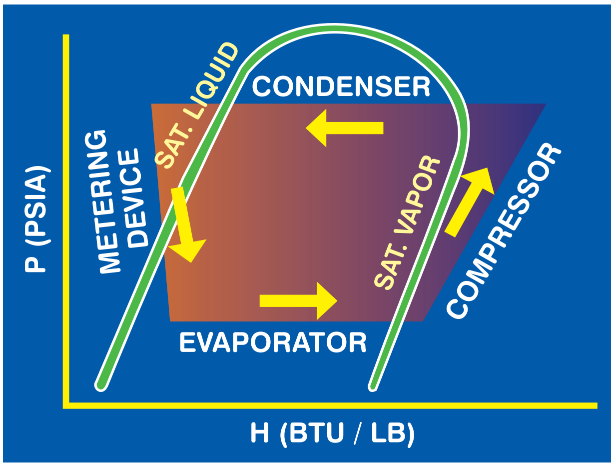

Some Misconceptions about the Refrigeration Cycle

And how use of the pressure/enthalpy diagram can help clear such doubts

By Ramesh Paranjpey
Technical Adviser & Consultant, Pune

During my career spanning over nearly 40 years I have come across many instances where due to lack of understanding of the basic refrigeration cycle, incorrect specifications are laid down by consultants, especially those consultants involved in chemical and process plant design. Most suppliers do not take exception and quote as per specifications which can lead to inefficient plant design. It is therefore essential to educate such people in highlighting the errors in their specifications so that these can be corrected. ASHRAE Handbook also confirms this in chapter 36 on refrigeration in the chemical industry stating that chemical engineers expect refrigeration, as any other utility, like water, steam or compressed air and do not understand that the refrigeration cycle performance is linked with the main chemical system behavior and a refrigeration plant cannot be turned on like a tap of water. Let me furnish some examples from my experience:

1. In a water-cooled chiller selection some consultants specify water entry temperature as 30° C and water outlet as 38°C with condensing as 44/45°C. When we pointed out that such high condensing temperatures are not desired, as it would lead to high power consumption and energy bills, they argued that with 8°C temperature rise in the condenser water, the circulated water quantity gets reduced thereby saving pumping cost. Similarly the heat exchanger would become more economical.

This is a total misconception, as any amount of saving in pumping cost and initial lower cost benefits are insignificant and are totally nullified against the higher power consumption of a refrigeration compressor. Similarly an air-cooled system can never be as efficient as a water-cooled system and many engineers try to justify use of air-cooled systems. Of course, there are other considerations for selecting an air-cooled option over a water-cooled system but it can never be due to power saving as can be seen in subsequent calculations.

2. Many refrigeration compressor manufacturers, especially from the US and Japan publish their compressor ratings with 15°F subcooling. Whereas most European manufacturers publish ratings at saturated conditions without any subcooling. The refrigeration compressor is a volume displacement machine and does not produce any subcooling on its own. It is also known that every degree of subcooling achieved in a system design increases capacity by approximately 0.5 % for R-22 refrigerant, without any extra power consumption and therefore the data for compressors published with subcooling looks unnecessarily attractive giving an impression that

continued on page 68

About the Author

Ramesh Paranjpey is a mechanical engineer with an M.Tech in refrigeration from IIT Bombay with over 35 years experience. He has worked in very senior positions starting with Kirloskar Pneumatic in Pune, Carrier Transicold in Bangalore and Singapore as well as Voltas-Air International Pune. Presently he works for himself as a technical adviser & consultant. He is an ASHRAE Fellow, past president ASHRAE W.I. chapter and past president ISHRAE Pune chapter. He can be contacted at *pramesh@vsnl.com*

continued from page 65

these are more efficient compressors with lower kW/TR values.

In reality, it is not an apple-to-apple comparison and one needs to carefully read conditions for which the data is published and apply suitable corrections. The subcooling section has to be built into the system by providing additional area or a separate subcooling section. If this is not done, one cannot get subcooling benefits. It should also be understood that only liquid can be subcooled and a mixture of vapour and liquid present in the condenser can never be subcooled and hence a separate arrangement for subcooling is a must if one wants to derive the benefit of subcooling.

