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Research Article

Determination of Condensation Heat Transfer Coefficient inside a Horizontal Pipe at High Pressure using Experimental Analysis

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Abstract

It is observed that simultaneous flow of liquid and vapor result in different flow regimes due to wetting characteristics of the liquid and interfacial shear and momentum exchange between the liquid and vapor based on the counteracting forces of gravity and vapor shear. In addition, flow regimes are also affected by channel cross section, orientation and based on whether the flow is external or internal flow. Two phase flow in a horizontal tube has widespread applications, particularly in the condensers and evaporators of refrigeration and air conditioning systems. The paper discusses the experimental analysis to evaluate the two phase heat transfer coefficient and pressure drop and an attempt is made to compare the experimental results with the existing correlations.

Keywords: *Two phase flow, High pressure, Flow regime, Heat transfer coefficient, Pressure drop, Experimental analysis*

1. Introduction

The pioneering work of Lockhart and Martinelli (R. Lockhart *et al*, 1947) forms the basis for modeling of two phase flow. They developed two phase multiplier approach to predict the frictional pressure drop of two phase flows using the separated flow model. Later, many correlations were developed for two phase multiplier, based on the experimental data of air-water mixtures, gas – oil mixtures etc at near atmospheric pressures and refrigerants for evaporating flows. Some of these correlations, viz., Lockhart and Martinelli correlation deviate by more than 100% in predicting the pressure drop of condensing flows or in general, two phase flows at high pressures. Yet, Lockhart and Martinelli correlation is widely used in the analytical modeling of condensing flows, viz., by Sarma *et al.* (P.K. Sarma *et al*, 2002), Li *et al,* (Li *et al*, 2000) etc.,

The quasi local experimental work reported in the literature is primarily to study the performance of alternative refrigerants. Shao *et al*, (Shao *et al*, 1998), Boissieux *et al*. (Boissieux *et al*, 2000), Smit*et al*. (Smit *et al*, 2002, Smit *et al*, 2002), Infante Ferreira *et al*. (C.A. Infante Ferreira *et al*, 2003) experimentally studied the performance of refrigerant mixtures to test their applicability as alternative refrigerants. Lee *et al*. (H.S. Lee *et al*, 2006) reported the performance of hydrocarbons for condensing flows.

Recent experimental work reported focuses on the development of better predictive procedures for thermal design of condensers. Aprea*et al*. (C. Aprea *et al*, 2003) studied the performance of refrigerants at low mass flux and reported that his experimental data was in good agreement with Dobson *et al*. (M.K. Dobson *et al*, 1998) correlation at low mass flux. Cavallini *et al*. (A. Cavallini *et al*, 2002) and Thome *et al*. (J. El Hajal *et al*, 2003, J. El Hajal *et al*, 2003) developed flow regime based correlations using large database of refrigerants reported in the literature. Though, these flow regime based correlations predicted the experimental data of halogenated refrigerants well, exhibited larger deviations at high reduced pressures as reported by Jiang *et al*. (Y. Jiang *et al*, 2006). Park *et al*. (C.Y. Park *et al*, 2008) experimentally studied the performance of ammonia and mentioned that these correlations deviate from his experimental data by more than 300%. This shows the semi empirical nature of these correlations and their limited applications.

Most of the analytical correlations developed for heat transfer were based on the assumption of annular flow regime. Some of the correlations, viz., Shah (M.M. Shah, 2009), Dobson *et al*. and Traviss *et al*. (D.P. Traviss, 1973), predict the heat transfer coefficient of pure refrigerants very well, but exhibit larger deviations for refrigerant mixtures.

The objective of the present study is primarily to design and fabricate an experimental setup for measuring the performance parameters of two phase flow, viz., heat transfer and pressure drop for three different refrigerants, R22, R134a and R407C using quasi local experimentation technique, where heat

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transfer information is obtained with associated small changes in vapor quality and is reported for the average vapor quality. Thus the heat transfer coefficients obtained are quasi local values for the average vapor quality of test section to evaluate two phase heat transfer coefficient and pressure drop of refrigerants, R22, R134a and R407C condensing inside a horizontal tube at an average saturation temperature of 40^0C. The experimental setup is validated using the experimental data of R22 reported in the literature.

2. Experimental Analysis

The primary focus in the present study is to allow the condensation of refrigerant inside the test section with a small decrement of vapor quality, (Δx), typically in the range of 0.13 to 0.33. The experimental heat transfer coefficient and pressure drop are reported for the average of inlet and outlet quality for the test section and hence are referred as quasi local values. Since the small quality decrements are considered, the test section heat duties are small.

Fig.1 Experimental Setup of Skid mount type

The test facility is skid mount type with all the components housed on MS frame as shown in Fig.1. All the components are mounted on the skid using vibration pads. The loft type water tanks are mounted on a separate MS structure. The set up basically consists of two main subsystems, viz., one refrigerant loop and three water loops.

2.1 Refrigerant Loop

The main components of refrigerant loop are shown in Fig. 2. Except test section which is concentric tube heat exchanger, all other heat exchangers are brazed plate type made of stainless steel.

