

Internal Heat Exchanger System Integration for R1234yf Refrigerant

2011-01-0128

Published
04/12/2011

Lothar Seybold
Adam Opel AG - General Motors Company

William Hill
MACRAE, LLC

Jean-Jacques Robin
Otto Egelhof GmbH & Co

Copyright © 2011 SAE International

doi:10.4271/2011-01-0128

ABSTRACT

This paper will examine a mobile air conditioning (MAC) system optimized for efficiency as well as evaporator cooling capacity. Different internal heat exchanger (IHx) capacities and various thermostatic expansion valve (TXV) parameters will be applied using R1234yf refrigerant. Factors that will be considered include IHx heat transfer and pressure drop, TXV superheat setting and slope, the effect of oil in circulation and how these factors impact the efficiency and capacity of the MAC system. The paper describes the test facility used and the test procedures applied.

INTRODUCTION

The United States Environmental Protection Agency (EPA) and the National Highway Traffic Safety Administration (NHTSA) have finalized regulations to reduce greenhouse gas emissions and improve vehicle fuel efficiency in order to reduce global warming. Air conditioning related credits can be earned with the Light-Duty Vehicle Greenhouse Gas Emission Standards and Corporate Average Fuel Economy Standards by implementing an internal heat exchanger into a MAC system [1]. By integrating an IHx into a MAC system the Coefficient of Performance (COP) can be increased at the same rate of increased cooling capacity. This improvement in efficiency reduces the energy and fuel consumption of the air conditioning (A/C) system.

The IHx has been commercialized in vehicles with R134a refrigerant since 2006. This paper will focus on the integration of a coaxial tube heat exchanger applied to R1234yf refrigerant, which summarizes the work to complete a systematic investigation of different IHx capacities and various thermostatic expansion valve parameters. The tests were performed on a system bench with a production-intent MAC system. Cooling capacity, COP, and compressor discharge temperature have been investigated and optimized through different TXV parameters and IHx capacity. In addition the effects of varying oil in circulation in the A/C loop on cooling capacity, COP, and compressor discharge temperature have been studied.

THEORY

The application of an IHx in vehicles with R134a A/C system has been reported in various publications, e.g. [2]. Since R134a and R1234yf have different thermodynamic properties, the cooling capacity, COP and compressor discharge temperature are lower when R1234yf is “dropped into” an A/C system designed for R134a [3]. The evaporator capacity and COP losses can be eliminated with use of an IHx and optimization of the TXV parameters [4]. The IHx increases COP at equal evaporator capacity as the refrigerant mass flow can be reduced due to increased enthalpy change across the evaporator. The refrigerant density change across the evaporator is lower for R1234yf than for R134a, which has an impact on the refrigerant velocity, pressure drop and heat transfer. Therefore reducing the pressure losses within

the A/C system components, especially in the suction line, IHX, compressor, evaporator, and condenser impacts significantly system efficiency and fuel consumption.

The major effects of implementing an IHX in a typical vehicle refrigerant cycle will be discussed in the following sections.

IHX DESIGN

In general an ideal IHX has the highest possible heat transfer with the smallest suction pressure drop. A high suction pressure drop can counter the benefit of an IHX or in worse case can result in a negative impact to the A/C system. Criteria for an optimized internal heat exchanger as used in R134a and R1234yf systems were studied in detail in reference [5]. An optimized IHX design must include the following features:

- maximum heat transfer
- minimum suction pressure drop
- limited liquid pressure drop
- minimum liquid volume

To quantify the relation between suction pressure drop and heat transfer, the absolute efficiency ratio as a characteristic number for an IHX was defined in [5]:

$$\beta = \frac{\dot{Q}_{real}}{\sqrt{\Delta p_{LP}}} \quad (1)$$

$\dot{Q}_{real}$ is the transferred heat from the liquid to suction side and Δp_{LP} the suction pressure drop. IHX with optimized absolute efficiency ratio β shows an improvement up to 35% for R1234yf refrigerant. That study showed an IHX needs to be optimized for each refrigerant used in order to achieve an optimal COP and cooling capacity. A coaxial tube IHX profile optimized for R1234yf refrigerant was used for this study.



