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Energy Efficiency of Refrigeration Systems

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ABSTRACT

Energy efficiency plays an important role in the development and operation of refrigeration systems. The method of the VDMA 24247-2 2 “Energy efficiency of refrigeration systems – Requirements for the system design and the components” were recently published. The method will be described within this paper, with the focus on the graphical interpretation.

1. INTRODUCTION

Cooling technology is of great economic and energetic importance. In 2009 about 14 % of electrical energy was used for refrigeration in Germany, (Preuss, 2011). IIR estimated that refrigeration and air conditioning use approximately 15 % of the world’s production of electricity, (IIR 2002). An energy efficient operation of vapor compression cycles helps to save electrical energy. Today this is associated with the saving of fossil fuels and reduction of carbon dioxide emissions. The evaluation of the energy efficiency requires an appropriate method. The exergetic analysis of refrigeration systems has long been known. The exergetic evaluation provides all the capabilities of individual components as well as complete refrigeration systems in detail. The method is a good way to optimize energy efficiency. However the exergetic analysis is, considered by many to be difficult and probably not often used by planners and operators.

In the VDMA standard sheet 24247 Part 2 “Energy efficiency of refrigeration systems – Requirements for the system design and the components” (VDMA, 2011) five key figures are presented, which promise more simplicity so more acceptance among the target group can be expected. These dimensionless figures allow a differentiated assessment of specific losses of a refrigeration system and its components in their interaction. Within this article the method and their figures are described clearly and finally compared with the exergetic efficiency (second law efficiency) of a refrigeration system.

2. REFRIGERATION

Naturally, heat flows from a system with a high temperature into a system of low temperature, if the systems are coupled by a heat-permeable boundary. The heat is transported by entropy. In addition, in a cooled system heat (and entropy) can be generated by dissipation of work. Cooling means to reduce the entropy of a system, while reducing the temperature to a level below the ambient temperature. Heat needs to be removed from a system, and pumped to a higher temperature level. To do this, a certain amount of energy is required. In figure 1 the scheme of a simple refrigeration system with its main components is shown. Arrows from the left and right, labeled with, indicate energy input respective to power input. Arrows from the bottom, labeled with $\dot{Q}$ show heat flowing into the cooled system or released to the heat sink. The refrigeration system is driven by the compressor with the power $P_{\text{oe-el}}$. The compressor works between the pressures corresponding to the temperatures T_{o} and T_{c}. Additional power is needed for the fluid transport (index: FT), on the cold side $P_{\text{FT-K}}$ and the warm side $P_{\text{FT-W}}$ and for defrost heating on the cold side $P_{\text{H-W}}$ (index: H). Incoming heat flows are from the heat source $\dot{Q}_{\text{oN}}$, dissipated electrical power for driving the fan $\dot{Q}_{\text{FT-K}}$ and for defrost $\dot{Q}_{\text{H-K}}$. Additional heat flow enters the suction line $\dot{Q}_{\text{iso}}$.

