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Evaporative Heat Transfer Coefficient of a HFC/HC Refrigerant Mixture Under Varied Heat Flux Condition – Low Heat and Mass Flux Application.

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ABSTRACT

An experimental study on in-tube flow boiling heat transfer of R-134a/R-290/R-600a refrigerant mixture and mineral oil has been carried out in varied heat flux test conditions. The heat transfer coefficients were experimentally arrived at temperatures between -8 and 5°C for mass flow rates of 3 to 5 g/s. Acetone is used as a hot fluid which flows in the outer tube of diameter 28.57 mm while the refrigerant mixture flows in the inner tube of diameter of 12.7 mm. By regulating the acetone flow conditions, the heat flux was maintained between 2 and 8 $kW.m^{-2}$ and the pressure of the refrigerant was maintained between 3.2 and 5 bar. Stratified and stratified-wavy flows prevail for the above conditions . The heat transfer coefficient was found to vary between 500 $W.m^{-2}.K^{-1}$ to 1500 $W.m^{-2}.K^{-1}$.The comparison of experimental results with the familiar correlations showed that the familiar correlations over predict.

1. INTRODUCTION

CFCs are to be phased out by 2010 as per Montreal Protocol final act of UNEP(1989). Many research and development efforts during the past decades have evolved R-134a and hydrocarbon blend as a prominent substitutes for R-12 (Devotta et al (1992,1998), Fatouh and El Kafafy(2006)). But if the existing R-12 systems are to be converted to work with R-134a, the compressor itself has to be changed as R-134a is immiscible with mineral oil. The matching oil for R-134a is POE oil which is hygroscopic in nature and may cause moisture entry into the circuit during service and maintenance operations. On the other hand even though hydrocarbons are miscible with mineral oil and also a suitable retrofit substitute for R-12, its flammable nature has caused concerns. Hence it is prudent to look for a refrigerant - oil pair that neither has service nor flammability issues.

Sekhar et al(2003,2004) that adding 9% of hydrocarbon blend to R-134a makes the resultant zeotropic mixture (M09) work well with mineral oil with enhanced performance. It can be a used as a drop-in-substitute for R-12 due to its better oil return characteristics. In this work the flow boiling heat transfer coefficients of this M09 in horizontal tube is studied for the conditions prevailing in the evaporator of a typical domestic refrigerator. The outcome of this work will form a basis for studies on enhanced tubes, mini channel and correlation development for M09.

2. FLOW BOILING - LOW HEAT AND MASS FLUX APPLICATIONS

The flow boiling process in a low heat and mass flux application is closely related to flow patterns and suppression of nucleation. Further, the heat transfer coefficient so evolved also depends on the mode of heat transfer used in the experiments. It is prudent to review the existing literature related to these applications to understand the heat transfer coefficient study and its implications.

There are large numbers of citation available for experimentation of flow boiling performed for annular flow pattern with high mass and heat flux conditions. Chen(1966) hypothesized that in a typical annular flow boiling process, there exist two distinguished regions which are the nucleation dominant region and the convective dominant region. In the former the heat transfer is predominantly due to nucleation and in the latter it is due to convective vaporization of liquid at the liquid-vapor interface where the nucleation is considered to be suppressed. Jung et al (1989a, 1989b), Shin et al (1997) and many other experimenters on refrigerants and mixtures have reported that a clear cut demarcating vapour quality can be realized between nucleate and convective boiling phenomenon in both mixture and pure fluids. It is also reported that in the convective region the heat transfer coefficients are independent of heat flux for a given pressure. For different mass flow rates, with a constant pressure and varying heat flux, the heat transfer coefficient lines merge into a single line once the nucleation is suppressed.

