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Abstract

Work presented at the annual meeting of the DKV in 2020 (online)

To meet the criteria of a sustainable use of energy in future, the development of refrigeration plants using CO2 as working fluid is of great importance. Especially integrated cycle layouts, covering a heat recovery for warm water supply and space heating, can contribute to increase the energy efficiency of supermarkets. Further, the utilization of ejectors for work recovery can enhance the energy efficiency. This paper deals with the analysis of different operation modes. The comparison of a combi ejector operation and an operation with expansion valve shows an improved energetic efficiency at higher ambient temperatures. The analysis reviled also a significant utilization of heat recover during ambient temperatures >15°C for the combi ejector operation compared to the HPV – operation. Comparison of fist law efficiency (COP) based on the high side pressure/gas cooler pressure of the plant reveals an COP increase for the combi ejector operation of 10.6 % at 80 bar.

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1 Paper

DKV-Tagung 2020, Magdeburg, AA III

Evaluation of ejector supported supermarket refrigeration systems

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Abstract

To meet the criteria of a sustainable use of energy in future, the development of refrigeration plants using CO_2 as working fluid is of great importance. Especially integrated cycle layouts, covering a heat recovery for warm water supply and space heating, can contribute to increase the energy efficiency of supermarkets. Further, the utilization of ejectors for work recovery can enhance the energy efficiency.

This paper deals with the analysis of different operation modes. The comparison of a combi ejector operation and an operation with expansion valve shows an improved energetic efficiency at higher ambient temperatures. The analysis reviled also a significant utilization of heat recover during ambient temperatures >15°C for the combi ejector operation compared to the HPV – operation. Comparison of fist law efficiency (COP) based on the high side pressure/gas cooler pressure of the plant reveals an COP increase for the combi ejector operation of 10.6 % at 80 bar.

Keywords:

Supermarket, Multi Ejector, Integrated systems for heating and cooling, measures for efficiency improvements

1 Introduction

For a long time, conventional refrigeration technology was based on hydrofluorocarbons (HFCs), which annually emit an amount of 0,35 Gt of carbon dioxide into the earth's atmosphere due to leakage and therefore contribute to global warming as short-lived climate forcer (Velders, 2015). During the past decades several international treaties, such as the Montreal Protocol and the Paris Climate Agreement, have established a pathway for policy makers to enforce the utilization of climate friendly refrigerants. Since then, carbon dioxide as working fluid gained increasing interest as low GWP, non-ozone depleting and non-flammable refrigerant with extraordinary properties. However, the uptake of CO_2 refrigeration plants is still slow, especially in supermarket systems. According to the findings of the IEA improved commercial refrigeration systems can be operated with only 60% of the average energy consumption, if there are still systems with 55% energy intensity above average (van der Sluis, 2017). Hereby, the focus is set on integrated solutions, providing cooling, heating and air-conditioning with on cycle.

Nowadays, the most promising plant design presents a parallel compression cycle, enhanced with ejectors for work recovery in order to reduce the exergetic expansion losses. This integrated plant configuration provides the shops with cooling and heating power. As project partner, Danfoss contributes its expertise in the Multi Ejector technology. This solution consists of several ejector cartridges, which are designed in a combination of fixed geometries and allows the plant operator to utilize ejectors at changing operating conditions. Several supermarkets throughout Europe were equipped with the system and are currently investigated to evaluate

Table 1 – System dimensioning capacities and reference temperatures

Service	Reference temperature	Cooling/Heating Capacity
LT	-31°C	16,55 kW
MT	-4°C/-6°C	43,80 kW
HR	40°C	30,00 kW

various energy characteristics in different operation modes and climates. This paper deals with the results of an energy analysis on a CO_2 parallel compression plant, installed in a supermarket in Germany.

2 System Description

2.1. Plant layout

The investigated configuration of ejector supported R-744 plants is installed in a supermarket in Germany. A parallel compression system with two evaporation levels, low and medium temperature cooling, was designed (see figure 1). The plant is equipped with a heat recovery unit to provide heating water, consisting of two plate heat exchangers and an inner brine circle. Table 1 presents the design dimensions of the system. Downstream the gas cooler, an internal heat exchanger further cools the high-pressure working fluid utilizing the cold flash gas. The receiver is fed either by the multi-ejector system or high-pressure expansion valve.

The plant builder decided to implement one high pressure lift gas ejector and one liquid ejector, differing in their inherent geometries. Both ejector cartridges are supplied with fluid coming from a common port at the

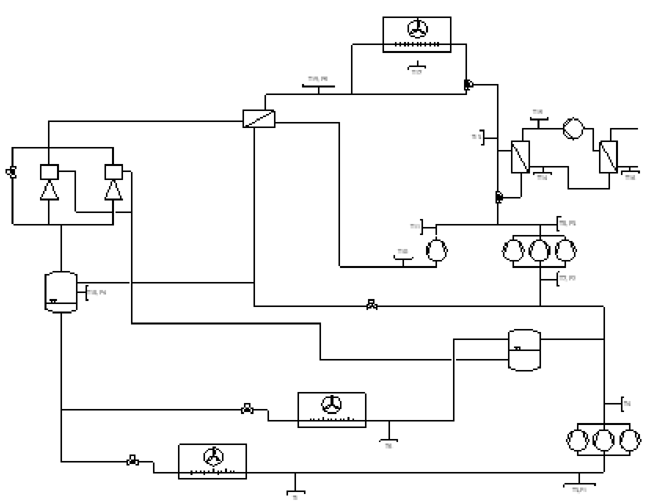

Figure 1 – PID parallel compression cycle enhanced with multiejector technology

accumulator. Depending on the ejector envelopes and operating conditions, the control of cartridges follows different operating strategies. Thus, the multi-ejector system has great importance in the high-pressure control strategy (Danfoss, 2020). The installed expansion devices allow three different expansion modes based on, i.e. expansion valve, high pressure lift ejector based and liquid ejector. The compressor packs are equipped with Bitzer fabricates. Within the LT and MT group one compressor works with additional frequency adaption in the range of 30–70 Hz. Estimated compressor capacities are presented in table 1 according to the results of the Bitzer software (Bitzer Kühlmaschinenbau GmbH, 2020). After the discharge port of the LT group, superheated gas is uniting with saturate flash gas from the accumulator. Furthermore, the flow mixes with two phase fluid coming from the constant pressure valve, regulating the receiver pressure. The following MT compressors lift the refrigerant to high side pressure level. Besides the constant pressure valve, the auxiliary compressor is an integral part of controlling the receiver pressure by sucking superheated flash gas. On the high-pressure side, the plant control has two ways of rejecting the heat. When there is no heat reclaim, the fluid is bypassed directly to the gascooler, rejecting the heat to the ambient. In times of high heat demand, the gascooler is bypassed. In subcritical operation the heat rejection is working as condensation process. In times of high ambient temperatures or having heat demand, the compressor control switches to transcritical operation, leading to a temperature glide in the heat exchangers. Especially for hot water supply transcritical heat rejection is advantageous due to the move of the pinch point towards the heat exchanger outlet. Moreover, Danfoss implemented the CALM system, to operate the MT evaporators with nearly no superheat (ALC ("adaptive liquid control") (Danfoss, 2018)). As a consequence, the suction pressure is increased, and the compressor pressure ratio is reduced. Hence, the compressor capacity decreases.

Table 2 – Nominal electric power input for compressors

Level	**Type**	**Power Supply**
LT	2JSL-2K	1,51 kW (3x)
MT	4JTC-15K	11,78 kW (3x)
AUX	4JTC-15K	12,14 kW (3x)
all		52,01 kW

2.2. Testing scheme

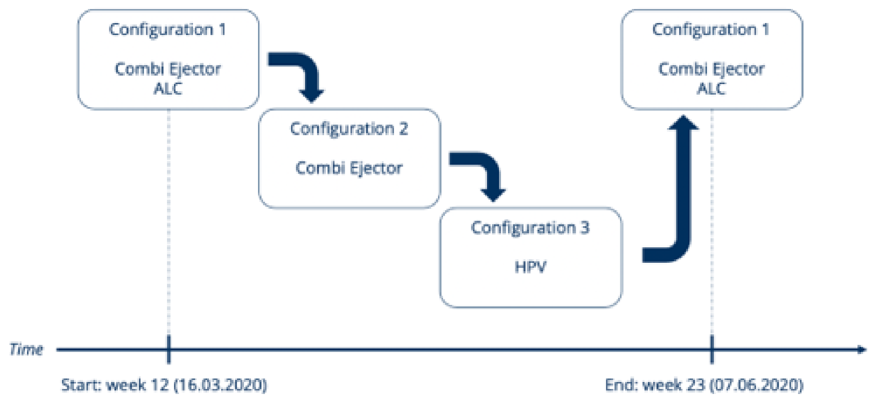

Figure 2 – Implemented testing scheme with operational changes

The Danfoss' controller allows a sophisticated testing scheme with three configurations (see figure 2) to gain an overview of the influences on pack performance and energy consumption as function of the ambient conditions. The plant operates for 4 weeks in one configuration mode and afterwards switches to the next mode. This operational cycle restarts after three months to cover various ambient temperatures throughout the year. The measurement data of the system parameters is provided from week 12 to week 23.

