ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 5, Issue 7, January 2016

Influence of hydraulic diameter on heat transfer and pressure drop during condensation of R-134a refrigerant in a brazed plate fin heat exchangers

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Abstract*— This paper presents the influence of hydraulic diameter on experimental heat transfer coefficient and pressure drop measured during R-134a saturated vapour condensation inside a brazed compact plate fin heat exchanger with serrated fins. The effects of saturation temperature (pressure), refrigerant mass flux, refrigerant heat flux, effect of fin surface characteristics and fluid properties are investigated. The average condensation heat transfer coefficients and frictional pressure drops were determined experimentally for refrigerant R-134a at four different saturated temperatures (34 ^{0}C, 38 ^{0}C, 40 ^{0}C and 42 ^{0}C). A transition point between gravity controlled and forced convection condensation has been found for a refrigerant mass flux around 20 kg/m^{2}s. In the forced convection condensation region, the heat transfer coefficients show a three times increase and 1.5 times increase in frictional pressure drop for a doubling of the refrigerant mass flux. The heat transfer coefficients show weak sensitivity to saturation temperature (Pressure) and great sensitivity to refrigerant mass flux and fluid properties. Test condensers are selected with three different hydraulic diameter (1.1894 mm , 1.345 mm and 1.7461 mm) serrated fins to analyse the affect of heat transfer and pressure drop characteristics. The condensation two phase heat transfer coefficient has decreased and frictional pressure drop has increased as hydraulic diameter reduced due to increase in heat transfer area and reduction in free flow area respectively. The frictional pressure drop shows a linear dependence on the kinetic energy per unit volume of the refrigerant flow. The correlations are provided for the measured heat transfer coefficients and frictional pressure drops.

***Index Terms*— Compact Plate fin heat exchangers, serrated fin surface, condenser, R-134a, Phase change, hydraulic diameter and experimental setup.**

Nomenclature

A	Heat transfer area (m^2)
a	Ratio of fin area to total area
Af	Free flow area (m^2)
CFD	Computational Fluid Dynamics
Cp	Specific heat capacity (J kg^{-1} K^{-1})
DM	De-Mineralized
Dh	Hydraulic diameter (m)
EES	Engineering Equation Solver
FPI	Fins per inch
f	Fanning friction factor
G	Mass flux (kg m^{-2} s^{-1})
g	Acceleration due to gravity (m s^{-2})
H	Specific Enthalpy (J kg^{-1})
h	Heat transfer coefficient (W m^{-2} K^{-1}), Fin height (m)
hp	Horse power
j	Colburn factor
KE	Kinetic Energy (J)
l	Serration length (m)
L	Flow length (m)
m	Mass flow rate (kg s^{-1})
Nu	Nusselt number (hDh k^{-1}), dimensionless
P	Pressure (Pa)
PHE	Plate Heat Exchanger
Pr	Prandtl number (μCp k^{-1}), dimensionless
Q	Heat load (kW)
q	Heat flux (kW m^{-2})
Re	Reynolds number
s	Fin spacing (m)
SCADA	Supervisory Controller & Data Acquisition System
T	Temperature (°C)
t	Fin thickness (m)
U	Overall heat transfer coefficient (W m^{-2} K^{-1})
V	Velocity (m s^{-1}), volume (m^3)
x	Vapor quality of refrigerant

Greek Symbols

Δ	Difference
λ	Thermal conductivity (W m^{-1} K^{-1})
μ	Dynamic viscosity, Ns m^{-2}
υ	Specific volume, m3 kg^{-1}
ρ	Density (kg m^{-3})
η_f	Fin efficiency
η_o	Overall surface efficiency

Subscripts

a	momentum
ave	average
c	manifold and ports
eq	equivalent
f	fin, frictional
g	gravity
i	Inlet
l, L	Liquid
o	Overall
p	plate
r	refrigerant
sat	refrigerant saturation
t	total
w	water

I. INTRODUCTION

In the world most of aircraft industries switching over to all electric or more electric systems. Environmental Control

System was one of the major secondary power consumers in the aircraft. The cabin pressurization and air conditioning was done by using engine bleed air. With the advanced technology, the trend is currently to use more electricity in Aircraft, this is called "All Electric Aircraft", in which part or all the systems are electrically driven [1]. All electric environmental control system has been chosen which works on Vapour Compression Refrigeration System. In vapour cycle system, condenser and Evaporator are heat exchangers. To meet the demand on performance, the volume and weight of the heat exchangers should be kept minimum. Typically, a heat exchanger is called compact if the surface area density is greater than 700 m^2/m^3 in either one or more channels of a two stream or a multi-stream heat exchanger, as defined by Shah et al.,[2]. There are also other actual and potential special applications in which compact heat exchanger passage geometry is needed to facilitate the highly efficient heating or cooling process involved in condensation or boiling.

Empirical correlations based on extensive experimental data have been obtained for the condensation heat transfer and frictional pressure drop characteristics of an ozone-friendly refrigerant HFC-134a (hydro fluorocarbon R134a) and other refrigerants in inside tubes and herringbone fins. Yi-Yie Yan et al., [3] reported experimental data on condensation heat transfer and pressure drop of refrigerant R-134a in a vertical plate heat exchanger (PHE). The results indicated that the condensation heat transfer is slightly better for a higher average imposed heat flux, but the associated rise in frictional pressure drop (ΔPf) was larger. Kang et al.,[4] and Han et al.,[5] studied on condensation of R-134a flowing inside helicoidal pipe. With the increase of mass flux, the overall condensation heat transfer coefficients of R-134a increased, and slowly the pressure drops also increased. Longo et al.,[6] investigated the affect of enhanced surfaces on refrigerant condensation heat transfer coefficients. Jokar et al.,[7] studied condensation and evaporation of refrigerant R-134a in PHEs with a plate corrugation inclination angle of 60^0. Wang et al.,[8] obtained the pressure drop characteristics of complete and partial condensation in a PHE. They found that during steam condensation in the PHE, the condensate film flow was laminar initially but changed to turbulent flow very quickly in the wavy channel.