However, in the case of domestic refrigerators and freezers, for which the present study is intended, the heat flux and mass flux are low. The predominant flow patterns corresponding to these conditions are stratified and stratified - wavy flow as mentioned in Aprea et al (2000) and Wattelet et al (1994). Stratified flow pattern differ from an annular pattern by the presence of an un-wetted perimeter. Whereas in stratified – wavy flow pattern, waves of flowing liquid splash on the side of the tube and thus increases the wetted perimeter intermittently. It is reported that in the evaporator tubes of these conditions, unlike annular flow, the nucleation is not completely suppressed even at higher vapour qualities. Therefore, both nucleate boiling and liquid convection are active heat transfer mechanisms in the global heat transfer process. It is reported that heat transfer coefficient is depended on heat flux through out the flow boiling process and the nucleation was not fully suppressed.

Jabardo et al (2000) has also reported that heat transfer coefficient at lower saturation temperature tend to raise than one at high saturation temperature. This was because the thermal resistance of the film attached to the tube surface is inversely proportional to the thermal conductivity of the liquid. Since the conductivity of the liquid diminishes with temperature, so does the heat transfer coefficient. Experiments of Ross et al (1987) on R-152a and R-13B1 claimed that at high pressure (4.75 bar) the effect of nucleation was not dissipated at higher vapor qualities. But the author also observed that when the pressure was lowered (around 1.5 bar) the nucleation was suppressed and the heat transfer coefficient was dominated only by convection.

It is to be noted that in most of the cases due to practical difficulty the length of the test section was always limited. Ross et al (1987) claimed that, applying high heat flux for a specified mass flux to achieve high vapor qualities in small test sections is not replicating the real conditions found in appliances. Shin et al (1997) claimed that it is not proper to use an electrically heated test section when the flow is stratified or tube is partially dried out. Jung et al (1989a, 1989b) suggested that to study refrigerant mixture the preferred mode of heating would be a counter-current heat exchanger. Kattan et al (1998a, 1998b) suggested that in a real time appliance the heat flux would never be constant and heat flux should not be imposed by the experimenter as done in electrically heated test sections. It is not a reasonable approximation to use a uniform heat flux for stratified and stratified-wavy flow patterns and the preferred mode of heating, especially for mixtures, would be through a counter current heat exchanger.

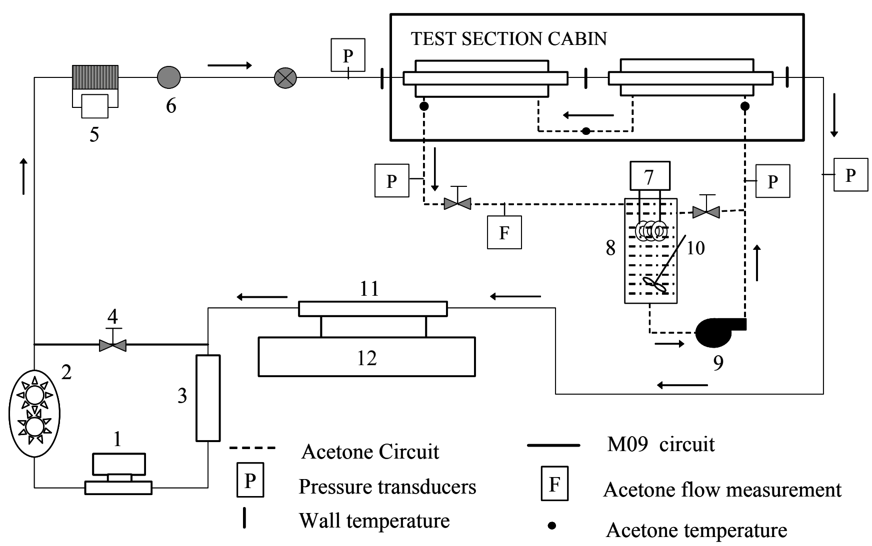
There are numerous correlations available in the published literature to predict the heat transfer coefficient. However it is reported in many literature such as Shin et al (1997) and Aprea et al (2000) that predictions through familiar correlations are far deviating from experimental results in many cases. It is needless to say that in a typical evaporator the refrigerant enters the compressor either in a saturated or in super heated state .. Hence it is vital to study the complete vaporization process of any refrigerant. However in many electrically heated test sections the higher vapour qualities are avoided due to apprehensions of tube burnout phenomenon.