The salient feature of the test facility is that it uses a separate compressor, expansion valve and receiver for each of the refrigerant. Three compressors for three different refrigerants are arranged in parallel circuit so that any one of the compressor is used at a given time. The compressors are properly isolated from each other with shutoff valves. Similarly three receivers and three expansion valves are arranged in parallel corresponding to each compressor with isolation from each other using shutoff valves. With this type of arrangement, only heat exchangers and refrigerant lines that are in common use are to be evacuated whenever a new refrigerant is charged, making the charging process simple compared to evacuating the compressor every time the refrigerant is changed. The presence of receiver in the refrigerant loop will cushion the fluctuations in the flow. The thermostatic expansion valve with external equalizer is used to make the entire evaporator surface available for heat transfer in case of large pressure drop in evaporator.

The refrigerant cycle is shown in Fig.2. The compressor exit is connected to the pre condenser where the refrigerant is desuperheated and also partly condenses depending upon the required inlet quality at the entrance of test section. A small quality decrement takes place in the test section based on the cooling load supplied by varying cooling water temperature or flow rate or both. Required amount of refrigerant can be supplied to the test section by allowing part of the flow through bypass valve which condenses in the bypass condenser for complete condensation as shown in Fig.2.

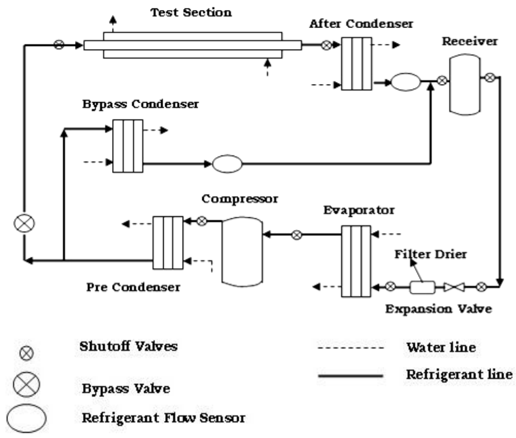


Fig.2 Schematic of Refrigerant Loop

The refrigerant from bypass condenser and after condenser mixes in the receiver as shown in Fig.2. The refrigerant then enters the expansion valve where the pressure is reduced. A filter drier following the expansion valve removes any presence of moisture and impurities. The low pressure refrigerant enters the evaporator and gets vaporized. The heat load to the evaporator is provided by a separate water loop. The low pressure vapor enters the compressor suction, thus closed refrigerant cycle is established.

2.2 Water Loop

Three water loops are designed; two on the condensing side and one on evaporating side. On the condensing side, one water tank is exclusively used for the test section to control the test section cooling load and another one is for the pre, after and bypass condensers together. Each water loop is connected to 1000 litre capacity loft type insulated water tanks Chillers and heaters are provided in each tank to maintain required water temperatures according to the experimental conditions. The water is circulated using four pumps, each of capacity, 720litre/hr for four heat exchangers on the condensing side, viz., test section, pre, after and bypass condensers. Magnetic flow sensors connected to the data acquisition system measures the water flow rate of test section. For other condensers, water flow rate is measured using analog magnetic flow meters.

2.3 Test Section

The test section is a concentric tube counter flow heat exchanger made of hard drawn copper with 3/8" diameter inner tube and 7/8" diameter outer tube with 1200 mm in length.

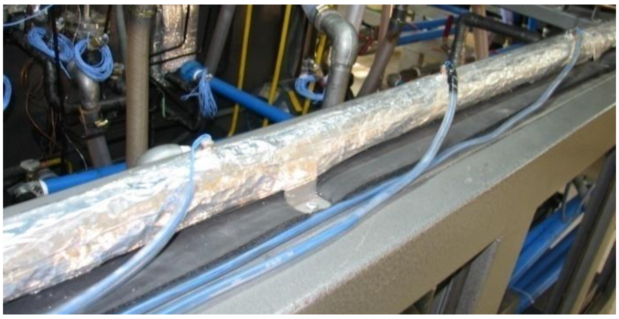


Fig.3 Test Section of Insulated Horizontal Pipe

The insulated test section is shown in Fig.3. The test section is insulated using 10 mm thick Armoflex insulation along with aluminum cladding to minimize heat losses. The inner tube is located concentrically by using brass end plates.

2.4 Instrumentation and Data Acquisition

Four T type thermocouple sensors with thermowells are provided at the inlet and outlet of each of the five heat exchangers in the refrigerant and water circuits. The pressure drop across the test section is measured using differential pressure sensor. Three temperature sensors with thermowells are located each at the exit of three expansion valves. Three pressure sensors are provided at the inlet of evaporator, after condenser and bypass condenser.

For test section, the water and refrigerant flow rates are measured using digital magnetic flow sensors. The refrigerant flow rate through the bypass condenser is also measured using digital magnetic flow sensor. The total refrigerant mass flow rate passing through the system is sum of the refrigerant flow rates obtained from the magnetic flow sensors of test section and bypass condenser. All these 32 sensors are connected to two 16-channel analog modules using the computerized POLSOFT data acquisition system.

Once the test facility is ready, the next step involves establishing the operating conditions viz., required water temperature at the inlet of test section, evaporator and other condensers. Then the water flow rate and temperature are adjusted in the pre condenser to get the set vapor quality at the inlet of test section. The required vapor quality decrements are achieved in the test section by controlling the water flow rate and the inlet water temperature. After achieving the required operating conditions, the system takes less than an hour to reach steady state conditions.

