

Internal Heat Exchanger Integration for a Dual Evaporator MAC System

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ABSTRACT

The United States Environmental Protection Agency (EPA) as well as the European Commission (EC) are developing test procedures to regulate mobile air conditioning system (MAC) efficiency to reduce greenhouse gas emissions and reduce global warming. In the United States, air conditioning related MAC credits can be earned by implementing an internal heat exchanger (IHX) into a MAC system. By integrating an IHX into a MAC system the, Coefficient of Performance (COP) can be increased at the same time increasing cooling capacity. This improvement in efficiency reduces the energy and/or fuel consumption of the MAC system.

This paper will compare various IHX plumbing configurations for a dual evaporator system with R1234yf refrigerant. A MAC system optimized for efficiency as well as evaporator cooling capacity is used to assess these different IHX plumbing configurations. Factors that will be considered include IHX heat transfer rate, suction pressure drop, and the impact of these factors on the COP and cooling capacity. The optimal IHX plumbing configuration for this dual evaporator system will be identified and its integration into the vehicle discussed. This paper also describes the MAC system, test facility used and the testing procedure applied.

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INTRODUCTION

In recent years, fuel efficiency is becoming more and more important to reduce greenhouse gas emissions. The EC Environment as well as other regulatory agencies like the EPA are developing test procedures to regulate MAC system fuel consumption and its impact on vehicle emissions. Various papers [1, 2] describe that by implementing an IHX into a MAC system, the system COP and cooling capacity can be increased at the same time. Reference [3] stated that by integration of an IHX, the COP can be increased up to 15% at the same level of cooling capacity. Improvement in system efficiency reduces energy consumption of the MAC system and consequently the fuel consumption of a vehicle.

The change to a new, environmentally-friendly air conditioning refrigerant, potentially R1234yf, may cause a cooling performance loss of approximately 9%. The physical refrigerant properties of R1234yf are slightly different from

the properties of R134a. As the specific enthalpy difference of R1234yf is smaller, mass flow must be increased to achieve the same cooling performance. The performance degradation can also be compensated by implementing an IHX. Reference [4] stated that the IHX will likely become standard with R1234yf systems. R1234yf is called environmentally-friendly because of a much lower Global Warming Potential (GWP) compared to R134a refrigerant.

Various internal heat exchangers have been commercialized in vehicles with R134a refrigerant and single evaporator MAC systems. This paper will focus on the integration of a typical coaxial "tube in tube" IHX for a dual evaporator MAC system applied to R1234yf refrigerant. This effort summarizes the work to complete a systematic investigation of different capabilities of IHX plumbing configurations. Cooling capacity, COP, and compressor discharge temperature have been studied with same Thermal

Expansion Valve (TXV) setting and slope. A deeper condenser analysis shows how the IHX placement affects the refrigerant outlet temperature of the condenser. The optimal IHX plumbing configuration for this dual evaporator MAC system was identified considering thermodynamic aspects at steady state test conditions. Beside the thermodynamic aspects, some pros and cons regarding the IHX integration into a typical vehicle engine compartment and refrigerant cycle were analyzed and summarized.

All tests were performed on a system bench with near production-intent MAC system components. This paper does not cover an investigation for dual evaporator systems where the second evaporator is used as a chiller, e.g., for battery cooling in hybrid vehicles. This paper also does not discuss more than two evaporators in a MAC system nor deeply investigate the optimal TXV settings for each IHX plumbing configuration.

BACKGROUND INFORMATION AND THEORY

The integration of a coaxial “tube in tube” IHX into a vehicle refrigerant cycle with a single evaporator was discussed in detail in reference [2]. Generally an IHX increases the COP at equal evaporator capacity as the refrigerant mass flow can be reduced due to enlarged enthalpy change across the evaporators. Increased compressor volumetric efficiency is realized by less re-expansion in the compressor cylinders as the liquid drops in the suction line will not evaporate in the compressor cylinder but in the IHX. These effects result in lower energy consumption of the system for the same capacity. The effect of an IHX in a dual evaporator MAC cycle is illustrated in Fig. 1.

