Experimental Investigation of Condensation Heat Transfer Characteristics of R-134a Vapor in Horizontal Heat Exchanger

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Abstract

An experimental investigation of refrigerant R-134a two-phase flow condensation heat transfer coefficient and pressure drop in condenser tube section of refrigeration system under different operating conditions is presented. The experimental and theoretical investigations are based on test conditions in range of 10 -17 kW/m² for heat flux, 42-63 kg/m²s for mass flux, vapor quality 1-0.03 and saturation temperature 44 to 49˚C. The experimental tests are conducted on test rig supplied with a test section to simulate the water cooled double pipe heat exchanger, which is designed and constructed in the present work. The experimental results have revealed that, the heat flux and mass flux have significant impacts on the heat transfer coefficient. The heat transfer coefficient was increased with increase in heat flux and mass flux at prescribed test conditions, where the enhancement in heat transfer coefficient was about 47% and 14% for relatively higher heat flux and mass flux, respectively. The enhancement in the heat transfer coefficient was about 51% for relatively lower saturation temperature 45.97˚C and 43% for higher vapor quality 0.88 compared to other values at constant test conditions. The pressure drop was higher in the range of 12% and 49% for relatively higher mass flux and heat flux respectively. The present work results have validated by comparison with predictive models and with similar research work results and the comparison has revealed an acceptable agreement.

Keywords: Heat transfer coefficient, Flow condensation, Heat flux, Mass flux.

Nomenclature

C_p: Specific heat at constant pressure	kJ/kg.K	µ: Dynamic viscosity kg/m.s
G: Mass flux	kg/m^2.s	ρ: Density kg /m^3
h: Specific enthalpy	kJ/kg	***Subscripts***
h_{fg}: Latent heat of condensation	kJ/kg	a: Top
h_z: Local heat transfer coefficient	W/m^2.℃	b: Bottom
ID: Inside diameter of pipe	m	fr: Frictional
k: Thermal conductivity	W/ m.℃	g: Vapor
L: Length of tube	m	i: Inlet
$\dot{m}$: Mass flowrate	kg/s	l: Liquid

OD: Outside diameter of pipe	m	m: Momentum
P: Pressure	Pa	o: Outlet
$\dot{Q}$: Heat transfer rate	W	pre: Pre-condenser
q: Heat flux	W/m^2	ref: Refrigerant, R134a
R: Thermal resistance	℃ /W	s: Sensible
T: Temperature	℃, K	st: Static
x: Vapor quality		sat: Saturation
		t: Test section
Greek letters		w: Tube wall
Δ: Denotes a difference or gradient.		wt: Water

1. Introduction

Condensation is a phase change process from vapor to liquid which represents an efficient way of heat removal as the latent heat of condensation provides a high heat transfer coefficient. Condensation heat transfer characteristics are important in many engineering and industrial applications, such as air-conditioning and refrigeration, steam power plants, cooling of nuclear reactors, heating and cooling process. Condensation of refrigerants inside heat exchanger channels was investigated by many researchers. [1] investigated experimentally the two-phase heat transfer coefficient of pure R134a condensing inside a smooth tube with inner diameter 8.1 mm and 500 mm length in tube heat exchanger vertical downward flow at high mass flux . The test runs are performed at average saturation condensing temperatures between 40-50℃, the mass fluxes are between 260 and 515kg m^{-2} s^{-1} and the heat fluxes are between 11.3 and 55.3 kW m^{-2}. The results showed that, the experimental heat transfer coefficient increases with the average vapor quality and refrigerant mass flux and decreases with increases of condensation and tube wall temperature difference. [2] measured condensation heat transfer coefficients in mini channels with smaller measurement uncertainties than previously obtained using three specially designed copper test sections. The test included three channel geometries, square, triangular and semi-circular multiple parallel mini-channels cooled on three sides of 1mm hydraulic diameters. Condensation heat transfer coefficients were obtained for the range of mass flux, average quality, saturation pressure and heat flux. The results revealed that, mass flux and quality have significant effects on the condensation process, even at lower mass fluxes, while saturation pressure, heat flux, and channel shape had no significant effects. [3] investigated experimentally the heat transfer and observed the condensation flow patterns of refrigerant R-134a inside a single smooth tube with several inclination angles. The test-condenser was a 1.04 m long double pipe counter-flow heat exchanger with an inner diameter of 8.38 mm. The experiments are performed for seven different tube inclinations in the range of -90° to +90° (with 30° increments) and mass velocities between 53 and 212 kg/m^2.s . The best performance is achieved by the tube with an inclination angle of +30° for all refrigerant mass velocities. [4] developed models to study the pressure drop and heat transfer coefficients for condensation of hydrocarbons in smooth horizontal tubes over a wide range of conditions for pure natural fluids, propane and pentane . They were combined with first study a measure of the heat transfer coefficient and pressure drops during condensation of propane in smooth horizontal tubes by [5]. These correlations validated for tubes with internal diameters between 6 and 19 mm. The models showed improved agreement with the frictional pressure drop and heat transfer coefficient measurements in the database when compared to the predictions of the correlations available. The models predicted the frictional pressure drop and heat transfer coefficient for the propane database with average deviations of 3% and -1%, respectively. [6] investigated experimentally the condensation of pure refrigerant R134a vapor inside a smooth vertical tube under different operating conditions . The test section was made of a copper tube with an inside diameter of 7.52 mm and length of 1m. Experimental tests were conducted for mass fluxes in the range of 20 – 175 (kg/m^2. s) with saturation pressure ranging between 5.8 and 7 bars. Obtained results showed that average condensation heat transfer coefficient decreases with increasing saturation pressure or temperature difference and increases with the increasing mass flux at the same saturation pressure and temperature difference. The saturation pressure and temperature were very effective on the average condensation heat transfer coefficient. [7] measured condensation heat transfer coefficient of R410A inside three different circular tubes with an inner diameter of 6.61, 7.5 and 9.2mm . Two-phase fluid flow conditions include mass fluxes from 200 to 320 (kg/m^2. s), qualities between 0.1 to 0.9, and heat flux range from 5 to 20 kW/m^2 at a fixed saturation temperature of 48 ℃. The results showed that the average heat transfer coefficient increased with the increase of vapor quality, mass flux, and heat flux, but decreased with an inner diameter. [8] studied the condensing flow heat transfer coefficient and pressure drop of a multiport mini-channel aluminium tube with a natural hydrocarbon, propane R290 as working fluid flowing through a square section horizontal tube having an internal diameter of 1.16 mm and a condensing length of 259 mm. The results showed that, the two-phase friction pressure gradient increases with the increase of mass velocity and vapor quality and with the decrease of saturation temperature. The heat transfer coefficients showed to be increased with increases in vapor quality and mass velocity, while increases of saturation temperature were observed to reduce the heat transfer coefficient. [9] investigated the condensation heat transfer coefficient and pressure drop of methane in a horizontal smooth tube. The tests were conducted at saturation pressure of 2 - 3.5 Mpa with the mass flux of 99 - 255kg/m^2.s and fluid to wall temperature difference of 4.8-20.2 K throughout the vapor quality range. The influences of mass flux, saturation pressure, vapor quality and temperature difference were analysed and discussed. Some condensation heat transfer coefficients of ethane with larger temperature differences were also reported at different operating conditions. The experimental data were compared with many well-known correlations of condensation heat transfer coefficient and pressure drop .