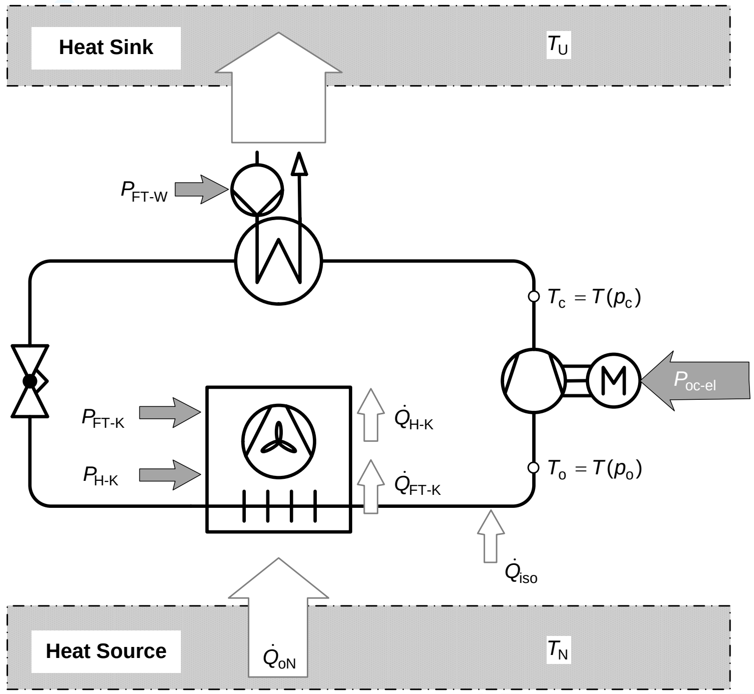

Figure 1: Scheme of a simple refrigeration system with energy flow

3. TERMS AND CONCEPT

The standardized terminology in this subject area is internationally diverse and heterogeneous, e. g. the definition of *EER* (Energy Efficiency Ratio) is different in U. S. and European standards. The VDMA standard (VDMA Einheitsblatt) 24247-2 is originally written in German language. So the meaning of the symbols and indices do not reveal itself in English language without further notice. To avoid confusion and misunderstandings the key terms are defined below. For energy assessment the term "energy efficiency" is often used. So far this term has not clearly been defined and needs to be explained. In general, efficiency describes the extent to which effort is used for an intended purpose. A high efficiency means: reach the goal with the least possible use of resources. For the purposes of the above mentioned definition, the energy efficiency of a system can be described by the ratio of refrigerating capacity and the input power. This ratio is known as the *COP* , the Coefficient of Performance

$$COP = \frac{\text{usefull refrigerating capacity}}{\text{total power input}} \frac{\mathrm{W}}{\mathrm{W}} \tag{1}$$

The *COP* is defined for exactly one operating condition of the refrigeration system under steady state conditions. Referring to figure 1, the

$$COP = \frac{\dot{Q}_{\mathrm{oN}}}{P} \frac{\mathrm{W}}{\mathrm{W}} \tag{2}$$

describes the efficiency for the given conditions. For the determination of an appropriate "energy efficiency" it is necessary to know the timing of the useful refrigerating energy and the energy input. The units J/J or kWh/kWh can be distinguished. This kind of energy efficiency should have a different name. It is a ratio like the *COP* , but the ratio is built with energies, so the term coefficient of energy, *COE* is defined here

$$COE = \frac{\text{usefull refrigerating energy}}{\text{total energy input}} \frac{\mathrm{J}}{\mathrm{J}} \tag{3}$$

In this context the heat removed from a cooled system is referred to as "generated refrigerating energy" Q_{oN}. It results

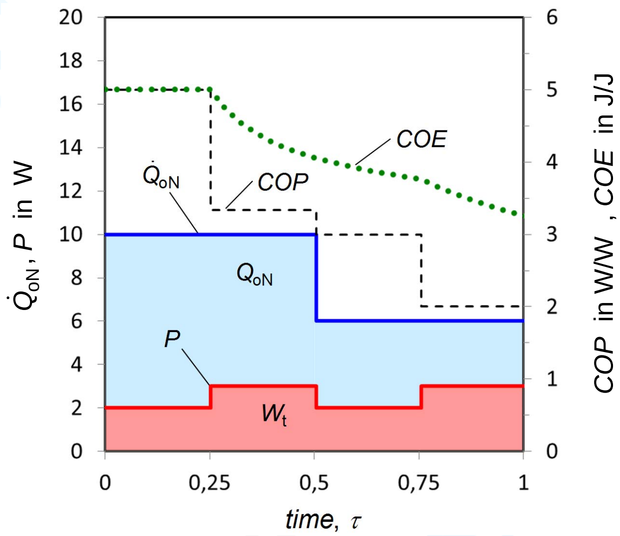

Figure 2: *COP* and *COE* of a cycle for different refrigerating capacities and driving powers, or different heat source and heat sink temperatures respectively.

from the integration of the refrigerating capacity over the time period $\Delta\tau = \tau_2 - \tau_1$. The input energy is technical work W_t to drive the whole system

$$COE_{\Delta\tau} = \frac{Q_{oN}}{W_t} = \frac{\int_{\tau_1}^{\tau_2} \dot{Q}_{oN}\,d\tau}{\int_{\tau_1}^{\tau_2} P\,d\tau}\,\frac{J}{J} \tag{4}$$

