Technical article

Energy saving on the high-pressure side of a refrigerating plant

Dipl. – Phys. Peter Roth

Head of Test Dpt.

Main topics:

- Presentation of effects on the refrigerating plant's COP when the condensing temperature is decreased with different refrigerants used. Which effects to these changes have at the cooling point at different temperature conditions (normal cooling/low-temperature cooling/air conditioning)?
- Summary of the most important influencing variables on the condensing temperature with regard to planning and operation of the refrigerating plant.
- Background information for determining the minimum condensing temperature.
  Which factors limit the condensing temperature?
- Comparison of different annual ambient temperature patterns and distributions using the example of three European cities and their influence on the achievable condensing temperatures and thus the partial load efficiency.
- Estimation of influence of power consumption of condenser fans on the overall efficiency of a refrigerating plant.

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Güntner AG & Co. KG
Hans-Güntner-Straße 2 - 6
82256 Fürstenfeldbruck
GERMANY

1. Introduction

Energy saving is a subject that has been discussed on several levels for many years now. It is explained and discussed between people from many different social classes – from theoretical papers drafted by academic institutes to TV talk shows. But do we really want to save "energy"? From a physical aspect, the total energy of the planet is practically inexhaustible, especially considering an enormous amount of energy flows continuously from the sun on to earth. The actual difficulties are the requirements and the results of converting energy carriers into technically usable energy. Therefore, the term "energy saving" really means "energy conversion with the least possible effort". On the one hand, this means that we should strive to achieve the most economically efficient and, consequently, inexpensive method to satisfy our "thermal" needs. On the other hand, we want our elementary living conditions, such as a clean environment, an adequate supply of healthy food and stable social systems to suffer as little as possible as a result of the energy conversion processes. It is especially this second, non financial aspect that was dealt with in the second part of the Fourth Assessment Report of the United Nations Intergovernmental Panel on Climate Change in March 2007 and that attracted a lot of attention. Güntner AG & Co. KG is also committed to these values and, for this reason, has embedded them in the company guidelines for many years.
Especially with heat exchangers it is quite obvious that one has to think about energy flows. Since air is used as a heating source and as a heat sink in many refrigeration systems, finned heat exchangers are used. The design and operating mode of these heat exchangers has a considerable effect on the energy consumption of a refrigeration system. In the following article, the thermodynamic interconnections on the high pressure side – the side of a refrigeration system that emits heat – will be discussed in theory and practice.

2. Effect of the condensation temperature on the coefficient of performance (COP) of a refrigerant system

The ratio between benefit and input is used to describe the efficiency or Q factor. In a refrigeration system the heat extracted from a chilled good or a cold storage room is the benefit and the work needed to do this, such as electrical driving power on the compressor is the input.

$$Güte = \frac{Nutzen}{Aufwand}$$ in a refrigeration system, this is $$COP = \frac{\dot{Q}_0}{\dot{W}}$$

Güte	Efficiency
Nutzen	Benefit
Aufwand	Input

where *COP* = Coefficient of Performance, $\dot{Q}_0$ = refrigerating capacity in watts and $\dot{W}$ = electrical compressor power in watts. Below we will consider only the single-stage vapour compression process to generate refrigeration as this is the most commonly used system. In this process, heat is withdrawn from a heat source, such as a cold storage room, which is then used to evaporate liquid refrigerant. The refrigerant vapour is compressed in the compressor with an enthalpy rise and is condensed again into the liquid phase on the condenser, thus giving off heat. The ratio of absorbed to emitted heat and the electrical work needed for this depend on the substance used and its temperature. These interconnections can be illustrated very clearly in the log(p)-h diagram of the respective refrigerant (Figure1).

