# HERMETIC COMPRESSOR PERFORMANCE EVALUATION APPLYING A ZEOTROPIC HYDROCARBON BLEND INSTEAD OF CFC-12

Martien Janssen, Patrick Beks
Re/genT B.V. Meerenakkerweg 1
5652 AR Eindhoven, The Netherlands

## ABSTRACT

This paper deals with the evaluation of hermetic compressor performances when changing from the refrigerant CFC-12 to a zeotropic mixture of isobutane and propane. Performances are evaluated experimentally on a compressor calorimeter test bench for both refrigerants. Test results are compared to a theoretical analysis. It is shown that the definition of the evaporation temperature is essential for a proper comparison. Furthermore, the effect of the refrigerant change on the cooling capacity and efficiency (COP) will be evaluated over a large temperature operating range. Increases or decreases in efficiency can be observed depending on the operating point; efficiency increases are typically found at relatively high evaporation temperatures. Further, differences in pressure ratios between both refrigerants have an effect on the cooling capacities depending on the compressor design. In literature only limited information is available regarding the influence on the cooling capacity of the application of isobutane and propane in a CFC-12 refrigeration system. This study focuses on this specific aspect.

## NOMENCLATURE

| | | | |
|---|---|---|---|
| $\alpha$ | correction factor for valve losses [-] | $p_s$ | suction pressure [bar] |
| $\rho$ | density [kg/m$^3$] | $p_d$ | discharge pressure [bar] |
| $\Delta h$ | enthalpy difference of the evaporator [kJ/kg] | $V_{swept}$ | swept volume of the compressor [m$^3$] |
| $f$ | frequency [Hz] | $V_{dead}$ | dead volume of the compressor [m$^3$] |
| $\dot{m}$ | massflow [kg/s] | $Q$ | cooling capacity [W] |
| $\eta_v$ | volumetric efficiency [-] | | |

## 1. INTRODUCTION

The performance of three different hermetic CFC-12 compressor designs has been evaluated with CFC-12 and a zeotropic hydrocarbon blend, which consists of 43.15 weight percentage propane (R-290) and 56.85 weight percentage isobutane (R-600a). This mixture is referred to as “HC-blend” throughout this paper and shows a so-called “temperature glide”. To make a proper comparison between the performance of the compressors on both refrigerants, the compressors were tested on a compressor calorimeter test bench. It needs mentioning that the HC- blend can be used as a “drop in” for CFC-12 based systems. To ensure proper functioning of the refrigeration system after being recharged with the HC-blend, special attention is paid to the influence on the cooling capacity. In the following table the swept volumes of the compressors tested are given, as well as their cooling capacity and COP at ASHRAE conditions (for CFC-12). All compressors tested are of the low back pressure type.

| Compressor | Swept volume [cm$^3$] | Cooling capacity measured [W] | COP measured [-] |
|---|---|---|---|
| A | 18.27 | 407 | 1.11 |
| B | 5.2 | 126 | 1.01 |
| C | 6.65 | 215 | 1.32 |

***Table 1**: General properties of the compressors: Performances are given for CFC-12 at ASHRAE test conditions /1/).*

It needs mentioning that in a practical refrigeration system the temperature glide of the refrigerant in the evaporator affects the heat transfer characteristics from the evaporator to the device being cooled (e.g. compartment of a refrigerator). This study concentrates on the effect of changing the refrigerant in compressor calorimetric circumstances. In practical refrigeration systems other factors will play a role as well, such as the temperature glide on the evaporator, internal heat transfer coefficient, etc.

## 2. TESTING METHODOLOGY

Before the actual start of the tests, the compressors were operated for at least 100 hours. During this "running in" period, special attention was paid to performance changes, because it is known that some compressors slightly change the capacity during initial operation. The performance tests, as well as the 100 hours compressor running-in period, were carried out on a "secondary fluid type" calorimeter (Method A, ISO 917/1/).

