Towards More Economical Methods of Investigating the Energetic and Heat Transfer Properties of New Refrigerants

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ABSTRACT

This paper describes experimental and modelling studies on a secondary refrigerant calorimeter, slightly modified in design but with more instrumentation than a standard calorimeter, having the objective of the simultaneous determination of average evaporative heat transfer coefficients and cycle COP. Results are given for two pure component refrigerants, R134a and R22, and two binary non-azeotropic mixtures of R22 and R142b in the proportions by mass 60:40 % for the first mixture and 40:60 % for the second mixture over a range operating conditions.

INTRODUCTION

Refrigeration two-phase flow and heat transfer research has been stimulated by the demand for accurate design methods and cycles that have an acceptable energy efficiency when new benign refrigerants, especially blends, are in use. Most blends (in particular non-azeotropic mixtures) exhibit heat transfer and thermodynamic behaviour which is considerably different from the single fluids they replace.

Accurate theoretical evaluation of the evaporative behaviour of refrigerants is hampered by the large uncertainties in heat transfer correlations used to describe the complex convective boiling process. A large number of methods for estimating heat transfer coefficients have been proposed in the literature, the majority of which lack generality and have an unknown accuracy beyond the bounds of the experimental data on which they are based. Many researchers, for instance; Rohlin (1994), Darabi et al., (1995) have reported poor agreement between their experimental measurements and some well known heat transfer correlations. In many cases, researchers have resorted to modifying an existing correlation to achieve a best fit to their data. Many surveys of currently available correlations for predicting heat transfer coefficients in boiling and condensation in pipes, for both pure fluids and mixtures, have been published in recent years; for example; Wang and Chato (1994), Stephan (1995), Darabi *et al.* (1995) and Thome (1996).

The thermodynamic properties of most new refrigerants can be determined with good accuracy, but predicting the actual behaviour in real systems (especially for the blends) still remains imprecise. In practical systems, the presence of lubricating oil, non-condensable gases and other unidentified impurities (all of which are difficult to incorporate into thermodynamic models) leads to a lack of agreement between theoretical results and actual measured behaviour. This necessitates a practical evaluation of the heat transfer behaviour. However, testing in special purpose equipment is laborious, time consuming and expensive, hence the need for more economic methods. The authors of this paper report their first attempts at developing such methods.

EXPERIMENTAL APPARATUS

The equipment has the same features as a basic vapour-compression refrigeration system, a schematic of which is shown in Figure 1. The equipment is identical in its essential features to that described by Pearson (1995), but has additional instrumentation for measuring wall and refrigerant temperatures in the evaporator and refrigerant mass flow. Refrigerant and adjacent wall temperatures are measured at regular intervals through the coil by T-type thermocouples and a 32 channel data logger. A controller/logger measures and records pressures and operates safety cut outs. Compressor shaft torque and speed are measured to close limits. The direct expansion (DX) coil which serves as the primary evaporator is wound as a single pass 220 mm diameter open coil consisting of sixteen turns of 15.8 mm o.d. (0.9144 wall thickness) copper tubing.

TEST FLUIDS AND OPERATING CONDITIONS

Two single fluids, R134a and R22, and two binary non-azeotropic mixtures of R22 and R142b in the proportions by mass 60:40 % for the first mixture (mix1) and 40:60 % for the second mixture (mix2) were tested. The tests for the heat transfer study were carried out at various evaporating temperatures ranging from minus 15°C to plus 2.5°C, all at a condensing temperature of 40°C. The vessel temperature was varied from 0°C to 10°C to give vessel-refrigerant temperature differences ,ΔT_{VR}, of 7.5, 10, 12.5 and 15K . The range of mass flow rates is estimated to be 4 g/s to 15.8 g/s. When the mass flow was measured, the estimate was found to be within 5%.

