Proceedings of EnCon2007
1st Engineering Conference on Energy & Environment
December 27-28, 2007, Kuching, Sarawak, Malaysia

ENCON2007-030

Refrigeration Efficiency Improvement via Heat Transfer Enhancement

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Abstract—**The criticality in the improvement of refrigeration system efficiency has now shifted to heat exchangers that need to work across the least possible temperature difference. With smaller temperature differences across the evaporator and condenser, the compressor work is reduced and energy efficiency is improved. To reduce these temperature differences, techniques of heat transfer enhancement are needed. This forms the basis of the present study in which heat transfer characteristics of enhanced annuli are studied experimentally in a water-cooled condenser. Normally superheated refrigerant vapour enters the condenser in turbulent flow conditions, and exits as subcooled liquid which is in the transition region of flow. The annulus-side coefficients are determined using the Wilson Plot technique. The results for the augmented surface are presented graphically and compared with the non-augmented case. For turbulent flow of vapour in plain annuli, the form of the correlation by Monrad and Pelton was satisfactory. For vapour in enhanced annuli, the correlation of Knudsen and Katz determined the basic form. The enhanced annulus outperforms the plain tube by about 25%-30% for superheated vapour heat transfer coefficient. The new empirical constants resulted in good prediction for turbulent superheated vapour flow in the annulus for both plain and enhanced annuli. For the single phase heat transfer correlation proposed, all predicted data are within ±5% of experimental values. Though R-22 has been chosen for the present study, the dimensionless form of presentation of results enables the correlations to be applied to other refrigerants.**

Keywords: heat transfer enhancement, refrigeration systems, energy efficiency improvement

I. INTRODUCTION

Condensation heat transfer, both inside and outside horizontal tubes, plays a key role in refrigeration, air-conditioning and heat pump applications. In recent years the science of condensation heat transfer has been severely challenged by adoption of substitute working fluids and new enhanced surface of heat exchangers. Well-known and widely established correlations to predict heat transfer during condensation may seem to be quite inaccurate in some new applications, and consequently a renewed effort is now being dedicated to the characterization of flow conditions and associated predictive procedures of heat transfer with condensing vapour. Tremendous research efforts have been made to miniaturize thermal systems and identify innovative technologies for heat transfer enhancement. Heat transfer enhancement techniques are classified as: *passive enhancement techniques*- not involving any direct application of external source of power, and *active enhancement techniques*- which require direct application of external power. Passive techniques, in general, have lower energy cost, lower heat transfer augmentation, and large pressure drops when compared to active techniques. Depending on the application and economic feasibility, the designer has to decide on the type of augmentation techniques used. Among such applications, the use of Double-Tube heat exchanger demonstrates an impressive potential for enhanced condensation heat exchangers. The WSHP unit is a reversible-cycle heat pump unit which uses water as the heat source when running in the heating mode and as the heat sink when in the cooling mode. For this purpose, a water-to-refrigerant heat exchanger is used. Generally, this could be a double tube or plate heat exchanger, figure 1.

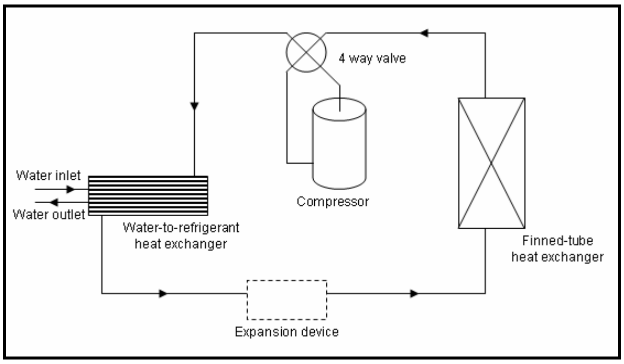

Figure 1. Refrigerant circuit for WSHP unit

The refrigerant circuit is completed with a compressor, expansion device and a finned-tube heat exchanger. The Double Tube heat exchanger is a very efficient design. The water and refrigerant are in direct thermal contact through the walls of the inner tubes. The water and the heat-transfer fluid are in counterflow. This is of particular advantage when desuperheating or subcooling occurs in the heat

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exchanger along with phase change. This type of heat exchanger has two loops similar to those described in the shell-and-tube heat exchanger.