During the tests the compressors were supplied with 230V, 50Hz. The compressors operated at an ambient temperature of 32°C; the "evaporator exit" as well as the "subcooler outlet" were maintained at 32°C. At a condensation temperature of 45, 55 and 69°C a range of evaporation temperatures, from -35 to +7°C, were set and subsequently measurements were performed.

Before the actual measurements the calorimeter refrigeration system was cleaned to eliminate any residual oil or refrigerant from previous tests. To ensure that no refrigerant remained in the oil when changing the refrigerant, the system was evacuated thoroughly whilst the oil was still at a relatively high temperature. This guarantees that only the refrigerant charged (CFC-12 or the HC-blend) was circulating in the refrigeration circuit.

Tests with CFC-12 and the HC-blend were performed with the same compressor oil. Due to the differences in solubility between the HC-blend and CFC-12 the actual viscosity of the oil in the compressor sump may slightly differ during the tests. However, the real effect on the viscosity has so far not been clearly established /4/. For this investigation, the effect of charging a HC-blend in a drop-in situation is of interest. For this reason the oil was not changed.

## 3. CHANGE IN COMPOSITION OF A ZEOTROPIC MIXTURE

As noted the HC-blend is a zeotropic mixture. When measuring compressor performances applying zeotropic mixtures it is important to consider that the refrigerant may change in composition inside the refrigeration system. In case this occurs, the composition of the actual mixture as pumped by the compressor is different from the composition of the mixture as charged to the test circuit. This generally results in an incorrect measurement of the compressor performance. Therefore, change in composition should be avoided during measurements of zeotropic mixtures.

Basically, there are three components in the calorimetric refrigeration circuit where change in composition may occur:

1. Evaporator: Tests have been performed with a "secondary fluid type calorimeter" /2/, which consists of a vessel in which the evaporator is constructed of a spirally wounded tube. The liquid vapour mixture entering the evaporation coil completely evaporates along the tube, and the liquid hold-up is small. For this reason the change in composition is assumed to be small, this in contrast to pool boiling evaporators;
2. Compressor shell: Due to different solubility of the mixture components in the compressor oil, change in composition could take place. However, considering the ratio of the oil content in relation to the refrigerant content of the calorimeter, and taking into account that during all measurements the compressor oil was at a relatively high temperature, this effect can only play a minor role;
3. Liquid accumulator in the high pressure line (after the condenser): The composition of the vapour will differ from the composition as charged. However, in the calorimeter only a small accumulator

is applied with a small vapour content. Calculations have shown that this yields only minor consequences (smaller than 2% on the composition).

In summary, when the calorimetric circuit is properly charged with the mixture components no problems are expected. So it can be assumed that the circulating composition is similar to the charged composition. A too low or too high total quantity of refrigerant in the calorimeter generally yields poor control behaviour (e.g., the subcooled liquid conditions cannot be maintained when the calorimeter is not sufficiently charged). The procedure applied during the tests in fact guarantees that the calorimeter has been adequately charged throughout all tests.

## 4. DEFINITION OF THE CONDENSATION AND THE EVAPORATION TEMPERATURES

The zeotropic mixture is characterised by a so called "temperature glide". This means that the condensation and evaporation processes do not take place at a constant temperature (which is the case for pure refrigerants such as CFC-12). Therefore, a definition of the condensation and evaporation temperature is required.

In this study the evaporation temperature is defined as the average of the evaporator inlet temperature and the saturated vapour temperature (dew point). The evaporator inlet temperature must be calculated using the pressure and the subcooling conditions in the liquid line (see Figure 1). This method of calculating the evaporation temperature best represents the performance of a compressor on an actual system, since it addresses the actual average temperature of the refrigerant inside the evaporator. It is recognised that this definition actually makes the evaporator temperature dependent on the condenser subcooling condition. In Table 2 the influence of the subcooling conditions are given. The condensation temperature is taken as the average of dew and bubble point temperatures at the condensation pressure.

