HEAT EXCHANGER OPTIMISATION USING ZEOTROPIC REFRIGERANT R-455A

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Abstract

R404A is largely used in commercial refrigeration systems. Given the relatively high Global Warming Potential (GWP=3943) of R404A, this refrigerant has been under scrutiny by regulators, and will be phased out following the recent F-gas regulation. Recently, the low GWP refrigerant mixture R455A has been proposed as a viable option for R404A replacement. Changing refrigerant thermophysical properties may have a relevant impact on the performance of the heat exchangers, especially if glide management has not been taken into considerations. In this paper, a finned coil condenser is used as reference. An advanced simulation model is used in order to compare the heat transfer performance of the candidate refrigerant mixture R455A with those of R404A referring to typical operating temperatures at various refrigerant flow configurations.

Introduction

The European Union's new F-Gas regulations (Regulation (EC) No 517/2014) specifies that beginning on January 1, 2020 new refrigerators and freezers for commercial use (hermetically sealed equipment) can operate only with refrigerants having 100-year Global Warming Potentials (GWP_{100}) lower than 2500 and then is decreased to 150 after January 1, 2022.

As a consequence, it is very likely that recently built plants will need to be operated in the next years with refrigerants having GWP lower than 2500. This paper does not aim to review the different fluids proposed as replacement: considering that no single pure fluid has been proposed so far as a drop-in replacement of R404A, the present work aims at highlighting the peculiarities of refrigerant mixtures performance during condensation in comparison to R404A. Just a few of the proposed mixtures have an azeotropic behaviour. Most of them show a temperature glide between 3 and 10 K, depending on the composition and on the operating conditions.

Among the proposed mixtures, R455A is a ternary blend with the composition R744/R32/R1234yf (3/21.5/75.5 mass%), having a GWP_{100} of 146 and showing a temperature glide of 10 – 12 K, depending on the working conditions. It was considered as a possible drop-in substitute in a commercially available R404A self-contained freezer by Sethi et al. (2016). They found that the 24-h energy consumption of the compressor is 6% lower for R455A.

Thermodynamic Assessment

In figure 1 the most relevant properties of R455A are compared with R404A, according to REFPROP 9.1 (Lemmon et al. 2013).

It is well known that the use of zeotropic mixtures should be properly addressed both from a thermodynamic point of view and in terms of the heat exchangers design.

Referring to ideal thermodynamic cycles, prof. Cavallini in 1995 observed that Lorenz cycle should be considered as the reference, instead of the Carnot cycle when a zeotropic mixture is used. Furthermore, he demonstrated that, for a given temperature lift of the secondary fluid, the advantages in terms of COP for the mixture is limited (2 – 3 %) when the temperature glide is in the order of 10 K. This consideration is referred to ideal cycles only, with perfect count-current configuration of the heat exchangers, that is not achievable when using air heat exchangers: in this case only cross flow configuration or quasi-counter flow configuration is possible. Furthermore, the presence of refrigerants with different volatility, causes an additional mass resistance that penalizes the heat transfer coefficient during condensation and boiling. This penalization is expected to be more relevant for mixture with larger temperature glide. Kondou et al. (2016) experimentally evaluated the condensation heat transfer coefficient of the ternary mixture R744/R32/R1234ze(E) (4/43/53 mass%) with 11 K temperature glide at 40°C saturation temperature, inside a 5.35 mm (reference internal diameter) microfin tube at G=200 kg $m^{-2}s^{-1}$ mass flux. They found that the Cavallini et al (2009) correlation with the Silver, Bell and Ghaly correction (as suggested by Cavallini) can be used for properly evaluate the heat transfer coefficients. Figure 2 reports a comparison between R404A and R455A at several G and fixed 40°C average saturation temperature (dew-bubble temperature average for an isobaric condensation) and vapour quality x=0.5 inside the 7.7 mm microfin tube described in Table 1. It can be observed that R455A HTCs are very similar R404A ones.

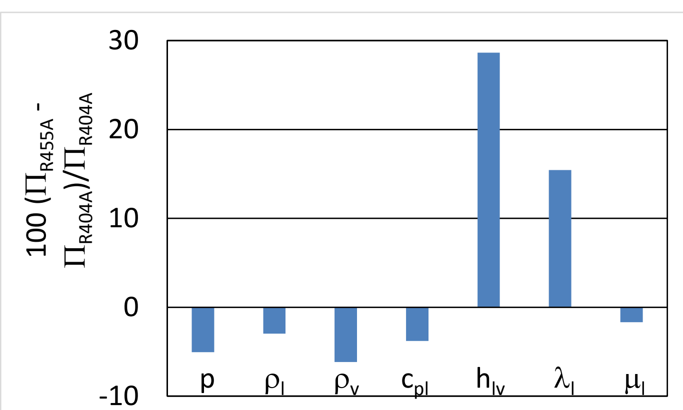

Figure 1. Comparison of some relevant thermodynamic and thermophysical properties for R455A and R404A

The comparison of the HTCs tells only a part of the story: as a matter of fact, also the refrigerant pressure drop during the phase change process should be taken into account, since it affects the local saturation temperature and hence the actual local driving temperature difference between the two fluids. Cavallini et al. (2010) proposed the so-called Penalty Factor approach (PF) to compare the heat transfer performance of different fluids. In figure 3, the PF of R404A and R455A are compared: the lower the penalty factor, the better is the heat transfer performance. However, one should consider that in an air cooled heat exchanger, the most relevant thermal resistance is on the air side, so the small difference in the refrigerant side performance can be considered negligible on the overall thermal transmittance.