3 Methodology

Table 3 – Sensor accuracies

Sensor type	Range	Accuracy
Pressure sensor 1	$-1bar \leq p \leq 59bar$	±0,8% FS
Pressure sensor 2	$-1bar \leq p \leq 159bar$	±0,8% FS
Temperature sensor	$-70^{\circ}C \leq \vartheta \leq 180^{\circ}C$	$\pm(0{,}3°C + 0{,}005 \cdot \vartheta)$
Watt meter	n.a.	±0,5%

Within in the measuring system of the investigated plant, the plant builder did not implement mass flow metering devices. The installed wattmeter has no capability of measuring each compressor power as well. Therefore, it was necessary to determine the compressor mass flows and powers with approximations. All com- pressors are Bitzer fabricants. The company provided approximative polynomials to simulate various compressor parameters based on DIN EN 12900. The coefficients were calculated with a defined superheat, which needed to be chosen in the software in advance. To reach the highest possible accuracy of the equations, the actual superheat of each compressor level (LT, MT and AUX) needed to be evaluated. A promising approach was to use the median value of the superheat from the chosen time frame in order to reduce the influence of peaks. Subcritical and transcritical polynomials were implemented in accordance to the manufacturer. In subcritical operation the polynomial function is dependent on the evaporation ϑ_0 and the condensation ϑ_C temperature.

$$y_{i,sub} = c_1 + c_2\vartheta_0 + c_3\vartheta_C + c_4\vartheta_0^2 + c_5\vartheta_0\vartheta_C + c_6\vartheta_C^2 + c_7\vartheta_0^3 + c_8\vartheta_0^2\vartheta_C + c_9\vartheta_0\vartheta_C^2 + c_{10}\vartheta_C^3 \qquad \text{Eq. 1}$$

The transcritical operation requires high-side pressures above the critical point, where no condensation process is possible. Hence, the high-pressure p_H removes the condensation temperature as the second variable.

$$y_{i,tran} = c_1 + c_2\vartheta_0 + c_3p_H + c_4\vartheta_0^2 + c_5\vartheta_0p_H + c_6p_H^2 + c_7\vartheta_0^3 + c_8\vartheta_0^2p_H + c_9\vartheta_0p_H^2 + c_{10}p_H^3 \qquad \text{Eq. 2}$$

The LT compressors only operate below the critical point and therefore only apply the subcritical polynomial. However, the MT and AUX compressors are also operating above the critical point. Hence, the program had to consider both types of polynomials depending on the high side pressure.

Moreover, one compressor of each pressure level group (LT, MT, AUX) was equipped with one frequency inverted drive within the range of 30Hz – 70Hz. The controller saved the nominal frequency in %. A conversion of the frequency parameter with a correction factor was needed to adjust the compressor equation (see equation 5.3), since the Bitzer Software just provided the polynomials at one frequency (50Hz in this case). The parameter and the initial compressor polynomial were multiplied to gain a linear dependency of the network's frequency and the change of mass flow and power (see Equation 1), giving an acceptable accuracy. A similar approach can be found in literature pointing out deviations of less than 0,5%, (Vindenes, 2018).

$$a = 0{,}008 \cdot f[\%] + 0{,}6 \qquad \text{Eq. 3}$$

$$y^*_{i,sub/tran} = a \cdot y_{i,sub/tran}$$

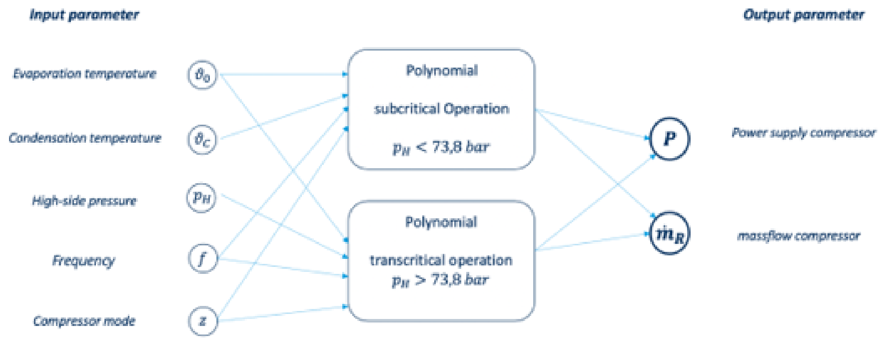

Figure 3 – Implemented compressor model

The compressor model was programmed in Python and is illustrated in figure 3. The fluid data was obtained by the open source database Coolprop (Bell, 2014).

The heat loads of the Gascooler (GC) and the heat recovery (HR) as well as the refrigeration load of all low temperature evaporators (LT) were determined by the enthalpy difference multiplied with the corresponding mass flow (see eq. 4).

$$\dot{Q} = \dot{m} \cdot (h_{i+1} - h_i) \qquad \textit{Eq. 4}$$

Furthermore, it was not possible to compute the load of the all MT evaporators in an enthalpy balance, since the mass flow balance was indefinite in this case. Therefore, the MT load had to be calculated via an energy balance (see Equation 5), neglecting possible heat losses in the lines and components. Following this approach, the overall real heat loss was summarized in the value of $\dot{Q}_{MT}$, which had to be considered in the evaluation of uncertainties.

$$\dot{Q}_{MT} = \dot{Q}_{GC} + \dot{Q}_{HR} - \sum_i P_{i,el,compressor} - \dot{Q}_{LT} \qquad \textit{Eq. 5}$$

The total COP, according to the first law of thermodynamics with all heat loads, is defined as:

$$COP_{tot} = \frac{\dot{Q}_{MT} + \dot{Q}_{LT} + \dot{Q}_{HR}}{P_{MT} + P_{LT} + P_{AUX}} \qquad \textit{Eq. 6}$$

In respect to changing ambient temperatures, variable loads and load ratios, it seems beneficial to investigate the second law efficiency, allowing an easy comparison of different systems. The temperature references are chosen according to the indicated reference states for the plant.

$$\eta_{tot} = \frac{\dot{Q}_{MT}\left(\frac{T_{amb}}{T_{MT,ref}} - 1\right) + \dot{Q}_{LT}\left(\frac{T_{amb}}{T_{LT,ref}} - 1\right) + \dot{Q}_{HR}\left(1 - \frac{T_{amb}}{T_{HR,ref}}\right)}{P_{MT} + P_{LT} + P_{AUX}} \qquad \textit{Eq. 7}$$

4 Results and Discussion

4.1. Cooling and heating loads

Figure 4 presents the heat capacities of LT, MT and HR. If the ambient temperature is rising, the capacity of LT and MT show an increasing trend. The variation of MT data at lower ambient temperatures is caused by the method of computation of the heat capacity (see Equation 5). The fluctuations of HR in this region can contribute to higher $\dot{Q}_{MT}$ values, while the power supply of the compressors does not change. This effect leads to increased uncertainty in this region and a high variation. Further, the data of $\dot{Q}_{MT}$ is determined with an offset, caused by the neglected sum of heat losses within the system. However, the results indicate a rather constant ratio of Q_{MT}/Q_{LT} in the order of 2/1, whereas the heat recovery of the system in operation also beyond typical ambient temperatures where building heat demand is expected.