2. Experimental Setup and Test Conditions

2.1 Experimental setup

The experimental apparatus is schematically shown in the **Fig.1** . The refrigeration system is of 310W capacity operating with R-134a and including a test section to simulate the condenser channel. It mainly consists of a reciprocating compressor with 124W capacity, pre-condenser which is double pipe heat exchanger, test section, refrigerant flow meter, capillary tube, electrically heated evaporator, water pump, pressure transmitters, two water flow meters, pressure gages, thermocouples and data loggers, and many other accessories as illustrated in **Fig.2**. The counter flow tube in tube coaxial type heat exchanger with 279 mm length and 15.9 mm inner tube diameter is used as a pre-condenser to desuperheat the refrigerant vapor in the cycle before entering the test section . The single-phase refrigerant enters the pre-condenser as a superheated vapor and exchange heat with cooling water to obtain the required vapor quality at the test section entrance . The cooling load (heat flux) applied in condenser test section is regulated by adjusting the cooling water flow rate entering test section. While the mass flux of the refrigerant is regulated via bypass loop connected in parallel with test section line to maintain the specified test conditions. Temperatures of the tube wall at test section and refrigerant at different locations in the test rig are measured using calibrated K-type thermocouples with range of -200 ℃ to 1250 ℃ and the temperatures reading are displayed using data loggers of model BTM-4208SD and EXTECH-TM500 with 12 channels as shown in the **Fig.1**. The experimental investigations of the refrigerant R134a flow condensation heat transfer coefficient and pressure drop in the condenser test section are limited by operating conditions in range of 10.49 – 17.17 kW/m^2 for heat flux, 42 -64 $kg/m^2.s$ for mass flux, vapor quality 0.03 – 1 and saturation temperature 44 to 49°C. The accuracy of the experimental measurements is illustrated in the **Table 1**.

2.2 Test Section

A water cooled double-pipe heat exchanger with a counter-flow arrangement was designed and fabricated as the test section to simulate the condenser of refrigeration system with 310W capacity and to conduct the experimental investigations of the refrigerant flow condensation heat transfer characteristics. The test section consists of inner smooth copper pipe with 5.8 mm inner diameter, 7.8 mm outer diameter and 800 mm length and outer copper pipe (shell) with 22.2 mm inner diameter and 25.6 mm outer diameter as shown in **Fig.3.** The test section was wrapped with 10 mm thick polyurethane insulation to reduce heat losses. Ten thermocouples are fixed with equally distance on the outer surface of inner tube at five locations along the length of the tube, and at each location two thermocouples are mounted, one at the top and one at the bottom of tube surface to measure the average temperature of the refrigerant at each location. Two other thermocouples are inserted inside inner tube to measure the refrigerant temperature at the inlet and outlet of the test section. Temperatures of the cooling water are measured using two thermocouples inserted inside outer tube at the inlet and outlet of the water line. Two transparent tubes of 5.8 mm inner diameter and 150 mm length are connected in the inlet and outlet of the test section to visualize the flow pattern.

