condensation process and the heat transfer rate of the system. To be assured that the measured data are reliable, 20% of the experimental runs were selected randomly to go through double and triple checks on different days. The results were acceptable. One of the plain tube results is shown in figure (2). To check the integrity of the test set-up, plain tube results are achieved and compared to the correlation proposed in ref. [14]. This comparison is presented in figure (3), where a reasonable agreement is observed. Figure (4) aims to elicit the influence of the vapor quality on the heat transfer coefficient. In this figure, the heat transfer rates for 5 different mass velocities are presented. Moreover, for each of these mass velocities, 6 different tests with various vapor qualities were carried out under the same condensing pressure, and hence a wide range of vapor quality is covered. This figure mostly indicates that if all other parameters remain untouched, the heat transfer coefficient augments with the mass velocity increment. In fact, higher refrigerant velocities lead to a more turbulent flow regime in the condensing layer, and consequently, it would accelerate the condensation process.

On the other hand, it is obvious from figure (4) that the reduction of vapor quality in the test condenser results in a certain decrement of heat transfer rate. This phenomenon also is rooted in the fact that in higher vapor qualities, the liquid refrigerant layer on the inner walls of the tube is thinner and they will cause less thermal resistance on the wall.

Experimental results for the coiled wire equipped tube for three different wire diameters of 0.5, 1 and 1.5 mm with a 10 mm pitch were recorded. Also, three various pitches of 8, 10, and 13 mm were tested with a 1.5 mm wire diameter. The specifications of these coiled wire equipped tubes are presented in table (2). In this table, D is the inner diameter of the tube, D_o is the outer diameter, D_e is the effective diameter, L is the length of the tube, e is the coiled wire thickness P is the coil pitch, and α is the swirling angle.

