

Experimental Measurements of Condensation Heat Transfer Coefficients for Refrigerant HFO-1234yf

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ABSTRACT

Experimental tests were conducted on a parallel flow condenser with HFO-1234yf as the working fluid on an AC system bench to determine average and local heat transfer coefficients during condensation of HFO-1234yf for mass flow rates that are typically encountered from idle to highway speeds (800 to 3000 rpms). A condenser from MY 2008 medium sized sedan was used for this investigation. All original OEM parts were used with the alternate refrigerant. Same TXV set-point was used with HFO-1234yf. The magnitude of the measured heat transfer coefficient for condensation were found to be 8~12% lower in comparison to HFC-134a. The magnitudes of the pressure drop during condensation were of the same magnitude as HFC-134a system. The information from this investigation can be used to in the design of condensers for mobile air conditioning systems with HFO-1234yf as the working fluid.

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INTRODUCTION

Refrigerant Hydro Fluoro Olefin (HFO)-1234yf is currently being considered as an alternate replacement for HFC-134a for mobile air conditioning system (MACS) to address the global warming concerns. This refrigerant has been developed jointly by DuPont and Honeywell. These two companies have been doing basic fundamental studies (Koban, 2009; Spatz, et. al, 2008; Figure 1) on this refrigerant. Based on recent information from 2012 SAE's Thermal Management System Symposium (TMSS) meeting at Scottsdale, AZ, DuPont has set-up a plant in China and is ready to start production. Over the last 3 years, a number of OEMs, suppliers (Meyer, 2008) have investigated HFO-1234yf refrigerant for automotive application. SAE has also conducted tests with this refrigerant for MACS through SAE's cooperative research project (CRP).

CalsonicKansei has also been actively working in this area. The author had conducted the following experimental investigations with HFO-1234yf as the working fluid:

- i. AC system performance as drop-in (Mathur, 2010a)
- ii. AC performance with charge optimization and TXV set point (Mathur, 2010a)
- iii. AC system performance enhancements with a suction line heat exchanger (Mathur, 2011a)
- iv. Evaporator performance (Mathur, 2011b)

v. Condenser performance (Mathur, 2010b)

vi. Experimental measurements of evaporator heat transfer coefficients (Mathur, 2012)

This study reported in this paper represents continuation of investigation at CalsonicKansei on HFO-1234yf as the working fluid for MACS. The current investigation focuses on experimental measurement of condensation heat transfer coefficient in a parallel flow condenser by using HFO-1234yf as the working fluid. An AC system with a parallel flow condenser from MY 2008 medium sized sedan was used for this investigation. To the best of author's knowledge, this is the first paper in open literature that presents experimentally measured average and local heat transfer coefficients during condensation for HFO-1234yf.

PRINCIPLE OF OPERATION OF A PARALLEL FLOW CONDENSER

Schematic of an AC loop is shown in Figure 2. The function of the condenser is to condense the superheated refrigerant vapors coming out of the compressor to subcooled liquid at the exit of the condenser. A photograph of a typical parallel flow condenser is shown in Figure 3 - This condenser was used for the current investigation. A parallel flow condenser consists of flat tubes (Sugihara, 1990; Mathur, 1999) that have multiple flow ports or channels. The refrigerant is supplied directly to the tubes through the header. Louvered fins are used for automotive heat

exchangers. A typical refrigerant tube is 2 mm in thickness with widths ranging from 14-22 mm with 6 to 12 flow ports or channels. Refrigerant tubes with widths ranging from 10-14 mm and thickness less than 2mm are also being used today in the industry. Currently extruded and folded tubes designs are being used for condensers for MACS. Folded tubes are being used to reduce the overall manufacturing costs; to reduce weight of the condenser; and to reduce system refrigerant charge. The flat tubes have less projected frontal area to the air stream that results in lower air-side pressure drop. The performance of a parallel flow condenser is superior in comparison to serpentine condensers (Mathur, 1998) as the refrigerant is distributed in multiple tubes. Also, the refrigerant pressure drop in parallel flow condenser is much smaller in comparison to a serpentine condenser due to the same reason.