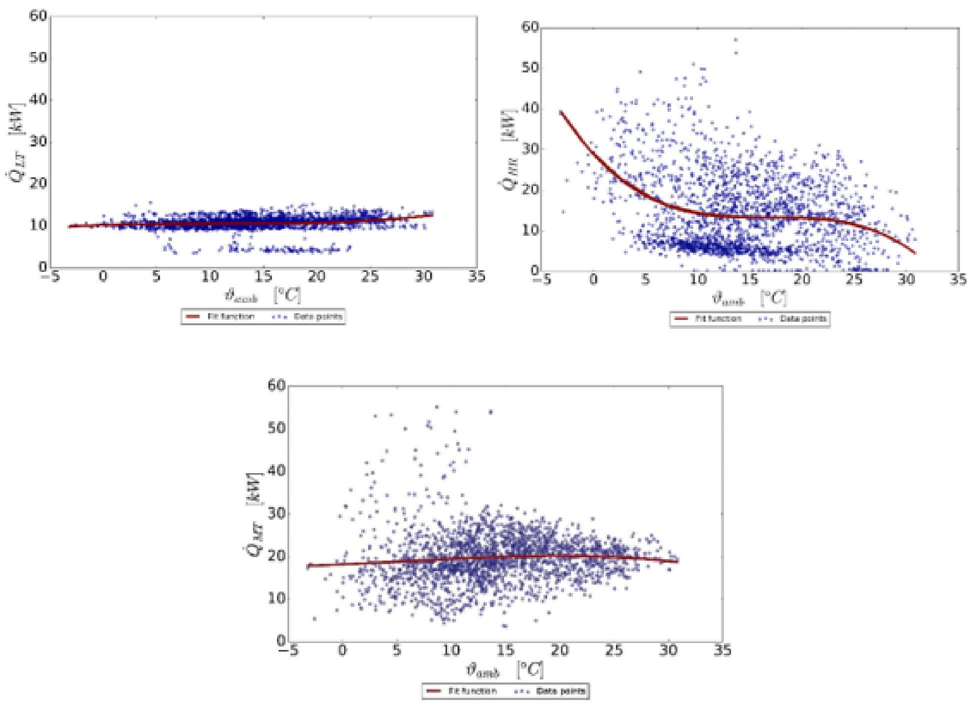

Figure 4 – Cooling and heating loads as function of ambient temperature

4.2. Comparison combi ejector mode vs. baseline (HPV)

In this section the focus is set on the comparison of selected energy parameters of two configurations: plant operation with combi ejector (green) and the operation with the HPV (red), representing the baseline. The measured electric energy consumption per hour in combi ejector mode was higher due to increased HR demand at low temperatures ($\vartheta_{amb} < 15°C$). At higher ambient temperatures both fit curves converge and show an intersection at ($\vartheta_{amb} \approx 20°C$). At higher ambient temperatures both fit curves converge and show an intersection at ($\vartheta_{amb} \approx 20°C$). In both operations, the compressor power was increasing to enable a proper heat rejection at high temperatures (transcritical operation with higher pressure ratio). With larger sample size, the energy consumption in combi ejector mode is expected to be lower at higher ambient temperatures.

The exergy efficiency, the combi ejector mode clearly benefited from the reduced throttling losses by utilization of the combi ejector unit. In general, the exergetic efficiency increases until $\vartheta_{amb} \approx 17°C$, caused by a higher exergetic load of the LT and MT evaporators at higher ambient temperatures (see Eq. 7). Further, a contrary effect can be seen at rising ambient temperatures. The exergetic efficiency decreases, since the capacity of the compressors rises significantly due to the transcritical operation. The maximum efficiency improvement of combi ejector mode can be observed at $\vartheta_{amb} = 21°C$ with an efficiency plus of 5,04% ($\Delta\eta_{ex} = 0{,}01$) compared to the baseline operation. Additionally, it can be observed that the combi ejector mode is less efficient than the HPV mode in times with higher HR demand, which is consistent with the trend of the energy consumption.

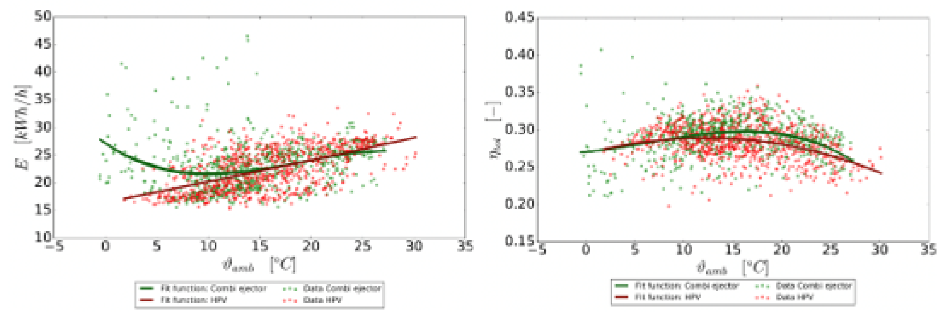

Figure 5 – Left- Measured energy consumption, Right - Second law efficiency

Analyzing the heat loads, energy consumption and the second law efficiency in detail indicated a much higher utilization of heat recovery functionality of the plant during combi ejector operation compared to the HPV operation, also indicated by the increased number of datapoints at pressures >80 bar for the combi ejector mode in Figure 6 -left hand side. The heat recovery functionality increases, independently of the ambient temperature, the discharge pressure set point of the MT and parallel compressor up to >80 bar. Comparing the heat recovery operation in HPV mode (Figure 6 rigth hand side) and of combi ejector model (Figure 6 left hand side) shows an COP increase of 10.6 % at 80 bar from 3.30 to 3.65, respectively.

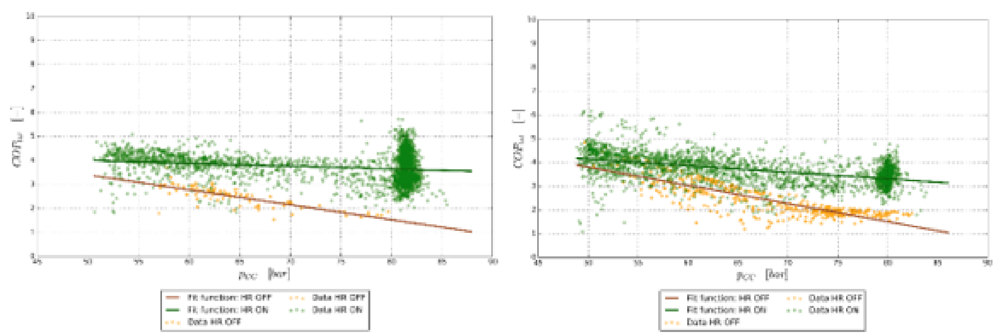

Figure 6 – Heat recovery effect on first law efficiency. Left - combi ejector mode, Right – HPV mode

5 Summary

This work emphasizes the importance of introducing carbon dioxide as natural working fluid in integrated commercial refrigeration plants in order to reduce their carbon footprint. The extraordinary thermodynamic properties of CO_2 and its corresponding application possibilities were pointed out. Especially at high ambient temperatures or in transcritical operation the utilization of an ejector device contributes to reduce the exergetic losses due to an isentropic expansion. Danfoss introduced the multi-ejector technology to combine different ejector geometries in one system. Together with an adaptive liquid management the plant efficiency can be improved significantly. The investigated plant was analyzed with a Python program. Due to missing watt meters (could not be installed due to Covid-19) at the compressors power supply as well as mass flow metering devices a compressor model was established to determine the missing parameters. The approach shows promising results for mass flow and power values. In the analysis, major plant operation modes were compared regarding their power consumption, first- and second law efficiency as function of the ambient temperature. When the cycle was operated with the combi ejector technology, the calculated results show significant enhancements of the first-law efficiency towards higher ambient temperatures. The improvements of the second-law efficiency were even higher at medium ambient temperatures. The analysis reviled a significant utilization of heat recover during ambient temperatures >15°C for the combi ejector operation compared to the HPV – operation.

Comparison of fist law efficiency (COP) based on the high side pressure/gas cooler pressure of the plant reveals an COP increase for the combi ejector operation of 10.6 % at 80 bar.

Due to the relatively short timeframe only a part of the prevailing ambient conditions was covered from the measuring system. A comprehensive overview about the plant will be possible, when the controller unit gathers data from all conditions.

Further work will be dedicated to the validation of the computed figures with an improved measuring system, including a higher sampling rate (<1 min) and watt meters for each compressor set. The high uncertainty of the determination of $\dot{Q}_{MT}$ need to be addressed. To compare the energy characteristics with other cycles, key performance indicators can be determined area-depended (i.e. cooling room size, shop area or display surface as presented by VDMA "Quickcheck" (Heinbokel, 2019)).

