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AN EXPERIMENTAL TEST RIG TO STUDY IN-TUBE CONDENSATION OF VARIOUS REFRIGERANTS

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Abstract

An increase in global warming potential and other environmental concerns are demanding new environmentally friendly refrigerants. For investigation of system performance of newly developed refrigerants, the convective heat transfer coefficient in two-phase flow for both boiling and condensation is required. In the literature, numerous works have been done on in-tube condensation of various refrigerants and it is noticed that most of the works are based on conventional test rigs involving dual refrigeration arrangement without compressor for the in-tube condensation analysis. This work is mainly focused on developing an innovative low cost highly efficient in-tube condensation test rig for flow condensation analysis of refrigerants. The in-tube condensation test rig is successfully fabricated, and the experiments are conducted for R134a refrigerant. The experiments are conducted for mass flux varying from 150 -320 kg/m^{2}s with different vapor qualities ranging from 0.3 to 0.9. Further, the results obtained are validated against MM Shah's unified correlation and the comparative results are also presented it is observed that the experimental results are in good agreement with that of Shah's correlation results.

Keywords: In-tube condensation, Heat Transfer Coefficient (HTC), R134a.

Nomenclature

A_{ti}	:	Inside the surface area of pipe (m^2)
c_{pw}	:	Specific heat of water (J/kgK)
m_r	:	Mass flow rate (kg/s)
h	:	Enthalpy in (kJ/kg)
G	:	Mass Flux of Refrigerant (kg/m^{2}s)
P_r	:	Prandtl number
p_r		Reduced pressure
Q	:	Heat transfer rate (W)
T	:	Temperature (°C)
X	:	Dryness fraction
Z		MM Sha's Correlating factor

Greek Symbols

α	:	Heat transfer coefficient of refrigerant (W/m^2 K)
λ	:	Thermal conductivity (W/m.K)

Subscripts

avg	:	average
exp	:	experimental
i	:	inside, refrigerant side
g	:	gaseous /vapor
f	:	friction factor of pipe
l		liquid
lo	:	liquid only
MMS	:	MM Shah's
nu	:	Nusselt number
p		pipe
PC	:	pre-Condenser
o	:	outside
r	:	refrigerant
s	:	saturated
w	:	water
TC	:	condenser Test Section
1	:	inlet
2	:	exit

1. Introduction

Increased environmental concern and usage of air conditioning require zero ozone depletion potential (ODP) and low global warming potential (GWP) and energy-efficient refrigeration systems. The trend of the last decades forces researchers to invent new refrigerants with zero ODP and energy-efficient heat exchangers. The condenser is such a heat exchanger that it needed to be designed for higher efficiency. Such works have been conducted by researchers to investigate refrigerants in the test section.

Andrea Diani et al. [1], Anand Kumar Solanki et al. [2], A Cavallini et al. [3], Ki Jung Park et al. [4], SHAO Li et al. [5], and Dongsoo Jung et al. [6] conducted an experimental investigation to find condensation HTC in the condenser. Almost, all of them were used pre and post confessors,

Anand Kumar Solanki et al. [2] and Dongsoo Jung et al. [6], Linlin Wang et al. [7], were used secondary condenser secondary axillary chillers as post condensers.

M.K Dabson et al. [8] conducted an experiment to study the HTC and flow pattern of different refrigerants in horizontal tubes. Stratified, wavy, wavy annular, annular, annular mist, and slug flow patterns were observed during condensation. All regimes were characterized by HTC that depended on the difference between wall and refrigerant temperature but were nearly independent of mass flux. They have developed new correlations for different flow regimes and done a comparative study with other sources.

L. LIEBENBERG et al. [9] reviewed recent works on the flow pattern during condensation of various refrigerants. They have strongly proposed the flow pattern influence the heat transfer characteristic during condensation of refrigerants.

R. Suliman et al. [10] conducted experimental tests to study the influence of flow patterns during condensation of R134a with small mass flux. They have developed an improved flow regime nearly at the transition region with a 6 % absolute mean deviation

Subsequently, M.K Bashar et al. [11] Alberto Cavallini et al. [12], J.W. Coleman et al. [13] are conducted experiments to study HTC and flow patterns during the condensation of refrigerants. They developed flow regime maps and the transition lines which can be used to predict the flow pattern or regime.