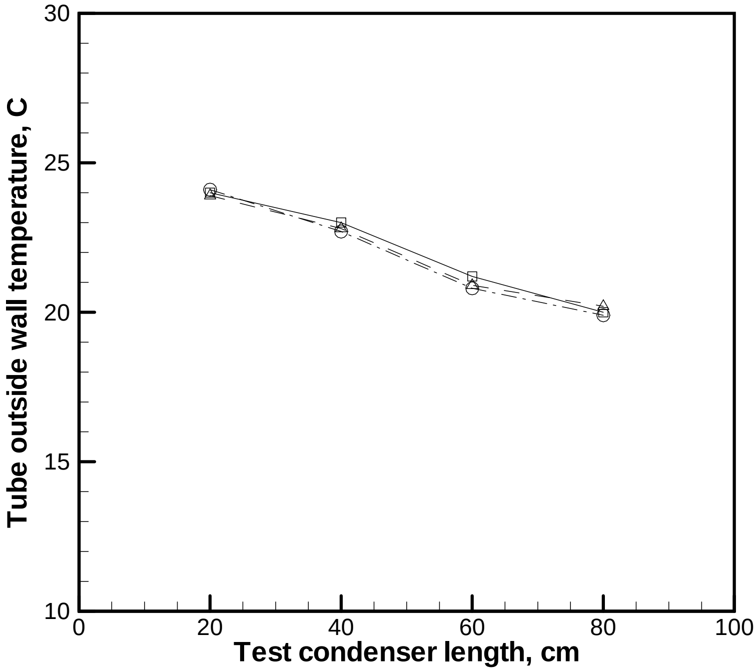

Figure 2 A repetitivity result for the plain tube

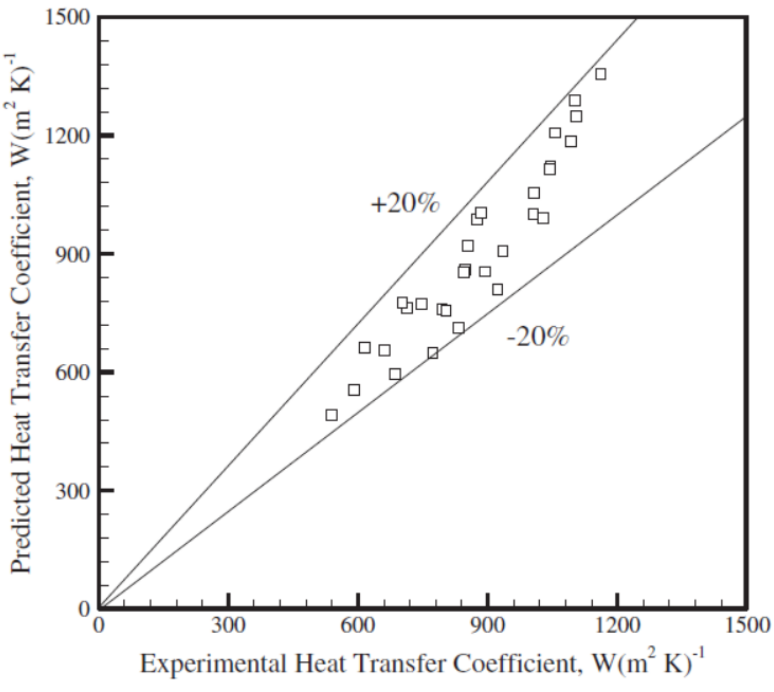

Figure 3 Comparison of the present heat transfer coefficients with the results of ref. [14]

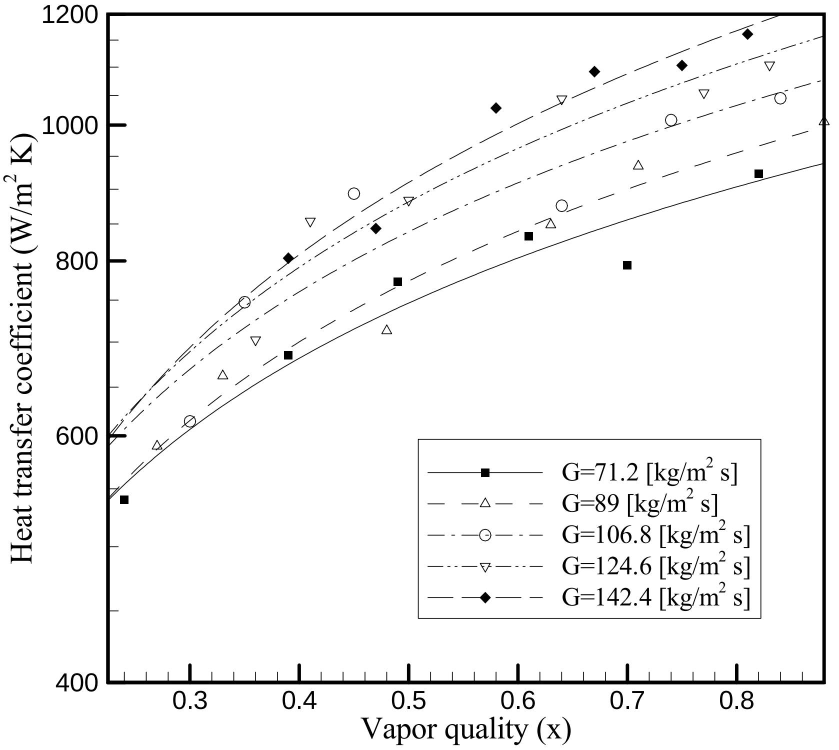

Figure 4 Variation of the heat transfer coefficient versus the vapor quality for different mass velocities

Table 1 dimensions and specifications of the coiled wire equipped tubes

Tube set	D [mm]	D_o [mm]	D_e [mm]	L [mm]	e [mm]	P [mm]	α [degree]
A	14.1	15.9	11.9	1000	0.5	10	73
B	14.1	15.9	10.4	1000	1	10	73
C	14.1	15.9	9.1	1000	1.5	10	73
D	14.1	15.9	8.4	1000	1.5	8	76.6
E	14.1	15.9	9.9	1000	1.5	13	69
F	14.1	15.9	---	1000	---	---	---