Condenser performance is dependent on the following design parameters: tube and fin design, tube and fin pitch, height, length and depth of the tube and fin, number of tubes in a given pass; number of pass; materials of components; and location of the inlet and outlet connections. A good condenser design should have relatively small areas for desuperheating and subcooling process, and hence, the area for condensation should be maximized (Mathur 1997, 1998, 1999, 2001).

Further details on the design and performance information on parallel flow condensers have been presented by the authors in an SAE paper (Mathur, 2012).

CURRENT INVESTIGATION

Condensation heat transfer coefficient (Incropera, 1981) is defined by the following equation:

$$h_{c, av} = Q / [A_{cond} (T_{sat, av} - T_{av wall})] \quad (1)$$

The above parameter, condensation heat transfer coefficient is essentially the average tube side heat transfer coefficient. For the above equation it is assumed that the refrigerant vapor is in saturated state, i.e., the thermodynamic quality is 1 at condenser inlet; and vapor is completely condensed at the outlet, i.e., thermodynamic quality is 0 at the condenser outlet.

To determine, local condensing heat transfer coefficients we can use [equation 1](#) by replacing local properties for wall temperature and saturation temperatures and by using appropriate local heat transfer rates in small elements. It should be noted that the local heat transfer coefficients (Mathur, 1999) are determined by using local thermodynamic & thermo-physical properties, flow pattern, pressure gradient, oil circulation ratio and void fraction.

[Table 1](#) shows the design details of the condenser that is used for this study. The size of the condenser finned area is as follows: H= 407.5mm; L= 670 mm; T=16mm. The condenser has a total of 42 tubes with two passes. The first pass has 35 tubes and the second one has a total of 7 tubes. The second pass is for subcooling. For this experimental investigation,

only the first pass was used - see [Figure 4](#). The pressure and temperature was measured at the inlet and outlet of the first pass and the center tube was instrumented to determine the wall surface temperatures. The center tube is tube 18th in 35 tubes. The tube was instrumented to measure the surface temperatures ([Figure 4](#)) at 10%, 30%, 50%, 70% and 90% from the inlet of the condenser. As stated above saturated conditions were maintained the condenser inlet and no sub-cooling was allowed at the tube outlet. The following section presents the details of the experiments.

EXPERIMENTAL TESTS

Tests were conducted in the system bench test facility. The test facility is instrumented in accordance with the ASHRAE standard (ASHRAE, 1997; [ASHRAE Standard 42.1, 1987](#); [ASHRAE Standard 37, 1988](#)). The facility is capable of measuring the performance of the system using air-enthalpy and refrigerant-enthalpy methods. Per ASHRAE standard, the data from airside should be within $\pm 6\%$ of the refrigerant data. The actual measured data was within $\pm 3\%$. The test facility is fully instrumented to record the operating parameters, e.g., pressure and temperatures of the refrigerant, and dry and wet bulb temperatures at the strategic points. This test facility consists of two rooms (See [Figure 5](#)) - one simulating the vehicle cabin and the other room simulating the engine compartment ([Figure 6](#)). The HVAC unit is placed in the room simulating the vehicle cabin while the condenser and compressor are placed in the room simulating the vehicles engine compartment. All parts are assembled in these two rooms based on the vehicle configuration. [Figure 7](#) shows the walls between the interior and engine compartment room that simulates the firewall. [Figure 8](#) shows the control panel in the engine compartment. All cables (voltage, rpm, thermo couples, etc) come to this control panel. All these cables are attached to the controls as shown in [Figure 5](#). A similar control panel is installed in the room simulating the cabin interior. Evaporator freeze is controlled by cycling the compressor via a thermister located in the evaporator.