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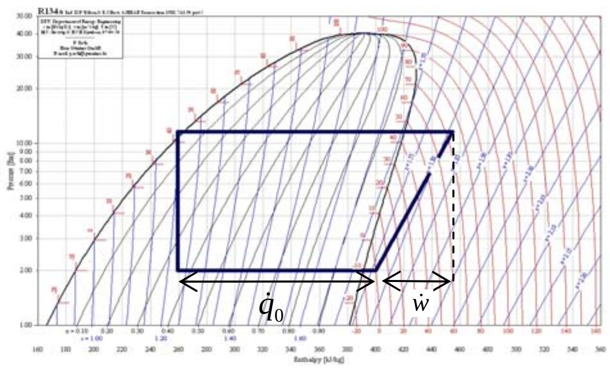

Figure 1: Single-stage refrigerant system

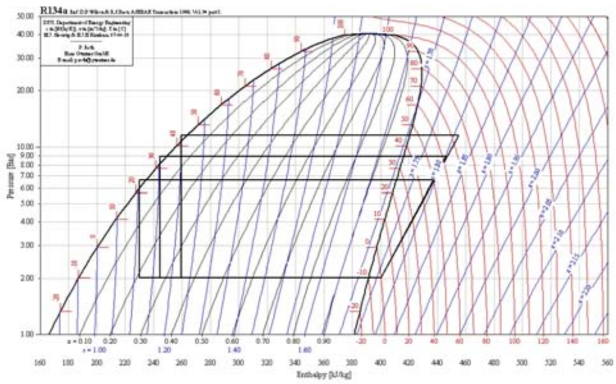

Figure 2: Three refrigerant systems with the same evaporation temperature and different condensation temperatures

In Figure 2 it can be seen that when the condensation temperature is reduced the refrigeration capacity increases and, at the same time, the required compressor power is reduced. Both mechanisms contribute to an

increase in the COP. This connection is calculated in Table 1 for several refrigerants and various temperatures.

Refrigerant		t_c=45 °C	t_c=35 °C	t_c=25 °C
R134a	t_o=0 °C	**3.43**	**4.7**	**6.94**
	t_o=-10 °C	**2.57**	**3.37**	**4.59**
R404A	t_o=-10 °C	**2.23**	**3.08**	**4.33**
	t_o=-40 °C	**1.05**	**1.38**	**1.78**
R717 (NH3)	t_o=-10 °C	**2.65**	**3.38**	**4.51**

Table 1: COP of different refrigerant systems; entropic efficiency of compressor: 0.7; Δt_{oh}=10 K for t_o=0, -10 °C; Δt_{oh}=20 K for t_o=-40 °C; Δt_u=3 K

Refrigerant:		t_c=40 °C	t_c=30 °C
R134a	t_o=0 °C	**3.3 %**	**3.9 %**
	t_o=-10 °C	**2.7 %**	**3.1 %**
R404A	t_o=-10 °C	**3.1 %**	**3.6 %**
	t_o=-40 °C	**3.4 %**	**2.6 %**
R717 (NH3)	t_o=-10 °C	**2.3 %**	**3.1 %**

Table 2: Average percentage improvement of the COP when the condensation temperature is reduced by 1 K

In Table 2 the average percentage improvement in the COP is shown when the condensation temperature is reduced by 1 K for two different temperature ranges. This means, for example, that with R404A lowering the condensation temperature from 43°C to 41°C with a constant evaporation temperature of –10°C the COP improves by two times 3.8%, or about 8%. The result of this is that you need 8% less mechanical compression work and, hence, 8% less electrical work. This opens up enormous savings potentials.

The condensation temperature has an important effect on the energy consumption of a refrigeration system. It is therefore very important to focus on keeping the condensation temperature as low as possible.

3. Parameters impacting the condensation temperature

This raises the question as to which measures and variables can influence the condensation temperature. For this, we must consider two fundamental equations for calculating heat exchangers:

$$(1)\quad \dot{Q}_c = \dot{m} \cdot \Delta h_c$$

$$(2)\quad \dot{Q}_c = k \cdot A \cdot (t_c - t_L)$$

$$\Rightarrow t_c - t_L = \frac{\dot{m} \cdot \Delta h_c}{k \cdot A}$$

$$(3)\quad \boxed{t_c = \frac{\dot{m} \cdot \Delta h_c}{k \cdot A} + t_L}$$

where
$\dot{m} =$ Kältemittelmassestrom
$\Delta h_c =$ spezifische Verflüssigungsenthalpie
$k =$ Wärmedurchgangskoeffizient
$A =$ Wärmeaustauscherfläche
$t_L =$ Lufteintrittstemperatur