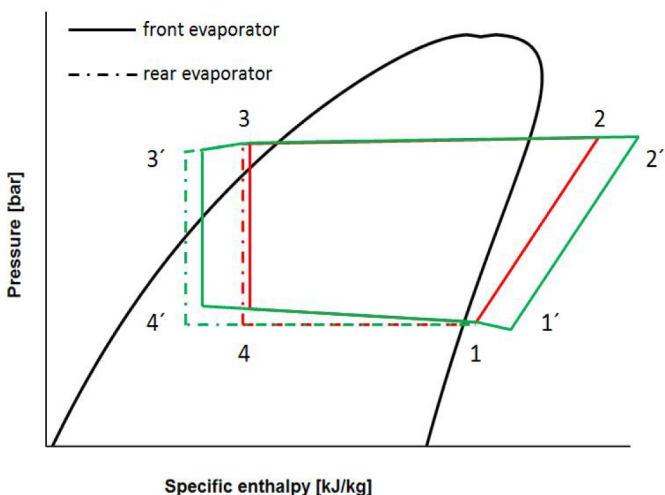


Figure 1. Dual evaporator MAC cycle with and without internal heat exchanger

The refrigerant cycle labeled with 1-2-3-4 is the theoretical refrigerant process without an IHX, with small

pressure drops in the condenser and evaporators, with a limited subcooling at the outlet of the condenser and a limited superheat at the outlet of both evaporators. A higher refrigerant mass flow to the front evaporator causes a higher pressure drop. The refrigerant cycle with an IHX is labeled with 1'-2'-3'-4'. The distance between 4-4' and 3'-3 represents the additional subcooling provided by the heat transfer in the IHX. Hereby the specific enthalpy in each evaporator is larger than in the refrigerant cycle without an IHX. This additional specific enthalpy improves the cooling capacity at equal refrigerant flow rates. As a consequence, the specific enthalpy of the suction vapor increases with the same amount $h_{1'} - h_1 = h_{3'} - h_3$. This addition of heat at the suction side will increase the compressor inlet temperature. Furthermore, the density of the suction vapor decreases due to the increased low-side pressure drop in the IHX, and therefore the refrigerant mass flow and compressor torque will decrease too. Further details are described for a single evaporator system in reference [5].

In this paper, the introduction of an IHX into a system with two evaporators is addressed. The major differences of implementing an IHX into a refrigerant cycle with two evaporators will be discussed in the following sections.

System Stability

System stability can be more of an issue with two evaporator systems since the two TXVs can interact. Interacting TXVs can result in flow variations to each evaporator as both try to satisfy their flow criteria. Typically, the front TXV will be designed with a cross-charged bulb element, and the rear TXV will be designed with parallel charge. Cross-charged means two or more different gases are in the TXV bulb. Parallel-charged means a single gas is in the bulb. Typically the gas in the bulb is the same as the system charge. During the testing, some instability was observed, but all data represents the average over at least 10 minutes so the variation is averaged over this time period.

At low charge conditions, the rear TXV wants to flow more refrigerant to the rear evaporator to satisfy the rear evaporator load. This can cause inadequate refrigerant flow to the front TXV and reduced performance in the front evaporator. Two charge determination tests (rear blower on / off) have been run to assure adequate charge is in the system to minimize this concern.

Pressure Drops

Rear evaporators are often designed with more passes than front evaporators. This keeps the refrigerant velocity higher to minimize the trapping of oil in the rear evaporator. The refrigerant flow is typically half of that which flows through the front evaporator. For this reason, the size of refrigerant lines to the rear evaporator should not be as large as those to the front evaporator, particularly the suction line or compressor return line. This smaller dimension also minimizes any tendency for trapping oil in these lines. Each plumbing configuration had different pressure drops on the

suction side due to the different routing of the lines which is discussed more detailed in the result section.

