

Experimental Investigation with Cross Fluted Double-Pipe Suction Line Heat Exchanger to Enhance AC System Performance

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ABSTRACT

An experimental investigation has been carried out to quantify the performance enhancements with a suction line heat exchanger (SLHX) in an AC system. An off-the-shelf double pipe cross fluted SLHX is used for this investigation. System level bench tests are conducted with an AC system from a 2008 MY mid-sized sedan. The cabin interior condition is held constant at 25°C and 50% RH. The dry bulb temperature for the engine compartment is varied from 25 to 45°C. The compressor speed is varied from 800 to 3000 rpm and the air velocity over the condenser is varied from 2 to 10 m/s. Based on the tests conducted on the AC system without and with SLHX, system performance (COP) has been improved by 7%. Additional tests have been planned with modified SLHX.

INTRODUCTION

The performance of the AC system can be improved by adding a suction line heat exchanger (SLHX) into the system. This will have a direct influence on the thermal comfort for the occupant along with fuel economy and exhaust emissions. An SLHX in an AC loop provides the following benefits (ASHRAE, 2006): (i) Increasing system performance, (ii) Sub-cooling liquid refrigerant to prevent flash gas formation at inlets to expansion valve, (iii) fully evaporating any residual liquid that may remain in the suction line prior to reaching the compressor.

A number of papers are available in the open literature on enhancing AC system performance using the suction line heat exchangers. Majority of the data is available for the commercial and industrial refrigeration applications (ASHRAE, 2006). Domanski et. al. (1994) and Klien et. al. (2000) presented the use of internal heat exchanger (IHx) to improve the system performance for the refrigeration systems. Some publications are available for the mobile air conditioning systems. Bowe et. al. (1999, 2001) presented the use of a SLHX with R744 as the working fluid. Preissner et. al. (2001) and Zhang et. al. (2002) presented the use of SLHX for R-134a systems. In a recent paper (Kurata et. al., 2007), used a double pipe internal heat exchanger for the front AC system that improved the cooling performance by 6~12% over the base case; and the COP by 12~15% over the baseline AC system. The inner diameter of the pipe was 19.1mm and the heat exchanger was 0.4m in length. This resulting capacity of the heat exchanger was approx 650W. The following section describes the working principle of a suction line heat exchanger.

PRINCIPLE OF OPERATION OF A SUCTION LINE HEAT EXCHANGER IN AN AC LOOP

Figure 1 shows a schematic (Stoecker, 1982) of the baseline AC system that does not have an SLHX. Figure 2 shows (Stoecker, 1982) the modified system that contains an SLHX to improve the system performance. The coefficient of performance (COP) for these two systems is given (Wark, 1981) by the following equations:

$$\text{COP}_{\text{Baseline}} = (h_1 - h_4) / (h_2 - h_1) \quad (1)$$

$$\text{COP}_{\text{SLHX}} = (h_5 - h_6) / (h_2 - h_1) \quad (2)$$

$$\text{COP}_{\text{Carnot}} = T_L / (T_H - T_L) \quad (3)$$

As is clear from these two figures and equations 1 and 2, the refrigeration effect for the second system is larger than the baseline system. The refrigeration effect represents the amount of cooling under the process 4 to 1 for baseline and 5 to 6 for the system with the SLHX. With the use of an SLHX in the system, we have increased the cooling area by moving the thermodynamic state (TXV inlet) left of the saturation liquid line (i.e., more sub-cooling) and moving the thermodynamic state right (evaporator outlet) of the saturation vapor curve (i.e., increasing super heat).