For the given arrangement of double tube, the refrigerant flows in the annulus whereas the cooling water flows in the inner tube, shown in figure 2.

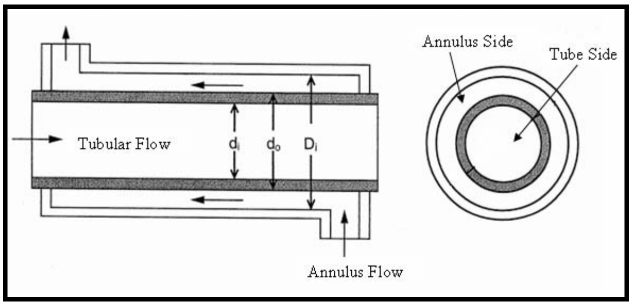

Figure 2. Concentric tube configuration (double tube heat exchanger)

II. OBJECTIVES AND SCOPE

The purpose of the study is to obtain the single phase heat transfer coefficient for the superheated vapor refrigerant flow in the annulus of the double tube exchanger. This research is conducted because the inner tube used in the current double-tube design is a 3D corrugated outer surface enhancement for copper tube (Turbo-C Copper Top Cross); it is difficult to generate a mathematical model for such complex geometries, no standard equation for the heat transfer coefficient is available for the Turbo-C copper top cross, and modification of equations from previous literature will require extensive measurements of pressure, temperature of the refrigerant and water side, and the surface temperature of the fin and fin root surface. Thus, with the above mentioned motivation, the objective of the current research is to characterize the heat transfer performance of the Turbo-C Copper Top Cross tube for superheated vapour. The enhancement effect of the Turbo-C Copper Top Cross tube was also studied for comparison with a plain tube. Experiments were conducted to obtain the necessary data which will be used for parametric analysis of the following independent parameters.

Table 1. Parametric Range for Experiments

No.	Experimental Condition
1	Test Fluid – R22 Chlodifluoromethane
2	Test Section (a) Plain Tube (b) Copper Top Cross

III. WILSON PLOT METHOD

A useful technique for determining individual resistance from an overall resistance was devised by Wilson (Briggs et al [1]) in 1915. Wilson was interested in determining the effects of water temperature and velocity on the overall coefficient for a steam condenser. In most heat exchanger condensing processes, the primary measurements taken connect the energy transferred to the temperature difference between two fluid streams. The overall rate of heat transfer can be calculated from:

$$Q = U\ A_O \Delta T \tag{1}$$

The objective of the condensation test is directed at determining the condensate heat transfer resistance from the overall thermal balance. When a vapour condenses on the outside of a tube which is internally cooled by flowing liquid, figure 3, the component thermal resistances are coupled to the overall resistance by:

$$\frac{1}{A_O U} = R_O + R_W + R_i = \frac{1}{A_O h_O} + R_W + \frac{1}{A_i h_i} \tag{2}$$

It is noted that there will be some heat loss through the outer steel tube as the outer surface of the condenser coil will not be insulated during use in actual unit applications. This heat loss is an added advantage to the entire heat rejection process by the condenser coil. Since the thermal conductivity of steel is less compared to the thermal conductivity of copper ($k_{steel} << k_{copper}$), the outer surface is assumed to be adiabatic. Additional precaution is taken by insulating the outer steel pipe with 1" thick Superloon insulation material to create an adiabatic outer surface. Equation (2) can then be transformed to:

$$\frac{1}{U} = \frac{A_O}{A_i}\frac{1}{h_i} + \left[A_O R_W + \frac{1}{h_O} \right] \tag{3}$$

Equation (3) can than be interpreted as a linear function which has the linear form $y = mx + b$, if R_w and h_o are constant, and it is the basis of the Wilson Plot method. Given such a test series, a line can be plotted as shown in Fig. 3.