The time interval $\Delta\tau$ applied to the relationship, has to be specified (e. g. day, season, year). For constant operation conditions or for a very short time period *COP* and $COE_{\Delta\tau}$ are equal. An example is given in figure 2. For different heat source and heat sink temperatures, the coefficient of performance (*COP*) and the $COE_{\Delta\tau}$ are shown. It becomes clear that the $COE_{\Delta\tau}$ cannot be calculated from the *COP* and vice versa. It has be mentioned that this $COE_{\Delta\tau}$ is different to the Seasonal Energy Efficiency Ratio or *SEER* (AHRI, 2011) and the European Seasonal Energy Efficiency Ratio or *ESEER* (Eurovent, 2006), because these numbers are calculated directly from *EER* and *COP*. *SEER* and *ESEER* are defined for comparison of similar systems only. With *COP* and *EER* it is not possible to estimate their theoretical limit. The efficiency of a system can be described more precisely by setting the above "energy efficiency" of a real process in relation to a certain (ideal) reference process. Referring to the above mentioned considerations it can be defined

$$\eta_{COP} = \frac{\text{Power Efficiency of a real Process}}{\text{Power Efficiency of a Reference Process}} = \frac{COP_{\text{real}}}{COP_{\text{Reference Process}}}. \tag{5}$$

$$\eta_{COE} = \frac{\text{Energy Efficiency of a real Process}}{\text{Energy Efficiency of a Reference Process}} = \frac{COE_{\text{real}}}{COE_{\text{Reference Process}}}. \tag{6}$$

The reference process is arbitrary. To ensure the comparability of different refrigeration processes, it is reasonable to choose the CARNOT cycle as a reference. In the VDMA Einheitsblatt 24247-2 the *COP* of the refrigeration system is compared with the *COP* of a CARNOT cycle between the temperatures of the heat source T_N and heat sink T_U. The ratio referred to is an "energy efficiency level" or "total energy efficiency". It is identical to the "exergetic efficiency" of the refrigeration system, also called the "second law efficiency". Therefore the energy efficiency calculated by the VDMA-Einheitsblatt is valid for one operation condition.

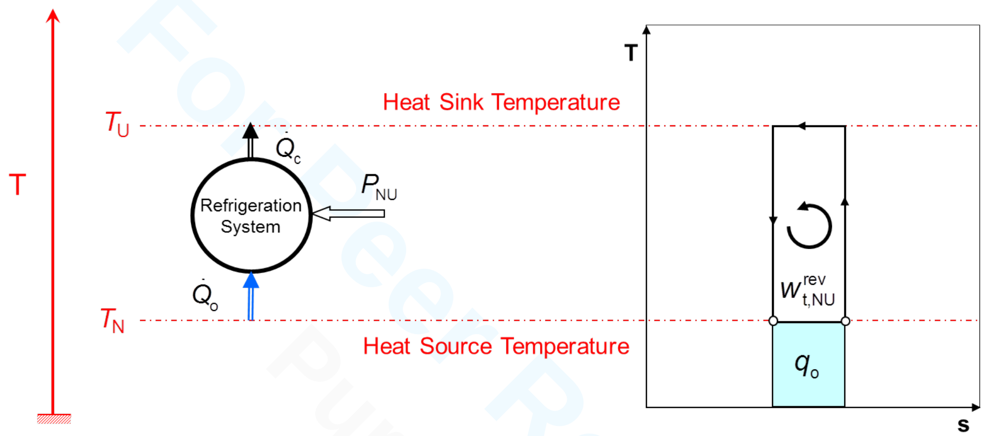

Figure 3: Specific work $w^{rev}_{t,NU}$ and specific heat q_o of the CARNOT cycle between the temperature levels T_N and T_U in a temperature-entropy-diagram (schematically)

4. EVALUATION OF ENERGY EFFICIENCY

In this chapter the characteristic numbers of VDMA 24247-2 will be presented, to describe the performance and energy efficiency of a refrigeration system. The assessment is based on the formation of relationships, or quality levels, which also allow the evaluation of partial system functions. To illustrate the principles, the CARNOT cycle is considered first. Subsequently, the observations are expanded to a real vapor compression cycle.

