Technical article

Energetic considerations for regulatory systems for condensers under partial load

Peter Roth

Head of Basic Development

R&D Department

Core topics:

- Influence of the condensation temperature on the COP
- Controlling the condensation temperature via ambient temperature-dependent set point shifting (extended standard)as opposed to controlling via a fixed limit value for the condensation temperature
- New control algorithm Energy Balance Function: Minimization of joint power consumption of both the compressor and the condenser fans
- Resulting energy saving potentials when reducing speed of condenser fans while the power consumption of the compressor increases slightly

Keywords:

Condensers, energy consumption, control, optimization

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

Introduction

It is well known that the condensing temperature chosen for a cooling circuit has a considerable effect on its COP. The lower the system's condensing temperature, the higher will be its efficiency and the less energy it will need to achieve a given refrigeration or air-conditioning performance. It is usual to consider the power consumed by the compressor to be the only power expended, often taking its efficiency into account. If the total power consumption of the refrigeration or air-conditioning unit is taken to include the electrical power consumed by the condenser fans, the above correlation will be correct under design conditions, i.e. at full load. But as soon as the refrigeration unit begins to operate at partial load, or under different operating conditions such as higher ambient temperatures, the power consumption of the condenser fans may increase to the same order as that of the compressor itself, or even higher. In these circumstances it may even be more efficient to reduce the speed of the condenser fans. This will lead to a higher condensing temperature and hence to higher power consumption in the compressor, but this needs to be weighed against the energy saved by the fans in order to judge whether energy can overall be saved in this way.

Previously, the condensing temperature of refrigeration and air-conditioning equipment was almost always regulated using P or PI controllers with one or two fixed set points. This article describes one familiar and one entirely new regulation strategy that aim to save more energy than these standard controllers.

Ambient temperature-dependent set point shifting

When refrigeration and air-conditioning equipment is operated with an energy-conscious low set point for the condensing temperature, alert system operators have often observed that, in hot weather, the system's condenser fans are running unexpectedly fast, even when the system load is low, for example while production is at a standstill. However, this situation will only stand out as unexpected if the system's load situation is explicitly known, because high outdoor temperatures do always lead one to expect high condenser fan speeds. Such cases do not occur in systems with higher condensing set points and therefore higher energy consumption; because the condenser fans are switched to normal operation at an early stage to ensure that the condensing temperature does not drop below its set point. This relationship is clearly illustrated by the following diagram:

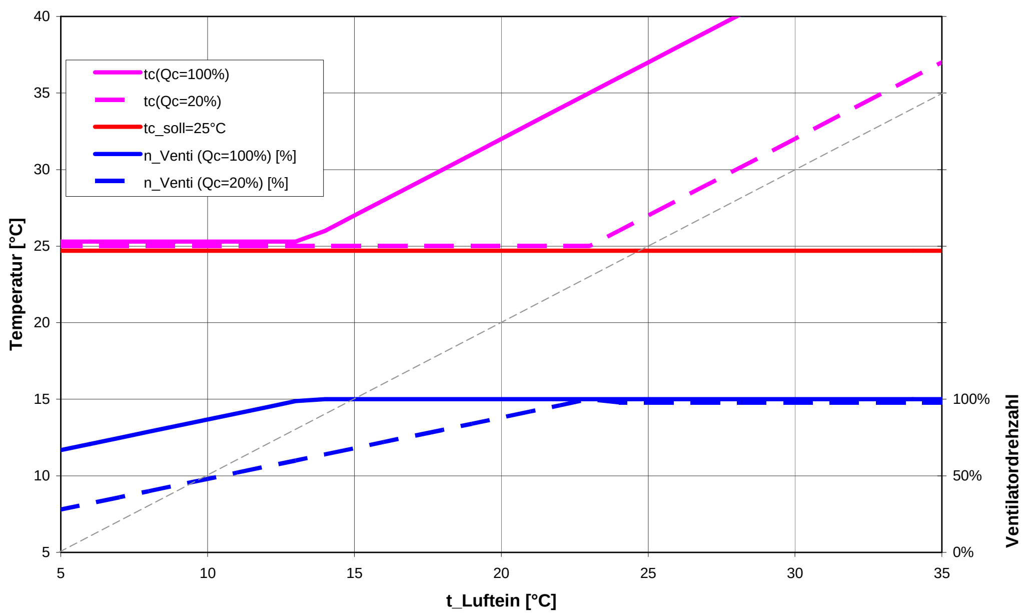

Diagram 1: Standard condenser regulation with tc_set point = 25 °C for a unit designed with dt1 = 12 K

Temperatur [°C]	Temperature [°C]
t_Luftein [°C]	t_air_intake [°C]
Ventilatordrehzahl	Fan speed
tc_soll = 25°C	tc_setpoint = 25°C
n_Venti (Qc = 100%) [%]	n_fan (Qc = 100%) [%]
n_Venti (Qc = 20%) [%]	n_fan (Qc = 20%) [%]

This example assumes a condenser designed for an intake temperature difference (difference between condensing and air intake temperatures) of 12 K. This means that under full load, the condensing temperature is always 12 K higher than the air intake temperature. Viewed simply, this is initially independent of the absolute air intake temperature. For example, with an air intake temperature of 28 °C and condenser fans running at full speed, the system will settle down to a condensing temperature of 40 °C. Let us also assume that the condensing set point on the controller has been set to 25 °C. If the air intake temperature drops below 13 °C, the controller will begin to reduce the fan speed, even if the system is running at full load, to prevent the condensing temperature dropping below its set point. Allowing this to happen could violate the design conditions of the expansion valve and cause the evaporator to lose performance.

If the system is running at partial load, for instance at weekends, when production does not require so much refrigeration, then assuming the fans are still operated at their nominal speed, the intake temperature difference will be reduced. The reason for this is the physical fact that, to a first approximation and under identical ancillary conditions, the performance of a heat exchanger is proportional to the driving temperature difference.