The airflow to the HVAC unit is supplied through an air machine that is designed based on ASHRAE standard (ASHRAE, 1997; [ASHRAE Standard 42.1, 1987](#); [ASHRAE Standard 37, 1988](#)). Cabin interior is maintained at 25°C and 50% RH; and engine compartment room temperature is varied from 25 to 45°C. The compressor rpm is varied from 800 to 3000 rpm; the condenser face velocity is varied from 2 m/s to 10 m/s; and the evaporator airflow is varied from 5, 7 and 9 m³/min (83 to 150 lps). These three evaporator airflows were selected for the investigation that will provide different cooling loads. This affects the total heat rejection in the condenser. Two condenser test conditions were tested - one that simulated idle condition (2 m/s) and the other that simulated 22 mph (35kph) vehicle speed (10 m/s). Condenser performance beyond 10 m/s becomes independent of the condenser face velocity or vehicle speed. [Table 2](#) shows the details of the test parameters.

It should be noted here that on the system bench, the airflow over the condenser is uniform (Figure 6). In a typical vehicle due to the presence of the bumper, the airflow over the condenser is highly non-uniform that negatively affects the condenser heat rejection rate.

Same lubricant (PAG) as base case was used with HFO-1234yf. Both systems used 140cc of PAG oil. The refrigerant was used as “drop-in” refrigerant in the existing system. This resulted an oil circulation ratio (OCR) that varied between 2.3~3.5% based on ASHRAE standard 41.4 (1996). All original OEM parts were used with the alternate refrigerant. Same TXV set-point was used with HFO-1234yf. REFPROP program was used to determine the properties of HFO-1234yf. An add-on file for REFPROP computer program for refrigerant HFO-1234yf was provided by Honeywell & DuPont. It should be noted here that tests are ongoing with lubricating oils that are compatible with HFO-1234yf.

For these tests, the discharge hose was cooled via a wet cool blanket to de-super heat the superheated refrigerant vapor coming out of the compressor. The saturated conditions at the condenser inlet were confirmed by measuring the condenser inlet pressure (hence, the saturation temperature) and the bulk fluid temperature of the refrigerant vapor at the inlet. The condenser heat rejection rate was precisely controlled to ensure no sub-cooling at the exit of the condenser. Also, for some cases, the compressor rpm was very slightly varied to give the above conditions. For example, for a compressor rpm of 800, the actual rpm was varied ± 10 rpm to provide the desired test conditions. However, this variation was within the uncertainty of the sensor that measure compressor rpm.

Pressure and temperatures are measured around the loop to determine the performance of the AC system. The pressure transducers have an accuracy of $\pm 0.25\%$ over the full scale. This corresponds to $\pm 0.09 \text{ kg/cm}^2$ for the high side pressures for the AC loop. Thermocouples with a measurement accuracy of $\pm 1.0 \text{ }^\circ\text{C}$ were used to measure the air and refrigerant temperatures. This resulted in an experimental measurement uncertainty of $\pm 8.5\%$ for the condensation heat transfer coefficients.

RESULTS AND DISCUSSIONS

Figures 9,10,11,12,13,14 show the performance of the condensation heat transfer coefficients and pressure gradients for HFO-1234yf as the working fluids. These tests have been conducted with the cabin maintained at 25°C with a relative humidity of 50% and evaporator airflow rate of $9 \text{ m}^3/\text{min}$. For these graphs, the points are the experimental data points and the curve is the best fit 3rd order polynomial curve.

Average Condensation Heat Transfer Coefficients

Figure 9 shows the average condensing heat transfer coefficient at an average condensing saturation temperature

of 50°C with the sink temperature of 25°C . The condensing heat transfer coefficient is plotted as a function of refrigerant mass flow rate. The magnitude of the average condensation coefficient increase with the refrigerant mass flow rate due to an increase in the forced convection. HFC-134a shows similar trend (Mathur, 2011c) - however, the magnitudes for HFC-134a are higher by 8~12% in comparison to HFO-1234yf. Figure 10 shows the average condensing heat transfer coefficient at an average condensing saturation temperature of 58.5°C with a sink temperature of 35°C . The magnitudes of the condensing heat transfer at this condition are higher and follow the same trend as in Figure 9.

