GUIDELINE

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Refrigerant Glide and Effect on Performances Declaration

Scope and Purpose

As a consequence of the EU regulation 517/2014 (F-gas), the future synthetic refrigerants with a low global warming potential will be to a large degree non-azeotropic mixtures with a significant temperature glide (some over 6 K). These refrigerants change temperature during the evaporating and condensing process at constant pressure. Many works comparing the performances of different refrigerants have been published referring to mid-point temperature where no evident correlation between mid-point and pressure can be easily determined. The scope of this guideline is to give an overview of the relation between the dew and mid temperature for non-azeotropic mixtures, highlight the difference in performance and give a recommendation for converting the performance of the compressor regulated by the existing standards to mid-temperature performance. In this case, any misunderstanding and wrong interpretations of the result are avoided.

1. Refrigerant Mixtures Classification

Refrigerant blends are mixtures of two (binary) or more pure refrigerant fluids. The refrigerant mixtures can be divided into:

1. Azeotropic refrigerant mixtures
2. Near-azeotropic refrigerant mixtures
3. Non-azeotropic refrigerant mixtures, also known as zeotropic mixtures

Near-azeotropic mixtures are mixtures with small temperature variation during phase change and a small difference in composition in liquid and vapour phases at equilibrium. The widely used refrigerants R404A and R410A belong to this class.

a) Azeotropic Mixtures

The azeotropic (refrigerant) mixtures are usually binary mixtures that behave like a pure fluid, i.e., under constant pressure they condense and evaporate at a constant temperature and the composition of the mixture in the vapour and liquid phases are considered to be the same. In a refrigeration cycle represented in a p-h diagram (figure 1), the compressor raises the pressures from the evaporating to the condensing pressure. Evaporation and condensation happen at constant pressure during the process change. A single temperature defines either the evaporation or the condensing pressure.

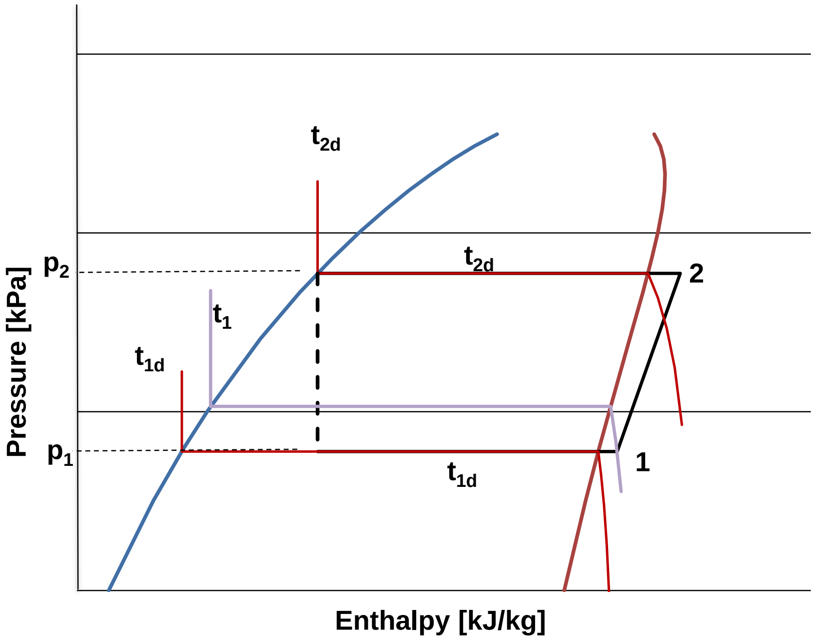


Figure 1: p–h diagram for azeotropic mixture
evaporating pressure: p_1 (t_{1d}), condensing pressure: p_2 (t_{2d})

b) Non-Azeotropic Mixtures

Non-azeotropic (refrigerant) mixtures, also known as zeotropic mixtures, exhibit a significant temperature variation during constant pressure phase change, such as condensation and evaporation. Also, at equilibrium the composition in vapour and liquid phases will be different. The term "glide" is widely used to describe the temperature change during the evaporation and condensation process.