This case is also covered by Diagram 1. At an air intake temperature of 23 °C, the controller begins to reduce the fan speed. If the air intake temperature is higher than this, we now get the undesirable situation where the fans are running at full speed although only a small amount of refrigeration or air-conditioning is needed. This may even lead to the condenser fans consuming more power than the compressor, which is not at all satisfactory, energy-wise. This same situation could equally be expressed by saying that, from a certain external temperature, a conventional compressor controller can no longer regulate the system according to the load, or can no longer recognise the load situation of the refrigeration or air-conditioning unit.

To relieve this situation, Güntner’s condensing pressure controllers have long incorporated the optional operating mode “ambient temperature-dependent set point shifting”. In this mode, as is customary, the set point has to be defined; it is sensible to choose here the lowest admissible condensing temperature. Additionally, a second parameter measured in Kelvin, namely the “set point shift”, has to be defined. The controller also needs to know the current air intake temperature, which is obtained using a temperature sensor. This method is based on the following rule:

The set point used to regulate the fan speed will be the value specified to the controller, except where the air intake temperature plus the shift gives a higher value. If it does, then this total is used as the new set point.

$$tc\_soll = \max\begin{cases} t\_LE + Schiebung \\ tc\_min \end{cases}$$

The current condensing set point is now no longer fixed: whenever the outdoor temperature exceeds a specific value the set point will move with it, at a fixed temperature difference – the “shift” – to the current air intake temperature. The resultant operating conditions are shown in Diagram 2.

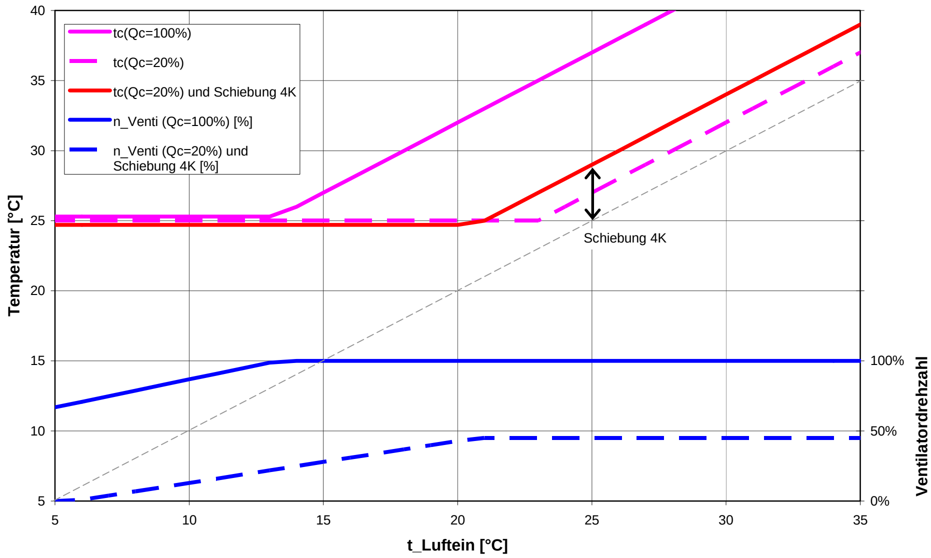

Diagram 2: Ambient temperature-dependent set point shifting with tc_set point = 25 °C and a shift of 4 K for a unit designed for dt1 = 12 K

Temperatur [°C]	Temperature [°C]
t_Luftein [°C]	t_air_intake [°C]
Ventilatordrehzahl	Fan speed
tc (Qc = 20%) und Schiebung 4K	tc (Qc = 20%) and shift 4K
n_Venti (Qc = 100%) [%]	n_fan (Qc = 100%) [%]
n_Venti (Qc = 20%) und Schiebung 4K [%]	n_fan (Qc = 20%) and shift 4K [%]
Schiebung 4K	Shift 4K

The design conditions and ancillary constraints stated for Diagram 1 have been retained, i.e. tc_min = 25 °C and dt1 = 12 K. The newly introduced shift is assumed to be 4 K. In operating situations with a high refrigeration or air-conditioning load the speed of the condenser fans is unchanged. In a full load situation, the condensing temperature will always be 12 K higher than the air intake temperature, and will therefore behave exactly as in the standard controller solution. Moreover, if the air intake temperature drops below 13 °C the fan speed will, as before, be reduced. If the intake temperature difference at the condenser now drops appreciably, as a result of low refrigeration or air-conditioning load, the condensing temperature resulting from fans running at full speed with a high air intake temperature will then be lower than the total of air intake temperature and shift. Since, in controller mode “ambient temperature-dependent set point shifting”, this total represents the current set point, the controller will need to reduce the fan speed to avoid dropping below the new set point. And this is exactly the reaction we are looking for. The speed and hence the power consumption of the condenser fans is reduced while the system is running at partial load, even though the condensing temperature is higher than its configured 25 °C set point.

It should not be forgotten that this will result in a lower COP for the refrigeration system as compared to a conventional controller, because we have deliberately accepted the higher condensing temperature caused by reducing the fan speed. However, the overall electrical power consumed by the refrigeration system, i.e. the total consumption of the compressor and the condenser fans, may very well be lower than without the “ambient temperature-dependent set point shifting” option, i.e. at 100 % fan speed.

This effect is relevant because the power consumption of the fans increases with the cube of their speed, while the airflow volume increases only linearly with the speed. Among other factors, the refrigeration unit’s overall power consumption is determined by the partial thermal load on the condenser, the unit’s COP and the energy efficiency class of the condenser and condenser fan. It follows that an appropriate value for the shift will thus depend on these parameters.

The size of the shift can be calculated or estimated from the system data, or can be derived empirically. Basically, the higher the COP of the refrigeration or air-conditioning unit the higher should be the value chosen for the shift, because higher COP values correspond to a lower ratio of compressor capacity to fan power consumption. This means that the effect of the condenser fans' energy consumption is an increasingly significant proportion of the power consumption of the system as a whole. The shift chosen for an air-conditioning system will thus be higher than for deep-freeze applications, where it may be possible to entirely dispense with it because the system's overall power consumption will always be dominated by the compressor.