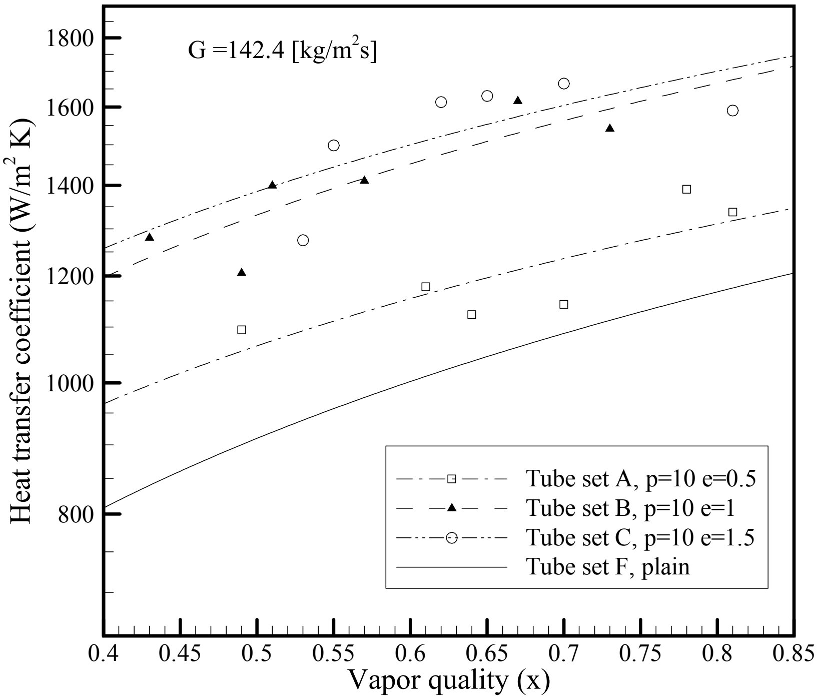

Figure 5 Variation of the heat transfer coefficient versus the vapor quality for coiled wire equipped and plain tubes of A, B, C, and F at G=142.4 [Kg/m^2s]

Figure (5) illustrates the influence of coiled wire thickness on the heat transfer coefficient. As can be seen, the addition of coiled wire in the tube results in an outstanding heat transfer enhancement. The highest increment was observed in the tube equipped with the thickest coil, i. e. tube D. This coiled wire can increase the mean heat transfer coefficient of the system at G=142.4 (Kg/m^2s) up to 80% regarding a plain tube.

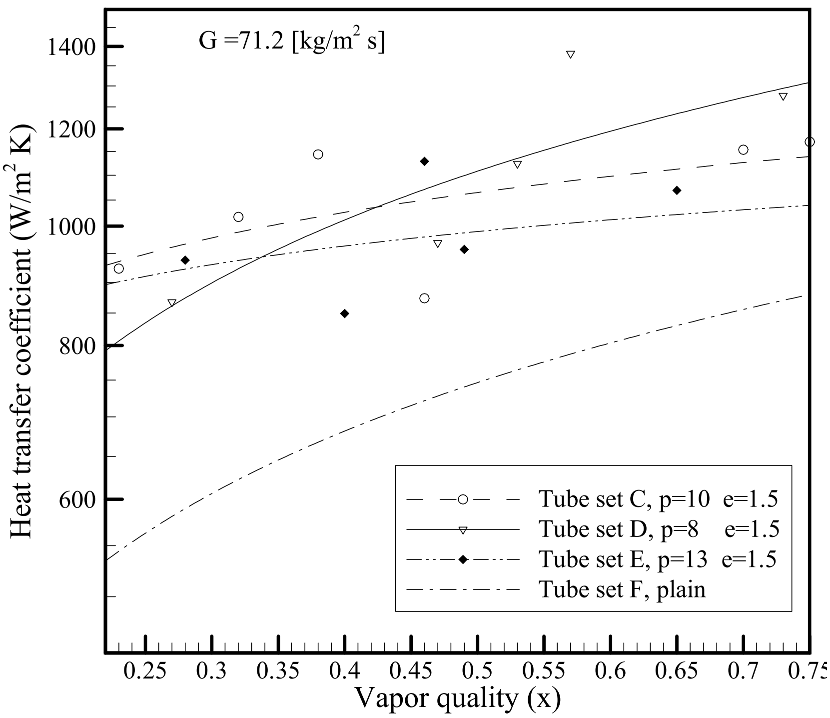

Figure 6 Variation of the heat transfer coefficient versus the vapor quality for coiled wire equipped and plain tubes of C, D, E, and F at G=71.2 [Kg/m^2s]