Figure 2 shows a p-h diagram representing the refrigeration cycle with a non-azeotropic mixture. The condensing pressure p_2 and evaporating pressure p_1 are considered to be constant throughout the change of state process. The lines of constant temperature are sloping, as illustrated in the figure. The temperature at which condensation starts is called the dew point, denoted here as t_{2d}. As condensation progresses, the temperature falls to t_{2f} (bubble point). During the evaporation process, the temperature changes from the temperature at the inlet of the evaporator t_{1e} to the dew point temperature t_{1d}. Superheat occurs after evaporation is complete, raising the temperature to t_1 (suction temperature at the compressor inlet). Compressors are rated according to this cycle, with the evaporating and condensing pressures expressed as dew point temperatures.

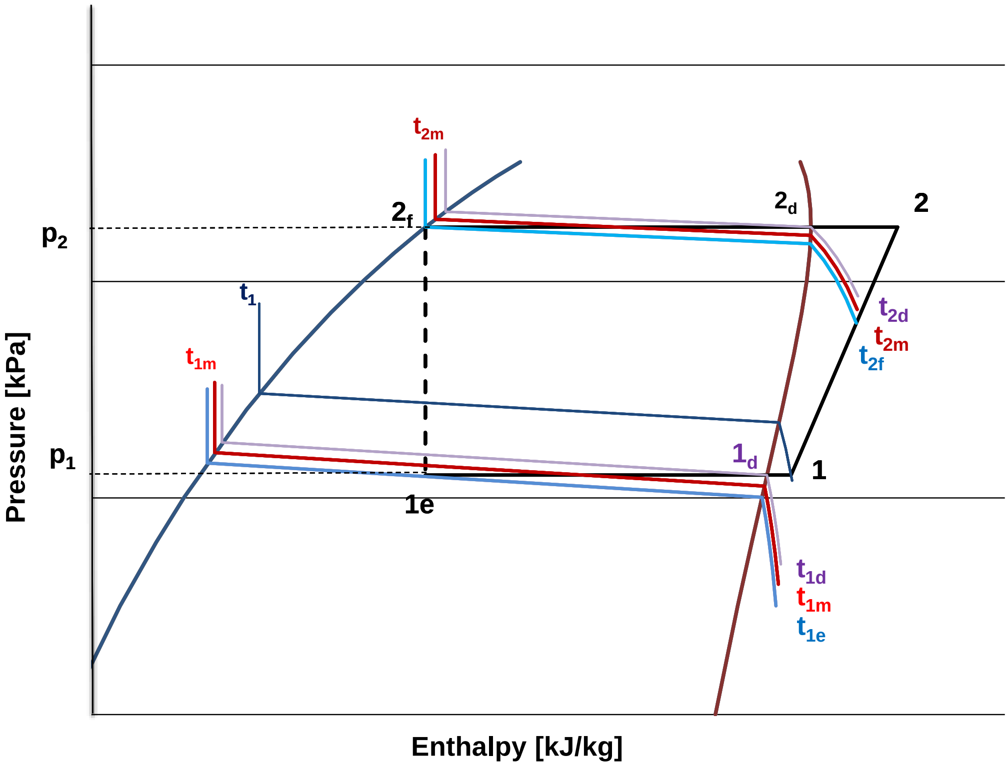


Figure 2: p-h diagram for non-azeotropic mixture

The question then arises as to which temperature along each change of state process should be used to define the evaporating and condensing temperatures. A mean temperature may be defined for purposes of analysis to represent the actual system performance or for comparing blends with pure refrigerants. Compressor standards agree for using dew point temperatures because they allow for a clear correlation between pressures and temperatures.

2. Performances Declaration

a) Dew Point Protocol

The evaporating and condensing temperatures are defined as the dew temperatures t_{1d} and t_{2d} as reported in figure 2. A single temperature now defines the compressor inlet (evaporation) pressure and it is independent of the condensation process. The definition of superheat is easily calculated as the difference between compressor suction temperature and evaporating temperature. The liquid sub-cooling is however still calculated with respect to the bubble point.

b) Mid-point Protocol“

The condensing temperature could be defined as the arithmetic mean of dew t_{2d} and bubble temperatures t_{2f}, and the evaporating temperature is likewise defined as the arithmetic mean of t_{1e} and t_{1d}.
For condensing, at given discharge pressure the dew temperature t_{2d} and the bubble temperature t_{2f} are fixed, hence the mid temperature is only pressure dependent. Therefore, reference can be easily made for the mid temperature: $t_{2m} = (t_{2f} + t_{2d}) / 2$.