Moreover, the effective shift value for a low energy efficiency class condenser will always he higher than one for condensers in a higher class, which is obvious really, because the energy efficiency class is defined as the ratio of the nominal condenser capacity Q_c_nom to the fans' electrical power consumption P_el. All condenser manufacturers certificated by Eurovent provide this information for all their equipment.

Equation 1

$$R = \frac{\dot{Q}_{c\_nom}}{P_{el}}$$

Class	Energy consumption	R
A++	extraordinarily low	240 < R
A+	extremely low	160 < R < 240
A	very low	110 < R < 160
B	low	70 < R < 110
C	medium	45 < R < 70
D	high	30 < R < 45
E	very high	30 < R

A condenser in energy efficiency class E will need to consume more than three times as much electrical drive energy in order to achieve the same thermal condenser capacity as one in energy efficiency class A. It follows that fans in a lower energy efficiency class need to be switched to a slower speed sooner than those in a higher energy efficiency class.

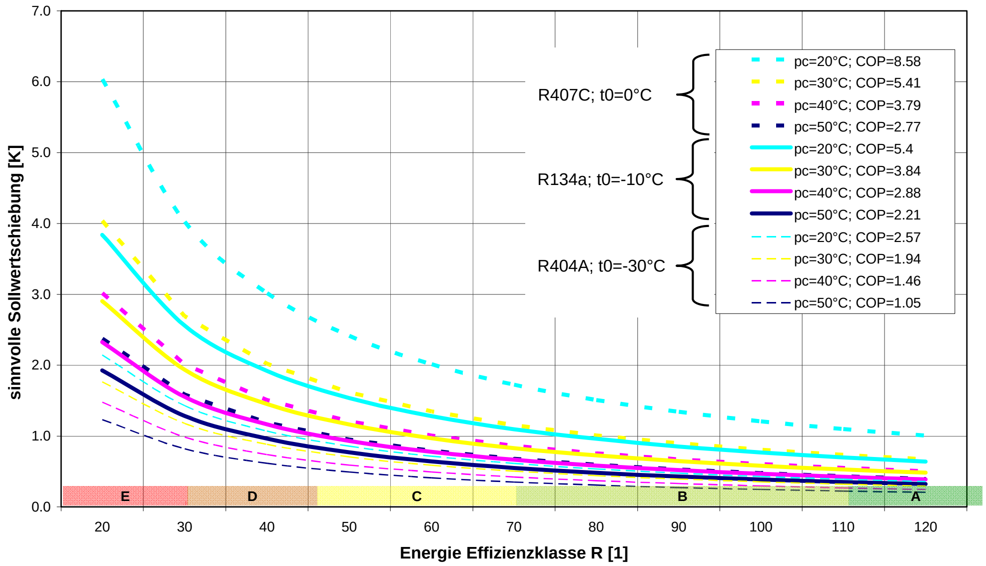

Diagram 3: Appropriate values for "ambient temperature-dependent set point shifting" for a unit designed for dt1 = 12 K and an isentropic condenser efficiency of 0.7

maximal sinnvolle Sollwertschiebung [K]	Maximum useful setpoint shift [K]
Energie Effizienzklasse R [1]	Energy efficiency class R [1]

The above diagram summarises a number of useful set point shift values for specific thermal conditions and a condenser designed for 12 K.

The "ambient temperature-dependent set point shifting" option is an easy way of reducing the total energy consumption of a refrigeration or air-conditioning unit under partial load, but this is only the first step of a genuine optimisation.

energy balance function

It would be advantageous to know the operating point that represents the minimum energy consumption of a refrigeration or air-conditioning unit under given ancillary conditions. This means not merely reducing the power consumption of the compressor drive and condenser fan, but minimising their total value, i.e. finding the energetically optimal operating point. The fact that there must be such a minimal operating point is clear from the following diagram.

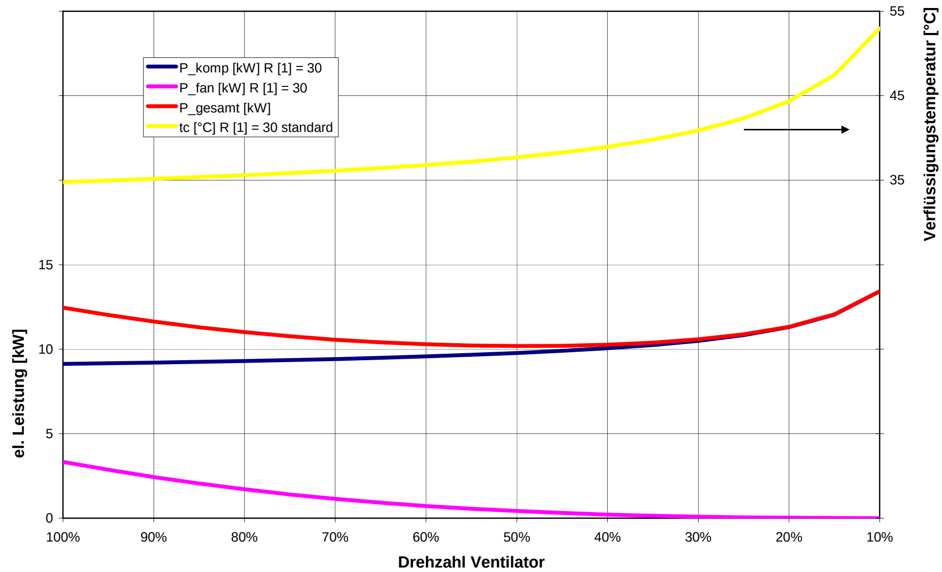

Diagram 4: Condensing temperature and power consumption of compressor and condenser fan with a partial refrigeration load of 40 %,
Ancillary conditions: R134a; t0 = -10 °C; t_air_intake = 27 °C; tc_min = 30 °C

el. Leistung [kW]	Electrical power [kW]
Drehzahl Ventilator	Fan speed
Verflüssigungstemp [C°]	Condensing temperature [°C]

For the purposes of this example we assume a partial load on the system of 40 %. The power consumption of the fan and the compressor are shown separately for varying fan speed. The absolute value of the fan's power consumption depends on both the energy efficiency class of the compressor and the third power of its current speed. The absolute value of the compressor's power consumption depends on the mass flow rate of the refrigerant, i.e. from the partial load, and from the (varying) current COP of the system. Assuming the evaporation temperature is held constant, the COP depends primarily on the condensing temperature, which is also shown in the diagram. The refrigerant and the compressor used, together with its efficiency characteristics, are assumed to be given.

