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Optimization potential index (OPI): An evaluation method for performance assessment and optimization potential of chillers in HVAC plants

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Abstract

Exergy analysis allows us to determine the quality of energy sources and losses due to irreversibilities through a system. However, the quantification of possible improvements as compared with the state of the art in technology is complicated. Typically, it is referred to the thermodynamic ideal, which is not achievable in practice. Therefore, this paper introduces the exergy optimization potential index, a new key figure based on exergy analysis and technical standard values, in order to assess the achievable performance and to determine possible improvements in vapor compression refrigeration plants with cold water distribution. By dividing the plant into different subsystems (dry cooler, refrigeration machine, cold storage & transport and cooling location), each of them can be assessed individually. Furthermore, by comparing the actual exergy effort with reference values, the interpretation of the results becomes straightforward. The applicability of the method is demonstrated on **two theoretical test cases and** on a real system. The investigated refrigeration plant performs well in general, **which is revealed with an average optimization potential index inferior to 0. However, the subsystem dry cooler shows potential for improvement in the period of May to mid of July. Also, three out of seven cooling locations have performance issues, which is indicated with an average optimization potential index of at least 0.07.** Overall, the electrical exergy input has a major impact on the optimization potential index. This reveals the importance of minimizing the electrical energy usage, as it is the main overhead in the operating cost of refrigeration plants.

Keywords: optimization potential evaluation method, exergy analysis, vapor compression refrigeration plants, cold water units, optimization potential index, OPI

Highlights

- A new optimization potential index (OPI) is proposed.
- Local information of subsystem optimization potential is revealed.

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- The key figure evaluation procedure is presented.
- The application is shown with **two test cases and** experimental data of a field plant.
- Simple application into red / green indicator is possible.

Nomenclature

Abbreviations

<i>COP</i>	coefficient of performance
<i>EER</i>	energy efficiency ratio
<i>HVAC</i>	heating, ventilation and air-conditioning
<i>OPI</i>	optimization potential index
<i>SIA</i>	Swiss society of engineers and architects
<i>VCRM</i>	vapor compression refrigeration machine
<i>VDMA</i>	mechanical engineering industry association

Variables

<i>B</i>	exergy [J]
\dot{B}	exergy flow rate [W]
<i>b</i>	specific exergy [J/kg]
<i>f</i>	amplification factor [-]
<i>h</i>	specific enthalpy [J/kg]
\dot{m}	mass flow rate [kg/s]
<i>p</i>	pressure [Pa]
<i>Q</i>	thermal energy [J]
\dot{Q}	heat flow rate (thermal power) [W]
<i>S</i>	entropy [J/K]
\dot{S}	entropy production [W/K]
<i>s</i>	specific entropy [J/(kg K)]
<i>T</i>	temperature [K]
<i>t</i>	time [s]
<i>V</i>	volume [m ³]
<i>W</i>	electrical energy [J]
\dot{W}	electrical power [W]

Subscripts

<i>0</i>	reference state for exergy analysis
<i>amb</i>	ambient
<i>app</i>	approach
<i>avg</i>	average
<i>C</i>	condenser
<i>Car</i>	Carnot
<i>CL</i>	cooling location
<i>CP</i>	circulating pump
<i>CPR</i>	compressor
<i>CST</i>	cold storage & transport
<i>D</i>	distribution
<i>DC</i>	dry cooler
<i>E</i>	evaporator
<i>el</i>	electrical
<i>ex</i>	exergetic
<i>gen</i>	generated
<i>in</i>	component / system input
<i>kin</i>	kinetic
<i>L</i>	loss / destroyed
<i>m</i>	mass
<i>out</i>	component / system output
<i>pot</i>	potential
<i>rev</i>	reversible
<i>RM</i>	refrigeration machine
<i>th</i>	thermal
<i>w</i>	work

Superscripts

*	reference value according to technical standards
<i>acc</i>	acceptable
<i>adq</i>	adequate

Greek letters

ΔT	temperature difference [K]
η	efficiency [-]

1. Introduction

A large amount of the energy consumption in buildings is due to the heating, ventilation and air conditioning (HVAC) systems, where one key component is the refrigeration machine. For large cooling loads, mostly vapor compression refrigeration machines (VCRM) with cold water distribution are deployed to generate, store and transport the cold. Therefore, HVAC systems and VCRM are subject of investigations, in order to increase their effectiveness and to reduce the primary energy demand [1, 2].

Typically, energy analyses (or first law analyses) are carried out and the respective energy efficiency is determined. The latter describes the ratio of a benefit to a certain effort, in order to assess the performance of the thermodynamic system. In the case of VCRM, the coefficient of performance (COP) and energy efficiency ratio (EER) are widely used as indicators of efficiency [3, 4].

In addition, exergy analyses (or second law analyses) were carried out for various applications in the built environment, such as the building envelope [5, 6], photovoltaics [7, 8], HVAC systems [9, 10], heat pumps [11, 12] and refrigeration machines [13, 14]. Exergy is defined as the unrestricted convertible part of energy which can be obtained from any form of energy by reversible interaction with a defined reference environment [15]. Therefore, exergy can be seen as a measure of energy quality and is the counterpart to the unusable energy portion, the anergy. Furthermore, exergy is a non conservative quantity and is destroyed due to irreversibilities in thermodynamic processes, e.g. heat exchange, which represents a reduction in obtainable work or thermal potential. Exergy losses are related to the entropy production [16] and exergy analysis thus associates energy principles with the second law of thermodynamics. Therefore, such an analysis highlights different aspects of the energy utilization, which are not visible in purely energetic considerations. In exergy analysis the quality of different energy flows and the respective losses throughout a system can be determined on a detailed level. By applying the exergy method, the energetic potential of different energy sources is comparable and a relative comparison between refrigeration plants or arbitrary systems is feasible. When assessing refrigeration plants, the exergy flows can be followed through the whole system from the input to the output. Thus, the location as well as the magnitude of the losses can be identified in every subsystem. Furthermore, by considering the exergy input of auxiliary devices, such as circulating pumps, the configuration of hydraulic networks in the refrigeration plant can be evaluated. An unfavorable design of the hydraulic circuit results in an elevated electrical exergy input of circulating pumps, as the associated losses in the piping, due to wall friction, are increased. The latter plays an important role in practice for the hydraulic integration of the refrigeration machine in the plant. Thus, the exergy analysis is also useful when it comes to the subject of system improvement.