It should be noted here that by increasing sub-cooling in the condenser, we are increasing the area in the heat exchanger (condenser) that is occupied by single-phase liquid flow. The magnitude of heat transfer coefficients is much lower for single-phase liquid flows in comparison (Mathur, 1998, 1999, 2001) to the condensation process. Similarly, by increasing the superheat at the evaporator outlet, we are increasing the area in the heat exchanger (evaporator) that is occupied by single-phase vapor flow. The magnitude of heat transfer coefficients are very low (Mathur, 1997, 1998, 2000) for the single-phase vapor flow in comparison to two-phase flow boiling. Hence, increasing sub cooling and superheat will typically result in a reduction in the overall performance of the heat exchanger. Therefore, the degree of sub-cooling and super heating must be optimized that results in an optimum performance for the heat exchangers and system.

Maximum achievable theoretical COP of a reversible Carnot cycle is given by equation 3. In this equation, T_L represents the temperature of the cooled space and T_H represents the ambient temperature. Based on this equation, the maximum COPs can be computed at ambient temperatures of 25, 35, and 45°C; with a fixed cabin temperature of 25°C. The calculated COPs are ∞ at 25°C; 30.8 at 35°C; and 15.9 at 45°C. At 25°C ambient temperature the COP is infinity as the denominator of equation 3 becomes 0 ($T_H - T_L$).

The effectiveness of the suction line heat exchanger can be determined by using the following equation:

Vapor Side:

$$\begin{aligned} \epsilon_{\text{vapor}} &= (\Delta T_{\text{suction side}}) / (\Delta T_{\text{Max}}) \\ &= (\Delta T_{\text{suction side}}) / (T_{\text{Liquid inlet}} - T_{\text{Suction inlet}}) \end{aligned} \quad (4)$$

Liquid Side:

$$\begin{aligned} \epsilon_{\text{Liquid}} &= (\Delta T_{\text{Liquid side}}) / (\Delta T_{\text{Max}}) \\ &= (\Delta T_{\text{Liquid side}}) / (T_{\text{Liquid inlet}} - T_{\text{Suction inlet}}) \end{aligned} \quad (5)$$

The above two parameters are essential in determining the influence of an SLHX on the system performance. These numbers can be used to help make a decision if the thermodynamic process has been sufficiently altered

to increase refrigeration effect to improve system COP by increasing liquid subcooling at condenser outlet and vapor superheat at evaporator outlet for a given design of SLHX.

Following sections describes the experimental set-up and test details.

CURRENT INVESTIGATION

An experimental investigation has been carried out to determine the influence of SLHX on the system performance. A double pipe cross fluted SLHX suited for the application is used for this investigation. This SLHX is an off-the shelf product (Doucette Industries, 2005) that was purchased from the market. Cross-fluted pipe enhances the heat transfer coefficient that helps in performance enhancements. The refrigerant suction side pipe diameter is 1/2" (12.7mm) and the refrigerant liquid flows through the annulus of equivalent pipe diameter of 1/4" (6.35mm). The length of the SLHX is 11.5" (292.1mm). This heat exchanger is made from copper. System level bench tests are conducted with an AC system from a 2008 MY mid-sized sedan. Both SLHX and the liquid lines were installed in the AC loop such that the SLHX was by-passed for the baseline tests (See Figures 8 and 9). For tests with the SLHX, the vapor and liquid refrigerant connections were made to the SLHX.

Tests were first conducted with the baseline system, i.e., without the SLHX. Tests were then repeated by maintaining each test condition by installing an SLHX in the AC loop. Details of the experiments are presented in the next section. Figure 3 shows the details of the SLHX used for this investigation. Figure 4 shows a picture of the cross section of the SLHX.

EXPERIMENTAL TESTS

Tests were conducted in the system bench test facility. The test facility is instrumented in accordance with the ASHRAE standard (ASHRAE, 1997). The facility is capable of measuring the performance of the system using air-enthalpy and refrigerant-enthalpy methods. 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. The various operating parameters are fed into the data acquisition system for data reduction. This test facility consists of two rooms (See Figure 5) – one simulating the vehicle cabin (say room A, Figure 6) and the other room simulating the engine compartment (say room B, Figure 7). 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 8 shows the SLHX installed in the AC loop and Figure 9 shows the SLHX and the liquid by-

pass line. Figure 10 shows the walls between the interior and engine compartment room that simulates the firewall. Figure 11 shows the control panel in the engine compartment. A similar control panel is installed in the room simulating the cabin interior.