As the fan speed is reduced, the condensing temperature will increase in order to maintain the specified air-conditioning or refrigeration performance. This leads to a reduction of the COP, which is reflected in the compressor's increased power consumption. The sum of these two power consumption figures exhibits a recognisable minimum at about 50 % fan speed. In this example we have thus determined the most energy efficient operating point, subject to having stipulated as known the refrigerant, the compressor, the energy efficiency of the condenser, the partial load, the evaporation temperature, the air intake temperature and the minimum condensing temperature (which in this case has no effect).

We shall now show what a generally valid mathematical model for this task might look like, and what conclusions we can derive from it.

Our aim is to set up a fundamental equation that describes all the possible operating conditions for the condenser of a refrigeration or air-conditioning unit depending on all the relevant influencing factors.

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Equation 2 $$f\,(Q_c, Q_{c_d}, t_c, t_{air\_intake}, dt1_d, n, n_d, ...) = 0$$

This equation includes a number of parameters that were laid down in our premise and do not need to be varied, such as the nominal condenser capacity Q_c_d, nominal intake temperature difference dt1_d and nominal fan speed n_d. The current air intake temperature t_air_intake and current condenser capacity Q_c are not fixed quantities as such, but they are not varied during our search for the energetic minimum because they can be considered to be fixed ancillary conditions for a given moment in time. In Equation 3 these unvarying quantities are underlined.

Equation 3 $$f\,(\underleftrightarrow{Q_c}, \underline{Q_{c_d}}, t_c, \underleftrightarrow{t_{air\_intake}}, \underline{dt1_d}, n, \underline{n_d}, ...) = 0$$

As variable quantities there remain the fan speed n and the condensing temperature tc. If either of these is known, the other follows, and we might represent this as solutions to our fundamental equation in the two variables, as shown in Equation 4 and Equation 5.

Equation 4 $$n = f_n(Q_c, t_c, ...)$$

Equation 5 $$t_c = f_{t_c}(Q_c, n, ...)$$

Although the scope and complexity of our fundamental equation mean that an analytically correct solution is not possible, it is nevertheless possible to find an arbitrary number of pairs of values for tc and n that fulfil the equation. Each such combination of tc and n represents a set of possible and feasible operating conditions for the condenser.

Equation 6
$$\begin{gathered}
f\,(\underleftrightarrow{Q_c}, \underline{Q_{c_d}}, t_c\langle 1\rangle, \underleftrightarrow{t_{air_intake}}, \underline{dt1_d}, n\langle 1\rangle, \underline{n_d}, ...) = 0 \\
f\,(\underleftrightarrow{Q_c}, \underline{Q_{c_d}}, t_c\langle 2\rangle, \underleftrightarrow{t_{air_intake}}, \underline{dt1_d}, n\langle 2\rangle, \underline{n_d}, ...) = 0 \\
f\,(\underleftrightarrow{Q_c}, \underline{Q_{c_d}}, t_c\langle 3\rangle, \underleftrightarrow{t_{air_intake}}, \underline{dt1_d}, n\langle 3\rangle, \underline{n_d}, ...) = 0 \\
\vdots
\end{gathered}$$

For each of these sets of possible operating conditions we can determine the power consumed by the compressor and the condenser fans, and hence the total consumption.

Equation 7 $$P_g = P_{el\_compressor} + P_{el\_fan}$$

Equation 8 $$P_g = P_{el\_compressor}(Q_c, COP(refrigerant, t_c, t_0, \eta(t_c, t_0)), \ldots) + P_{el\_fan}(R, n, \ldots)$$

As we already explained, the current condenser capacity Q_c and the system's coefficient of performance, COP, depending among other things on the refrigerant and the compressor performance, are the principal factors affecting the power consumption of the compressor. The principal factors affecting the power consumption of the fans are the energy efficiency of the condenser and their speed. Now all we need to do is find the pair of values for tc and n that produces the lowest overall power consumption P_g.

Equation 9

$$\begin{aligned} P_g\langle 1\rangle &= P_{el\_comp}(t_c\langle 1\rangle,\ldots) + P_{el\_fan}(n\langle 1\rangle,\ldots) \\ P_g\langle 2\rangle &= P_{el\_comp}(t_c\langle 2\rangle,\ldots) + P_{el\_fan}(n\langle 2\rangle,\ldots) \\ P_g\langle 3\rangle &= P_{el\_comp}(t_c\langle 3\rangle,\ldots) + P_{el\_fan}(n\langle 3\rangle,\ldots) \\ &\vdots \end{aligned}$$

To illustrate this mathematically, each value of the partial load corresponds to a curved surface in three-dimensional space for which we have to find the minimum. If we go a step further and include the partial load as an additional parameter, we can generate a surface for the partial load and the condensing temperature whose height is then the overall power consumption. Applying Equation 4, we could also use the condenser fan speed for the second coordinate axis instead of the condensing temperature, since these two values, although not analytically resolvable, are implicitly directly interdependent.