## 5. THEORETICAL COOLING CAPACITY CALCULATIONS

In addition to the practical compressor tests, theoretical calculations /3/ have been performed comparing CFC-12 and the HC-blend in a refrigeration cycle. For these calculations the following formula is used:

$$\frac{Q_{HC-blend}}{Q_{CFC-12}} = \frac{\dot{m}_{HC-blend}\Delta h_{HC-blend}}{\dot{m}_{CFC-12}\Delta h_{CFC-12}} = \frac{\rho_{HC-blend}V_{swept}f\Delta h_{HC-blend}}{\rho_{CFC-12}V_{swept}f\Delta h_{CFC-12}} = \frac{\rho_{HC-blend}\Delta h_{HC-blend}}{\rho_{CFC-12}\Delta h_{CFC-12}} \quad (1)$$

If the definition for the evaporation and condensation temperature as given before is used, the calculations show that the cooling capacity for the HC-blend is approximately 13% lower than for CFC-12 (at standard ASHRAE conditions /1/). However, the capacity reduction is only 3% when the evaporator entrance temperature is chosen as the reference evaporation temperature. The capacity reduction is 23% when the dew point conditions are chosen as reference. These calculations demonstrate the sensitivity of the data with respect to the reference point chosen. In Figure 2 a graph is presented where this sensitivity is shown at the conditions described. In this figure it can also be noted that the reduction in cooling capacity depends on the actual evaporation temperature; the higher the evaporation temperature the smaller the decrease in capacity.

If the evaporation temperature is defined as the average of the evaporator inlet and dew point temperature, the subcooling condition affects this average temperature. In the following table the theoretical reduction of the cooling capacity is presented for different subcooling conditions at 55°C condensation temperature, -23.3°C evaporation temperature and 32°C evaporator exit conditions.

| Subcooled liquid temperature [°C] | Capacity compared to CFC-12 [%] |
|---|---|
| 10 | -10 |
| 32 (ASHRAE) | -13 |
| 55 (no subcooling) | -17 |

***Table 2**: Theoretical difference in cooling capacity HC-blend compared to CFC-12 (the evaporation temperature is defined as the average of the evaporator inlet and dew point temperature)*

This table shows that an increase of the subcooling yields a smaller cooling capacity decrease maintaining the same evaporation temperature. This means that for relatively large subcooling conditions the HC-blend performs relatively better.

## 6. RESULTS

**Running in performance**

As mentioned, the compressors were operated for at least 100 hours before the actual tests were started. During these tests the performance was registered. For all compressors no changes in performance were noted during this period.

**Compressor performance**

In Table 3 the differences in cooling capacity, input power and COP between the HC-blend and CFC-12 are presented.

| Compressor | Cooling capacity [%] | Input power [%] | COP[%] |
|---|---|---|---|
| A | -21 | -19 | -2 |
| B | -12 | -9 | -2 |
| C | -15 | -17 | +3 |

***Table 3**: Differences between HC-blend and CFC-12 at ASHRAE test conditions /1/.*

As can be seen in the table, the cooling capacity decreases from 12 to 21%. As noted, the theoretical calculation predicts a decrease of 13%. The cooling capacity decreases measured for compressor B and C correspond relatively well with this 13% value. For compressor A, the capacity reduction is 21%, which is substantially larger than for B and C. The cause of this effect will be discussed later. The input power of the compressors decreases with approximately the same value as the cooling capacity which results in only small changes in COP values.

