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Experimental Investigation of InTube Condensation of HFO-1234yf

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Abstract

An increase in global warming potential and other environmental concerns are demanding new environmentally friendly refrigerants. For effective performance of refrigeration and air conditioning system condenser design play a vital role. Experiments were conducted to study the condensation of R1234yf refrigerant inside a copper tube of 8.4 mm in diameter and 750 mm in length. The heat transfer coefficients of refrigerant were calculated against mass flux varying from 150–300 kg/m^2 s, quality of refrigerant 0.3 to 0.8, and saturated temperature 30 and 500 C. The experimental heat transfer coefficients were compared to the heat transfer coefficients by the recent MM Sha correlation. The experimental results are in good agreement with an absolute mean deviation of nearly 20% with the MM Sha heat transfer coefficient.

Keywords: *In-tube condensation, Heat Transfer Coefficient (HTC), R1234yf.*

Greek Symbols

α	:	Heat transfer coefficient of refrigerant (W/m^2 K)
μ	:	Dynamic viscosity (Ns/m^2)
Δ	:	Difference
λ	:	Thermal conductivity (W/m.K)

1.0 Introduction

Due to the widespread use of CFCs in refrigeration and air conditioning systems around the world, environmental hazards such as global warming and ozone depletion were incurred and observed during the 1990s. As a result, the Kigali agreement proposed an amendment to the Montreal Protocol in 1987 to phase out CFCs. In 1997, the goal was to reduce greenhouse gas emissions (GHG). phasing out HFCs as an amendment to the Kyoto Protocol [1].

Extensive research and development activities were conducted to find alternative refrigerants to fill the gap in refrigeration and air conditioning systems. These efforts resulted in the creation of the R134a refrigerant.

R134a is a non-ozone-depleting refrigerant. However, R134a has a global warming potential (GWP) of 1430, it must be replaced soon with a more environmentally friendly refrigerant.

Somchai Wongwises et al. [6], conducted an experiment for a comparative study of R134a with R-290, R-600, and R600a. The following proportions, 50%/40%/10%, 70%/25%/50%, and 70%/25%/50%. For various parameters, the refrigeration COP and cooling capacities were calculated. When compared to other proportions, including pure R-134a,

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the 70/25/50 proportion has the best cooling capacity.

G. Arslan et al. [17], carried out an experiment to determine the heat transfer coefficient (HTC) of R134a inside a vertical smooth tube-in-tube heat exchanger of 0.5 mm diameter for various mass flow rates and quality in the condenser test section.

The energy balance equation was used to calculate HTC. They proposed an equation for condensation HTC after analyzing the experimental results. The HTC of the refrigerant increases with increasing vapour quality and decreases with increasing saturation temperature, according to their findings.

Andrea Diani et al. [22], Anand Kumar Solanki et al. [23], and SHAO Li et al.[7], and were conducted many experimental investigations to find condensation HTC in the condenser. They were almost all used as pre and post-confessors with a pump to circulate the refrigerant. They also used an auxiliary condenser.

Ki Jung et. al [13], conducted an experiment for a comparative study on R123a and R1234yf in plain, fin, and Turbo C tubes. They found that the R1234yf has the capacity to replace the R134.

R134a is hazardous to the environment because it has a global warming potential of 1340. Minor and Spatz [8], proposed R1234yf as an alternative to R134a in 2008, which has a global warming potential of less than 1, whereas R134a has a global warming potential of 1340.

Pamela Reasor et al [2], conducted a comparison study of R1234yf and R134a performance. They simulated condensation and evaporation by taking into account refrigerant inlet temperature, quality, and mass flow rate. They discovered that heat load capacity is within 2.4% and the difference in outlet temperature is only 2 K. Furthermore, the results of their investigation revealed that R1234yf has close thermal values and the best drop-in replacement potential of R134a.

L. Liebenberg et.al [9], reported recent research on the flow patterns of different refrigerants during condensation. They are convinced that the flow pattern has an effect on the heat transfer properties of refrigerants during condensation.