Figure 1. Schematic coaxial tube IHX cross section for R1234yf

TXV DESIGN

This paper studies the design criteria required in order to optimize the TXV parameters using R1234yf refrigerant along with an IHX MAC system. An optimized TXV design must include the following features:

- adiabatic throttling
- controlled superheat at evaporator outlet within the optimal range
- enable enough refrigerant mass flow during cool down to meet cooling capacity requirements

Initially the appropriate TXV capacity at maximum cooling performance has to be selected. The TXV capacity is a function of system parameters such as maximum refrigerant flow required in transient operation, superheat at evaporator outlet, subcooling at TXV inlet, condensing pressure, evaporating pressure, and valve throttling area (q). The throttling area is a function of throttle geometry and the valve stroke, while the valve stroke is a function of the spring stiffness (o), diaphragm design and material properties (m), bulb charge (n), and TXV setting and slope, see Fig 2.

The maximum stroke and valve throttling area were not limiting factors and therefore the basic design of the TXV was considered as fixed. The main parameters used to optimize the TXV performance were slope and setting.

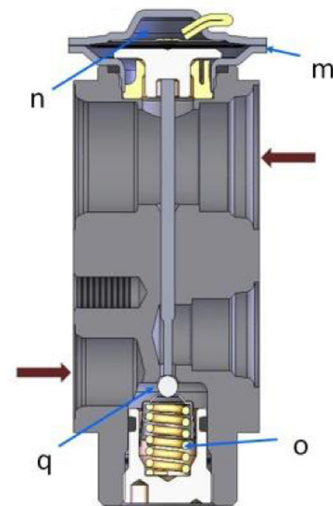


Figure 2. Schematic TXV cross section

EVAPORATOR DESIGN

The IHX provides enthalpy gain by increasing the subcooling of the refrigerant before the TXV. As a consequence of reducing specific enthalpy at the evaporator inlet, the specific enthalpy of the suction vapor will increase and therefore the compressor inlet and outlet temperatures will increase. Due to

the pressure and temperature properties of R1234yf, the compressor discharge temperature is lower for the same condensing pressure, which allows the use of higher IHX capacity as compared to R134a. The change in the evaporator inlet refrigerant quality might suggest that changes to the evaporator pass configuration are necessary to maintain a proper temperature distribution and reduce evaporator pressure drop to increase MAC system efficiency.

The theoretical benefit of additional subcooling before the evaporator at the same evaporating and condensing temperature for both refrigerants is given by equations 2 and 3. The absolute gain of the specific enthalpy Δh_{abs} is:

$$\Delta h_{abs} = h_{w/oSC} - h_{withSC} \quad (2)$$

Where $h_{w/oSC}$ is the enthalpy at evaporator inlet when no subcooling is available and where h_{withSC} represents the enthalpy at evaporator inlet when subcooling is available. The relative gain of specific enthalpy Δh_{rel} for the evaporator is:

$$\Delta h_{rel} = 100 \cdot \left(\frac{\Delta h_{withSC}}{\Delta h_{w/oSC}} - 1 \right) \quad (3)$$

Figure 3 and 4 illustrating the absolute and relative enthalpy gain for a subcooling range from 0 K to 20 K.

The absolute specific enthalpy gain is quite similar for both refrigerants. There is a slightly benefit for R134a. But the relative specific enthalpy gain in the evaporator is higher for R1234yf. Therefore the benefit of an IHX should be greater in the R1234yf MAC system compared to the R134a system.

The specific enthalpy difference in the evaporator at the same evaporating temperature for R134a and R1234yf is smaller for R1234yf and therefore, the mass flow must be increased to achieve the same evaporator capacity. Even if the density of R1234yf is higher than R134a, the volumetric cooling capacity of R1234yf is lower which requires a higher refrigerant flow too. The higher refrigerant flow causes higher pressure drop in the components of the A/C system and reduces system efficiency. This impact can be eliminated by introducing an IHX to increase the specific enthalpy across the evaporator.