Kältemittelmassestrom	Mass flow of refrigerant
Spezifische Verflüssigungsenthalpie	Specific condensation enthalpy
Wärmedurchgangskoeffizient	Heat transfer coefficient
Wärmeaustauscherfläche	Surface area of heat exchanger
Lufteintrittstemperatur	Air intake temperature

In the framed formula (3) all the thermodynamic-related parameters impacting the condensation temperature are named. In detail, this means:

- The higher the amount of refrigerant to be condensed, the higher the condensation temperature must be if the other underlying conditions remain constant. One could also say that the cooling load and hence the converted amount of refrigerant must be kept as low as possible to achieve low condensation temperatures.
- The specific condensation enthalpy is a substance property and cannot be influenced other than by the choice of refrigerant.
  Since in the specific condensation enthalpy the compressor outlet temperature is included as the right limit, attention must be paid that the hot gas outlet temperature is not unnecessarily high as the result of a low quality compressor. It must also be ensured that the overheating of the refrigerant during evaporation is not unnecessarily high, as this also leads to a considerable increase in the condensation enthalpy.
- The heat transfer coefficient should be as high as possible. This is an equipment-specific property. As a producer of heat exchangers, Güntner constantly strives to develop more efficient heat exchangers with the help of optimised fin profiles and tube geometries.
- The greater the surface area of a heat exchanger, the lower the condensation temperature can be – with the same boundary conditions. The entire surface area of the heat exchanger must also be free and uncontaminated to participate in the heat transport.
- It is almost trivial that the condensation temperature is linked directly to the air intake temperature – in other words, the ambient temperature. This means, of course, that the heat exchanger must have access to as much fresh air as possible. Consequently, they should not be installed in pits or in narrow spaces to avoid them sucking in heated air (recirculation).

The commonly held idea that the condensation temperature is dependent on the power or size of the compressors is only half true, as a larger compressor helps increase the condensation temperature only because of its greater mass flow and not because it can "develop a higher pressure", as can clearly be seen in the formula above.

4. Limits for reducing the condensation temperature

As shown above, reducing the condensation temperature can increase the efficiency of the unit and thus reduce drive energy. But this raises the question as to how much one can reduce the condensation temperature. What are the technical limits as regards lowering the temperature?
Figure 1 shows that the two pressure levels are connected via the relaxation apparatus and also via the pressure increasing device (compressor). Both could have an influence on the determination of a minimum possible condensation temperature.
Some brief research of compressor manufacturers showed that there are no serious technical or thermodynamic restrictions for low condensation temperatures as regards compression technology. With two people who were contacted the first reaction was in regard to a lowest permitted high pressure, "This deployment limit doesn't exist." However, it is possible that various compressor models are more or less suitable for low condensation temperatures. With rotary screw compressors an external oil pump may be needed and it must be ensured that a minimum oil temperature is maintained. In the case of reciprocating compressors the minimum forces needed to open the valve plates and the associated reduction in COP set a lower limit for the minimum condensation pressure. But these restrictions are far removed from the defining factor for the minimum condensation temperature: the expansion apparatus.
All expansion apparatus, with the exception of relaxation machines, which can be regarded more as rare curiosities and which have never been used in commercial refrigeration or cooling systems, are valves or capillaries. With almost all expansion valves the pressure difference created by them has the main effect on the volumetric flow rate that passes through them. Figure 3 shows the refrigeration capacities of a thermostatic expansion valve with three different nozzles in relation to the pressure difference.

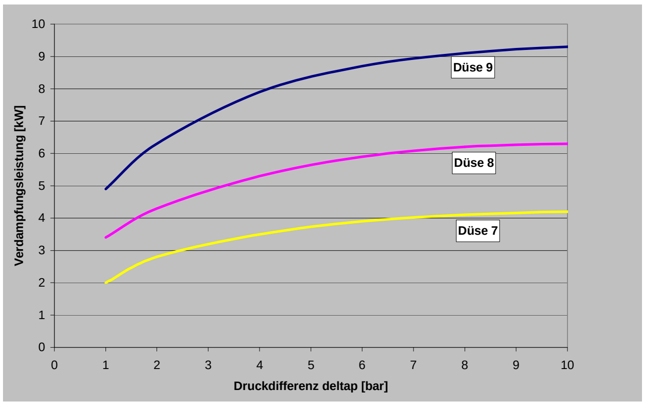

Figure 3: Refrigeration capacity of three different thermal expansion valves

Verdampfungsleitstung	Evaporation capacity
Druckdifferenz deltap (bar)	Pressure difference delta p (bar)

