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Compressor Performance Definition for Refrigerants with Glide

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Abstract

Compressor performance data is normally expressed in terms of saturated suction and discharge temperatures. For glide refrigerants such as R407C there is a need to determine the appropriate temperatures to define the suction and discharge conditions. The midpoint approach advocates the use of the temperatures that are midpoints of the condensation and evaporation processes, as plotted in a p-h diagram. The midpoint protocol will give a good system performance comparison with non-glide refrigerants such as R22 in many systems, and has been widely used. However, for compressor data, the dew point protocol has advantages. The paper compares the results, in terms of compressor capacity and COP (EER) values when the midpoint and dew point definitions are applied to specific compressor rating points. The comparison is extended to a range of suction and discharge conditions, and the effects of superheating and subcooling are considered. The paper makes reference to International, European and ARI standards for the presentation of compressor performance data.

Introduction

It is well known that a pure substance has a fixed boiling point, but with a mixture comprised of two or more components, the boiling point depends on the proportion of the components. Also, the composition will change to some extent during the boiling process. R502 was the first refrigerant mixture to be widely adopted and it has the special property that there is little or no change in the boiling point during the evaporation process. For all practical purposes, it may be treated as a single substance and such refrigerants are classified as azeotropes.

Some new refrigerant mixtures (R400 series) that are used in lieu of the chlorinated refrigerants have a measurable temperature change during the phase change processes. Such refrigerants are classified as zeotropic refrigerants. 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, which is a mixture of R134a, R125, and R32. The suffices A, B, C and so on are used to designate the relative composition of the various constituents, while the suffix 'a' in R134a denotes an isomer. While in the case of R404A and R410A the glide is small and normally neglected, the glide associated with R407C can be as much as 7K. This is large enough to have an effect on the refrigeration cycle and the description of compressor performance.

Whilst this paper is primarily concerned with compressor performance, it is system performance that is the final objective, and so it is appropriate to mention that there has been significant work comparing the performance of R22 and R407C. Y. Hwang et al. [1] have reported that R407C, which has vapor pressures similar to that of R22 has a 6.8% lower cooling seasonal performance and a 4.7% lower heating seasonal performance compared to R22. As pointed out by the authors in their paper, the results of this study compare well with results presented by AREP (Godwin 1993 [2]). Xin Liu [3] clarifies that the temperature glide of non-azeotropic refrigerants benefits a finite heat source or sink (water-cooled machine) by matching the temperature change of the water or air streams passing through a heat exchanger while there is a penalty for an infinite heat source or sink (air-cooled machine). Haselden [4] has demonstrated significant system COP benefits using a purpose designed system, which exploits the potential of mixture properties.

Cycle Definition

For R407C, the shape of the well-known Pressure-Enthalpy (p-h) refrigerant diagram is 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 sloping, as illustrated in the figure. 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. The composition shift during the process is however, quite small. Further superheat 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. These are the temperatures used to define the rows and columns of tabulated compressor data, where they are normally shown as "Evaporating Temperature" and "Condensing Temperature". 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. This "real mean temperature" would then be that temperature, which if it prevailed along the entire length of the heat exchanger, would result in the same heat transfer rate as that associated with the actual temperature distribution. The Alternative Refrigerants Evaluation Program (AREP) Technical Committee, in its report published in January 1995 examined several approaches for rating compressors. Included in this report is an overview of the methods for rating compressors with zeotropes by Richard E. Cawley [5]. He concludes that it is not practical to use a rating system for zeotropes that will allow direct comparison with pure refrigerants. Baxter and Rice [6] point out that the use of mean temperature is of greater relevance to system designers. They also recommend that the superheat and subcooling be calculated with respect to the mean temperatures to simplify the procedure and yet obtain a more general and accurate compressor performance for use in system models. While ASERCOM [7] agrees that a fair comparison is possible by using performance data based on mean evaporating and condensing temperatures, its members expressed preference for using dew point temperatures despite an associated understatement in the compressor performance, because of its relative ease of use. In an earlier work, Connon [8] had also proposed the use of mean condenser and evaporator temperatures for comparing blends with pure refrigerants.

For the purposes of this paper, two commonly used approaches, the mid-point and the dew-point definitions are considered.

Mid-point Protocol

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 temperature value midway through the evaporating and condensing process, as depicted on a p-h diagram, is used.

