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A novel method for estimation of the annual energy assessment of using an external subcooler in a refrigeration machine

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Abstract

The additional subcooling of the refrigerant is an interesting way to improve the efficiency of the refrigerant system. On behalf of the Swiss Federal Office of Energy, the seasonal saving potential of using an external subcooler in a refrigeration machine considering application limits and plant specific conditions was investigated based on computer simulations.

The result of the investigation is presented as a guideline. With this method, a user can determine the optimum subcooling for his application with the given environmental conditions. By considering the operating limits of the refrigeration machine and the environmental conditions, the method can be used directly for the planning and adjustment of a refrigeration machine.

The results show that a refrigeration machine for air conditioning with a direct evaporator and the refrigerant R290 (propane) can significantly improve efficiency by using an external subcooler. There is an annual energy saving potential of 7% if the external subcooler is used with cold wastewater.

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Keywords: Subcooling ; Seasonal saving potential ; Refrigerant cycle ; External subcooler ; Refrigerating machine

1. Introduction

1.1. Related research on the subject

In compression refrigeration systems, subcooling of the liquefied refrigerant is an important issue that can affect the efficiency of the entire cycle. There are many studies in the literature that deal with the subcooling of the refrigerant. This concerns, for example, the type of subcooling that can be realized externally or by means of an internal heat exchanger. Experimental investigations on subcooling with refrigerants R134a and R1234yf have been carried out by *Pottker and Hrnjak* [1], who investigated the operation with internal heat exchanger regarding the efficiency. It was shown that the efficiency of a refrigeration cycle system with R1234yf can be increased up to 18% by using an internal heat exchanger. With R134a, the efficiency could be increased by 9%.

The choice of refrigerant has a significant impact, as various other studies have shown. *Ansari et al.* [2] investigated the effects of subcooling on the operation of the refrigeration machine. They investigated the refrigerants R1243zf and R1233zd(E), which were not considered in our research. The results were compared with the performances of the refrigerant R134a and have shown that R1233zd(E) performs better with subcooling than R134a, while 1243zf is slightly less efficient both energetically and exergetically.

Furthermore, some studies have already been conducted on supercooling by ejector. *Yilmaz and Erdinc* [3] investigated a new ejector subcooling system, where part of the refrigerant is expanded after the condenser.

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The tests were carried out on seven different refrigerants. With a COP improvement of 20% and an exergy efficiency improvement of 18% the best performance was obtained by R1234yf.

1.2. Purpose of subcooling

Subcooling is necessary to avoid vapor bubbles (flashgas) in front of the expansion valve which is essential for stable and safe operation of the refrigerating machine.

Vapor bubbles before the expansion valve reduce the refrigeration capacity [4] and can lead to cavitation in the expansion valve, which can damage it. If the valve is operated in this condition for a longer period of time, there is a risk of wet operation, as the valve no longer closes completely due to the damage.

With electronic injection control, the amount of refrigerant injected into the evaporator is metered by means of an expansion valve so that the superheat at the evaporator outlet remains constant. The mass flow depends on the pressure difference across the expansion valve, the cross-sectional area, and the inlet density of the refrigerant. The expansion valve changes the cross-sectional area by its degree of opening and thus regulates the flow or the refrigerant charge level in the evaporator. If the refrigerant is not sufficiently subcooled, flashgas may form in the liquid line due to pressure or heat losses, which significantly interferes with the control of the injection rate. The influence of flashgas on the injected mass flow and superheat is shown schematically in Figure 1.

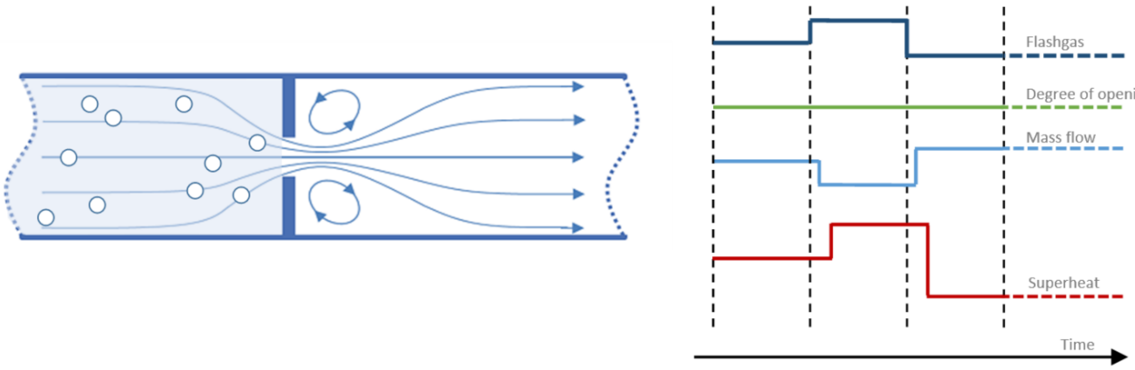

Fig. 1. Influence of flashgas on mass flow and superheat

The gas bubbles have a significantly lower density than the liquid refrigerant. If gas bubbles appear in the liquid refrigerant upstream of the expansion valve, the quantity of refrigerant injected is reduced without the opening position of the expansion valve having changed. This reduction in mass flow leads to an increase in superheat and the controller opens the expansion valve as a countermeasure.