4.1. Calculation methods to determine the minimum condensation temperature

For the design of condensers and evaporators the values on the compressor connection sides are always given as the relevant pressure data for calculation in the corresponding equilibrium temperatures. This means that for the condenser the condensation temperature at the inlet of the condenser is given as a nominal value. But on the other hand, the evaporation temperature is given at the outlet of the evaporator. These definitions are practical, as the pressure losses during phase changes are device specific and should not have an effect on the greatest possible thermodynamic temperature difference. All the relevant standards also take account of this convention. Several pressure losses have to be overcome from when the refrigerant enters the condenser until it exits the evaporator.

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$$(4) \quad p_c = p_0 + \Delta p_{Rohrleitung} + \Delta p_{Expansionsventil} + \Delta p_{Verteiler} + \Delta p_{Verdampfer} + \Delta p_{Verflüssiger}$$

mit

$$p_c = \text{Verflüssigungsdruck}$$

$$p_0 = \text{Verdampfungsdruck}$$

$$\Delta p_{Verflüssiger} = \text{interner Druckverlust Verflüssiger}$$

$$\Delta p_{Rohrleitung} = \text{Flüssigkeitsrohrleitungsverluste inkl. Armaturen}$$

$$\Delta p_{Expansionsventil} = \text{Druckverlust Expansionsventil}$$

$$\Delta p_{Verteiler} = \text{Druckverlust Kältemittelverteiler}$$

$$\Delta p_{Verdampfer} = \text{interner Druckverlust Verdampfer}$$

Rohrleitung	Tubing
Expansionsventil	Expansion valve
Verteiler	Distributor
Verdampfer	Evaporator
Verflüssiger	Condenser
mit	where
Verflüssigungsdruck	Condensation pressure
Verdampfungsdruck	Evaporation pressure
Interner Druckverlust Verflüssiger	Internal pressure loss in condenser
Flüssigkeitsrohrleitungsverluste inkl. Armaturen	Fluid tubing losses, incl. fittings
Druckverlust Expansionsventil	Pressure loss, expansion valve
Druckverlust Kältemittelverteiler	Pressure loss, refrigerate distributor
Interner Druckverlust Verdampfer	Internal pressure loss, evaporator

It is assumed that the condensation and evaporation temperatures are known in the first step. The internal pressure losses of the condenser and the evaporator are calculated and provided in the Güntner configuration program GPC. But as a rule they are so small that they can be ignored. The GPC also calculates the pressure loss of the refrigerant distributor. The tubing and fitting losses must be calculated by the system designer. In this way it is possible to determine the dominating pressure difference across the expansion valve. With additional data, such as the type of refrigerant and the required refrigeration capacity, it is possible to choose a suitable expansion valve based on the manufacturer's data. This is described in Figure 4.

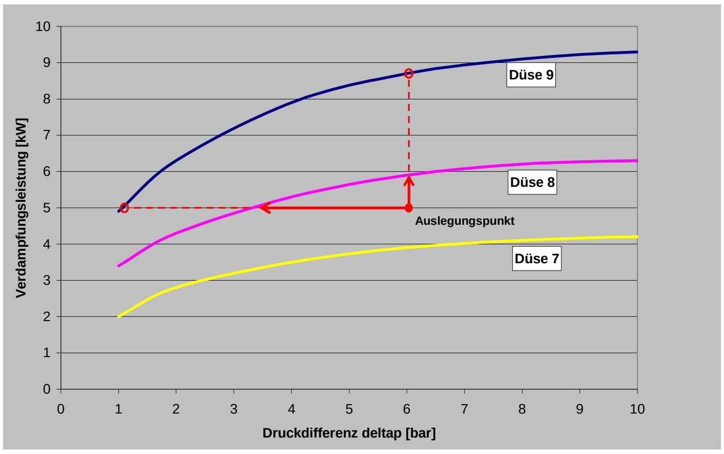


Figure 4: Configuration of thermal expansion valve and minimum permitted pressure difference