The average evaporating temperature t_{1m} is the average temperature between the inlet temperature in the evaporator t_{1e} and saturated vapour temperature t_{1d}: $t_{1m} = (t_{1e} + t_{1d}) / 2$.
The evaporator inlet temperature, and hence mid evaporating temperature, changes with the condensing pressure as illustrated in Figure 3. Similarly, the mid evaporating temperature is also dependent on the extent of sub-cooling. Measurement of the evaporating pressure is thus no longer sufficient to define the mid evaporating temperature and t_{1e} is expressed as function of the evaporating pressure, the condensing pressure and the sub-cooling $t_{1e} = f(p_1, p_2, \text{sub-cooling})$. Superheat definition can also be misinterpreted when using mid-point data. Superheat is the difference between the temperature at the compressor suction inlet, t_1 and the (dew point) temperature at the end of the evaporation process, t_{1d}. When the evaporating temperature is defined as mid-point, the dew point temperature has to be calculated before the superheat can be found – $t_{1d} = t_{1e} + 2(t_{1m} – t_{1e})$. Obviously, the lack of correlation between the evaporating temperature and the evaporating pressure renders this approach somewhat difficult.
Hence, it is obvious that any performance related to mid-point temperature could create a misunderstanding if insufficient information is given.

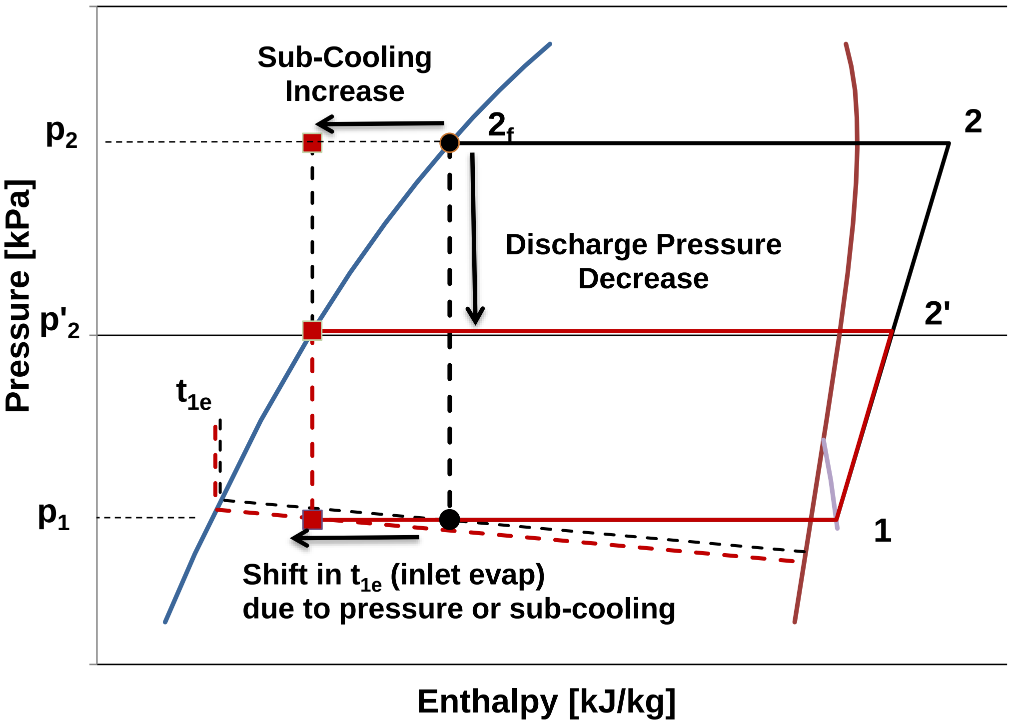


Figure 3: Effect of condensing pressure or the sub-cooling on evaporating mid-temperature

When an economizer cycle is applied (Fig. 4), the mid-temperature t'_{1m} depends on the outlet temperature t_{10} of the economizer liquid sub-cooler. Hence, the mid-temperature is changing with the sub-cooling at the same evaporating and condensing pressure. Thus, referring to mid-temperature introduces a further complication for economizer application.