In phase change the heat transfer of a compact heat exchanger, in addition to fluid properties and geometrical parameters, the heat flux, mass flux and vapour quality also affect the heat transfer coefficients. Extensive experimental data and empirical correlations were obtained for the condensation heat transfer and pressure drop characteristics of refrigerant R-134a and other refrigerants on inside helicoidal pipe and Herringbone fins by Yi-Yie Yan et al.,[3]; Kang et al.,[4]; Longo et al.,[6]; Giovanni A Longo,[7,8,9,10 & 11] and Amir Jokar et al.,[14]. However, none of these correlations can't be used directly for the present study, where fin geometry is different, as these correlations do not have a predictive character.

The components holding the largest amount of refrigerant in a refrigeration unit are the condenser and evaporator. Therefore Palm [15] indicated that, heat exchangers with small internal volume on refrigerant side should be used for same required system performance. So, compact heat exchangers are having higher compact values compared to plate heat exchangers. The compactness of brazed plate fin heat exchanger results in low refrigerant charge and lower environmental impact. Quantitative data are required before a heat exchanger can be designed as condenser, and this data can be generated only through a dedicated experimental program accompanied by rigorous or numerical analysis. The literature survey indicates that very limited research efforts have been devoted to the condensation of R-134a in Compact plate fin heat exchangers and no data is available for serrated type fin surfaces in open literature.

Plate fin heat exchangers with a variety of fin-types have been designed for these applications. The surface shape of the plate determines the surface performance, which is described by non-dimensional heat transfer coefficient and friction factor relationships [16]. The mechanism of condensation in compact heat exchangers is essentially those of gravity-controlled film condensation, for which a Nusselt-type theory applies and a shear-controlled process in which convective mechanism dominate [16]. Compact heat exchangers have an advantage again, in general, in that condensate drainage process is augmented by surface tension drawing the liquid into the corners of the surface, which act as preferred drainage paths and thinning the liquid layer on the active surface [16].

This paper presents the average condensation heat transfer coefficients and frictional pressure drop measured during R-134a inside small brazed compact plate fin heat exchanger. The effects of hydraulic diameter, refrigerant mass flux, saturation temperature (Pressure), average heat transfer coefficient and frictional pressure drops are investigated. The correlation for average refrigerant heat transfer coefficient and frictional pressure drop are developed for the selected serrated fin surfaces.

II. EXPERIMENTAL SET-UP AND PROCEDURES

The experimental test facility is established, to study the condensation of R-134a in different types of serrated fins used in compact plate fin heat exchangers. Schematic diagram of the experimental test facility is shown in Fig. 1, which has five main loops and SCADA system. Specifically, the system consists of a refrigerant loop, Condenser loop, Evaporator loop, super heater loop and De-super heater loop. Refrigerant R134a is circulated in the refrigerant loop, whereas De-mineralized (DM) water is used in all other loops. The condenser and evaporator loops are capable of supplying DM water flow at a constant temperature in the range of 5 ^{0}C to 60 ^{0}C with stability within ± 0.5 K. In order to obtain different conditions of R-134a in the test heat exchanger, it is required to control the temperature and flow rate of the working fluids in the other loops.

ISSN: 2277-3754
ISO 9001:2008 Certified
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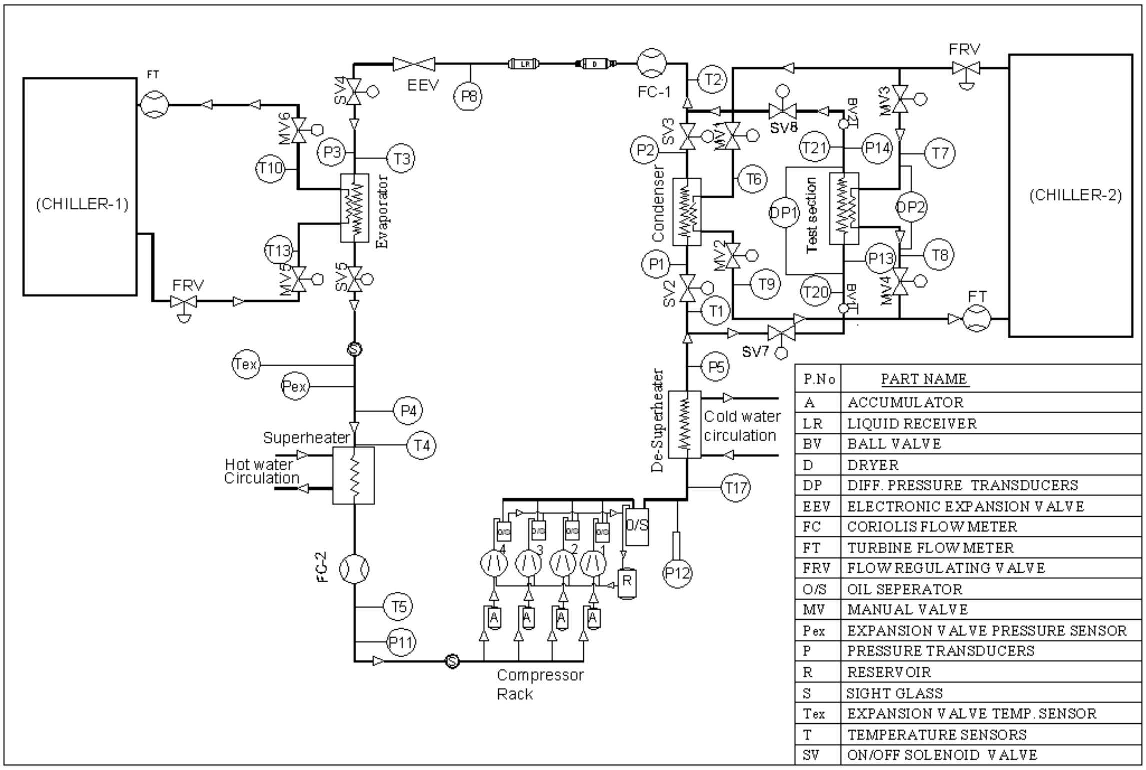