In a horizontal tube that has been consistently heated, experimental research on the flow boiling of the pure refrigerants R134a and R123 and their blends was conducted by Lim T et al [5]. Through tubular sight glasses with an internal diameter of 10 mm placed at the test section's inlet and outflow, the flow pattern was studied. At a pressure of 0.6 MPa, tests were conducted with heat flux ranges of 5–50 kW/m^2, vapour quality ranges of 0–100%, and mass velocities of 150–600 $kg/m^2/s$.

Samaneh Daviran et al. [12], conduct a similar comparative study in a microchannel condenser with a constant cooling capacity of 3.5 kW and refrigerant mass flow of 113 kg hr^{-1}. The constant cooling capacity of HFO-R1234yf HTC, according to them, is very close to that of HFC R134a, and the COP of R1234yf is 18% higher than that of HFC R134a.

J. Navarro-Esbri et al. [14], performed an experimental study for R1234yf and R134a in a VCR system, taking into account many variables, and they discovered that the COP of R1234yf is 8 to 9% lower than that of R134a.

Jignesh K et al. [21], examined the performance of R134a and R1234yf in an automobile test rig to compare the performance of R134a and R1234yf in an automobile test rig. According to the data from the literature for hydrodynamic analysis, the COP of 1234yf is 6.5 % lower than that of R134a, however, R1234yf can be used as an alternative substitute for R134a.

The automotive condenser was numerically simulated by Gurudatt et al.[15]. They employed Shah's correlation to forecast the refrigerant HTC in two phases, and their findings reveal that the better the vapour quality is, the higher the heat transfer coefficient will be, especially around 0.8 dryness fraction [5].

Table 1: Comparison of R134a and R1234yf refrigerants Properties [8]

Thermodynamic Property	R-134a	R-1234yf
Chemical Name	Tetrafluoro-Pthylene	Tetrafluoro-Propene,
Chemical formula	CF3CH2F	C3H2F4
Boiling point at 1 atm	-26°C	-29°C
Critical temperature	101.1°C	95°C
Critical pressure	40.6 bar	33.82 bar
Latent heat kJ/kg	209.5 kJ/kg	180.25 kJ/kg
Temperature glide	0	0
Ozone depletion potential	0	0
Flammability rating	A2L Refrigerant - Mildy Flammable	A1Non-Flammable
Global warming potential	1340	Less than 1

MM Sha [10] has been conducting research for the past three decades to develop exact correlations for two-phase flow condensation in various conditions. In 1979, MM Sha proposed a new correlation that was widely accepted at the time. Later, some limitations for low mass flow rate and moderate pressure were discovered.

MM Shah [25] developed an improved correlation with good agreement with 22 different refrigerants in 2009. The impra oved correlation identified three flow regimes, I, II, and III, which are turbulent, laminar, and mixed, respectively.

Thome et al. [4], Garimella et al. [3], Dalkilic et al. [11], Kim and Mudawar et al. [15], Awad et al. [16], and Del Col et al. [18] examined and compared the MM Sha correlation extensively. The results showed a mean deviation of 22%.

In 2016 MM Sha [25] developed two alternative generalized correlations for the condensation of refrigerants by introducing the effect of vapour and liquid phase surface tension, as well as Weber's numbers, during condensation. Furthermore, modified correlations were reported in [24], and the same correlations were used to validate the current experimental HTC.

HFOs condensation inside microfin tubes with small diameters is still unexplored. Numerous experimental studies on in-tube forced condensation of refrigerants have been conducted in the literature, with almost identical experimental setups involving a dual refrigeration system [6].

HFOs condensation inside tubes is still unexplored. This paper presents the first-order results of R1234yf refrigerant in-tube condensation and a comparison with the MM Sha correlation. And also expensive refrigerant pumps and other instruments are used in previous literature and experimental setups. This research focuses on an innovative in-tube condensation test rig.

2.0 Experimental Test Rig Description

It is made up of an evaporator compressor, condenser filter, drier, and a thermostatic expansion valve (TXV). The refrigerant from the evaporator exit enters the compressor as saturated or super-heated vapour and is compressed to high temperature and high-pressure superheated vapour. The first super-heated vapour refrigerant enters the post-condenser, where it is quality controlled before being fed into the actual condenser test section.