The evaporator inlet temperature, and hence evaporating temperature, changes with the condensing pressure as illustrated in Figure 2. Similarly, the evaporating temperature is also dependent on the extent of subcooling. Measurement of the evaporating pressure is thus no longer sufficient to define the 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 defined as mid-point, the dew point temperature has to be calculated before the superheat can be found. Obviously, the lack of correlation between the evaporating temperature and the evaporating pressure renders this approach somewhat difficult.

Dew-Point Protocol

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 is easily calculated as the difference between compressor suction temperature and evaporating temperature. The liquid subcooling is however still calculated with respect to the bubble point.

Standard Rating Conditions for Compressors

The current standards for presentation of tabular compressor data, including ARI554, 1999 [9] all specify dew point for the definition of evaporating and condensing temperatures EN12900: 1999[10] states that the rating conditions should be 10K (18°F) superheat and zero subcooling. ISO 9309 [11], currently being drafted, is also expected to specify the same conditions. A suction gas temperature of 20°C (68°F) is also allowable, but 10K (18°F) is more appropriate for most air conditioning applications. Both the ARI and EN standards refer to specific high evaporating temperature (air conditioning) rating points as shown in Tables 1 and 2. The ARI standard specifies a mass flow rating, and while this overcomes possible misunderstandings associated with subcooling, data for capacity and EER will still be needed.

M.Paulus-Lanckriet et al. [14] report that any fractionation of the R407C refrigerant which could result in a variation of composition from the nominal values could affect the performance measured. In their analysis of liquid and vapor composition of refrigerants during calorimetry, Sundaresan and Watkins [15] conclude that no significant fractionation occurred with R407C. They suggest that in the calorimeter, there is very little chance for pooling and with a constant circulation, there is less chance for fractionation.

Conversion to Dew Point Data

Compressor performance data is derived from polynomial equations as defined in the above mentioned standards. Many sets of coefficient data exist for which the performance definition is mid point. These data sets can quite readily be used to generate dew point data. The methodology for this can be explained with reference to Figure 3 by describing the process for a single point, the ARI air conditioning point. The task is to find the compressor capacity using the mid point coefficients, given the dew point evaporating and condensing pressures.

Step 1. Use refrigerant property data to establish the suction and discharge pressures, **P1** and **P2** from the dew point temperatures - 7.2°C/54.4°C (45°F/130°F). This enables the cycle diagram to be drawn, as shown in Figure 4.

Step 2. Now that **P2** is established, the point $\mathbf{T2_{bubble}}$ may be found using refrigerant properties. Calculate mid point condensing temperature, $\mathbf{T2_{mid}}$, by taking the arithmetic mean of $\mathbf{T2_{bubble}}$ and $\mathbf{T2_{dew}}$.

Step 3. Because the temperature at the evaporator inlet, $\mathbf{T1_{evap\ inlet}}$, is unknown, it is necessary to make reference to enthalpy values. The enthalpy at the evaporator inlet $\mathbf{h_e}$ is equal to the liquid enthalpy. With **P1** established, the values $\mathbf{h_b}$ and $\mathbf{h_d}$ may be found from refrigerant properties. This enables $\mathbf{T1_{evap\ inlet}}$ to be calculated using the following equation in which all the other values are now known:

$$\frac{T1_{evapinlet} - T1_{bubble}}{T1_{dew} - T1_{bubble}} = \frac{h_e - h_b}{h_d - h_b}$$

Calculate mid point evaporating temperature, $T1_{mid}$, by taking the arithmetic mean of $T1_{evap\ inlet}$ and $T1_{dew}$. It needs to be noted that there is an approximation insofar as the enthalpy is not a linear function of temperature. Strictly speaking, an iterative process should be used. Investigation has shown that the maximum error in evaporating temperature is 0.1K (0.2°F) which has a negligible impact on performance values.

Step 4. Use the mid point temperatures established in steps 2 and 3 in the mid point polynomial equations to obtain the capacity (and other parameters) from the mid point coefficients.

The mid point temperatures thus found, 5.3°C, 52.2°C (42°F, 126°F) are lower than the dew point values in Step 1. The result is lower capacity and power values when using the dew point definition, because the suction pressure, and hence suction density and mass flow are decreased by approximately 7%. The change in discharge pressure has very little effect. A small compensation arises because the enthalpy difference is slightly increased, resulting in an overall capacity drop of approximately 5%.