In order to increase the system effectiveness, tools and procedures were being developed to examine the energy usage and to determine optimization potentials. Field tests were carried out, in order to develop a standardized methodology to systematically monitor refrigeration systems and their efficiency [17]. The study investigated the implementation of the energy efficiency rating approach of the standards VDMA 24247-2 [18] and VDMA 24247-7 [19]. Three different subsystems are suggested for the assessment: refrigeration

machine, the refrigeration plant (including refrigeration machine and dry cooler) and total refrigeration system (including refrigeration machine, dry cooler, hydraulics including storage and cold utilization). Furthermore a total coefficient of performance (TCOP) and a total energy performance factor (TEPF) are defined with respect to the chosen system boundary. Additionally, to allow a differentiated assessment of the refrigeration machine, five efficiency key figures are proposed in the standards. Four of them to describe each a different component (cold production, heat transport, fluid transport and cold utilization efficiency) and one to describe the whole system (energy efficiency level). As a result, an energy consumption monitoring was possible in most of the cases and potentials for optimization as well as efficiencies at different operating conditions were identified. However, difficulties occurred due to the measuring concepts of some field plants, as not all needed variables were available for monitoring. Thus, the implementation of the energy efficiency assessment for existing systems was difficult or impossible with the pursued method. A model-based key performance indicator (KPI) was introduced to monitor telecommunication cooling systems and to suggest improvements if the cooling efficiencies are not met [20]. The KPI compares the actual energy efficiency index with a reference one, where the latter represents the target performance of a specific application. The approach to split the investigated system into subsystems was also applied for a chilled water system of a campus [21]. It was divided into four subsystems (chilled water distribution, refrigeration, steam turbine drive and heat rejection), in order to carry out an energy-, exergy- and cost analysis of the plant. Fang *et al.* evaluated the operation performance of an HVAC system by partitioning it into three subsystems (cooling water, chilled water and air handling unit) and by applying ideal exergy flow models [22]. By minimizing the exergy destruction in the subsystems, the ideal operation level was defined. Five different control strategies were assessed, where the best strategy yields a performance improvement of 11.9 % compared to the most unfavorable operating condition. Menberg *et al.* carried out a comprehensive exergy analysis of a ground-source heat pump in cooling and heating mode, where subsystems were chosen accordingly for a detailed evaluation of the plant (e.g. down to the detail of mixing valves) [23]. The investigation discovered different exergy losses in the components with respect to the operating mode. The exergy performance in heating mode is mentioned to be twice as high as in cooling operation. The authors state the need of additional research regarding variable boundary conditions as well as the study of alternative reference environment definitions, in order to allow an analysis under real operation conditions. In another study, the framework of an international collaboration regarding low exergy systems for buildings and community systems is presented [24]. The use of exergy analysis is emphasized to capture all the relevant aspects of the energy usage and to decrease further the exergy demand of buildings as well as reducing the primary energy demand. To optimize the energy usage and therefore reducing the exergy losses, the exergy path was followed within the whole energy chain from the conversion of primary energy up to the building envelope through different subsystems. The application of measurement concepts for efficiency investigations of refrigeration plants with an exergy based approach was investigated by Eisenhauer *et al.* [25]. A complete exergy analysis of refrigeration plants requires a large number of measured variables. In order to reduce their number, the authors suggest to extend the system boundaries, with the drawback of decreasing the level of detail, or to use thermodynamic relations and assumptions to determine the missing quantities. Further, the

authors state that the method from the standard VDMA 24247-2 is well suited for a relative comparison of similar refrigeration plants, but reaches its limitation when a deviation from the reference process is present. Moreover, the determination of typical reference values for system components, which are necessary for a comparison, is difficult.

Accordingly, research is needed in the development of efficiency assessment systems which are consistently applicable with the most common measuring concepts in refrigeration plants. On the other hand, the determination of typical reference values is required in order to allow an absolute comparison of different refrigeration plants in real operating conditions. Together with the findings highlighted above, a novel and general applicable method to determine the performance and the related optimization potential regarding the state of the art in technology of vapor compression refrigeration plants with cold water distribution is developed. The present work focuses on the whole refrigeration plant including the secondary hydraulic circuits present in the system (no devices with direct cooling and condensation). In order to exemplify this method, a comprehensive exergy analysis is carried out, where the exergy flow is examined from the input to the output. The refrigeration plant is split into reasonable subsystems, which leads to a reduction of the required measurement variables by still ensuring a sufficient level of detail. The exergy efficiency is assessed at various levels depending on the functionality investigated (e.g. cold transport), allowing a differentiated insight into the behavior of the refrigeration plant. Furthermore, the optimization potential index (OPI) is introduced, which is based on the actual and an achievable reference exergy efficiency with technical standards as baseline, revealing the improvement potential in each subsystem at a glance. With the definition of these reference values, an absolute comparison between different refrigeration plants independently of the system structure is achieved, resulting in a widely applicable assessment. Moreover, the interpretation of the results becomes simpler and may be carried out by laypersons who can, if needed, initiate corrective measures by specialists. In order to demonstrate the functionality of the proposed method, the rating system is **applied on two theoretical test cases as well as on a case study, while for the latter, experimental data gained from a field installation is used.**