Refrigerant Distribution

The TXV setting, the evaporator size, and pressure drop all affect the refrigerant flow or distribution between front and rear evaporators. It is important to check these items carefully when an internal heat exchanger is added to the system to confirm that the distribution is not shifted to “starve” the front evaporator for refrigerant flow. The performance of the front system is considered more critical to the overall acceptance of the HVAC system in the vehicle.

TEST SETUP AND MAC SYSTEM

IHX Plumbing Configurations Investigated

Several IHX plumbing configurations can be reasonable integration solutions for a dual evaporator system. To identify the optimal IHX plumbing configuration for this dual evaporator system, the following five plumbing configurations were assessed. Four of these plumbing configurations were also investigated on a MAC system bench and compared to the base system which has no IHX.

Plumbing configuration C5 was not studied on the system bench because of lowest evaluation rating and much higher HVAC system on-costs as discussed later in [Table 1](#). The length of the IHX was the same for each configuration. The base system without IHX is labeled with C0 having same suction line routing as the IHX configurations.

Design configuration 1

[Figure 2](#) shows the plumbing configuration labeled with C1. In this configuration both front and rear evaporator are connected to the IHX. The refrigerant to and from both evaporators is passing the IHX.

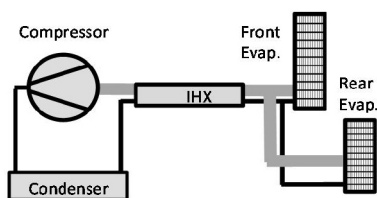


Figure 2. IHX plumbing configuration C1

Design configuration 2

[Figure 3](#) shows the plumbing configuration C2. In this configuration only the front evaporator is connected to the IHX. The refrigerant of the front evaporator is passing the IHX, while the refrigerant from the rear evaporator is bypassing the IHX.

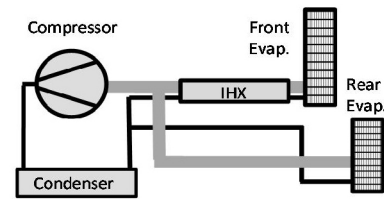


Figure 3. IHX plumbing configuration C2

Design configuration 3

[Figure 4](#) shows the plumbing configuration C3. The low pressure side of both front and rear evaporator is connected to the IHX. The liquid side of the rear evaporator is bypassing the IHX.

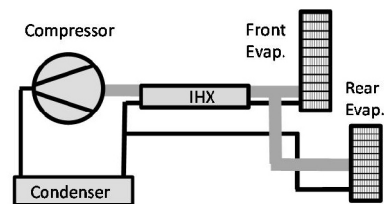


Figure 4. IHX plumbing configuration C3

Design configuration 4

[Figure 5](#) shows the plumbing configuration C4. The high pressure side of both front and rear evaporator is connected to the IHX. The suction side of the rear evaporator is bypassing the IHX.

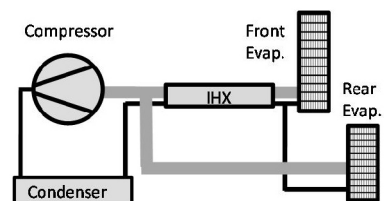


Figure 5. IHX plumbing configuration C4

Design configuration 5

[Figure 6](#) shows plumbing configuration C5. Each evaporator has its own separate IHX. In this configuration the capacity of each IHX should be balanced with its evaporator.

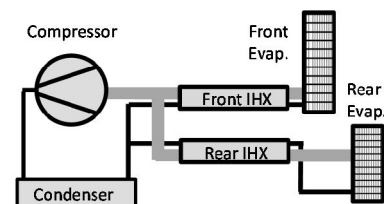


Figure 6. IHX plumbing configuration C5

Table 1. Comparison of IHX plumbing configurations

	C1	C2	C3	C4	C5
Design Complexity and Packaging	5	1	3	2	0
Additional Weight [g]	5	1	3	2	0
Additional Liquid Volume	4	2	2	4	1
Low Number of Parts/Joints	3	1	2	2	0
Vehicle Manufacturing	5	1	3	3	1
Vehicle Service	5	1	3	2	1
Refrigerant Plumbing Logistic	5	1	3	4	2
Refrigerant Plumbing Cost	5	1	3	2	0
Total	37	9	22	21	5

More specific details of each configuration and some pros and cons regarding the IHX integration into the vehicle are summarized in detail level in [Table 1](#).