In the present work a test section of 10m length with a counter current heat exchanging arrangement is used to study the variation of heat transfer coefficient of M09 undergoing flow boiling by allowing complete vaporization (x

= 1) under varied heat flux conditions. Flow pattern map reported by Thome (2005) is used to identify the flow pattern for a given operating condition inside the heated channel. The significance of nucleation contribution at high vapour quality in a low mass and heat flux application is highlighted. The paper also compares the experimental results with Gungor – Winterton(1985) , Kandlikar (1990) and Wattelet et al (1994) correlations. It is a known that the correlation evolved for pure fluids or for specific fluids might not successfully predict for any other new fluids. However, the model which gives minimum deviation could be selected for further modification to suit this working fluid, which is a continuation of this work. It is also needless to say that some of well known correlations are modified versions of existing correlations for pure or mixtures.

3. EXPERIMENTATION

The layout of the experimental test rig is shown in Figure 1. The test facility consists of three circuits which are 1. M09 circuit; 2. the acetone circuit to heat the M09; 3. the auxiliary cooling circuit to condense the M09. The entire test section is constructed using hardened copper tube. The inner tube of the test section is 12.7 mm in diameter with 1.02 mm thickness and the outer tube diameter is 28.58 mm with 1.21 mm thickness. The test section is split in to 20 small sub section of 0.5 m effective length. At every subsection, the inlet and outlet acetone temperatures and outer wall temperature of the inner tube carrying M09 are measured using PT100 RTD sensors (accuracy of ±0.15^{o}C). Since the considered mixture is zeotropic in nature, the temperature was measured at the top and bottom of the inner tube at every section . The average of these two temperatures is considered as T_{WO}. The temperatures at various locations are recorded using a data acquisition system. The entire test section is effectively insulated with thermorex foam, thermocole and glass wool.



1. Corriolis mass flow meter
2. Magnetically coupled gear pump
3. Liquid receiver
4. By – pass valve
5. Pre- heater
6. Sight glass
7. Temperature controller
8. Hot bath
9. Acetone pump
10. Stirrer
11. Tube – in – tube condenser
12. Auxiliary condensing unit

Figure 1 Schematic view of the Experimental setup

The M09 is pumped to the inlet of the test section (with 1 to 4^{o}C of subcooling) using a magnetically coupled sealess gear pump and it is boiled by the hot acetone which flows in counter-current direction. The vaporized M09 is then condensed back to a liquid receiver using an auxiliary condensing unit. The liquid is pumped back from the receiver to the test section via a mass flow meter. The cold acetone leaving the test section flows in to a temperature controlled bath where the acetone is heated and pumped back to the test section using a magnetically coupled centrifugal pump. The M09 is metered using a Corriolis mass flow meter with ± 0.1% accuracy. The pressure of the M09 is measured at the ends of the test section using an absolute piezo-resistive transducer with ±0.1% accuracy. Acetone is metered using a suitable glass tube rota-meter with ±0.5% accuracy. An electric pre-heater of 100 W is provided at the inlet of the test section to precisely control the entry condition of M09.

Parameters fixed during tests are pressure and mass flow rate of M09, mass flow rate of acetone, inlet temperature of acetone. Once steady state conditions are achieved, the signals from pressure transducers, mass flow meter and temperature sensors are logged into the data logger and transferred to spread sheet. The heat transfer coefficients are further reduced by dynamically linking a MATLAB program with the spreadsheets and REFPROP software. Using compressor manufacturer's catalogue the mass flow rates prevailing in commercial refrigerators are estimated and the operating conditions are selected as shown in Table 1.