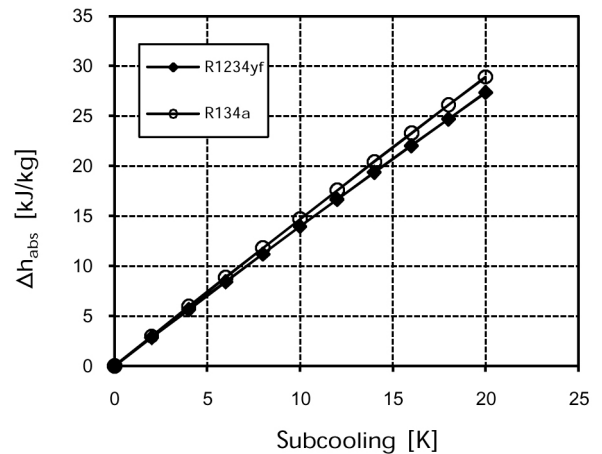


Figure 3. Absolute specific enthalpy gain at evaporator inlet vs. subcooling

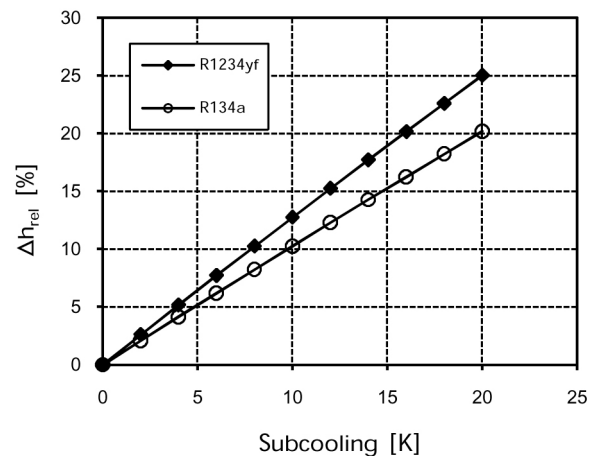


Figure 4. Relative specific enthalpy gain at evaporator inlet vs. subcooling

CONDENSER DESIGN

The higher compressor outlet temperature caused by an IHX increases the temperature difference between the ambient air and the refrigerant entering the condenser. The higher temperature difference can result in a higher condenser capacity and increased outlet air temperatures from the condenser. Changes to the condenser pass configuration can help to minimize this impact. The engine cooling capacity as well as the underhood temperatures should be reviewed as a result.

COMPRESSOR DESIGN

When the refrigerant superheat at compressor inlet becomes lower than approximately 5 K, the compressor volumetric efficiency is negatively affected. Liquid drops result in re-expansion in the compressor cylinder which reduces the volumetric efficiency. Through use of an IHX the liquid drops in the suction line will not evaporate in the compressor

cylinder but in the IHX and the volumetric efficiency of the compressor will be improved.

As the superheat at the compressor inlet is increased, the density of the suction gas is decreased resulting in reduced refrigerant mass flow for the same compressor displacement. The theoretical mass flow reduction as function of compressor inlet superheat is shown in Fig. 5:

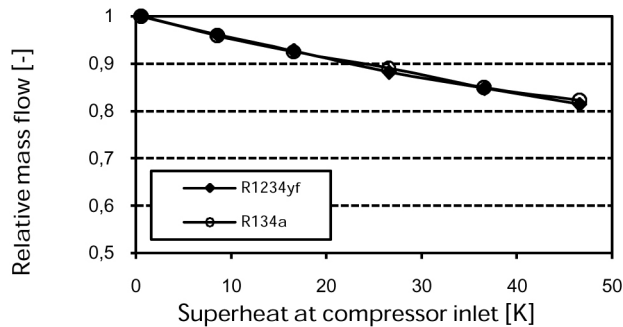


Figure 5. Theoretical impact of compressor superheat on the refrigerant mass flow at constant suction pressure