Fig. 1. Schematic View of the Experimental Test Rig

The refrigerant loop contains the compressor rack, accumulators, oil separators, De-Super Heater, Condenser, Test Condenser (Brazed compact plate-fin heat exchanger), liquid receiver, filter drier, sight glasses, electronic expansion valve, evaporator, super heater and Coriolis flow meter. Refrigerant loop is a basic Vapour Compression Refrigeration System, consisting of four parallel semi hermitically sealed variable speed reciprocating compressors having different capacity, Test Condenser, Condenser, Electronic Expansion valve and Evaporator. The variable speed reciprocating compressor rack is designed to control the flow rate between 0.005 kg/sec to 0.06 kg/sec in steps of 0.005 kg/sec using variable frequency drive. The flow rate was measured by a mass flow meter (Micro motion DN4) installed after the test condenser with an accuracy of ±0.05%.

The de-super heater installed in the downstream of compressor rack, controls the inlet conditions to test condenser by water cooling system. After condenser, the liquid refrigerant flows back to the receiver. The filter drier provides high water adsorption at low and high condensing temperatures, as well as at low and high degrees of humidity. The liquid line filter drier protects refrigeration system from moisture and solid particles. Coriol is flow meter is installed in the circuit after evaporator and after condenser to measure the refrigerant flow rate. The system also has sight glasses for physical verification of state of the refrigerant R-134a. The standard brazed plate heat exchanger is installed in the main circuit for calibration of test rig. After attaining the required conditions the condenser is bypassed by the test heat exchanger which is installed as a separate test section. This test condenser provides a water circuit to provide the cooling fluid, where the water flow rate can be varied between 0.08 kg/s to 0.7 kg/s. The pressure transducers and temperature sensors are located at the entrance and exit of the Test Condenser. The accuracy ranges of measuring devices are listed in the Table 1.

	Device	Type	Accuracy
1	Resistance thermometers	RTD	0.15°C
2	Thermocouples	Type K	0.5°C
3	Refrigerant flow meter	Coriolis effect	0.05%
4	Water flow meter	Turbine	0.25%
5	Pressure transducers	Stain gage type Range: 0-20 bar g	0.25% full scale
6	Differential pressure transducers	Stain gage type Range:0-5 PSI	0.025% full scale

Table 1: Accuracy ranges of measuring devices

III. BRAZED COMPACT PLATE FIN HEAT EXCHANGER

The test condensers were a brazed compact plate fin heat exchanger with serrated fin surface. The schematic of the serrated fin geometry is shown in Fig. 2. The flow in offset fins is highly turbulent even at low Reynolds number. The test condensers were basically a cross flow arrangement with two passes on the water side and one pass on R-134a side. The schematic representation of Test Core is shown in Fig. 3. It is made of Aluminum alloy because of its low density, high thermal conductivity and high strength at low temperature. The geometric data of fins used in the test condensers are given in Table 2.

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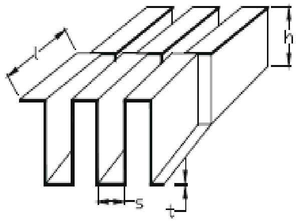
The basic principles of plate fin heat exchanger manufacture [17] are same for all sizes and materials. The fin surfaces, side-bars, parting sheets and cap sheets are held together in a jig under a predefined load, placed in a furnace and brazed to form the plate fin heat exchanger block. Vacuum brazing process was used for brazing of test condenser. The brazed test condenser with inserted wires is shown in Fig. 3(a). The headers and nozzles are then welded to the block, taking care that the brazed joints remain intact during the welding process. In all the three test condensers (TC1, TC2 and TC3), DM water side same type of fin was used. In R-134a circuit of test condensers TC1, TC2 and TC3 the hydraulic diameter of serrated fin was 1.1894 mm, 1.345 mm and 1.7461 mm respectively.

The temperatures of refrigerant and water at the inlet and outlet of the test condenser and evaporator are measured by using resistance thermometer (RTD). The refrigerant pressures at the inlet of test condenser and the evaporator are measured by strain gauge pressure transducer; the test condenser pressure drop is measured by a strain-gauge differential transducer. The refrigerant mass flow rate is measured by Coriolis Effect mass flow meter and water flow rates through the test condenser and evaporator are measured by turbine flow meter. The data acquisition system is used, to collect the data of pressure, temperature, differential pressure, Water flow rate, refrigerant flow rate at different test points and the repeatability of experimental data is ensured. It includes the Industrial PC with 187 Channel Proficy SCADA, which gives the measured data in the form of digital output. The pressure transducer, differential pressure transducer, turbine flow meter, Coriolis flow meter needs a power supply of 4-20 mA electric current as a driver for output.