The refrigerant has been condensed into a nearly liquid state. Between the post condenser and the differential pressure gauge, a filter cum drier was installed. The filter cum drier cleans the refrigerant loop of any moisture or foreign particles.

The refrigerant is then a sub-cooled liquid that is maintained by a filter drier. This liquid refrigerant is then isoenthalpically throttled to produce a low-temperature, low-pressure liquid-vapour mixture. The mixture then enters the evaporator, and the cycle begins again.

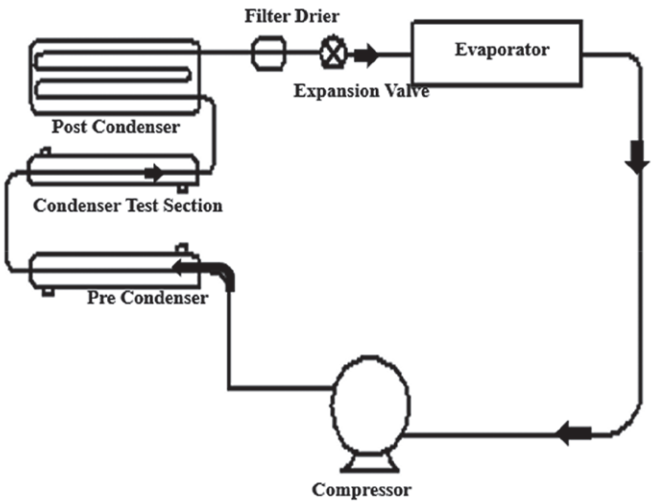

Figure 1: Line diagram of the in-tube condensation test rig

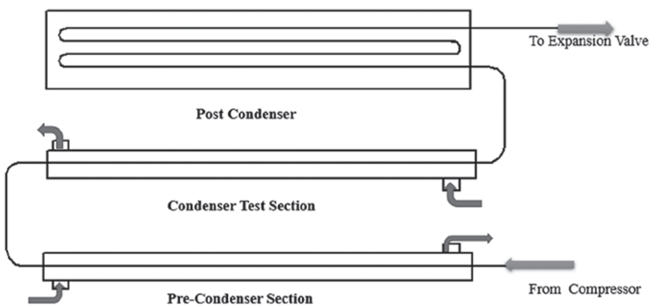

Figure 2: Test section, showing pre and post-condensers

Conduction of experiment

Experiments for various saturation temperatures are carried out by varying the coolant rate and compressor speed. The coolant mass flow rate of the pre-condenser was varied to control the quality of the refrigerant to the test section. As the temperature and pressure of the refrigerant changed, they were measured. The temperature was measured with precision using K-type thermocouples (0.05 per cent).

The plane copper tube used with an internal diameter of 8.4 mm, an outer diameter of 9.1 mm, and a length of 700 mm. The refrigerant is then a sub-cooled liquid that is maintained by a filter drier. This liquid refrigerant is then isoenthalpically throttled to produce a low-temperature, low-pressure liquid-vapour mixture. The mixture then enters the evaporator, and the cycle begins again.

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The plane copper tube used with an internal diameter of 8.4 mm, an outer diameter of 9.1 mm, and a length of 700 mm. The refrigerant properties were recorded using REFPROP software [24], and equations were solved by using MAT Lab software to determine the HTC of R1234yf refrigerant in the test section.

Data reduction

The mass flow rate of the refrigerant is measured by using Haaland equation [25] as follows.

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

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

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

$$m_r = \rho_l * A_P * V_l \quad \text{... (4)}$$

The quality of the refrigerant vapour at the pre-condenser section inlet is determined by the heat transfer rate in the pre-condenser.