Once the steady conditions are ensured from the logged data and from the temperature scanner, mass flow rate of water for the pre, bypass and after condensers; and for the evaporator is noted from the analog flow meters. Only those readings are taken for which the ratio of difference between the refrigerant side heat loss/gain and the coolant side heat gain/loss to the coolant side heat gain/loss is less than ±1%.

3. Data Analysis

The observations from the data acquisition unit are transferred to excel sheets and the calculations are performed. The first step involved in the data analysis is evaluation of average or mean vapor quality of test section. The second step is evaluation of two phase or sectional or local heat transfer coefficient from the energy balance of test section. The pressure drop is directly measured across the test section using differential pressure transmitter.

3.1 Average Quality of Test Section

Ignoring the axial conduction in the heat exchangers, the inlet and outlet vapor quality of test section is obtained by applying First Law of Thermodynamics to the pre condenser and test section.

Test Section Inlet Quality: The mass flow rate of the refrigerant passing through the pre condenser is sum of the mass flow rates through test section and bypass condenser. The heat lost by the refrigerant in the pre condenser is equal to the heat gained by the cooling water in the absence of heat losses as the heat exchanger is well insulated. Thus heat duty of pre condenser is obtained from the water side heat transfer, given by Eq. (5.1a).

$$Q_{PC,w} = m_{PC,w} C_{pw} (T_{PC,wo} - T_{PC,win}) = m_{PC,r} (i_{PC,rin} - i_{PC,ro}) \tag{1}$$

$$i_{PC,ro} = i_{TS,rin} = i_{PC,rin} - \frac{Q_{PC,w}}{m_{PC,r}} \tag{1a}$$

$i_{PC,rin}$, the enthalpy at the inlet of pre condenser is obtained from the pressure and temperature readings of the refrigerant using REFPROP, refrigerant property database. From Eq. (1a), the enthalpy of refrigerant leaving the pre condenser and entering the test section is obtained and hence the vapor quality at the inlet to the test section using Eq. (2).

$$x_{TS,rin} = \frac{i_{TS,rin} - i_{l,TS,rin}}{i_{lv,TS,rin}} \tag{2}$$

where, i_{lv} and i_l are latent heat and saturated liquid enthalpy respectively at the saturation temperature corresponding to test section inlet pressure.

Test Section Outlet Quality: Similarly, the test section outlet quality is obtained from the energy balance of test section given by Eq. (3). The refrigerant entering the test section is part of the total refrigerant leaving the pre condenser based on the flow rate requirement of the experimental conditions imposed. The remaining refrigerant will pass through the bypass condenser for complete condensation.

$$Q_{TS,w} = m_{TS,w} C_{pw} (T_{TS,wo} - T_{TS,win}) = m_{TS,r} (i_{TS,rin} - i_{TS,ro}) \tag{3}$$

$$i_{TS,ro} = i_{TS,rin} - \frac{Q_{TS,w}}{m_{TS,r}} \tag{3a}$$

From Eq. (3) and by using saturation liquid enthalpy and latent heat at saturation temperature corresponding to the test section exit pressure, the outlet vapor quality of test section is obtained.

$$x_{TS,ro} = \frac{i_{TS,ro} - i_{l,TS,ro}}{i_{lv,TS,ro}} \tag{4}$$

Using Eqs. (1) and (2), average vapor quality for the test section is calculated.

$$x = \frac{x_{TS,rin} + x_{TS,ro}}{2} \tag{5}$$

decrement, Δx for the test section is obtained. The average saturation temperature of the test section is the saturation temperature corresponding to the average of the inlet and outlet pressures of the test section. The average saturation temperature and corresponding latent heat of the refrigerant are obtained from REFPROP. Knowing the quality decrement for the test section, heat lost by refrigerant is calculated using Eq. (6a). The average heat duty of the test section is calculated using Eq. (6b) and the energy balance error for the test section can be calculated using Eq. (6c).

$$Q_{TS,r} = m_{TS,r} i_{lv,TS} \left(\Delta x\right) \tag{6a}$$

$$Q_{ave} = \frac{Q_{TS,r} + Q_{TS,w}}{2} \tag{6b}$$

and hence energy balance error (%) =

$$\frac{\left|Q_{TS,ave} - Q_{TS,w}\right|}{Q_{TS,w}} X100 \tag{6c}$$

Only those measurements are considered for which, the energy balance error is less than 1%.

3.2 Local Heat Transfer Coefficient

From the test section heat duty given by Eq. (1), from the average wall temperature measured on the outer surface of test section inner tube using six thermocouples and from the average saturation temperature of test section, the local heat transfer coefficient for the average quality of the test section is obtained using Eq. (7). Owing to high thermal conductivity of copper tube and its negligible thickness, the inner surface temperature of the copper tube is considered to be same as the outer surface temperature.

$$h = \frac{Q_{TS,w}}{\left(\Pi dL\right)\left(\bar{T}_s - \bar{T}_{wall}\right)} \tag{7}$$

Using the three refrigerants, performance parameters of two phase flow are studied in the pressure range of 10 - 16 bar. The experimental uncertainty in the measurement of heat transfer coefficient is found to be ±10.85% and in the measurement of pressure drop is ±0.081 kPa. The experimental data is compared with some of the widely used and recently developed flow regime based correlations.