If the proportion of flashgas at the inlet of the expansion valve is reduced, e.g., for a short time completely liquid, the mass flow increases due to the greater density, the superheat decreases, and the controller reduces the opening cross-section of the expansion valve as a counter-reaction. The system may start to oscillate and may no longer be stable [5].

1.3. Aim of the study

The investigations on additional subcooling were carried out for five different subcooling types, five different applications (each with different evaporating temperatures) and eight different refrigerants. All information and results on the investigations are given in the final report on refrigerant subcooling [6] (in German). This paper is limited to the investigation of subcooling with external subcooler for the air conditioning application with a direct evaporator (evaporating temperature of 10 °C) and the refrigerant R290.

The aim of the study is to present the result as a guideline. This allows a user to estimate the potential of additional subcooling in each case for his individual application and the given environmental conditions.

2. Methodology

2.1. Cycle with external subcooler

The investigations for this report were all carried out using computer simulations with MATLAB [7]. To show the influence of subcooling on the refrigeration circuit, a reference process with the same application and the same refrigerant, but without additional subcooling, is calculated in each case. Subsequently, the refrigeration circuit is supplemented by an external subcooler and calculated again. For a seasonal consideration, this procedure is repeated at different operating points. The comparison is shown schematically in Figure 2.

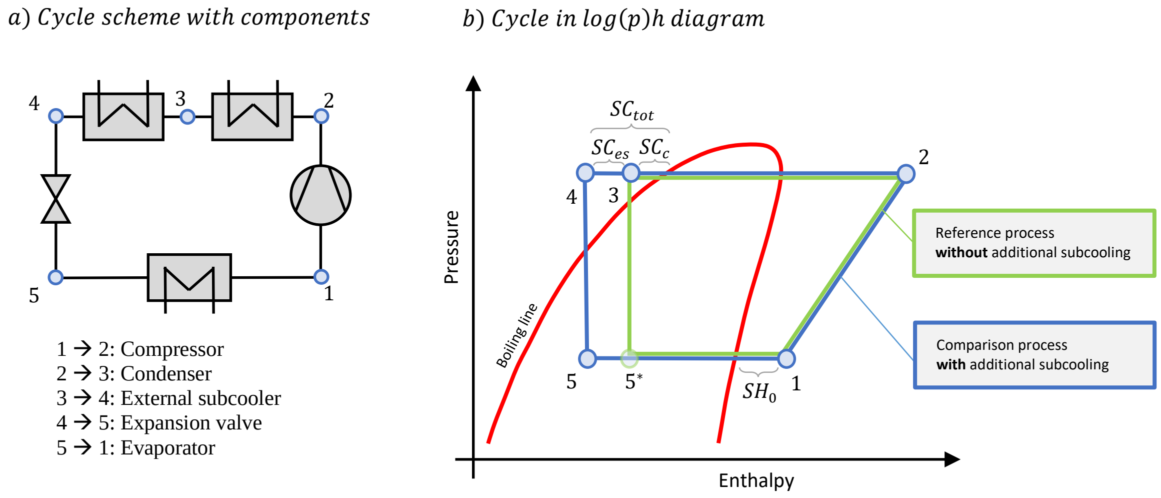

Fig. 2. a) The refrigeration cycle with its components. b) The reference process without additional subcooling (green) and the comparison process with additional subcooling (blue), drawn in log-p-h diagram.

In the case of an external subcooler, the refrigerant is subcooled after the condenser (3) with an additional heat exchanger (4). The total subcooling (SC_{tot}) therefore results from the subcooling by the condenser (SC_c) and the additional subcooling by the external subcooler (SC_{es}). How much the refrigerant can be subcooled depends on the temperature and capacity of the heat sink of the external subcooler. The external subcooling increases the enthalpy difference at the evaporator ($h_1 - h_5$) and thus the refrigeration capacity. The operating point of the compressor and its power consumption remain unchanged. The efficiency of the chiller therefore increases with increasing subcooling. For a correct assessment of the efficiency, however, the required auxiliary energies must also be taken into account. The estimation of the auxiliary energies requires a consideration of the entire system. If the efficiency of another system is reduced due to the subcooling or if additional energy is used to ensure the heat sink (e.g. pump capacities), this must be taken into account. Only if the energy saving at the refrigeration machine is more than the required auxiliary energies, the efficiency of the overall system increases.

