

High Efficiency Subcool Condenser

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ABSTRACT

Sustainable reduction of the CO₂ emissions of future vehicles goes hand in hand with the increase in efficiency of ancillary units. Of particular importance is the reduction of fuel consumption attributed to the vehicle's air conditioning system. The innovative development of a condenser with a 3-pass subcooling zone renders it possible to significantly increase the efficiency of the cooling circuit. This is applicable both in the case of current and future refrigerants.

The basis for increasing the efficiency of the condenser is rooted in the systematic analysis of the local heat transfer functions, pressure drops and implementation of an improved flow path configuration. The current condenser design, which includes the functionality of a receiver/dryer with a dedicated subcooling zone, is effective at providing stable liquid refrigerant to the TXV under a variety of conditions. By optimally adapting the flow cross-section to the density change of the refrigerant and increasing the subcooling function by means of a 3-pass flow subcooling zone, it is possible to achieve a significantly lower outlet temperature of the refrigerant. The locally higher pressure drop in the subcooling zone does not have a negative impact on the vapor compression circuit and can therefore be used to increase efficiency without the risk of losses.

The theoretical derivation, as well as simulation and measurement results for the new condenser, are presented and discussed in comparison with conventional designs. A new condenser specification has been derived and validated for the system-oriented evaluation of the component. Finally, the validation results from the circuit and vehicle measurements in the wind tunnel are presented. The newly developed condenser resulted in a significant increase of 4% of evaporator performance while maintaining mass and package space. Alternatively, weight and conduction depth can be reduced by 25% while maintaining capacity and system efficiency.

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INTRODUCTION

In the last decade, mobile air-conditioning (MAC) systems underwent continuous improvement in terms of full load performance, efficiency, weight and cost reduction. The change to a new environmental friendly refrigerant (R1234yf) poses an additional challenge. While the GWP (Global Warming Potential) of R1234yf is only 4 compared to 1,400 of R134a, implementation of the new refrigerant leads to an AC system performance loss of approximately 8%. Innovative countermeasures need to be identified and developed to overcome this performance degradation.

Besides the use of the new refrigerant and impact on full load performance, efficiency at partial and low load conditions is crucial to address the reduction of fuel consumption due to AC operation. Fuel consumption of ancillary units significantly contributes to the overall fuel consumption of a vehicle at real life conditions over the year. Even though not all standard cycles for vehicle fuel

consumption account for the use of the AC unit yet (AC is off), the real life fuel consumption is crucial to the end-customer. In addition to the judgment of the end-user, more fuel consumption cycles will be run with AC-ON and drive the efficiency of MAC systems in the future.

A high performance condenser in a MAC cycle including a receiver/dryer and a subcool area has to transfer the heat from the AC circuit, namely heat from evaporator and compressor, to the environment. With the target to improve the condenser performance, the idea of stable subcooling was developed already 15 years ago [1]. With the installation of the receiver between condensation and subcool area, saturated liquid refrigerant is provided to the subcool pass. With the further subcooling of the liquid refrigerant, the outlet enthalpy of the condenser is reduced and the effective enthalpy difference at the evaporator is higher. This leads to a wider process in the p,h-diagram and improves both performance and COP.

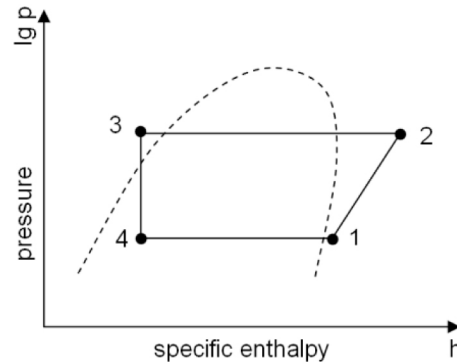


FIGURE 1. Typical MAC operation point in p-h-chart

The idea to further improve subcooling is obvious but limited by the space available for the KOMO (condenser module with receiver/dryer and subcool area). Increasing the subcool area can only be achieved by decreasing the number of tubes in the condenser, because the overall number of tubes in the heat exchanger is constant due to packaging constraints. Lower number of tubes for condensation effects the overall performance of the condenser and leads to a higher discharge pressure at the compressor and thus reduced COP. Hence, the target is to find the most effective way to increase subcooling with only moderate effect on high pressure.

