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SUCTION LINE HEAT EXCHANGER FOR R134A AUTOMOTIVE AIR-CONDITIONING SYSTEM

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ABSTRACT

The performance of an R134a automotive prototype air-conditioning system was tested in the laboratory with and without an internal heat exchanger. At a higher condenser air temperature of 40°C and a restrictive idling air flow rate of 1.0 m/s, the COP and the capacity increased on the order of 5 to 10 % with a suction line heat exchanger with 60 % effectiveness. If the system is operated not at the optimum expansion device setting or it is undercharged, the improvement in system performance is even higher. When designing a suction line heat exchanger, the low side pressure drop is critical and must be minimized so as not to compensate the performance improvement.

INTRODUCTION

The performance of R134a automotive air-conditioning systems has been continuously improved over the last years. However, the potential benefits associated with the implementation of a suction line heat exchanger (SLHX) - also termed internal heat exchanger - have not been paid much attention according to the published literature. A SLHX transfers energy from the refrigerant leaving the condenser to the suction gas, resulting in a lower inlet enthalpy at the evaporator, providing a higher cooling capacity. The competing effects with respect to performance are a larger enthalpy change across the compressor and a lower mass flow rate, both effects caused by the lower suction density. An R134a prototype system was operated with and without a SLHX, the performance comparison is presented in this paper.

EXPERIMENTAL SETUP

The SLHX was tested in the R134a automotive air-conditioning test facility at the Center of Environmental Energy Engineering (CEEE). The evaporator is placed in a closed loop, and an air handling unit controls the specified temperature and humidity of the air stream entering the evaporator. The air inlet and outlet temperatures are measured with a grid of 9 thermocouples upstream and downstream of the evaporator, the humidity entering and leaving is determined by two dew point meters. In addition, the moisture condensing on the coil is collected in order to verify the dew point measurement. The air flow rate is varied with a variable speed fan, and is calculated from the pressure drop across a nozzle in the loop. The air flow rate measurement was calibrated with an electric heater prior to the beginning of the experiments. From the temperature and humidity differences, and the air flow rate, the cooling capacity is calculated. The heat loss from the experimental test section to the ambient is calibrated and incorporated in the capacity calculation. A differential pressure transducer measures the air side pressure drop across the evaporator.

The condenser is mounted in an open duct within a variable climate environmental chamber. The air flow rate, inlet and outlet air temperatures, and capacity are calibrated, measured, and calculated in the same manner as for the evaporator.

The schematic setup of the R134a loop is shown in Figure 1. The compressor is placed in the same climate chamber as the condenser, and consequently, is exposed to the condenser air inlet temperature (outdoor temperature). This condition is close to the arrangement in a vehicle, where the compressor is exposed to slightly higher temperatures in the engine compartment. The compressor is driven by an electric motor operated with a variable frequency drive. A non-contact torque meter is mounted between the motor and the compressor, and together with the measured compressor speed, the input power to the compressor is determined. The refrigerant leaving the compressor passes the condenser and a mass flow meter, and can then be routed through the suction line heat exchanger, or bypass it. The suction line heat exchanger transfers heat to the suction gas, providing more cooling capacity and improving the cycle COP. After passing the expansion device (manually or electronically controlled), the refrigerant evaporates in the evaporator and enters the accumulator. Liquid refrigerant and oil are stored in this device - a small amount of the oil-refrigerant mixture passes through a bleed hole at the bottom inside of the accumulator to the main refrigerant line to ensure proper compressor lubrication. The refrigerant loop is equipped with instream thermocouples and pressure transducers at the inlet and outlet of the main components.

Note that the system is neither a fixed orifice system nor a TXV (thermostatic expansion device) arrangement. The superheat was varied using charge and expansion valve settings, and the degree of subcooling resulted from the particular system test conditions and components. The operation with a receiver, which controls the high side pressure and the degree of subcooling is planned in future tests with a new prototype.