2.2. Operating points

The operating point of a refrigeration machine is defined by evaporating and condensing temperature. These are defined as follows:

The **evaporating temperature** T_0 depends on the application and is constant for all operating points. For air conditioning with direct evaporator, the evaporating temperature is 10 °C. The influence of the vapor quality on heat transfer and distribution in the evaporator are not considered.

The **condensing temperature** T_C depends on the ambient temperature and is therefore variable. To consider the operation of the refrigeration machine at different ambient temperatures, the simulations are carried out with five different condensing temperatures at intervals of 5 °C each. These can be seen in Table 1.

Table 1. The five selected operating points in terms of their evaporating temperature T_0 and condensing temperature T_C

Operating point	Evaporating temperature T_0 (°C)	Condensing temperature T_C (°C)
1	+ 10	+ 45
2	+ 10	+ 40
3	+ 10	+ 35
4	+ 10	+ 30
5	+ 10	+ 25

2.3. Parameters of the cycle simulations

Apart from total subcooling, the input parameters for the cycle simulations are identical in both cycles. Table 2 shows the parameters used for the simulations of the reference process and the process with external subcooler. While in the reference process the total subcooling is the same as the subcooling in the condenser, in the external subcooler the total subcooling is varied from 5 K to 25 K (in 5 K steps) at each operating point. How much the refrigerant can be subcooled depends strongly on the available heat sink. This can very differently, depending on the object. Therefore, different degrees of subcooling are investigated in this study.

Table 2. Parameters of the cycle simulations for the circuits investigated

Cycle	Operating points (T_0 / T_C)	1	2	3	4	5
Both (reference and external subcooler)	Subcooling condenser [K]	2	2	2	2	2
	Superheating evaporator [K]	7	7	7	7	7
	Isentropic compressor efficiency [-]	0.7	0.7	0.7	0.7	0.7
Reference	Subcooling total [K]	2	2	2	2	2
External subcooler	Subcooling total [K]	5 - 25	5 - 25	5 - 25	5 - 25	5 - 25

2.4. Computer simulations

The investigations for this report were all carried out using computer simulations with MATLAB [7]. The fluid properties of the refrigerants were calculated using the software REFPROP [8]. The specific enthalpies are calculated for all four or five points of the cycle (see Fig. 2a) and 2b)). The specific enthalpy depends on the pressure and temperature of the fluid and can be calculated for each point. In the 2-phase area, this also allows the vapor quality to be calculated. Accordingly, the most important input parameters are the evaporation temperature T_0 and the condensing temperature T_C, which are listed in Table 1. With the extended cycle parameters of Table 2, the specific enthalpy is calculated for each point in the cycle. The process at the expansion valve (Fig. 2a): 4 → 5) is assumed to be isenthalp, so that $h_5 = h_4$ in each case (compare with Fig. 2b)).

The cooling capacity is calculated according to equation 1:

$$\dot{Q}_0 = \dot{m} \cdot (h_1 - h_5) \tag{1}$$

Since the mass flow $\dot{m}$ is the same everywhere in the refrigeration cycle with external subcooler as well as in the reference refrigeration cycle, the EER is calculated independently of the mass flow $\dot{m}$ according to equation 2:

$$EER = \frac{h_1 - h_5}{h_2 - h_1} \tag{2}$$

The calculation of the EER is performed for both the refrigeration circuit with external subcooling and the reference refrigeration circuit without additional subcooling. The savings potential corresponds to the change in efficiency and is calculated according to equation 3. For all 5 operating points this procedure is repeated.

$$\Delta EER = \frac{EER_{External\ SC}}{EER_{Ref.Process}} \tag{3}$$

3. Limits of subcooling

3.1. Minimum subcooling

The minimum subcooling is essential for stable and safe operation of the plant. Whether this leads to an increase in efficiency is secondary. Subcooling can be reduced by pressure losses and heat inputs between the receiver and the inlet to the expansion valve. The pressure losses are composed of the static height difference between the header and the expansion valve and the pressure losses of the pipelines and fittings. The influence of pressure loss and heat exchange on subcooling (SC) is shown schematically in Figure 3.

