Dealing With Glide Refrigerants

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This article explains the effect of glide on the basic refrigeration cycle and the presentation of compressor performance data

Introduction

It is well known that a pure substance has a fixed boiling point. The temperature of the boiling water in a kettle is the same no matter how much water remains in it. With a two or more component mixture the boiling point depends on the proportion of the components, and the composition will change to some extent during the boiling process – an effect which enables liquid composition to be perfected to a fine art by many distilleries. R502 was the first mixture to be widely adopted and it has the special property that there is either no change or very little change in the boiling point during the evaporation process. For all practical purposes, it may be treated as a single substance. The refrigerant classification committee of ASHRAE, designated a special refrigerant series, the R500 series for refrigerants of this type, which are called azeotropes.

With the phase out of chlorine containing refrigerants, a new look was taken at mixtures having a measurable temperature change during the boiling process. They are termed zeotropic refrigerants, and the R400 series was adopted for this category. The term “glide” is widely used to describe the temperature change – the temperature “glides” from one value to another during the evaporation and condensation processes. Mixtures having different properties can be made using one set of components in various proportions. The best known example is the R407 series. R407 tells us that the refrigerant is a mixture of R134a, R125, and R32. In order to standardise on certain component percentages suffices A, B, C and so on are used.

With R404A and R410A the glide is small and normally neglected, but with R407C it can amount to 7K and this is large enough to have an effect on the refrigeration cycle and the description of compressor performance.

Note that the upper case **A**, **B**, or **C** denotes different percentage compositions of given components, whereas the lower case “**a**” in R134a denotes a specific chemical isomer of a single substance.

Cycle Definition

For R407C the shape of the well known Pressure Enthalpy refrigerant diagram is unchanged, as shown in Figure 1. The condensing pressure P2 and evaporating pressure P1 are considered to be constant throughout the change of state process.

The lines of constant temperature are now sloping, as shown in blue and red. The temperature at which condensation starts is called the dew point, denoted here as T2(Dew). As condensation progresses, the temperature falls to T2(Bubble). The temperature during the evaporation process changes from T1(Evaporator Inlet) to T1(Dew), because the lighter components, R32 and R125, evaporate preferentially to the R134a, and so the remaining liquid becomes R134a rich, its boiling point gradually increasing until all the liquid is evaporated. Note that this does not mean that the lighter components boil away leaving liquid R134a at the very end of the process. The composition shift during the process is limited and quite small. Further superheat then occurs after evaporation is complete, raising the temperature to Ts, the suction temperature at the compressor inlet.

Compressors are rated according to this cycle, with the evaporating and condensing pressures expressed as saturation temperatures. The question arises as to which temperature should be used for the definition of saturated discharge temperature and saturated suction temperature.

Mid-point

The first and perhaps most logical definition is the mid-point or mean-point. The condensing temperature is defined as the arithmetic mean of T2(Dew) and T2(Bubble), and the evaporating temperature is likewise defined as the arithmetic mean of T1(Evaporator Inlet) and T1(Dew). The value of temperature midway through the evaporating and condensing process is used, and compressor data presented in this way gives the truest definition for comparison with R22.

The evaporator inlet temperature, however, changes slightly when the condensing pressure changes as illustrated in Figure 2. Because of this change, there is a change in the evaporating temperature (according to the mid-point definition) when the condensing condition shifts, although the evaporation pressure has not changed. A similar effect occurs if there is an addition of sub-cooling. Measurement of the evaporating pressure is no longer sufficient to define the evaporating temperature – and this is likely to lead to difficulties because most people are used to being able to take just a suction pressure gauge reading to find evaporating temperature.

Superheat definition can also be misinterpreted when using mid-point data. Superheat is the difference between the temperature at the end of the evaporation process, T(Dew) and the temperature at the compressor suction inlet, Ts. When the evaporating temperature is shown as mid-point, the dew point temperature has to be calculated before the superheat can be found.

There is a need for a definition whereby a single evaporating temperature can be associated with the evaporating pressure.

Dew-Point

With this definition, the evaporating and condensing temperature are defined as T1(Dew) and T2(Dew). A single temperature now defines the evaporation pressure and it is independent of the condensation process. The definition of superheat once again becomes (compressor suction temperature – evaporating temperature). Note that the reference for liquid subcooling is still the end of the condensation process, or bubble point. The subcooling is always defined as the temperature drop of the liquid from this point.

Compressor Data

Today, most R407C compressor data uses the mid-point definition. A frequently used rating reference is the ARI condition, 11.1K superheat and 8.33K subcooling. When interpreting data of this type the above comments need to be taken into account.