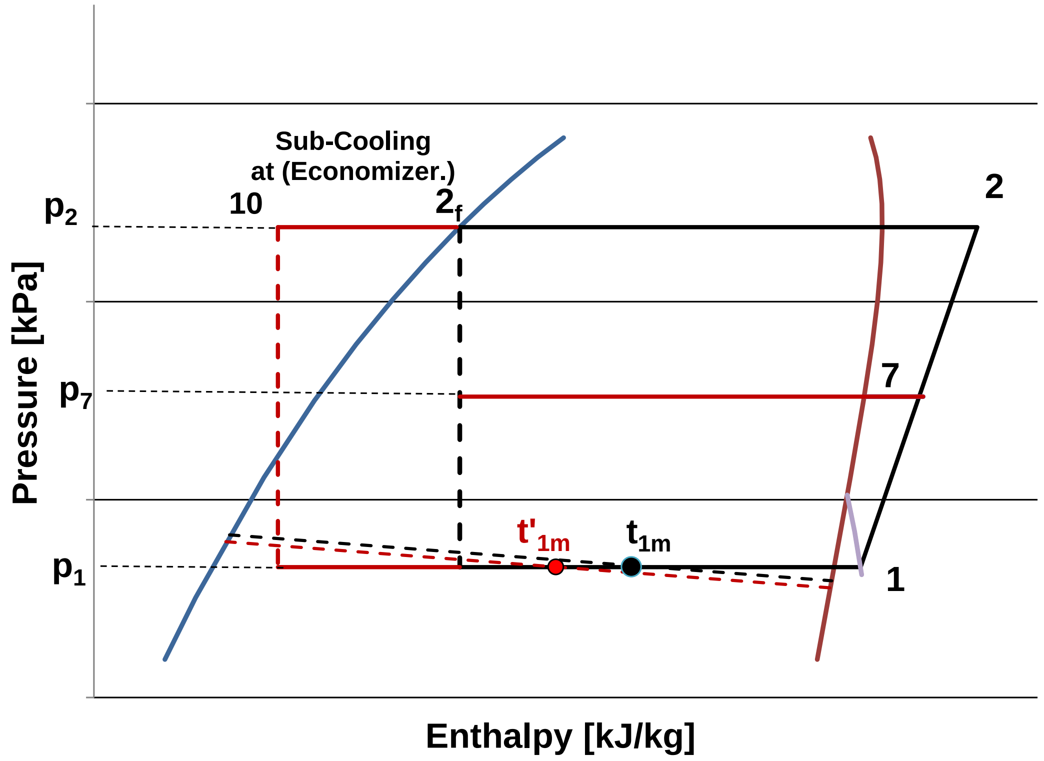


Figure 4: Effect of the Economizer outlet temperature on
Evaporating mid-point temperature

c) Compressor Performances

The compressor's performances declaration in Europe and North America are regulated respectively according to the standard EN 12900 and to ARI 540. The declared performance according to these standards shall comprise the refrigerating capacity or mass flow and the absorbed power, at dew points evaporating and condensing temperatures. The polynomial equation reported below is used either by EN 12900 as well by ARI 540 to generate the performances of the compressor:

$$X = C1 + C2 \cdot (S) + C3 \cdot D + C4 \cdot (S^2) + C5 \cdot (S \cdot D) + C6 \cdot (D^2) + C7 \cdot (S^3) + C8 \cdot (D \cdot S^2) + C9 \cdot (S \cdot D^2) + C10 \cdot (D^3)$$

where:
X is the refrigerating capacity (only EN 12900), absorbed power or mass flow, current
S is the evaporating temperature <u>at suction dew point</u>
D is the condensing temperature <u>at discharge dew point</u>,
C is a coefficient.

2) Performance Comparison

Figure 5 shows an example of the difference between the mid and dew temperatures for R407C. It proofs how the condensing temperature affects the mid evaporating temperature. The 2 lines are calculated at constant dew evaporating temperature.