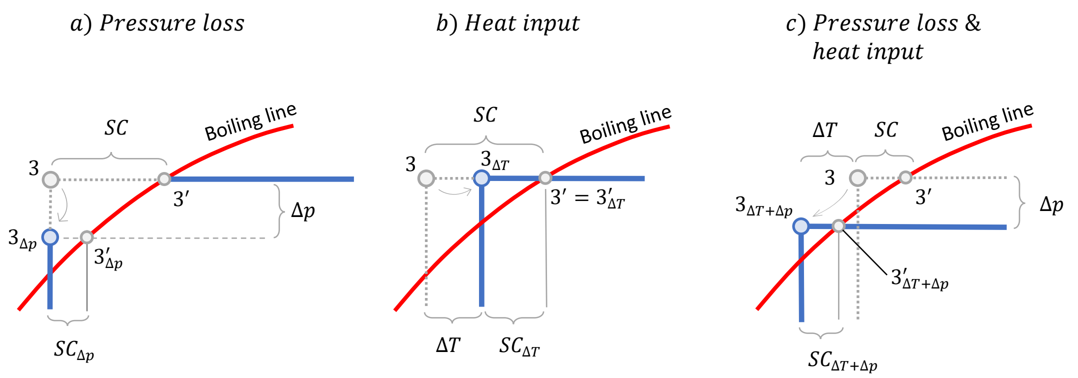

Fig. 3. Reduction of subcooling for a) due to pressure losses, for b) due to heat input, and for c) due to pressure losses and heat input

In Figure 3 the upper left section of the log p-h diagram is shown enlarged. In Figure 3a) the influence of the pressure loss, which is caused by pipelines, fittings, or lines with large height difference, is shown ($3 \rightarrow 3_{\Delta p}$). In case of pressure losses, the refrigerant temperature remains constant ($T_3 = T_{3\Delta p}$). However, the subcooling reduces because the boiling temperature reduces due to the pressure loss ($T_{3'} > T_{3'\Delta p}$). In the plot of Figure 3b), the influence of heat exchange between the pipe and the environment is shown ($3 \rightarrow 3_{\Delta T}$). Heat exchange takes place through the piping and fittings. Depending on fluid and ambient temperature, heat input or heat output results for the refrigerant. During heat exchange, the pressure and thus the boiling temperature remain constant ($T_{3'} = T_{3'\Delta T}$). The subcooling changes as the refrigerant heats up or cools down further due to the heat exchange. In Figure 3c), the combination of pressure losses and heat exchange is shown. These two effects can compensate each other. Due to the pressure loss, the boiling temperature of the refrigerant is reduced. However, if the ambient temperature is lower than the fluid temperature, the refrigerant is additionally cooled via the tube wall.

The minimum subcooling is therefore dependent on pressure losses and heat inputs and is therefore very plant specific.

3.2. Maximum subcooling

The maximum subcooling is primarily about optimizing efficiency and therefore varies depending on the refrigeration machine. In particular, the influence of large subcooling on the piping components and the vapor quality must be considered.

The subcooling beyond the minimum subcooling is only useful if it can increase the efficiency of the system. This may vary depending on the subcooling method, operating control, and refrigerant. How much the refrigerant can be subcooled is limited by the vapor quality at the outlet of the expansion valve. The influence of subcooling on the vapor quality can be seen in Figure 4.

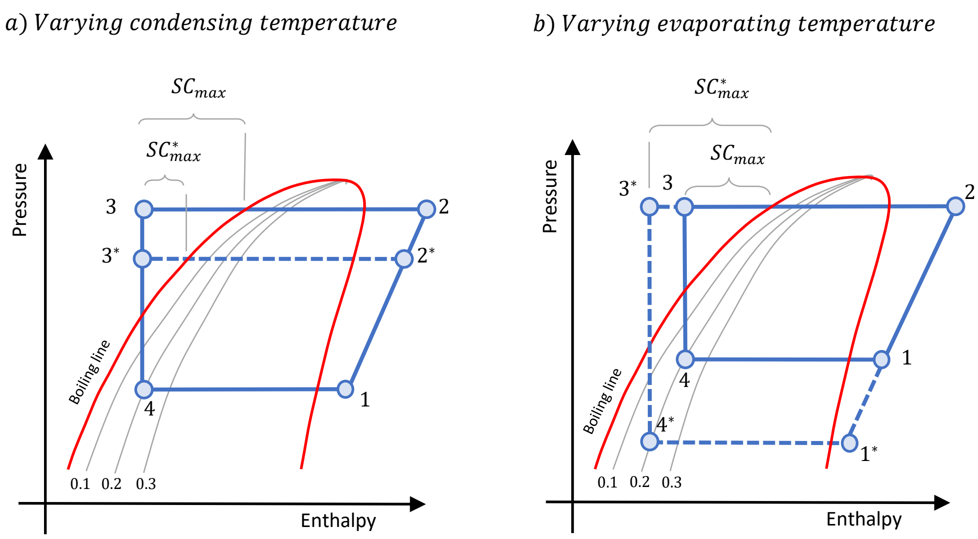

Fig. 4. Influence of subcooling on the vapor quality for a) varying condensing temperature and for b) varying evaporating temperature.