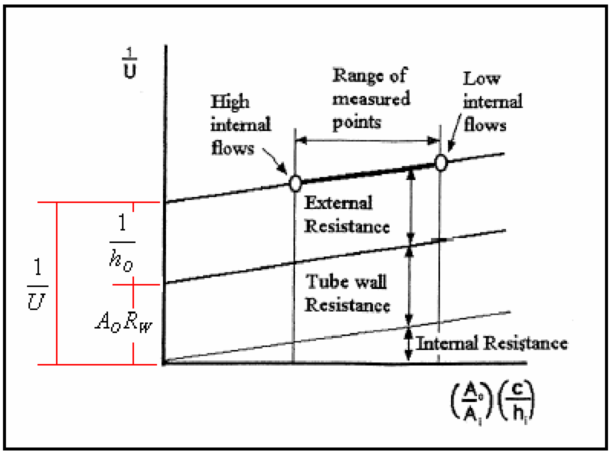

Figure 3. The Wilson Plot - general features

The experimental points extend from a high internal flow rate, with low overall heat transfer resistance, to a low internal flow rate and high overall heat transfer resistance. A line drawn through these points intersects the U^{-1} – axis at a point where the internal heat transfer resistance approaches zero, and thus the overall heat transfer resistance at this point is simply the sum of wall resistance and the external heat transfer resistance. The internal heat transfer coefficient, at any flow rate, can than be determined independently using (3). It should be noted that the accuracy and reliability of the Wilson Plot Method depends on the validity of the correlation used to describe the internal and external heat transfer coefficient.

IV. TEST FACILITY AND METHOD OF DATA ANALYSIS

The commissioning of the test section was carried out using HCFC R-22 as refrigeration medium in the annulus and water as coolant in the inner tube. The primary objective of these tests is to determine the correlation for overall heat transfer of the condensation process. The overall energy flow in the test section can be determined using three independent routes. These routes use:

- the temperature increase in the coolant flow
- the mass of condensate collected under the test section
- the circumferentially integrated temperature drop across the tube wall

Deans et al [2] has reported that the maximum difference in the calculated overall energy flows using these three routes to analyze the condensation process was less than 5%. The temperature increase in the coolant flow is chosen due to its ease in experimental measurements and the use of this method with respect to the Wilson Plot method. The test facility was designed and assembled to cater for this need. The test facility is capable of testing either plain or enhanced straight Double-Tube condenser at conditions typical of a vapour compression refrigeration system.

V. TEST FACILITY CONFIGURATION

The Double-Tube heat exchanger configuration used in this study is a one-pass, single-track system. The single-track system means that that only one test section and one refrigerant compressor may be installed in the system for individual testing, which simply means that the refrigerant is not circulated within the test facility during the data run. The refrigerant passes through the Double-Tube condenser only once as it travels from the high-side (compressor discharge line) to the low-side (compressor suction line). The single-track and single-pass system makes it possible to obtain one successful data point every time the facility is operated. If the operating conditions such as refrigerant mass flow rate or compressor discharge pressure are varied (non-geometric variables), it is possible to obtain additional data points without changing the test tube or compressor. The presence of the Electronic Expansion Valve (EXV) allows us to do this. Use of conventional expansion device such as the capillary tube here will require additional work repetition where the refrigerant circuit will have to be vacuumed, leak tested and recharged for the individual length of capillary tube needed to throttle the refrigerant flow. The two main variable components in this test facility are the test section and the refrigerant compressor. The facility is designed and installed with valves and fittings for both the refrigerant medium and the cooling water. This is to allow for quick and easy replacement of the test section and/or refrigerant compressor. Each refrigerant has an individual range of refrigerant flow rate, depending on the amount of refrigerant charge and compressor suction and discharge pressure. Three types of refrigerant compressor were chosen (1 HP, 2 HP, and 3HP) to provide a wide range of refrigerant mass flow rates. Straight horizontal test section with two types of inner tube was used in this study, i.e. (1) Plain Tube (Plain Annulus) and (2) Turbo-C (Enhanced Annulus).