ACKNOWLEDGEMENTS

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NOMENCLATURE

Symbol		*Suffixes and Acronyms*	
E	Electric energy consumption (kWh)	el	electric
P	Electrical Power demand (kW)	LT	Low temperature application
$\dot{Q}$	Thermal Power (kW)	MT	Medium temperature application
T	Temperature (K)	AUX	auxilliary
ϑ	Temperature (°C)	tot	total
COP	Coefficient of performance	0	evaporation
η	Efficiency	C	condensation
HPV	High-pressure expansion valve	amb	ambient
p	Pressure (Pa)	ex	exergetic
HR	Heat recovery		

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Bewertung von ejektorgestützten R744 Supermarkt-Kühlsystemen (Evaluation of ejector supported R744 Supermarket Refrigeration systems)
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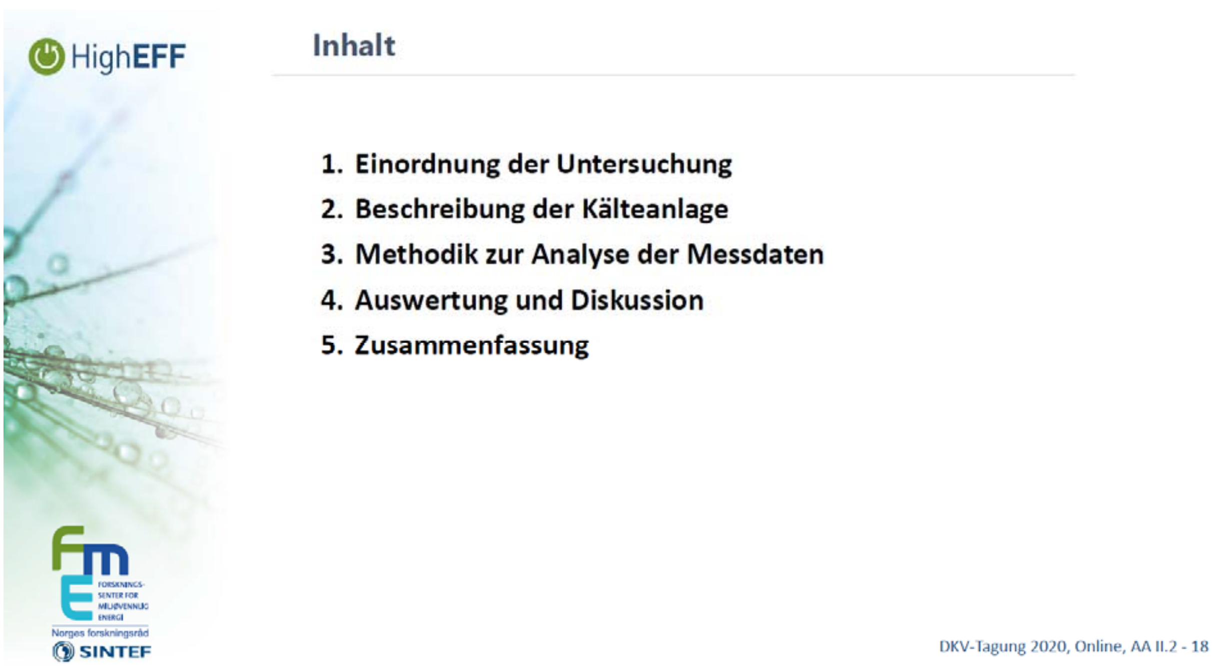
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3. Methodik zur Analyse der Messdaten
4. Auswertung und Diskussion
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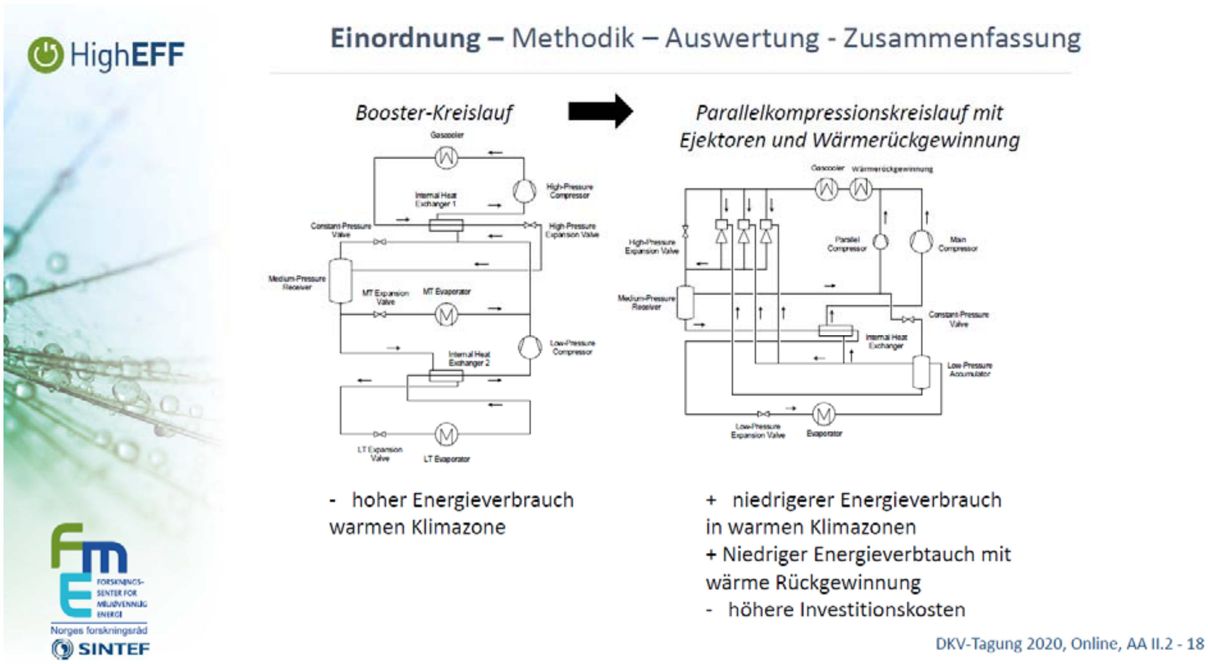
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Einordnung – Methodik – Auswertung - Zusammenfassung
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Parallelkompressionskreislauf mit Ejektoren und Wärmerückgewinnung
Gascooler
High-Pressure Compressor
Internal Heat Exchanger 1
Constant-Pressure Valve
High-Pressure Expansion Valve
Medium-Pressure Receiver
MT Expansion Valve