Performance Comparison

Table 1 identifies various rating conditions at which the performance is measured or calculated. The conditions differ in the extent of subcooling and superheating and accordingly represent different cycles on the p-h diagram. The pressures and temperatures corresponding to the different rating conditions, as calculated using the two approaches are also summarized in Table 1. The properties are calculated using the data provided by ASEREP [13].

The performance of the compressor at ARI using the midpoint approach is taken to be the base measure (100%). The performance at other conditions, calculated using the above recommended approach and also measured using a scroll compressor, are compared with respect to this base. The actual compressor test values are indicated in parentheses in Table 2. A very good correlation is observed between the calculated and measured values corroborating the validity of the recommended analytical method.

The values computed for a given rating condition using the two approaches are compared and represented in Figures 4, 5 and 6. It can be seen that when the dew point protocol is used, the capacity and the power calculated are less by approximately 5% with no appreciable difference in the COP (EER). As stated previously, the mass flow measured using the dewpoint protocol is about 6 - 7% lower than that measured using the midpoint protocol. Use of the ISO definition results in even lower capacity values because there is no subcooling effect. For specific values of subcooling or superheating the differences in performances, calculated using the two approaches, are similar. 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 7 illustrates the dew point capacity value as a percentage of the midpoint value for a typical scroll compressor for various evaporating and condensing temperatures. As can be seen from the graph, the difference ranges from 4% at 60°C (140°F) condensing to about 9% at 25°C (77°F) condensing, the dew point capacity being the lower of the two. However, for a given condensing temperature, the efficiency (COP / EER) calculated using the two approaches are comparable.

Conclusions:

(1) The paper presents a brief introduction to the characteristics of refrigerants with glide and associated implications on performance measurements from a compressor manufacturer's point of view.

(2) The real mean temperature would depend on the actual system design and a meaningful performance comparison between R407C and R22 cannot be made without reference to the system design.
(3) Because of the relative ease of use, the dew point approach is the better method for rating the compressor performance.
(4) Use of the dew point protocol consistently presents a lower capacity and power values than the midpoint protocol for a given compressor. The COPs (EERs) calculated from the two methods are however comparable.
(5) The results from one approach may be readily calibrated with respect to the other and the numbers thus meaningfully interpreted.

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(10) European Standard EN12900 Refrigerant Compressors – Rating Conditions, Tolerances and Presentation of Manufacturer's Performance data 1999.
(11) International Standard ISO 9309 (draft), Standard Rating Conditions for Compressors, p. 3.
(12) ARI Standard 540 Method for Presentation of Compressor Performance Data 1991.
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(15) Sundaresan, S.G., and Watkins, R.J., Copeland Corporation, Study of liquid and vapor composition of refrigerants during calorimetry, Proceedings of the 1993 International CFC and Halogen alternatives conference, Washington, D.C.

Point	Definition	Suction Press.	Discharge Press.	Liquid Temp.	Suction Temp.
1	**MID-POINT (ARI 1991)** 7.2/54.4 °C (45°F/130°F) 11.1K (20°F) SH, 8.3K (15°F) SC	**6.31 bar a** *91.5 psia*	**23.26 bar a** *337.3 psia*	**43.9°C** *111°F*	**20.45°C** *68.8°F*
2	**MID-POINT** 7.2/54.4°C (45/130 °F) 10K (18°F) SH, 0 SC	**6.27 bar a** *90.9 psia*	**23.26 bar a** *337.3 psia*	**52.2°C** *126°F*	**19.14°C** *66.4°F*
3	**MID-POINT** 5/50°C (41/122 °F) 10K (18°F) SH, 0 SC	**5.90 bar a** *85.6 psia*	**21.0 bar a** *304.5 psia*	**47.7°C** *118°F*	**17.04°C** *62.7°F*
4	**DEW-POINT(ARI, 1999)** 7.2/54.4 °C (45/130 °F) 11.1K (20°F) SH, 8.3K (15°F) SC	**5.89 bar a** *85.4 psia*	**22.1 bar a** *320.5 psia*	**41.7°C** *107°F*	**18.3°C** *65°F*
5	**DEW-POINT** 7.2/54.4°C (45/130 °F) 10K (18°F) SH, 0 SC	**5.89 bar a** *85.4 psia*	**22.1 bar a** *320.5 psia*	**50.0°C** *122°F*	**17.2°C** *63°F*
6	**DEW-POINT)(EN12900)** 5/50°C (41/122 °F) 10K (18°F) SH, 0 SC	**5.47 bar a** *79.3 psia*	**19.85 bar a** *287.8 psia*	**45.4°C** *113.7°F*	**15°C** *59°F*

Table 1. Physical Properties associated with the various Rating Points.