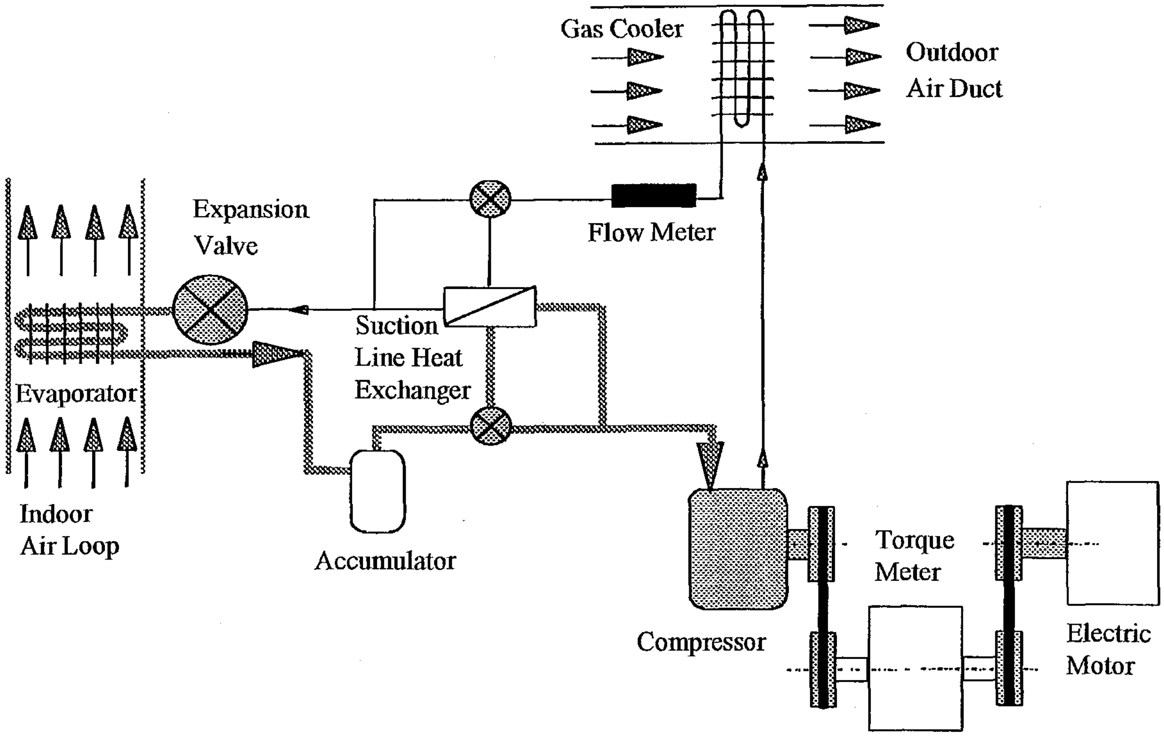


Figure 1: Schematic setup of the R134a cycle

COMPONENTS AND TEST CONDITIONS

A state-of-the-art piston compressor and custom made heat exchangers were employed in the R134a system. They are of comparable dimensions and performance when compared to current systems, the actual dimensions are not of primary interest in this study. The evaporator inlet temperature was set at 27°C with a relative humidity of 50 %, and the air flow rate was fixed at 580 m^3/h. The air velocity across the condenser was restricted to 1.0 m/s at a compressor speed of 1000 RPM, and increased to 2.5 m/s at 1800 RPM. The air inlet temperature was varied between 25 and 40°C (40 % humidity). These test conditions are summarized in Table 1.

Indoor			Outdoor			Compressor Speed [RPM]
Temperature [°C]	Relative Humidity [%]	Air Flow Rate [m^3/h]	Temperature [°C]	Relative Humidity [%]	Frontal Air Velocity [m/s]	
27	50	580	25 40	40	idling: 1.0 driving: 2.5	idling: 1000 driving: 1800

Table 1: Test conditions

RESULTS AND DISCUSSION

The remaining parameters to be adjusted during testing were the refrigerant charge and the opening of the expansion device. The expansion device was controlled so that the evaporation temperature was held constant at various values depending on the compressor speed. At 1000 RPM, the evaporation temperatures tested were 5°C, 8°C, and 12°C, where 8°C showed the best performance. At 1800 RPM, the capacity to be transferred at the evaporator is larger, therefore the evaporation temperatures tested were 2°C, 5°C, and 8°C, where 2°C showed the best performance. It is not possible to run lower temperatures as freezing of moisture on the evaporator coil must be avoided.

The performance of the system both with and without a suction line heat exchanger for a compressor speed of 1000 RPM and a condenser air inlet temperature of 40°C is shown in Figures 2 and 3. The data points to the left of the curves represent the performance of the system operated at undercharged conditions. The refrigerant leaves the condenser as a 2-phase fluid for higher evaporation temperatures (8°C), and with small amounts of subcooling at lower evaporation temperatures. The superheat at these operating conditions is on the order of 15 K, which is close to the air inlet temperature. The tests using a SLHX (with the same refrigerant charge) show a larger capacity and COP. Note that the system volume remained the same for tests with and without SLHX, as no system modifications were carried out, and only valve settings were changed.

The refrigerant charge was increased along the curves from left to right until the performance did not change significantly with increasing charge. For the higher evaporation temperature of 8°C, the charge was increased until the superheat was about 5 K, resulting in a subcooling at the condenser of about 20 K. At the lower evaporation temperature of 5°C, the superheat was about 10 K and the subcooling 30 K. Note that the degree of superheat and subcooling does not change significantly whether a SLHX is applied or not. For 8°C evaporation temperature, the capacity improved by 8 % and the COP by 10 %. At lower charge or lower evaporation temperatures (non-optimum operating points) - the benefit of the suction line heat exchanger can be even larger.