It can be shown that the task of cooling can ideally be done with a CARNOT cycle, index: C. The *COP* of such a cycle between the temperature levels T_N and T_U (index: NU) is defined as

$$COP_{NUC} := \frac{\dot{Q}_o}{P^{rev}_{NU}} = \frac{q_o}{w^{rev}_{t,NU}} = \frac{T_N}{T_U - T_N}. \tag{7}$$

The refrigerating capacity $\dot{Q}_o$ and the required power P are calculated from the mass flow of the refrigerant $\dot{m}_R$

$$\dot{Q}_o = \dot{m}_R \cdot q_o \qquad , \qquad P_{NU} = \dot{m}_R \cdot w^{rev}_{t,NU}. \tag{8}$$

The specific work for the reversible process $w^{rev}_{t,NU}$ and the specific refrigerating heat q_o of the CARNOT cycle are shown in figure 3 as areas in a temperature-entropy-diagram. The coefficient COP_{NUC} depends only on the temperatures of the heat source T_N and heat sink T_U. For a constant temperature lift between the heat source and the heat sink $T_U - T_U$ the COP_{NUC} increases with increasing heat source temperature. For a constant heat source temperature T_N the COP_{NUC} increases with decreasing temperature lift $(T_U - T_N)$.

4.1 Efficiency of Heat transfer

In a real system, a temperature difference in the heat exchangers is necessary for the transportation of heat. Therefore the temperature on the cold side of the refrigeration cycle T_o must be lower than the heat source temperature T_N, while the temperature on the warm side of the cycle T_c has to exceed the temperature of the heat sink, the temperature of the environment T_U.

A CARNOT cycle between the temperatures T_o and T_c requires more driving power input than a CARNOT cycle between the temperatures T_N and T_U, if it is assumed that both processes produce the same refrigerating capacity.

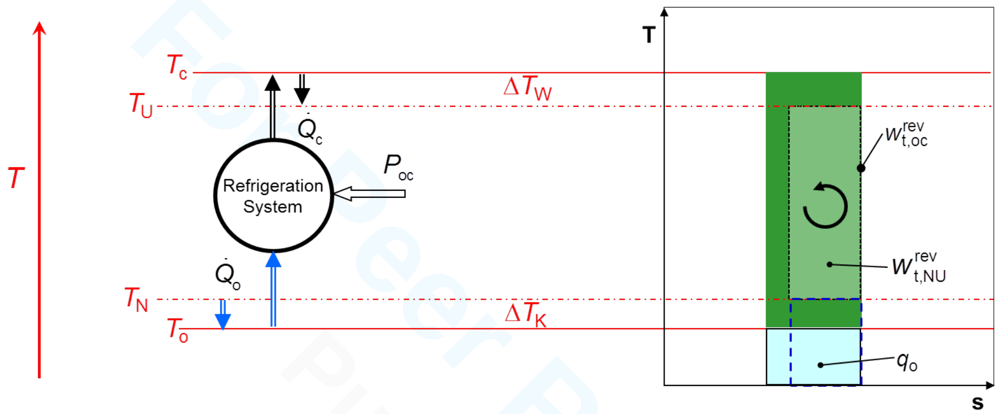

Figure 4: Comparison of CARNOT cycles between T_N and T_U respectively T_o and T_c.

A comparison of the refrigerating capacity of both processes performing the same refrigerating capacity results in

$$\eta_{WT} := \frac{COP_{ocC}}{COP_{NUC}} = \frac{w^{rev}_{t,NU}}{w^{rev}_{t,oc}} = \frac{T_o}{T_N} \cdot \frac{T_U - T_N}{T_c - T_o}, \tag{9}$$

with the coefficient of performance for the CARNOT cycle between T_N and T_U

$$COP_{NUC} = \frac{\dot{Q}_o}{P^{rev}_{NU}} = \frac{q_o}{w^{rev}_{NU}} = \frac{T_N}{T_U - T_N} \tag{10}$$

and the coefficient of performance for the CARNOT cycle between T_o and T_c

$$COP_{ocC} = \frac{\dot{Q}_o}{P^{rev}_{oc}} = \frac{q_o}{w^{rev}_{t,oc}} = \frac{T_o}{T_c - T_o} \tag{11}$$

The superscript "rev" refers to the fact that the CARNOT cycle is a reversible process. This ratio η_{WT} is called the efficiency of heat transfer (German: WT: Wärmetransport). A graphical interpretation of the specific work is possible in a $T,s-$diagram. (Please note: The following diagrams are schematically only, not to scale.) Due to the lower temperature T_o and the higher lift to T_c more specific work is necessary as shown in figure 4.