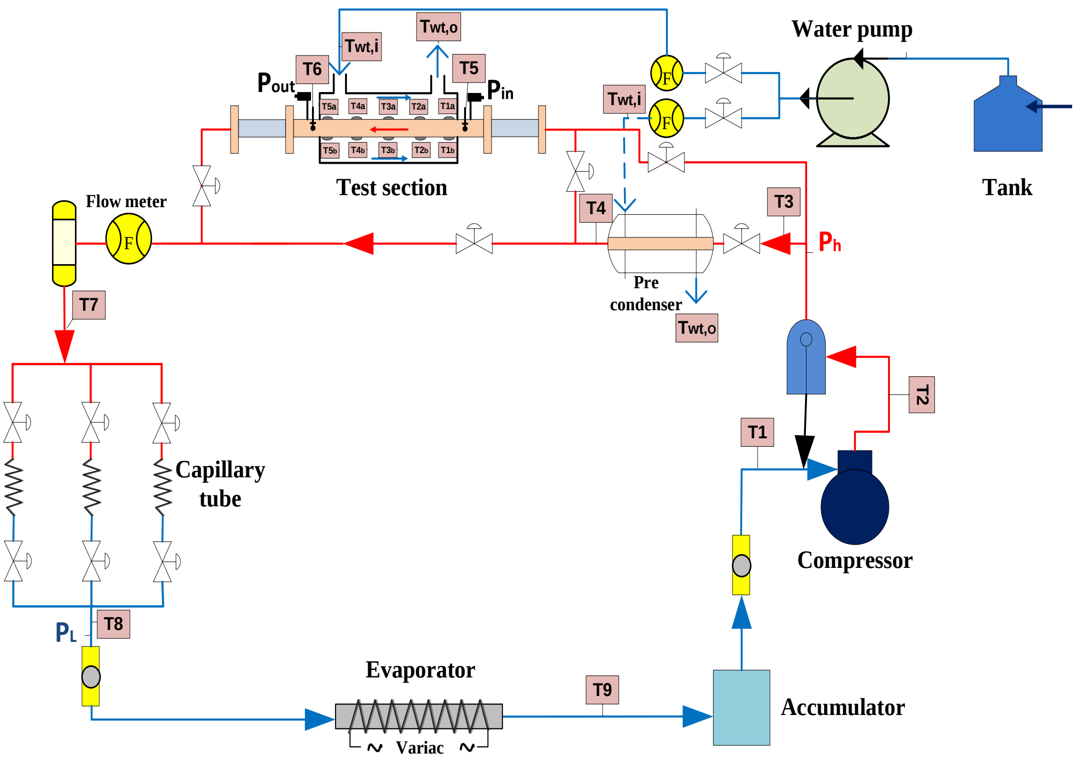

Figure 1. Schematic diagram of the experimental setup.