MT Evaporator
Low-Pressure Compressor
Internal Heat Exchanger 2
LT Expansion Valve
LT Evaporator
Wärmerückgewinnung
Parallel Compressor
Main Compressor
Constant-Pressure Valve
Internal Heat Exchanger
Low-Pressure Accumulator
Low-Pressure Expansion Valve
Evaporator
- hoher Energieverbrauch warmen Klimazone
+ niedrigerer Energieverbrauch in warmen Klimazonen
+ Niedriger Energieverbtauch mit wärme Rückgewinnung
- höhere Investitionskosten
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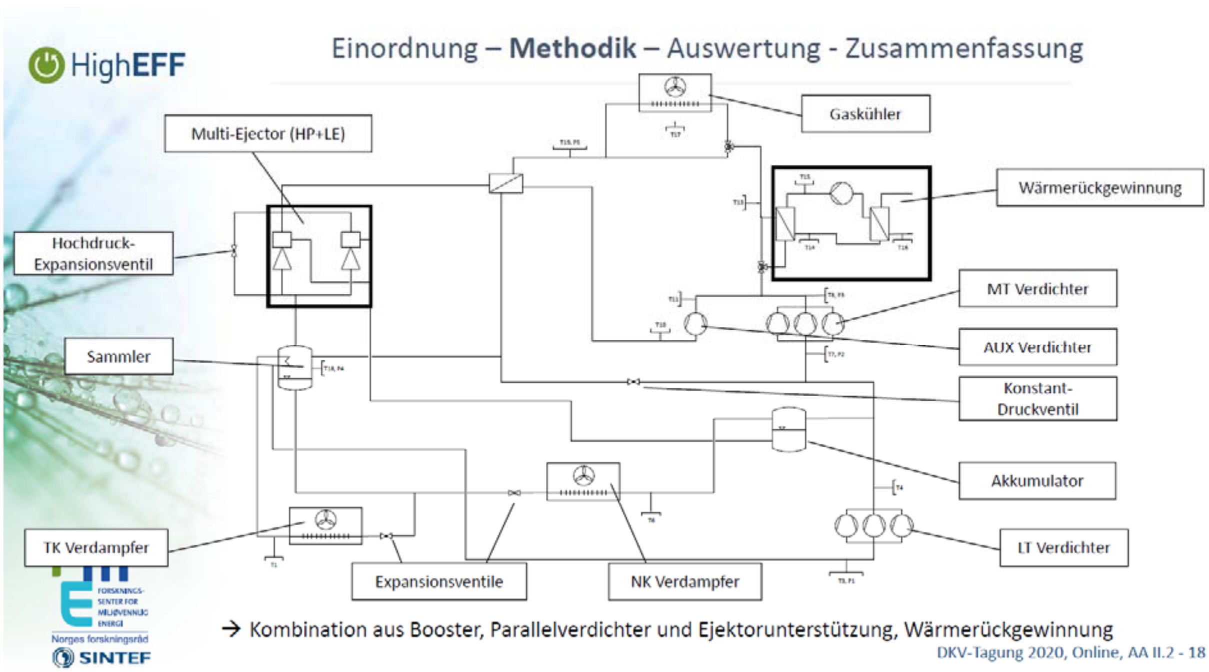
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Einordnung – Methodik – Auswertung - Zusammenfassung
Gaskühler
Multi-Ejector (HP+LE)
Wärmerückgewinnung
Hochdruck-Expansionsventil
MT Verdichter
AUX Verdichter
Sammler
Konstant-Druckventil
Akkumulator
TK Verdampfer
LT Verdichter
Expansionsventile
NK Verdampfer
→ Kombination aus Booster, Parallelverdichter und Ejektorunterstützung, Wärmerückgewinnung
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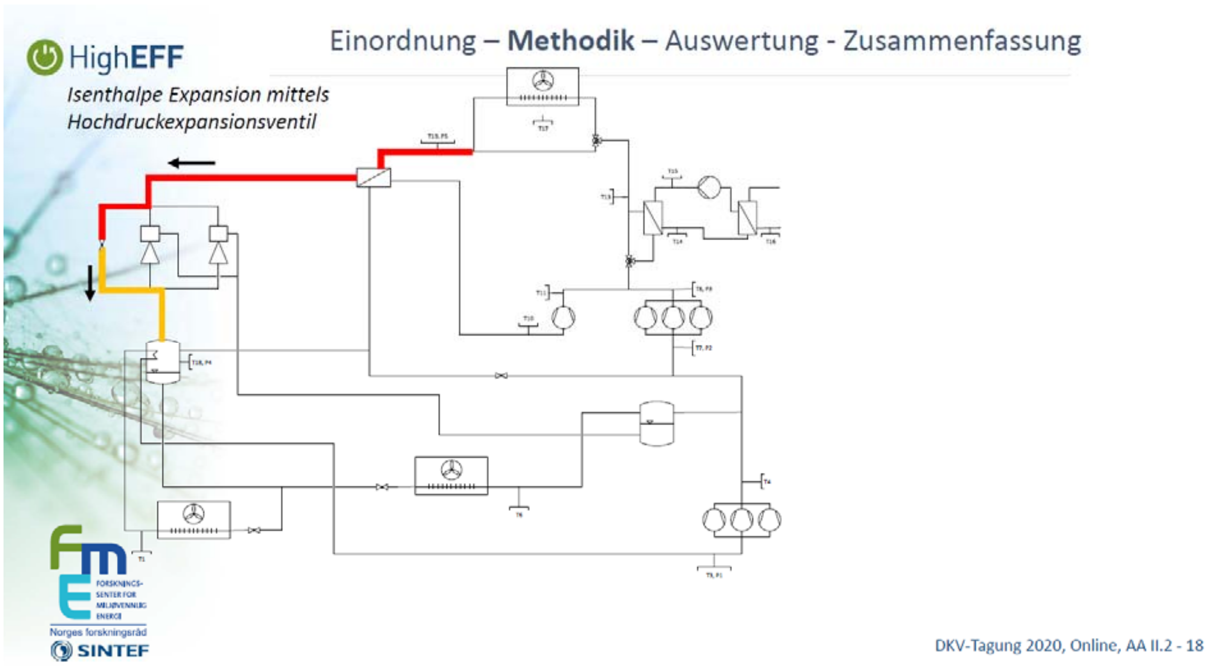
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Isenthalpe Expansion mittels
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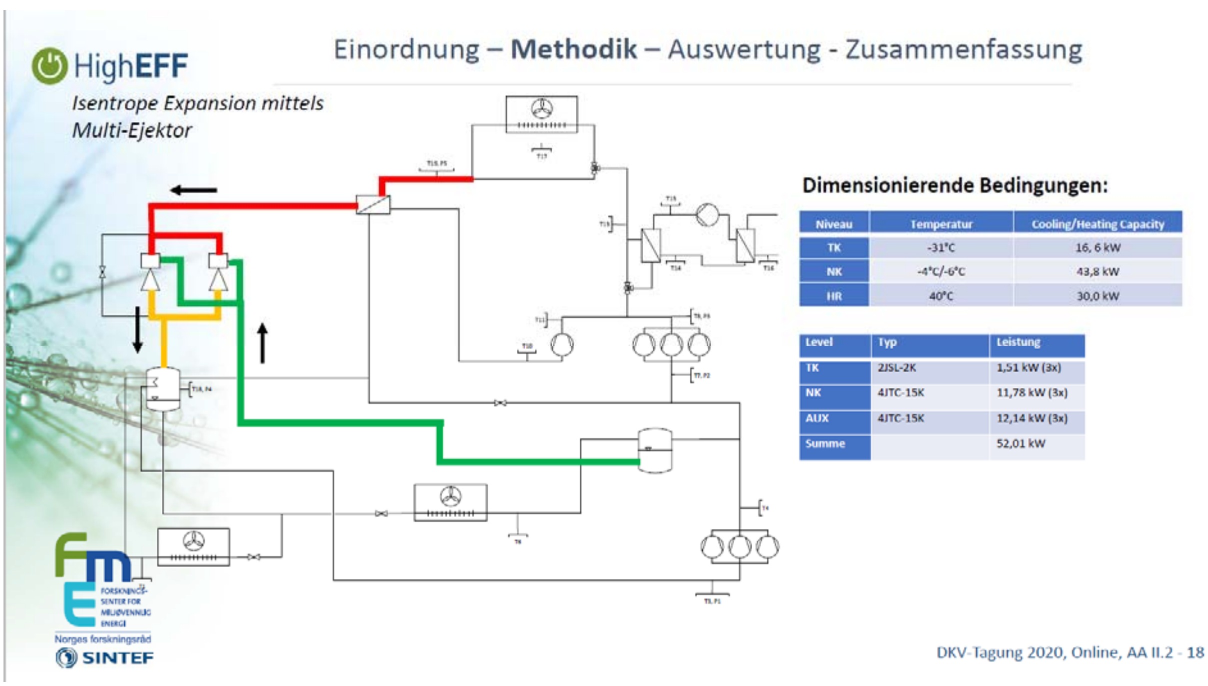
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Einordnung – **Methodik** – Auswertung - Zusammenfassung
Isentrope Expansion mittels
Multi-Ejektor
Dimensionierende Bedingungen:

Niveau	Temperatur	Cooling/Heating Capacity
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HR	40°C	30,0 kW

Level	Typ	Leistung
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NK	4JTC-15K	11,78 kW (3x)
AUX	4JTC-15K	12,14 kW (3x)
Summe		52,01 kW

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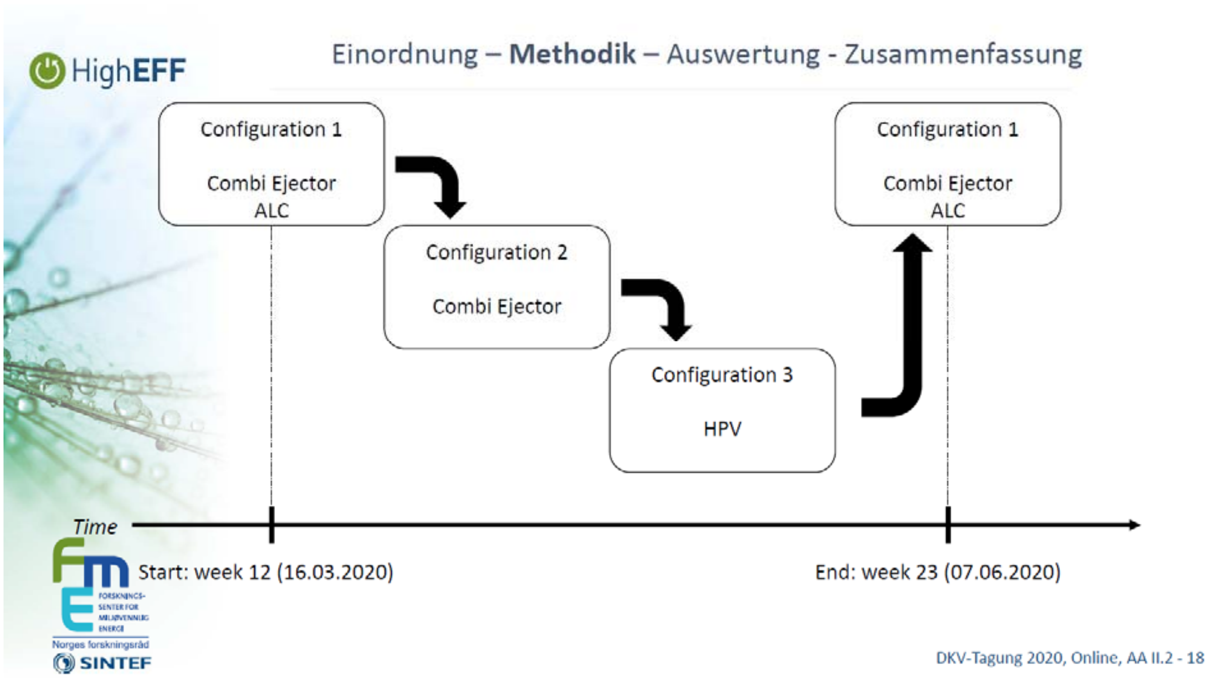
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Einordnung – Methodik – Auswertung - Zusammenfassung
Configuration 1
Combi Ejector
ALC
Configuration 2
Combi Ejector
Configuration 3
HPV
Configuration 1
Combi Ejector
ALC
Time
Start: week 12 (16.03.2020)
End: week 23 (07.06.2020)
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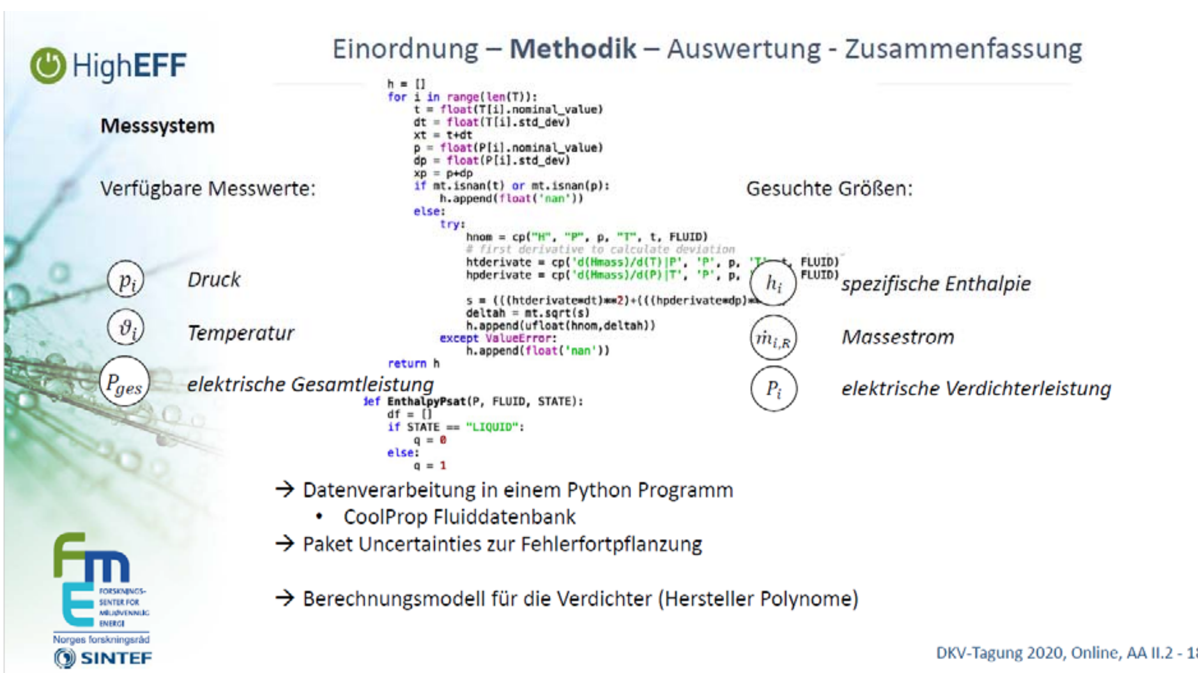
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Einordnung – Methodik – Auswertung - Zusammenfassung
Messsystem
Verfügbare Messwerte:
p_i Druck
ϑ_i Temperatur
P_{ges} elektrische Gesamtleistung
Gesuchte Größen:
h_i spezifische Enthalpie
$\dot{m}_{i,R}$ Massestrom
P_i elektrische Verdichterleistung
h = []
for i in range(len(T)):
t = float(T[i].nominal_value)
dt = float(T[i].std_dev)
xt = t+dt
p = float(P[i].nominal_value)
dp = float(P[i].std_dev)
xp = p+dp
if mt.isnan(t) or mt.isnan(p):
h.append(float('nan'))
else:
try:
hnom = cp("H", "P", p, "T", t, FLUID)
# first derivative to calculate deviation
deltah = mt.sqrt(s)
h.append(ufloat(hnom,deltah))
except ValueError:
h.append(float('nan'))
return h
def EnthalpyPsat(P, FLUID, STATE):
df = []
if STATE == "LIQUID":
q = 0
else:
q = 1
→ Datenverarbeitung in einem Python Programm
• CoolProp Fluiddatenbank
→ Paket Uncertainties zur Fehlerfortpflanzung
→ Berechnungsmodell für die Verdichter (Hersteller Polynome)
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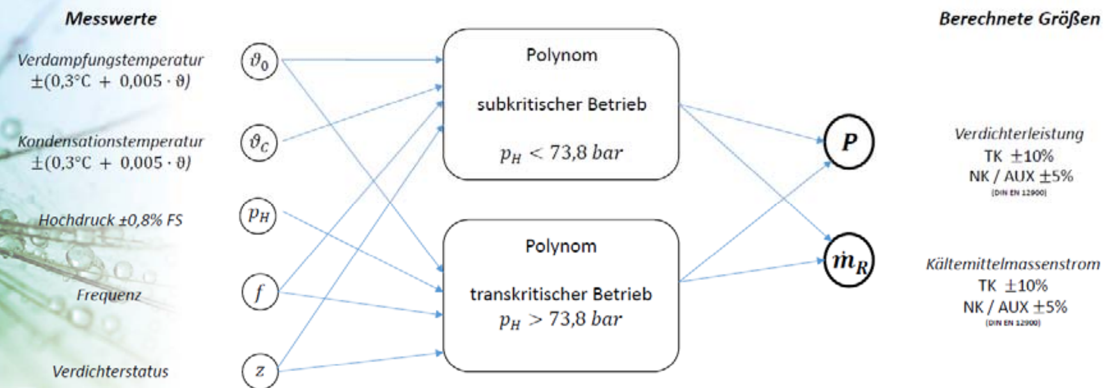
Einordnung – **Methodik** – Auswertung - Zusammenfassung

Leider verhinderte der COVID-19 Lockdown die Installation von Wattmetern an der Kälteanlage



Einordnung – **Methodik** – Auswertung - Zusammenfassung

Berechnungen der Wärmerückgewinnungs- und Gaskühler-wärmeströme:

$$\dot{Q}_i = \dot{m} \cdot (h_{i+1} - h_i)$$

NK Kälteleistung:

$$\dot{Q}_{MT} = \dot{Q}_{GC} + \dot{Q}_{HR} - \sum_i P_{i,el,compressor} - \dot{Q}_{LT}$$

Leistungszahl:

$$COP_{tot} = \frac{\dot{Q}_{MT} + \dot{Q}_{LT} + \dot{Q}_{HR}}{P_{MT} + P_{LT} + P_{AUX}}$$

Exergetischer Wirkungsgrad:

$$\eta_{tot} = \frac{\dot{Q}_{MT}\left(\frac{T_{amb}}{T_{MT,ref}} - 1\right) + \dot{Q}_{LT}\left(\frac{T_{amb}}{T_{LT,ref}} - 1\right) + \dot{Q}_{HR}\left(1 - \frac{T_{amb}}{T_{HR,ref}}\right)}{P_{MT} + P_{LT} + P_{AUX}}$$