In Figure 3 differences in cooling capacity, (ASHRAE conditions for condensation temperature, evaporator superheating and subcooling /1/), are given over an evaporation temperature range from –35°C to 7°C. Comparing these results with the theoretical differences from Figure 2 (“average”), yields the conclusion that the experiments show the same trend over the evaporator temperature range; the higher the evaporation temperature, the smaller the decrease in capacity. At relatively high evaporation temperatures the actual capacity decrease is lower than theoretically predicted. This difference could have been caused by a temperature effect of the suction gas. Namely, the suction gas is heated by the compressor itself before it actually enters the cylinder. This decreases the density of the gas inside the cylinder and therefore also the massflow of the compressor. A calculation was performed to show that at an equal cooling capacity the HC-blend suction gas entering the cylinder would be colder than the CFC-12 refrigerant. This effect is not taken into account in the theoretical model which explains why the difference in cooling capacity is smaller than predicted.

It is remarkable that for compressor A the decrease is relatively large at evaporation temperatures lower than –20°C. To investigate this effect a regression model has been applied using the following formulas:

$$Q = \alpha \rho f V_{swept} \eta_v \Delta h \tag{2}$$

Where the volumetric efficiency is defined by:

$$\eta_v = 1 - \frac{V_{dead}}{V_{swept}} \left( \left( \frac{p_d}{p_s} \right)^{\frac{1}{n}} - 1 \right) \text{ where } n_{CFC\text{-}12} = 1.13 \text{ and } n_{HC\text{-}blend} = 1.10 \tag{3}$$

As can be seen in formula (3), the pressure ratio of the refrigerants affects the volumetric efficiency. In Figure 5 the pressure ratios are given versus the evaporation temperature. From this figure one can conclude that the pressure ratio of the HC-blend is relatively high at low evaporation temperatures. At a low (-35°C) evaporation temperature the HC-blend yields a 5% higher pressure ratio, while at +5°C the pressure ratios are similar both for CFC-12 and the HC-blend. As shown above, the polytropic indexes of both refrigerants are not equal; the lower polytropic index for the HC-blend also yields a negative effect on the volumetric efficiency for the HC-blend.

The model is fitted to the experimental data with $\alpha$ and $V_{dead}/V_{swept}$ as the regression coefficients. In this way the ratio between the dead volume and the swept volume can be estimated. In Table 4 these ratios are given as well as the theoretical volumetric efficiencies at –30°C evaporation and 55°C condensation temperature (assuming the suction gas temperature entering the cylinder to be approximately 70°C).

| Compressor | Dead volume to swept volume ratio [-] | Volumetric efficiency CFC-12 [-] | Volumetric efficiency HC-blend [-] | Difference [%] |
|---|---|---|---|---|
| A | 0.065 | 0.41 | 0.32 | -22 |
| B | 0.053 | 0.52 | 0.45 | -13 |
| C | 0.035 | 0.69 | 0.64 | -7 |

***Table 4***: *Theoretical calculations for CFC-12 and HC-blend for each compressor.*

It can be noted that the dead volume calculated for compressor A is relatively large with respect to the other compressors. Due to this relatively large dead volume the volumetric efficiency of this compressor is relatively small at low evaporating temperatures, resulting in reduced cooling capacities. This is once more illustrated in Figure 4 where a graphical presentation is given with the theoretical difference in volumetric efficiency between CFC-12 and the HC-blend. In this figure it can be noted that for compressor A a drop in volumetric efficiency is particularly present at low evaporation temperatures. This effect explains experimental results, because for the cooling capacity (see Figure 3), a reduction in capacity was also found at low evaporation temperatures for compressor A.

Figure 6 gives COP differences for tests performed at 45, 55 and 69°C condensation temperatures. From this figure it can be concluded that for high evaporation temperatures the COP for the HC-blend is higher than for CFC-12 (a maximum COP increase of 17% was found at 7.2°C evaporation temperature). Concerning efficiency the HC-blend becomes a good option for high temperature applications.

## 7. CONCLUSIONS

The performance of three compressors was evaluated having converted from CFC-12 to a HC-blend. Test results and theoretical calculations yield the following conclusions:

- For the HC-blend the compressor data measured and presented are sensitive to the definition of the evaporation temperature. The average temperature between the evaporator inlet and the dew point temperature best represents the actual situation in the evaporator and has been selected in this study.