2. Exergy principles

Similarly to energy, exergy transfer is realised by heat, work and mass flow. By applying an exergy balance over a control volume, e.g. a complete system or one single component, the rate of change of exergy $\frac{dB}{dt}$ can generally be expressed as [26]:

$$\frac{dB}{dt} = \underbrace{\sum_i \left(1 - \frac{T_0}{T_i}\right) \dot{Q}_i}_{\dot{B}_{th}} + \underbrace{\left(\dot{W} - p_0 \frac{dV}{dt}\right)}_{\dot{B}_w} + \underbrace{\sum_j \dot{m}_{in,j} b_{in,j} - \sum_k \dot{m}_{out,k} b_{out,k}}_{\dot{B}_m} - \dot{B}_L \quad (1)$$

where \dot{B}_{th} represents the thermal exergy transfer rate by a heat flow rate \dot{Q}_i at the temperature level T_i with respect to a reference temperature T_0 , which is discussed separately in section [2.1]. \dot{B}_w denotes the exergy flow rate by actual mechanical or electrical power \dot{W} and surroundings work, where p_0 is the reference pressure and $\frac{dV}{dt}$ the rate of change of the control volume. Electrical, as well as mechanical, energy is high quality energy

and by definition pure exergy, while heat is low quality energy and therefore only contains a certain amount of exergy. By heat transfer processes, e.g. heat conduction through a wall, the work potential of the heat, and therefore the exergy, decreases due to irreversibilities. According to the second law of thermodynamics, any irreversible process produces entropy due to losses, e.g. friction. The so called destroyed exergy \dot{B}_L quantifies the loss in work potential and is related to the generated entropy at a constant reference temperature [16, 27]:

$$\dot{B}_L = T_0 \dot{S}_{gen} \geq 0 \quad (2)$$

where \dot{S}_{gen} represents the entropy generated in the process. Any real process generates entropy and therefore exergy is destroyed. The term \dot{B}_m in Eq. 1 indicates the exergy transfer rate in and out of the system by mass flow, where \dot{m}_{in} and \dot{m}_{out} are the incoming and outgoing mass flow rate, respectively. b_{in} and b_{out} denote the associated specific flow exergies, which can be described as [26]:

$$b = h - h_0 - T_0(s - s_0) + b_{kin} + b_{pot} \quad (3)$$

where h and s is the specific enthalpy and entropy, respectively. h_0 and s_0 represent the reference values at the reference environment with temperature T_0 and pressure p_0 . b_{kin} denotes the exergy of kinetic energy and b_{pot} the exergy of potential energy. The thermal exergy flow rate has the property of changing its flow direction with respect to the heat flow rate, depending on the system temperature. If $T_i > T_0$, the Carnot factor [28]:

$$\eta_{Car} = \left(1 - \frac{T_0}{T_i}\right) \quad (4)$$

is positive and the exergy flow rate has therefore the same direction as the heat flow rate. Conversely, the exergy flow rate has the opposite direction of the heat flow rate if $T_i < T_0$, as the Carnot factor is negative. In other words, thermal exergy is always transferred towards the reference environment. Regarding the thermal exergy, Shukuya introduced the concept of warm and cool exergy, depending on whether the system temperature is higher or lower than the reference [29].

2.1. Definition of the reference environment

In exergy analysis, the definition of the reference environment is crucial in order to carry out reliable evaluations and must be selected depending on the investigated system. Torío *et al.* examined different reference environments for buildings and their energy supply systems: the universe, the indoor air inside the building, the undisturbed ground and the ambient air surrounding the building, with the recommendation to use the current surrounding outdoor air as the reference state [30]. Furthermore, Pons demonstrated that the reference state temperature should be fixed and constant also for dynamic analysis [31]. Only if the reference temperature T_0 is constant (e.g. in Eq. 3), then exergy is a function of state which results from a linear combination of energy and entropy. Combining both findings, the averaged outdoor temperature T_{avg} is used over each assessment interval in the present work as reference temperature T_0 , defined as:

$$T_0 \hat{=} T_{avg} = \frac{1}{n} \sum_{t=0h}^{24h} T_{amb} \quad (5)$$

where T_{amb} represents the current surrounding outdoor air temperature and n the number of summands resulting from the present measurement interval (see section 4.2 for details). The same reference temperature is used for all investigated subsystem (see section 3 and 4). The approach of using averaged outdoor temperatures has been applied in other studies, e.g. for the analysis of a building [32] or air cooling systems in buildings [33].

3. Refrigeration plant overview

The basic form of a refrigeration plant with cold water distribution is depicted schematically in Fig. 1. The cold is generated by a vapor compression refrigeration cycle (see Fig. 1, subsystem refrigeration machine, RM), which cools down the secondary side liquid, consisting typically of a water-glycol mixture. The cold is stored and transported (see Fig. 1, subsystem cold storage & transport, CST) to the cooling location (see Fig. 1, subsystem cooling location, CL), e.g. to an air handling unit (AHU), while consuming electrical energy

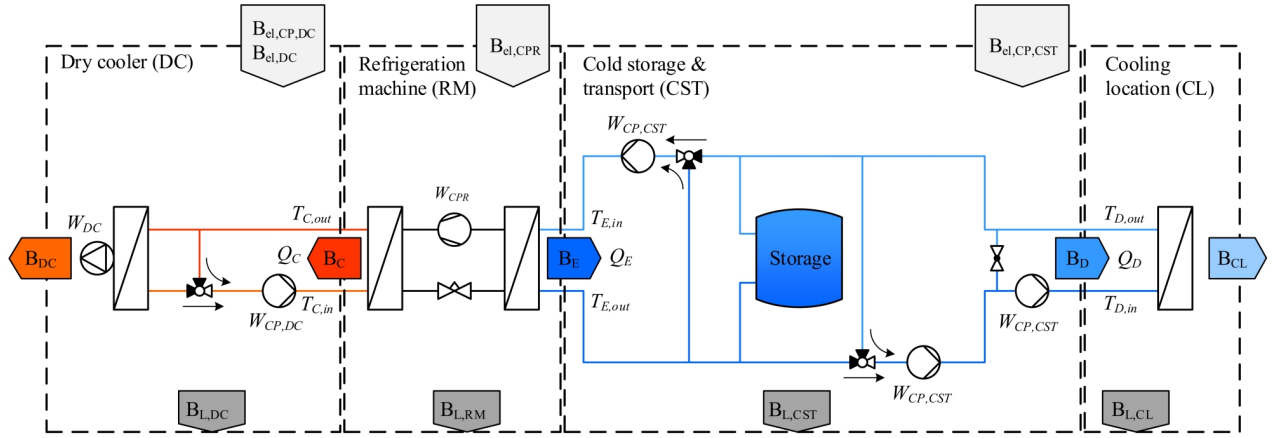


Figure 1: Generalized schematic of a typical vapor compression refrigeration system with cold water distribution and its subsystems. Arrows indicate the exergy inputs and outputs of each subsystem and variables in italic indicate measured quantities (see Table 1 and 2 for details). The blue and red color depicts the cold and hot side of the plant, respectively.