Small driving temperature differences in the heat exchangers (ΔT_K, ΔT_W) improve the efficiency of heat transfer η_{WT}. However, please note that small temperature differences are not possible for all applications, for example: drying of moist air and shock chilling or freezing. The transferred heat flow $\dot{Q}$ is calculated as

$$\dot{Q} = UA \cdot \Delta T_{log,m} \tag{12}$$

with U as the heat transfer coefficient and A as heat transfer area.

Only from the efficiency of heat transfer η_{WT} it is not clear whether a small temperature difference is achieved by a large heat-transmitting surface A or large heat transfer coefficients U. For example, a powerful fan can compensate a (too) small area by an increase of the air side heat transfer coefficient and thus improve heat transfer. For the fan, however, energy input is applied.

4.2 Efficiency of Fluid Transport

The total electrical power input of the refrigeration system is composed of the electrical power for the compressor (including oil sump heater, fan etc.) and for the units that enable the transport of the fluids outside the refrigerant circuit (pumps, fans). Furthermore, the electrical power is applied for an electric defrost heater if required. The total power input is calculated to

$$P_{\text{ges}} = P_{\text{oc-el}} + P_{\text{FT-K}} + P_{\text{FT-W}} + P_{\text{H-K}}. \tag{13}$$

These quantities are related to the mass flow of the circulating refrigerant. For a simple system with only one refrigerant circuit we provide the overall specific technical work

$$w_{\text{t,ges}} = w_{\text{t,oc}} + w_{\text{FT-K}} + w_{\text{FT-W}} + w_{\text{H-K}} \tag{14}$$

which can be represented as areas in the $T,s-$diagram. In figure 5 the areas belonging to the cold side are arranged below the specific technical energy for the cycle, while the work of the fluid transport on the warm side are above. The temperatures associated to the new areas have no meaning. It becomes clear that all the technical work has increased. The efficiency of fluid transport η_{FT} is defined as the ratio of the electrical power for the operation of the compressor to the total electrical power supplied to the system

$$\eta_{\text{FT}} := \frac{P_{\text{oc-el}}}{P_{\text{ges}}}. \tag{15}$$

For a simple system with only one refrigerant circuit it is

$$\eta_{\text{FT}} = \frac{w_{\text{t,oc}}}{w_{\text{t,oc}} + w_{\text{FT-K}} + w_{\text{FT-W}} + w_{\text{H-K}}}. \tag{16}$$

For a CARNOT cycle $w_{\text{t,oc}} = w_{\text{t,oc}}^{\text{rev}}$.

4.3 Efficiency of Cold Utilisation

Electrical energy supplied to the heat source, e. g. to drive the fans or for an electric defrost heater is fully converted to heat. The refrigeration system must remove this heat to hold the temperature at T_{N}. Thus, the useful refrigerating capacity is smaller than the "generated" refrigerating capacity by the compressor. An "efficiency of cold usage" is defined as the ratio of net refrigerating capacity and the refrigerating capacity

$$\eta_{\text{Qo}} := \frac{\dot{Q}_{\text{oN}}}{\dot{Q}_{\text{o}}} \tag{17}$$

For a simple system with only one refrigerant circuit it is

$$\eta_{\text{Qo}} = \frac{q_{\text{oN}}}{q_{\text{o}}} = \frac{q_{\text{o}} - q_{\text{FT-K}} - q_{\text{H-K}}}{q_{\text{o}}}. \tag{18}$$

In figure 6 it is shown that the area q_{o} is reduced to the net specific cooling capacity q_{oN}, by the dissipated power for fans and heater. If superheat in the suction line has to be considered, the net cooling capacity is reduced further.