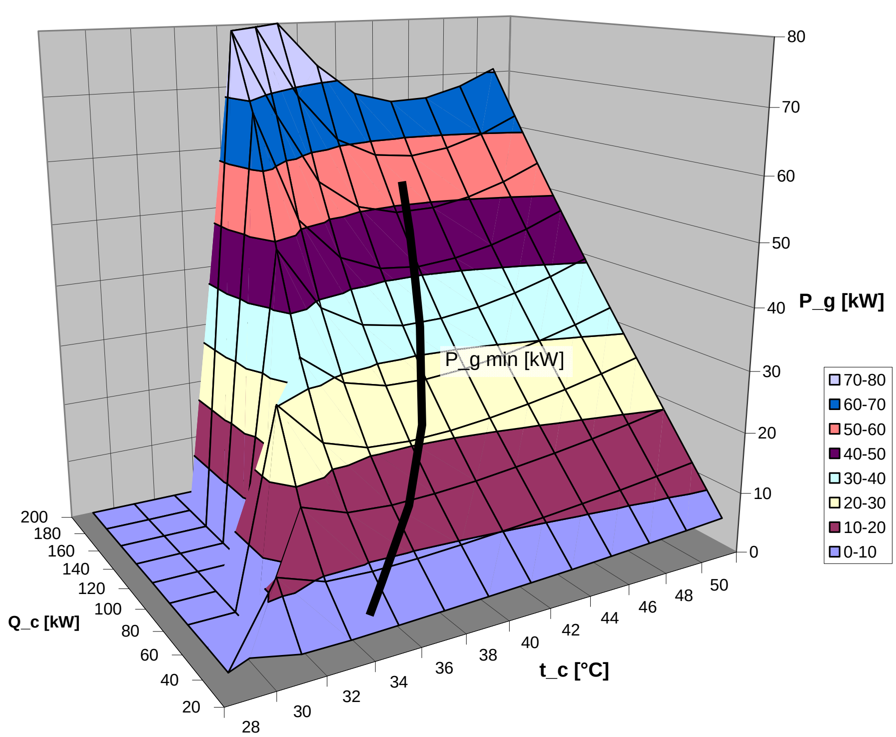

Diagram 5: Overall power consumption of a refrigeration plant depending on the condensing temperature and the current condensing power

At each lowest point on this surface, or we might say "along the river bed of this mountain", we find the energetically optimal condensing temperature and fan speed for each value of the partial load.

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Concrete results from the energy balance function

On the basis of these general correlations we have set up specific systems of equations containing actual materials data for refrigerants, characteristics of selected compressor designs and the thermal operating characteristics of Güntner condensers. Luckily, we find that these systems of equations are extremely "good-natured". It was possible to fulfil their convergence criteria in just a few iterations.

A number of these optimisation results are shown below.

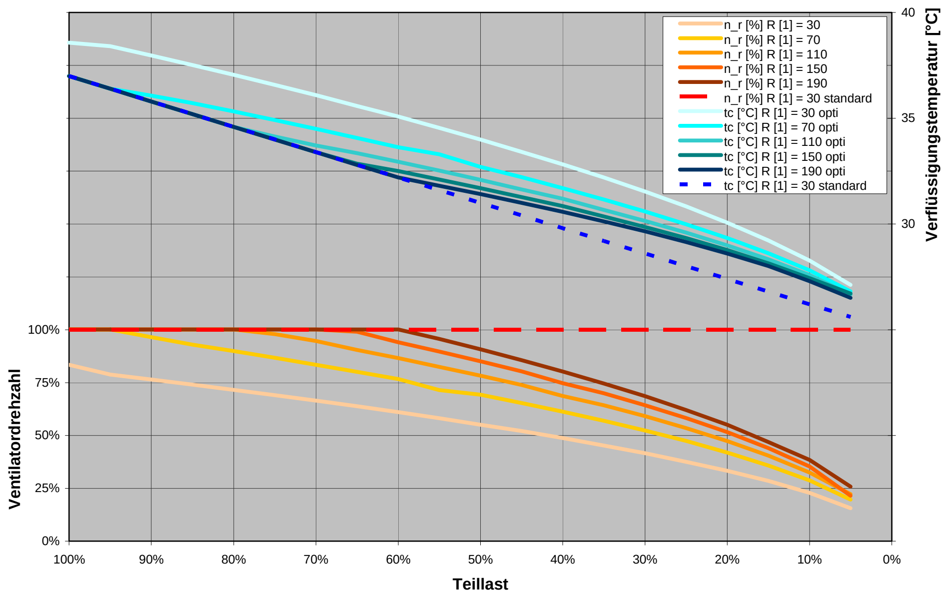

Diagram 6: Optimised condensing regulation compared to standard regulation, R134a; t0 = -10 °C; t_air_intake = 25 °C; tc_min = 25 °C

Ventilatordrehzahl	Fan speed
Teillast	Partial load
Verflüssigungstemperatur [°C]	Condensing temperature [°C]

For each of the different energy efficiency classes, Diagram 6 shows the relative condenser fan speeds and the resulting condensing temperatures mapped against the partial load. Here the chosen refrigeration unit used a medium-range reciprocating compressor and the refrigerant R134a, with an evaporation temperature of -10 °C, an air intake temperature of 25 °C and a minimum condensing temperature of 25 °C. The condenser was designed for an intake temperature difference of 12 K. The broken lines show the system's behaviour under standard regulation. Standard regulation attempts to reach and maintain the minimum condensing temperature without imposing any additional conditions (the special case of "ambient temperature-dependent set point shifting" is not considered here).

When the refrigeration system is running at full load, the operating conditions under standard regulation and energy balance function are identical, except for the condenser with the worst energy efficiency of 30. It is interesting that for this fan even at full load the energetic optimum makes it necessary to reduce the speed to 80 %, even though this increases the condensing temperature to above 38.5 °C. If this fan were to be operated at full speed, the condensing temperature would be 37 °C, but the system's overall power consumption would still be higher than at 80 % of the speed.

The lower the refrigeration load required of the system the more we need to reduce the fan speed, even when those fans have a high energy efficiency rating. When the partial load drops below 50 % it becomes energetically desirable to reduce the speed of even highly economical fans, such as those with an energy efficiency rating of 190. With standard regulation, regardless of the energy efficiency, the fan would be operated at its maximum speed even for an extremely small refrigeration load, because with an air intake temperature of 25 °C there is no possibility of reaching the 25 °C set point.

We have previously spoken in general terms of “energetically optimised operating conditions”, but have not yet quantified the absolute energy consumption figures. In Diagram 7 we therefore show the actual energy savings for the various energy efficiency classes, depending on the current partial load.