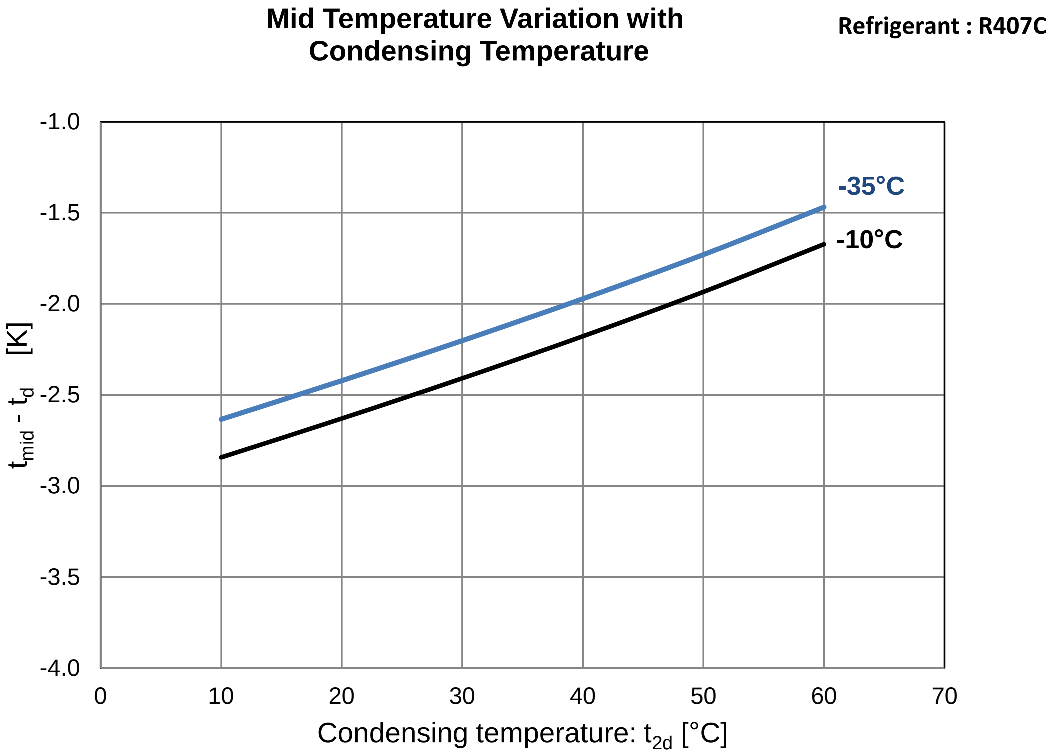
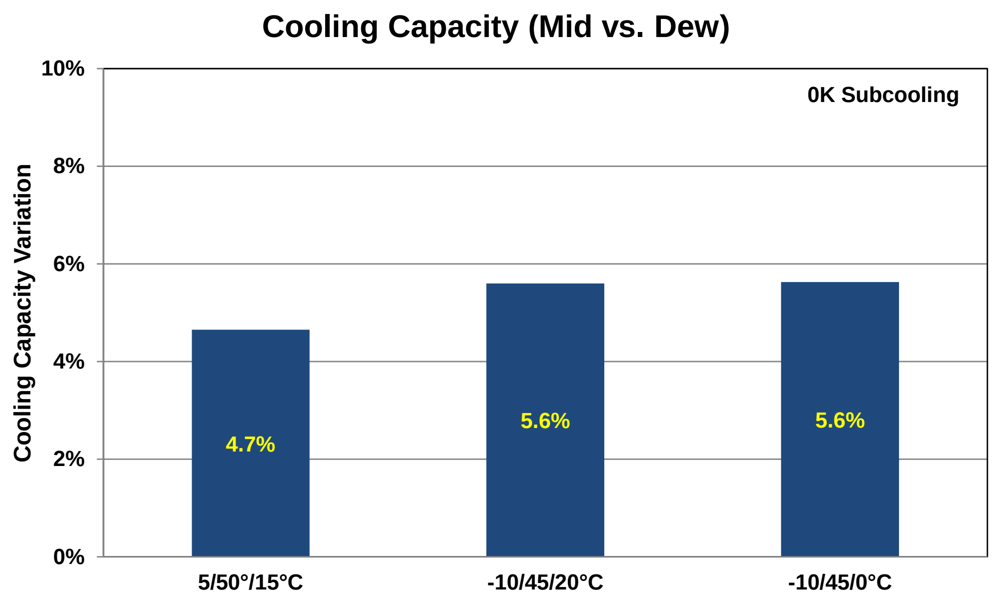


Figure 5: Mid vs. Dew evaporating temperature at various condensing pressure

Figure 6 illustrates the performance variation using the two approaches for a typical reciprocating or scroll compressor at -10 °C evaporating and 45 °C condensing temperatures. It can be seen that the capacity is lower by approximately 5 % for dew temperature reference with no appreciable difference in the COP. The system designer may properly interpret the data from the appropriate definition, but a casual observer may conclude that the compressor delivers less capacity when dew point definitions are used, although this is not the case.