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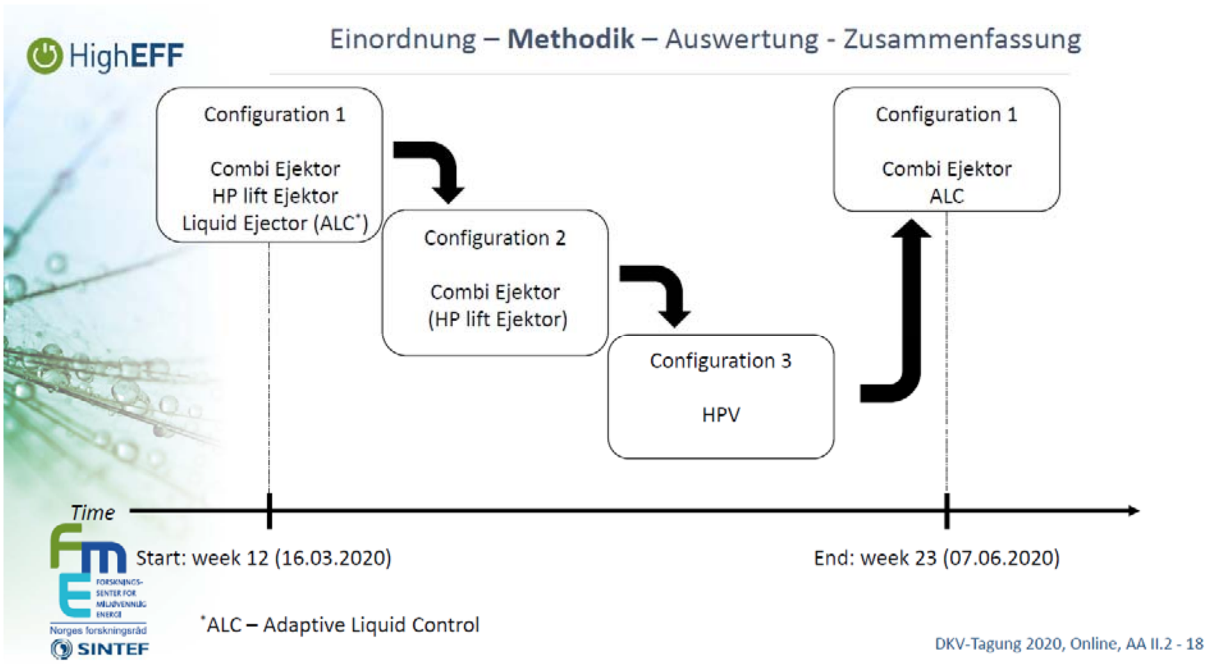
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Einordnung – Methodik – Auswertung - Zusammenfassung
Configuration 1
Combi Ejektor
HP lift Ejektor
Liquid Ejector (ALC*)
Configuration 2
Combi Ejektor
(HP lift Ejektor)
Configuration 3
HPV
Configuration 1
Combi Ejektor
ALC
Time
Start: week 12 (16.03.2020)
End: week 23 (07.06.2020)
*ALC – Adaptive Liquid Control
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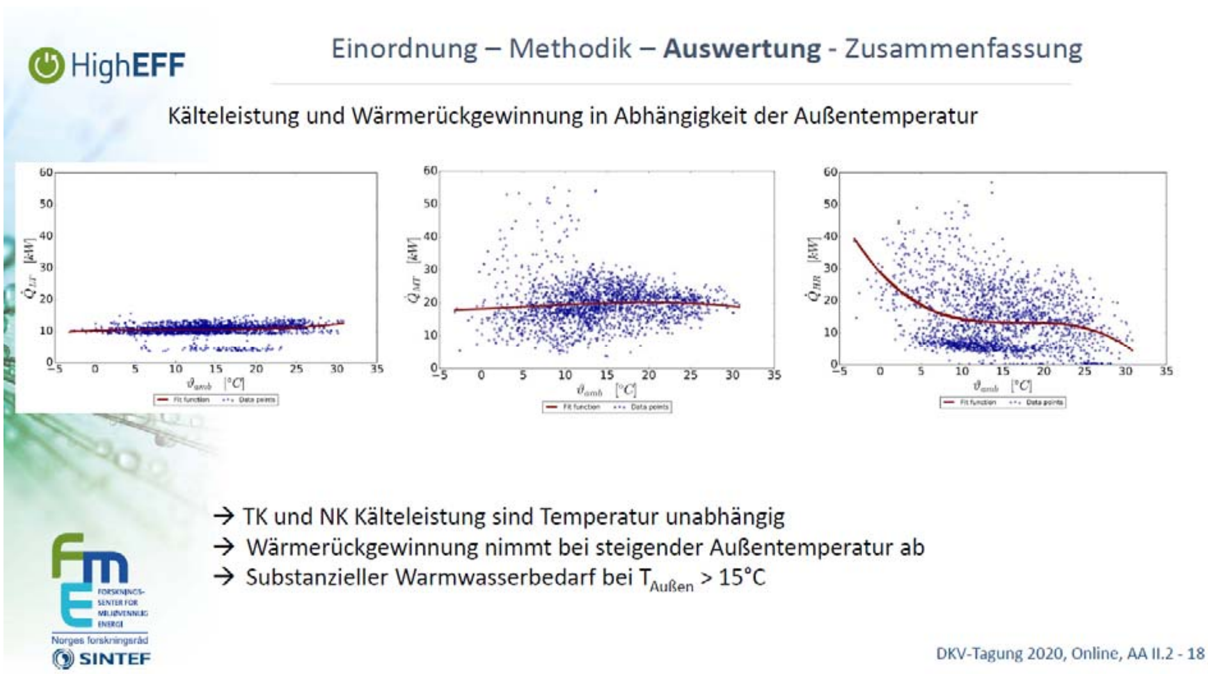
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Einordnung – Methodik – Auswertung - Zusammenfassung
Kälteleistung und Wärmerückgewinnung in Abhängigkeit der Außentemperatur
→ TK und NK Kälteleistung sind Temperatur unabhängig
→ Wärmerückgewinnung nimmt bei steigender Außentemperatur ab
→ Substanzieller Warmwasserbedarf bei T Außen > 15°C
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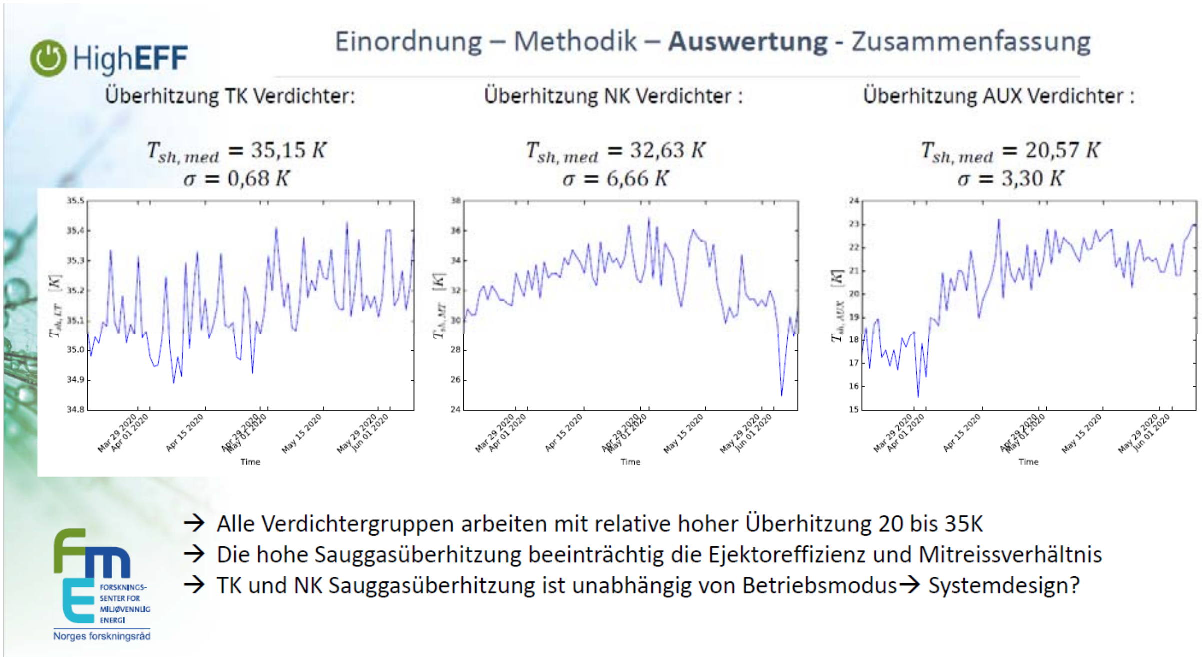
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Einordnung – Methodik – **Auswertung** - Zusammenfassung
Überhitzung TK Verdichter:
$T_{sh,\,med} = 35{,}15\ K$
$\sigma = 0{,}68\ K$
Überhitzung NK Verdichter :
$T_{sh,\,med} = 32{,}63\ K$
$\sigma = 6{,}66\ K$
Überhitzung AUX Verdichter :
$T_{sh,\,med} = 20{,}57\ K$
$\sigma = 3{,}30\ K$
Time
→ Alle Verdichtergruppen arbeiten mit relative hoher Überhitzung 20 bis 35K
→ Die hohe Sauggasüberhitzung beeinträchtig die Ejektoreffizienz und Mitreissverhältnis
→ TK und NK Sauggasüberhitzung ist unabhängig von Betriebsmodus→ Systemdesign?
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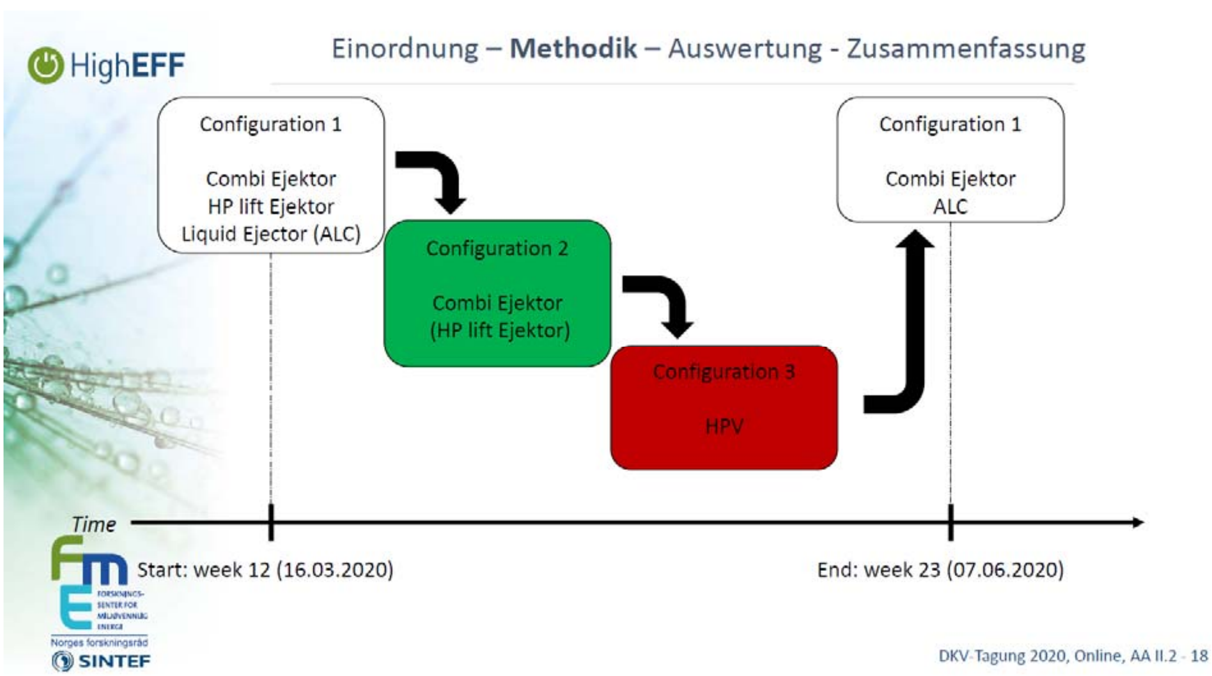
HighEFF
Einordnung – **Methodik** – Auswertung - Zusammenfassung
Configuration 1
Combi Ejektor
HP lift Ejektor
Liquid Ejector (ALC)
Configuration 2
Combi Ejektor
(HP lift Ejektor)
Configuration 3
HPV
Configuration 1
Combi Ejektor
ALC
Time
Start: week 12 (16.03.2020)
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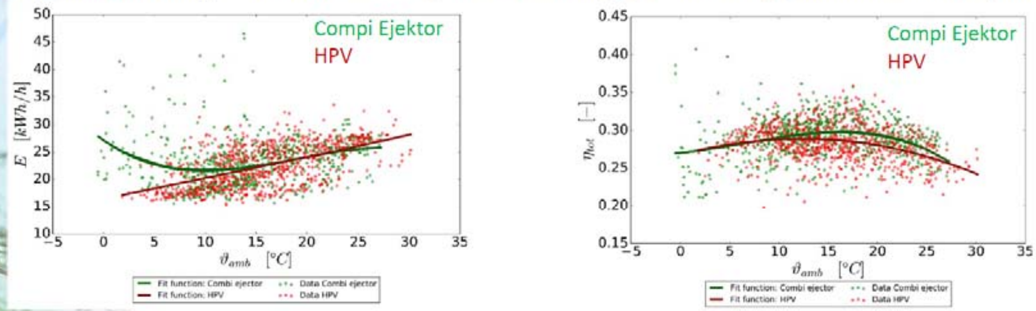
Einordnung – Methodik – **Auswertung** - Zusammenfassung