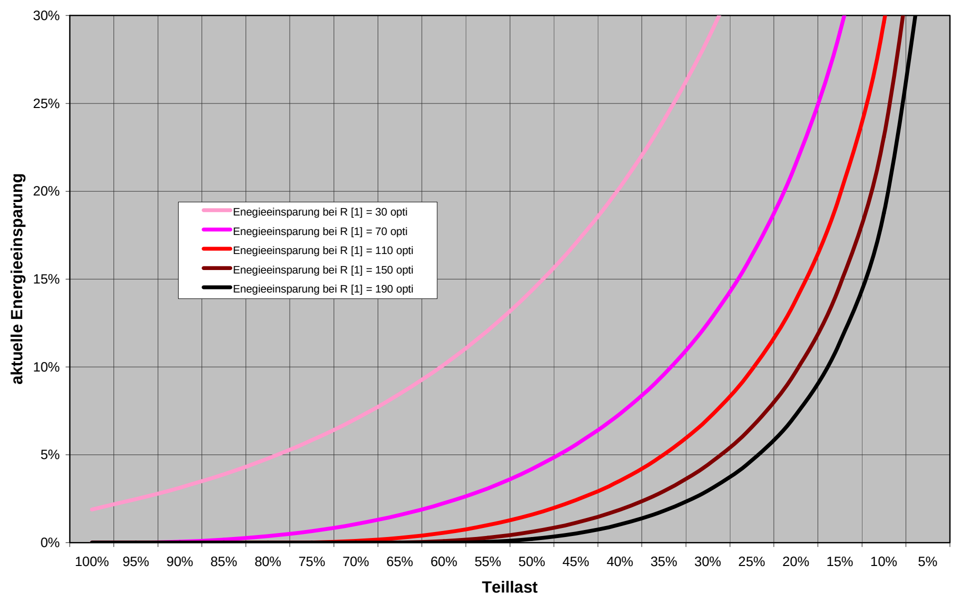

Diagram 7: Energy saved by optimised condensing regulation compared to standard regulation, R134a; t0 = -10 °C; t_air_intake = 25 °C; tc_min = 25 °C

aktuelle Energieeinsparung	Actual energy saved
Teillast	Partial load
Energieeinsparung bei R [1] = 30 opti	Energy saving for R [1] = 30 optimised
Energieeinsparung bei R [1] = 70 opti	Energy saving for R [1] = 70 optimised
Energieeinsparung bei R [1] = 110 opti	Energy saving for R [1] = 110 optimised
Energieeinsparung bei R [1] = 150 opti	Energy saving for R [1] = 150 optimised
Energieeinsparung bei R [1] = 190 opti	Energy saving for R [1] = 190 optimised

Here, the astonishing case of a condenser with energy efficiency 30 needing to be regulated down even at full load is placed in proportion by the fact that this saves relatively little energy over and above the standard regulation method, in other words, this is a “shallow energy minimum”. However, the further we go into the partial load range, the clearer become the potential energy savings. For example, this graph shows that, under a partial system load of 20 %, we can save more than 12 % of the total energy consumed using the conventional standard regulation method, even for a condenser with an energy efficiency rating of 110. We should however be aware that for this partial load, even the total power consumption is only about 15 % of that of a full load.

For the purpose of understanding the energy balance function it is useful to compile a graph showing the condensing temperature and fan speed against the air intake temperature. Diagram 8 shows such a curve for a partial load of 25 %. At low air intake temperatures there is no difference between the two methods of regulation, because the fan speed needs to be reduced in exactly the same way to maintain the minimum condensing temperature of 25 °C. From an ambient temperature of around 20 °C the fan speed is increased only very slowly. This causes the condensing temperature to rise faster than it would have done under standard regulation, but the increased energy consumption caused by this slightly higher condensing temperature is not so great as would have been caused by increasing the fan speed. Here, again, we see that the relative fan speeds of condensers with higher energy efficiency may be higher, because improving the COP has a greater effect.

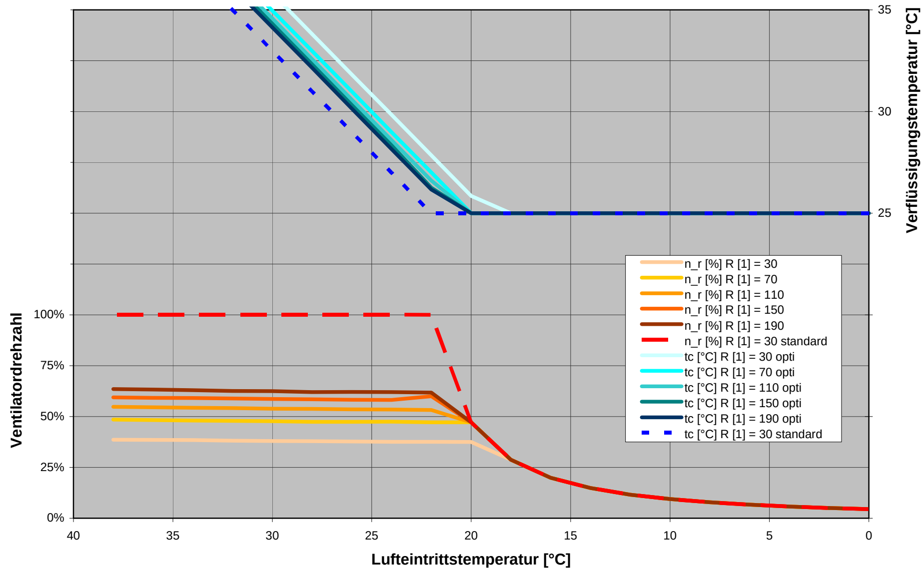

Diagram 8: Optimised condensing regulation compared to standard regulation, R134a; t0 = -10 °C; partial load 25 %; tc_min = 25 °C

Ventilatordrehzahl	Fan speed
Lufteintrittstemperatur [°C]	Air intake temperature [°C]
Verflüssigungstemp [°C]	Condensing temperature [°C]

In Diagram 9 we have assumed a day in the height of summer, with an air intake temperature of 35 °C and increased the minimum condensing temperature from 25 °C to 40 °C. Such high set points, forcing high and energetically disadvantageous condensing temperatures, are still frequently encountered in practice, because installers and users neglect to precisely design the system’s expansion valves. But even under these ancillary conditions the energy balance function is able to identify potential savings.