Local Condensation Heat Transfer Coefficients

Figure 11 shows the local condensing heat transfer coefficient at a sink temperature of 25°C and compressor rpm of 800. The local condensing heat transfer coefficient is plotted as a function of the distance from the inlet of the condenser. The magnitude of the local condensation coefficient decrease with distance from the inlet. This is due to the fact that more vapors are condensed in the flow direction. With more vapor condensing on the downstream from inlet the dry surface area is no longer available for refrigerant vapor to condense on. Ultimately a liquid film develops inside of the tube that further reduced local heat transfer coefficient. From theory, the condensing heat transfer coefficient is a function of liquid thermal conductivity and the film thickness ($h=k/t$). Based on this equation, condensation heat transfer coefficient will decrease with the increase of the film thickness. Conversely, the condensation heat transfer coefficient should be equal to infinity when the film thickness is “zero”. This case is true at the condenser inlet where the entire inner surface area is available for condensation. Figure 12 shows same data at a compressor rpm of 3000. As is obvious, at this condition, the local condensing heat transfer coefficients increase. From Figure 12 it is evident that the local heat transfer coefficient tends to follow the equation from theory ($h=k/t$) - the heat transfer coefficient approaches infinity at condenser inlet. However, the data from Figure 11 does not suggest this trend. This explains the differences between theory and practice. Also, point to be noted here is that the data was curve fitted that resulted in a maximum R square (R^2).

Pressure Drop During Condensation

Figure 13 shows the condenser refrigerant pressure drop as a function of refrigerant mass flow rate during condensation at 50°C and a sink temperature of 25°C . The pressure drop increase exponentially as a function of refrigerant mass flow rate. The magnitudes and trends for HFO-1234yf are similar to HFC-134a (Mathur, 2011c). This is due to two factors: (i) Latent heat for HFO-1234yf (see figure 1) is lower by approximately 22~23% in comparison to HFC-134a. Hence, for the same cooling capacity, the mass

flow rate for HFO-1234yf should be larger by the same factor resulting in higher average velocities for both phases; (ii) the liquid viscosity for HFO-1234yf is lower by approx. 28%; vapor viscosity is higher by 4% in comparison to HFC-134a. Hence, the overall resultant is approximately same magnitude of the pressure drops for both refrigerants. Figure 14 shows similar data for a condensation temperature of 58.5°C.

Note that the tests were conducted with HFO-1234yf and the baseline lubricant (PAG). Hence, changing lubricant can have a significant impact on the heat transfer coefficient (Mathur, 2012; Zeng & Webb, 1997). Hence, the author has planned to conduct these tests again with lubricants that are compatible with HFO-1234yf.

CONCLUSIONS

An experimental investigation has been carried out to experimentally determine condensation heat transfer coefficients with HFO-1234yf as the working fluid. System level bench tests are conducted with an AC system from a 2008 MY mid-sized sedan. Ambient conditions were varied for the tests from 25°C to 45°C by maintaining the cabin interior air temperature at 25°C and 50% relative humidity. The compressor operating condition was varied from 800 to 3000 rpm and the condenser face air velocity was varied from 2 to 10 m/s. Average and local condensing heat transfer coefficients have been presented in the paper. Based on the current investigation, the following are the main conclusions:

- The experimental heat transfer coefficient data for HFO-1234yf data shows similar trend as HFC-134a over the range of mass flow rate investigated. These are typical mass flow rates that are encountered in an automotive application.
- The magnitudes of the condensing heat transfer coefficients are lower by 8~12% for HFO-1234yf in comparison to HFC-134a
- Experimentally measured local condensation heat transfer coefficients for HFO-1234yf are presented as a function of distance from the tube inlet
- The pressure drop during condensation for HFO-1234yf are of similar magnitudes as HFC-134a
- The data can be used in the detailed design of condensers for HFO-1234yf system
- This is the first investigation in the open literature that deals with experimental measurements of condensation heat transfer coefficients for HFO-1234yf for automotive applications

FUTURE WORK

Simulation work has been planned to model the thermal and hydrodynamic performance with HFO-1234yf as the working fluid to accurately determine the performance under dynamic operation. Also experimentally investigate the maldistribution of refrigerant flow in the condenser tubes.

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