Elektrischer Energieverbrauch und exergetischer Wirkungsgrad in Abhängigkeit der Außentemperatur



→ Energieverbrauch ist vergleichbar, bei Temperaturen unter 15°C führt die Wärmerückgewinnung zu höherem Energieverbrauch

→ Exergetischer Wirkungsgrad zeigen Vorteile des Ejektorbetiebs $\vartheta_{amb} = 21°C$ +5,04%

→ Die Herausforderung in der Analyse liegt in der Tatsache begründet im Messzeitraum des Ejektorbetriebs ein hoher Anteil an Wärmerückgewinnung gedeckt werden musste

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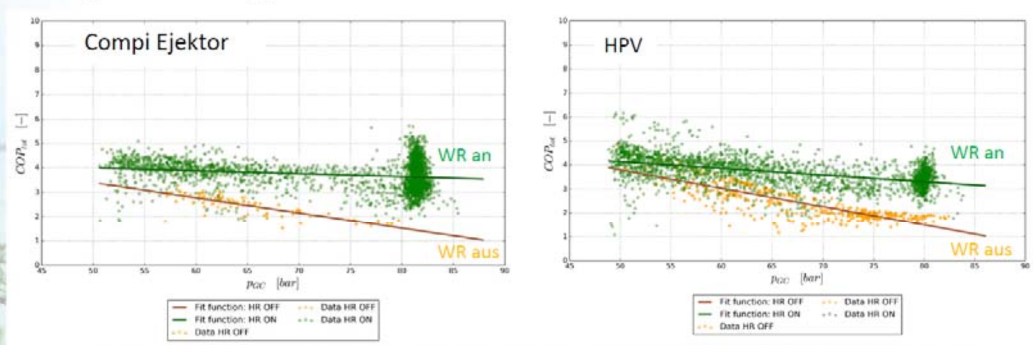
Einordnung – Methodik – **Auswertung** - Zusammenfassung

Leistungszahl in Abhängigkeit des Betriebsmodus und des Hochdrucks



→ Das Nutzerprofil des Supermarkts, respektive die Nutzung der Wärmerückgewinnung hat einen bedeutenden Einfluss aus Betriebsverhalten der Anlage

→ Die WR erhöht natürlicher Weise den el. Leistungsbedarf, folglich führt eine Betrachtung dieser in Abhängigkeit der Außentemperatur zur Fehlinterpretation

→ +10.6% höhere Leistungszahl im Wärmerückgewinnungs Betrieb bei 80 bar Hochdruck für den Ejektorbetrieb (Combi Ejektor 3.65, HPV 3.30)

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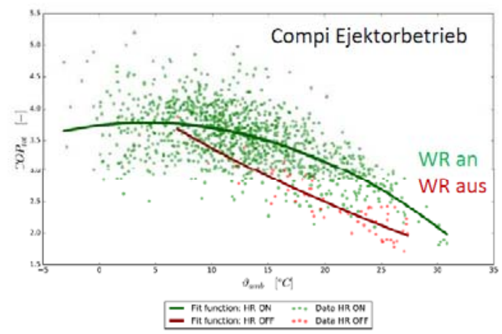
Einordnung – Methodik – **Auswertung** - Zusammenfassung

Leistungszahl in Abhängigkeit der WR und der Außentemperatur



→ Es besteht kein Zweifel daran Wärmerückgewinnung macht sinn definitiv sinn!
→ $COP_{ohne\ WR}$ 2,5 @ $\vartheta_{amb} = 20°C$
→ COP_{WR} 3,2 @ $\vartheta_{amb} = 20°C$

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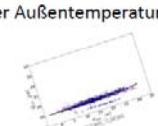
Einordnung – Methodik – Auswertung - **Zusammenfassung**

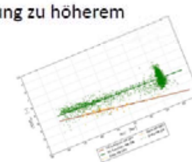
Analyse eines ejektorgestützten R744 Supermarkt-Kühlsystemes:

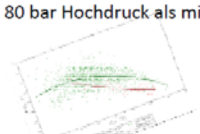
- TK + NK Kälteleistung Temperatur unabhängig, Wärmerückgewinnung nimmt bei steigender Außentemperatur ab aber substanzieller Warmwasserbedarf bei $\vartheta_{amb} > 15°C$
- Alle Verdichtergruppen arbeiten mit relative hohe Sauggasüberhitzung beeinträchtig die Ejektoreffizienz und Kälteanlagenperfromance → Systemdesign / Sauggasleitungslänge



Vergleich Ejektorbetrieb vs HPV

- Energieverbrauch ist vergleichbar, bei Temperaturen unter 15°C führt die Wärmerückgewinnung zu höherem Energieverbrauch
- Exergetischer Wirkungsgrad zeigen Vorteile des Ejektorbetiebs +5,04% bei $\vartheta_{amb} = 21°C$
- Das Nutzerprofil des Supermarkts, respektive die Nutzung der Wärmerückgewinnung hat einen bedeutenden Einfluss aus Betriebsverhalten der Anlage
- Die WR erhöht natürlicher Weise den el. Leistungsbedarf, folglich führt eine Betrachtung dieser in Abhängigkeit der Außentemperatur zur Fehlinterpretation
- Ejektorbetrieb +10.6% höhere Leistungszahl im Wärmerückgewinnungsmodus bei 80 bar Hochdruck als mit HPV (Combi Ejektor 3.65, HPV 3.30)
- Wärmerückgewinnung macht sinn definitiv sinn!
  $COP_{ohne\ WR}$ =2,5 (@ $\vartheta_{amb} = 20°C$) COP_{WR}=3,2 (@ $\vartheta_{amb} = 20°C$)





Ausblick

- Instalation von Wattmetern ist bereits abgeschlossen, Optimalisierung der Anlage, Weiterführung der Messreihe → Analyse eines Jahresverlaufes ist angestrebt

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