4. Validation of Test Facility

The test section is initially charged with the refrigerant, R22 and the local heat transfer coefficients are determined at different mass fluxes of 200, 400 and 600 kg/m^{2}s. R22 is used by many researchers for evaluating two phase heat transfer coefficient and its data is well reported in the literature. The recent experimental data reported by Cavallini *et al.* and Liebenberg (L. Liebennerg *et al*, 2005) for a condensing temperature of 40^{0}C is used to validate the test facility. The experimental data reported by Cavallini *et al.* is for a tube of inner diameter, 8 mm and length, 1m and that of Liebenberg is for a tube of OD 9.82 mm and an effective length of 1.5 m. The experimental heat transfer coefficient and pressure drop obtained from the present study is compared with that of Cavallini *et al.* at mass flux of 200 and 400 kg/m^{2}s and with that of Liebenberg at mass flux of 400 kg/m^{2}s. The graphs of comparison are presented in Figs. 4 and 5.

4.1 Comparison of Heat Transfer Coefficient

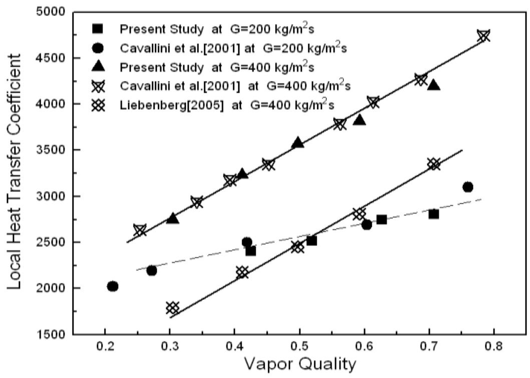


Fig.4 Comparison of Local Heat Transfer Coefficient of Present Study with other Experimental Data from Literature

From Fig. 4, it is observed that the heat transfer coefficient values obtained from the present study are in good agreement with Cavallini *et al*. data at the mass fluxes considered, but the heat transfer coefficients reported by Liebenberg are much lower with a constant deviation from the present study and from Cavallini *et al*. data.

At a mass flux of 200 kg/m^{2}s, heat transfer coefficient does not vary significantly with quality for both the data from the present study and from Cavallini *et al*. A rather insignificant variation of heat transfer coefficient with vapor quality as shown by dotted line in Fig. 4 is representative of stratified or stratified wavy flow regime where the vapor shear forces are negligible and the heat transfer is predominantly gravity driven. The operating conditions of the present study and of other studies considered for comparison are plotted on Thome *et al*. flow regime map for R22 as shown in Fig. 5. The derived flow regime, viz., of stratified wavy from the heat transfer characteristic of two phase flow of R22 at a mass flux of 200 kg/m^{2}s, matches with the predictions of Thome *et al*. flow regime map.

At a mass flux of 400 kg/m^{2}s, the data from present study and from Cavallini *et al*. shows linear variation with vapor quality as represented by solid line in Fig. 4. This behavior is observed with the annular flow regime where the dominant mode of heat transfer is forced convection of liquid film in the axial direction. These observations also match with the predictions of Thome *et al*. map as shown in Fig. 5.

The heat transfer data of Liebenberg plotted in Fig. 4 also shows a linear variation with quality representing the corresponding flow regime as annular. Accordingly, Thome *et al*. map in Fig. 5 shows that Liebenberg data also falls in annular flow regime. In addition, Fig. 4 shows that the slopes of the heat transfer data of present study and Cavallini *et al*. and Liebenberg are parallel. The constant deviation of Liebenberg data from that of present study and Cavallini *et al*. is possibly due to the heat duties and wall temperatures used in his experiment are different from those used in the present study and in Cavallini *et al*. experiments which affect the heat transfer coefficient as given in Eq. (7).

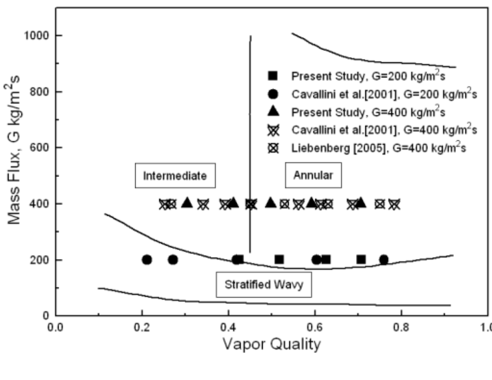


Fig. 5 Thome*et al*. (J. El Hajal*et al*, 2003) Flow Regime Map Showing Flow Regimes for the Present Study

From these observations, it can be concluded that the experimental data of present study is corroborated with the experimental data of Cavallini *et al*. and Liebenberg. The experimental data obtained using test facility could capture the physical phenomena by exhibiting a definite behavior of heat transfer coefficient with vapor quality for mass fluxes, 200 and 400 kg/m^{2}s. In addition, the dominant flow regime derived from the heat transfer characteristics at different mass fluxes matched with the predictions of Thome *et al*. map.

4.2 Comparison of Pressure drop

Fig. 6 shows that the pressure gradient obtained from the present study is in good agreement with Cavallini *et al*. data and also with Liebenberg data at 400 kg/m^{2}s. This is because the pressure drop data is function of mass flux and vapor quality, diameter and properties of fluid which are almost same for experimental studies considered for comparison. As shown in Fig. 6, the minor systematic variations in the pressure drop data taken from different sources of present study, Cavallini *et al*. and Liebenberg can be attributed to the minor differences in the dimensions in terms of length and internal diameter of the test section and experimental conditions. Thus from the comparisons of heat transfer coefficient and pressure drop as shown in Figs. 4 and 6 respectively, it is concluded that the test facility is able to predict the trends of performance parameters with mass flux and vapor quality. Thus the test facility is validated.