The airflow to the HVAC unit is supplied through an air machine that is designed based on ASHRAE standard. Cabin interior (room A) 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 to 9 m³/min.

Pressure and temperatures are measured around the loop to determine the performance of the AC system. The experimental uncertainty of COP was of the order of $\pm 3\%$ from the air side measurements. The pressure transducers have an accuracy of $\pm 0.25\%$ over the full scale. This corresponds to ± 0.09 kg/cm² for the high side pressures for the AC loop. Thermocouples with a measurement accuracy of ± 1.0 °C were used to measure the air temperatures.

For these experiments, a flow meter was initially not used in the liquid line. Reason being, it would have taken a number of changes to the existing plumbing that could have affected the system performance due to pressure drop. However, a refrigerant flow meter is being added to the system to record the refrigerant mass flow rate for energy balance, and to determine the heat transfer rate for the SLHX. Refrigerant mass flow rates have been calculated based on the compressor characteristics (volumetric efficiency as a function of rpm) and operating condition.

RESULTS AND DISCUSSIONS

Figures 12 to 14 show the COP of the baseline and the SLHX installed in the AC system. These tests have been conducted with the cabin maintained at 25°C with a relative humidity of 50% and evaporator airflow rate of 9 m³/min. These figures show the data for the condenser air face velocity of 2 m/s.

Figure 12 shows the data for an ambient condition of 25°C. At this condition, the performance enhancement with a SLHX is approx 2~3%. This is due to the fact that the overall ΔT between the cabin and the outside is 0°C. As this temperature difference increases, the enhancement of the system performance increases. This is shown in Figure 13 that shows the data for an ambient temperature of 35°C. This data shows an enhancement of the COP of 6~7%. At higher ambient temperature of 45°C (Figure 14), the enhancement of the COP ranges from 4~5%. Figure 15 shows a cross plot for the COP for the baseline and with an SLHX as a function of compressor rpm and ambient temperature.

Figure 16 shows the performance enhancements of the above data. The lowest set of data is for an ambient temperature of 25°C. The data for 35°C ambient temperature is a maximum around 7%. The data at 45°C is lower than at 35°C as the increase in the compression work is more than the gain in refrigeration effect. Hence, the net effect is a lower COP. The data at 45°C for a compressor RPM of 1500 seems to be low. This is due to experimental measurement error. The arrow points where the data should have been. As can be seen from this graph, all three data sets (25, 35, and 45°C ambient temperatures) follow the same trend. At lower compressor speeds, the refrigerant mass flow rate is lower resulting in lower magnitudes of flow boiling (two-phase) heat transfer coefficients in the evaporator resulting in a lower performance (Mathur, 1998; 2000). Hence, the difference in the performance (SLHX – Baseline) is lower at lower compressor rpm. As the compressor speed is increased, the mass flow rate increases throughout the system resulting in an improved evaporator performance due to higher two-phase flow boiling heat transfer coefficients. However, at very high compressor rpm, the mass flow rate becomes very high and suction line pressure drop (or thermodynamic irreversibility) starts to affect the thermal performance due to very high vapor velocities. Figure 17 shows a cross-plot of the percentage COP enhancement as a function of the ambient temperature for different compressor operating conditions.

Figure 18 shows the test data for a given operating condition for the condenser face air velocity of 10 m/s for compressor operating condition of 1500 rpm. A higher airflow rate over the front end or over the condenser enhances condenser, and hence, the AC system performance. The performance at 10 m/s is higher in comparison to air velocity of 2m/s (Figures 12-14).