Finned Coil Heat Transfer Performance

As a further step, in order to assess the behaviour of R455A in comparison with R404A, one finned coil condenser was considered among typical European production. The main geometrical characteristics are reported in Table 1, while the operating conditions for R404A are listed in Table 2. Different number of circuits in parallel, all arranged in a prevalent counterflow lay-out, were investigated (from 3 to 48).

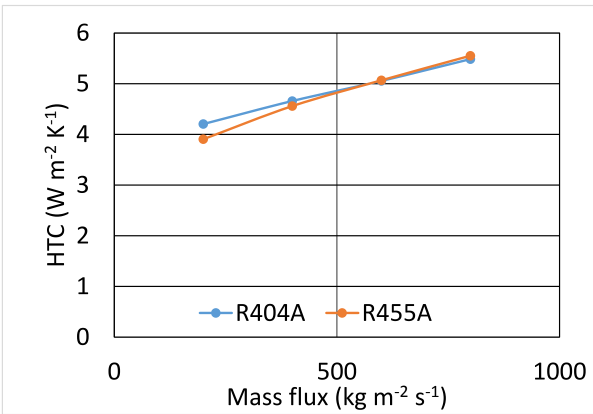

Figure 2 Effect of mass flux on the condensation heat transfer coefficient (HTC) at vapour quality 0.5 and average saturation temperature 40°C.

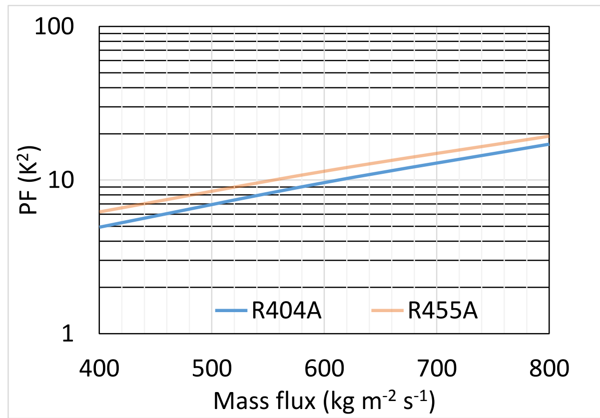

Figure 3. Penalty factor PF for 7.7 mm ID tube, 40 °C average saturation temperature, x=0.5.

To simulate the performance of the condenser, a detailed numerical code was used. The code is described in details in Zilio et al. (2015). The air side heat transfer coefficient was evaluated according to experimental data obtained from the manufacturer.

Table 1. Geometrical description of the condenser

Characteristics	
Longitudinal tube spacing [mm]	21.6
Transverse tube spacing [mm]	25
Tube length [mm]	1140
Inside tube diameter (fin tip) [mm]	7.7
Number of rows [-]	4
Number of tubes per row [-]	48
Number of refrigerant circuits [-]	4-48
Fin spacing [mm]	2.1
Type of fins	louvered
Tube inner surface	microfin

Table 2. R404A operating conditions for the condenser

Condensate / Superheating	[K]	0.0
Average condensation temperature (dew-bubble)	[°C]	40.0
Dew condensation temperature	[°C]	40.2
Air inlet condenser (dry bulb)	[°C]	25.0
Condenser air face velocity	[m s^{-1}]	2.0

The performance of R404A is reported in figure 4, at the conditions in Table 2, for the different number of circuits in parallel. R455A performance was evaluated by iteratively changing the average saturation temperature in order to achieve the same heating capacity obtained with R404A at any given finned coil circuitation. The corresponding saturation temperatures (dew-bubble, both calculated at the inlet pressure) are reported in the graph.

In order to maximize the exchanged heat flow rate with R404A, a number of circuits between 16 and 48 should be chosen. R455A can foster a reduction of the average saturation temperature up to 2.5 K when 16 or 24 circuits in parallel are considered. This result is a consequence of the better matching of refrigerant/air temperature profiles thanks to the relevant R455A temperature glide. This feature offers an opportunity for reducing the compressor discharge pressure and hence the compression work.

Conclusions

In this paper R455A and R404A heat transfer performance is compared thermodynamically, by means of the Penalty Factor approach and then with reference to a commercially available finned coil condenser simulated by means of a detailed 3-D simulation code. The analysis showed that R455A can be a viable option from the point of view of the condenser performance when considering a drop-in scenario of R404A. However, attention should be paid in the design of a quasi-counterflow design of the finned coil circuits, in order to avoid performance penalizations as a consequence of the R455A temperature glide.

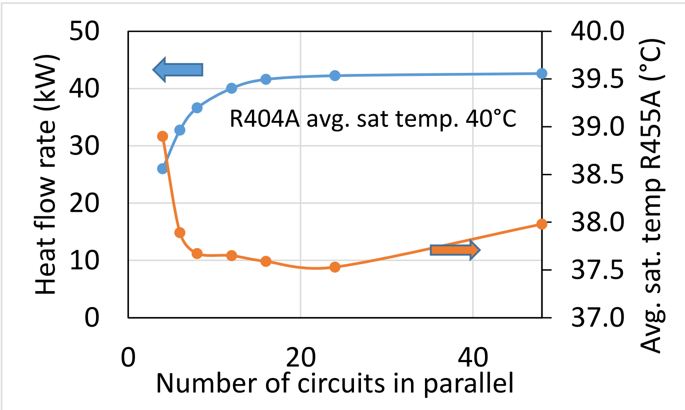

Figure 4. Finned coil exchanged heat flow rate and R455A average saturation temperature as a function of number of circuits in parallel. (p_{sat} R404A @ 40°C: 1822 kPa / p_{sat} R455A @ 37.5 °C: 1659 kPa)

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