3. The third myth is regarding useful superheat. If one studies in detail the thermodynamic cycle, superheat is never useful as it increases the specific volume at the entry of the compressor thereby reducing the mass flow rate and thus the cooling capacity. It is useful in the sense that it only helps in protecting the compressor by reducing the chances of getting liquid at the suction of the compressor.

Similarly if superheat is produced in the evaporator, the vapour zone area becomes larger, thereby making the evaporator less efficient, as expensive heat transfer area is used for superheating rather than for latent heat transfer by way of evaporation, which is the main function of an evaporator. The most efficient system is one without any superheating of suction gas, which is possible with all flooded coolers, predominantly used in ammonia systems or centrifugal machines where power consumption becomes a main criterion for selection due to very high capacities of large plants which work round the clock.

All these concepts will become clear once we look at the pressure/ enthalpy diagram and study various conditions and their effect on system performance.

We will use HCFC-22 refrigerant for our study and with bench mark values of +40°C saturated condensing temperature and +5°C saturated evaporating temperature, considering no subcooling of liquid and no superheating of suction gas. Similarly, equipment and piping pressure losses and heat gains are not considered for the sake of ease of understanding. The compression process is assumed to be isentropic. This situation is considered as *Condition 1*. Refrigeration capacity required is assumed as 10 ton for calculation of mass flow rate. Refrigeration properties are taken from the Danfoss software on Refrigeration Utilities.

Condition 1 (Bench Mark Cycle)

An introduction to the pressure /enthalpy diagram

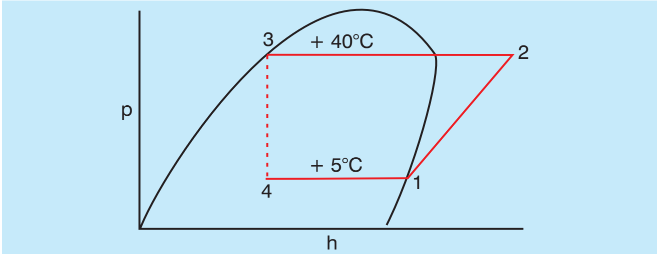

Figure 1

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy h_1-kJ/kg	Specific Vol. V_1 m³/kg	Enthalpy h_2-kJ/kg	T_2 °C	Enthalpy $h_3=h_4$ kJ/kg
+5	+40	5.838	15.335	407.152	0.040362	431.044	55.763	249.674

From this data we can derive this useful information :

1. Refrigeration capacity: $h_1–h_4$= (407.152)–(249.674) = 157.478 kJ/kg
2. Power required for compression: $h_2–h_1$= (431.044) – (407.152) = 23.892 kJ/kg
3. Coefficient of performance (COP): $h_1–h_4 \div h_2–h_1$ = (157.478) / (23.892) = 6.5912
4. Compression ratio: discharge pressure/suction pressure = (15.335) / (5.838) = 2.6267
5. Discharge temperature at the end of isentropic compression: 55.763°C
6. Specific volume at 1: 0.040362 m³/kg
7. Mass flow rate = 1/specific volume = 1÷ 0.040362= 24.7757 kg/m³
8. Mass flow rate to get 10 ton capacity: 10x12660/157.478 = 803.9218 kg/hr × 0.040362 m³/kg = 32.44 m³/hr
9. Heat rejection in condenser = $h_2–h_3$ = (431.064) – (249.674) = 181.39 kJ/kg

Condition 2

Always keep the saturated discharge temperature as low as possible.

(Discharge temperature can increase due to various factors such as undersized condenser, reduced / low water flow, blocked condenser tubes, strainer, overcharge, non

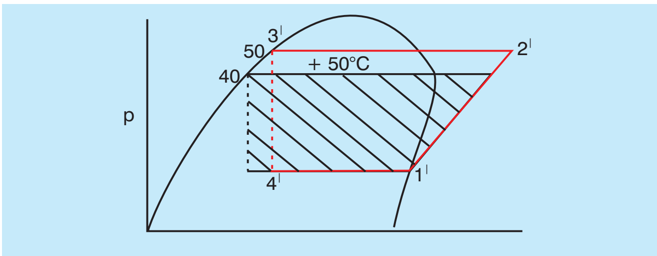

Figure 2

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy h'_1-kJ/kg	Specific Vol. V_1 m³/kg	Enthalpy h'_2-kJ/kg	T'_2 °C	Enthalpy $h'_3=h'_4$ kJ/kg
+5	+50	5.838	19.423	407.152	0.040362	437.210	69.559	263.253

continued on page 70

continued from page 68

condensibles etc.)

