Factors to Consider in Converting Compressor Rated Capacity to Actual Capacity

One of the greatest areas of confusion and misunderstanding in the air conditioning and refrigeration industry is that of published compressor capacity ratings. Historically, each compressor manufacturer has developed their own rating system, and unfortunately the test conditions have varied. As a result, there have been problems in making accurate comparisons between compressors from different manufacturers.

Since the rated compressor capacity takes full credit for any increase in enthalpy (the engineering term for heat content) due to increased return gas temperature, normally the rated compressor capacity is higher with higher return gas temperatures. The effect on compressor capacity of higher return gas temperatures is a factor of internal compressor design, and will vary with each compressor model. There really is no all purpose correction factor, but for replacement purposes, it is sufficiently accurate on welded air conditioning compressors to allow 3% for the capacity change when comparing compressors rated with 95°F return gas as opposed to 65°F return gas.

A second major area of misunderstanding arises because compressors in actual systems seldom operate at exactly the same conditions as the published rating. Obviously, some adjustment must be made to published data to estimate the capacity at the actual field conditions.

A third complicating factor is the fact that there are differences of opinion and what appears to be misunderstanding by some engineers in the industry regarding the relationship between liquid subcooling and suction gas superheat that occurs in the liquid to suction heat exchanger.

Published Compressor Ratings For Copelametic® Compressors

Copelametic refrigeration compressors for many years had specification data published with rating conditions of 65°F return gas and 0°F subcooling. In recent years, confusion has arisen because ARI Standard 520 called for 5°F subcooling for low temperature ratings, and 15°F subcooling for medium and high temperature ratings, and new data sheets were issued on that basis. However, the liquid subcooling rating was not universally accepted in the industry, and in some cases alternate data sheets were issued. The water was further muddied because R-22 high temperature Copelametic compressors are basically considered air conditioning models, and data was published with 20°F superheat, while in many cases the same compressor was used in refrigeration applications where all other data was based on 65°F return gas.

A fixed rating point is essential so that consistent quality checks can be made to monitor compressor performance on the production line. In establishing rating conditions, the compressor is run on a precise calorimeter and capacity credit is taken for the liquid subcooling and return gas temperatures as stated. In making comparisons with compressors rated on a different basis, or in calculating capacity at different conditions, corrections must be made as necessary.

Liquid Subcooling

Liquid subcooling, although frequently included in a compressor rating, has no relation to compressor operation or performance. It is defined as heat extracted from liquid refrigerant condensed at a given saturation temperature and pressure by some heat sink outside the refrigeration system. It is often called “natural subcooling” and is typically the subcooling resulting from the condenser performance.

In reality, it comes from system operation and it can be logically argued that liquid subcooling should not be included in compressor ratings. The rationale for doing so has been convenience on the basis that the typical system would operate with some equivalent degree of subcooling, and the rated compressor capacity would thus reflect more accurately the expected system capacity. Regardless of the pros or cons, it is a fact of life that liquid subcooling does exist in many compressor ratings. Changes in liquid temperature reflect known changes in enthalpy, and the enthalpy change in the liquid for any temperature change at a constant pressure can be easily calculated from available tables.

Mechanical subcooling is the term used to describe further cooling of the liquid refrigerant by refrigeration-typically cooling the liquid going to the low temperature cases by the medium temperature system on a supermarket rack. To the extent that this subcooling can be maintained by insulation, the subcooled refrigerant can deliver more effective refrigeration at a higher efficiency level in the low temperature case due to the elimination of flash gas.

The third form of subcooling is that obtained through the use of a liquid to suction heat exchanger, and this type is the one most frequently misunderstood. The mere act of transferring heat from the liquid to the vapor does not in itself create any additional refrigeration effect, since the decreased enthalpy of the liquid may be offset by the decreased mass flow through the compressor. To the extent that the increase in suction gas temperature resulting from heat transfer from the liquid replaces heat pickup that might occur outside the refrigerated space, there is definitely a gain.

For example, assume the suction gas leaves a refrigerated case at 0°F, goes underground, and is warmed up to 40°F by heat picked up from the ground, and further increases in temperature to 65°F in the machine room. If a heat exchanger can be used, increasing the temperature of the gas to 40°F before it goes underground, and the gas still reaches the compressor at 65°F, there is a definite gain in capacity represented by the 40°F increase in suction gas temperature.