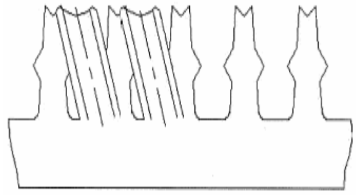

Figure 4. Illustration of enhanced surface (Turbo-C)

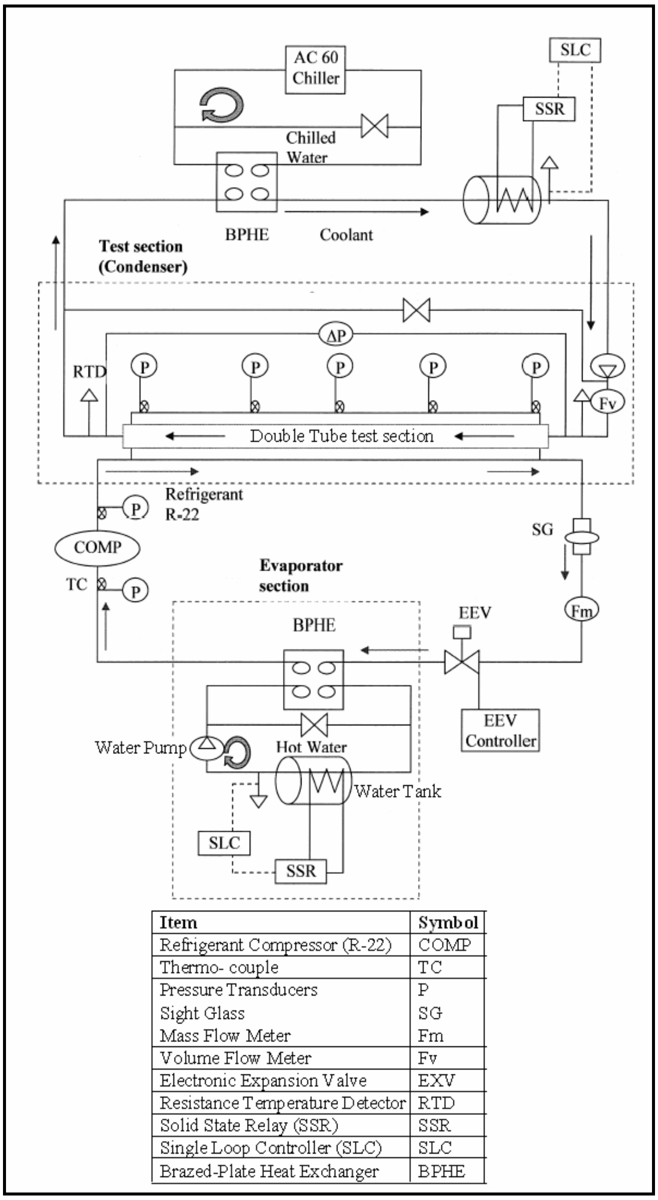

Figure 5: Schematic Diagram of Experiment

Table 2: Description of the Test Section Tubes Used

Description	**Plain Tube**	**Turbo-C**
OD	22.22 mm	22.22 mm
ID	17.82 mm	17.82 mm
Length	2980 mm	2980 mm
Outer Surface	Smooth Surface	3-D Integral Fin
Inner Surface	Smooth Surface	Smooth Surface
Other Information	N.A.	Pitch of Fin = 0.75 mm Pitch of Corrugation = 5 mm

VI. DATA REDUCTION & EXPERIMENTAL UNCERTAINTIES

In the Dry Run Test the compressed vapour enters the test section as superheated vapour and exits in the same phase. The heat transferred to the cooling water is kept sufficiently low to allow the vapour temperature at exit to be above the saturation temperature; hence no phase change occurs. This will allow us to obtain the single phase heat transfer coefficient for superheated vapour.

$$Q_{SH} = \dot{m}_O (\Delta h)_{SH} = \overline{U}_{SH} A_O (LMTD) \quad (4)$$

$$\frac{1}{\overline{U}_{SH} A_O} = \frac{1}{\overline{h_{SH}} A_O} + R_W + \frac{1}{\overline{h}_i A_i} \quad (5)$$

Uncertainties in the experimental data were calculated based on the propagation of error method, described by Kline and McClintock [3]. Accuracy for various measurement devices, refrigerant properties and water properties are given in Table 3, 4 and 5.

Table 3. Uncertainties of various measurement devices

Parameter (Make, Type)	Uncertainties
Water Volume Flow (YOKOAWA, Magnetic Flow Meter)	$\pm$ 0.08% of reading
Refrigerant Mass Flow Meter (YOKOGAWA, Coriolis)	$\pm$ 0.05% of reading
Refrigerant Pressure (HAWK, Pressure Transducer)	$\pm$ 0.18 psig
Water temperature (CHINO, Pt-100 RTD)	$\pm$ 0.01^{0}C
Refrigerant Temperature (VANCO, Type-T Thermocouple)	$\pm$ 0.06^{0}C

Data provided by OYL R&D Center (M) Sdn. Bhd.