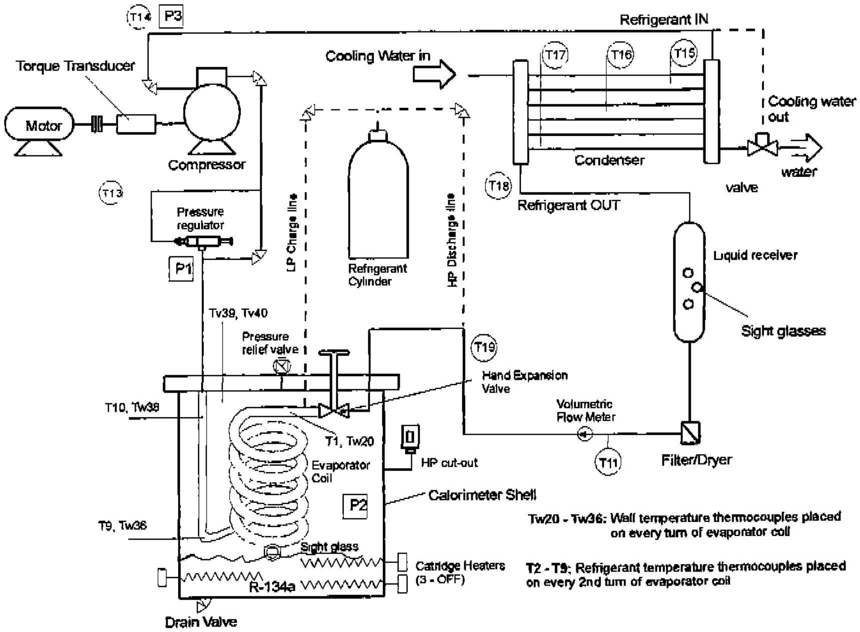

Figure 1: Schematic of test rig

MATHEMATICAL MODEL OF THE EVAPORATOR

Figure 2 exemplifies the actual temperature profiles for a pure fluid (R134a) and a non-azeotropic blend (R22/R142b 40/60 %wt). The complexity of non-azeotropic mixture temperature profiles and inherent difficulty in deducing the onset of dry-out is apparent.

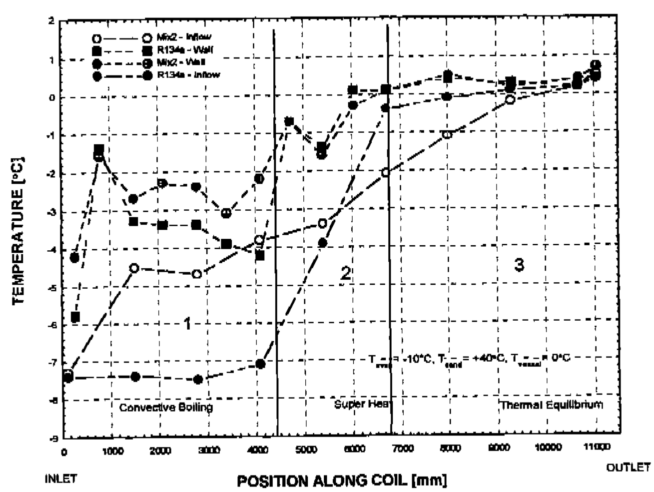

Figure 2: Actual temperature profiles for R134a and Mix2; Evap temp = -10°C , Vessel temp = 0°C Cond temp = 40°C , $\dot{m}_{R134a}$ = 7.55 g/s ; $\dot{m}_{Mix2}$ = 5.51 g/s.

Three distinct heat transfer regions for the pure fluid may be identified from the temperature profiles; the two-phase forced convection boiling region, the superheat region, and the thermal equilibrium region. Determination of the magnitude of the boiling region 1 (Figure 2) and its associated heat transfer is the essence of any endeavour to simulate a DX evaporator. The quantity of heat absorbed in superheating the refrigerant to the outlet temperature is small in comparison with the heat transferred in boiling (usually less than 5% of the total refrigerating capacity).

The heat transfer problem consists of two parts; condensation of the heat transfer media on the outside of the tube and heating of the refrigerant inside the tube; boiling in the two-phase region and superheating in the dried out region. The model developed to simulate these processes is intended to be useful for dealing with refrigeration design, for instance, comparative performance of benign refrigerants, optimisation of refrigerant blend composition and evaporator size for any given set of operating conditions.