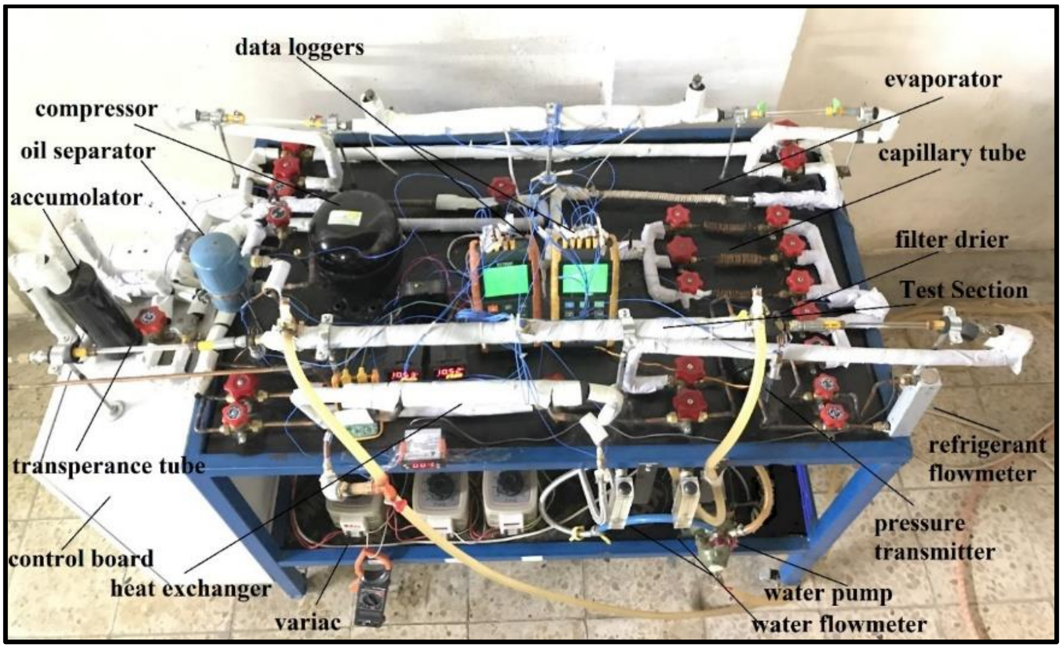

Figure 2. Test rig

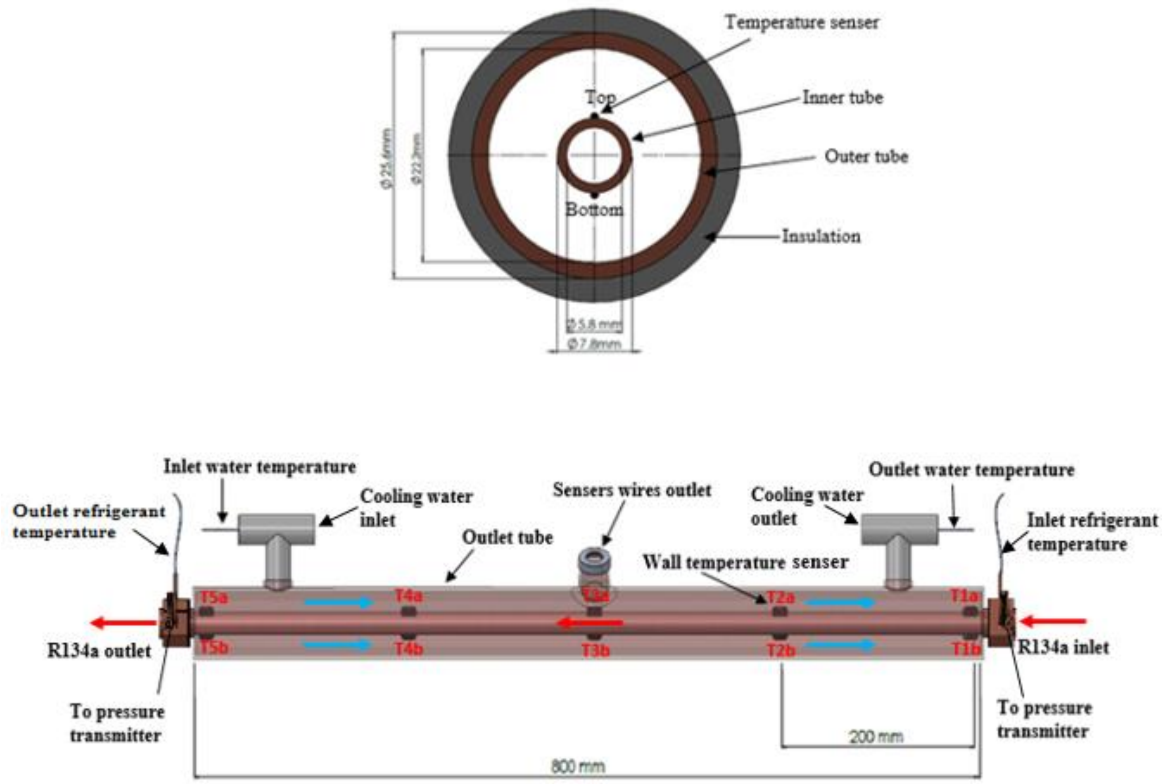

Figure 3. Condenser test section.

Table1. Accuracy of the experimental measurements

Variables	Measurement accuracy
Pressure gauge [kPa]	± 0.1 %
Pressure transmitter [kPa]	± 0.04 %
Temperature readers [°C]	± 1%
Refrigerant flow meter [kg/s]	± 0.01 %
Water flow meter [kg/s]	± 0.2 %

3. Experimental Data Analysis

To calculate the flow condensation heat transfer coefficient and pressure drop from the experimental data, the following considerations are supposed:

1- Heat transfer in the axial direction of the test section tube is neglected.
2- Heat flux is uniform along the condenser tube.
3- Pressure drop from the inlet to outlet pressure is a linear function of tube length.

3.1 Heat Transfer Coefficient

The heat removal rate for water flowing in the annulus of the test section and pre-condenser can be determined by [6]:

$$Q^{\cdot} = m^{\cdot}_{wt} \cdot cp_{wt} \cdot (T_{wt,o} - T_{wt,i}) \quad (1)$$

The rate of heat rejected by refrigerant in the test section is calculated by [10]:

$$\dot{Q}_S = \dot{m}_{ref} \cdot (h_i - h_o) \tag{2}$$

$$q = \frac{\dot{Q}}{\pi \cdot ID \cdot L} \tag{3}$$

The local heat transfer coefficient of the refrigerant flow condensation in the test section (condenser tube) is calculated by:

$$h_z = \frac{q}{(T_{ref} - T_{wi})} \tag{4}$$

The external wall temperature T_{wo} for each axial location (z) along the test section tube was assumed to be the average of measured temperatures around the tube cross section and calculated by:

$$T_{wo} = \frac{T_a + T_b}{2} \tag{5}$$

The mean inner wall temperature at each position (z), $T_{wi,z}$ is calculated using one dimensional heat conduction across [11]:

$$T_{wi,z} = T_{wo} + (\dot{Q} \cdot R) \tag{6}$$

$$R = \frac{\ln\left(\frac{OD}{ID}\right)}{2 \cdot \pi \cdot L_z \cdot k} \tag{7}$$

L_z : The length of tube subsection (increment in the axial position along tube) (m)

The refrigerant saturation temperature along the test section at each position (z) $T_{ref.z}$ is calculated depending on the local saturation pressure $P_{sat,z}$ at any location (z) as follows:

$$P_{sat,z} = P_{in} - L_z \cdot \left(\frac{\Delta P}{L}\right) \tag{8}$$

Where:

$$\Delta P = P_{in} - P_{out} \tag{9}$$

P_{in}, P_{out} : is the refrigerant pressure at inlet and outlet of the test section tube respectively .