For both refrigerants the relative mass flow reduces in the same rate when superheat increases at compressor inlet. The internal losses in the compressor should also be increased as the density decreases, resulting in reduced volumetric efficiency in the compressor. The increased suction gas temperature also increases the bearing temperatures in the compressor, reducing the viscosity of the fluids in the area of the bearings and potentially reducing the losses as long the oil film is not lost due to these higher temperatures. As the absolute R1234yf mass flow is increased, the internal pressure losses in the compressor are higher and should be reviewed.

EXPERIMENTAL SETUP AND TEST METHOD

A/C SYSTEM DESCRIPTION

For this study an A/C system with production-intent components designed for a mid-sized vehicle was used. This A/C system was chosen since it represents a broad spectrum of MAC performance levels for many vehicles. The components of this A/C system are:

Compressor:

Variable displacement with 140ccm, externally controlled, 125 ml oil PAG ND 8, 120 mm pulley. The oil is not the optimized one for R1234yf.

Evaporator:

Optimized for R134a, flat-tube and fins, 250 mm × 210 mm × 50 mm installed in the production HVAC module with production blower, 2 × 3 passes with 10-7-10 tubes

Condenser:

Optimized for R134a, flat-tube and fins with integrated receiver dryer and integrated subcooling pass, 635 mm × 390 mm × 15 mm, 3 passes with 17-11-8 tubes and 1 pass for subcooling with 5 tubes

TXV:

Block design, 1.5 tons, cross or parallel charged

IHX:

Counter flow coaxial tube, 500 mm length (except for the investigation of the IHX length influence), 25mm outer diameter, the IHX profile is optimized for R1234yf refrigerant which means minimized suction pressure drop, minimized liquid volume, limited liquid pressure drop, and maximum heat transfer as discussed in [5]. The IHX routing has one 45 degree and one 20 degree bend.

TEST BENCH DESCRIPTION

To allow a wide range of test conditions and similar thermal conditions as in a vehicle occur, the test bench has two chambers with separate air conditioning units (h, i). The A/C components were positioned similar as installed into the vehicle. Even nearly the serial production plumbing was used. In the first chamber (a) the HVAC module (c) including TXV and in the second chamber (b) the A/C components of the vehicle engine compartment compressor (g), condenser (d), IHX and plumbing were installed, see Fig. 6a and 6b.