Convective Boiling Region

It is assumed that the two-phase flow is steady, incompressible, uniform, one-dimensional and homogeneous. Further, the skin friction force imposed by the high velocity vapour phase on the liquid phase and the variation of kinetic and potential energy along the path is neglected.

Energy Balance

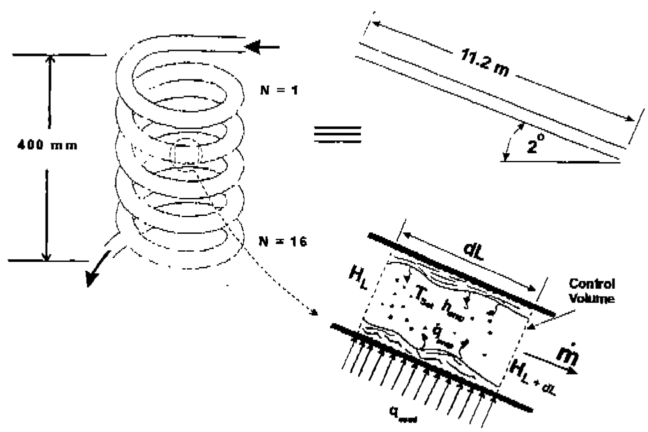

Figure 3: Simplified representation of evaporator showing control volume of fluid in forced convective boiling.

For the control volume shown in Figure 3, an energy balance on an element of length dL gives the following result:

$$dL = \frac{Gd_i(1+\Omega)(H_{fg}dx + (1-x)\,Cp_l\;dT_{sat} + Cp_v dT_{sat})}{4h_{evap}\Omega(T_\infty - T_{sat})} \qquad \textbf{(1)}$$

where $\Omega = \dfrac{d_o}{d_i}\dfrac{h_\infty(L)}{h_{evap}(L)}$; T_∞ is the temperature inside the shell, $h_\infty(L)$ is the local condensation heat transfer coefficient, H_{fg} is the enthalpy of vaporisation [kJ/kg] and G is the mass flux [kg/m^{2}s].

By integrating equation 1 from the known inlet quality until complete evaporation is achieved, i.e., at $x \approx 1$, the length of tube in which evaporation occurs can be obtained. However, integration of equation 1 is not a straight forward task due to the complex dependence of the heat transfer coefficient on quality, heat flux, flow regime and tube dimensions. A review of existing correlations shows that a great many of them have been developed for simple geometric configurations (vertical and horizontal tubes) and constant heat flux and/or wall temperature conditions. Such idealised conditions are clearly not obtainable in the vast majority of practical applications including this one. Consequently, application of most heat transfer correlations to the present case is only possible given certain modelling postulations, such as, neglecting the influence of the lubricating oil.

Heat transfer coefficients for all the refrigerants are calculated using the Jung and Radermacher (1989) correlation

Condensation Heat Transfer of a Single Fluid

The heat transfer on the outside of the coiled evaporator tube can be modelled by making use of the relationship for the heat transfer coefficient derived from Nusselt's film condensation theory. A detailed discussion of this derivation can be found in the open literature, for example, Collier and Thome (1994).

From Nusselt's analysis, the heat transfer coefficient, $h_{\infty 1}$, for condensation on a single, inclined horizontal tubes in stagnant vapour is given by:

$$h_{\infty 1} = 0.725\left[\frac{k_l^3 \rho_g g \cos\Theta\,(\rho_l - \rho_v) H_{fg}}{d_o \mu_l (T_\infty - T_w)}\right]^{\frac{1}{4}} \tag{2}$$

where Θ is the angle of inclination to the horizontal [°], g is acceleration due to gravity [m/s], k is thermal conductivity [w/m^{2}k], ρ is density [kg/m^3] and μ is viscosity [Ns/m^2].