3.2 Refrigerant Vapor Quality:

The vapor quality at the inlet of the test section tube (x_{in}) is expressed in term of the local enthalpy as follows [10]:

$$x_{in} = \frac{h_{i,t} - h_{l,i}}{h_{fg,i}} \tag{10}$$

Where: $h_{l,i}$ is the specific enthalpy of the liquid refrigerant at the inlet of test section and $h_{fg,i}$ is the latent heat of condensation of the refrigerant at the inlet of test section (from R134a thermo-physical properties table) .

The specific enthalpy of the refrigerant at the inlet of test section $h_{i,t}$ represents the specific enthalpy of the refrigerant at the outlet of pre-condenser, which can be determined by applying an energy balance on the pre-condenser as follows :

$$h_{i,t} = h_{i,pre} - \frac{Q^{\cdot}_{pre}}{m^{\cdot}_{ref}} \tag{11}$$

Where: $h_{i,pre}$ is the specific enthalpy of the refrigerant at the inlet of pre-condenser (kJ/kg).

Vapor quality (dryness fraction) of the refrigerant at each position (x_z) is calculated by:

$$x_z = x_{in} - L_z \cdot \left(\frac{\Delta x}{L}\right) \tag{12}$$

Where: Δx is the vapor quality difference of the refrigerant between inlet and outlet of the condenser tube which is expressed by:

$$\Delta x = x_{in} - x_{out} \tag{13}$$

x_{out}: Vapor quality of the refrigerant at the outlet of test section tube which is calculated by:

$$x_{out} = \frac{h_{o,t} - h_{l,o}}{h_{fg,o}} \tag{14}$$

$h_{l,o}$: Specific enthalpy of the liquid refrigerant at the outlet of the test section.

$h_{fg,o}$: Latent heat of condensation of the refrigerant at the outlet of the test section.

$h_{o,t}$: Specific enthalpy of the refrigerant at the outlet of the test section tube which is determined by applying an energy balance on the test section [12]:

$$h_{o,t} = h_{i,t} - \frac{Q^{\cdot}}{m^{\cdot}_{ref}} \tag{15}$$

3.3 Frictional Pressure Drop

Total pressure gradient in the test section tube is calculated by:

$$-\frac{dp}{dz} = -\frac{\Delta P}{L} \tag{16}$$

$$\Delta P = \Delta P_{fr} + \Delta P_m + \Delta P_{st} \tag{17}$$

Where, ΔP is determined from equation (9) and:

ΔP_{fr} : Frictional pressure drop of the refrigerant flow due to shear at the tube surface and at the vapor-liquid interface in the test section (kPa).

ΔP_m : Momentum pressure drop due to the acceleration of the two phase refrigerant flow in the test section (change in kinetic energy) (kPa).

ΔP_{st} : Pressure drop of the refrigerant flow due to the static pressure change in the test section (kPa).

ΔP : Total pressure drop of the refrigerant flow in the test section tube (kPa) which is determined experimentally using the following equation:

Pressure drop of the refrigerant flow due to the static pressure change ΔP_{st} can be neglected because the test section is a horizontal tube and thus no change in the static pressure along the tube, so the frictional pressure drop ΔP_{fr} can be determined by:

$$\Delta P_{fr} = \Delta P - \Delta P_m \tag{18}$$

Where , ΔP_m is determined using homogeneous model by [13]:

$$\Delta P_m = G^2 \cdot \left(\frac{1}{\rho_g} - \frac{1}{\rho_l} \right) \cdot |\Delta x| \tag{19}$$

$|\Delta x|$: The absolute value of the vapor quality change along the test section tube.

4. Results and Discussion

4.1 Experimental results

Figures 4 and 5 show the effect of heat flux measured in range of 10.49 to 17.17 kW/m^2 on the local heat transfer coefficient at fixed mass fluxes 53.17 and 63.63 kg/m^2.s respectively. It can be observed that, the heat transfer coefficient is increased with increasing of vapor quality and continuously decrease with vapor quality for all the values of heat flux. The heat transfer coefficient was directly proportional with heat flux and relatively higher value of heat transfer coefficient was observed at heat flux 17.17 kW/m^2 . The increase in heat flux at fixed difference between refrigerant and tube wall temperatures lead to enhance the heat transfer corresponds to newton law of convection heat transfer. A significant reduction in heat transfer coefficient was noticed at lower vapor quality (x<0.3) and the local heat transfer coefficient at constant mass flux 63.63 kg/m^2 was higher for heat flux 17.17 in range of 47% than that for relatively lower heat flux 10.49 kW/m^2. The variation of local heat transfer coefficient with vapor quality at fixed heat flux 10.49 kW/m^2 and different mass fluxes 42.64, 53.17 and 63.63 kg/m^2.s is shown in Fig.6. It can be seen that the higher values of heat transfer coefficient are achieved by a greater mass flux 63.63 kg/m^2.s due to the contribution of forced convection which is influenced by relatively higher refrigerant mass flow rate and then greater flow velocity in condenser channel. The same behaviour of heat transfer coefficient can be observed in Fig. 7 for heat flux 17.17 kW/m^2 . It is evident that, the values of local heat transfer coefficient was higher at 17.17 kW/m^2 compared to heat flux of 10.49 kW/m^2 resulted from the effect of cooling load in condenser tube at constant test conditions . The local heat transfer coefficient at constant heat flux 17.17 kW/m^2 was higher for mass flux 63.63 kg/m^2 by about of 14% than that for relatively lower mass flux 42.64 kg/m^2 .