For the past three decades, MM Sha [14] has been doing research to develop exact correlations for two-phase flow condensation in different conditions. In 1979 MM Sha have proposed a new correlation which was accepted widely during those days. Later it was observed some limitations for low flow rate and moderate pressure. In 2009 MM Shah [15] developed an improved correlation, good agreement with 22 different refrigerants. In the improved correlation, three flow regimes I, and III were identified namely turbulent, laminar, and mixed respectively.

Further MM Sha correlation was extensively examined and compared with correlations developed by Thome et al. [17], Garimella et al. [18], Chen et al. [19], Dalkilic et al. [20], Kim and Mudawar et.al [21], Awad et al. [22] and Del Col et al. [23]. The results were shown a 22% mean deviation. And MM Sha included Weber's numbers because of vapor and liquid phase surface tension during condensation. And modified correlations were reported in [16] same correlation was used to validate the present experimental HTC.

The literature shows that numerous experimental studies were done on in-tube condensation of refrigerants using a similar experimental setup that involves a dual refrigeration system. And experimental setups are involves expensive refrigerant pumps. This work is on an innovative in tube condensation test rig.

2. Experimental Test Rig.

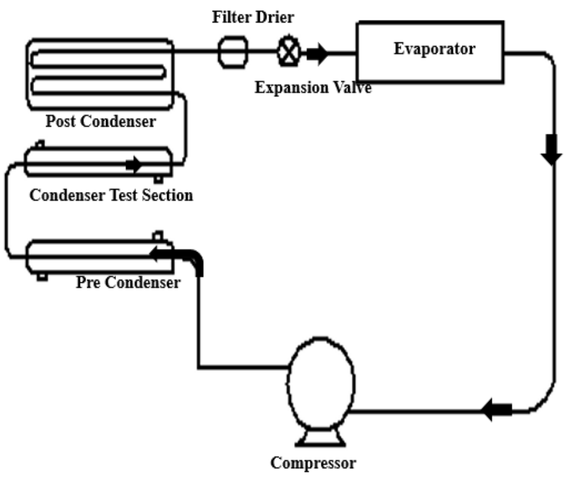

Figure 1 Line diagram of in-tube condensation test rig.

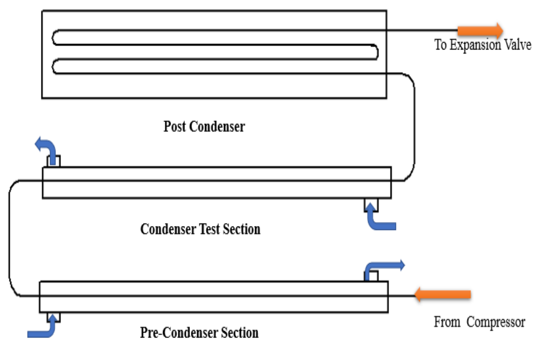

Figure 2 Test section, showing pre and post condensers.

The experimental set-up was designed and fabricated to study the condensation of R-134a inside a plain horizontal tube. The schematic diagram for the experimental setup is shown in Fig. 1.

It consists of an evaporator compressor condenser filter drier and thermostatic expansion valve. The refrigerant from the exit of the evaporator enters the compressor as saturated or super-heated vapor, where it is compressed to high temperature and high-pressure superheated vapor. The first super-heated vapor refrigerant enters the post condenser where the refrigerant is controlled with desired quality and then it is fed into the actual condenser test section.

The refrigerant is condensed into the nearly liquid phase. A filter cum drier was fitted between the post condenser and differential pressure gauge for the removal of any moisture or foreign particle present in the refrigerant loop.

And then refrigerant is a sub-cooled liquid which is ensured by filter drier. This liquid refrigerant is then throttled is-enthalpically to a low-temperature low-pressure liquid-vapor mixture. This mixture enters the evaporator and the cycle repeats.