- The cooling capacity of the HC-blend is 7 to 35% smaller than for CFC-12 over the evaporation temperature evaluated. This reduction could be critical in applications where the blend is applied as a drop-in and where the cooling capacity is already hardly sufficient. The ratio between dead volume and the swept volume plays a role here: the higher the ratio the higher the drop in cooling capacity due to the smaller volumetric efficiency. A trend in cooling capacity difference is noted (by the tests and the theoretical calculation); the higher the evaporation temperature the lower the reduction in capacity applying the HC-blend. It needs mentioning that for relatively large subcooling conditions the HC-blend performs relatively better.
- Regarding the efficiency (COP) it can be concluded that the HC-blend yields higher efficiencies at relatively high evaporation temperatures. With the blend an energy consumption reduction can be obtained at this temperature level, while the opposite effect is noted at low evaporation temperatures. This may also explain why different results are reported in various studies /4/.

## ACKNOWLEDGEMENTS

The authors are grateful to FKW, Hannover, Germany and INFRAS, Zürich, Switzerland for their support of and approval for this publication.

## REFERENCES

/1/ ASHRAE standard test condition: -23.3°C evaporation temperature, 55°C condensation temperature; subcooled liquid 32°C and suction gas temperature 32°C

/2/ ISO 917, “Testing of refrigerant compressors”, Second edition, September 15, 1989.

/3/ NIST, “REFPROP 5.1”: Refrigerant property data, 1996.

/4/ UNEP, United Nations Environment Programme “Study on the Potential for Hydrocarbon Replacements in Existing Domestic and Small Commercial Refrigeration Appliances, part 1: HC-desk survey”, January 1999, (ISBN 92-807-1765-0).