4.4 Efficiency of Cold Production

For the above considerations a CARNOT cycle was assumed for cooling. A real cycle, however, consists of irreversible processes. Let us assume that only the compression process and expansion process are associated with entropy production. The ratio of the *COP* of a CARNOT cycle between T_{o} and T_{c} is

$$COP_{\text{ocC}} = \frac{q_{\text{o}}}{w_{\text{t,co}}^{\text{rev}}} = \frac{T_{\text{o}}}{T_{\text{c}} - T_{\text{o}}} \tag{19}$$

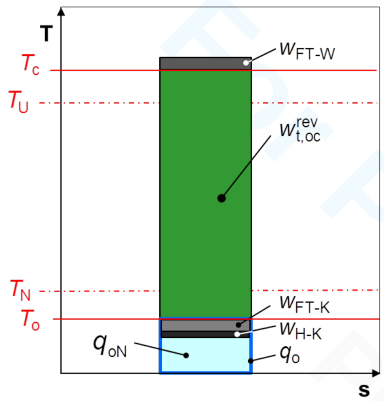

Figure 5: Specific work in the $T,s-$diagram

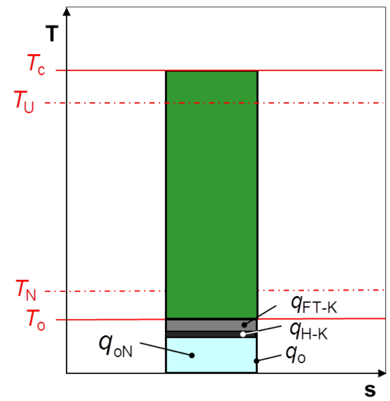

Figure 6: Specific net cooling capacity

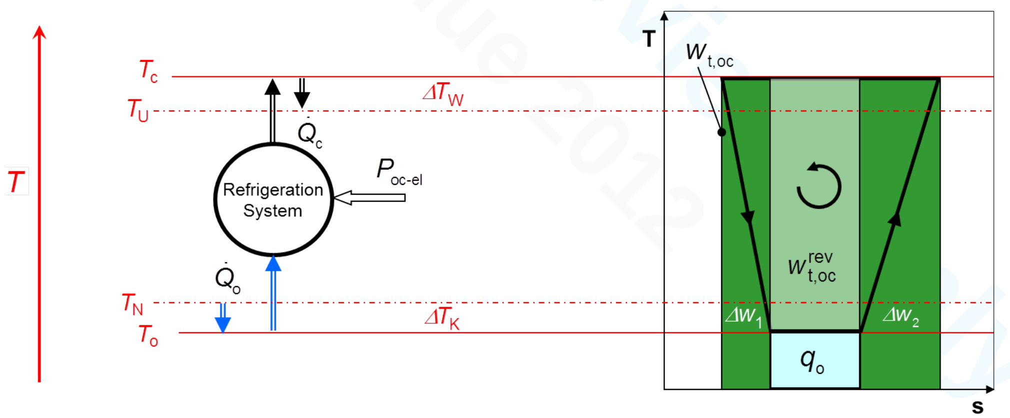

Figure 7: Comparison of a CARNOT cycle and an irreversible cycle

and the *COP* of an irreversible cycle, that operates between the same heat source and sink temperatures is

$$COP_{oc}^{\#} = \frac{q_o}{w_{t,oc}}. \tag{20}$$

Thus an efficiency of cold production can be defined as

$$\eta_{KC} := \frac{COP_{oc}^{\#}}{COP_{ocC}} = \frac{w_{t,co}^{rev}}{w_{t,co}} = \frac{T_c - T_o}{T_o} \cdot \frac{q_o}{w_{t,co}} \tag{21}$$

with

$$w_{t,oc} = w_{t,oc}^{rev} + \Delta w_1 + \Delta w_2. \tag{22}$$

A larger specific work is required for the irreversible processes, as shown in figure 7.

4.5 Total energy efficiency

The product of the derived efficiencies describes the overall energy efficiency of a system. It is named “level of energy efficiency” and defined as

$$\eta_{\text{ges}} = \eta_{\text{KC}} \cdot \eta_{\text{WT}} \cdot \eta_{\text{FT}} \cdot \eta_{\text{Qo}}. \tag{23}$$