The effect of saturation temperature of the refrigerant R-134a on the local heat transfer coefficient at constant mass flux 63.63 kg/m^2.s for three tested values of temperature 48.51℃, 47.64 ℃ and 45.97 ℃ is shown in Fig.8. It can be seen that, the increase of saturation temperature leads to a significant variation in the heat transfer coefficient. This behaviour can be explained by the fact that, the rise in saturation temperature will increase the value of refrigerant–tube wall difference and leads to a reduction in condensation heat transfer coefficient at constant mass and heat fluxes. The value of local heat transfer coefficient for saturation temperature 45.97℃ was higher of 51% than that for a relatively higher temperature 48.51℃ for constant mass flux. Fig.9 depicts the effect of inlet vapor quality of the refrigerant R-134a on the local heat transfer coefficient in the condenser tube at a constant heat flux 17.17 kW/m^2 and mass flux 63.63 kg/m^2. S . It can be observed for the three different tested values of the inlet vapor quality 0.8868, 0.6915, and 0.6455, the local heat transfer coefficient tends to partially increase with inlet vapor quality through most values of local vapor quality in the condenser tube due to the dominance of forced convective condensation at high vapor quality . The local heat transfer coefficient for vapor quality 0.8868 was higher in range of 43% than that for a relatively lower value 0.6455. Fig. 10 shows the frictional pressure drop variation with vapor quality for mass fluxes (42.64 and 63.63 kg/m^2.s) and constant heat flux 17.17 kW/m^2. It can be seen that, the pressure drop of the refrigerant for both mass fluxes decrease with vapor quality along condenser tube, but the pressure drop for mass flux 63.63 kg/m^2.s was higher than that for 42.64 kg/m^2.s in range of 12% due to the shear effect of the refrigerant flow on inner surface of the tube. A similar trend of pressure drop variation can be observed in Fig.11 for constant mass flux 63.63 kg/m^2.s. and different heat fluxes, but the pressure drop for heat flux 17.17 kW/m^2 was relatively higher than that for 10.49 kW/m^2 in the range of 49% at constant test conditions.

4.2 Comparison of Experimental and Predictive Models Results

The theoretical results are determined using three of the most quoted predictive models, Shah, Cavallini and Thome [14]-[16]. These theoretical results are calculated by engineering equations solver (EES) software based on similar operating conditions and plotted with experimental results for comparison. The comparison between theoretical and experimental results of heat transfer coefficient as a function of vapor quality for mass flux 63.63 kg/m^2.s and heat flux 17.17 kW/m^2 is shown in Fig.(12). It can be seen from this figure that the theoretical models have well predicted the results within prescribed test conditions. The trend of the theoretical and experimental results is approximately similar with average deviation in range of (8%), (47%) and (98%) for Shah, Thome and Cavallini respectively. This difference in results is due to the assumptions made to the theoretical models and measurement errors in the experimental work. It can be concluded that, Shah and Thome models have revealed a best prediction of heat transfer coefficient and tend to be closer to the experimental results compared to Cavallini model. The experimental results of the present study are validated by comparison with other research work results of [7]. It can be observed that, the trend of both results is approximately similar with some deviation resulted from the differences in operating conditions .

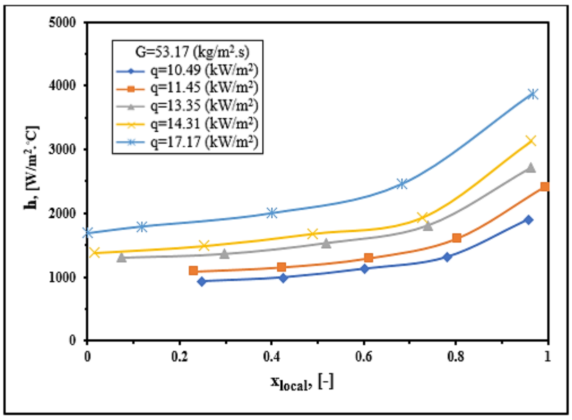

Fig.(4) Effect of heat flux on the local transfer coefficient at G=53.17kg/m^2.s

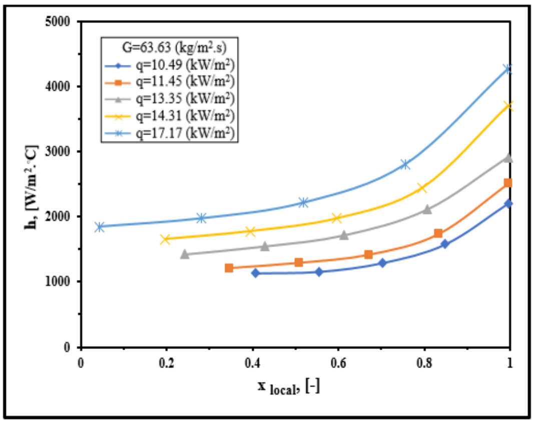

Fig.(5) Effect of heat flux on the local heat heat transfer coefficient at G=63.63kg/m^2.s