Increasing the subcooling reduces the vapor quality at the outlet of the expansion valve. Thus, the minimum vapor quality there defines a limit to how much the refrigerant can be subcooled. Figure 4a) shows the influence of the condensing temperature (operating point) on the maximum possible subcooling. A reduction of the condensing temperature also reduces the maximum possible subcooling. Figure 4b) shows that a reduction in the evaporating temperature (application) increases the maximum possible subcooling.

4. Results

4.1. Overview

The results of this investigation show the influence of different amounts of subcooling on the efficiency of the system. At the same time, different auxiliary variables are displayed, which are important for the estimation of the minimum necessary and maximum possible subcooling.

The combination of these factors allows a plant specific and seasonal assessment of efficiency improvement. Identical plots have been created for different subcooling types, refrigerants and applications. Therefore, auxiliary variables are also shown, which are not of interest for the assessment in the case of external subcooling. The plots can be seen in the appendix of the final report of the project [6].

Figure 5 shows the influence of additional subcooling on the efficiency as well as important auxiliary variables for a system with an external subcooler and refrigerant R290, for an application with T_0 = 10 °C (air conditioning with a direct evaporator).

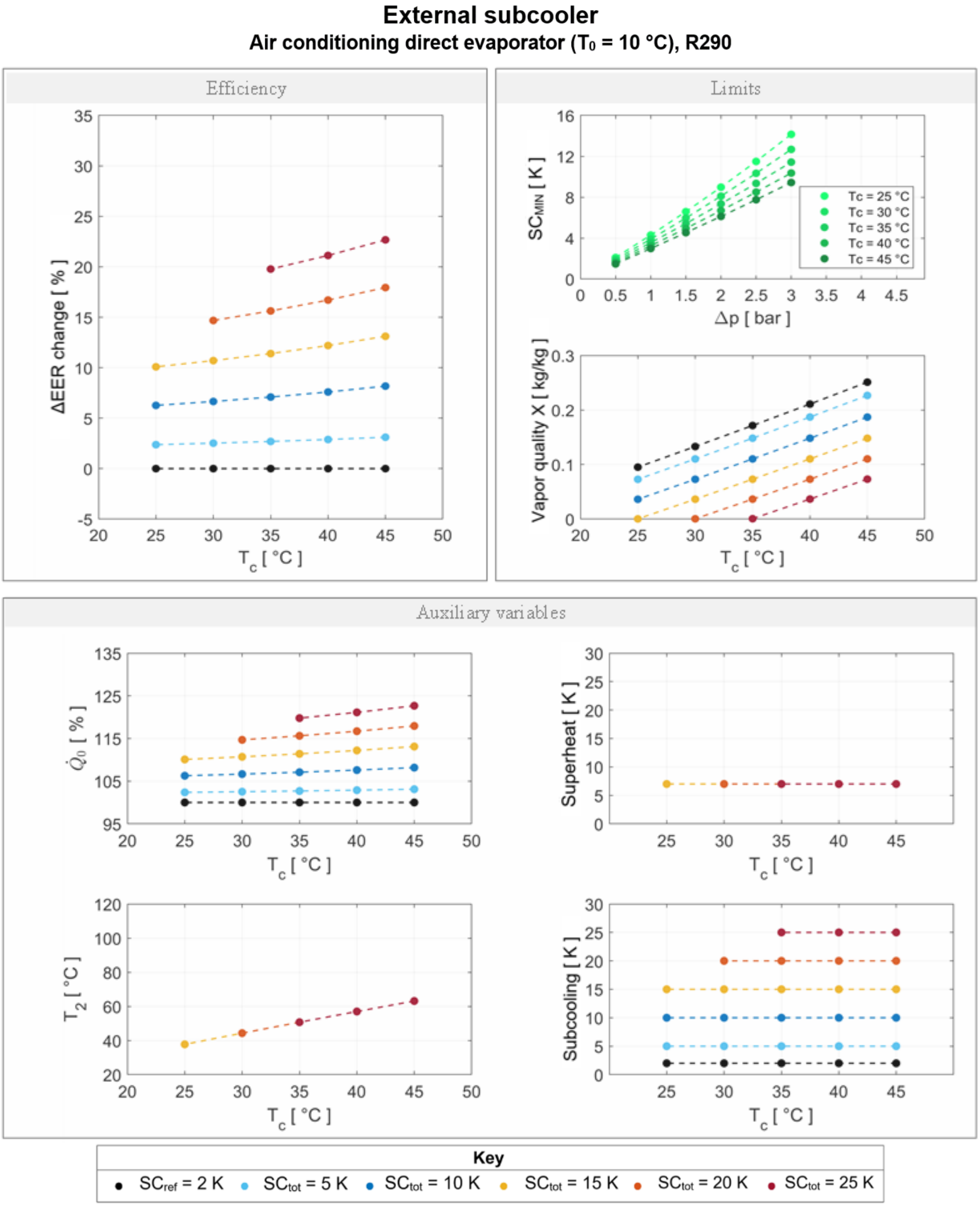

Fig. 5. Influence of additional subcooling regarding efficiency, limits, and auxiliary variables for an example of a refrigeration machine