For performance comparison purposes it is always best to work with zero subcooling, because this avoids any ambiguity in the definition. The effect of changing a given compressor data set from mid-point to dew-point format is to make an adjustment to the values of the saturated evaporating and condensing temperatures. From the point of view of the tabular data user, a condition of say, -10º C/ +45º C now refers to slightly different pressures, P1 and P2. The new pressures are a little lower than previously, and this results in slightly lower capacity figures for the stated operating condition. The chart shown in Figure 3 may be used as a guide to obtaining dew point data from the mid-point data for a typical scroll compressor using R407C.

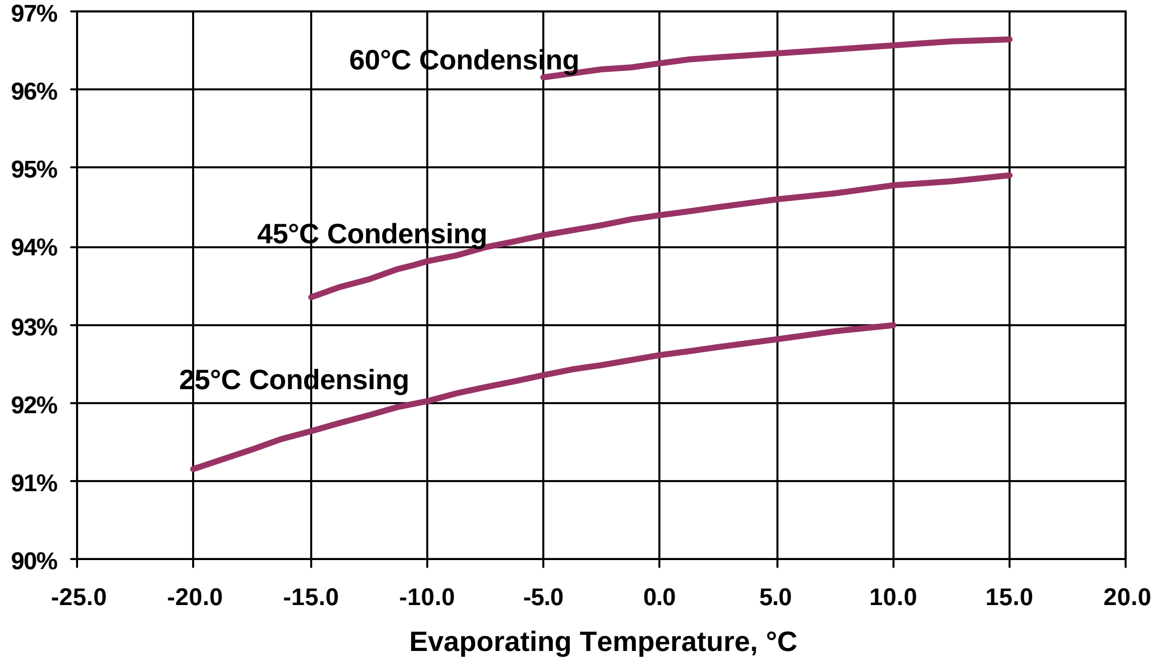
Standard Rating Conditions

The International Standard ISO 9309 and the European Standard EN12900:1999 for Presentation of Manufacturers' Performance Data specify dew point for the definition of evaporating and condensing temperatures. The rating conditions should be 10K superheat and zero subcooling. 20°C suction gas temperature is also allowable, but 10K is more appropriate for most air conditioning applications. Both the ARI and EN standards refer to specific high evaporating temperature (air conditioning) rating points and Table 1 shows typical scroll performance relative to the commonly quoted ARI mid point definition, 7.2°C Evaporating, 54.4°C condensing. The new ARI standard (ARI 554, 1999) also refers to dew point conditions.

Data for an individual compressor changes by the amount shown in the table when moving from the old mid point definition, because of the rating condition differences. The system designer can interpret the data from the appropriate definition, but the casual observer may conclude that the compressor delivers less capacity when dew point definitions are used, and this is not the case.

In the past all Copeland R407C compressor data was presented using the mid-point definition. Now, with Copeland SELECT software version 4.0, R407C data is available in dew-point or mid-point format with the computer taking out the need for complex calculation for parameters at different points in the cycle.