to drive the circulating pumps. The cooled room itself and the building envelope are not considered in the present work. To remove the heat of the chiller condenser, another secondary cycle with a dry cooler and circulating pump is present (see Fig. 1, subsystem dry cooler, DC). The hydraulic circuit in the diagram is not intended to represent the optimum solution, but to illustrate the assignment of pumps, valves, etc. to the subsystems. The division is based on considerations regarding the formulation of performance key figures and the study of various piping & instrumentation diagrams of real systems. Additionally, in some plants, subsystems for heat utilization and free cooling are present (not shown in the illustration and not discussed in the present paper).

Furthermore, Fig. 1 shows the measured thermodynamic quantities in every subsystem in order to determine the corresponding exergy in- and outputs, which are indicated with arrows (see Table 1 and 2 for details). The B_L arrows indicate the destroyed exergy of the respective subsystems. The exergy input is generated on

the one hand by electrical energy of the compressor, circulating pumps as well as dry cooler fans and on the other hand by thermal and flow exergy exchange over the subsystem boundaries. The corresponding exergy quantities as well as key figures are calculated according to section 4. In larger plants multiple chillers and

Table 1: Measured variables in each subsystem for the exergy computation according to Fig. 1

Subsystem	Measured variables
DC	Condenser thermal energy Q_C
	Condenser inlet temperature $T_{C,in}$
	Condenser outlet temperature $T_{C,out}$
	Cirulating pump electrical energy $W_{CP,DC}$
	Dry cooler fan electrical energy W_{DC}
RM	Compressor electrical energy W_{CPR}
CST	Evaporator thermal energy Q_E
	Evaporator inlet temperature $T_{E,in}$
	Evaporator outlet temperature $T_{E,out}$
	Cirulating pump electrical energy $W_{CP,CST}$
CL	Cold distribution thermal energy Q_D
	Cold distribution inlet temperature $T_{D,in}$
	Cold distribution outlet temperature $T_{D,out}$

Table 2: Exergy in- and outputs of the subsystems according to Fig. 1

Subsystem	Exergy input	Exergy output
DC	Cirulating pump exergy $B_{el,CP,DC}$	Dry cooler thermal exergy B_{DC}
	Dry cooler fan electrical exergy $B_{el,DC}$	
	Condenser exergy B_C	
RM	Compressor exergy $B_{el,CPR}$	Condenser exergy B_C
		Evaporator exergy B_E
CST	Cirulating pump exergy $B_{el,CP,CST}$	Cold distribution exergy B_D
	Evaporator exergy B_E	
CL	Cold distribution exergy B_D	Cooling location exergy B_{CL}

cooling locations are often integrated in parallel. Accordingly, several refrigeration machine (RM) and cooling location (CL) subsystems may exist, where each of them is evaluated individually. As a rule, the dry cooler (DC) and cold storage & transport (CST) subsystem occur only once per refrigeration plant, while they absorb the exergies of all RM and feed all CL subsystems. Any mixing circuits (T-connections, 3-way valves, etc.) and auxiliary devices are assigned to the DC or CST subsystem. When defining the subsystem boundaries in

the layout of a real installation, it must be ensured that all needed quantities to calculate the exergy flows are measured.

3.1. Theoretical test cases

Two theoretical test cases with a refrigeration plant as shown in Fig. 1 are analyzed in the present study. The first case represents an adequate operating refrigeration system, while the second simulates a faulty operation due to a fouled condenser. Therefore, a larger temperature difference between the heat exchanger in- and outlet as well as an increased electrical energy consumption of the circulating pumps in the subsystem DC is assumed. For both test cases, a steady-state operation of the refrigeration system with daily values according to Table 3 is considered.

Table 3: Defined daily values of temperatures, thermal and electrical energies for both evaluated test cases in each subsystem.

Subsystem	Variables	Values test case 1 (adequate operation)	Values test case 2 (faulty operation)
DC	Q_C	2246.4 MJ	2246.4 MJ
	$T_{C,in}$	27 °C (300.15 K)	29 °C (302.15 K)
	$T_{C,out}$	29 °C (302.15 K)	35 °C (308.15 K)
	$W_{CP,DC}$	22.5 MJ	49.9 MJ
	W_{DC}	59.1 MJ	59.1 MJ
RM	W_{CPR}	518.4 MJ	518.4 MJ
CST	Q_E	1728.0 MJ	1728.0 MJ
	$T_{E,in}$	19 °C (292.15 K)	19 °C (292.15 K)
	$T_{E,out}$	12 °C (285.15 K)	12 °C (285.15 K)
	$W_{CP,CST}$	21.6 MJ	21.6 MJ
CL	Q_D	1555.2 MJ	1555.2 MJ
	$T_{D,in}$	13 °C (286.15 K)	13 °C (286.15 K)
	$T_{D,out}$	17 °C (290.15 K)	17 °C (290.15 K)
all	T_0	22 °C (295.15 K)	22 °C (295.15 K)

3.2. Examined field plant

Additionally to the test cases, an existing refrigeration plant located in the city of Winterthur, Switzerland is investigated in the present work as a case study. The field plant consists of five parallel refrigeration machines with 950 kW cooling power each and ammonia as refrigerant. The machines and the distribution networks are located underground, where the hydraulic circuit supplies seven different buildings with cold. The main application is for office space cooling. Furthermore, three rooftop dry coolers with a nominal power of 2000 kW and 11 circulating pumps with a nominal volume flow rate ranging from 69.5 to 485 $\frac{\text{m}^3}{\text{h}}$ are present.