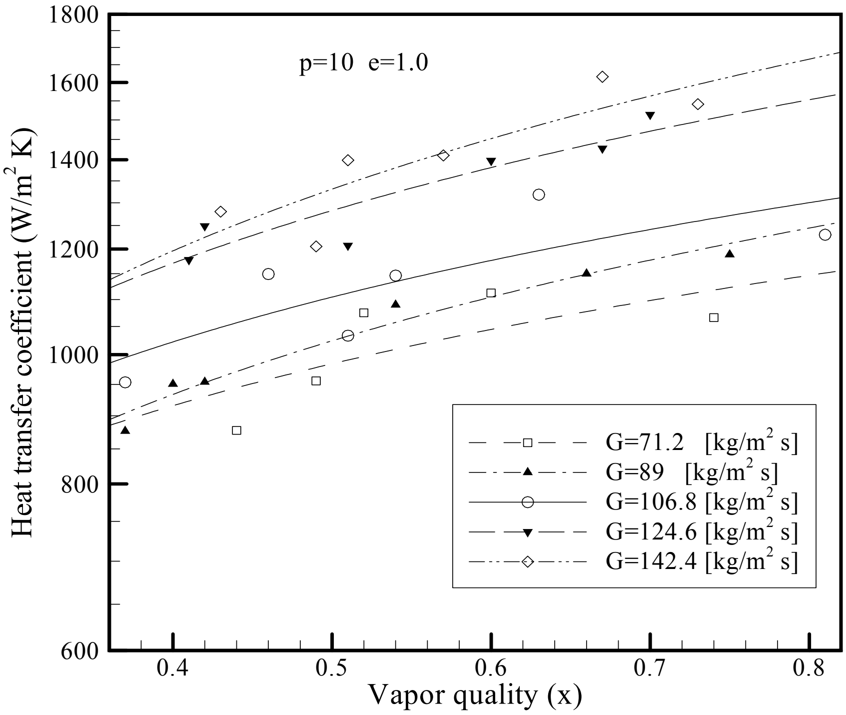

Figure 7 Variation of the heat transfer coefficient versus the vapor quality for different mass velocities in pipe B

Moreover, the behavior of the heat transfer coefficient with variations of coiled wire pitch is presented in figure (6). This figure indicates that as the coiled wire thickness augments the heat transfer rate of the condenser increases as well. On the other hand, when the coil pitch is decreased, the heat transfer rate would increase. This enhancement can reach up to 31.6% when the pitch is declined from 13 to 8 mm. This phenomenon occurs due to the increment of flow friction with the reduction of the coiled wire pitch. This friction would increase the flow turbulence and consequently, it increases the heat transfer rate.

Finally, figure (7) demonstrates the variations of heat transfer coefficient versus vapor quality in tube B for five different mass velocities. This figure indicates that the heat transfer coefficient reduces by the vapor quality reduction in all mass velocities and coiled wire patterns. This happens due to the reduction of liquid layer thickness on the tube walls in higher refrigerant qualities which may cause less thermal resistance on the tube wall. In addition, one can deduce from this figure that when all other parameters are fixed, generally the heat transfer coefficient would increase with an increment of the mass velocity. This behavior is well-predicted and is in absolute accordance with the flow and heat transfer rates of the plain tube.

6 Conclusions

The flow and heat transfer characteristics of R-600 refrigerant condensation was experimentally investigated in a plain and several coiled wire equipped tubes. The following results were obtained from the experiments:

1. The heat transfer coefficient is increased on the tube walls with an increment in the mass velocity of the refrigerant, vapor quality, and mass velocity of the coolant liquid.
2. It was observed that the coiled wire can enhance the heat transfer rate up to 80% with regard to the plain tube.

3. With an increment of the coiled wire thickness, the heat transfer coefficient would increase when all other parameters are fixed, and therefore the highest rate of heat transfer was observed in the tube containing the coiled wire with the biggest thickness.
4. Also, when the coiled wire pitch is reduced, the heat transfer rate augments. This behavior is seen to be general for high-quality, but not for low-quality refrigerant vapors.