Verdampfungsleistung	Evaporation capacity
Auslegungspunkt	Design point
Düse	Nozzle
Druckdifferenz	Pressure difference

The position of the design point in Figure 4 is because under the original defined boundary conditions a pressure difference of 6 bar is available to the expansion valve and the evaporator is to provide nominal power of 5 kW. You would thus choose nozzle 8 as the smallest nozzle to provide the required refrigeration capacity. The nominal power of this valve in this condition is about 6 kW. As the evaporator can dissipate only 5 kW, the valve would adjust to an opening degree of about 85%. The expansion valve could also provide the nominal refrigeration capacity of 5 kW even at a pressure difference of 3.5 bar. In this condition the expansion valve would then be 100% open. But this also means that the condensation pressure could be reduced by 2.5 bar without the system falling below the nominal refrigeration capacity. The refrigeration system could be operated with a higher level of efficiency. If you chose a larger nozzle, you could even reduce the condensation pressure by 5 bar. However, in the configuration case (Δp = 6 bar) this would mean that the valve would have to be operated at just 50% open. This is already a borderline case for thermostatic expansion valves, as their control quality is reduced the more they are closed.
Consequently, the choice of expansion valve is always an optimisation task that must be resolved taking account of good control behaviour of the expansion valve and a low pressure difference. The load curve of the cooling unit should also be considered here. If rapid, large load fluctuations are expected in cooling operations, you would be better choosing smaller nozzles with better control. For example, in cold storage rooms that are filled only seldom you could also choose a larger nozzle.
As soon as you have chosen a nozzle, the minimum required pressure difference is also defined and, hence, the minimum possible condensation temperature for this refrigeration unit.
Since all Güntner frigen evaporators can be ordered for the refrigerants R134a and R404A/R507 with attached thermostatic expansion valve, a TEV module has been added to the Güntner configuration software GPC. With this module it is now possible to configure thermostatic expansion valves exactly to the operating conditions without having to change to a different program, such as from an expansion valve manufacturer, and re-enter the boundary conditions. The minimum condensation temperature and opening degree in the configuration are also calculated and output in this module.

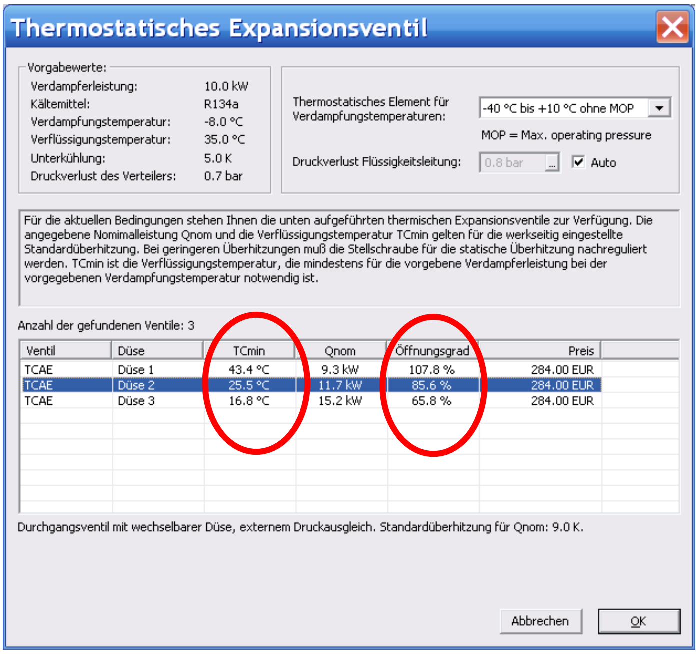


Figure 5: Configuration module for thermostatic expansion valves in GPC

In a refrigeration system with several refrigeration units or several evaporators, the minimum condensation temperature is defined for each evaporator. Consequently, the evaporator with the highest minimum condensation temperature determines the minimum condensation temperature for the entire refrigeration system. If just one expansion valve is chosen badly, there is a risk that the condensation temperature is increased for the complete system, which would mean more, and above all, unnecessary energy consumption.