Table 1 : Operating range

Parameters	Range
$\dot{m}_R$ $(g.s^{-1})$	3 to 5
p_R (bar)	3.2 to 5
q $(kW.m^{-2})$	2 to 8

4. DATA REDUCTION

The reported heat transfer coefficients are average heat transfer coefficient of every sub-sections. The tube wall temperatures (T_{WO}), acetone temperature (T_A) , pressure at the inlet and the exit of the test section (P_R), mass flow rate of M09 ($\dot{m}_R$), acetone mass flow rate ($\dot{m}_A$) are the observed data. The actual location of incipience and completion of saturated boiling of M09 in the test section is not known directly. Hence the acetone temperature distribution is polynomially fit against the length of the test section ($T = f(L)$) . Based on the inlet condition, saturation condition and the latent heat of evaporation of the M09 , the entire length of the test section is divided into sub-cooled, two phase and super heated length. Sub-cooled length is estimated by equating the acetone heat load to the sub-cooled load of the M09 as shown in Equation 1. Further, balance of acetone heat load is iteratively balanced against the latent heat of vaporization ($\dot{m}_R$. Δi_{fg}) along the length of the test section. This gives vapour quality ($x = f(p - \Delta p , hf + \Delta i_{fg})$) and M09 equilibrium temperature for a given vapour quality ($T_R = f(p - \Delta p, x)$) at the end of every sub section. The length after x = 1 is the superheated length which is not considered for the study.

The pressures are measured only at the inlet and outlet of the test section. Since the pressure drop is negligible in a low mass flow rate application, a linear variation of pressure drop is assumed along the test section. Considering the heat flux, the Fourier radial heat conduction equation and the measured outer wall temperature (T_{WO}), the inner tube wall temperature (T_{WI}) can be calculated using equation 2 . Finally the heat transfer coefficients can be evaluated using equation 3. REFPROP is used to estimate the thermo-physical properties of the fluids.

$$(\Delta L)_{SUB} = \frac{\dot{m}_R Cp_R (T_{BUB} - T_{SUB})}{\dot{m}_A Cp_A \left(\frac{dT}{dL}\right)_A} \quad (1)$$

$$T_{WI} = T_{WO} - \frac{q_{SEG} \ln \frac{D_O}{D_I}}{2\pi k (\Delta L)_{SEG}} \quad (2)$$

$$h_R = \frac{q_{SEG}}{(T_{WI} - T_R)} \quad (3)$$

5. VALIDATION

The error analysis for the heat transfer coefficient is carried out by applying the uncertainty analysis as suggested in Moffat (1988). The input to the heat transfer coefficient is the heat flux (q) and the wall super heat. In this counter-current heat exchanger the heat flux (q) depends on mass flow rate($\dot{m}_A$) and temperature difference of acetone. The overall uncertainty in estimating h_R is found to be within ± 6.5 to ± 12.2 % The repeatability check conducted for the lowest mass flow rate of 3.5 g.s^{-1} at 3.2 bar indicated that the deviation of heat transfer coefficient is within ± 1.5 % only. The test results were validated with R-134a and compared with Wattelet et al (1994) which was evolved with similar conditions and the average deviation is found to be within ± 7%.

6. RESULTS AND DISCUSSION

The experiment has been conducted for the range of operating conditions mentioned in Table 1. The mass fluxes corresponding to 3 to 5 g.s^{-1}, are 28 to 56 kg.m^{-2}.s^{-1}.It is found that this flow conditions result in stratified and stratified-wavy pattern which were confirmed using flow pattern maps of Thome (2005).

As mentioned earlier, in a counter - current heat exchanger test section, the local heat flux varies along the length. Figure 2 shows the heat flux variation for a constant acetone inlet condition and M09 pressure of 4 bar at different mass flow rates . It can be seen that heat flux during incipience of boiling is higher for low flow rates. This is because at lower flow rate the nucleation resistance is low and the boiling occurs rapidly than at higher flow rate. This high level of flashing at low mass flow rate results in higher heat flux being taken from the acetone. However, in later part of the tube there will not be sufficient liquid (M09) in the test section to sustain the same heat flux. But at higher flow rate due to increased nucleation resistance by the flow velocity in the initial stage of boiling, the heat flux starts with a low value. However as the flow proceeds due to increased vapor velocity of the M09, the heat flux increases due to convective vaporization. In the later part of the tube, the heat flux rapidly decreases due to deficiency of liquid M09.