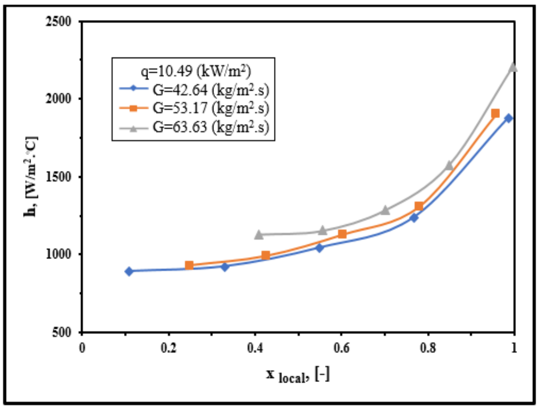

Fig.(6) Effect of mass flux on local heat transfer coefficient at q=10.49 kW/m^2

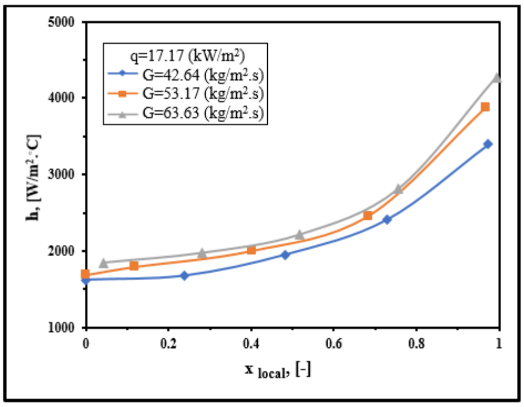

Fig.(7) Effect of mass flux on local heat transfer coefficient at q=17.17 kW/m^2

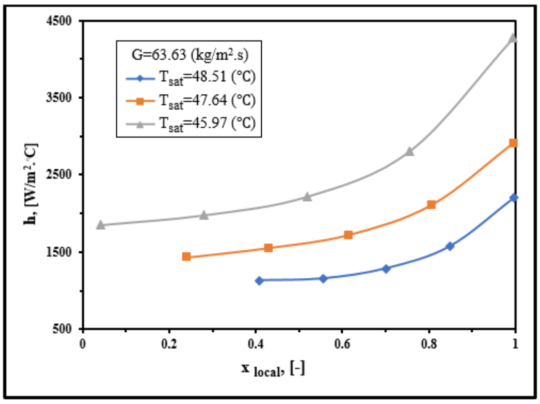

Figure 8. Effect of saturation temperature on local heat transfer coefficient at G =63.63kg/m^2.s

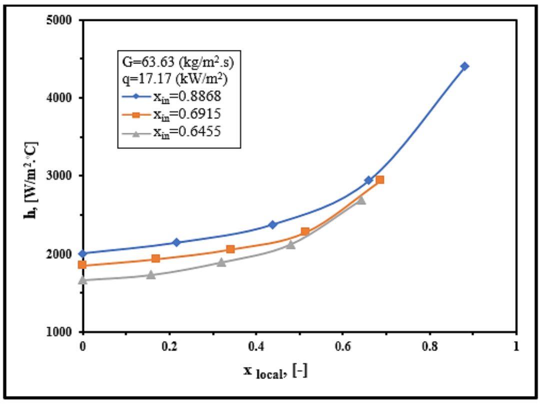

Figure 9. Effect of inlet vapor quality on local heat transfer coefficient at G =63.63kg/m^2.s

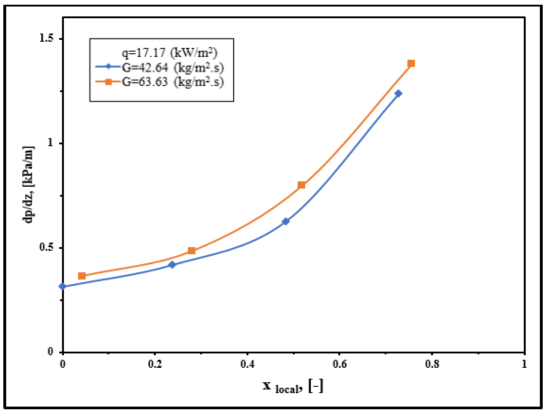

Figure 10. Variation of pressure drop with vapor quality for different mass fluxes and q= 17.17 kW/m2.

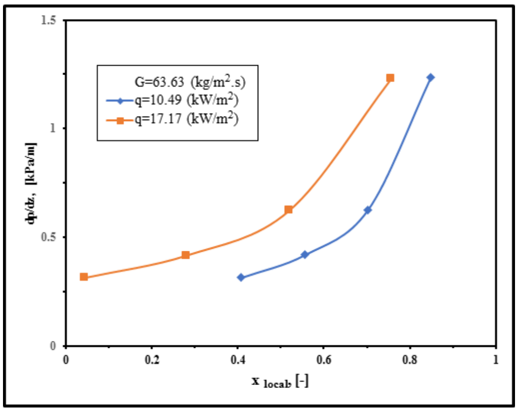

Figure 11. Variation of pressure drop with vapor quality for different heat fluxes and G= 63.63 kg/m2.s

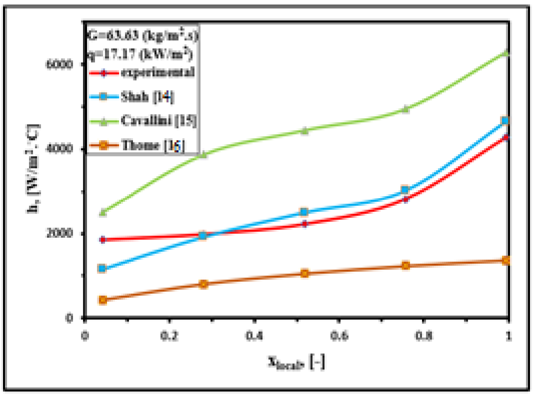

Figure 12. Predicted and experimental heat transfer coefficient as a function of vapor quality for G= 63.63 kg/m2.s and q= 17.17 kW/m2.