There will be an increase in capacity if the suction gas is raised to a higher temperature by a liquid to suction heat exchanger before entering the compressor. For example, assume a compressor is operating with 45°F gas entering the compressor. Utilizing a heat exchanger to raise the gas temperature to 65°F will increase the compressor capacity.

If the increase in suction gas temperature in a heat exchanger affects the temperature of the gas entering the compressor, the increase in capacity must be measured only by the effect on the suction gas, since the change in mass flow may offset some of the resultant liquid subcooling. In no case can credit be taken for both the changes in enthalpy in the liquid and the suction vapor, since this would be duplication of values.

It may help to visualize the refrigerated case and the heat exchanger within a black box, with liquid entering and suction vapor leaving. With or without a heat exchanger, the liquid entering does not change - only the suction gas temperature changes.

The limiting factor in the use of heat exchangers may actually be compressor discharge temperature, since the discharge temperature changes almost proportionally with the return gas temperature, and overheating remains the major source of compressor failure on low temperature systems.

Tables 1 to 5 show the adjustment to compressor rated capacity for changes in liquid temperature resulting from natural or mechanical subcooling.

Any capacity increase resulting from liquid to suction heat exchange is more accurately calculated from Tables 6 to 10.

Table 1
Increase In Compressor Rated Capacity Per 10°F Change In Liquid Temperature For Natural or Mechanical Subcooling
R-12 and R-22
For all Evaporating Temperatures

Condensing Temperature	65ºF Return Gas	50ºF Return Gas
130°F	5.2%	5.3%
120°F	4.8%	5.0%
110°F	4.5%	4.7%
100°F	4.2%	4.3%
90°F	4.0%	4.1%

Table 2
Increase In Compressor Rated Capacity Per 10°F Change In Liquid Temperature For Natural or Mechanical Subcooling
R-502

Condensing Temperature	65ºF Return Gas	50ºF Return Gas
Low Temp		
130°F	7.5%	7.9%
120°F	6.8%	7.0%
110°F	6.3%	6.5%
100°F	5.8%	6.0%
90°F	5.4%	5.5%
Med. Temp		
130°F	7.8%	8.3%
120°F	7.0%	7.5%
110°F	6.4%	6.8%
100°F	5.9%	6.3%
90°F	5.5%	5.8%

Table 3
Increase In Compressor Rated Capacity Per 10°F Change In Liquid Temperature For Natural or Mechanical Subcooling
R-22
For all Evaporating Temperatures

Condensing Temperature	65ºF Return Gas	40ºF Return Gas
140°F	5.6%	6.0%
130°F	5.1%	5.5%
120°F	4.7%	5.0%
110°F	4.4%	4.7%
100°F	4.1%	4.4%
90°F	3.9%	4.1%

Table 4
Increase In Compressor Rated Capacity Per 10°F Change In Liquid Temperature For Natural or Mechanical Subcooling
R-134A
For all Evaporating Temperatures

Condensing Temperature	65ºF Return Gas	40ºF Return Gas
140°F	6.9%	7.6%
130°F	6.3%	6.9%
120°F	5.8%	6.3%
110°F	5.4%	5.8%
100°F	5.0%	5.4%
90°F	4.6%	5.0%

Table 5
Increase In Compressor Rated Capacity Per 10°F Change In Liquid Temperature For Natural or Mechanical Subcooling
R-404A
For all Evaporating Temperatures

Condensing Temperature	65ºF Return Gas	40ºF Return Gas
140°F	15.8%	18.5%
130°F	12.4%	14.3%
120°F	10.2%	11.6%
110°F	8.7%	9.8%
100°F	7.6%	8.4%
90°F	6.8%	7.4%

Heat Transfer Into The Suction Vapor

On a compressor specification sheet showing capacity with 65°F return gas, the compressor capacity has been measured on a calorimeter taking credit for all of the heat content of the suction gas at 65°F. This can only be achieved with a liquid to suction heat exchanger located at the outlet of the refrigerated space with sufficient capacity to increase the suction gas temperature to 65°F.

As a practical matter, this seldom if ever happens on an actual installation. It is quite common to have 65°F gas entering the compressor, but some part of the heat gain typically enters the suction line from outside the refrigerated space, and thus reduces the published capacity. Tables 6 to 10 show the effect on capacity at different conditions.

As pointed out previously, the same figures may be used to calculate the benefit of a liquid to suction heat exchange by determining how much of the outside heat gain can be replaced by heat exchange from the liquid.