5. Effect of outdoor temperatures on the condensation temperature based on actual temperature patterns

As mentioned in the previous section, the outdoor temperature and, consequently, the air intake temperature at the condenser have major effects on the condensation temperature. Figures 6 and 7 show the hourly average of the outdoor temperatures of three European cities over a year in two different ways.

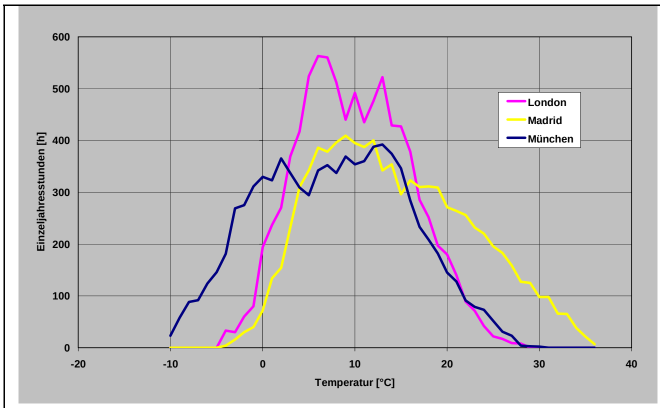


Figure 6: Temperature frequency distribution for one year for Madrid, London and Munich *(METEONORM 5.0; 2003)*

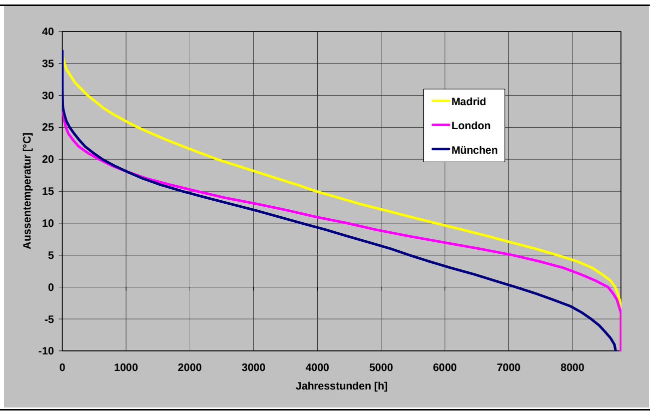


Figure 7: Cumulative temperature frequency distribution for one year for Madrid, London and Munich

Einzeljahresstunden	Individual yearly hours
Temperatur	Temperature
Außentemperatur	Outdoor temperature
Jahresstunden	Yearly hours

Figure 6 shows the number of hours in a reference year for every temperature in 1 degree Kelvin increments. It can be seen that in London the number of hours with very low or very high temperatures is low compared to the other two cities Munich and Madrid. London has more of a balanced climate with no extreme values. The average annual temperature is lowest in Munich and highest in Madrid. All of these properties have an effect on the energy consumption of a refrigeration system.
In Figure 7 the number of hours where the outdoor temperature was above a certain temperature are added together. This form of presentation can also be described as cumulative temperature frequency distribution. This presentation is especially suitable for illustrating the effects of lowering the condensation temperature.