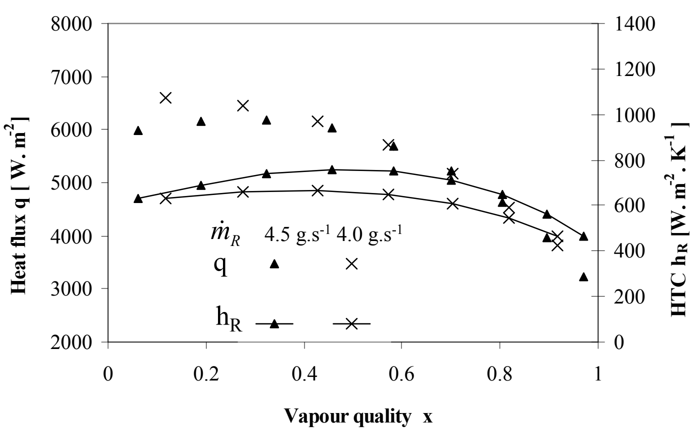


Figure 2 : Variation of heat flux with respect to dryness fraction

The variation of heat transfer coefficient against vapour quality is also shown in figure 2. It is seen that at lower M09 flow rates higher rate of nucleation causes a rapid removal of heat from the surface, which results in a sudden drop in inner wall temperature and hence the wall super heat also decreases leading to higher heat transfer coefficient. The observed trend of heat transfer coefficient with respect to vapour quality is due to higher nucleation in the beginning which subsides as the vapor fraction increases and the convective heat transfer compensates to some extent due to increased vapor velocity.

The variations of heat transfer coefficient against vapour quality for a constant pressure and mass flow rate for different level of mean heat fluxes are shown in figure 3. It is known that in an annular flow, for a constant pressure and mass flow rate, the heat transfer coefficient is independent of heat flux once the nucleation is suppressed. Further the heat transfer coefficient lines will merge in to single line (Jung et al (1989a, 1989b) and Shin et al (1997)) after the point of suppression. But in this case of stratified flows it can be seen that lines do not merge completely which indicates that the heat transfer coefficient is a strong function of heat flux even at higher dryness fraction because of nucleate boiling prevailing even at high vapor quality in a low mass and heat flux application.

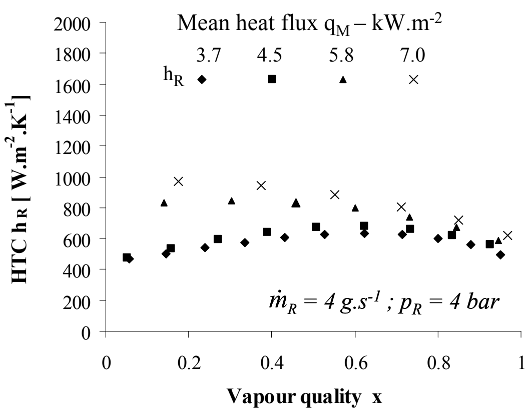


Figure 3: Influence of mean flux on heat transfer coefficient at 4 bar

In the absence of an exclusive correlation for the considered mixture, a comparison of the experimental results with prediction based on familiar correlations for other pure fluids or mixtures is not unusual . Hence the experimentally evolved heat transfer coefficient values of M09 are compared with correlations. This would help in identifying a correlation (from existing correlations) that closely predicts the heat transfer coefficient of the present mixture and the same correlation can be considered for suitable modification to improve accuracy. A comparison of experimental heat transfer coefficient with the correlation result such as Gungor and Winterton (1986), Kandlikar (1990), Wattelet et al (1994) marked as GW86, KN90 and WT94 respectively is shown in Figure 4. The Kandlikar correlation requires a fluid dependent number (Fl) for M09, which is not readily available. The value of R-134a (Fl = 1.63), which is also 91% of the mixture, is considered for Kandlikar correlation. It is found that the prediction of these correlations is mostly scattered between +80% to -40% deviation. However, Kandlikar correlation predict fairly well for this mixture and around 60 % of the experimental data fall within ± 25% deviation. Hence it is necessary to evolve a suitable correlation for M09 applicable to stratified and stratified - wavy flow regimes which will be presented in the next part of the work.