Table 6
Loss in Compressor Rated Capacity Due to Heat Gain into Suction Line Outside Refrigerated SpacePer 10°F Increase in Suction Gas Temperature
R-12

Conditions	100ºF Condensing Temperature		110ºF Condensing Temperature		120ºF Condensing Temperature	
	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas
Low Temp. 0ºF subc ooling 50ºF subcooling	2.6% 2.1%	2.7% 2.2%	2.7% 2.2%	2.8% 2.3%	2.8% 2.3%	2.9% 2.4%
Med. Temp. 0ºF subc ooling 50ºF subcooling	2.7% 2.3%	2.8% 2.3%	2.8% 2.3%	3.0% 2.4%	3.0% 2.4%	3.1% 2.5%
High Temp. 0ºF subc ooling 50ºF subcooling	2.9% 2.4%		3.1% 2.5%		3.2% 2.6%	

Table 7
Loss In Compressor Rated Capacity Due To Heat Gain Into Suction Line Outside Refrigerated SpacePer 10°F Increase In Suction Gas Temperature
R-502

Conditions	100ºF Condensing Temperature		110ºF Condensing Temperature		120ºF Condensing Temperature	
	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas
Low Temp. 0ºF subc ooling 50ºF subcooling	3.1% 2.5%	3.3% 2.5%	3.4% 2.6%	3.5% 2.7%	3.6% 2.7%	3.8% 2.8%
Med. Temp. 0ºF subc ooling 50ºF subcooling	3.4% 2.6%	3.5% 2.7%	3.6% 2.7%	3.8% 2.8%	3.8% 2.8%	4.1% 3.0%

Table 8
Loss In Compressor Rated Capacity Due To Heat Gain Into Suction Line Outside Refrigerated Space Per 10°F Increase In Suction Gas Temperature
R-22

Conditions	100ºF Condensing Temperature		110ºF Condensing Temperature		120ºF Condensing Temperature	
	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas
Low Temp.						
0ºF subc ooling	2.2%	2.3%	2.3%	2.3%	2.4%	2.5%
50ºF subcooling	1.9%	1.9%	1.9%	2.0%	2.0%	2.0%
Med. Temp.						
0ºF subc ooling	2.5%		2.5%		2.7%	
50ºF subcooling	2.0%		2.2%		2.2%	

Table 9
Loss In Compressor Rated Capacity Due To Heat Gain Into Suction Line Outside Refrigerated Space Per 10°F Increase In Suction Gas Temperature
R-404A

Conditions	100ºF Condensing		110ºF Condensing		120ºF Condensing	
	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas
Low Temp.						
0ºF Subcooling	3.5%	3.6%	3.7%	3.9%	4.1%	4.3%
50º Subcooling	2.6%	2.7%	2.8%	2.9%	2.9%	3.0%
Med. Temp.						
0ºF Subcooling	3.8%	4.1%	4.2%	4.4%	4.5%	4.8%
50º Subcooling	2.9%	3.1%	3.1%	3.2%	3.2%	3.4%

Table 10
Loss In Compressor Rated Capacity Due To Heat Gain Into Suction Line Outside Refrigerated Space Per 10°F Increase In Suction Gas Temperature
R-134a

Conditions	100ºF Condensing		110ºF Condensing		120ºF Condensing	
	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas	65ºF Return Gas	50ºF Return Gas
Med. Temp.						
0ºF Subcooling	2.9%	3.0%	3.1%	3.2%	3.2%	3.4%
50º Subcooling	2.3%	2.4%	2.4%	2.5%	2.6%	2.6%
High Temp.						
0ºF Subcooling	3.1%	-	3.3%	-	3.5%	-
50º Subcooling	2.5%	-	2.6%	-	2.7%	-

Change in Return Gas Temperature

Compressors are rated either for a fixed return gas temperature, or for a fixed amount of superheat in the suction gas. Suction vapor entering the compressor at any other temperature will change the compressor capacity. Three factors are involved. A decrease in return gas temperature will reduce the enthalpy, or heat content of the entering vapor, while the cooler gas will be more dense, thus increasing the mass flow through the compressor, and these two factors tend to offset one another. A change in return gas temperature will affect the heat transfer within the compressor.

The net effect of these changes on a typical compressor with different refrigerants is shown in Table 11.