Point	Definition	Mass Flow	Capacity	Power	COP
1	**MID-POINT (ARI 1991)** 7.2/54.4 °C (45°F/130°F) 11.1K (20°F) SH, 8.3K (15°F) SC	**100%** **(100%)**	**100%** **(100%)**	**100%** **(100%)**	**100%** **(100%)**
2	**MID-POINT** 7.2/54.4°C (45/130 °F) 10K (18°F) SH, 0 SC	**100%** **(99.9%)**	**90%** **(89.3%)**	**100%** **(99.6%)**	**90%** **(89.6%)**
3	**MID-POINT** 5/50°C (41/122 °F) 10K (18°F) SH, 0 SC	**94%** **(93.7%)**	**89%** **(88.2%)**	**91%** **(90.3%)**	**98%** **(97.6%)**
4	**DEW-POINT(ARI, 1999)** 7.2/54.4 °C (45/130 °F) 11.1K (20°F) SH, 8.3K (15°F) SC	**93%** **(93.1%)**	**95%** **(94.8%)**	**95%** **(94.6%)**	**100%** **(100.2%)**
5	**DEW-POINT** 7.2/54.4°C (45/130 °F) 10K (18°F) SH, 0 SC	**94%** **(93%)**	**86%** **(84.9%)**	**95%** **(94.5%)**	**91%** **(89.9%)**
6	**DEW-POINT) (EN12900)** 5/50°C (41/122 °F) 10K (18°F) SH, 0 SC	**88%** **(86.7%)**	**85%** **(83.3%)**	**86%** **(85.9%)**	**98%** **(96.8%)**

Note: Numbers in parentheses represent actual performance values measured using a typical scroll compressor, relative to the ARI performance calculated using midpoint approach. Numbers not enclosed in parentheses are calculated values.

Table 2. Scroll Compressor Performance At Various Rating Conditions.

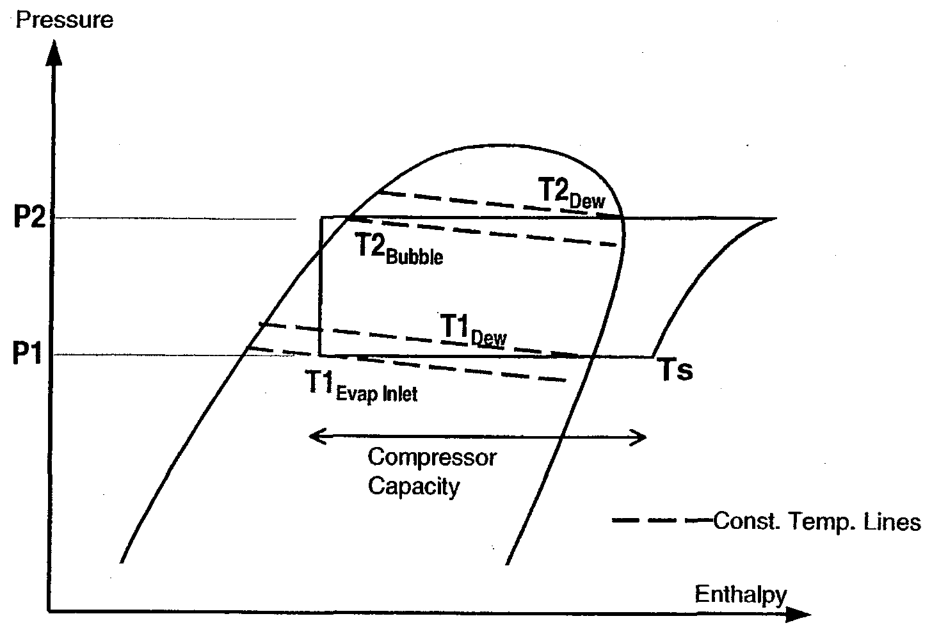

Figure 1. Refrigeration cycle showing lines of constant temperature in the two phase region