A rating of five means best, zero means lowest. The plumbing configurations C1 to C4 received the highest ratings and were therefore investigated on a MAC system bench. Out of these four configurations, C1 has the highest rating in [Table 1](#) and is the preferred solution when vehicle integration aspects are considered.

MAC System Description

An A/C system with near production-intent components designed for a large midsize vehicle was used. This A/C system represents a typical MAC performance for a high volume large midsize vehicle. The components of the A/C system used are described in the following section:

Compressor

The compressor is a variable displacement type with 180 cc displacement, externally-controlled, 110 mm pulley diameter, and 195 ml PAG oil optimized for R1234yf.

Front evaporator

The front evaporator is a “flat-tube and fin” design with 250 mm × 238 mm × 58 mm. Its pass configuration is 2 × 2 and optimized for R134a refrigerant. During testing the evaporator is installed in the production HVAC module with an integrated blower module.

Front TXV

The front TXV is a typical block design with 1.5 tons, nominal settings 2.25 barg and 3.1 barg, cross charged, and optimized for R1234yf systems with IHX.

Rear evaporator

The rear evaporator is a flat-tube and fin design, 142 mm × 235 mm × 55 mm installed in the production HVAC module with an integrated blower module. Its pass configuration is 2 × 3 and optimized for R134a.

Rear TXV

The rear TXV is a block design with 1.2 tons, settings 1.7 barg and 2.7 barg, parallel charged, and without shut off valve. This setting is not optimized for IHX and R1234yf.

Condenser

The condenser is a “flat-tube and fin” design with integrated receiver dryer and integrated subcooling pass with 635 mm × 390 mm × 15 mm. It has 3 passes with 17-11-8 tubes and 1 pass for subcooling with 5 tubes and is optimized for R134a.

Internal heat exchanger

The IHX is a typical extruded coaxial “tube in tube” design, counter flow, 490 mm length, and 25 mm outer diameter. The IHX routing has one 90 degree and four 45 degree bends. The IHX profile is optimized for R1234yf.

TEST BENCH AND TEST CONDITIONS

Test Bench Description

A schematic layout of the test bench is shown in [Fig 7](#). The test bench has three separate air conditioning loops.

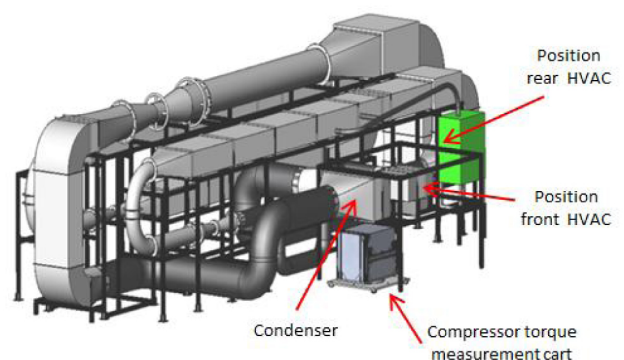
**Figure 7. Test bench layout, schematic**