The diagram and calculations will show you why.

Assume that the discharge pressure is higher than the bench mark +40°C (in Condition 1) and the design is based on a higher condensing temperature, say +50°C.

From this data we can derive that when the discharge pressure is higher than expected :

1. Refrigeration capacity: $h'_1 - h'_4$ = (407.152) – (263.253) = 143.899 kJ/kg vs the bench mark 157.478 kJ/kg. *Capacity reduces!*
2. Power required for compression: $h'_2 - h'_1$ = (437.210) – (407.152) = 30.058 kJ/kg vs the bench mark 23.892 kJ/kg. *Power consumption increases!*
3. Coefficient of performance (COP): $(h'_1 - h'_4) \div (h'_2 - h'_1)$ = 143.899/30.058 = 4.7873 vs the bench mark 6.5912. *Efficiency drops!*
4. Compression ratio : discharge pressure / suction pressure = (19.423) / (5.838) = 3.326 vs the bench mark 2.6267. *Increases or volumetric efficiency drops!*
5. Discharge temperature at the end of isentropic compression: 69.559°C vs the bench mark 55.763°C. *Increases and compressor runs hotter, leading to more wear & tear.*
6. Specific volume at 1 : 0.04362 m³/kg vs the bench mark 0.040362 m³/kg. *Remains the same!*
7. Mass flow rate to get 10 ton capacity=12660 ×10/ (143.899) = (879.7837) kg/hr × (0.040362) = 35.50 m³/ hr vs the bench mark 32.44 m³/hr. *Since more mass flow is required to get the same capacity, a bigger compressor is required.*
8. Heat rejection in condenser = $h'_2 - h'_3$ = (437.210) – (263.253)= 173.957 × $\frac{157.478}{143.899}$ = 190.3724 kJ/kg vs the bench mark 181.39 kJ/kg. *Heat rejection increases! Requires a bigger condenser. Also ratio of heat rejection/ cooling capacity increases as cycle becomes less efficient.*

Conclusion

Keep/select/design as low a saturated discharge temperature & pressure as possible to get the best performance without affecting the design conditions in the premises or process fluid outlet temperature. Ensure the required pressure drop for the TXV, if used.

Condition 3

Always keep the saturated suction temperature as high as possible. The following diagram and calculations will show you why.

Assume that the saturated suction temperature is lower than the bench mark +5°C (as per Condition 1) and the design is based on a lower evaporating temperature, say + 0°C.

This can happen due to liquid line obstruction, evaporator coil or fan-motor damaged, less refrigerant charge, moisture in the system, under sizing of liquid line or expansion valve etc.

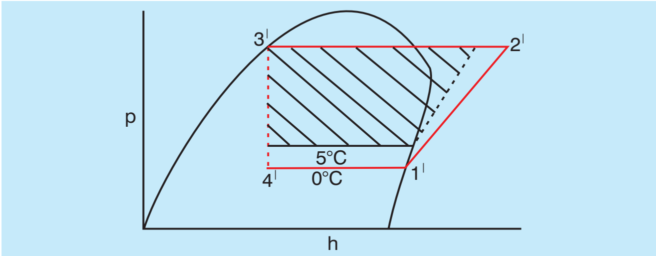

Figure 3

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy h'_1-kJ/kg	Specific Vol. V_1 m³/kg	Enthalpy h'_2-kJ/kg	T'_2 °C	Enthalpy $h'_3=h'_4$ kJ/kg
+0	+40	4.976	15.335	405.370	0.047143	433.429	58.43	249.674

From this data we can derive the following:

1. Refrigeration capacity = $h'_1 - h'_4$ = (405.370) – (249.674) = 155.696 kJ/kg vs the bench mark 157.478 kJ/kg. *Capacity reduces!*
2. Power required for compression: $h'_2 - h'_1$ = (433.429) – (405.370) = 28.059 ×157.478/155.696 = 28.38kJ/kg vs the bench mark 23.892kJ/kg. *Power consumption increases!*
3. Coefficient of performance (COP): $(h'_1 - h'_4) \div (h'_2 - h'_1)$ = 155.696/28.059=5.54 vs the bench mark 6.5912. *Efficiency drops!*
4. Compression ratio: discharge pressure/suction pressure = 15.335/4.976=3.0818 vs the bench mark 2.6267. *Increases or volumetric efficiency drops!*
5. Discharge temperature at the end of isentropic compression: 58.43°C vs the bench mark 55.763°C. *Increases and compressor runs hotter, leading to more wear & tear!*
6. Specific volume at 1 : 0.047143 m³/kg vs the bench mark 0.040362 m³/kg. *Increases!*
7. Mass flow rate to get 10 ton capacity = 12660×10/ 155.696 = 813.123 kg/hr × 0.047143 m³/kg = 38.33 m³/hr vs the bench mark 32.44 m³/hr. *Since more mass flow is required to get the same capacity, a bigger compressor is required!*
8. Heat rejection in condenser = $h'_2 - h'_3$ = 433.429 – 249.674 = 183.755×157.478/155.696 = 185.858kJ/kg vs the bench mark181.39 kJ/kg. *Heat rejection increases! Requires a bigger condenser. Also ratio of heat rejection / cooling capacity increases as the cycle becomes less efficient.*

Conclusion

Keep/select/design as high a saturated suction temperature & pressure as possible to get the best performance without affecting the design conditions in the premises or process fluid outlet temperature.

Condition 4

Always keep suction gas superheat to the minimum. This illustration is based on external superheat after the evaporator.

continued on page 74

continued from page 70

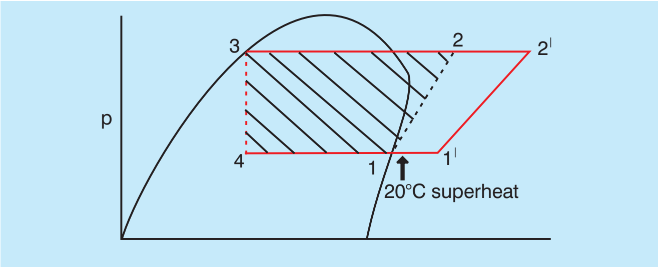

Figure 4

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy $h^{|}_{1}$-kJ/kg	Specific Vol. V_1 m³/kg	Enthalpy $h^{|}_{2}$-kJ/kg	$T^{|}_{2}$ °C	Enthalpy $h_3=h_4$ kJ/kg
+5(+20°C) superheat	+40	5.838	15.335	421.903	0.044647	448.414	75.742	249.674

From this data we can derive this useful information:

1. Refrigeration capacity: $h_1–h_4$ = (407.152) – (249.674) = 157.478 kJ/kg
2. Power required for compression: $h^{|}_{2}–h^{|}_{1}$ = (448.414) – (421.903) = 26.511 vs the bench mark 23.892 kJ/kg. *Hence as superheat increases, power consumption increases.*
3. Coefficient of performance (COP): $(h_1–h_4) \div (h^{|}_{2}–h^{|}_{1})$ = 157.478/26.511 = 5.94 vs the bench mark 6.5912. *System efficiency drops!*
4. Compression ratio : discharge pressure/suction pressure = 15.335/5.838 = 2.6267. *Remains unaltered, as superheat does not alter the compressor discharge or suction pressure.*
5. Discharge temperature at the end of isentropic compression: 75.742°C vs the bench mark 55.763°C. *Discharge temperature increases. Compressor runs hotter, leading to more wear & tear.*
6. Specific volume at 1 : 0.044647 m³/kg vs the bench mark 0.040362 m³/kg. *Increases!*
7. Mass flow rate to get 10 ton capacity = 12660 × 10/157.478 = 803.92 × 0.044647 = 35.8926 m³/hr vs the bench mark 32.44 m³/hr. *Since more mass flow is required to get the same capacity, a bigger compressor is needed.*
8. Heat rejection in condenser = $h^{|}_{2}–h_3$ = 448.414–249.674 = 198.74 kJ/kg vs the bench mark 181.39 kJ/kg. *Requires a bigger condenser! Also the ratio of heat rejection / cooling capacity increases as the cycle becomes less efficient.*