THERMODYNAMIC ANALYSIS

AC System Performance and COP

Fig. 1 shows a typical operation point of a MAC system plotted in the pressure enthalpy chart. (1→2 compression, 2→3 heat rejection to cooling module air flow via condenser, 3→4 throttling, 4→1 heat absorption from cabin air flow via evaporator)

The evaporator performance can be calculated from the refrigerant's specific enthalpy difference across the evaporator and the refrigerant mass flow:

$$\dot{Q}_{Evap} = \dot{m}_R \cdot (h_1 - h_4)$$

The efficiency of the system is expressed by the coefficient of performance (COP) which is calculated as the ratio of evaporator performance and mechanical power input to the compressor:

$$COP = \frac{\dot{Q}_{Evap}}{P_{Comp}} \approx \frac{h_1 - h_4}{h_2 - h_1}$$

To achieve the target of increasing the evaporator performance there are two main possibilities. The first is to increase the refrigerant mass flow which requires a compressor having a greater displacement or being operated at higher speed. The second, more efficient way is to reduce the specific enthalpy of the refrigerant entering the evaporator (h_4). Since the throttling of the refrigerant in the expansion device is virtually isenthalpic ($h_4 = h_3$), this is equivalent to

reducing the specific enthalpy of the refrigerant leaving the condenser.

To reduce the refrigerant's specific enthalpy at the condenser outlet, usually the subcooling of the refrigerant is increased. Considering a given face area and hence a constant total number of tubes of the condenser, this is achieved by increasing the number of tubes for subcooling the refrigerant and reducing the number of tubes for condensation. The reduction of the number of tubes for condensation leads to an increase in high pressure because the condensation area is smaller. In addition to the negative effects of a rise in high pressure on the refrigerant mass flow and the COP, it is shown in various recent studies that there is an optimal subcooling value [2, 3, 4]. Figure 2 shows that if the subcooling value is increased above this optimum value, the specific enthalpy at the condenser outlet begins to rise again though the subcooling is still increasing.

Following this knowledge, it is clear that the right approach to increase evaporator performance is not to maximize subcooling but to minimize the refrigerant temperature at the condenser outlet. This leads to another degree of freedom for development: the pressure drop in the subcooling region.

Figure 3 shows a detail of the pressure-enthalpy-chart. Point A indicates the beginning of the subcooling region, points B and C the end of the subcooling region, Point B being at a high subcooling value (a) at the same pressure level as point A. Point C has a much lower subcooling value (b) at a lower pressure due to the pressure drop in the subcooling region and the gradient of the boiling curve. However, if the increased pressure drop of the subcooling A→C is caused by an improved heat transfer due to higher refrigerant flow velocities, the point C can be at a lower refrigerant temperature than point B though its subcooling value is lower. As a result, employing the freedom to locally raise the pressure drop in the subcool area, the overall performance can be improved. Hence, for the specification of a condenser the refrigerant pressure drop of the subcool part should be separated from the condensation part. The steeper line in the p,h diagram is just a part of the pressure drop which takes place in the further throttling in the valve.