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

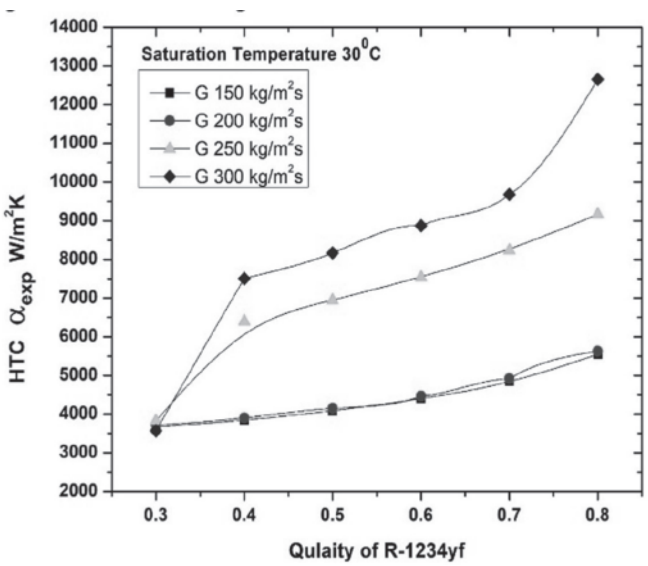

Figure 3: Refrigerant performance on condensation HTC for 30°C

The heat carried by water is the same the heat lost by the refrigerant in the pre-condenser section.