References

[1] Dehghan, M., Valipour, M.S., and Saedodin, S., "Microchannels Enhanced by Porous Materials: Heat Transfer Enhancement or Pressure Drop Increment", Energy Conversion and Management, Vol. 110, pp. 22-32, (2016).

[2] Vutukuru, R., Pegallapati, A.S., and Maddali, R., "Suitability of Various Heat Transfer Fluids for High Temperature Solar Thermal Systems", Applied Thermal Engineering, Vol. 159, pp. 113973, (2019).

[3] Sadaghiani, A.K., Saadi, N. S., Parapari, S. S., Karabacak, T., Keskinoz, M., and Kosar, A., "Boiling Heat Transfer Performance Enhancement using Micro and Nano Structured Surfaces for High Heat Flux Electronics Cooling Systems", Applied Thermal Engineering, Vol. 127, pp. 484-498, (2017).

[4] Giovannelli, A., and Bashir, M.A., "High-temperature Cavity Receiver Integrated with a Short-term Storage System for Solar MGTs: Heat Transfer Enhancement", Energy Procedia, Vol. 126, pp. 557-564, (2017).

[5] Bie, Y., Li, M., Chen, F., Królczyk, G., and Li, Z., "Heat Transfer Mathematical Model for a Novel Parabolic Trough Solar Collecting System with V-shaped Cavity Absorber", Sustainable Cities and Society, Vol. 52, pp. 101837, (2020).

[6] Varun, Garg, M. O., Nautiyal, H., Khurana, S., and Shukla, M. K., "Heat Transfer Augmentation using Twisted Tape Inserts: A Review", Renewable and Sustainable Energy Reviews, Vol. 63, pp. 193-225, (2016).

[7] Choi, S.U., and Eastman, J.A., "Enhancing Thermal Conductivity of Fluids with Nanoparticles", Argonne National Lab., IL (United States), (1995).

[8] Das, S.K., Choi, S.U., and Patel, H.E., "Heat Transfer in Nanofluids-a Review", Heat Transfer Engineering, Vol. 27, No. 10, pp. 3-19, (2006).

[9] Wang, X. Q., and Mujumdar, A.S., "A Review on Nanofluids-part I: Theoretical and Numerical Investigations", Brazilian Journal of Chemical Engineering, Vol. 25, No. 4, pp. 613-630, (2008).

[10] Yu, W., and Xie, H., "A Review on Nanofluids: Preparation, Stability Mechanisms, and Applications", Journal of Nanomaterials, Vol. 2012, pp. 1-17, (2012).

[11] Keblinski, P., Phillpot, S. R., Choi, S. U. S., and Eastman, J. A., "Mechanisms of Heat Flow in Suspensions of Nano-sized Particles (Nanofluids)", International Journal of Heat and Mass Transfer, Vol. 45, No. 4, pp. 855-863, (2002).

[12] Hwang, K.S., Jang, S.P., and Choi, S.U., "Flow and Convective Heat Transfer Characteristics of Water-based Al2O3 Nanofluids in Fully Developed Laminar Flow Regime", International Journal of Heat and Mass Transfer, Vol. 52, No. 1-2, pp. 193-199, (2009).

[13] Akers, W., Deans, H., and Crosser, O., "Condensing Heat Transfer within Horizontal Tubes", Chem. Eng. Progr., Vol. 54, (1958).

[14] Boyko, L., and Kruzhilin, G., "Heat Transfer and Hydraulic Resistance during Condensation of Steam in a Horizontal Tube and in a Bundle of Tubes", International Journal of Heat and Mass Transfer, Vol. 10, No. 3, pp. 361-373, (1967).

[15] Soliman, H., "The Mist-annular Transition during Condensation and its Influence on the Heat Transfer Mechanism", International Journal of Multiphase Flow, Vol. 12, No. 2, pp. 277-288, (1986).

[16] Fard, A.M., Mirjalily, S.A.A., and Ahrar, A.J., "Influence of Carbon Nanotubes on Pressure Drop and Heat Transfer Rate of Water in Helically Coiled Tubes", Journal of Enhanced Heat Transfer, Vol. 26, No. 3, pp. 217-233, (2019).