The actual test section is as shown in "Fig 2" which involves pre and post condensers also. The condenser employed in this work is the flat-tube water-cooled condenser. The high temperature, high-pressure super-heated vapor will enter the pre condenser first. In the pre condenser, the quality of the refrigerant i.e dryness fraction is varied with the help of regulating the cooling water supply rate. The controlled quality of the two-phase refrigerant is then entered into the test section. In the test section, the amount of heat rejected by the refrigerant is the energy balance equation. The test section also has the facility of controlled cooling water circulation. From the exit of the test section, the refrigerant is then sent into the post condenser section where it will be completely condensed by the high rate of water circulation to ensure the liquid refrigerant enters the TXV and filter drier is kept between post condenser and TXV. The unique feature of this test rig is that all the three sections, pre condenser, test section, and post condenser are made in a single condenser with sections. This eliminates the secondary cooling system required for post condensation which is common in conventional in tube condensation test rigs. Thereby this is cost-effective.

3. Conduction of experiment

The experiments are conducted for different saturation temperatures by varying coolant rate and compressor speed. Refrigerant quality to the test section was controlled by the pre condenser, by varying coolant mass flow rate. The change in refrigerant temperature and pressure were recorded. K-type thermocouples with an accuracy of (± 0.05 %) were used to measure temperature.

An 8.4 mm internal diameter, 9.1 mm outer diameter, and 750 mm long plane copper tube was used.

The properties of refrigerant were recorded using REFPROP software [23], E.W. Lemmon et al [24], and all the equations were solved by MAT Lab software to find HTC of R134a refrigerant at test section.

4. Data reduction.

The mass flow rate of the refrigerant is measured by using Haaland equation as follows.

$$\Delta P = \frac{l_p * \rho_l * V_l^2}{2 * d_i} \quad (1)$$

$$\frac{1}{f} = -1.8 \log\left[\left(\frac{e}{3.7 * d_i}\right)^{1.11} + \frac{6.9}{R_{el}}\right] \quad (2)$$

$$R_e = \frac{\rho_l * V_l * d_l}{\mu_l} \quad (3)$$

$$m_r = \rho_l * A_P * V_l \quad (4)$$

The refrigerant vapor quality at the pre-condenser section inlet depends on the heat transfer rate in the pre-condenser.

$$(Q_w)_{PC} = m_w * c_{pw}(T_{w2} - T_{w1})_{PC} \quad (5)$$

Heat carried by the water is equal to the heat lost by the refrigerant in the pre-condenser section.

$$(Q_w)_{PC} = (Q_r)_{PC} \quad (6)$$

the enthalpy of the refrigerant at the exit of pre-condenser is

$$(h_{r2})_{PC} = (h_{r2})_{PC} - (Q_r)_{PC}/m_r \quad (7)$$

The vapor quality of refrigerant is calculated as

$$X_1 = \frac{(h_{r1})_{PC} - h_f}{h_{fg}} \quad (8)$$

Similarly, quality of the refrigerant X_2 at exit of the test section is calculated as follows.

$$X_{avg} = \frac{X_1 + X_2}{2} \quad (9)$$

The HTC of the refrigerant is calculating as follows

$$(Q_w)_{TC} = A_{ti} * \alpha * (T_{rs} - T_{avg}) \quad (10)$$

$$\alpha_{exp} = \frac{(Q_w)_{TC}}{A_{ti} * (T_{rs} - T_{avg})} \quad (11)$$

The condensation HTC of R134a in smooth tube at different operating conditions are obtained. In this section, the effect of mass flux, vapor quality, saturation temperature and refrigerant performance on condensation heat transfer coefficient are reported.