Conclusion

Super heat is always bad for refrigeration cycle efficiency. In direct expansion plants we normally keep this to around 5° to 6°C to protect the compressor from liquid entry. Electronic expansion valves have a faster response and hence are being increasingly used as they work with low superheat settings and hence from the same evaporator more area is available for latent heat transfer than for superheating and thus the cycle efficiency improves. Ideally, flooded systems with a saturated suction with no superheat gives best efficiency and performance. Hence most of the big chillers where power consumption is critical like centrifugal machines or screw chiller packages work on flooded operation, with no superheat. It is a myth to mislead people by calling it useful superheat. Superheat is never useful for the refrigeration cycle except that it protects the compressor from possible liquid refrigerant at the suction valve. Any amount of superheat achieved in the evaporator is in reality a loss, since expensive area of the evaporator is being used for superheating whereas in reality it should have been used for latent heat transfer i.e. evaporation.

Condition 5

Always increase sub cooling as much as possible.

The following diagram and calculations will show you why:

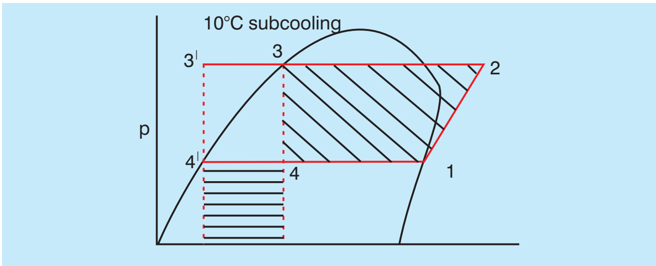

Figure 5

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy h_1-kJ/kg	Specific Vol V_1 m³/kg	Enthalpy $h^{|}_{2}$-kJ/kg	$T^{|}_{2}$ °C	Enthalpy $h^{|}_{3}=h^{|}_{4}$ kJ/kg
+5	+40 with 10°C sub cooling	5.838	15.335	407.152	0.040362	431.064	55.763	236.651

This condition considers external subcooling without use of refrigeration

From this data we derive this useful information

1. Refrigeration capacity: $h_1–h^{|}_{4}$ = (407.152) – (236.651) = 170.501kJ/kg vs the bench mark 157.478 kJ/kg. *Increases!*
2. Power required for compression: $h_2–h_1$ = (431.064) – (407.152) = 23.892 vs the bench mark 23.892 kJ/kg. *Remains unaltered!*
3. Hence subcooling increases capacity without any increase in power
4. Coefficient of performance (COP): $(h_1–h^{|}_{4}) \div (h_2–h_1)$ = 170.501/23.892 = 7.1363 vs the bench mark 6.5912. *System efficiency improves!*
5. Compression ratio : discharge pressure/suction pressure = 15.335/5.838 = 2.6267. *Remains unaltered, as subcooling does not alter the compressor discharge or suction pressure.*
6. Discharge temperature at the end of isentropic

continued on page 76

continued from page 74

compression: 55.763°C vs the bench mark 55.763°C. *Does not increase, remains the same!*

7. Specific volume at 1 : 0.040362 m³/kg vs the bench mark 0.040362m³/kg. *Remains the same! However as cooling capacity per kg of refrigerant has increased, less mass flow to get same capacity is required. Hence a smaller compressor can do the job!*
8. Mass flow rate to get 10 ton capacity = 12660 × 10/ 170.501 = 742.517 × 0.040362 = 29.96 m³/hr vs the bench mark 32.44 m³/hr. *Since less mass flow is required to get the same capacity, a smaller compressor is needed!*
9. Heat rejection in condenser = $h_2 - h'_3$ = (431.064) – (236.651) = 194.413 kJ/kg vs the bench mark 181.39 kJ/kg. *Requires a bigger condenser!*

Conclusion

As can be seen subcooling the liquid always is beneficial. It adds to the capacity without increasing power consumption. Subcooling also ensures that the metering device receives liquid only. If gas bubbles are present in the liquid at the entry, it causes many problems as is known to all of us. Any degree of subcooling is not possible and depends on the cooling medium temperature available as well as the saturated discharge temperature. It also adds to the cost but advantages more than compensate for this additional cost.