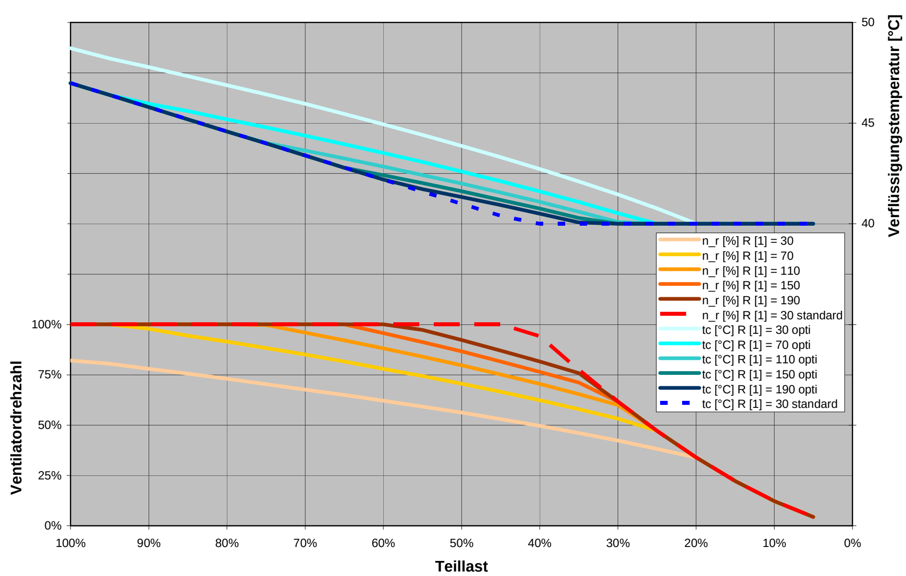

Diagram 9: Optimised condensing regulation compared to standard regulation, R134a; t0 = -10 °C; t_air_intake = 35 °C; tc_min = 40 °C

Ventilatordrehzahl	Fan speed
Teillast	Partial load
Verflüssigungstemperatur [°C]	Condensing temperature [°C]

In this case, from a partial load of 40% the standard regulation method would reduce the fan speed to avoid the condensing temperature dropping below 40°C. GMM-eta is also bound by this set point and throttles back depending on the energy efficiency of the fan when the condensing temperature reaches 40°C. For small partial loads there is thus no difference between GMM-eta and the standard regulation method, nor can the energy situation be any better, as Diagram 10 shows. However, depending on the energy efficiency of the compressor, for partial loads of between 50% and 40% it is possible to achieve a marked reduction in overall power consumption compared to standard regulation.

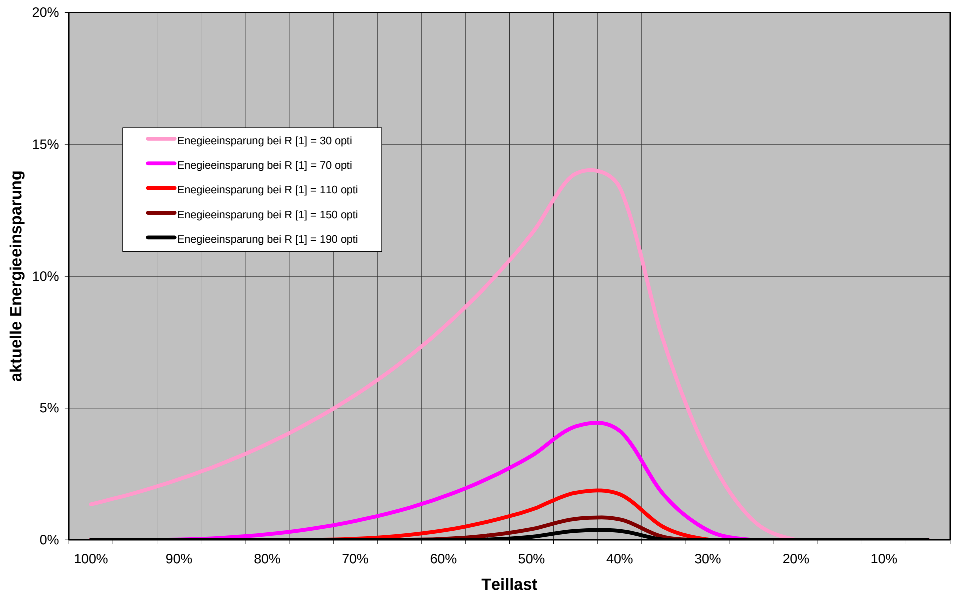

Diagram 10: Energy saved by optimised condensing regulation compared to standard regulation, R134a; t0 = -10 °C; t_air_intake = 35 °C; tc_min = 40 °C

aktuelle Energieeinsparung	Actual energy saved
Teillast	Partial load
Energieeinsparung bei R [1] = 30 opti	Energy saving for R [1] = 30 optimised
Energieeinsparung bei R [1] = 70 opti	Energy saving for R [1] = 70 optimised
Energieeinsparung bei R [1] = 110 opti	Energy saving for R [1] = 110 optimised
Energieeinsparung bei R [1] = 150 opti	Energy saving for R [1] = 150 optimised
Energieeinsparung bei R [1] = 190 opti	Energy saving for R [1] = 190 optimised

As we explained before, the COP of a refrigeration or air-conditioning unit is one of the significant factors affecting the potential savings from optimised operation of the condenser fans. This becomes obvious if we compare the fixed drive power of the condenser fan with the drive power of the compressor, which steadily decreases as the COP increases. Thus the choice of refrigerant, the specified evaporation temperature and the efficiency of the compressor together determine the potential savings from the use of the energy balance function. In Diagram 11 you can see a number of refrigerants with different evaporation temperatures mapped against the partial load.