The heat transfer coefficient, $h_{\infty N}$, for the N^{th} tube can be estimated from Kern's equations (Collier, 1994) as:

$$h_{\infty N} = h_{\infty 1}\left(N^{\frac{5}{6}} - (N-1)^{\frac{5}{6}}\right) \tag{3}$$

Computational Scheme

The temperature and quality of the refrigerant at inlet to the evaporator, the mass flow rate and the refrigerating effect are estimated from an ideal cycle simulation program. Thermodynamic properties are computed using REFPROP Version 5.10 (NIST, 1996). Following recommendations by the originators of the REFPROP subroutines, pure refrigerant properties were calculated using the Modified Bennedict-Webb-Rubbin (MBWR) equation of state while the Extended Corresponding States (ECS) model was used for the estimation of mixture properties

The unknown boiling length is divided into a number of divisions of unknown, unequal lengths, but with the same heat absorbed in each section. The iterative procedure uses a succession of wall temperatures to calculate the condensation and convective boiling heat transfer coefficients. Convergence is achieved when the condensation heat transfer equals the evaporative heat transfer.

RESULTS

Measured temperatures in the evaporator coil were consistently greater than saturation temperatures calculated from the pressure measured during experiments (typically 2 – 2.5K for the single component refrigerants and 4 – 5K for the R22-R142b blends). The higher saturation temperatures are probably due to the unwanted but difficult to avoid presence of air and water in the system. The presence of lubricating oil, the non-existence of thermal equilibrium at any cross section of the tube caused by partial wetting of the inner tube walls (especially severe when flow is stratified), and the turbulent and vigorous nature of the boiling process may also contribute. However, the refrigerating capacity computed from a cycle simulation program using the measured mass flow rate gave refrigerating capacities that were in excellent agreement with the measured values.

Experimental and Predicted Heat Transfer Coefficients

The average heat transfer coefficients are obtained by first estimating the boiling length and the average temperature differences driving condensation and convective boiling from the refrigerant and wall temperature profiles (e.g., Figure 2). The average heat flux for the section is then calculated using the estimated boiling area and the heat energy required to evaporate the refrigerant to the dry saturated condition. This procedure entails a degree of uncertainty, especially in the case of the blends. Figure 4 shows the average heat transfer coefficients for three of the refrigerants at an effective vessel – refrigerant temperature difference, ΔT_{VR}, (obtained from theoretical considerations) of 10K. The blends (mix1 and mix2) are seen to give poorer heat transfer coefficients compared to R134a.

Figure 4 also shows the heat transfer coefficients predicted using the evaporator model. The predicted heat transfer coefficients are lower than the measured values. This is mostly due to the fact that the evaporator model assumes that the evaporation occurs at the saturation temperature corresponding to the measured pressure. Therefore, the calculated difference between the tube wall and the boiling refrigerant temperature will be greater than that measured, hence the lower heat transfer coefficients.

A summary of the test conditions is given in Table1.

Table 1: Summary of tests at ΔT_{VR} = 10K

T_e [°C]	Δ_{VR} [°C]	R134a	R22	Mix1 R22/142b (60/40)	Mix2 R22/142b (40/60)
		$\dot{m}$ [g/s]	$\dot{m}$ [g/s]	$\dot{m}$ [g/s]	$\dot{m}$ [g/s]
-10	10	7.55	10.77	7.18	5.51
-5	10	10.01	13.69	8.99	7.24
-2.5	10	11.26	14.35	10.32	7.98
0	10	12.44	15.27	11.42	8.97

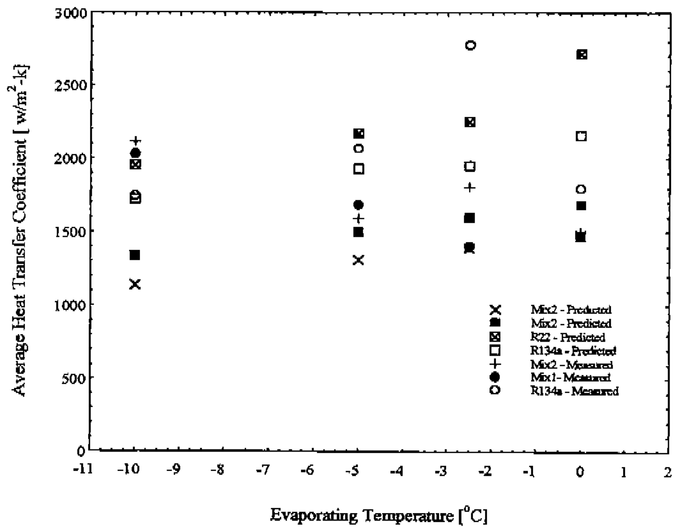

Figure 4: Average experimental heat transfer coefficients and predicted heat transfer coefficients at a vessel-refrigerant temperature difference of 10K.