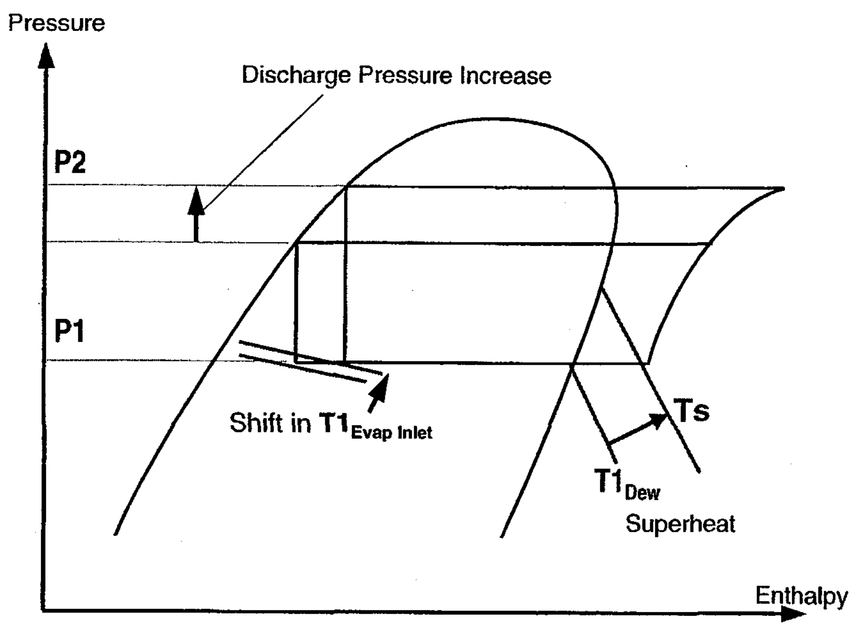

Figure 2. Refrigeration cycle illustrating the shift in evaporator inlet temperature caused by discharge pressure increase, and the definition of superheat