Table 2. System bench test conditions

	Compressor	Condenser		Front Evaporator				Rear Evaporator		
No.	compressor speed	temp. air inlet	air speed air inlet	temp. air inlet	air mass flow	rel. humidity air inlet	set point air outlet	temp. air inlet	air mass flow	rel. humidity air inlet
	[min-1]	[°C]	[m/s]	[°C]	[kg/h]	[RH %]	[°C]	[°C]	[kg/h]	[RH %]
1	900	45	1,5	45°C	540	25%	3	30°C	390	25%
2	2500	45	3	45°C	540	25%	3	30°C	390	25%
3	4000	45	4	45°C	540	25%	3	30°C	390	25%
4	900	35	1,5	35°C	540	40%	3	25°C	390	40%
5	2500	35	3	35°C	540	40%	3	25°C	390	40%
6	900	35	1,5	35°C	390	40%	3	25°C	390	40%
7	2500	35	3	35°C	390	40%	3	25°C	390	40%
8	900	35	1,5	35°C	390	40%	3	25°C	0	40%
9	2500	35	3	35°C	390	40%	3	25°C	0	40%
10	900	25	1,5	25°C	390	80%	3	25°C	390	40%
11	2500	25	3	25°C	390	80%	3	25°C	390	40%
12	900	25	1,5	25°C	240	80%	3	25°C	240	40%
13	2500	25	3	25°C	240	80%	3	25°C	240	40%
14	900	25	1,5	25°C	240	50%	3	25°C	240	40%
15	2500	25	3	25°C	240	50%	3	25°C	240	40%
16	900	25	1,5	25°C	240	80%	3	25°C	0	40%

In the evaporator loops, the air temperature can be conditioned from 10°C to 55°C and the air mass flow varied from 180 kg/h to 700kg/h. The ambient air temperature in the condenser loop can be heated up to 65°C. A condenser air mass flow from 1000 kg/h to 4800 kg/h is possible. In all loops the air mass flow is measured by Venturi meters. Both evaporator conditioning loops have separate humidifiers. The temperature is measured before and after each heat exchanger with a grid of 16 thermo couples. The humidity in each evaporator inlet and outlet is measured by capacitive sensors. At the evaporator inlet, the relative humidity is measured in the HVAC modules. To assure better mixture of the air, the relative humidity is measured at the outlet side a few meters after the evaporator in the air conditioning loop. Ducts from the outlet of the module to the section where the humidity is measured are well insulated. The air enthalpy, needed for the calculation of the air side capacities, is calculated by air temperature, pressure and humidity.

The compressor is powered by an 11 kW electric motor and is mounted on the A/C system carrier. The torque meter is mounted to another cart and connected to an electric motor by a belt, see description in [Appendix 3](#). The output side of the torque meter is connected with the compressor by a second belt. Corrections are not applied for belt losses. The accuracy of the torque meter is ± 0.04 Nm.

Sound velocity sensors are used to determine the oil circulation rate for each evaporator loop. The oil circulation rate is calculated from the measured speed of sound, pressure and temperature in the subcooled refrigerant. The refrigerant

mass flow is corrected for the oil circulation rate. The refrigerant mass flow of the front and rear evaporator is measured with a Coriolis sensor and sound velocity sensor placed before each TXV according [Appendix 2](#).

Temperature and pressure sensors are placed on all relevant locations as shown in [Appendix 2](#). All lines are prepared with specific adapters to install the appropriate sensors. Different adapter types are shown in [Appendix 4](#). All temperature and pressure sensors are calibrated. The pressure drop between the front evaporator outlet and compressor inlet is measured by a 3 bar differential sensor. The pressure drop between the rear evaporator outlet and compressor inlet is measured by a 1 bar differential sensor.

Air side pressure drop is measured by differential sensors to evaluate the influence of condensate in the evaporators. Differential sensors also used to measure the air pressure before HVAC and ambient pressure.

Test Conditions

The applied test conditions are shown in [Table 2](#). Test conditions below 25°C evaporator air inlet temperature were not tested due to the limitations of accuracy of the compressor torque and the refrigerant mass flow.

The test points from 13 to 16 were conducted with a de-stroked compressor. When the air temperature set point was reached the compressor was controlled manually with a PWM controller, using the average evaporator air outlet grid temperature as the controlling input. Test points 8, 9, and 16 had no airflow to the rear evaporator even though the ambient

environment of the HVAC module and consequently the TXV was maintained.