Parameter	Unit	Common	TC1	TC2	TC3
		DM Water side	Refrigerant side		
Type		Serrated (Lanced & Offset)			
Material		Aluminium			
Fin frequency	FPI	28	30	28	18
Fin height	mm	5	3.05	5	3.8
Fin thickness	mm	0.127	0.1016	0.127	0.254
Serration length	mm	3.175	3.175	3.175	1.588
No of layers	No.	2	1	1	1
Hydraulic diameter	mm	1.345	1.1894	1.345	1.7461

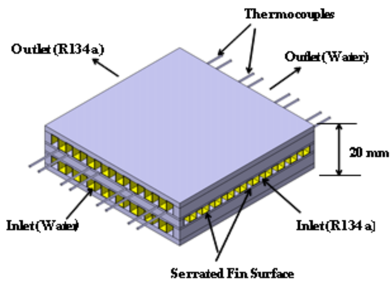
Table 2: Geometric data of fins in Test Condenser (TC)

a) 3D view of serrated fin



b) Fin dimension

Fig. 2 Schematics of Serrated Fin Geometry



a) Test Condenser Core

b) Test Condenser

Fig. 3 Schematic diagram of Test Condenser

IV. VALIDATION OF TEST FACILITY

Initially the testing was carried out with industrial brazed PHE in condenser loop to validate the test facility. During operation, the vapour refrigerant R-134a coming out from a compressor rack condenses in the condenser and the liquid refrigerant flows through Coriolis flow meter, collector tank and filter dryer. The quality of R-134a at the condenser inlet was kept at desired value by adjusting the temperature and flow rate of DM water in De-Super heater. The liquid refrigerant then passes through evaporator after expansion in the electronic expansion valve. Any change of the system variables will lead to fluctuations in the temperature, pressure and flow. The heat balance test is performed in the test condenser and evaporator after the system reaching steady state, a deviation to the extent of 5% was considered acceptable, and measured data within the range are

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ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
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considered for further calculation. The thermal properties of the R-134a were taken from thermo physical properties of refrigerants (ASHRAE Fundamental hand book [18]). The experimental data obtained were validated with empirical correlations given by Giovanni A Longo [10 & 12] and Nusselt [19] for plate heat exchangers. The results are found in good agreement in these comparisons.

A. Comparison with open literature

The two phase analytical heat transfer coefficients and frictional pressure drop for the plate heat exchangers are already well established. A commercial brazed plate heat exchanger has been analyzed for a saturation temperature of 36 ^{0}C. Most of the authors like Giovanni A Longo [10 & 12], Yi-Yie et al. [3] and Amir Jokar et al. [14] who did extensive experimental work on plate heat exchangers followed the same methodology. For the validation of the test rig, experiments were conducted on brazed plate fin heat exchanger at condensation saturated temperature 36 ^{0}C as per the methodology followed by the earlier authors [3,10,12 & 14]. The experimental results of plate heat exchanger are compared with Giovanni A Longo [10] and Nusselt [19] results as shown in Fig. 4(a) and Fig. 4(b) for heat transfer coefficient and frictional pressure drop respectively. Experimental results are in well accordance with the analytical results given by Giovanni A Longo [10] for the low mass flux region and the variations are found to be about 8% in heat transfer coefficient and 11% in frictional pressure drop values. Giving exact reasons for variation of these factors may not be possible due to involvement of so many parameters such as manufacturing aspects, testing conditions and uncertainties in measurements.

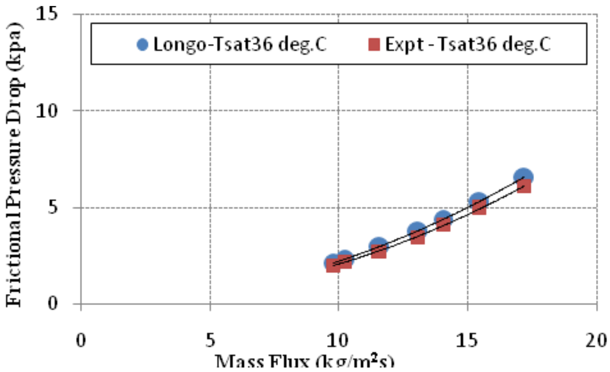

Fig. 4(a): Validation of mass flux vs. ΔP_f

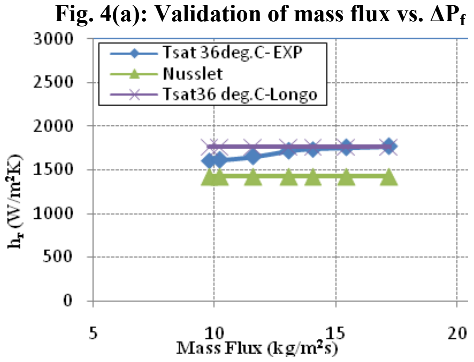

Fig. 4(b): Validation of mass flux vs. h_r

V. DATA REDUCTION METHOD

The data reduction and analysis is carried out in the present investigation to deduce the heat transfer rate from the refrigerant flow to the water flow in the test unit. The amount of heat transfer from the refrigerant to the water can be determined from the energy conservation principle. Therefore, the heat transfer from the refrigerant to the water can be calculated from the following equations:

$$Q_w = m_w C_{pw}(T_{wo} - T_{wi}) \tag{1}$$

Based on Newton's law of cooling, the overall heat transfer coefficient can be determined by

$$Q_w = U A_w \Delta T_{ln} \tag{2}$$

Where ΔT_{ln} , is the logarithm mean temperature difference defined as

$$\Delta T_{ln} = \frac{T_{wo} - T_{wi}}{\ln\frac{T_{rsat} - T_{wi}}{T_{rsat} - T_{wo}}} \tag{3}$$