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

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

$$(h_{r2})_{PC} = (h_{r1})_{PC} - (Q_r)_{PC}/m_r \quad \text{... (7)}$$

The vapour quality of refrigerant is calculated as

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

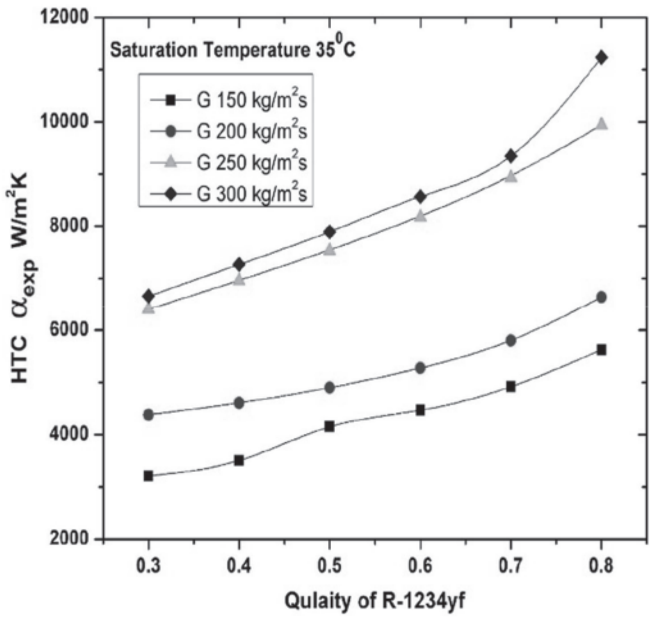

Figure 4: Refrigerant performance on condensation HTC for 35°C

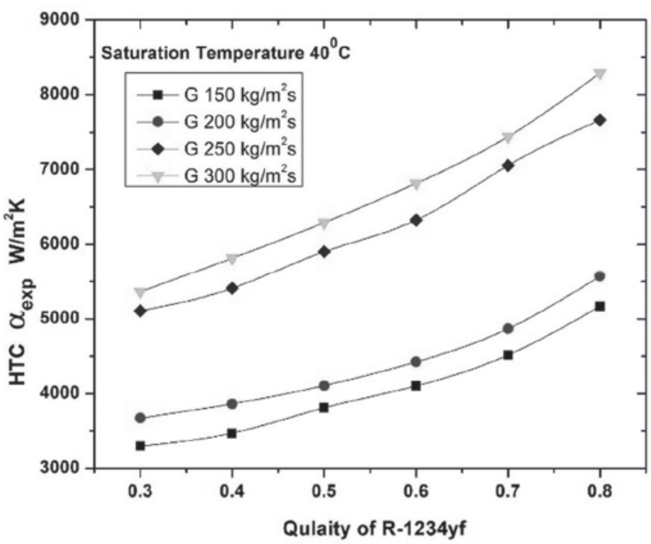

Figure 5: Refrigerant performance on condensation HTC for 40°C

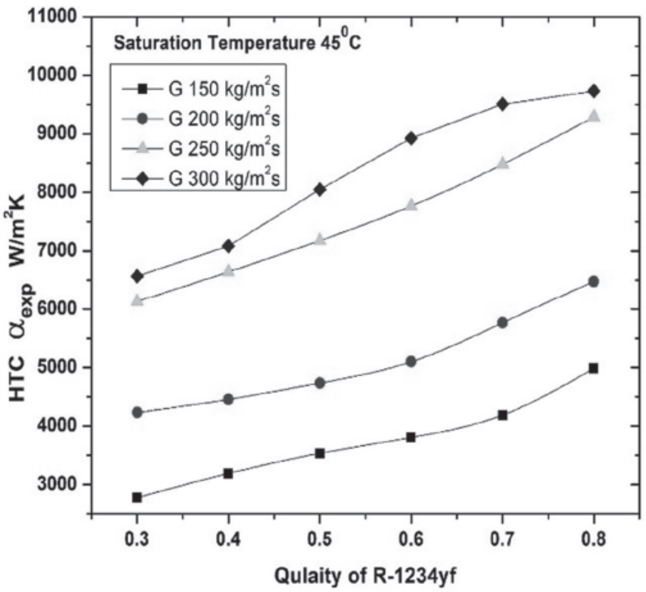

Figure 6: Refrigerant performance on condensation HTC for 45ºC

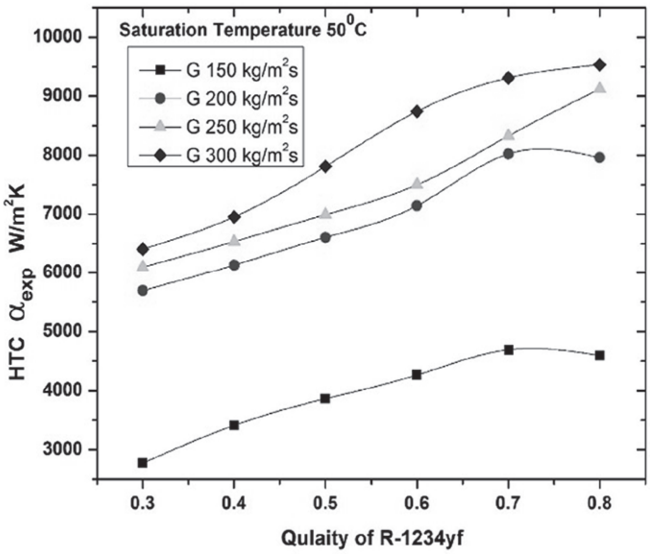

Figure 7: Refrigerant performance on condensation HTC for 50ºC

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

$$X_{avg} = \frac{X_1+X_2}{2} \qquad \text{... (9)}$$

The HTC of the refrigerant is calculating as in Figures 3 to 7.

HTC by MM Sha's correlation [18]

$$p_r = \frac{\mu_l * c_{pl}}{K_l} \qquad \text{... (10)}$$

$$Z = \left(\frac{1}{X} - 1\right)^{0.8} p_r{}^{0.4} \qquad \text{... (11)}$$

$$\alpha_{lo} = \frac{0.023\,(R_{elO})^{0.8} P_{rl}{}^{04}\, K_l}{d_i} \qquad \text{... (12)}$$

$$\alpha_l = \alpha_{lo}\left[1 + \frac{3.8}{Z^{0.95}}\right]\left(\frac{\mu_l}{14*\mu_l}\right)^{(0.0058+0.557*p_r)} \qquad \text{... (13)}$$

$$\alpha_{nu} = 1.32 * R_{e_{lo}}{}^{\frac{-1}{3}}\left[\frac{\rho_l(\rho_l-\rho_g)*g*k_l{}^3}{\mu_l}\right] \qquad \text{... (14)}$$

Jg is a dimension less velocity factor defined as

$$J_g = \frac{X*G}{(g*d_i*\rho_{g*}(\rho_l-\rho_g))^{0.5}} \qquad \text{... (15)}$$

MM Shah et.al [14] defined three regimes I, II, and III. *Jg* for regime I is defined as J_{g1}

$$J_{g1} = 0.9 * (gZ + 0.263))^{-0.6} \qquad \text{... (16)}$$

Jg for regime II is defined as J_{g2}

$$J_{g2} = 0.95 * (1.254 + 2.272 * Z^{1.249}) \qquad \text{... (17)}$$

MM Sha's HTC calculated asper following condition

If $Jg = J_{g1}$ then regime is I regime

$$\alpha_{MMS} = \alpha_l \qquad \text{... (18)}$$

If $Jg = J_{g2}$ then regime is II regime

$$\alpha_{MMS} = \alpha_l + \alpha_{nu} \qquad \text{... (19)}$$

$$AMD = \frac{1}{M}\sum_{i=1}^{i=M}\frac{|\alpha_{exp}-\alpha_{MMS}|}{\alpha_{exp}} \qquad \text{... (20)}$$