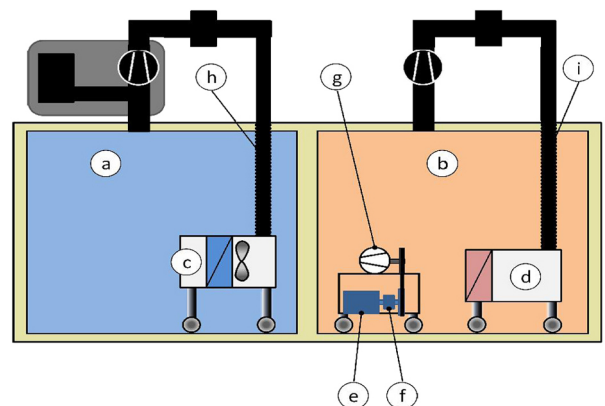


Figure 6a. Schematic of the test bench without IHX and plumbing

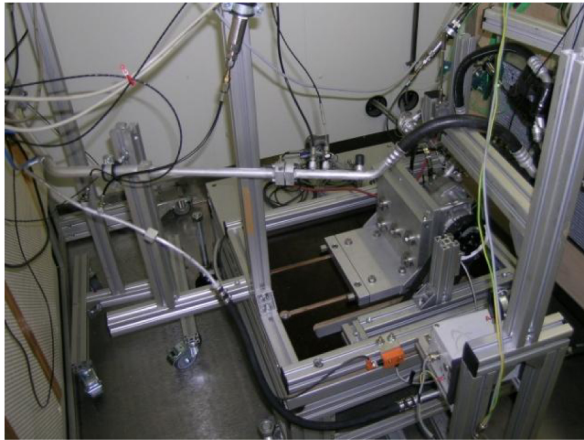


Figure 6b. Chamber (b) with the IHX

The compressor is mounted on the top of a support frame with the electric motor (e) under it to avoid the area surrounding the compressor being impacted by the driving motor. The torque meter (f) is placed on the drive motor shaft. The torque measurement includes the belt transmissions losses since this was not corrected. So, it can be expected that the COP values are somewhat underestimated. The accuracy of the torque measurements is ± 0.3 Nm. At 15°C ambient the torque is below 3 Nm and therefore the compressor power is inaccurate, which results in some uncertainties in COP calculation. Usually, the refrigerant mass flow rate is measured between condenser outlet and IHX inlet where the refrigerant is liquid. In case of low subcooling, an accurate measuring of the refrigerant mass flow rate becomes impossible. Therefore the Coriolis mass flow sensor was placed between compressor outlet and condenser inlet where the refrigerant is always superheated. In addition, this position keeps the refrigerant charge close to production system, since the added volume of the flow meter contains gas, not liquid refrigerant. The oil circulation rate (OCR) is calculated from the measured speed of sound, pressure and temperature in the subcooled refrigerant after the condenser. The mass flow was corrected for the oil circulation rate. While the condenser is mounted in a frame, the evaporator is installed in the production HVAC module. Therefore the condenser air flow is uniform, while the evaporator air flow is similar to the flow in the vehicle. In order to compare different IHX lengths, all system interfaces remain in the same position, and the total suction line always had the same length (with and without IHX).

Refrigerant pressure transducers and temperature sensors are placed at all important positions in the circuit. All refrigerant temperature sensors are from type PT-100 and are immersed in the refrigerant while measured pressures are absolute. Pressure drop along the IHX is measured by a differential pressure sensor. On the air side, thermocouples from type K are arranged on the grids upstream and downstream from the evaporator and condenser, see [Figure 6c](#):

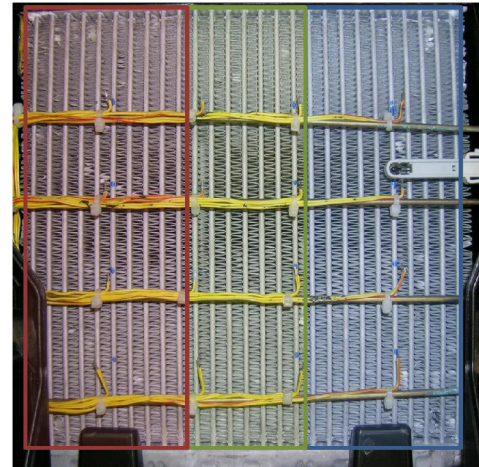


Figure 6c. Evaporator temperature grid vs. paths

The air relative humidity is measured with two capacitive sensors after the evaporator. The reported value is the average of these two sensors.

TEST CONDITIONS

The applied test conditions are shown in [table 1](#). These test conditions reflect the typical bandwidth of all important vehicle operating conditions for single evaporator and dual evaporator A/C systems.

The system parameters were measured in stable conditions after a stabilization period of approximately 10 minutes. Data were recorded for 10 minutes with a data acquisition frequency of 1 Hz. Test results are the average of the data over this time. The reproducibility of the results was checked by repeating several tests.

The tests 5 to 7 were conducted with a de-stroked compressor as the air temperature target was reached. The compressor was controlled manually with a PWM controller, using the average evaporator grid temperature as an input.

TEST RESULTS AND DISCUSSION

CHARGE DETERMINATION

The amount of refrigerant to choose for production charging can be chosen in a number of ways. In most cases, it is chosen by carefully testing the system at a variety of charges and plotting the results as shown in [Fig. 7](#):