Table 4. Uncertainties of refrigerant R-22 properties

Predicted Properties	Uncertainties	Source
Density	$\pm$ 0.1%	Kamei et al. [4]
Isobaric Heat Capacity	$\pm$ 1.0%	
Viscosity	$\pm$ 2.1%	Klein et al. [5]
Thermal Conductivity	$\pm$ 3.7%	McLinden et al. [6]

Properties data obtained from ASHRAE Fundamentals Handbook 2001 [7]

Table 5. Uncertainties of predicted water properties

Predicted Properties	Uncertainties	Source
Density	$\pm$ 0.02%	Wagner and Pru*B* [8]
Isobaric Heat Capacity	$\pm$ 0.3%	
Viscosity	$\pm$ 0.5%	Kestin et al. [9]
Thermal Conductivity	$\pm$ 0.5%	

Properties data obtained from ASHRAE Fundamentals Handbook 2001 [37]

Uncertainties in the single phase heat transfer coefficient (superheated vapour and subcooled liquid) and the condensation heat transfer coefficient were calculated for various test runs in the smooth and enhanced annulus as a root-sum-square (RSS) method. Experimental results and the associated uncertainties are listed in Table 6. The uncertainties are dominated by the uncertainties associated with the refrigerant and water properties. Higher uncertainties were found at higher refrigerant mass flow rate.

Table 6. Uncertainty analysis for experimental data

Test Sequence	Plain Tube h (W/m^2.K)		Turbo-C h (W/m^2.K)	
Compressor	Highest	Lowest	Highest	Lowest
1 HP	345.57 $\pm$ 5.48%	284.80 $\pm$ 4.80%	456.70 $\pm$ 4.96%	391.32 $\pm$ 5.10%
2 HP	688.13 $\pm$ 4.90%	508.12 $\pm$ 3.88%	779.12 $\pm$ 4.23%	731.46 $\pm$ 3.89%
3 HP	979.44 $\pm$ 4.42%	817.18 $\pm$3.72%	1076.89 $\pm$ 3.87%	991.88 $\pm$ 3.86%

VII. TEST RESULTS

The test results discussed above accomplished the objective of this study whereby the single phase heat transfer coefficient and the condensation heat transfer coefficient are obtained. The literature search failed to identify a condensation correlation for the current Turbo-C tubes which makes it difficult to compare and verify our results. Hence the comparisons are made with the plain annulus result which will be a base case to illustrate the effect of the 3-D Integral Fin Surface Enhancement. Equation (6) is used to obtain the single phase heat transfer coefficient from experimental results for superheated vapour.

$$\overline{h}_{SH} = \left[\left(\frac{A_i \Delta T_{(LMTD)SH}}{Q_{SH}} \right) - \left(A_O R_W \right) - \left(\frac{A_O}{A_i \, \overline{h}_i} \right) \right]^{-1} \quad (6)$$

When the resulting heat transfer coefficient obtained from the experiment was plotted against the changes in Re, Pr and *k* (due to the change in properties of refrigerant for each individual data run), a coherent curve was obtained. The results of the single phase heat transfer coefficients for superheated vapour, for enhanced and plain annuli, are illustrated in figure 6. Both tests show a consistent trend, as expected, where the heat transfer coefficient increases with increase of mass flux.

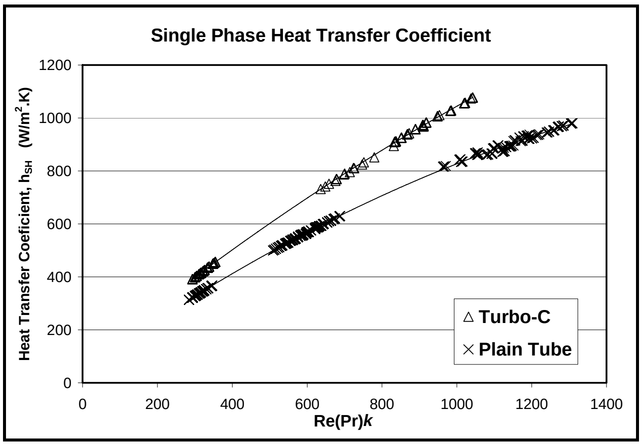

Figure 6. Comparison of single phase heat transfer coefficient

The 3-Dimensional integral fin present for the Turbo-C annulus acts as an extended surface. The higher heat transfer coefficient for superheated vapour, when compared to plain annulus is attributed to the effective surface area. This condition explains the trend illustrated in figure 6 which may be summarized as:

a) The Turbo-C tube outperforms the plain tube by about 25%-30% for superheated vapour.
b) The single phase heat transfer coefficient increases with increasing mass flux.