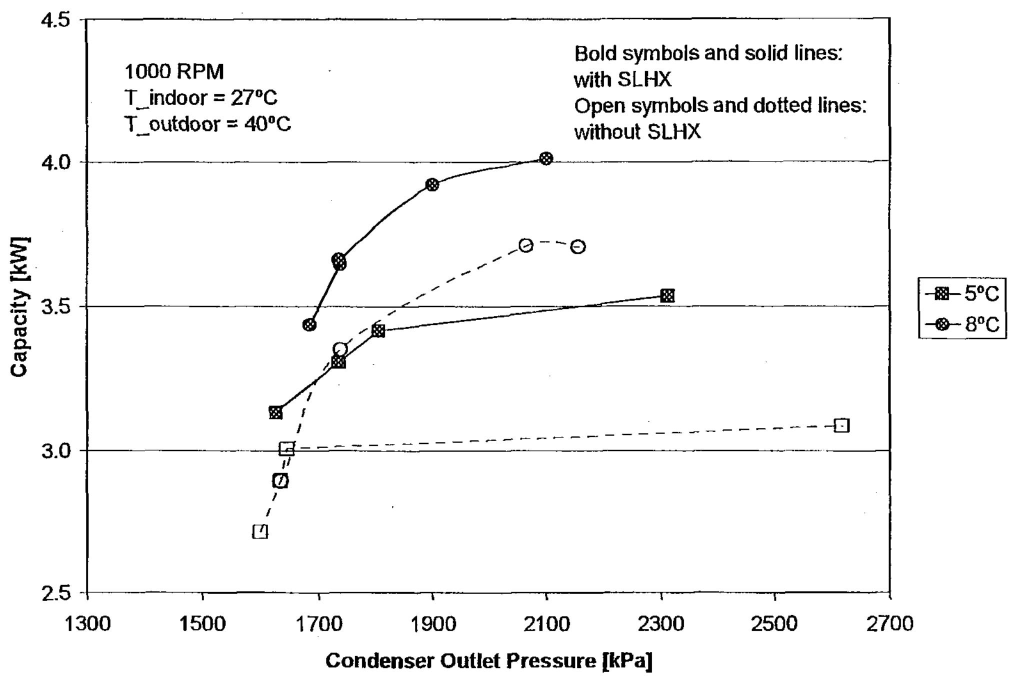
At a compressor speed of 1800 RPM, the dependence is similar, however the performance improvement with the suction line heat exchanger is only on the order of 5 %. At lower condenser air inlet temperatures of 25°C at 1000 RPM, the performance with and without SLHX was equivalent, however, at undercharged conditions and not optimum evaporation temperatures, the improvement was 0 to 5 %. At 25°C condenser air inlet temperature and 1800 RPM, no performance increase was noticeable.

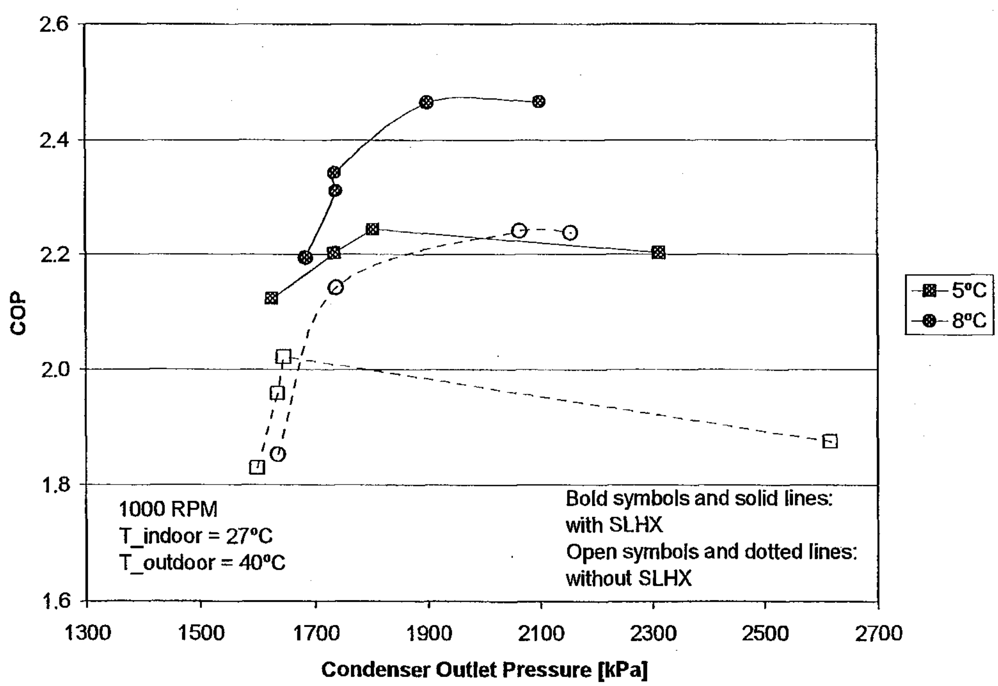
When designing a SLHX, the refrigerant pressure drop is a major parameter to consider. The high pressure side is not very crucial as the pressure drop in the subcooled region is low due to the high density of the liquid, and it is not difficult to design the heat exchanger with a low pressure drop. Moreover, the small high side pressure drop does not change the refrigerant temperature significantly. However it should be avoided that the refrigerant gets into a 2-phase state, as the pressure drop of 2-phase fluid in the liquid line gets significantly larger. On the suction side however, the pressure drop reduces the suction density resulting in a lower mass flow rate, a higher pressure ratio, both leading to a lower system performance.