Figure 5 shows the result as an overview. The top left box shows the potential for EER improvement as a function of condensing temperature for subcooling values of 5 - 25 K. However, in order to determine the efficiency improvement, the limits of subcooling (top right box in Fig. 5) and the auxiliary variables (bottom box in Fig. 5) must also be considered.

The diagrams in Figure 5 allow a comprehensive assessment of the additional subcooling as well as the expected efficiency gain at different operating points. The use of the result, as shown in Figure 5, is described in the following subsection.

4.2. Interpretation and use of the result

To be able to use the result, the heat sink of the external subcooler must be known. As an example, a use of a cold wastewater, which allows cooling of the refrigerant to constant 15 °C, is investigated. The maximum possible subcooling, given by the heat sink, is thus dependent on the operating point and is between 10 - 30 °C. To estimate which subcooling or which efficiency increase can be achieved with this heat sink, the following three limits must be considered and drawn in the upper right diagram in Figure 5 (Limits) and are shown larger for explanation in Figure 6:

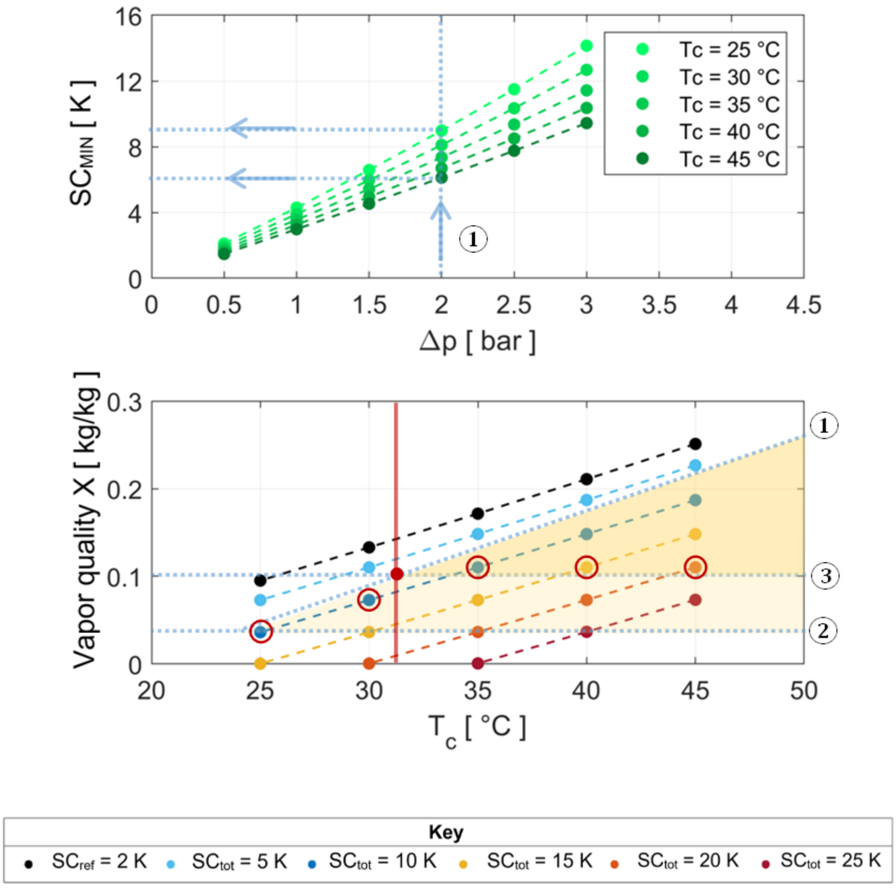

Fig. 6. Description to capture the limits of subcooling

① Minimum subcooling

As described in chapter 4.1, minimal subcooling is necessary for safe and stable operation of the system. Based on the upper diagram, the minimum subcooling can be estimated due to the plant-specific pressure losses. In this example, a plant with large differences in height and therefore 2 bar pressure losses is examined. Due to the pressure losses a minimum subcooling of 6 - 9 K is necessary. This value can also be plotted in the lower diagram and shows the lower limit of subcooling.