## FIGURES

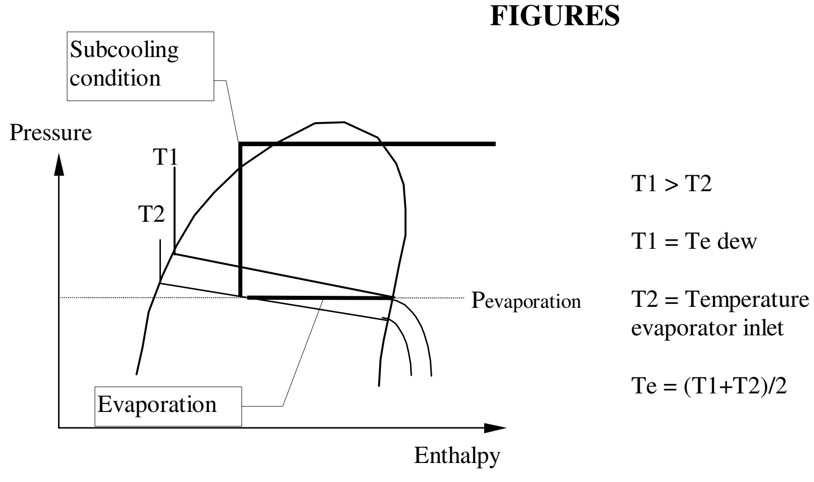


***Figure 1**: Pressure-enthalpy diagram of a blend.*

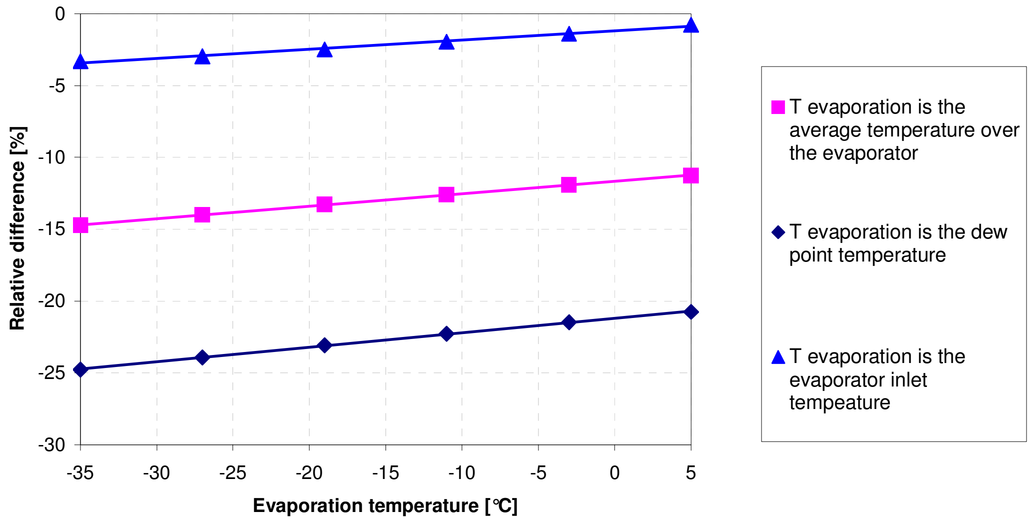


***Figure 2****: Relative difference between capacity of the HC-blend and CFC-12 representing the sensitivity to the evaporation temperature definition (55°C condensation, 32°C evaporator inlet and 32°C subcooled liquid conditions).*

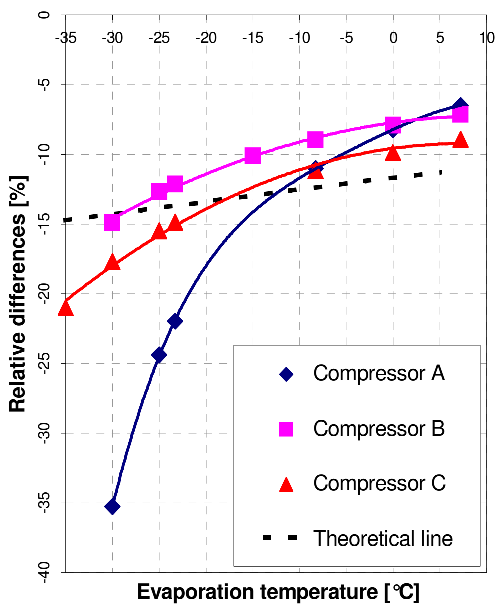


***Figure 3****: Relative difference between the cooling capacity of HC-blend and CFC-12 at 55°C condensation. Also the theoretical line (see also Figure 2) based on the average evaporation temperature is given (the evaporation temperature is taken as the average between the inlet and the dew point temperature).*

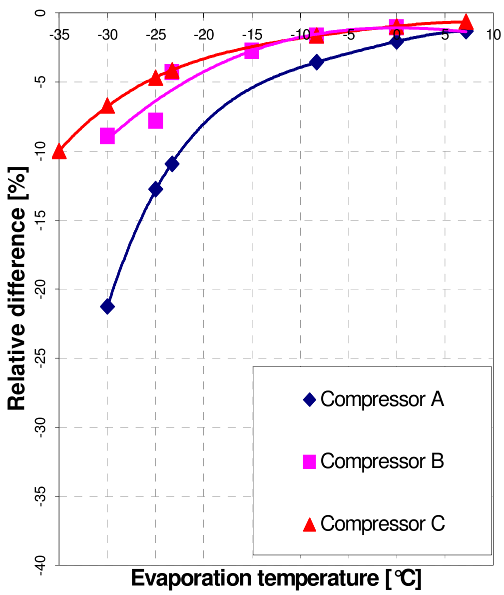


***Figure 4****: Relative difference between the theoretical volumetric efficiency of the HC-blend and CFC-12 at 55°C condensation temperature (for the evaporation temperature the average between the inlet and the dew point temperature is taken).*