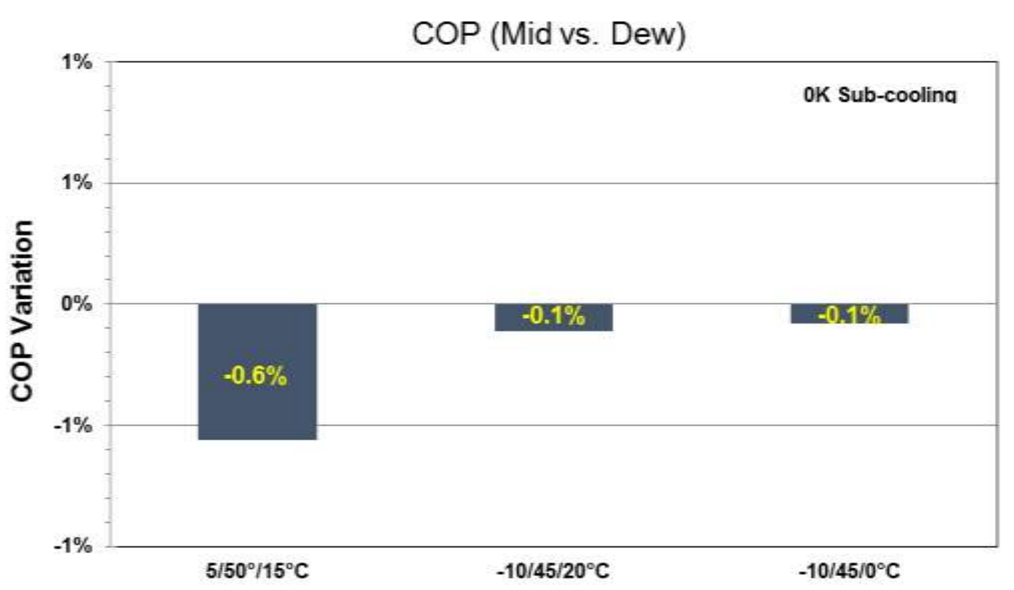


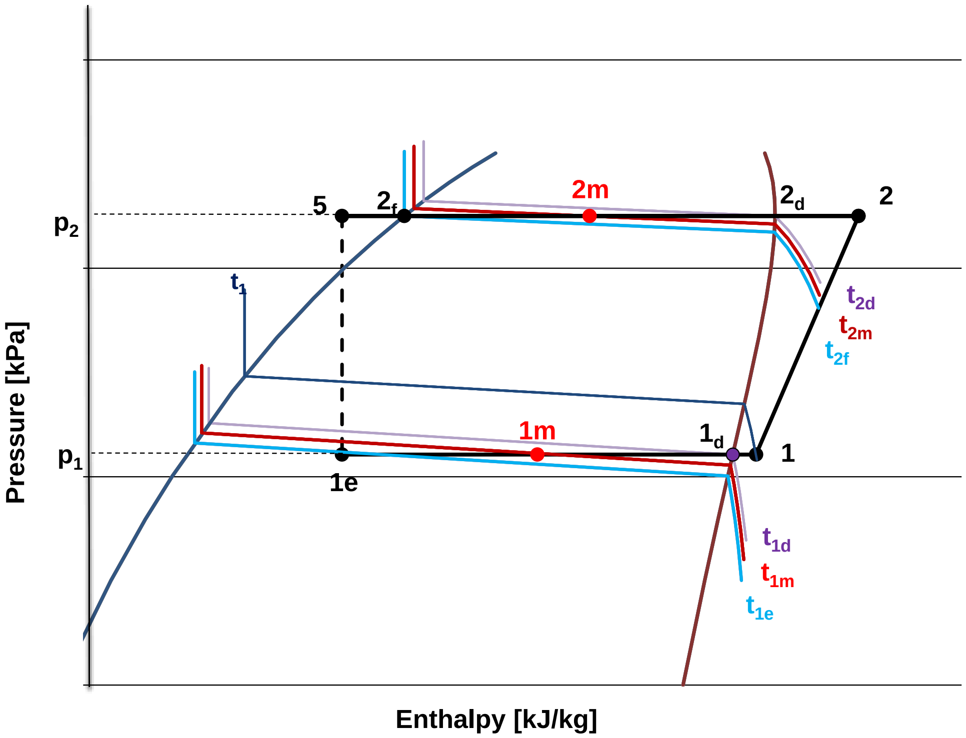
Figure 6: Reciprocating and Scroll Compressor performance variations according to dew and mid temperatures

Recommendations of using Mid-point temperature

This is a recommended method to convert dew (point) temperatures to mid-point temperatures. It allows comparing compressor performance data for different refrigerants.