Condition 6

This condition takes into account both superheating of suction gas and subcooling of liquid refrigerant normally achieved by use of suction – liquid line heat exchangers.

This condition considers external subcooling with out use of refrigeration.

From this data we can derive this useful information.

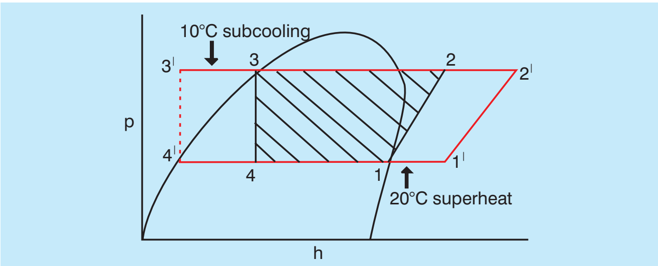

Figure 6

Evap. Temp. °C	Cond. Temp. °C	Suct.Pr. Bar	Disc.Pr. Bar	Enthalpy h'_1-kJ/kg	Specific Vol. V'_1 m³/kg	Enthalpy h'_2-kJ/kg	T'_2 °C	Enthalpy $h'_3=h'_4$ kJ/kg
+5(+20°C) superheat cooling	+40 with 10°C sub	5.838	15.335	421.903	0.044647	448.414	75.742	236.651

1. Refrigeration capacity: $h_1 - h'_4$ = 407.152 – 236.651 = 170.501kJ/kg vs the bench mark 157.478. *Capacity increases!*
2. Power required for compression: $h'_2 - h'_1$ = (448.414) – (421.903) = 26.511 vs the bench mark 23.892 kJ/kg. *Increases!*
3. Coefficient of performance (COP): $(h_1 - h_4) \div (h'_2 - h'_1)$ = (170.501) / (26.511) = 6.431 vs the bench mark 6.5912. *System efficiency is slightly lower. Although capacity has increased due to subcooling the power consumption has also increased due to superheating.*
4. Compression ratio : discharge pressure/suction pressure = 15.335/5.838 = 2.6267. *Remains unaltered.*
5. Discharge temperature at the end of isentropic compression: 75.742°C vs the bench mark 55.763°C. *Increases, leading to higher wear & tear.*
6. Specific volume at 1 : 0.044647 m³/kg vs the bench mark 0.040362 m³/kg. *Increases!*
7. Mass flow rate to get 10 ton capacity = 12660 ×10/ 170.501 = 742.517 × 0.044647 = 33.15 m³/hr vs the bench mark 32.44 m³/hr. *Less mass flow which would have been required due to subcooling is nullified due to increase in specific volume on account of superheat. Hence to get the same capacity, more or less the same compressor swept volume would be required.*
8. Heat rejection in condenser = $h'_2 - h_3$ = (448.414) – (249.614) = 198.8 kJ/kg vs the bench mark 190.3724 kJ/kg. *Requires a bigger condenser!*

Conclusion

A suction /liquid line heat exchanger is a useful device as it helps in subcooling liquid there by giving additional capacity. The resulting superheat increase ensures that the compressor has less chance of liquid coming through the suction gas. Selection of expansion valve and location of bulb needs to be studied to ensure that superheat due to expansion valve and through subcooler do not add and lead to abnormal superheating.

Summary

For any design, whether it is for air conditioning or process plant, the best theoretical efficiency is the Carnot cycle, which means heat absorption at the same conditions as the space/cooling medium temperature to be maintained and rejecting heat at the same temperature as the heat sink. This is, in actual practice not possible and hence we design heat exchangers with a certain optimum temperature differences. However, a designer must keep in mind that closing the temperature differences between condensing and evaporating will always improve the system performance. Similarly it should be kept in mind that there is nothing like useful superheat. Superheat is always bad where as subcooling is always useful. In short, lower discharge pressures, higher suction pressures, low superheat, high subcooling and lower compression discharge temperature is the best formula for any vapour compression refrigeration system design. ❖