Moreover, two cold storages with a capacity of 3.5 m³ are integrated to the refrigeration system. Accordingly, the refrigeration plant is split into 14 different subsystems. For the present analysis, measurements from May to August 2018 are evaluated.

4. Performance and optimization potential evaluation approach

4.1. Performance key figures

One of the most used performance indicators for energy systems is the efficiency or first law efficiency when referring to a thermodynamic process. It generally describes the ratio between the useful and supplied energy or power and is typically denoted coefficient of performance *COP* for refrigeration machines [34]:

$$COP = \frac{\dot{Q}_E}{\dot{W}_{CPR}} \quad (6)$$

where \dot{Q}_E denotes the cooling capacity of the evaporator and \dot{W}_{CPR} the power input of the compressor. The *COP* is typically larger than 1. To account for losses due to irreversibilities, in order to highlight different aspects of the energy usage in the system and to determine optimization potentials, the exergy or second law efficiency is used. For refrigeration cycles, the exergy efficiency η_{ex} is defined as:

$$\eta_{ex} = \frac{|\dot{B}_E| + |\dot{B}_C|}{\dot{W}_{CPR}} = 1 - \frac{\dot{B}_L}{\dot{W}_{CPR}} \quad (7)$$

which relates the thermal exergy output of the condenser \dot{B}_C and the evaporator \dot{B}_E to the compressor input. If the condenser heat is ejected directly to the environment and thus the exergy is destroyed, the exergy efficiency is defined as follows [28, 34]:

$$\eta_{ex} = \frac{|\dot{B}_E|}{\dot{W}_{CPR}} = \frac{\dot{W}_{CPR,rev}}{\dot{W}_{CPR}} = 1 - \frac{\dot{B}_L}{\dot{W}_{CPR}} \quad (8)$$

which relates the ideal power input $\dot{W}_{CPR,rev}$ of a reversibly working machine (no losses) to the actual compressor effort, meaning that a reversibly operating refrigeration machine needs to supply at least the same amount of work as the cooling load exergy of the evaporator. In a more general form, by simply balancing exergy output to input, e.g. for a cold storage where no work is supplied or consumed, the exergy efficiency, also denoted exergy grade, is written as [15]:

$$\eta_{ex} = \frac{\dot{B}_{out}}{\dot{B}_{in}} = 1 - \frac{\dot{B}_L}{\dot{B}_{in}} \quad (9)$$

where \dot{B}_{in} and \dot{B}_{out} represent the total exergy in- and output of the system, respectively. The exergy efficiency yields 1 when assessing a reversible process and 0 if the energy is completely degraded. Therefore, it delivers a realistic estimation of the system performance, meaning the ability to produce a high quality energy output with a certain exergy input in a defined process, and thus giving insight to the energy utilization.

4.2. Optimization potential index

However, an absolute quantification of the performance is problematic, as the exergy efficiency is referred to a thermodynamic ideal which is not achievable in reality. Assuming two refrigeration plants A and B with $\eta_{ex,A} = 0.2$ and $\eta_{ex,B} = 0.3$, it is obvious that plant B is more efficient than plant A. Nevertheless, there is no evidence that plant B possesses further optimization potential comparing to the state of the art in technology, which is of greater importance in practice. This lets us introduce the optimization potential index OPI:

$$OPI = 1 - \frac{\eta_{ex}}{\eta_{ex}^*} = 1 - \frac{\frac{\dot{B}_{out}}{\dot{B}_{in}}}{\frac{\dot{B}_{out}^*}{\dot{B}_{in}^*}} = 1 - \frac{\dot{B}_{in}^*}{\dot{B}_{in}} \quad (10)$$

where η_{ex} represents the actual exergy efficiency determined with measurements and η_{ex}^* a reference or baseline exergy efficiency defined by technical standards and regulations. \dot{B}_{in}^* describes the reference exergy input to the system determined by technical norms. The optimization potential evaluation is therefore based on the effective compared to the technically optimal effort, while the same benefit is achieved. The evaluation of a subsystem is carried out on the assumption that the adjacent subsystems perform in the same way. Thus, the optimization potential index indicates how the real system would behave compared to the reference system in exactly the same situation. In the present work, all reference exergies which are not solely computed with measured quantities but calculated with reference values from technical standards, are marked with an asterisk. By using the optimization potential index to assess a refrigeration system, the interpretation of the results becomes simpler and the significance is greater in practice, as the reference point can vary depending on technological conditions. $OPI \approx 0$ is achieved with good engineering, the technical requirements are exceeded for an optimization potential index inferior to 0 and a value significantly superior to 0 indicates potential for improvement (Fig. 2a). Thus the performance and optimization potential of the refrigeration system is visible at a glance. Making an example: assuming the OPI of every subsystem is below 0 and therefore, the refrigeration plant operates according to the technical requirements. Then, due to a malfunction of a circulating pump in the subsystem CST, the energy consumption is increased. The issue is revealed with an increased value of the OPI in the mentioned subsystem and points out an optimization potential. The problem can be identified by plant operators by tracking the key figure over time, e.g. with a daily check. Therefore, the OPI delivers a first localization of the problem with respect to the chosen subsystems. If the location of the malfunction is known, a more detailed study may be carried out in order to determine the faulty component and to initiate corrective measures to increase the performance of the refrigeration plant.