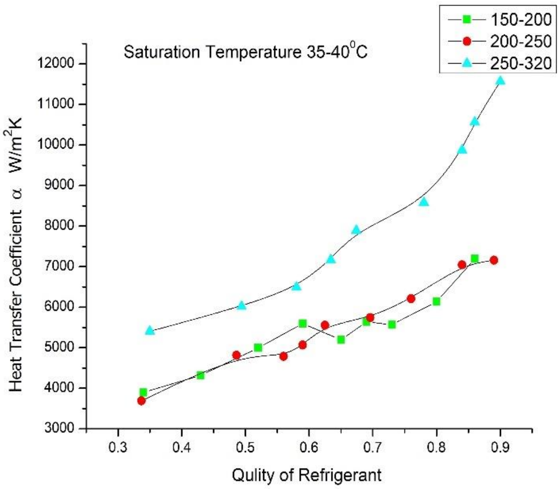

Figure.3. Refrigerant performance on condensation HTC for 35-40^0 C

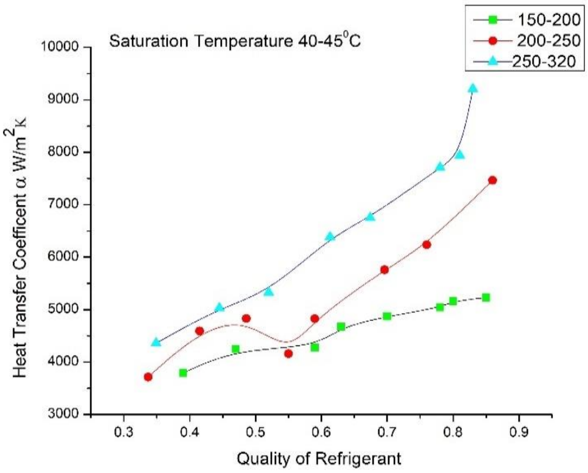

Figure.4. Refrigerant performance on condensation HTC for 40-45^0 C.

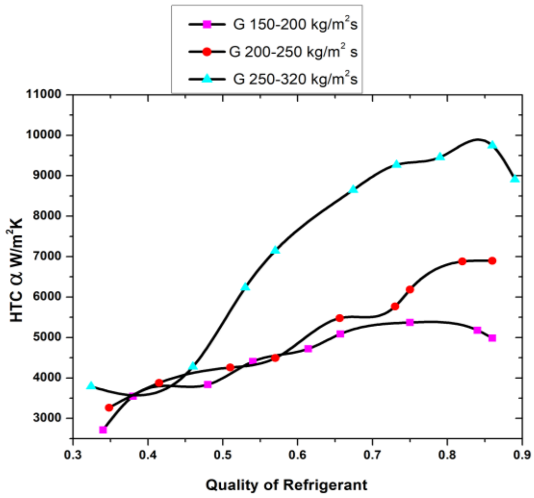

Figure.4. Refrigerant performance on condensation HTC for 45-50^0 C.

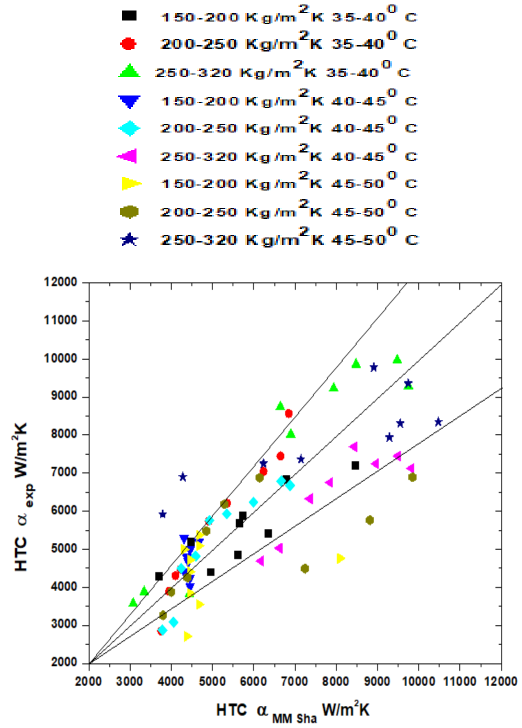

Fig.7. HTC of R134a refrigerant by Experimental vs. MM Sha values.