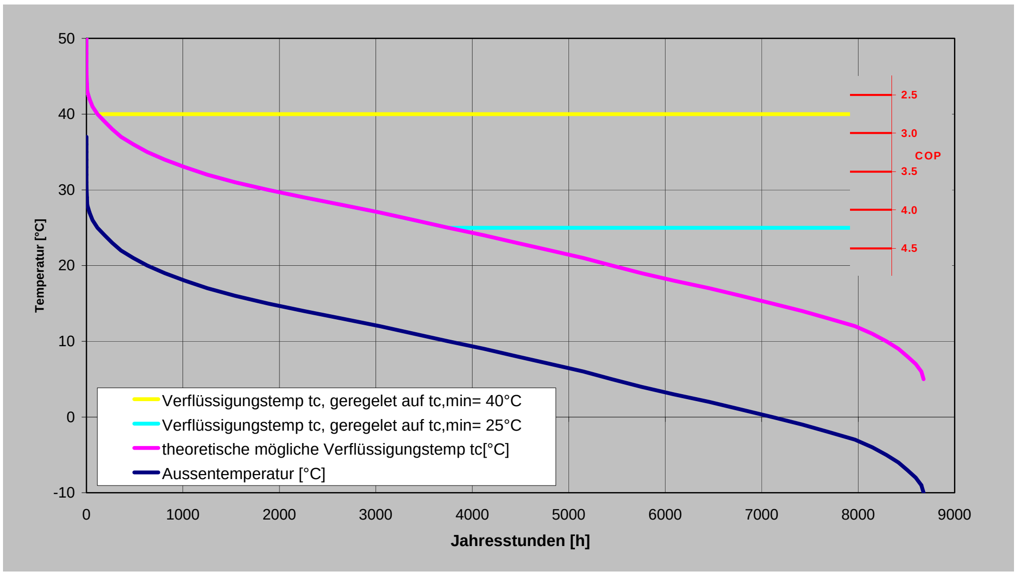


Figure 8: Auswirkungen eines reduzierten, geregelten Verflüssigungstemperatur am Beispiel von München Temperatur	Effects of a reduced controlled condensation temperature, using Munich as an example Temperature
Jahresstunden	Yearly hours
Veflüssigungstemp tc, geregelt auf tc, min = 40°C	Condensation temp. tc, controlled to tc, min = 40°C
Veflüssigungstemp tc, geregelt auf tc, min = 25°C	Condensation temp. tc, controlled to tc, min = 25°C
Theoretisch mögliche Verflüssigungstemp tc (°C)	Theoretically possible condensation temp. tc (°C)
Außentemperatur (°C)	Outdoor temperature (°C)

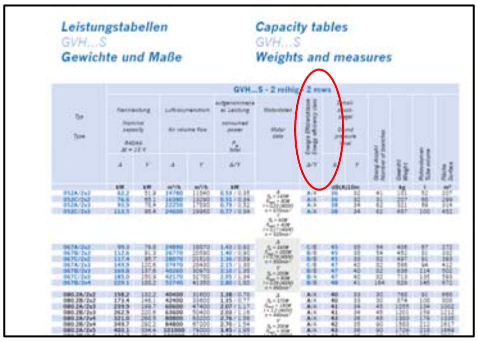
Figure 8 explains the effects of a reduced condensation temperature on the basis of the outdoor temperature distribution of Munich (blue line).
Assuming that a constant refrigeration capacity has to be provided over the entire 8,760 hours in a year, the temperature difference between the condensation temperature and the air intake temperature is influenced mainly by the size and heat transfer behaviour of the condenser (see Formula (3)). For the case shown in Figure 8 this temperature difference is 15 K. The theoretically required condensation temperature follows the air temperature with the same difference for the entire time. With consideration of actual temperature-related heat transfer coefficients and other technical side effects, this difference will show slight deviations from the original value that was assumed to be constant. However, these deviations are so small that they have absolutely no effect on the expressive power of the generalised case. In the case of the cooling load reduction with falling outdoor temperatures, e.g. for air conditioning systems, it is possible to determine a reduced difference via the formula (3), but for the sake of simplicity this will not be considered here.

Since a minimum condensation temperature has to be maintained due to the expansion valves that are used, the yellow and light blue lines were drawn in for different sizes of expansion valves. A minimum condensation temperature of 40°C was determined for a small expansion valve (yellow line). This means that the condenser fans can be operated at full speed for only a few very hot hours in the year. After the 200 hottest hours you would have to reduce the speed to maintain the minimum condensation temperature of 40°C. The refrigeration system is then operated at a COP of 2.7 regardless of the outdoor temperature. If larger valves were used, you would be able to achieve a minimum condensation temperature of 25°C. This means that the condenser fans could be operated at full speed for about 4,000 hours per year. As opposed to the first case, every hour that the fans run at full speed increases the overall efficiency of the system. The maximum achieved COP is now 4.2. This results in enormous energy savings; in this example 28%.
Unfortunately, in practice one often finds systems where the minimum condensation temperature has not been calculated. On site, a very high general value is set for the condensation pressure controllers "to be on the safe side". These systems are operated at bad levels of efficiency unnecessarily and with no benefit whatsoever. A simple adjustment of the potentiometer for the condensation pressure controller can often save cash immediately with no disadvantages (see Table 2). But you should check whether this measure causes a situation where in some refrigeration units the minimum condensation temperature for the evaporator is too low, which would mean that the full nominal refrigeration capacity would no longer be available for this refrigeration unit. However, these cases occur only in normal fan operation; that is, at lower outdoor temperatures than those assumed in the configuration case. In some applications the refrigeration load is also reduced and the reduced refrigeration capacity is adequate. Otherwise you could also consider changing the expansion valve nozzle for this evaporator.