The systematic approach to determine the OPI is based on a balance of the exergy flows over the subsystem boundaries. For practical reasons, it is not always possible to measure all the needed quantities of the subsystems to determine η_{ex} and η_{ex}^* , which is the case for the subsystems DC and CL. In these cases, the OPI is evaluated directly with the exergy inputs, which are either computed from measurements, technical standards or derived with an exergy balance (Eq. 1 and 3). The electrical power input of the compressor and auxiliary units is included as exergy input, which is both consistent by deriving the exergetic efficiency from the exergy balance and plausible from a functional point of view. The electrical power consumption of the

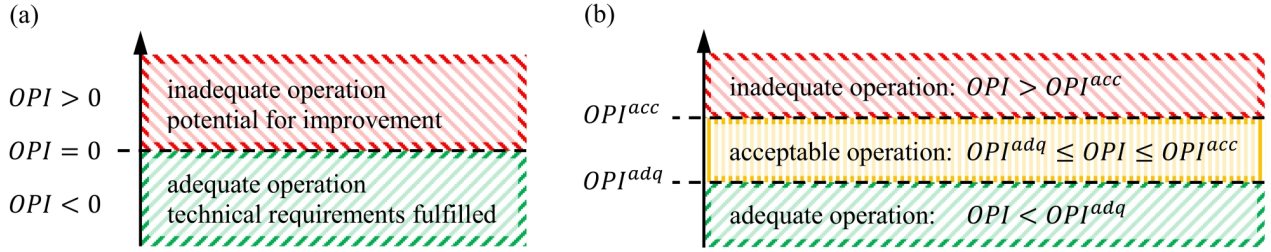


Figure 2: Optimization potential index scale for determining the operating condition of the refrigeration system and revealing improvement capabilities with (a) a basic and (b) a detailed assessment.

circulating pumps, for example, is a necessary expense for the functioning of the subsystem. The exergy input is needed in order to push the fluid through the piping in the secondary hydraulic cycle while overcoming the wall friction. Furthermore, a portion of the supplied electrical energy is converted into heat, which increases the temperature of the fluid. Both processes are associated with exergy losses. Consequently, a system with a well designed hydraulic circuit requires a lower exergy input of the circulating pumps. The exergy input of auxiliary devices was also included in other studies [23, 35]. The same applies to the dry coolers, as they allow an increased heat transfer by forced convection. The exergy efficiency of the dry cooler is maximized if the optimum between fan power and heat transfer is achieved.

For the analysis of a refrigeration plant, data from a steady operating phase is usually averaged and evaluated. However, the latter is problematic as the ambient air temperature fluctuates over the day and possibly too few stationary phases over the evaluation period are present which match the selection criteria. With a dynamic approach, additional difficulties arise in the evaluation of the instationary exergy balance. Therefore, a quasi-stationary approach is proposed in the present work, where steady-state is assumed over the measurement interval. Thus, the exergy flow rate is numerically integrated by using the trapezoidal rule to calculate the exergy of each measurement interval. As the cooling load in refrigeration plants usually follows a daily rhythm, it is proposed to evaluate the key figure on a daily basis. Consequently, the optimization potential index is formed with the sum of the calculated exergy inputs of every measurement interval over the day:

$$OPI = 1 - \frac{\sum_{t=0}^{24h} B_{in}^*}{\sum_{t=0}^{24h} B_{in}} \quad (11)$$

The number of summands varies depending on the measurement interval. The field plant data available is recorded at an interval of 5 minutes which results in 288 summands for the daily assessment in the present evaluation. The response time of the investigated system is less than 10 minutes and therefore the measurement interval is insufficiently short to correctly describe the start up and shut down behavior of the refrigeration machines. However, this error is assumed to be negligible small regarding the key figure assessment interval of 24 hours.

4.2.1. Subsystem dry cooler

The optimization potential index of the subsystem DC (see [Fig. 1](#)) is:

$$OPI_{DC} = 1 - \frac{\sum_{t=0}^{24h} \left[\sum_i B_{C,i}^* + \sum_j B_{el,CP,DC,j}^* + \sum_k B_{el,DC,k}^* \right]}{\sum_{t=0}^{24h} \left[\sum_i B_{C,i} + \sum_j B_{el,CP,DC,j} + \sum_k B_{el,DC,k} \right]} \quad (12)$$

where the exergy input is summed up over all present refrigeration machines and auxiliary devices, which feed the subsystem. The condenser exergy B_C is given by:

$$B_C = Q_C \left(1 - \frac{T_0}{T_C} \right) \quad (13)$$

with the condenser thermal energy Q_C and the logarithmic mean temperature of the condenser T_C expressed by the following equation:

$$T_C = \frac{T_{C,in} - T_{C,out}}{\ln \left(\frac{T_{C,in}}{T_{C,out}} \right)} \quad (14)$$

where $T_{C,in}$ and $T_{C,out}$ are the secondary side condenser in- and outlet temperatures, respectively. Electrical energy is per definition pure exergy. Therefore, the exergy input of circulating pumps $B_{el,CP,DC}$ is defined as follows:

$$B_{el,CP,DC} = W_{CP,DC} \quad (15)$$

where $W_{CP,DC}$ represents the circulating pump electrical energy. Similarly, the exergy input of dry cooler fans $B_{el,DC}$ corresponds to:

$$B_{el,DC} = W_{DC} \quad (16)$$

with W_{DC} , the respective electrical energy of the dry cooler fans. The reference condenser exergy B_C^* is expressed by the following equation:

$$B_C^* = Q_C \left(1 - \frac{T_0}{T_C^*} \right) \quad (17)$$

where T_C^* describes the reference temperature of the condenser. The following relation is introduced:

$$T_C^* = T_0 + \Delta T_{app} + \frac{T_{C,out} - T_{C,in}}{2} \quad (18)$$

where ΔT_{app} represents the temperature difference in the dry cooler heat exchanger, meaning the heat transfer medium outlet to ambient air inlet temperature difference. A target value of the latter is 6 K for dry coolers as reported by the technical standard VDMA 24247-8 [\[36\]](#). The necessary electrical energy of the auxiliary devices should not exceed a certain percentage of the total condenser thermal energy, which is defined by the electro-thermo amplification factors according to the technical standard SIA 382/1 [\[37\]](#) listed in [Table 4](#). It is proposed to calculate the reference exergy input of circulating pumps $B_{el,CP,DC}^*$ according to:

$$B_{el,CP,DC}^* = \frac{1}{f_{el,th,CP,DC}} \sum_i Q_{C,i} \quad (19)$$

where $f_{el,th,CP,DC}$ represents the electro-thermo amplification factor according to [Table 4](#). Correspondingly, it is proposed to compute the reference exergy input of dry cooler fans with the following equation:

$$B_{el,DC}^* = \frac{1}{f_{el,th,DC}} \sum_i Q_{C,i} \quad (20)$$

where $f_{el,th,DC}$ is the associated electro-thermo amplification factor for dry cooler fans according to [Table 4](#).