② Maximum possible subcooling (Heat sink)

The maximum possible subcooling depends on the heat sink of the external subcooler and its seasonal behavior. In this case, the heat sink is cold wastewater, which allows subcooling to a constant temperature of 15 °C. As can be seen in Figure 4, a steady outlet temperature from the subcooler means different degrees of subcooling, depending on the condensing temperature. This is shown in the Figure 6 by line ②.

③ Maximum subcooling (vapor quality)

The maximum possible subcooling is limited by the heat sink. An additional limitation is the minimum vapor quality at the outlet of the expansion valve. In this case, a minimum vapor quality of 0.1 is examined. This value can be defined by the designer's experience or by the expansion valve's operating limits. This is shown in the Figure 6 by line ③.

Based on the three plant specific limits, a range of possible subcooling results. Between the minimum necessary ① and the maximum possible subcooling ②, which are determined from the pressure losses and the heat sink, there is a range of subcooling which is theoretically possible (yellow area). Considering the minimum vapor quality ③, this area can be further restricted.

At high condensing temperatures (T_C > 31 °C), both the minimum subcooling and the minimum vapor quality can be maintained. All operating points between the two lines are theoretically possible. For efficiency reasons, the optimum subcooling is as close as possible to the minimum vapor quality, provided those other auxiliary variables, such as the end of compression temperature, are not in a critical range.

At low condensing temperatures (T_C < 31 °C), it is not possible to ensure the minimum subcooling without falling below the minimum vapor quality. The designer of the machine should consider operating points in this area in more detail.

Before determining the increase in efficiency of the individual operating points, the influence on other important parameters in the refrigeration cycle can be checked using the additional diagrams. Especially in applications with large pressure differentials, it is possible that the end of compression temperature T_2 comes into a critical range. To verify that the selected subcooling is feasible, the end of compression temperature T_2 must be within the compressor's operating limits. The compressor operating limits are defined by the manufacturer. Figure 7 shows the diagram for estimating the end of compression temperature, which is at the auxiliary variables in Figure 5.

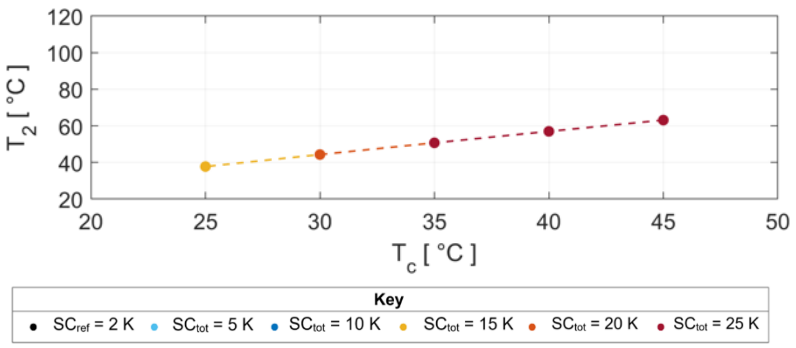

Fig. 7. Plot for estimation of the end of compression temperature (T_2)

The potential efficiency increases from the additional subcooling can be determined in Figure 8.
It shows the savings potential ΔEER of the external subcooler at different operating points. In the case with a wastewater heat sink of 15 °C ΔEER is between 6 – 18 % depending on the condensing temperature T_C. For condensing temperatures of 25 - 35 °C, a total subcooling of 10 K is possible. This was determined using the limits of subcooling (see Fig. 6). The total subcooling for a condensation temperature of 40 °C is 15 K (yellow line), at T_C = 45 °C is SC_{tot} = 20 K (red line).