VIII. Correlations for Plain and Enhanced Annuli

This experimental work was conducted to develop design correlations so that the results of the study could be directly used by the HVAC community for design and development of double tube condenser systems. The first step in this effort was comparison of the current data with available correlations. The correlations available in the literature range from purely theoretical ones to those purely empirical in nature. All fluid properties required in the data reduction and the correlations were evaluated using property tables in ASHRAE Handbook of Fundamentals 2001 [7]. All subsequent analysis of correlations is given in non-dimensional form. Comprehensive reviews of literature led to selection of a few correlations which best represent the annulus geometries and flow characteristics of the fluids in the test section. The selected correlation was compared with experimental work on plain tube. This is done to assess the applicability of the selected correlations. The one that most accurately represents the range and conditions of interest was used as the starting point. These correlations give the basic representation of the average heat transfer coefficients for a given set of parametric conditions. Next, it was assumed that the presence of the fluid inlet and exit fittings (both refrigerant and cooling water) and surface enhancement did not alter the form of the heat transfer coefficient substantially and that any difference present can be handled by adjusting the empirical constants.

Table 5.1 Individual correlations for test conditions

Smooth annulus [10]
$\left(\overline{Nu}\right)=\frac{\left(\overline{h}\right)D_h}{k}=0.020\left(\mathrm{Re}_v\right)^{0.8}\left(\mathrm{Pr}\right)^{1/3}\left(\frac{D_i}{d_O}\right)^{0.53}$
for $\mathrm{Re}_v => 10{,}000$ (7)
Enhanced annulus [11]
$\left(\overline{Nu}\right)=\frac{\left(\overline{h}\right)D_h}{k}=0.04\left(\mathrm{Re}_v\right)^{0.87}\left(\mathrm{Pr}\right)^{0.4}\left(\frac{s}{D_h}\right)^{0.4}\left(\frac{e}{D_h}\right)^{-0.19}$
for $\mathrm{Re}_v => 10{,}000$ (8)

Considering the current case of condensation of pure vapour (R-22) flowing on a horizontal water-cooled tube, this process can be divided into three flow regions, namely, superheated vapour, condensation and subcooled liquid. Correlation development can generally lead to a deeper understanding of the heat transfer phenomenon that occurs in the annulus side. However, it is optimistic to expect such explanation in the present case in which one fluid is investigated for one diameter ratio. The correlation for forced convection in tubes is given in the form of:

$$Nu=\frac{h\ d}{k}=C(\mathrm{Re}^m)\,\mathrm{Pr}^{1/3} \tag{9}$$

This common correlation due to Sieder-Tate is a simple approach which reveals the physical aspect of the experimental theory, but this is not the form of correlation which gives the best result. Another available correlation that has a similar form gives a lower scatter:

$$Nu=\frac{h\ d}{k}=C(\mathrm{Re}^m)\,\mathrm{Pr}^n \tag{10}$$

where m is a function of Pr. Considering the conditions of our test, it is safe to assume that the changes in the Pr value of the superheated vapour and subcooled liquid of the condensation process have a minor influence on the way Re affects the Nu correlation. Hence if the index m, as yet to be found, is constant, a simple linear regression will reveal the value of C and m in Eq. (9). In the Dry Run Test for superheated vapour flow, the compressed vapour enters the test section in superheated state and exits at the same phase. The heat transferred to the cooling water is kept sufficiently low to allow the vapour exit temperature to be above the saturation temperature and to avoid phase change

$$Q_{SH}=\dot{m}_O(\Delta h)_{SH}=\overline{h}_{SH}A_O(LMTD) \tag{11}$$

$$\overline{h}_{SH}=\frac{\dot{m}_O(\Delta h)_{SH}}{A_O(LMTD)} \tag{12}$$

$$\overline{Nu_{SH}}=\frac{\overline{h_{SH}}d_h}{k}=C(\mathrm{Re}^m)\,\mathrm{Pr}^n \tag{13}$$

$$\ln\left(\overline{Nu_{SH}}\right)=\ln(C)+m\ln(\mathrm{Re})+\left(\frac{1}{n}\right)\ln(\mathrm{Pr}) \tag{14}$$