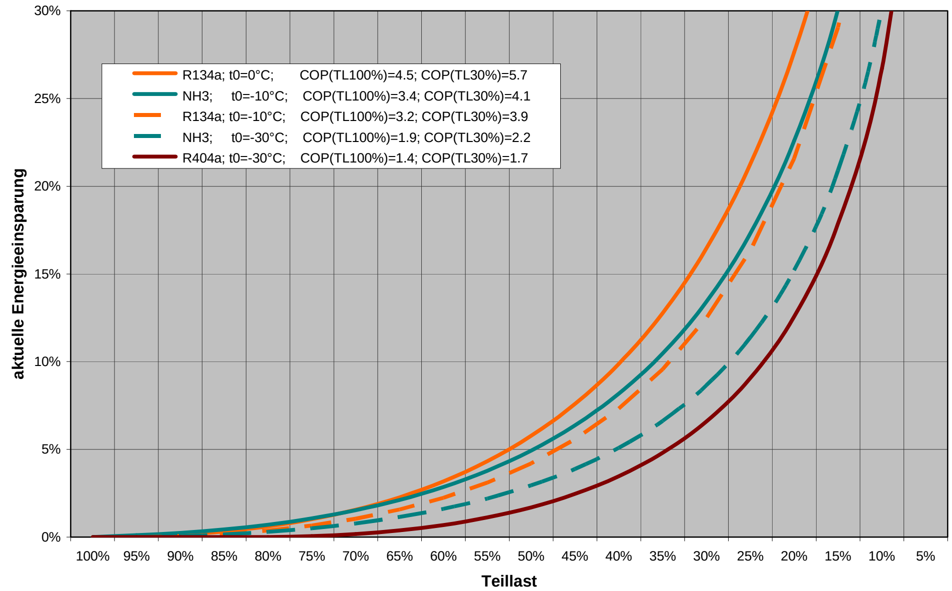

Diagram 11: Energy saved by optimised condenser regulation compared to standard regulation with different refrigerants and evaporation temperatures; energy efficiency = 70; t_air_intake = 25 °C; tc_min = 25 °C

aktuelle Energieeinsparung	Actual energy saved
Teillast	Partial load

As you would expect, the air-conditioning application using R134a and an evaporation temperature of 0 °C achieves the best savings of the solutions shown here. For an ordinary refrigeration application with an evaporation temperature of -10 °C, the energy saved with ammonia and R134a are almost identical. Also as we expected, the worst result is that for the deep-freeze application using R404a, because its COP is lower than that of all the four other applications.

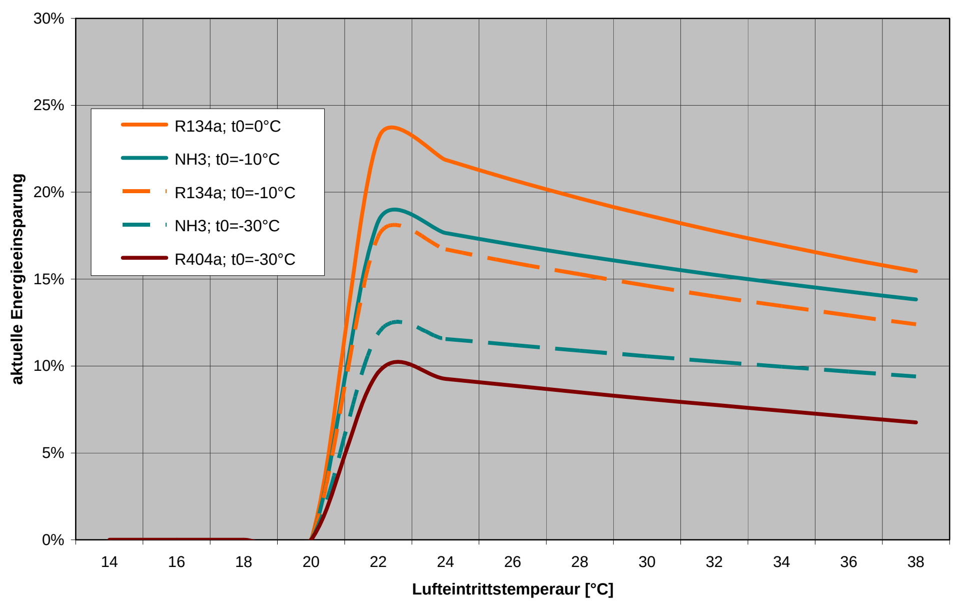

Diagram 12: Energy saved by optimised condenser regulation compared to standard regulation with different refrigerants and evaporation temperatures; energy efficiency = 70; partial load = 25 %; tc_min = 25 °C

aktuelle Energieeinsparung	Actual energy saved
Lufteintrittstemperatur [°C]	Air intake temperature [°C]

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This behaviour is also reflected in the graph showing the air intake temperatures. We can also observe that, in every case, the energy saved becomes less as the temperature of the air intake rises, because the COP of any cooling circuit is reduced as its condensing temperature increases, and this in turn reduces the potential savings from lowering the fan speed.

The energy savings produced by the energy balance function are based on reducing the speed of the condenser fan while increasing the compressor power somewhat less. Aside from the energy savings there is one extremely welcome side-effect, namely the reduced noise emissions from the condenser resulting directly from the reduced speed. Particularly on warm summer evenings at the weekends, the neighbours will be grateful for the reduced noise level, and the system operator will be saving money at the same time.

Summary

This article was able to show that regulating the speed of condenser fans for optimum energy consumption, taking into consideration the total power consumption of compressor and condenser fans, can produce markedly higher energy savings than conventional regulation using only P or PI controllers. The use of standard controllers, particularly for systems running under partial load with a combination of high air intake temperatures and low minimum condensing temperatures, leads to extremely energy inefficient operating situations. The energy saving results from the fact that a fan's power consumption increases with the cube of its speed, while its airflow volume increases approximately linearly. Energy-optimised regulation needs to know not only the condensing temperature, which has always been the case, but also the air intake temperature at the condenser, so that it can coordinate the condenser capacity with the performance of the refrigeration or air-conditioning unit, which as we know is determined by the refrigerant, the pressure level and the effectivity of the condenser.