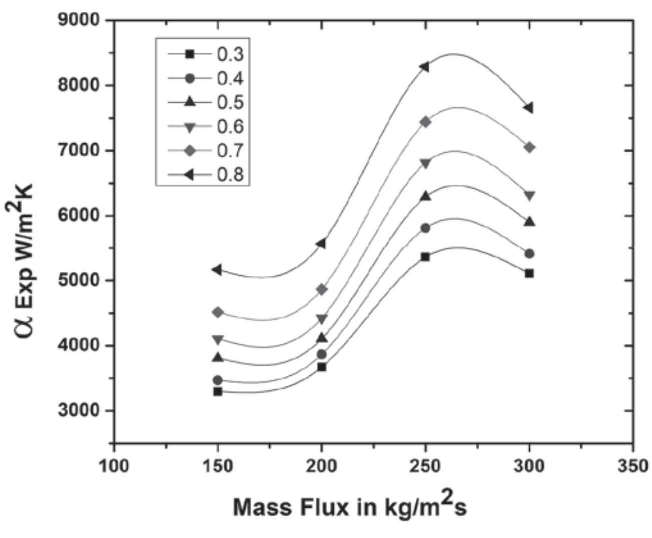

Figure 8: HTC of R1234yf refrigerant by Experimental vs. MM Sha values

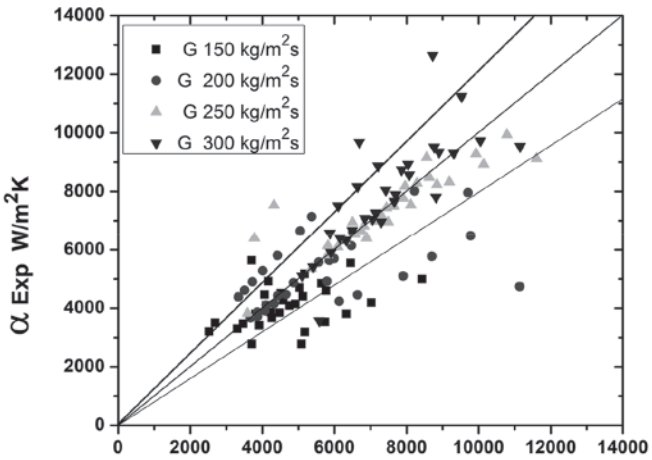

Figure 9: HTC of R1234yf refrigerant by Experimental vs. MM Sha values

The HTC of refrigerant is calculated as per the regime condition that occur during condensation. And equations from 12 to 17 solved by using MAT Lab and Refprop software's. And HTC using MM Shas correlations were recorded for different saturation temperature, mass fluxes and vapour qualities.

The mean absolute deviation for experimental HTC and MM Sha's correlations is calculated by using equation (20).

3.0 Results and Discusssions

It has been seen from the Figs. 1, 2, 3 and 4 as mass flux increases for a fixed condensation temperature the two phase heat transfer coefficient also increases and it will be maximum at around 0.8 dryness fraction. The HTC is almost praotional to the mass flow rate and decreseses after 250 kg/m^{2}s

The absolute mean deviation of experimental values is nearly 20% percentage shown in Fig.9 and also it shows almost 76 points fall in the bandwidth of 20% out of 98.

4.0 Conclusions

The following conclusions were drawn based on the experimental work conducted in this study.

- As the mass flow rate increases the HTC increases and also HTC is higher at the higher vapour quality of R1234yf.
- Experimental results are in good agreement with MM Sha correlation.
- HTC value increases with increase in mass flow rate and tend to decrease for further increasing in mass flow rate. Future work

A correlation need to be developed with high accuracy for predicting the HTC for different operating conditions.

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