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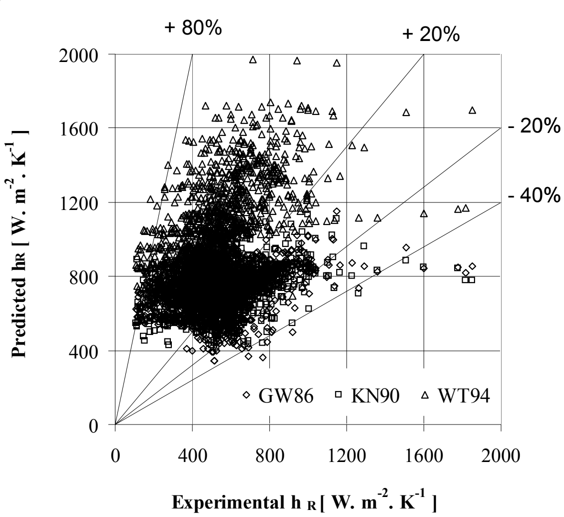

Figure 4 : Deviation plot

7. CONCLUSIONS

- Experiments on M09 (R-134a/ R-290/ R-600a: 91% / 4.068% /4.932% by mass) performed at low mass flow rates revealed that nucleation effects are present even at high vapor quality.
- It is observed that for a low mass flow rate ($\dot{m}_R$) of 3 to 5 g.s^{-1} and heat flux of 2 to 8 kW.m^{-2}, the heat transfer coefficient is estimated to be falling between 500 to 1500 W.m^{-2}.K^{-1}.
- The results also indicate that in stratified and stratified - wavy flow, the HTCs are completely dependent on heat flux even at moderate qualities when pressure (p_R) and mass flow rate($\dot{m}_R$) are kept constant.
- It is found that the predicted heat transfer coefficient values using familiar correlations are scattered between +80% to -40% deviation.
- Kandlikar correlation predicts fairly well the heat transfer coefficient of this mixture when the flow is stratified. Around 60 % of the experimental data fall within ±25% deviation.

Nomenclature:

A	cross section area	[m^{-2}]
Cp	specific heat capacity	[J.kg^{-1}.K^{-1}]
D	diameter of inner tube	[m]
Fl	Fluid parameter	
g	gravity constant (9.81)	[m.s^{-2}]
G	mass flux ($\dot{m}_R$/A)	[kg.m^{-2}.s^{-1}]
h, HTC	heat transfer coefficients	[W.m^{-2}.K^{-1}]
i	enthalpy	[J.kg^{-1}]
k	Copper Therm. Cond.	[W.m^{-1}.K^{-1}]
L	length	[m]
$\dot{m}_R$	mass flow rate	[g.s^{-1}]
q	heat flux	[W.m^{-2}]
T	Temperature	[K], [^{o}C]
x	vapour quality	

Subscripts

A	acetone
BUB	bubble point
DEW	dew point
fg	latent heat
I	inner side, inner tube
M	mean values
O	outer side
R	local M09 condition
SAT	saturation
SEG	segmental
SUB	sub-cooled
TP	two phase
WI	inside wall of inner tube
WO	outside wall of inner tube

REFERENCES

Aprea, C., Rossi, F., Greco, A., 2000, Experimental evaluation of R22 and R407C evaporative heat transfer coefficient in a vapour compression plant, Int. J. Refrigeration , vol. 23 , no. 5: p. 366-377.

Chen, J.C., 1966, Correlation for boiling heat transfer to saturated fluids in convective flow, Industrial and Engineering Chemistry, Process Design and Development, vol. 5 : p. 322-329.