Where Trsat is the average saturation temperature of the refrigerant derived from the average pressure measured on refrigerant side and Twi, Two are the water temperatures at the inlet and outlet of the test unit. The average condensation heat transfer coefficient (hr) on refrigerant side of the test unit is derived from the global heat transfer coefficient (U) assuming no fouling resistance.

$$\frac{1}{U} = \frac{1}{\eta_{or} h_r} + \frac{t}{\lambda\left(\frac{A_p}{A_r}\right)} + \frac{1}{\eta_{ow} h_w\left(\frac{A_w}{A_r}\right)} \tag{4}$$

where Aw is the water side area, hr and hw are the refrigerant side and water side heat transfer coefficients respectively and η0 is the overall fin surface efficiency, which is defined as;

$$\eta_o = 1 - a(1 - \eta_f) \tag{5}$$

in which 'a' is the ratio of the fin area to the total area,

$$\eta_f = \frac{\tanh(ml)}{ml} \tag{6}$$

Where $m = \sqrt{\frac{2h_w(1+\frac{t}{l})}{\lambda * t}}$ and $l = \frac{h}{2}$ (7)

Equation (4) shows that, if the heat transfer coefficient at the water side is known, the average condensation heat transfer coefficient hr can be directly calculated from the experimental data. For general cases the Wilson plot technique could be used for finding both heat transfer coefficients as explained in Kumar et.al.[20] and Styrylska et.al.,[21]. This method has been widely applied for separating individual resistances from an overall resistance. Unfortunately, due to lack of data, it is not possible to apply this method in the present investigation using the test rig. Hence, the water side heat transfer coefficient hw is estimated separately as explained in the following sections.

A. *Water side heat transfer coefficient*

Generally, the single-phase heat transfer coefficient can be expressed with the Colburn j factor,

$$j_w = \frac{h_w}{G\,C_p} Pr^{2/3} \tag{8}$$

For the present fin geometry, the Reynolds number for water flow is defined as

$$Re_w = \frac{\rho V D_h}{\mu} = \frac{G D_{hw}}{\mu} \tag{9}$$

Where

$$G = \frac{m}{A_f} \tag{10}$$

And

$$D_{hw} = [2(s-t)h]/[(s+h) + (\frac{t*h}{l})] \tag{11}$$

An extensive literature survey has been carried out to find a correct correlation on the water side heat transfer coefficient for serrated fin surface. The heat transfer coefficient and friction factor correlations presented by Pallavi and Ranganayakulu ., [22], using CFD are compared with Weiting [23], Manglik and Bergles [24] and Joshi ,Webb [25], which were obtained from the experiments with air flow. Sen Hu and Keith E Herold, [26-27] have published the affects of the Prandtl number on Colburn j factor for water and polyalphaolefin (PAO) fluids and claimed that, air models over-predict the j factor for liquids and found that the heat transfer coefficient of liquids is approximately two times larger than that of air. Hence, a detailed analysis has been carried out using ANSYS Fluent tool to estimate the 'j' and 'f' factors for Serrated fins in accordance with Ranganayakulu et.al.,[28] the detailed description of this procedure is reported in Ramana Murthy et.al., [29].

The correlation for water side heat transfer coefficient for the above selected serrated fin surface in accordance with Ramana Murthy K V et.al.,[29] in laminar range and turbulent range are as follows:

for Laminar Range ($100 \leq Re \leq 800$)

$$Nu_w = 0.049\, Re_w^{0.69} Pr^{\frac{1}{3}} \tag{12}$$

Then, $$h_w = 0.049 \left(\frac{\lambda_w}{D_h}\right) Re_w^{0.69} Pr^{\frac{1}{3}} \tag{13}$$

for Turbulent Range ($1000 \leq Re \leq 15000$)

$$Nu_w = 0.016\, Re_w^{0.85} Pr^{\frac{1}{3}} \quad , \tag{14}$$

Then, $$h_w = 0.016 \left(\frac{\lambda_w}{D_h}\right) Re_w^{0.85} Pr^{\frac{1}{3}} \tag{15}$$

The steady state conditions for the test condenser are determined by checking the heat balance between the released heat from R-134a and the heat absorbed by the cooling water. The amount of heat released from R-134a can be calculated by the following equation:

$$Q_r = m_r\, \Delta H \tag{16}$$

Where Qr, mr and ΔH are the heat load, mass flow rate and enthalpy drop of R-134a across the test condenser respectively. The criterion used for the steady state condition during the test is given below:

$$\frac{|Q_r - Q_w|}{Q_{ave}} \leq 5\% \tag{17}$$

Where Q_{ave}, is the average value of heat exchanged between R-134a and the cooling water and is determined by the following relation:

$$Q_{ave} = \frac{1}{2}(Q_r + Q_w) \tag{18}$$

B. Pressure drop measurements

To measure the frictional pressure drop on the refrigerant side is computed in accordance with Yi-Yie Yan et al.,[3] by subtracting the pressure losses at the test condenser inlet and exit manifolds and ports (ΔPc) then adding the momentum pressure rise (deceleration, ΔPa) and the gravity pressure rise (elevation, ΔPg) to the total pressure drop measured (ΔPt).