Condensing mid temperature:	$t_{2m} = (t_{2f} + t_{2d}) / 2$
Sub-cooling:	$Dt_{sub} = t_{2f} - t_5$
Evaporating mid temperature:	$t_{1m} = (t_{1e} + t_{1d}) / 2$
Gas Superheat at the compressor inlet	$t_{sh} = (t_1 - t_{1d})$

The above temperature conversion provides more than a rough comparison of *compressor performance data.*

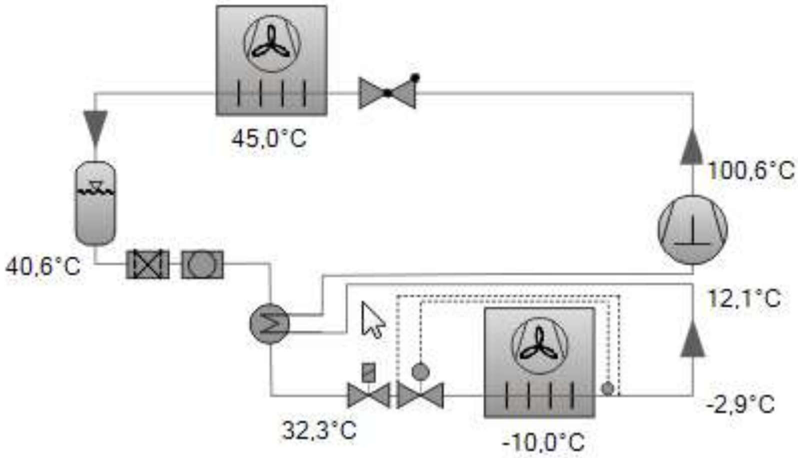


Nomenclature

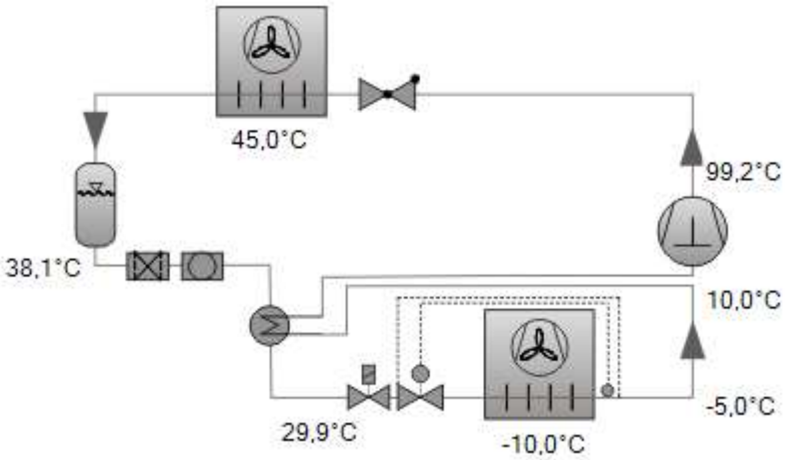
1 : compressor inlet
2 : compressor outlet
d: dew
m: mid
f: fluid
2_f: liquid refrigerant at bubble point corresponding to the compressor discharge pressure

5: refrigerant fluid at the inlet of the expansion device
7: intermediate pressure (economizer)
10: liquid outlet of the economizer
t1: refrigerant gas temperature at the inlet of the compressor
t_{1d}: dew point temperature at suction pressure
t_{2d}: dew point temperature at discharge pressure
t_{1m}: mid-point temperature at suction pressure
t_{2m}: mid-point temperature at discharge pressure
t_{1e}: refrigerant temperature inlet at the evaporator

Example **mid**-point: R407C // -10/45/SH after evap. 5 K total 20 K/SC inside condenser 2 K



Example **dew** point: R407C // -10/45/SH after evap. 5 K total 20 K/SC inside condenser 2 K



Reference

- EN 12900:2013 - Refrigerant compressors – Rating conditions, tolerances and presentation of manufacturer's performance data
- EN13771-1: 2003- Compressors and condensing units for refrigeration - Performance testing and test methods — Part 1: Refrigerant compressors
- ARI standard 540: 2004 – Performance rating of positive displacement refrigerant compressors and compressor units