By plotting ln (Nu) against ln (Re) for the results of Dry Run Test for both plain tube and Turbo-C, the single phase heat transfer coefficient for superheated vapour and subcooled liquid was obtained. Figure 7 shows the relationship between heat transfer coefficient and superheated vapour flow in the annulus of plain and enhanced annulus in non-dimensional numbers. Considering the Reynolds Number for the test results is >10000, the flow in the plain annulus is taken as turbulent region. Equation (7) was used to evaluate the Nusselt type correlation which is given by Monrad and Pelton [10]. Thus, the single phase heat transfer coefficient for superheated vapour for the plain annulus is:

$$\left(\overline{Nu}\right)_{\substack{PLAIN\ TUBE\\ VAPOUR}}=0.0146\left(\mathrm{Re}\right)^{0.8293}\left(\mathrm{Pr}\right)^{1/3}\left(\frac{D_i}{d_O}\right)^{0.53} \tag{15}$$

Equation (8) was used to evaluate the Nusselt type correlation which is given by Knudsen and Katz [11]. Thus, the single phase heat transfer coefficient for superheated vapour flow for enhanced annulus is:

$$\left(\overline{Nu}\right)_{\substack{TURBO-C\\ VAPOUR}}=0.0248\left(\mathrm{Re}\right)^{0.8412}\left(\mathrm{Pr}\right)^{0.4}\left(\frac{s}{D_h}\right)^{0.4}\left(\frac{e}{D_h}\right)^{-0.19} \tag{16}$$

Comparison of experimental Nusselt value using (15) and (16) against predicted Nusselt value is illustrated in figure 8.

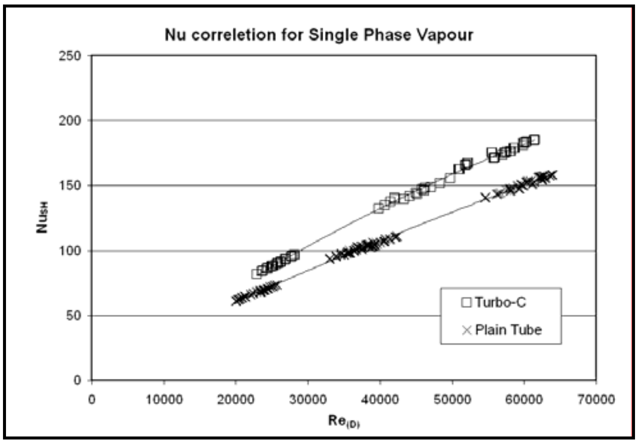

Figure 7. Comparison of single phase heat transfer coefficient non-dimensional number

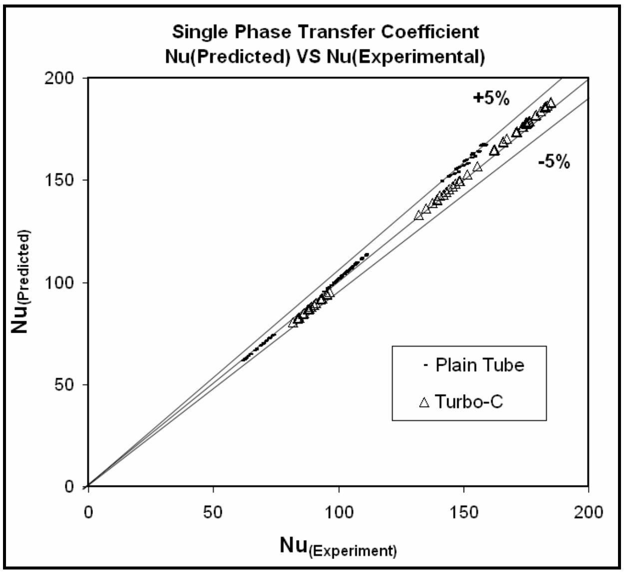

Figure 8. Comparison of experimental Nusselt against number predicted Nusselt number for superheated vapour