[17] Mozaffari, J., Mirjalily, S.A.A., and Ahrar, A.J., "Experimental Investigation of Enhancing Influence of Al2O3 Nanoparticles on the Convective Heat Transfer in a Tube Equipped with Twisted Tape Inserts", Thermal Science, Vol. 25, No. 1B, pp. 541-551, (2021).

[18] Luu, M., and Bergles, A., "Experimental Study of the Augmentation of In-tube Condensation of R-113", Iowa State University College of Engineering, (1979).

[19] Akhavan-Behabadi, M., Kumar, R., and Rajabi-Najar, A., "Augmentation of Heat Transfer by Twisted Tape Inserts during Condensation of R-134a Inside a Horizontal Tube", Heat and Mass Transfer, Vol. 44, No. 6, pp. 651-657, (2008).

[20] Salimpour, M., and Yarmohammadi, S., "Effect of Twisted Tape Inserts on Pressure Drop during R-404A Condensation", International Journal of Refrigeration, Vol. 35, No. 2, pp. 263-269, (2012).

[21] Salimpour, M., and Yarmohammadi, S., "Heat Transfer Enhancement during R-404A Vapor Condensation in Swirling Flow", International Journal of Refrigeration, Vol. 35, No. 7, pp. 2014-2021, (2012).

[22] Salimpour, M.R., and Gholami, H., "Effect of Inserting Coiled Wires on Pressure Drop of R-404A Condensation", International Journal of Refrigeration, Vol. 40, pp. 24-30, (2014).

[23] Agrawal, K. N., Kumar, A., Akhavan-Behabadi, M. A., and Varma, H. K., "Heat Transfer Augmentation by Coiled Wire Inserts during Forced Convection Condensation of R-22 Inside Horizontal Tubes", International Journal of Multiphase Flow, Vol. 24, No. 4, pp. 635-650, (1998).

[24] Akhavan-Behabadi, M., Salimpour, M. R., Kumar, R., and Agrawal, K. N., "Augmentation of Forced Convection Condensation Heat Transfer Inside a Horizontal Tube using Spiral Spring Inserts", Journal of Enhanced Heat Transfer, Vol. 12, No. 4, (2005).

[25] Hernandez III, A.C., and Fumo, N., "A Review of Variable Refrigerant Flow HVAC System Components for Residential Application", International Journal of Refrigeration, Vol. 110, pp. 47-57, (2020).

[26] Deng, N., Gao, J., Cai, R., Jing, X., Zhang, Y., Hao, R., and Li, M., "Experimental Investigation on Matching a Conventional Water Source Heat Pump with Different Refrigerants for Supplying High Temperature Water", Applied Thermal Engineering, Vol. pp. 114668, (2020).

[27] Dilip Raja, N., Subesh, T., Surrya Prakash, D., Venkatasudhahar, M., Ramesh, V., and Logesh, K., "Retracted Article: Experimental Investigation on the Effect of Shell Side Interstitial Twisted Tapes in the Performance of Counter Flow Shell and Tube Heat Exchanger", International Journal of Ambient Energy, Vol. 40, No. 8, pp. 1-8, (2019).

Nomenclature

T_{ws}	Outer wall average temperature (*K*)
T_{wo}	Temperature of test condenser walls (*K*)
Q	Exerting heat by the coolant water (*W*)
$\dot{m}_w$	Coolant water mass flow rate ($kg\ t^{-1}$)
C_{pw}	Thermal capacity of water ($J\ kg^{-1}\ K^{-1}$)
q	Radial heat flux ($W\ m^{-2}$)
ΔT_w	Temperature drop in the tube wall (*K*)
D	Inner diameter of the tube (*m*)
D_o	Outer diameter of the tube (*m*)
K_w	Thermal conductivity ($W\ m^{-1}\ K^{-1}$)
T_{wi}	Inner wall mean temperature (*K*)
T_s	Saturated temperature of the vapor (*K*)
h	Convection coefficient ($W\ m^{-2}\ k^{-1}$)
D_e	Effective diameter (*m*)
L	Length of the tube (*m*)
e	Coiled wire thickness (*m*)
P	Coil pitch (*m*)
α	Swirling angle (*degree*)