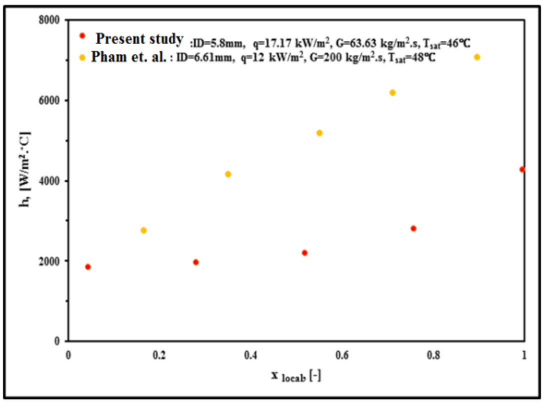

Figure14. Comparison of present study experimental results with Pham *et.al.* 2015 work.

Conclusion

The experimental and theoretical results of the flow condensation heat transfer of the refrigerant R-134a in condenser test section for refrigeration system under different operating conditions of, heat flux, mass flux, vapor quality and saturation temperature are presented and can be concluded as follows :

1- The heat flux has a significant impact on value of local heat transfer coefficient, where the percentage rise for relatively higher heat flux 17.17 kW/m^2 was about 47% compared to10.49 kW/m^2 at constant test conditions .

2- The enhancement in the heat transfer coefficient was about 14% when the refrigerant mass flux increased from 42.64 $kg/m^2.s$ to 63.63 $kg/m^2.s$ at constant test conditions .

3- The enhancement in the heat transfer coefficient was about 51% for the saturation temperature 45.97 ˚C compared to 48.51 ˚C at a fixed refrigerant mass flux and heat flux .

4- For prescribed tested values of heat flux 17.17 kW/m^2 and mass flux 63.63 $kg/m^2.s$, the local heat transfer coefficient has increased in range of 43% for relatively higher inlet vapor quality 0.8868 compared with lower value of vapor quality 0.6455 .

5- The pressure drop of the refrigerant flow in the condenser test section was significantly influenced by mass flux and heat flux. The pressure drop was increased with heat flux and mass flux in range of 12% and 49% respectively .

6- The comparison between the experimental and theoretical results has showed an acceptable agreement with average deviations resulted from the assumptions made to the theoretical models and measurement errors of the experimental work .

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استقصاء تجريبي لخصائص انتقال الحرارة لتكثيف بخار مائع التثليج في R-134a مبادل حراري افقي

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الخلاصة

قدم هذا البحث دراسة تجريبية لمعامل انتقال الحرارة وانحدار الضغط خلال تكثيف مائع ثنائي الطور في حالة جريان لمائع التثليج R-134a في مقطع أنبوب مكثف لمنظومة تبريد تحت ظروف تشغيل مختلفة. الاستقصاء العملي والنظري اعتمد ظروف اختبار تم تحديدها، للفيض الحراري ((10-17 كيلوواط/متر مربع، الفيض الكتلي (42-63)كغم/متر مربع. ثانيه، نسبة الجفاف للبخار (1-0.03) ودرجة حرارة التشبع للمائع من (44 الى 49) درجة مئوية. الاختبارات العملية نفذت باستخدام جهاز فحص يتضمن مقطع فحص لمحاكاة مبادل حراري من نوع الانبوب المتداخل مبرد بالماء والذي صمم ونفذ خلال الدراسة الحالية. أظهرت النتائج العملية وجود تأثير واضح للفيض الحراري والفيض الكتلي على قيم معامل انتقال الحرارة. قيم معامل انتقال الحرارة قد ازدادت مع زيادة الفيض الحراري والفيض الكتلي ضمن ظروف الاختبار المحددة، حيث كانت نسبة التحسين في معامل انتقال الحرارة بحدود 47% و14% لقيم الفيض الحراري والفيض الكتلي الأعلى نسبيا على التوالي. نسبة التحسين في معامل انتقال الحرارة كانت بحدود 51% و43% لقيم درجة حرارة التشبع ونسبة الجفاف للمائع 45.47°C و 0.88 على التوالي مقارنة بالقيم الأخرى عند ظروف تشغيل ثابته. انحدار الضغط كان اعلى بحدود 12% و 49% لقيم الفيض الكتلي والفيض الحراري الأعلى نسبيا على التوالي. تم التثبت من نتائج الدراسة الحالية من خلال المقارنة مع نتائج النماذج النظرية وكذلك مع نتائج دراسة مشابهه وقد أظهرت المقارنة توافق مقبول.

الكلمات المفتاحية: معامل انتقال الحرارة، تكثيف في حالة الجريان، الفيض الحراري، الفيض الكتلي.