Test Data Acquisition

All system parameters are measured in stable conditions after a stabilization period of approximately 10 to 35 minutes. Data have been recorded for 6 to 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 has been checked by repeating several test sequences.

Test Data Analysis

The test results were analyzed by balancing the air and refrigerant side capacities. This balance is maintained within 5% at most points and if not, the test condition is repeated. Analysis included in depth analysis of the refrigerant properties and pressure-enthalpy diagrams in addition to the air properties of the system.

TEST RESULTS AND DISCUSSION

Charge Determination

A refrigerant charge determination was performed for each IHX plumbing configuration at 2000 compressor rpm, 40°C ambient, and 40% relative humidity. The front blower was set to 540 kg/h and 390 kg/h for the rear. For example, the plumbing configuration C1 showed a plateau of the subcooling after the condenser at 9 K which corresponds with a refrigerant charge between 860 g and 960 g. An amount of 920 g refrigerant charge, which is almost at the middle of the plateau, has been chosen. This slightly higher amount of refrigerant compared to an A/C system installed in a vehicle is caused by the high pressure hoses, which connect the separate condenser chamber with the HVAC module and the compressor. The subcooling after the condenser is used instead of the subcooling before TXVs to choose the refrigerant charge, since the latter is influenced by the heat transfer of the IHX.

Each charge determination was repeated with rear blower off to ensure subcooling at condenser outlet when refrigerant is allocated in the rear evaporator. The criteria for evaluation are to confirm that subcooling still exists when the airflow is turned off to the rear evaporator at the charging condition.

Using the same charge determination approach for the other configurations, the charge amounts chosen are shown in Table 3. Charge amounts are slightly different because of liquid volume changes caused by the different plumbing configurations.

Table 3. Selected charge amounts

Configuration	Charge Amount	Comment
C0	840 g	Baseline system
C1	920 g	All refrigerant passes through IHX
C2	940 g	Only refrigerant from the front evaporator passes through the IHX
C3	935 g	All suction gas passes through IHX. Liquid from rear is by-passed
C4	920 g	All liquid passes through the IHX. Suction gas from rear is by-passed

Oil Circulation

OCR was measured in the front evaporator loop and in the rear evaporator loop separately. No significant difference in OCR was measured between both loops at the test conditions evaluated. Also no significant OCR influence caused by the IHX placement was observed. As the compressor is designed with an oil separator, the OCR rate was below 3.3% for all test points.

Refrigerant Mass Flow Distribution

IHX placement affects the refrigerant mass flow distribution between the front and rear evaporators. When bypassing the IHX the total effective cross-sectional area is higher due to parallel line routing. This increased flow area is causing a lower pressure drop in the lines which increases the mass flow at equal compressor power. Figure 8 shows how the rear-to-front evaporator mass flow ratio is increased when the refrigerant of the rear evaporator is bypassing the IHX.

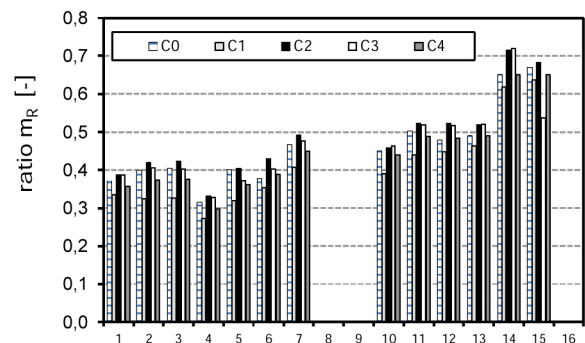


Figure 8. Refrigerant mass flow ratio, all flow is to the front evaporator in the test points 8, 9, and 16

Configuration C2 has the highest evaporator mass flow ratio as both the liquid and suction lines to the rear evaporator are bypassed. Beside this, the pressure below the diaphragm on the TXV is reduced allowing the TXV to open more in the rear evaporator. C1 has a lower ratio and the best weighted energy consumption as will be discussed later.