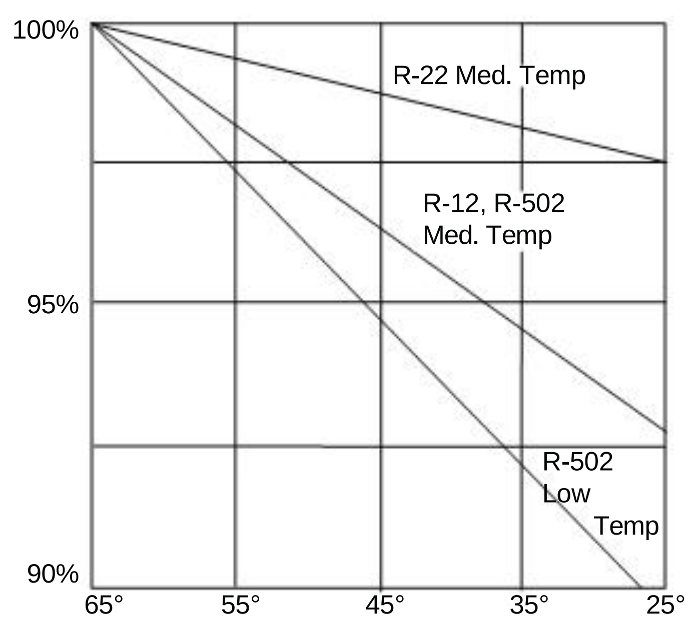
Note the dramatic difference in the performance with R-22 as opposed to R-12 and R-502. R-22 has a much larger refrigeration effect per pound than the other refrigerants, while being much less dense than R-502 vapor. The net effect is that in accessible hermetic compressors used for refrigeration duty, changes in return gas temperature with R-22 have little effect on compressor capacity.

There appears to be little effect on power consumption with low, return gas temperatures. Changes are so small they are difficult to measure, but appear to average about 1% increase in power consumption for each 30°F decrease In return gas temperature.

Effect of Return Gas Temperature on Compressor Capacity

Additional information is also included about R-22 and the newer non-ozone depleting HFC refrigerants R-404A and R-134a. This information is located in Tables 12 and 13. These tables reflect the net change of capacity with setting the evaporator temperature to 0ºF with a varying condensing and return gas temperature. Values are given for 65ºF and 40ºF return gas. This is due to the fact that semi-hermetic compressors are rated at 65ºF return gas and hermetic compressors are rated at 40ºF.

Table 11
Effect Of Return Gas Temperature On
Compressor Capacity



Return Gas Temperature, °F

Table 12

Loss in Compressor Rated Capacity Due to Return Gas Temperature With Respect to 65ºF Return Gas

R-22

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	6.0%	4.7%	3.1%	1.8%
20ºF	4.2%	3.4%	2.3%	1.4%
30ºF	2.9%	2.7%	1.8%	1.2%
40ºF	2.0%	1.9%	1.4%	0.9%
65ºF	0.0%	0.0%	0.0%	0.0%

R- 404A

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	28.6%	22.9%	16.0%	11.7%
20ºF	19.5%	15.7%	11.0%	8.2%
30ºF	15.1%	12.1%	8.5%	6.4%
40ºF	10.7%	8.6%	6.1%	4.5%
65ºF	0.0%	0.0%	0.0%	0.0%

R-134a

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	13.2%	12.2%	9.4%	7.3%
20ºF	9.3%	8.4%	6.6%	5.2%
30ºF	7.3%	6.6%	5.1%	4.0%
40ºF	5.3%	4.7%	3.7%	2.9%
65ºF	0.0%	0.0%	0.0%	0.0%

Table 13

Loss in Compressor Rated Capacity Due to Return Gas Temperature With Respect to 40ºF Return Gas

R-22

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	3.5%	2.8%	1.8%	1.0%
20ºF	1.8%	1.5%	1.0%	0.5%
30ºF	0.9%	0.8%	0.5%	0.3%
40ºF	0.0%	0.0%	0.0%	0.0%
65ºF	+2.3%	2.0%	+1.4%	+0.9%

R-404A

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	20.1%	15.6%	10.5%	7.5%
20ºF	9.9%	7.7%	5.2%	3.8%
30ºF	4.9%	3.8%	2.6%	1.9%
40ºF	0.0%	0.0%	0.0%	0.0%
65ºF	12.0%	+9.4%	+6.5%	+4.8%

R-134a

Return Gas	140ºF Condensing Temperature	130ºF Condensing Temperature	110ºF Condensing Temperature	90ºF Condensing Temperature
0ºF	8.3%	7.9%	6.0%	4.5%
20ºF	4.2%	3.9%	3.0%	2.3%
30ºF	2.1%	2.0%	1.5%	1.2%
40ºF	0.0%	0.0%	0.0%	0.0%
65ºF	+5.6%	+4.9%	+3.8%	3.0%

Converting Rated Capacity to Actual

By evaluating the effect of actual operating conditions on published ratings, reasonable estimates can be made of actual effective system capacity, and in some cases, the effect of system design changes can be evaluated.