$$\Delta Pf = \Delta Pt - \Delta Pc + \Delta Pa + \Delta Pg \tag{19}$$

The momentum and gravity pressure drops are estimated by the homogeneous model for two phase gas-liquid flow as per J.G.Collier [30] as follows:

$$\Delta Pa = G^2 (v_g - v_l)|\Delta x| \tag{20}$$

$$\Delta Pg = g\, \rho_m\, L \tag{21}$$

Where v_g and v_l are the specific volume of liquid and vapour phase, |Δx| is the absolute value of the vapour quality change between inlet and outlet and

$$\rho_m = [X_m/ \rho_g + (1-X_m)/\rho_l]^{-1} \tag{22}$$

is the average two-phase density between inlet and outlet calculated by the homogeneous model at the average vapour quality Xm between inlet and outlet. The pressure drop in the inlet and outlet manifolds and ports are empirically estimated, in accordance with Shah and Focke [31] as follows:

$$\Delta Pc = 1.5\, G^2/ (2\, \rho_m) \tag{23}$$

VI. ANALYSIS OF THE RESULTS

Using the CFD correlations on water side, the average heat transfer coefficient of refrigerant has been calculated for the three different serrated fin surfaces used in Test Condenser. The condensation tests were carried out with R-134a down-flow and water horizontal–flow at four different saturated temperatures (34 ^{0}C, 38 ^{0}C, 40 ^{0}C and 42 ^{0}C) to determine the effect of saturated temperature on the condensation heat transfer and pressure drop of R-134a inside a brazed plate fin heat exchangers. The tests were conducted on three different Test Condensers to study the effect of hydraulic diameter. Table.3 indicates the operating conditions in the condenser under experimental tests.

The mass flow rate and heat flux of R-134a was varied from 16 to 46 kg/m^{2}s and 7 to 33 kW/m^2 respectively in the test condenser. Fig. 5 and Fig. 6 shows the effect of refrigerant mass flux on refrigerant heat transfer coefficient for refrigerant saturated temperatures 34 ^{0}C and 40 ^{0}C respectively at three different hydraulic diameters of serrated fin geometries. Results are indicating that the effect of hydraulic diameter much significant on heat transfer coefficient. When the hydraulic diameter decreases for the same mass flux and heat flux the heat transfer coefficient increased due to increased heat transfer area. Also, due to capillarity in brazed plate fin surface, the thickness of condensate film reduces and increases the heat transfer coefficient. The effect of refrigerant mass flux on heat transfer coefficients were constant when the mass flux is less than 20 kg/m^{2}s, probably the condensation is controlled by gravity, which is shown in the Fig. 5 and Fig. 6. At low

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 5, Issue 7, January 2016

refrigerant mass flux (< 20 kg/m^{2}s) the condensation heat transfer coefficients are not depend on mass flux and are predicted by Giovanni [10]. For higher refrigerant mass flux (> 20 kg/m^{2}s), the heat transfer coefficient increased as mass flux increased due to forced convection condensation, shown in the Fig. 5. and Fig. 6. For higher refrigerant mass flux (> 20 kg/m^{2}s) the heat transfer coefficients depend on mass flux and are well predicted by Akers et al.[32]. The average refrigerant heat transfer coefficient increased by 21% for a 32% reduction of hydraulic diameter.

Table 3: Operation Conditions during experimental tests

Set	Psat (bar'g')	Tsat (^{o}C)	Quality		G_r	q_r
			Xin	Xout	(kg/m^{2}s)	(kW/m^2)
1st	8.66±0.2	34±0.5	0.94-1	0-0.07	16-42	6.9-31
2nd	9.1±0.2	36±0.5	0.97-1	0-0.03	15-44	8.7-33
3rd	9.6±0.2	38±0.5	0.94-1	0-0.06	25-41	12-30
4th	10.04±0.2	40±0.5	0.93-1	0-0.07	15-45	8.3-33

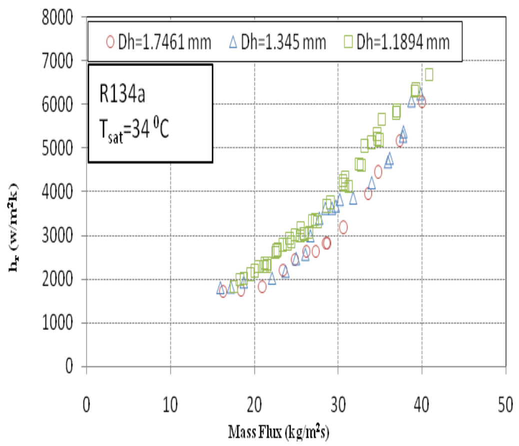

Fig. 5. Refrigerant mass flux vs. Refrigerant heat transfer coefficient at T_{sat} =34^{0}C

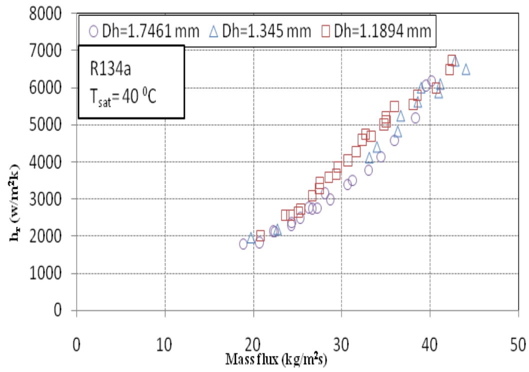

Fig. 6. Refrigerant mass flux vs. Refrigerant heat transfer coefficient at T_{sat} =40^{0}C

Fig. 7 and Fig. 8 indicates the effect of refrigerant mass flux on refrigerant frictional pressure drop for refrigerant saturated temperatures 34 ^{0}C and 40 ^{0}C respectively at three different hydraulic diameters of serrated fin geometries. Results show that, the frictional pressure drop increases as the mass flux increased. Also, it is observed that the frictional pressure drop increases as the serrated fin hydraulic diameter decreases due to decrease in free flow area. The frictional pressure drop varies from 86% to 96% of the refrigerant side total pressure drop. The refrigerant side frictional pressure drop increased by 18% for a hydraulic diameter reduction of 32% at a constant mass flux.