Figure 3. Dew point capacity value as a percentage of mid point value for a typical scroll compressor



Rating Condition	Capacity	Power	COP
MID-POINT RATING POINT, 7.2/54.4 °C **(ARI, 1991)** Conditions, 11.1K Superheat, 8.3K Subcooling	**100**	**100**	**100**
MID-POINT RATING POINT, 7.2/54.4°C At 10K Superheat, Zero Subcooling	**- 10%**	**- 0%**	**- 10%**
MID-POINT RATING POINT, 5/50°C Conditions, 10K Superheat, Zero Subcooling	**- 11%**	**- 9%**	**- 2%**
DEW-POINT RATING POINT, 7.2/54.4 °C **(ARI, 1999)** Conditions, 11.1K Superheat, 8.3K Subcooling	**- 5%**	**- 5%**	**- 0%**
DEW-POINT RATING POINT, 7.2/54.4°C at 10K Superheat, Zero Subcooling	**- 14%**	**- 5%**	**- 9%**
DEW-POINT RATING POINT, 5/50°C **(EN12900)** Conditions, 10K Superheat, Zero Subcooling	**- 15%**	**- 14%**	**- 2%**

Table 1. Scroll Compressor Performance Relative to Mid-Point ARI Air Conditioning Point

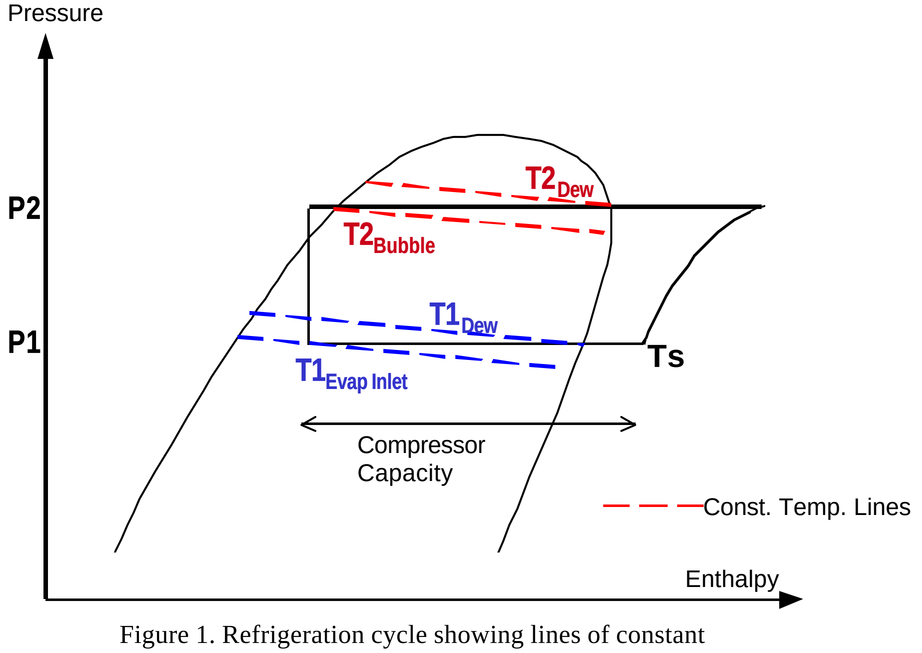


Figure 1. Refrigeration cycle showing lines of constant temperature in the two phase region

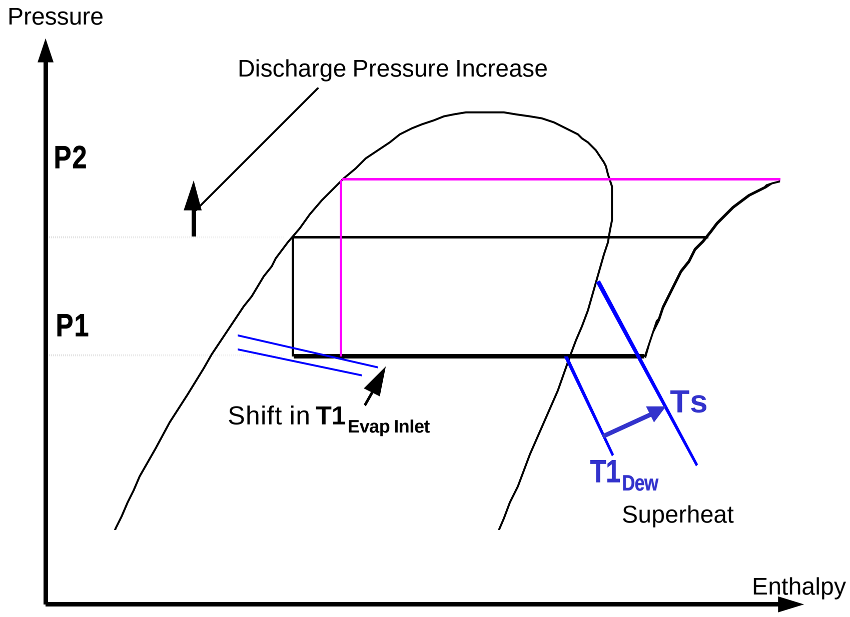


Figure 2. Refrigeration cycle illustrating the shift in evaporator inlet temperature caused by discharge pressure increase, and the definition of superheat

Figure 3. Dew point capacity value as a percentage of mid point value for a typical scroll compressor