Experimental Investigation of Condensation Heat Transfer Coefficient of Refrigerants during in Tube Condensation

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Introduction

Heat transfer performance of a refrigerant is an important aspect of the design of new refrigeration systems or retrofit of existing ones. Though there are accurate and sufficient heat transfer performance data and correlations for established and well-known refrigerants, data for alternative fluids are not yet well established or available in public domain. This situation makes experimental investigation of heat transfer performance of alternative refrigerants important. This paper briefly describes the design and fabrication of a test rig to find the condensation heat transfer coefficients of mixture refrigerants in plain tubes in the presence of lubricant, and presents initial results obtained for R134a. The results are compared with theoretical values obtained using correlations to demonstrate the validity of the measurements so that the procedure could be extended to refrigerant mixtures.

Heat exchangers of the test rig were initially designed using selected correlations for common refrigerants. Figure 1 presents a schematic of the test rig fabricated as a part of the research, and available at the Applied Thermodynamics Laboratory of the University of Peradeniya. The test rig includes an open type compressor with provision to vary the refrigerant volume flow rate by varying the flow speed. It also includes a desuperheater, condenser, pre heater and evaporator. There are three separate fluid circuits or loops; refrigerant, cooling water and heating water, in the apparatus. The pre heater provides a measurable thermal load to the evaporator for the steady running for the system. Heat given up by condensing the refrigerant is measured using the temperature rise of a known flow rate of cooling water.

Test rig design, assembly and running

The design of the rig includes sizing the heat exchangers, compressor, expansion valve and refrigerant pipes. Instrumentation of the test rig is another important aspect where sensing elements for pressure and temperature of the refrigerant have to be embedded into the rig accurately, particularly to capture the parameters during phase changing flow.

Sizing the evaporator and the condenser was based on the log mean temperature difference (LMTD) heat exchanger design method. Theoretical values of convective heat transfer coefficient of the refrigerant were obtained using Dittus and Boelter (1985) and Shah (1979) correlations. A counter flow configuration is used in both heat exchangers and they are arranged as concentric tube (tube in tube) heat exchangers and thermally insulated.

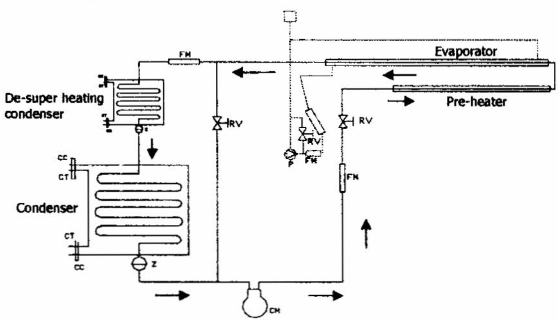

Figure 1. Schematic diagram of the test rig

The refrigerant flow rate was varied by varying the speed of the compressor. During testing, the evaporator was loaded using a stream of hot water generated with an electric heater. During the test the inlet refrigerant to the main condenser must not be in a superheated state. Therefore, the flow rate of water and refrigerant should be adjusted as necessary to achieve this. The flow rate could be measured to a precision of ± 0.0001 m^3 and temperature to a precision of ± 0.1 °C. The refrigerant pressure and temperature at the inlet and outlet of the condenser section is measured with pressure transducers and thermocouples, respectively.

The thermo-physical properties of refrigerants were obtained from a refrigerant property database which uses all the recent correlations and mixing rules to estimate refrigerant properties (McLinden *et al.*, 1998).

Data reduction

In-tube average heat transfer coefficient, $h_{r\text{-}av}$, is calculated from the expression for overall heat transfer coefficient U as given in Eq (1).

$$\frac{1}{h_{r-av}} = \frac{\pi}{K} - \frac{1}{h_w d_{out}} - \frac{1}{2\pi} \ln \frac{d_{out}}{d_{in}} \qquad (1)$$

Where, K is the thermal conductivity and, d_{in} and d_{out} represent inner and outer diameters of the tube respectively.

The waterside heat transfer coefficient, h_W, is obtained using Dittus-Boelter correlation (Dittus and Boelter, 1985). The overall heat transfer coefficient per unit length of the test section can be estimated using Eq (2), where *C* and *m* are specific heat and mass flow rate, *L* is the length of the test section, ΔT_m is the log mean temperature difference, ant T_w is the water temperature.

$$U = \frac{C_W m_w (T_{w1} - T_{w2})}{L \Delta T_m} \qquad (2)$$

Results and discussion

Table 1 presents some of the initial results obtained with refrigerant R134a, after commissioning the test rig. These give an idea of the order of magnitude of the difference between average in-tube condensation convective heat transfer coefficients obtained from measurements under practical conditions and the corresponding theoretical values for similar flow conditions. Compared to the theoretical values, the results of the test rig are on average about 7% to 8% smaller.

Invariably there are certain aspects of the experiment where the ideal conditions stipulated in the development of correlations are difficult to achieve. The existence of pressure drops, certain unavoidable heat losses, variation of flow patterns (compared to those for which the correlations are developed) and certain assumptions used in data reduction such as negligible wall thicknesses etc contributes to differences in the experimental and theoretical values.

Conclusions

The test rig was able to produce results for R134a to better than 10% accuracy referred to predictions based on the correlations. For the purpose of engineering design, accuracy of this order of magnitude is acceptable for vapour compression systems so that the experiment can be extended to other refrigerants to obtain in-tube convective heat transfer coefficients during condensation using the same test rig under practical conditions encountered in refrigeration system operation.

Table 1. Comparison of results obtained from test rig with theoretical estimates

Trial	Refrigerant flow rate kg/s x10^3	Convective heat transfer coefficient (W/m^2K)	
		Experimental[1]	Theoretical[2]
1	4.225	267.7	304.5
2	4.630	301.3	327.1
3	8.712	510.0	521.0

1 - Obtained using the test rig given in Figure 1

2 - Obtained using correlation in Ref [1] and [2]

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