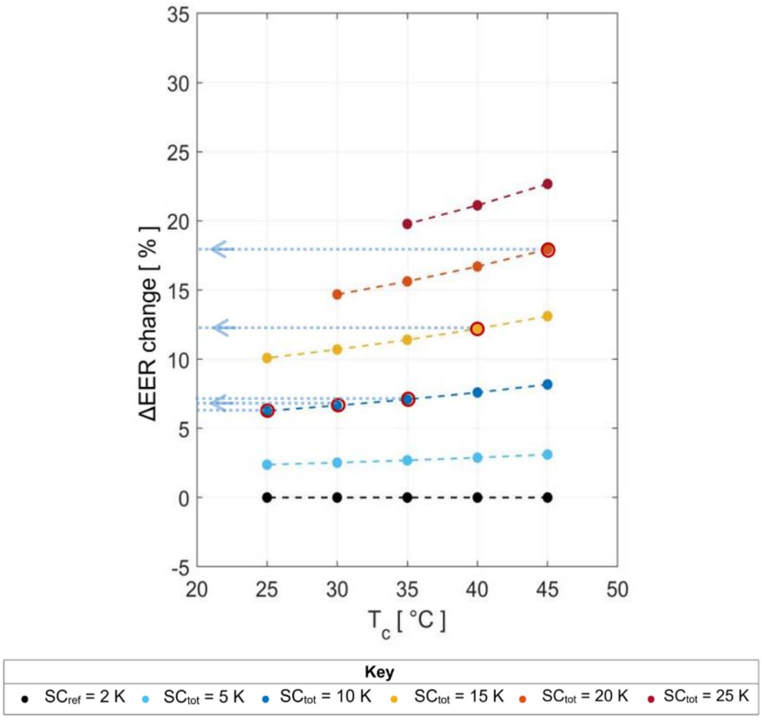

Fig. 8. Change in EER due to subcooling for different operating points

The efficiency improvement values can be used as the basis for an economic efficiency calculation. It should be noted that the operating points vary in frequency depending on the location. The individual operating points must therefore be weighted differently to estimate an annual energy saving, which is described in chapter 5.

5. Application for profitability calculation

The evaluation of the economic efficiency depends on the investment costs, the savings potential of the subcooling type, and the required auxiliary energies. The diagrams created in this report primarily help to estimate the annual savings potential of the subcooling type (ΔEER_a), which is determined according to equation 4.

$$\Delta EER_a = \sum_{i=1}^{5} w_i \cdot \Delta EER_i \tag{4}$$

In the previous chapter, the savings potential (ΔEER) of external subcooling by means of cold wastewater at different condensing temperatures was estimated. Figure 8 shows that the EER increases between 6 - 18 % depending on the condensing temperature.

For a seasonal consideration, the individual operating points must be weighted according to their energetic share of the annual consumption (w_i). The weighting of the individual operating points depends on the location and planned use (load profile). The final report on the investigation of refrigerant subcooling [6] shows in the appendix weighting factors for various locations in Switzerland and load profiles. The methodology for the energetic weighting of several operating points is described in detail in the final report on refrigeration compressors [9] (in German). Table 3 shows the seasonal consideration for the Basel site as an example. The load profile of the application must also be considered.

Table 3. Example of annual savings potential based on seasonal consideration

Operating point	1	2	3	4	5
Condensing temperature [°C]	45	40	35	30	25
Weighting factor [w_i]	0.01	0.05	0.10	0.14	0.70
Savings potential [ΔEER]	18 %	12 %	8 %	7 %	6 %
Annual savings potential [ΔEERa]			**7 %**		

The results in Table 3 show the energy weighting of the individual operating points for the Basel site. The annual savings potential is 7 %. This is due to the hourly frequency of ambient temperatures at the investigated site. It can be seen that the operating points with the highest savings potential are significantly less relevant in terms of energy. It should be noted that this only applies to the refrigeration machine and that any possible changes in the consumption of auxiliary energies have not been considered.

6. Conclusion

The studies on seasonal energy savings potential with additional subcooling have provided a useful basis for proper subcooling. The plots in Figure 5 can be used to determine the limits and energy savings potential of a refrigeration cycle with external subcooling.

It has been shown that there is a significant savings potential for refrigeration machines with direct evaporator and R290 as refrigerant if they can be operated with an external subcooler with constant heat sink.

Based on the seasonal evaluation, an economic efficiency calculation can be performed in addition to the annual savings potential. The study shows that the savings potential is strongly dependent on the operating point due to the limitations. For a seasonal evaluation, it is therefore always necessary to consider several operating points. However, the results are not only important for estimating the energy savings potential, but also for setting the operating limits. Compliance with the limits is important for safe and reliable operation of the refrigeration machine.

The study was carried out only based on computer simulations, which were verified by experts. There is no experimental comparison of the results. Accordingly, it would be interesting if the investigations could be compared with measurements on real refrigeration machines.

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Nomenclature and Abbreviations

T_0	Evaporating temperature	COP	Coefficient of Performance
T_C	Condensing temperature	SC	Subcooling
ΔT	Temperature difference	EER	Energy Efficiency Ratio
Δp	Pressure loss	ΔEER	Savings potential
SC_{ref}	Subcooling for a reference process		
T_2	End of compression temperature		
h_i	Specific enthalpy		
$\dot{m}$	Refrigerant mass flow		
$\dot{Q}_0$	Cooling capacity		
ΔEER_i	Savings potential at an operating point to calculate the annual savings potential		

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