Table 4: Electro-thermo amplification factors in each subsystem for dry cooler fans and circulating pumps according to standard SIA 382/1 [37](#).

Subsystem	Electro-thermo amplification factor		Threshold value
DC	Dry cooler fan	$f_{el,th,DC}$	≥ 28 ($\approx 3.5\%$ of Q_C)
	Circulating pump	$f_{el,th,CP,DC}$	≥ 85 ($\approx 1.2\%$ of Q_C)
CST	Circulating pump	$f_{el,th,CP,CST}$	≥ 65 ($\approx 1.5\%$ of Q_E)

4.2.2. Subsystem refrigeration machine

The optimization potential index proposed for the subsystem RM (see [Fig. 1](#)) is:

$$OPI_{RM} = 1 - \frac{\sum_{t=0}^{24h} B_{el,CPR}^*}{\sum_{t=0}^{24h} B_{el,CPR}} \quad (21)$$

where the exergy input of the compressor $Ex_{el,CPR}$ is given by:

$$B_{el,CPR} = W_{CPR} \quad (22)$$

with the compressor electrical energy W_{CPR} . The reference exergy input of the compressor $B_{el,CPR}^*$ is computed with an exergy balance over the refrigeration machine according to:

$$B_{el,CPR}^* = B_E + B_C^* + B_{L,RM} \quad (23)$$

where B_E is the actual exergy output of the evaporator and B_C^* the reference exergy output of the condenser according to [Eq. 17](#) (reference situation of the subsystem DC). The destroyed exergy $B_{L,RM}$ is assumed to be identical as in the actual situation, which results in a stricter assessment and is computed according to:

$$B_{L,RM} = B_{el,CPR} - B_E - B_C \quad (24)$$

If measurements and thermodynamic properties of the refrigeration cycle were available, a more detailed determination of the destroyed exergy could be carried out with a thermodynamic model of the cycle process. However, the analysis should then be carried out specifically for the plant under investigation and is not applicable generally. As electrical energy is pure exergy, OPI_{RM} may be written as the ratio of the actual COP to a reference coefficient of performance COP^* according to the following equation:

$$OPI_{RM} = 1 - \frac{COP}{COP^*} = 1 - \frac{\frac{Q_E}{W_{CPR}}}{\frac{Q_E}{W_{CPR}^*}} = 1 - \frac{W_{CPR}^*}{W_{CPR}} \quad (25)$$

where Q_E represents the evaporator thermal energy, W_{CPR} the actual and W_{CPR}^* the reference compressor electrical energy.

4.2.3. Subsystem cooling location

The optimization potential index of the subsystem CL (see [Fig. 1](#)) is:

$$OPI_{CL} = 1 - \frac{\sum_{t=0h}^{24h} B_D^*}{\sum_{t=0h}^{24h} B_D} \quad (26)$$

where the cold distribution exergy B_D is calculated according to:

$$B_D = Q_D \left(1 - \frac{T_0}{T_D} \right) \quad (27)$$

with the cold distribution thermal energy Q_D and the logarithmic mean temperature of the cold distribution T_D expressed by the following equation:

$$T_D = \frac{T_{D,in} - T_{D,out}}{\ln \left(\frac{T_{D,in}}{T_{D,out}} \right)} \quad (28)$$

where $T_{D,in}$ and $T_{D,out}$ are the cold distribution in- and outlet temperatures, respectively. The same procedure

Table 5: Reference in- and outlet cold distribution temperatures for defined air-conditioning applications according to standard SIA 382/1 [37](#).

Air-conditioning application	Reference inlet temperature $T_{D,in}^*$	Reference outlet temperature $T_{D,out}^*$
No dehumidification	$\geq 14^\circ\text{C}$	$\geq 20^\circ\text{C}$
Partial dehumidification	$\geq 10^\circ\text{C}$	$\geq 16^\circ\text{C}$
Controlled dehumidification	$\geq 6^\circ\text{C}$	$\geq 12^\circ\text{C}$

is applied for the reference cold distribution exergy:

$$B_D^* = Q_D \left(1 - \frac{T_0}{T_D^*} \right) \quad (29)$$

with the reference temperature T_D^* expressed by:

$$T_D^* = \frac{T_{D,in}^* - T_{D,out}^*}{\ln \left(\frac{T_{D,in}^*}{T_{D,out}^*} \right)} \quad (30)$$

where $T_{D,in}^*$ and $T_{D,out}^*$ represent the reference in- and outlet temperatures. It is proposed to define the reference values for the cold distribution depending on the air-conditioning application according to the technical standard SIA 382/1 [37](#) listed in [Table 5](#).