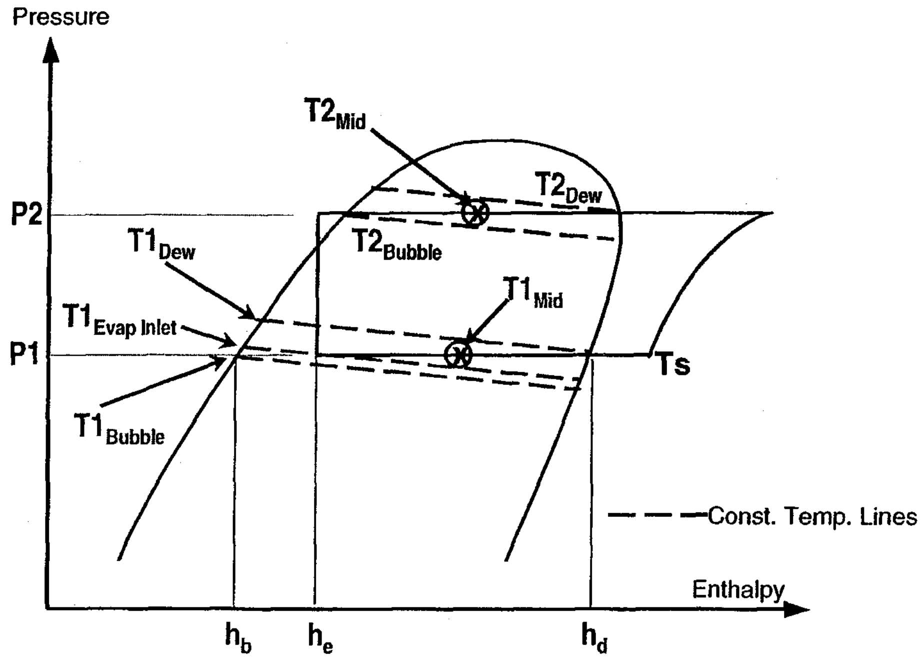

Figure 3. Calculation of Mid Point

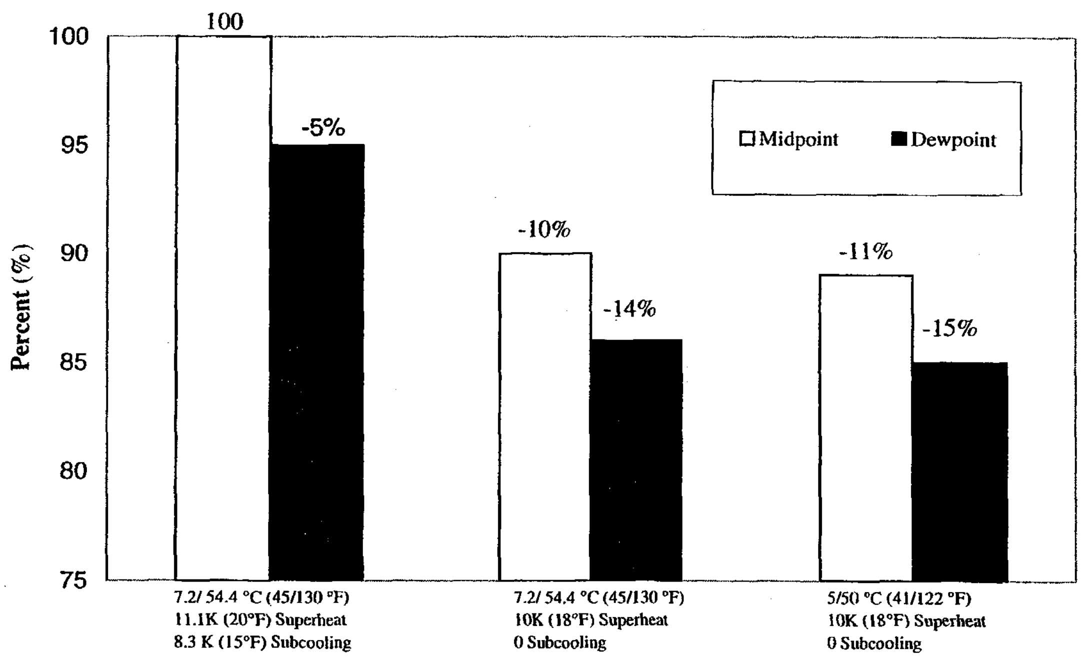

Figure 4 Comparison Between Midpoint And Dewpoint Capacities For Different Conditions