Devotta, S., Gopichand, S., 1992, Comparative assessment of HFC134a and some refrigerants as alternatives to CFC12, Int J Refrigeration, vol.15, no.2: p. 112–118.

Devotta, S., Parande, M.G., Patwardhan, V.R., 1998, Performance and heat transfer characteristics of HFC-134a and CFC-12 in a water chiller, Applied Thermal Engg, vol.18, no.7: p. 569 – 578.

Fatouh, M., El Kafafy, M. , 2006, Assessment of propane/commercial butane mixtures as possible alternatives to R134a in domestic refrigerators, Energy Conversion and Management, vol.47: p 2644–2658.

Gungor, K.E., Winterton, R.H.S., 1986, A general correlation for flow boiling in tubes and annuli, Int. J. Heat Mass Transfer, vol. 29: p. 351-358.

Gungor, K.E., Winterton, R.H.S., 1987, Simplified general correlation for saturated flow boiling and comparisons of correlations with data, Chem. Eng Res. Des., vol.65, p. 148-156.

Jabardo, J.M.S., Filho, E.P.B., 2000, Convective boiling of halocarbon refrigerants flowing in a horizontal copper tube an experimental study, vol. 23: p 93 – 104.

Jung, D.S., McLinden, M., Radermacher, R., Didion, D., 1989a, Horizontal flow boiling heat transfer experiments with a mixture of R22/R114, Int. J. Heat Mass Transfer, vol 32: p. 131-145.

Jung, D.S., McLinden, M., Radermacher, R., Didion, 1989b, D., A study of flow boiling heat transfer with refrigerant mixtures. Int. J. of Heat and Mass Transfer, vol. 32 : p. 1751-1764.

Kandlikar, S.G., 1990, A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes, J. of Heat Transfer, vol. 12: p. 219-228.

Kattan, N., Thome, J.R., Favrat, D., 1998a, Flow boiling in Horizontal tubes: Part 1 – Development of a diabatic two-phase flow pattern map, J. of Heat transfer, vol.120 , no.1: p. 140-147.

Kattan, N., Thome, J.R., Favrat, D., 1998b, Flow boiling in horizontal tubes : Part 3 – Development of a new heat transfer model based on flow pattern, J. of Heat transfer, vol. 120 , no.1: p. 156-165.

Moffat, R.J., 1988, Describing the uncertainties in experimental results, Experimental Thermal and Fluid Science, vol. 1: p 3-17.

REFPROP, NIST Standard Reference Database 23, Version 7.01.

Ross, H., Radermacher, R., Di Marzo, M., Didion, D., 1987,cHorizontal Flow boiling of pure and mixed refrigerants, Int. J. Heat Mass Transfer, vol.30, p. 979-992.

Sekhar, S.J., Kumar, K.S., Lal, D.M., 2004, Ozone friendly HFC134a/HC mixture compatible with mineral oil in refrigeration system improves energy efficiency of a walk in cooler, Energy Conversion and Management, vol. 45: p. 1175-1186.

Sekhar, S.J., Premnath, R.P., Lal, D.M., 2003, On the performance of HFC134a/HC600a/HC290 mixture in a CFC12 compressor with mineral oil as lubricant, Ecolibrium-Journal of Australian Institute of Refrigeration, vol. 2, p. 24-29.

Shin, J.Y., Kim, M.S., Ro, S.T., 1997, Experimental study on forced convective boiling heat transfer of pure refrigerants and refrigerant mixtures in a horizontal tube, Int. J. Refrigeration, vol.20: p. 267-275.

Thome, J.R., 2005, Update on advances in flow pattern based two-phase heat transfer models, Experimental Thermal and Fluid Science, vol. 29 : p. 341-349.

United Nations Environment Programme, 1989, Montreal protocol on substances that deplete the ozone layer. Final Act.

Wattelet, J.P., Chato, J.C., Souza, A.L., Christoffersen, B.R., 1994, Evaporative characteristics of R12, R134a and a mixture at low mass fluxes, ASHRAE Trans. Symposia vol.2 ,no.1 : p. 603 – 615.

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