Boiling Length - Measurements and Theoretical Results Compared

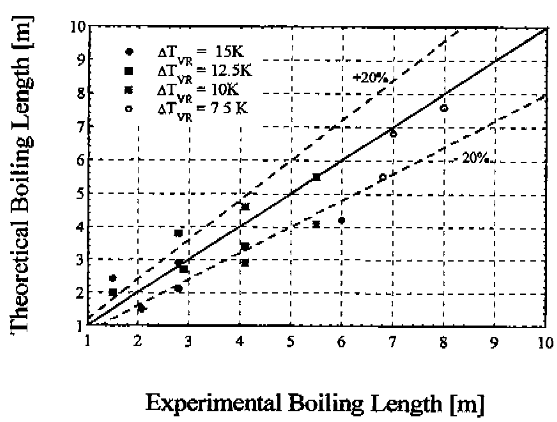

Figure 5: R134a – Predicted and measured boiling lengths compared

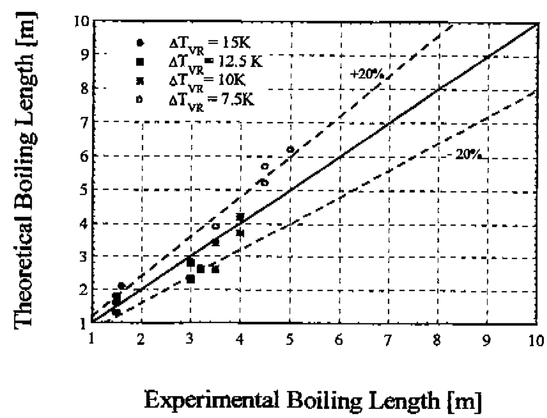

Figure 6: Mix2 – Predicted and measured boiling lengths compared

Figures 5 and 6 show the results of boiling length calculations for R134a and Mix2, respectively. In the majority of cases, the model under-predicts boiling length because the calculations are performed at greater temperatures differences than the measured values. The fact that lubricating oil was not included in the computations also contributes to the scatter observed in the predicted lengths. However, the majority of the predicted boiling lengths are within ±20% of the measured values.

CONCLUSION

The capability of the present test facility to simultaneous determine the heat transfer and energetic properties of refrigerants has been demonstrated. The experimental results show that the R22-R142b mixtures have lower refrigerating capacity and poorer heat transfer performance. However, the mixtures were seen to have a higher COP than R22. R134a had a better COP than all the refrigerants.

Results from the modelling study are encouraging considering the uncertainties in both condensation and convective boiling heat transfer correlations and, also, the fact the influence of the lubricating oil was not included in the model.

Future Work

There are numerous opportunities for further research using the present test equipment. It would be interesting to repeat the tests and measure the oil concentration in the liquid refrigerant. Tests with oil free refrigerants would also be useful in studying the effect of lubricating oil on the Pressure-Temperature-Volume behaviour and on heat transfer coefficients. Further, improvements in instrumentation and mathematical modelling with a view to increasing the accuracy of heat transfer measurements and theoretical calculations, respectively, are planned.

NOMENCLATURE

C_p	Specific heat capacity at constant pressure	[kJ/kg.K]	h	Heat transfer coefficient	[W/m^2.K]
d_i	Internal diameter of tube	[m]	L	Length	[m]
d_o	Outer diameter of tube	[m]	T	Saturation temperature	[°C]
H	Specific enthalpy	[kJ/kg]	x	Mass dryness fraction	[-]

Subscripts

cond	Condensation	∞	Ambient condition
evap	Evaporation	**sat**	Saturation
L	Liquid	**w**	Wall
v	Vapour phase		

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