6. Effect of the energy consumption of the condenser fans on the coefficient of performance (COP) of a refrigerant system

All the previous data about the COP of a refrigeration system consider only the drive energy at the compressor as input. Every condenser is equipped with electrically operated fans to increase the heat transfer and thus keep the surface area of the heat exchanger as small as possible. The drive energy of these fans can be considered in the overall COP of the refrigeration system.
For several years participants in the EUROVENT certification program have been obliged to state the energy efficiency class in their catalogues. The energy efficiency class is a measure for the ratio of the electrical drive power to the standard condenser power (see Table 3). Powerful fans with a high speed and a large air volume require large electrical drive powers and are thus classified as having "high energy consumption". Weaker fans with a lower speed require less drive power, transport less air and are thus classified as low energy consumers.

Class	Energy consumption	R
A	extremely low	R > 110
B	very low	70 < R < 110
C	low	45 < R < 70
D	medium	30 < R < 45
E	high	30 < R



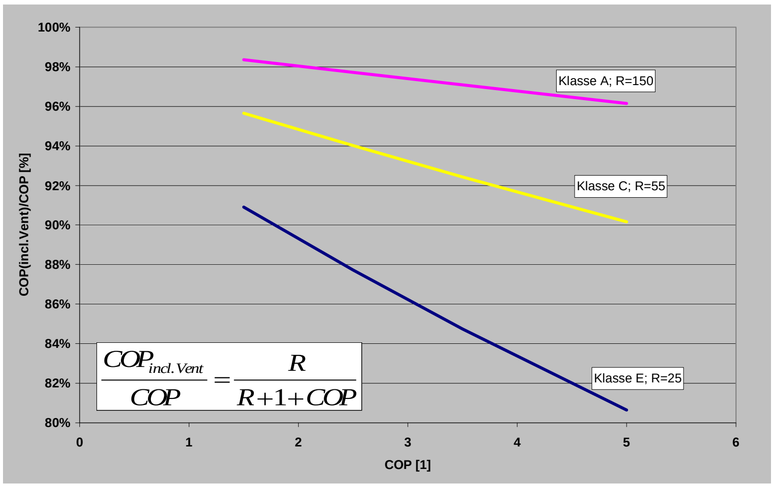


Table 3: Energy efficiency class acc. to EUROVENT, catalogue data

Figure 8: Percentage reduction of COP if the fan power is taken into account in relation to the energy efficiency class of a condenser

Figure 8 shows the decline of the previously used COP without fan drive power if this power is considered. This decline also depends on the original, simple COP. Two statements can be taken from this diagram:

- The use of fans with a high energy consumption can lead to a considerable decline in the overall COP (up to 20%).
- The negative effect of the fan power on overall efficiency is more serious the higher the COP of a refrigeration system is.

If you want an energy-saving refrigeration system, you must also use energy-saving fans. But as these energy-saving fans have a lower air capacity, devices with a larger heat exchanger surface area have to be chosen. This also has the advantage that the noise emission is less due to the lower speed of the fans, which is a very welcome side effect to energy saving.

7. Summary

The condensation temperature has a major effect on the efficiency of a refrigeration system. The lower this is, the lower the drive energy for compression, which automatically reduces the running costs of the refrigeration system.
There are various ways of lowering the condensation temperature. The size of the heat exchanger surface area and the type and control of the condenser fans have the biggest influence. In practice very many systems are still operated with unnecessarily high condensation temperatures, as the connection between the choice of expansion valve and the condensation temperature have not been calculated in detail and they thus run with relatively high safety allowances. These calculations can now be carried out very easily with the TEV module of Güntner's configuration software.
The choice of condenser fans can also help save energy. The classification of the energy efficiency of condensers in catalogues provides important information for choosing the right system. The environment benefits when highly efficient condensers are chosen, not only as a result of saving energy and running costs, but also due to the associated noise reduction.