IX. Overview Remarks

The overall objective of the present study was to develop the single phase heat transfer coefficient for superheated vapour. The correlation by Monrad and Pelton [10] was used for plain annuli. For heat transfer in enhanced annuli, the correlation by Knudsen and Katz [11] determined the basic form. The new empirical constants resulted in good prediction for turbulent superheated vapour flow in the annulus for both plain annulus and enhanced annulus. On examining the accuracy of the single phase heat transfer correlation proposed, all predicted data is within the $\pm$5% of experimental value. The superheated vapour flow of plain annulus has an absolute deviation of $\pm$3.12% and $\pm$2.52% respectively. Similar results are observed for the superheated vapour and subcooled liquid flow for enhanced annulus with +3.91% and +2.46% receptively.

Acknowledgment

The authors would like to acknowledge with appreciation, the help and support from OYL R&D CENTER, in providing all facilities for this research work.

References

[1] Briggs, D.E., Young, E.H., "Modified Wilson plot techniques for obtaining heat transfer correlations for shell and tube heat exchangers," *Chemical Engineering Progress Symposium Series*, No. 92, Vol. 65, 1969, pp. 35-45.

[2] Deans, J., Sinderman, A., Morrison, J.N.A., "Use of the Wilson plot method to design and commission a condensation heat transfer test facility," *Two-Phase Flow Modelling and Experimentation*, Edizioni ETS, 1999.

[3] Kline, S., and McClintok, F., "Describing uncertainties in single-sample experiments," Mechanical Engineering, Vol. 75, Jan 1953, pp. 3-8.

[4] Kamei, A., Beyerlein, S.W., and Jacobsen, R.T., "Application of nonlinear regression in the development of a wide range formulation for HCFC-22," *International Journal of Thermophysics*, Vol. 16, No. 5, 1995, pp. 1155-1164.

[5] Klein, S.A., McLinden, M.O., "An improved extended corresponding states method for estimation of viscosity of pure refrigerants and mixtures," *International Journal of Refrigeration*, Vol. 20, No. 3, 1997, pp. 208-217.

[6] Mclinden, M.O., Klein, S.A., Perkins, R.A., "An improved extended corresponding states model of thermal conductivity refrigerants and refrigerant mixtures," *International Journal of Refrigeration*, Vol. 20, No. 3, 1997, pp. 208-217.

[7] ASHRAE Fundamentals Handbook, Appendix E Thermophysics Properties, 2001.

[8] Wagner, W., and Pru*B*, A., "New international formulation for the thermodymanic properties of ordinary water substance for general and scientific use," *Journal of Physical and Chemical Reference Data*, Vol. 31, No. 2, 2002.

[9] Kestin, J., Sengers, J.V., Kamgar-Parsi, B., and Levelt Sengers, J.M.H., "Thermophysical properties of fluid H_2O," *Journal of Physical and Chemical Reference Data*, Vol. 13, No. 1, 1984.

[10] Monrad, C.C., and Pelton, *J.F., Transection AIChE*, Vol. 38, 1942, pp. 593.

[11] Knudsen, J.G., and Katz, D.L., *Chemical Engineering Progress*, Vol. 46, 1950, pp. 490.

Nomenclature

A	area (m^2)
c	Coefficient of the heat transfer correlation
d	diameter (m)
h	convective heat transfer coefficient (W/m^2K)
Δh	latent heat of vaporization (J/kg)
k	thermal conductivity (W/mK)
Pr	Prandtl Number
Q	rate of heat transfer (W)
R	thermal resistance (K/W)
Re	Reynolds Number
T	temperature (K)
U	overall heat transfer coefficient (W/m^2K)
ΔT	change in temperature (K)
ρ	density (kg/m^3)
μ	viscosity (kg/ms)
LMTD	Log mean temperature difference
$\dot{m}$	mass flow rate (kg/hr)

Subscripts:

c	coolant
cw	coolant wall
i	internal
l	liquid
o	outside
s	tube surface
sat	saturation
SH	superheated vapour
w	wall