Example # 1:

The ZF06K4E-TF5 refrigeration scroll compressor operating with R-404A is rated at 6,820 Btu/hr at -25°F evaporator, 100°F condensing, 0°F liquid subcooling, and 65°F return gas.

What is the estimated actual effective capacity at the case if there is 10°F natural liquid subcooling, the suction gas leaves the case and heat exchanger at +10°F, picks up heat from the ground and machine room, and enters the compressor at 40°F?

1. Effect of 40°F return gas from Table 12	-5.3%
2. Effect of 10°F subcooling from Table 5	+8.4%
3. Loss due to heat gain of 30°F in suction line from Table 9	-10.7%
Net Correction	-7.6%

Actual net capacity available
6,820 Btu/hr x 92.4% = 6,300 Btu/hr

Example # 2:

Given the same compressor, what is the actual effective capacity if the 10°F subcooling remains as in Example #1, the suction gas is raised to 40°F before leaving the case and heat exchanger, and the return gas enters the compressor at 65°F?

1. Effect of 10°F subcooling from Table 5	+8.4%
2. Loss due to heat gain of 25°F in suction line from Table 9	-8.3%
Net Correction	+0.1%

Actual net capacity available
6,820 Btu/hr x 100.1% = 6,830 Btu/hr

Example # 3:

The Discus model 6DH3-200E-TSK is rated at 136,000 Btu/hr with R-134a at 15°F evaporator, 110°F condensing, 0°F liquid subcooling, and 65°F return gas.

What is the effective capacity if applied in a supermarket with liquid at saturation entering a liquid to suction heat exchanger in the compressor room, the subcooled liquid run in insulated lines to the case, the suction gas leaving the case at 20°F, the suction vapor warming to 40°F before entering the heat exchanger, and the suction vapor warming to 50°F when leaving the heat exchanger and entering the compressor?

1. Loss due to heat gain of 20°F insuction line from Table 10	-6.4%
2. Loss due to decrease in return gas temperature of 15°F from Table 12	-3.3%
Net Correction	-9.7%

Actual net capacity available
136,000 Btu/hr x 90.3% = 122,810 Btu/hr

In a similar fashion, with interpolation, an evaluation can be made of most refrigeration systems operating at other than rating conditions. The same factors may be applied to compressors with different rating conditions for comparison purposes.

Example # 4:

The ZF06K4E-TF5 refrigeration scroll compressor operating with R-404A is rated at 6,820 Btu/hr at -25°F evaporator, 100°F condensing, 0°F liquid subcooling, and 65°F return gas.

a. What is the estimated actual effective capacity at the case if there is no natural liquid subcooling, the suction gas leaves the case and heat exchanger at +10°F (superheat), picks up heat from the ground and machine room, and enters the compressor at 65°F?

b. What is now the effective capacity if the suction gas picks up less heat (due to shorter or insulated lines) and enters the compressor at 40°F?

c. By what percentage is the compressor "oversized" in the second scenario?

a. Gas entering compressor at 65°F

1. Effect of 65°F return gas from Table 12	0.0%
2. Effect of zero subcooling from Table 5	0.0%
3. Loss due to heat gain of 55°F in suction line from Table 9	-19.3%
Net Correction	-19.3%

Actual net capacity available
6,820 Btu/hr x (100-19.3)% = 5,504 Btu/hr

b. Gas entering compressor at 40°F

1. Effect of 40°F return gas from Table 12	-5.3%
2. Effect of zero subcooling from Table 5	+0.0%
3. Loss due to heat gain of 30°F in suction line from Table 9	-10.5%
Net Correction	-15.8%

Actual net capacity available
6,820 Btu/hr x (100-15.8)% = 5,742 Btu/hr

c. With a return gas temperature of 40°F instead of 65°F, the compressor delivers a net capacity of 5,742 Btu/hr instead of 5,504 Btu/hr at the evaporator. This is a net, 238 Btu/hr more, or, 3.5% "more than the rated value" (19.3%-15.8% = +3.5%). The compressor in this case appears to be oversized by about 3.5% for this application.