Including the individual definitions it results in

$$\eta_{\text{ges}} = \frac{COP_{\text{oc}}}{COP_{\text{ocC}}} \cdot \frac{COP_{\text{ocC}}}{COP_{\text{NUC}}} \cdot \frac{P_{\text{oc-el}}}{P_{\text{ges}}} \cdot \frac{\dot{Q}_{\text{oN}}}{\dot{Q}_{\text{o}}} = \frac{\dfrac{\dot{Q}_{\text{o}}}{P_{\text{oc-el}}}}{\dfrac{T_{\text{o}}}{T_{\text{c}} - T_{\text{o}}}} \cdot \frac{\dfrac{T_{\text{o}}}{T_{\text{c}} - T_{\text{o}}}}{\dfrac{T_{\text{N}}}{T_{\text{U}} - T_{\text{N}}}} \cdot \frac{P_{\text{oc-el}}}{P_{\text{ges}}} \cdot \frac{\dot{Q}_{\text{oN}}}{\dot{Q}_{\text{o}}}. \tag{24}$$

The reduction to the most important terms gives

$$\eta_{\text{ges}} = \frac{COP}{COP_{\text{NUC}}} = \frac{\dot{Q}_{\text{oN}}}{P_{\text{ges}}} \cdot \frac{T_{\text{U}} - T_{\text{N}}}{T_{\text{N}}}. \tag{25}$$

The level of energy efficiency η_{ges} is equal to the exergetic efficiency, the second law efficiency of the system. (Remark: The term energy efficiency is used, even though the definition is based on powers.)

A transfer of the above considerations to a refrigeration system with a real vapor compression cycle is easily possible. In figure 8 an example demonstrates the above mentioned areas for a basic vapor compression cycle which is much closer to a real vapor compression cycle than a CARNOT cycle. An adiabatic compressor and an isenthalpic expansion are assumed. The power supply for auxiliary units and other heat sources are not shown here, to preserve the simplicity of the diagram.

The white area in the center represents the specific input power for a comparable CARNOT cycle $w^{\text{rev}}_{\text{t,NU}}$. It is the specific exergy of the specific refrigerating capacity.

The whole colored area including the white area minus the square area of q_{o} is the specific energy input for the cycle with an adiabatic compressor.

Additional to the energy of the vapor compression cycle the energy input for fluid transport and heating are shown in figure 9. If assumed that the energy input w to the heat source (fan and defrost energy) is dissipated to q heat ($q_{\text{FT-K}}$ and $q_{\text{FT-W}}$), the respective areas can be drawn on the $T,s-$diagram. One can clearly see that the input becomes bigger, while the net refrigerating capacity becomes smaller.

In figure 10 the specific exergy losses of all processes are shown, e_{v12} for the compressor, e_{v23} for the condenser, e_{v34} for the expansion valve and e_{v41} for the evaporator etc.

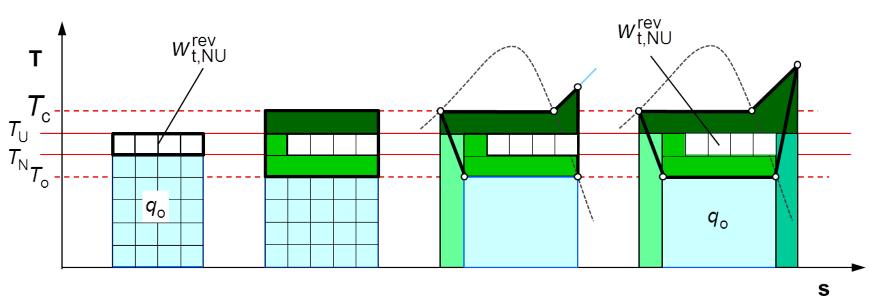

Figure 8: Specific energy as areas in a T,s-diagram, schematically for the ideal CARNOT cycle, a CARNOT cycle with heat transfer losses, a vapor compression refrigeration cycle with adiabatic and reversible compression and vapor compression refrigeration cycle (from left).