6. HTC by MM Sha's correlation.

$$P_r = \frac{\mu_l * c_{pl}}{c_{pk}} \tag{12}$$

$$Z = \left(\frac{1}{x} - 1\right)^{0.8} p_r^{\ 0.4} \tag{13}$$

$$\alpha_{lo} = \frac{0.023\,(R_{elo})^{0.8} P_{rl}^{\ 04}\, K_l}{d_i} \tag{14}$$

$$\alpha_l = \alpha_{lo}\left[1+\frac{3.8}{Z^{0.95}}\right]\left(\frac{\mu_l}{14*\mu_l}\right)^{(0.0058+0.557*p_r)} \quad (15)$$

$$\alpha_{nu} = 1.32 * {R_{e_{lo}}}^{\frac{-1}{3}}\left[\frac{\rho_l(\rho_l-\rho_g)*g*{k_l}^3}{\mu_l}\right] \quad (16)$$

Jg is a dimension less velocity factor defined as

$$J_g = \frac{X*G}{(g*d_i*\rho_{g*}(\rho_l-\rho_g))^{0.5}} \quad (17)$$

MM Shah et.al [23] defined three regimes namely I, II, and III, and regime II is not defined in the same literature.

J_g for regime I is defined as

$$J_{g1} = 0.9*(gZ+0.263))^{-0.6} \quad (18)$$

J_{g2} for regime II is defined as

$$J_{g1} = 0.95*(1.254+2.272*Z^{1.249}) \quad (19)$$

MM Sha's HTC calculated asper following condition

If Jg =Jg1 then regime is I regime

$$\therefore \ \alpha_{MMS} = \alpha_l \quad (20)$$

If J_g =J_{g1} then regime is II regime

$$\therefore \ \alpha_{MMS} = \alpha_l + \alpha_{nu} \quad (21)$$

$$MAD = \frac{1}{M}\sum_{i=1}^{i=M}\frac{|\alpha_{exp}-\alpha_{MM\,S}|}{\alpha_{exp}} \quad (22)$$

The HTC of refrigerant is calculated depends on the regime that occur during condensation process. And equations from (12) to (21) solved by using MAT Lab and Refprop software's. And HTC for MM Shas correlations were recorded for different saturation temperature, mass fluxes and vapor qualities.

The mean absolute deviation for experimental HTC and MM Sha's correlations is calculated by using eqn (22). Results and Discussion

The condensation HTC of R134a in smooth tube at different operating conditions are reported. In this section, the effect of mass flux, vapor quality, saturation temperature and refrigerant performance on condensation HTC are reported.

Fig.3 shows experimental HTC verses different vapor quality of R134a. The effect of mass flux vapor qulity and saturation temperature on condensation of R134 are clariffied. Fig.3 shows effect of mass flow rate at different range from 150-330 kg/m^{2}s on heat transfer coefficient at low saturation temperature 35-40^{0}C. Fig.3 depicts the HTC increses with incresing mass flux up to 0.8 vapor qulti and its strts to decline.

Similarly Fig.4 shows experimental HTC verses different vapor quality of R134a at 40-45^{0}C. In Fig.4 it clearly shows that the HTC is incresing with increse of mass flow rate at sam vapor qulity. This is because mainly as mass flus increases turbulence exist in flow inside the smooth tube.

Mainly, as mass velocity increases in the tube, the vapor speed and thin liquid film raises, and hence more turbulence exits in the flow. The HTC of R134a is increased nearly by 1.4 times at all vapor qualities for mass flux ranging from 150-320 kg/m^{2}s show higher HTC.

7.Validation of results

In **Fig.6** experimental results were validated with M.M Shah's correlation and reported.

It was reported that the experimental HTC and MM Sha's HTC values shows 20 % AMD at wide range of operating conditions.

The experimental HTC values are falls about 82% data point within the error band of ±20%.

7. Conclusions

The following conclusions can be drawn from the present study:

1) The results obtained from this experimental set up is in very good agreement with those obtained from mm Shas's corelation results.

2) This is an unique in tube condensation test rig which can be extended to study system performance of refrigerants in an VCRS cycle .

3) Using this experimental set up any refrigerant can be analysed for in tube condensation perfrormance with very small modifications and suitable correlation can be obtaied which is a key element in design of condenser.

4) This experimental set up can also be extended to perform flow boiling analysis of refrigerants with suitable modification of evaporator.

Referrance

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