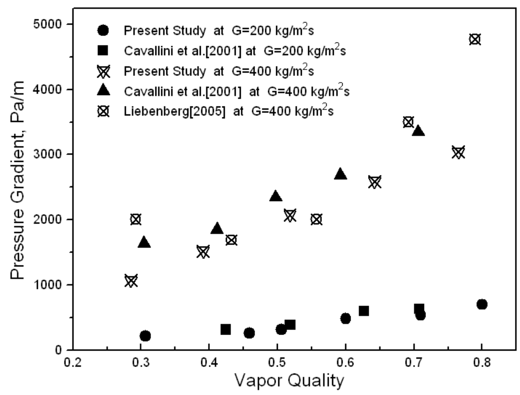


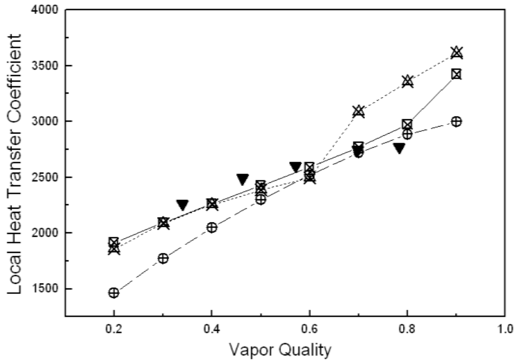
Fig.6 Comparison of Local Pressure Gradient of Present Study with Experimental Data from Literature

5. Experimental Heat Transfer Coefficient with Correlations

Shah (17) correlation, an annular flow correlation based on two phase multiplier approach, Dobson *et al*. correlation with two broadly classified flow regimes of gravity driven and shear based and flow regime based Cavallini *et al*. correlations are used for the comparison of experimental data. Sweeny multiplier (M.K. Dobson *et al*, 1998) is used for Dobson *et al*. correlation for zeotrope, R407C. Shah, Dobson *et al*. and Cavallini *et al*. correlations represent the progressive development in the modeling of condensing flows inside a tube.

5.2 Comparisons for R134a

Fig. 7a shows that at low mass flux, all correlations under predict the experimental data, but only up to medium vapor quality.



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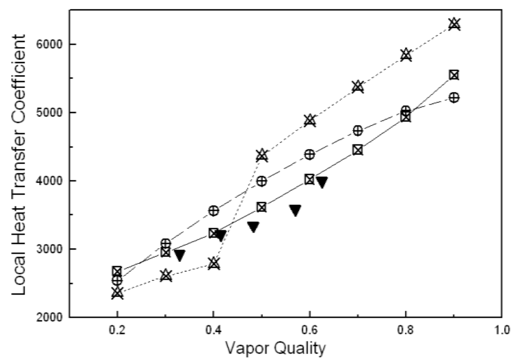

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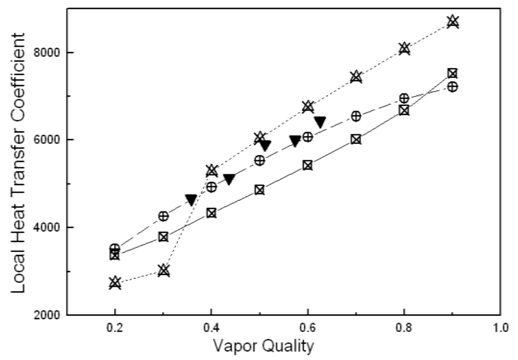

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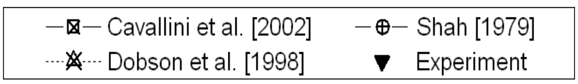
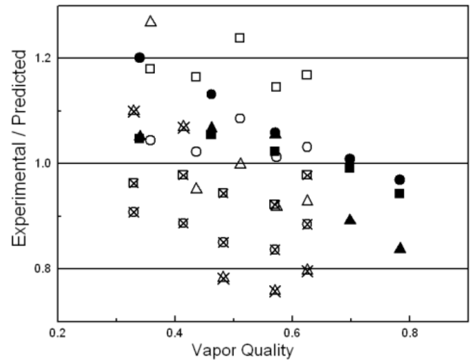


Fig.7 Comparison of Experimental Heat Transfer Coefficient of R134a with Correlations

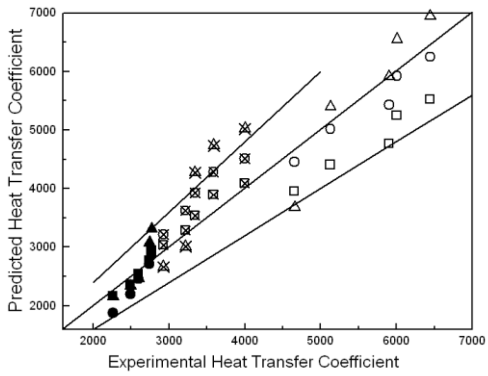
At high vapor quality, the correlations over predict the experimental data. As shown in Fig. 7, the Cavallini *et al*. correlation closely follows the experimental data at low and medium mass fluxes and at high mass flux, the experimental data scatters close to the Shah correlation.

Fig. 8b shows that all the data points fall within ±20% deviation band except the points representing Dobson *et al*. correlation at medium mass flux.