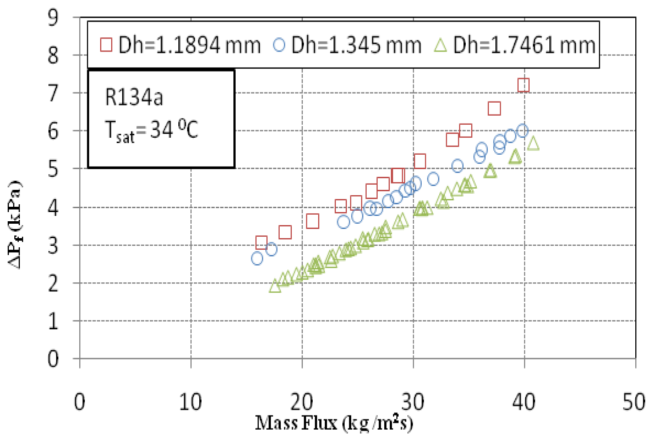

Fig. 7. Refrigerant mass flux vs. Refrigerant Frictional Pressure drop at T_{sat}=34 ^{0}C

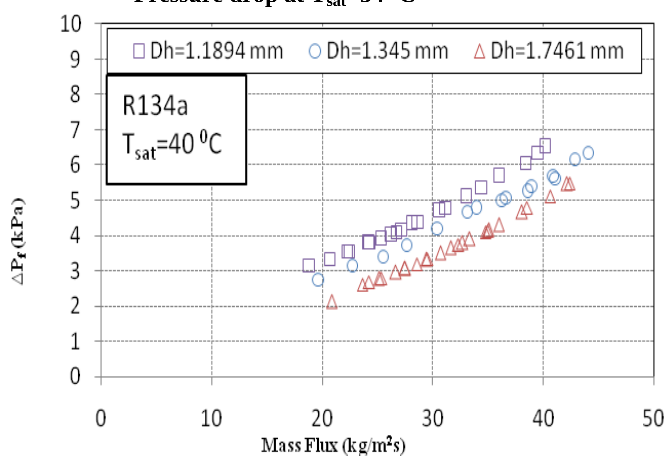

Fig. 8 Refrigerant mass flux vs. Refrigerant Frictional Pressure drop at T_{sat}=40 ^{0}C

All the parameters like flow rates, inlet and outlet pressures and temperatures and pressure drops across the test condenser for both refrigerant and water sides were recorded. The equations of state of R-134a are taken from EES [33]. Using these data, uncertainty analysis has been carried out for estimation of uncertainties in heat transfer and pressure drop parameters using the Engineering Equation Solver (EES) [33] program developed for such a purpose. The analysis indicates the overall uncertainty within ±16.7% for the refrigerant condensation heat transfer coefficient measurement and within ±4.5% for the refrigerant measured frictional pressure drop. The detailed results of the different parameters and their estimated uncertainties are tabulated in Table 4.

Fig.9 indicates the comparison between experimental and calculated (proposed) condensation average refrigerant heat transfer coefficient correlation to the present data. It shows that most of the experimental values are within ± 20%. Fig.10 shows the comparison between experimental and calculated (proposed) condensation frictional pressure drop correlation to the present data. It is found the average deviation is about ± 20% between the frictional pressure drop correlation and the measured data.

Parameter	Uncertainty
Plate length, Width(m)	± 0.0005
Plate area (m2)	± 0.00007
Water inlet temperature, Twi (°C)	± 0.15
Water outlet temperature, Two (°C)	± 0.15
Refrigerant inlet temperatures, Tri (°C)	± 0.15
Refrigerant outlet temperatures, Tro (°C)	± 0.15
Refrigerant inlet pressures, P (%)	± 0.25
Differential Pressure, ΔP (%)	± 0.025
Heat flow rate, Q (%)	± 6.8
Refrigerant mass flux, Gr (%)	± 1.2
Refrigerant heat flux, qr (%)	± 2
Overall heat transfer coefficient, U (%)	± 7.2
Water heat Transfer coefficient, hw (%)	± 5.3
Refrigerant average heat transfer coefficient, h_r (%)	± 16.7
Refrigerant Frictional pressure drop, ΔP_f (%)	± 4.5

Table 4: Parameters and estimated uncertainties

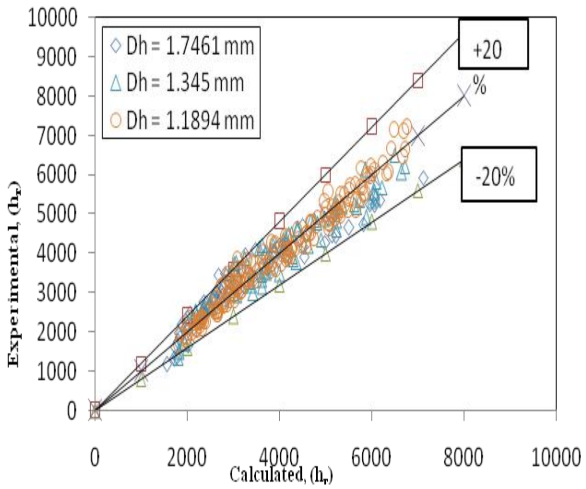

Fig.9. Comparison between experimental and calculated average refrigerant heat transfer coefficient

The correlations have been expressed in terms of equivalent Reynolds number, liquid prandtle number, liquid thermal conductivity and fin hydraulic diameter. The power law expressions have been used for determining the two phase refrigerant heat transfer coefficient and two phase refrigerant friction factor as a function of the equivalent Reynolds number and dimensionless fin parameters.