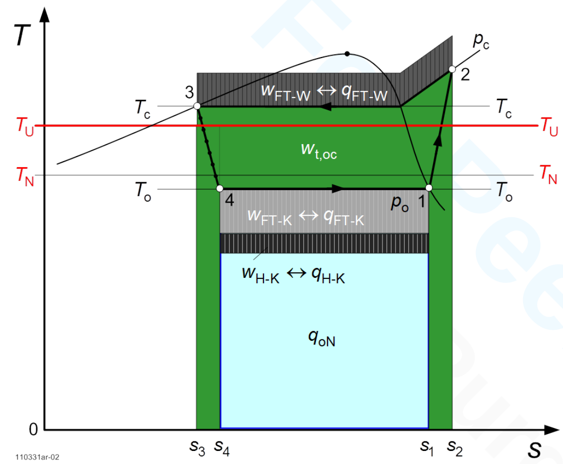

Figure 9: Specific energies of the system in a temperature-entropy-diagram

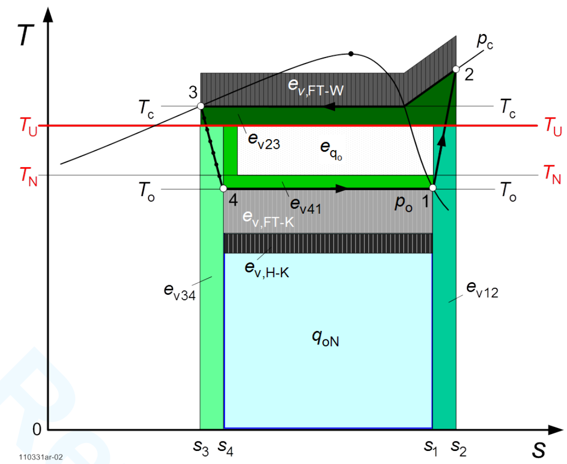

Figure 10: Specific exergy losses (e_V) in a temperature-entropy-diagram

5. OUTLOOK

The method can be expanded to a set of conditions to optimize a design or the operation of a system. Additionally, the method can be extended to assess the compressor, to understand and explain more details. With the following equation an example is given

$$\frac{\dot{Q}_{\mathrm{o}}}{P_{\text{oc-el}}} = \frac{\dot{Q}_{\mathrm{o}}}{\dot{Q}_{\mathrm{oV}}} \cdot \frac{\dot{Q}_{\mathrm{oV}}}{\dot{Q}^*_{\mathrm{oV}}} \cdot \frac{\dot{Q}^*_{\mathrm{oV}}}{\dot{W}_{\mathrm{tV}}} \cdot \frac{\dot{W}_{\mathrm{tV}}}{P_{\mathrm{m}}} \cdot \frac{P_{\mathrm{m}}}{P_{\text{oc-el}}}, \tag{26}$$

with

$\dot{W}_{\mathrm{tV}}$	Power given to the gas
P_{m}	Mechanical power
$P_{\text{oc-el}}$	Electrical power
$\dot{Q}_{\mathrm{oV}}$	Refrigerating capacity of the compressor
$\dot{Q}^*_{\mathrm{oV}}$	Refrigerating capacity of the compressor with subcooling

6. CONCLUSION

The method published in VDMA 2424-7-2 is capable to evaluate the energy efficiency at constant operation conditions. The final result is equal to the exergetic efficiency. The method is not as powerful as a details exergetic analysis, but it is capable to improve a lot of systems. Due to the fact that it seems to be easier, the acceptance in application could be higher than the exergetic analysis.

NOMENCLATURE

Symbols

A	Heat transfer area	(m^2)	$\dot{W}$	Power	(W)
COE	Coefficient of energy	(J/J),(Wh/Wh)	w_t	Specific technical work	(J/kg)
COP	Coefficient of performance	(W/W)	Δ	Difference	
e	Specific exergy	(W/kg)	η_{KC}	Efficiency of cold production	(W/W)
$\dot{m}$	Mass flow	(kg/s)	η_{WT}	Efficiency of heat transfer	(W/W)
P	Power	(W)	η_{FT}	Efficiency of fluid transport	(W/W)
$\dot{Q}$	Heat flow	(W)	η_{Qo}	Efficiency of cold utilisation	(W/W)
q	Specific heat	(J/kg)	η_{ges}	Total efficiency of a system	(W/W)
T	Temperature	(K)	τ	Time	(s)
U	Heat transfer coefficient	(W/(m^2 K))			

Subscripts

c	Condensation	NC	From T_N to T_U
C	CARNOT	o	Evaporation
el	Electrical	oc	From T_o to T_c
FT	Fluid transport	rev	Reversible
ges	Total	R	Refrigerant
H	Heating	U	Ambient
K	Cooling	V	Compressor
m	Mechanical	v	Loss
N	Benefit	WT	Heat transfer

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