Fig. 8a shows the deviation graph for R134a. Cavallini *et al.* correlation under predicts the experimental data up to medium qualities and then over predicts with a deviation of 5 – 6% for a low mass flux of 200 kg/m^{2}s. For a medium mass flux, it over predicts the experimental data with a deviation of 5% and for high mass flux, it under predicts with a larger deviation of 18% .



(a)



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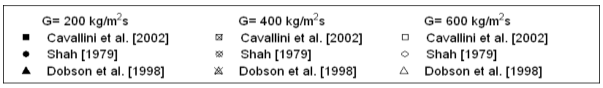


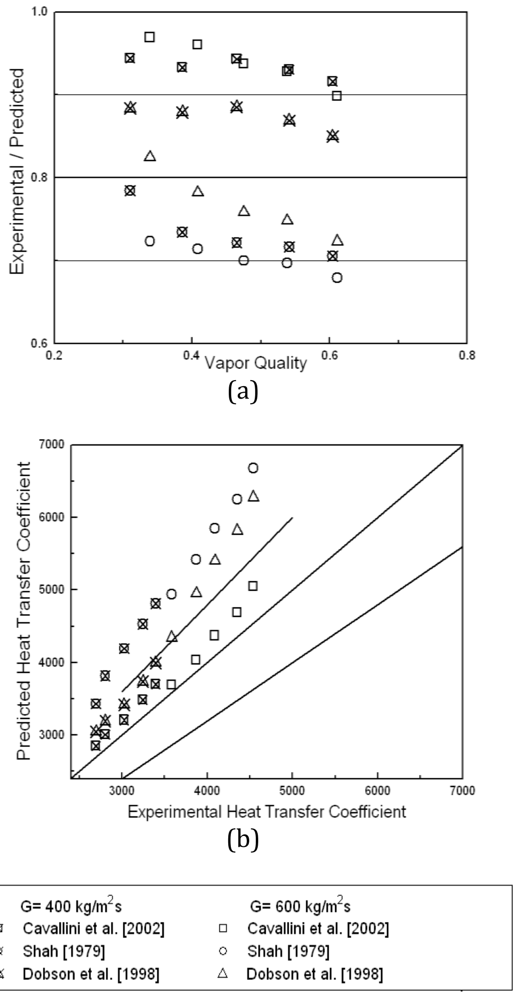
Fig.8 a) Deviation Graph b) Parity Graph for R134a

Shah correlation under predicts the experimental data with a deviation of 12% for a low mass flux, over predicts with a deviation of 16% for medium mass flux and closely follows the experimental data with a deviation of 5% at higher mass flux. Dobson *et al*. correlation exhibits a deviation of 12% at low mass flux, 19% at medium mass flux and 8% at high mass flux.

Comparison on the basis of Flow Regimes: For R134a Cavallini *et al*. correlation predicted the experimental data well in stratified wavy, intermediate and slug-annular regimes with a deviation of 5%, but exhibited larger deviations with annular regime data with 18% deviation. Shah correlation predicted the experimental data with good agreement only for annular regime. For other flow regimes, it exhibited higher deviations. The annular flow correlation of Dobson *et al*. predicted the experimental data in annular regime with a deviation of 8%. For other flow regimes, it exhibited larger deviations in the range of 12 – 19%.

5.3 Comparisons for R407C

Fig. 9 shows the comparisons for R407C. Fig. 9 shows that most of the data falling beyond +20% deviation line.



(a)

(b)

Fig.9 Comparison of Experimental Heat Transfer Coefficient of R407C with Correlations
a) Deviation Graph b) Parity Graph

As shown in Fig. 10a Cavallini *et al*. correlation shows a deviation of 6 - 7% at medium and high mass flux. Dobson *et al*. correlation exhibits a deviation of 13% at medium mass flux and 23% at higher mass flux. In spite of Sweeny multiplier used for zeotrope, R407C, Dobson et al. correlations shows larger deviations from experimental data. Comparatively higher deviations from experimental data are exhibited by Shah correlation with a deviation 27% at medium mass flux and 30% at high mass flux.

For R407C, Cavallini *et al*. correlation predicted the experimental data with good agreement for intermediate and annular flow regimes. Irrespective of flow regime, all other correlations predicted the experimental data with larger deviations.