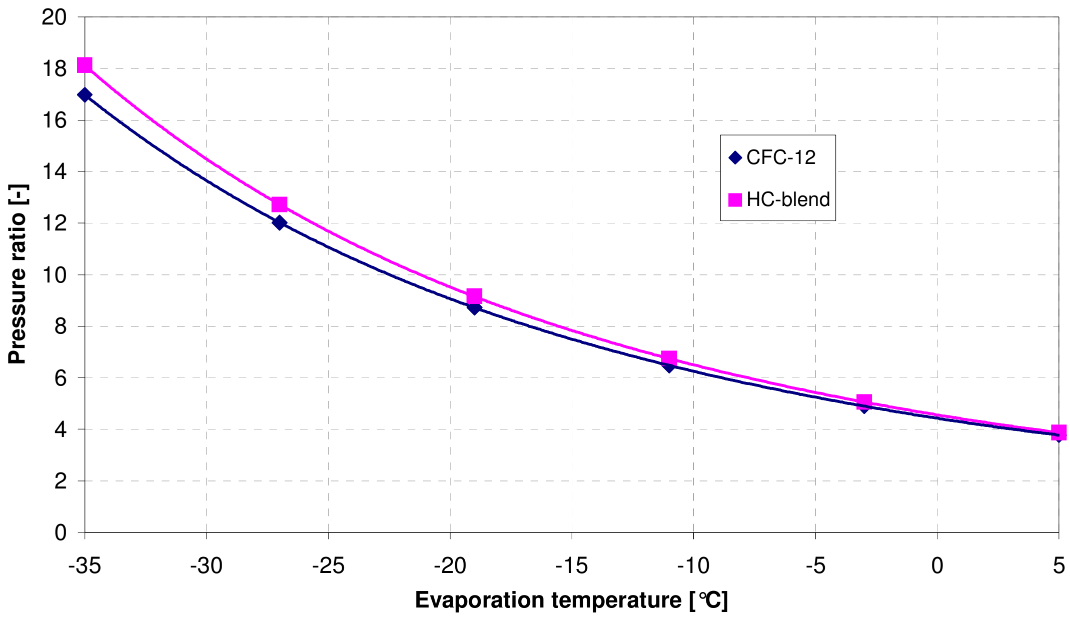


*Figure 5: Pressure ratios of CFC-12 and HC-blend (for the evaporation temperature the average between the inlet and the dew point temperature is taken).*

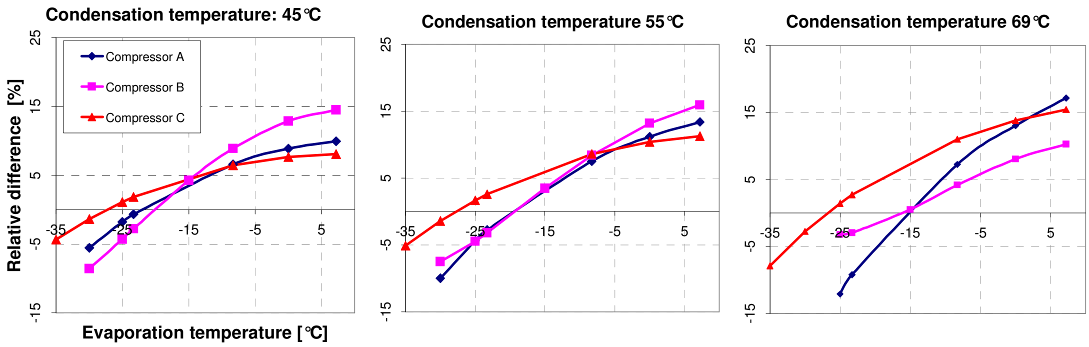


*Figure 6: Relative difference between the COP of the HC-blend and the COP of CFC-12 at 45, 55 and 69°C condensation temperature (for the evaporation temperature the average between the inlet and the dew point temperature is taken).*