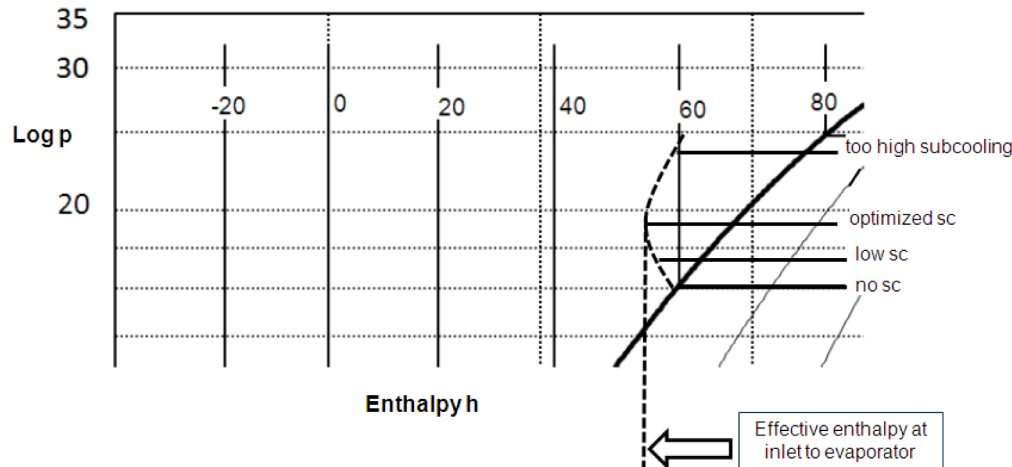


FIGURE 2. Optimal subcooling [2]

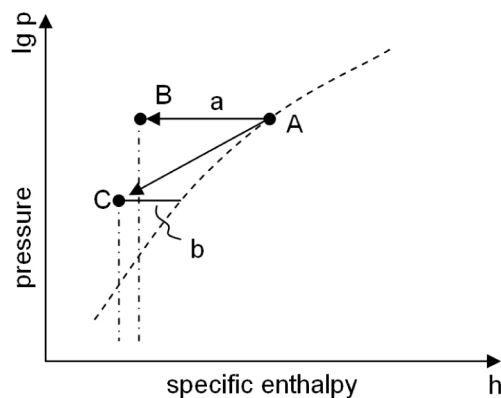


FIGURE 3. isobaric and non-isobaric subcooling

CHARACTERIZATION OF THE INNOVATION

Assuming a given heat exchanger matrix which is well-engineered in terms of overall heat transfer coefficient and balanced heat transfer areas on the refrigerant and air sides, in this paper the discussion is limited to the refrigerant side analysis. The flow passes of a state of the art condenser are shown in [figure 4](#). The condensation part is partitioned into 3 passes with a declining number of flat tubes of 21-13-5. This is done to adapt the cross section for the refrigerant to the changing density on its way through the condenser. In fact, the density changes by a factor of 20 from a superheated gas at the inlet and a more or less saturated liquid at the end of condensation and inlet to the receiver. The refrigerant is collected in the receiver/dryer, which then provides saturated liquid to the subcooling part with 4 tubes in the example. This definition results in a subcooling of the liquid refrigerant of 5-7 K at standard conditions. The internal heat transfer coefficients $\alpha_{i,m}$ in [figure 4](#) is defined as the mean value over

each pass and set to 100% for each section for the following discussion.

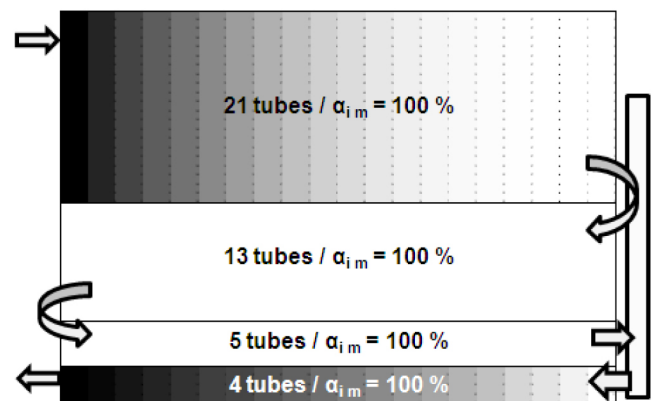


Figure 4. Flow passes of a 4-pass series condenser 21-13-5=4.

The analysis of the resulting refrigerant flow velocity in the tubes along the flow path is shown in [figure 5](#) and the resulting internal heat transfer coefficient $\alpha_{i,m}$ in [figure 6](#) in a qualitative comparison to the maximum coefficient. From the graphs it is obvious that at every next flow pass the velocity and respectively heat transfer coefficient is adjusted to a higher level. The overall heat transfer coefficient and thus the efficiency of the condenser is higher compared to a condenser with only one pass for condensation plus one pass for subcooling. This comparison is also shown in [figure 6](#), where the heat transfer coefficient of the 2-pass condenser is printed as dotted line. Of course, the geometry of the tubes need to be designed so that the cross section of the parallel tubes results in a moderate internal pressure drop with positive contribution to the effectiveness of the condenser in a given AC circuit.