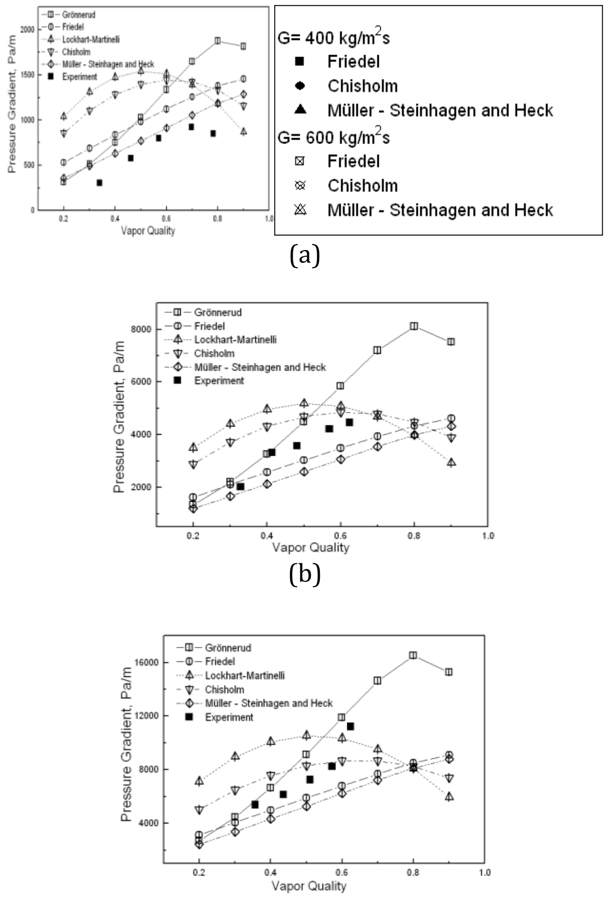
The comparison of experimental heat transfer coefficients of present study with that of correlations shows larger deviations for the experimental data of R407C in the range of 20 – 30%. The maximum deviation exhibited by the Cavallini *et al*., Shah and Dobson *et al*. correlations is in the range of 18%, 16% and 19% respectively, excluding the data of R407C.

6.Experimental Pressure Gradient with Correlations

The pressure drop measured across the test section for condensing flows represents the frictional as well as momentum pressure drop. For the small vapor quality decrements considered in the experimental analysis, the magnitude of momentum pressure drop is negligible compared to the frictional pressure drop. Hence, the measured pressure drop is compared with the frictional pressure drop correlations from literature without deducting the momentum pressure drop component.

6.1 Comparisons for R134a

On an average as shown in Fig. 10, Friedel correlation (L. Friedel, 1979) under predicts the experimental data with a deviation of 22% for the medium mass flux and 20% for the higher mass flux. Müller - Steinhagen and Heck correlation (H. Müller-Steinhagen *et al*, 1986) predicts the experimental data



(a)

(b)

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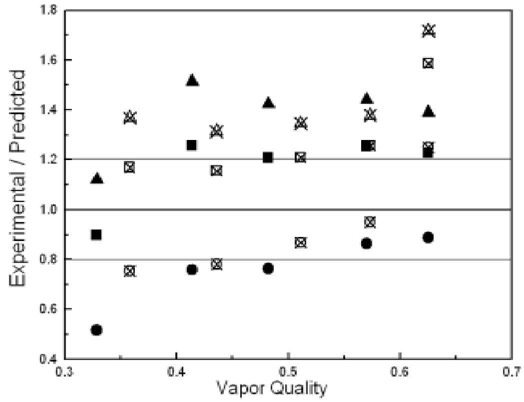

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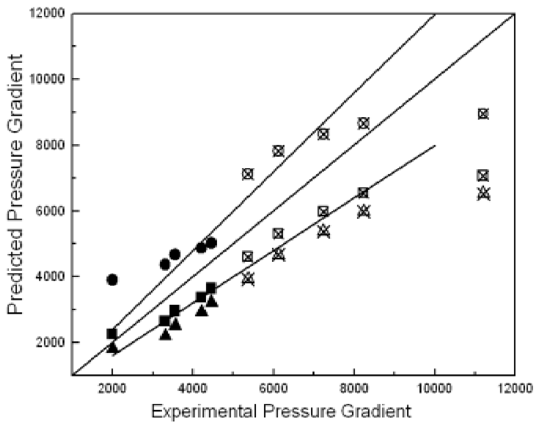


(e)

Fig.10 Comparison of Experimental Pressure Gradient of R134a with Correlations

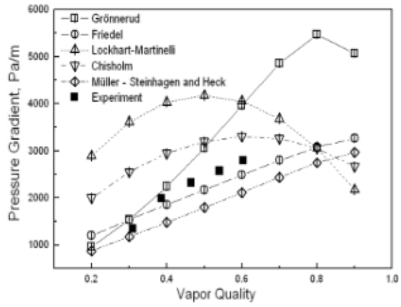
with a deviation of 42% and 35% for medium and high mass fluxes respectively. Chisholm correlation (D. Chisholm, 1973) over predicts the experimental data for R134a with a deviation of 22% for medium and higher mass fluxes.

6.2 Comparisons for R407C

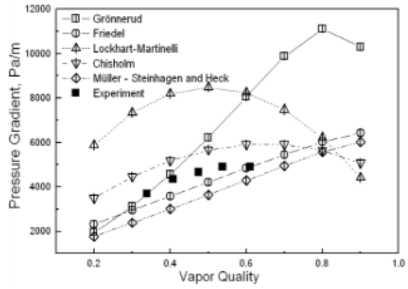
The comparison of experimental pressure gradient of R407C with correlation is shown in Fig. 11 where, the Friedel correlation closely follows the experimental data while all the other correlations exhibit higher deviations.

Fig. 11c shows that only Friedel correlation falls within a deviation band of ±20% with all other correlations exhibiting more than ±20% deviation.

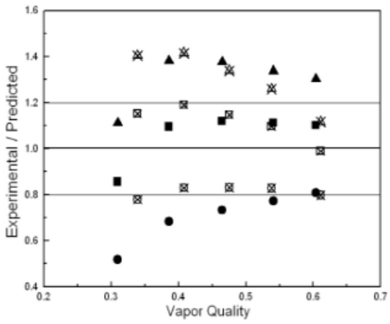
Friedel correlation predicts the experimental pressure gradient of R407C with a deviation of 10% for the medium mass flux and with a deviation of 17% for higher mass flux as shown in Fig. 11c. Chisholm correlation predicts the experimental data better than Müller - Steinhagen and Heck correlation with a deviation of 23% for medium mass flux and 19% for higher mass flux. Müller - Steinhagen and Heck correlation under predicts the experimental data with a deviation of 33% at medium and high mass fluxes as shown in Fig. 11c and 11d.