The heat exchanger used here was designed carefully so that the low side pressure drop is negligible. It was below the measurement accuracy of 10 kPa in all operating conditions tested.

Due to the focus on the low pressure drop, the heat exchangers effectiveness ranged only from 55 to 65 %.

The system was tested with a second SLHX. At the mass flow rates encountered in the system, the low side pressure drop was between 40 and 70 kPa. Although having an effectiveness of 60 %, the pressure drop is too high and annihilates the performance benefit completely.





Figures 2 and 3: Cooling capacity and COP with and without SLHX

To further quantify the performance decrease associated with an additional pressure drop at the low pressure side, calculations were carried out with a program based on fitted cycle parameters from the experimental data set. For the same test conditions as the data presented in Figures 2 and 3, the penalty of the pressure drop on the low pressure side of the suction line heat exchanger is shown in Figure 4. A 50 kPa pressure drop reduces capacity and COP by about 10 %.

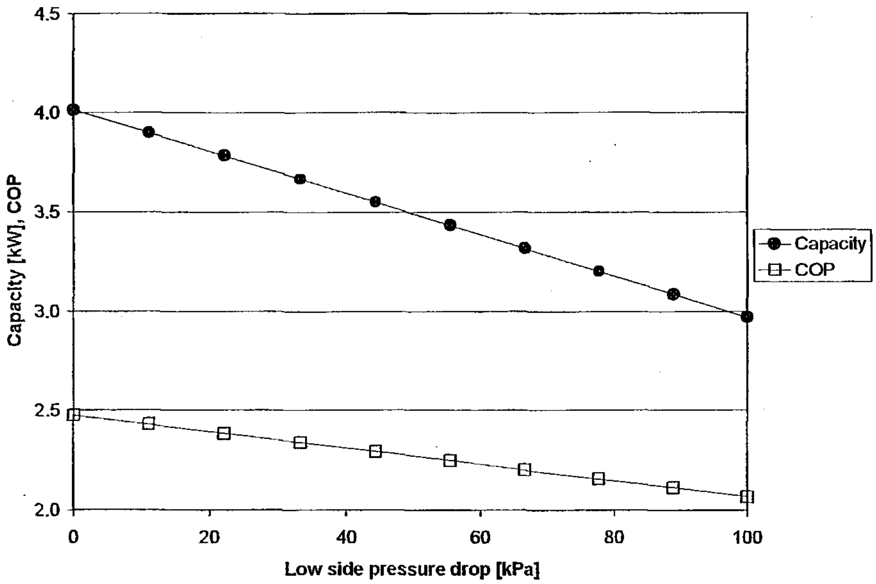


Figure 4: Performance penalty of pressure drop at the suction side

CONCLUSIONS

Depending on the operating conditions and design, a suction line heat exchanger can improve the performance of an automotive system. At higher air temperatures of 40°C at the condenser and with a low air velocity of 1.0 m/s, which typically occurs in idling conditions, a suction line heat exchanger can improve the capacity as well as the COP by about 5 to 10 %. At higher air velocities and lower temperatures, the benefit nearly vanishes. If the air-conditioning system is operated at either not at the optimum evaporator temperature or with insufficient charge (both of which can occur in a vehicle), the benefit of the suction line heat exchanger can be even larger.

After having demonstrated that the performance improvements of a suction line heat exchanger are not negligible, a further design with a higher efficiency is planned. However, the pressure drop at the low pressure side is the most important parameter to consider in a suction line heat exchanger design. With the new heat exchanger, a broader range of operating conditions is planned to be tested.

ACKNOWLEDGMENTS

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