The experimental measurement uncertainty of COP is of the order of $\pm 3\%$. This magnitude is based on the instrumentation that is used for the test equipment. For the current investigation, the magnitude of the maximum COP enhancement was measured as +7%. All test data with the SLHX shows a positive improvement in the AC system performance (COP) and there is no scatter in the data (i.e., some data with positive enhancements and some with negative enhancements). Hence, we can safely conclude that the AC system under investigation with a SLHX will enhance the AC system performance in all operating conditions.

Figure 19 shows the effectiveness of the SLHX for the vapor side. At low compressor speeds, the effectiveness of the heat exchanger is higher as the refrigerant mass flow rate is lower. This results in a larger residence time for the refrigerant within the SLHX to exchange energy with the liquid side, consequently resulting in a larger effectiveness. However, at higher compressor speeds, the refrigerant flow rate becomes

very high resulting in a significant reduction of the residence time within the SLHX. This results in a lower overall temperature difference (out – in), thereby, decreasing the effectiveness (see equation 4). The magnitudes of the effectiveness for all operating conditions were in the range from 22 to approximately 30%. The test data at 45°C, 800 rpm seems to be low – this probably is due to measurement error. From this graph one can also see that the effectiveness at a given compressor rpm is nearly constant. Hence, if we need to increase the performance enhancement more than 7%, then we must increase the size of the SLHX (length and diameter).

Figure 20 shows the effectiveness of the SLHX for the liquid side. The data follows the same trend as for the vapor side. However, the magnitudes of the effectiveness for the liquid side are lower in comparison to the vapor side. Note that the minimum heat capacity between the hot (liquid) and cold (vapor) fluid is for the vapor side (Incropera & Dewitt, 1981). Reason being, the specific heat of liquid is more than the specific heat of vapor (since mass flow rate is the same, or, $m_L C_{p,L} = m_V C_{p,V}$). Hence, the minimum heat capacity would be for the vapor phase. Therefore, vapor phase will see a larger overall temperature difference across the SLHX in comparison to the liquid. Hence, the liquid overall temperature difference is lower resulting in lower effectiveness.

Discussions on the COP Enhancements: A COP improvement number of 7% is lower than the number reported by Kurata, 2007. One of the main reasons is that the SLHX used for this investigation is about 65% in length in comparison to the double pipe heat exchanger that Kurata had used. Also, Kurata's internal pipe diameter was larger (19.1mm) in comparison to the pipe diameter (12.7mm) used for this study. Hence, the SLHX used by Kurata is approximately two times in surface area compared to the SLHX used for this study. Also, we need to keep in mind that an off the shelf SLHX was used for this investigation.

Improvement in Fuel Consumption with SLHX: Improving AC system performance with a SLHX will have a positive effect on the vehicle fuel consumption. Based on the literature search, enhancement of the AC system performance (COP) will result in an improvement in the fuel consumption by 1~2%. This also depends on the drive cycle that is selected for conducting tests. The objective of this investigation was primarily determining the thermodynamic performance enhancement of an existing AC system with a SLHX. Hence, the fuel consumption investigation was not carried out. However based on the data available in the literature (e.g., Kurata, 2007), it is expected that the fuel consumption will be reduced by approximately 1~1.5%. This will also result in a reduction of emissions from the tailpipe.

CONCLUSIONS

An experimental investigation has been carried out with a cross fluted double-pipe suction line heat exchanger to enhance AC system performance. 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. Based on the test results, the AC system was improved by 7% with a SLHX over the baseline system.

FUTURE WORK

Tests are planned to increase the size of the SLHX that would provide a minimum of 10% performance enhancement over the baseline system. Additional AC performance tests have been planned by installing a flow meter in the AC loop for energy balance. Also, fuel consumption tests are also being planned.

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NOMENCLATURE

ϵ	- Effectiveness, %
COP	- Coefficient of performance, Dimensionless
h	- Enthalpy, kJ/kg
T	- Temperature, °K