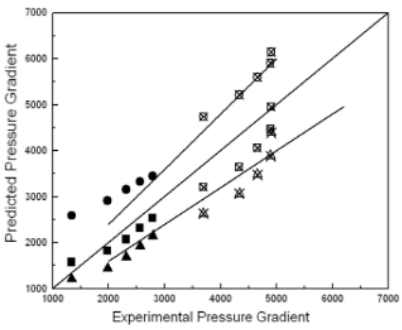
(a)



(b)



(c)



(d)

G= 400 kg/m^{2}s	G= 600 kg/m^{2}s
■ Friedel	⊠ Friedel
● Chisholm	⊗ Chisholm
▲ Müller - Steinhagen and Heck	⌘ Müller - Steinhagen and Heck

Fig.11 Comparison of Experimental Pressure Gradient of R407C with Correlations

Conclusions

The experimental analysis is performed for mass flux of 200, 400 and 600 kg/m^{2}s for a pressure range of 10 – 16 bar at a condensing temperature of 40^{0}C. The experiments are conducted with average vapor quality of test section in the range of 0.3-0.71.

- The experimental set up has shown good repeatability by exhibiting a heat balance error of less than 1% for 98% of the runs, in spite of higher heat duties involved for the refrigerants considered in the present study.
- The uncertainty analysis performed on experimental heat transfer coefficient results in an average uncertainty of ±10.85%. The uncertainty involved with pressure drop is ±0.065% or ±0.081 kPa.
- The validation of test facility with R22 shows that the performance parameters are following the physics involved and shows a definite behavior for runs at low, medium and high mass fluxes.
- The experimental data points plotted on Thome *et al.* flow regime map shows that it spreads in stratified wavy, slug and annular flow regimes.

Experimental Heat Transfer Coefficient

- At low mass flux of 200 kg/m^{2}s, the variation of two phase or local heat transfer coefficient is not significant with vapor quality. At high mass fluxes of greater than 400 kg/m^{2}s, a linear variation of heat transfer coefficient with quality is observed. These trends corroborate with the heat transfer characteristics of the flow regimes predicted by Thome *et al.* flow regime map, viz., stratified wavy and annular.
- The experimental heat transfer coefficients of R134a are almost same of that of R22 as their liquid property combination parameter, φ which is the non dimensional combination of thermal and transport properties, is approximately same for both the fluids.
- The experimental heat transfer coefficients of R407C are approximately 13% lower compared to that of R22 due to high mass transfer resistance of mixture refrigerant though its liquid property combination parameter is 16% higher than that of R22.

Comparison with Correlations

The comparison of experimental data obtained in the present study with Shah, Dobson *et al*. and Cavallini *et al*. correlations resulted into following conclusions.

- Cavallini *et al.* correlation predicted the experimental data falling in stratified, slug and annular flow regimes with a minimum deviation of 5% and a maximum deviation of 18%.
- Other correlations predicted the experimental data with good agreement for only a particular flow regime. Shah correlation exhibited better predictions for annular regime and for fluids with high reduced pressure. Similarly Dobson *et al*. correlation exhibited better predictions for stratified wavy regime.

Experimental Pressure Drop

- Low pressure refrigerant, R134a exhibits high pressure drop as its liquid to vapor density ratio and liquid viscosity being higher compared to the other two refrigerants.
- On an average, the pressure drop of R407C is almost same as that of R22, as both have same value of liquid to vapor density ratio.

Comparison with Correlations

The comparison of experimental pressure drop data with Lockhart-Martinelli ,Grönnerud (R. Grönnerud, 1972), Chisholm, Friedel and Müller-Steinhagen and Heck correlations resulted into following conclusions.

- Lockhart and Martinelli correlation which is widely used in the modeling of condensing flows exhibits a deviation of more than 100% from the experimental data of all the three refrigerants considered in the present study.
- Among the pressure drop correlations considered for comparison, Friedel correlation predicts the experimental pressure drop with good agreement with a minimum deviation of 5% and maximum deviation of 22%.
- All most all the pressure drop correlations exhibited higher deviations for low pressure refrigerant, R134a
- Thus the comparison shows scope for the development of better predictive procedure for pressure drop at high pressures.

Nomenclature

C_p	Specific heat, $kJ/kg-K$
d	Internal Diameter, m
G	Mass Flux, $kg/m^2 s$
h	Heat Transfer Coefficient, $W/m^2\ K$
i	Specific Enthalpy, J/kg
i_{lv}	Enthalpy of Condensation, J/kg
L	Length of the Tube, m
m	Mass flow rate, kg/s
p	Pressure, Pa
$\frac{dp}{dz}$	Pressure Gradient, Pa/m
Q	Total Heat Transfer Rate, W
Re	Reynolds Number
T	Temperature
x	Vapor Quality

Subscripts

in	Inlet
w	Water
o	outlet
r	Refrigerant
s	Saturation
PC	Pre Condenser
TS	Test Section
v	Vapor
$wall$	Wall

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