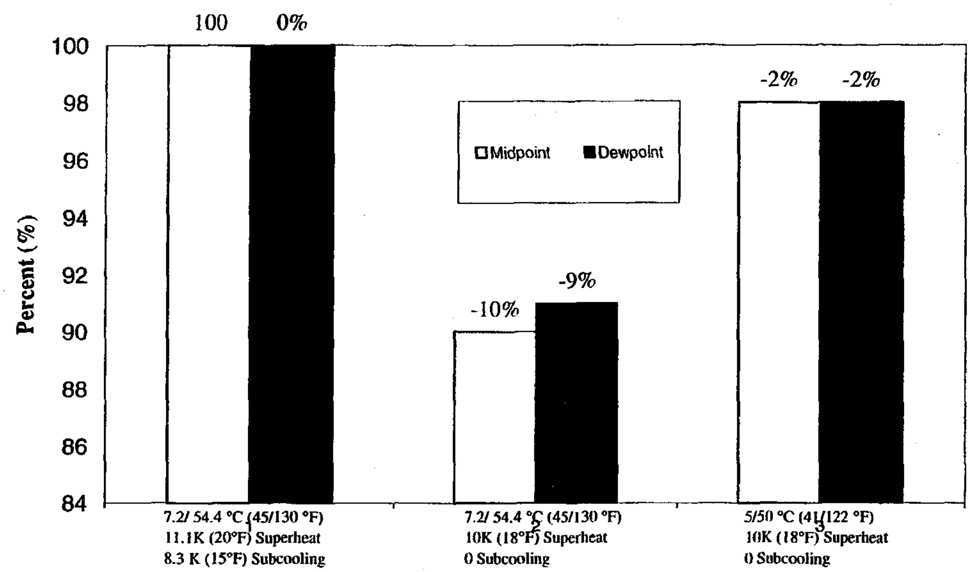

Figure 5 Comparison Between Midpoint And Dewpoint COPs For Different Conditions

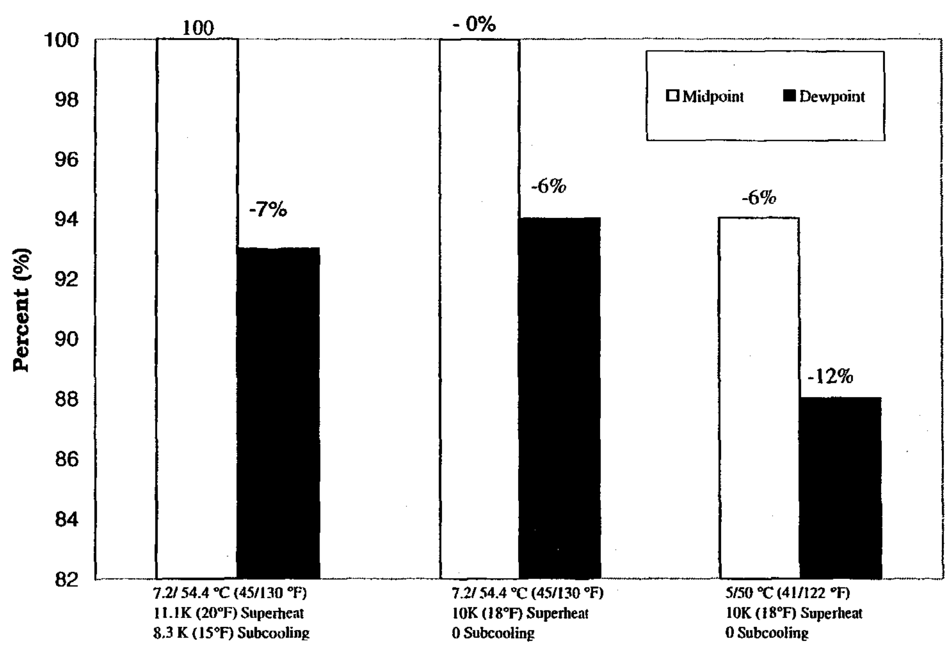

Figure 6 Comparison Between Midpoint And Dewpoint Mass Flows For Different Conditions

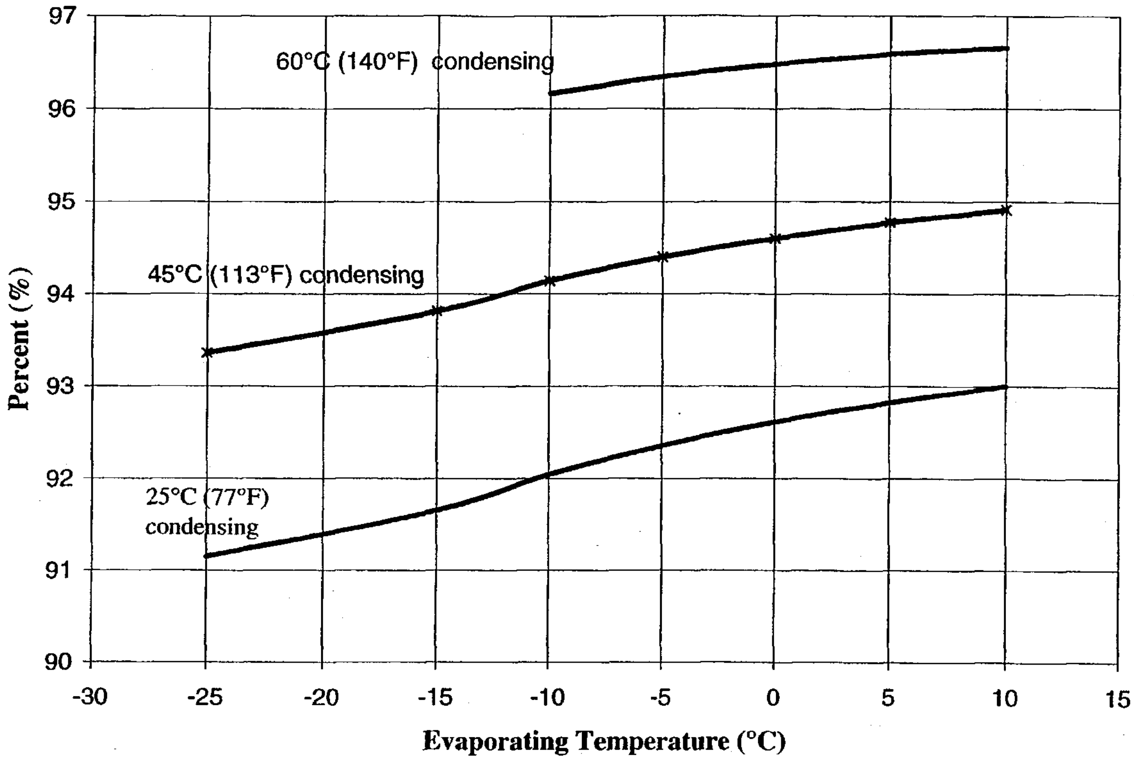

Fig. 7 Dew point capacity value as a percentage of midpoint value for a typical scroll compressor.