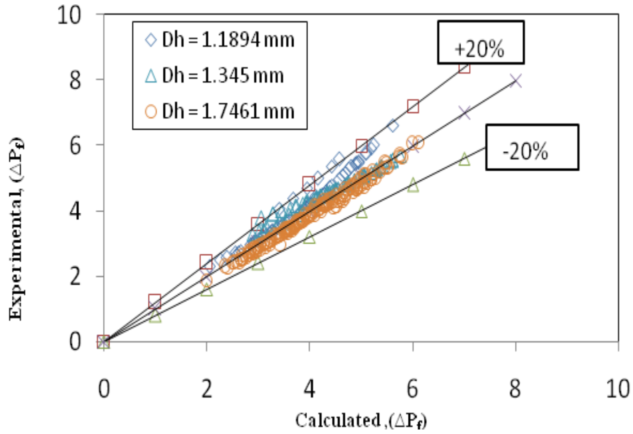

Fig.10. Comparison between experimental and calculated refrigerant frictional pressure drop

The two phase refrigerant condensation heat transfer coefficient (h_r) is functionally related to (λ_L/D_h), Re_{eq}, Pr_L and T_{sat} and it can be represented as:

$$h_r = B(\lambda_L/D_h)^{a1} Re_{eq}^{a2} Pr_L^{a3} \quad (24)$$

where B, a1, a2 and a3 are constants.

Therefore, the proposed equation for this particular serrated fin surface as follows:

$$h_r = 1.583 \times 10^{-4}(\lambda_L/D_h)^{0.3627} Re_{eq}^{1.5678} Pr_L^{4.0336} \quad (25)$$

The above correlations correctly predict 93% of the hr data for equivalent Reynolds number range of 600 to 1800, and root-mean-square error is 3.5%.

Similarly, the two phase serrated fin surface frictional pressure drops are functionally related to (KE/V) and it can be represented as:

$$\Delta P_f = C(KE/V)^{b1} \quad (26)$$

Where C and b1 are constants.

Therefore, the proposed equation for this particular serrated fin surface as follows:

$$\Delta P_f = 1.3023(KE/V)^{0.49769} \quad (27)$$

The above correlations correctly predict 88% of the ΔP_f data for equivalent Reynolds number range of 600 to 1800, and root-mean-square error is 5.5%.

No other experimental data are available in the open literature on the condensation average saturated heat transfer coefficient and pressure drops for R-134a in serrated fin surface. So, the obtained experimental results could not be compared with others.

VII. CONCLUSION

The experimental investigation was conducted to measure the condensation heat transfer coefficient and pressure drop of R-134a in a small brazed plate fin heat exchanger with serrated fin with three different hydraulic diameters. The serrated fins of hydraulic diameter 1.189 mm, 1.345 mm and 1.7461 mm are selected for condensation experiments. The

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 5, Issue 7, January 2016

development of test rig to generate the R-134a condensation heat transfer coefficient and pressure drop in brazed compact plate fin heat exchangers with Serrated (Lance & Offset) fins have been discussed. The experiments were conducted for refrigerant R-134a at four different saturated temperatures (34 ^{0}C, 38 ^{0}C, 40 ^{0}C and 42 ^{0}C) with the mass flux ranging from 16 to 46 kg/m^{2}s. The effects of the mass flux of R-134a, average two phase saturated condensation heat transfer coefficient, core frictional pressure drop and test condenser total pressure drop were discussed in detail. The experimental results illustrates that the condensation heat transfer coefficient and frictional pressure drop increased with refrigerant mass flux. The saturation temperature shall not be much significant on heat transfer coefficient when the mass flux is less than 20 kg/m^{2}s which corresponds to a liquid equivalent Reynolds number around 800 and probably condensation is controlled by gravity as discussed by Giovanni A Longo [10]. For higher refrigerant mass flux (> 20 kg/m^{2}s), the heat transfer coefficient depend on mass flux and forced convection condensation occurs and well predicted by Akers et.al [32]. The experiments were carried out on three different hydraulic diameter serrated fins. The results are indicated that the average refrigerant heat transfer coefficient increased by 21% for a reduction of 32% hydraulic diameter. As the reduction in hydraulic diameter reduces the free flow area and it increases the pressure drop. The frictional pressure drop result shows the 18% increase in refrigerant frictional pressure drop for 32% reduction of hydraulic diameter. A generalized two phase heat transfer coefficient and two phase frictional pressure drop correlation have been developed based on the experimental data. The results indicate as the refrigerant mass flux increases the associated heat transfer coefficient and frictional pressure drops also increased. For a refrigerant mass flux lower than 22 kg/m^{2}s, the two phase saturated refrigerant heat transfer coefficients are not dependent on mass flux. The experimental results follow the trend of average R-134a condensation heat transfer coefficient and associated frictional pressure drop with mass flux are followed as presented by the different authors [4-14]. An empirical correlation was proposed on the basis of the investigation for the calculation of the average two phase condensation heat transfer coefficient. Correlations are presented for the measured average condensation heat transfer coefficient and frictional pressure drop.

ACKNOWLEDGEMENT

The Authors wish to acknowledge Aeronautical Development Agency for allowing publication of the paper.

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