4.2.4. Subsystem cold storage & transport

The optimization potential index proposed for the subsystem CST (see [Fig. 1](#)) is:

$$OPI_{CST} = 1 - \frac{\sum_{t=0}^{24h} \left[B_E^* + \sum_j B_{el,CP,CST,j}^* \right]}{\sum_{t=0}^{24h} \left[\sum_i B_{E,i} + \sum_j B_{el,CP,CST,j} \right]} \quad (31)$$

where the exergy input is summed up over all present refrigeration machines and auxiliary devices, which supply the subsystem. An exception is the reference evaporator exergy B_E^* , as it is determined over an exergy balance (see [Eq. 35](#)) for all present chillers and can therefore not be allocated to each one individually. The evaporator exergy is given by:

$$B_E = Q_E \left(1 - \frac{T_0}{T_E} \right) \quad (32)$$

with the evaporator thermal energy Q_E and the logarithmic mean temperature of the evaporator T_E expressed by the following equation:

$$T_E = \frac{T_{E,in} - T_{E,out}}{\ln \left(\frac{T_{E,in}}{T_{E,out}} \right)} \quad (33)$$

where $T_{E,in}$ and $T_{E,out}$ correspond to the secondary side evaporator in- and outlet temperature, respectively. Identically to the subsystem DC, the exergy of circulating pumps is:

$$B_{el,CP,CST} = W_{CP,CST} \quad (34)$$

with $W_{CP,CST}$, the corresponding electrical energy of the circulating pumps. The reference evaporator exergy input B_E^* is computed with an exergy balance over the cold storage & transport boundaries as follows:

$$B_E^* = \sum_k B_{D,k}^* + B_{L,CST} - \sum_j B_{el,CP,CST,j}^* \quad (35)$$

where B_D^* is the reference exergy output of the cold distribution (reference situation of the subsystem CL) and $B_{el,CP,CST}^*$ the reference circulating pump exergy input. Similarly to the subsystem DC, it is proposed to calculate the reference exergy input of circulating pumps with:

$$B_{el,CP,CST}^* = \frac{1}{f_{el,th,CP,CST}} \sum_i Q_{E,i} \quad (36)$$

where $f_{el,th,CP,CST}$ represents the electro-thermo amplification factor according to [Table 4](#). The destroyed exergy $B_{L,CST}$ is assumed to be identical as in the actual situation, which results in a stricter assessment and is defined as:

$$B_{L,CST} = \sum_i B_{E,i} + \sum_j B_{el,CP,CST,j} - \sum_k B_{D,k} \quad (37)$$

Similarly to the subsystem RM, a more detailed assessment of the losses is possible if additional boundaries are defined over single components, e.g. cold storages and mixing valves. However, this results in additional measured variables and the analysis must be carried out specifically for the plant under investigation, which is not applicable for a general assessment.

5. Results and discussion

5.1. Test cases

By applying the method described in section 4.2 and using the test case data of Table 3, the corresponding exergy values and the OPI of each subsystem are determined. Fig. 3a depicts the optimization potential index of every subsystem in the two investigated theoretical test cases. It is apparent, that in adequate operation (test case 1) all the OPI are below 0. Thus, the technical requirements are met. In test case 2, a fouled condenser is assumed, which results in a faulty operation of the refrigeration plant. The issue is revealed with an OPI of 0.052 in the subsystem DC and 0.014 in the subsystem RM, indicating potential for improvement.

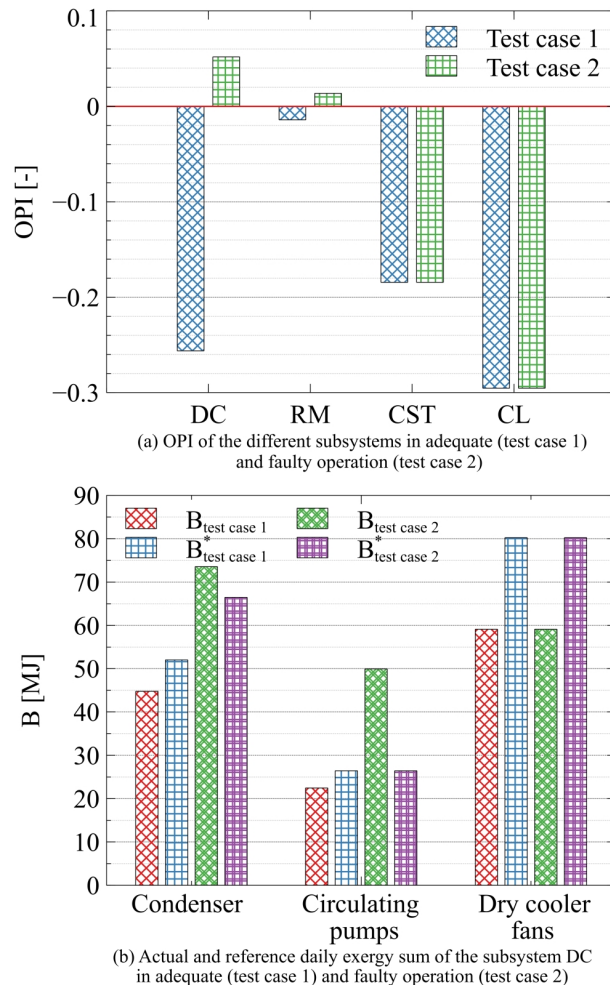


Figure 3: Results of the test case 1 (adequate operation) and test case 2 (faulty operation): (a) Optimization potential index of the subsystems dry cooler, refrigeration machine, cold storage & transport and cooling location, (b) actual and reference daily exergy sum of the different components in the subsystem DC.

A detailed analysis is feasible by investigating the exergy inputs of every component in the subsystem DC (see Fig. 3b). In test case 1, all the actual exergies B are inferior to the references B^* , resulting in an OPI_{DC} of -0.26 . In test case 2, an increased exergy input of the circulating pumps compared to the reference is observed,

in order to overcome the additional pressure drop in the fouled heat exchanger and to ensure a sufficient heat transfer. Also, the condenser exergy input is elevated compared to the reference due to the given increased temperature difference. The latter affects not only the subsystem DC, but also the subsystem RM, resulting in an OPI value superior to 0. The theoretical test cases demonstrate the functionality of the method, where the OPI delivers information about the refrigeration plant performance and indicating potential for improvement if a faulty operation is present.

5.2. Field plant

Following, the presented method is applied with measurements from a field plant as a case study. Fig. 4 and 7 show the daily optimization potential index (OPI) (y-axis) of every subsystem in the field plant under investigation in function of the date (x-axis) according to the available experimental data. The red line depicts the boundary between the adequate (technical requirements fulfilled) and the inadequate (potential for improvement) operating conditions.

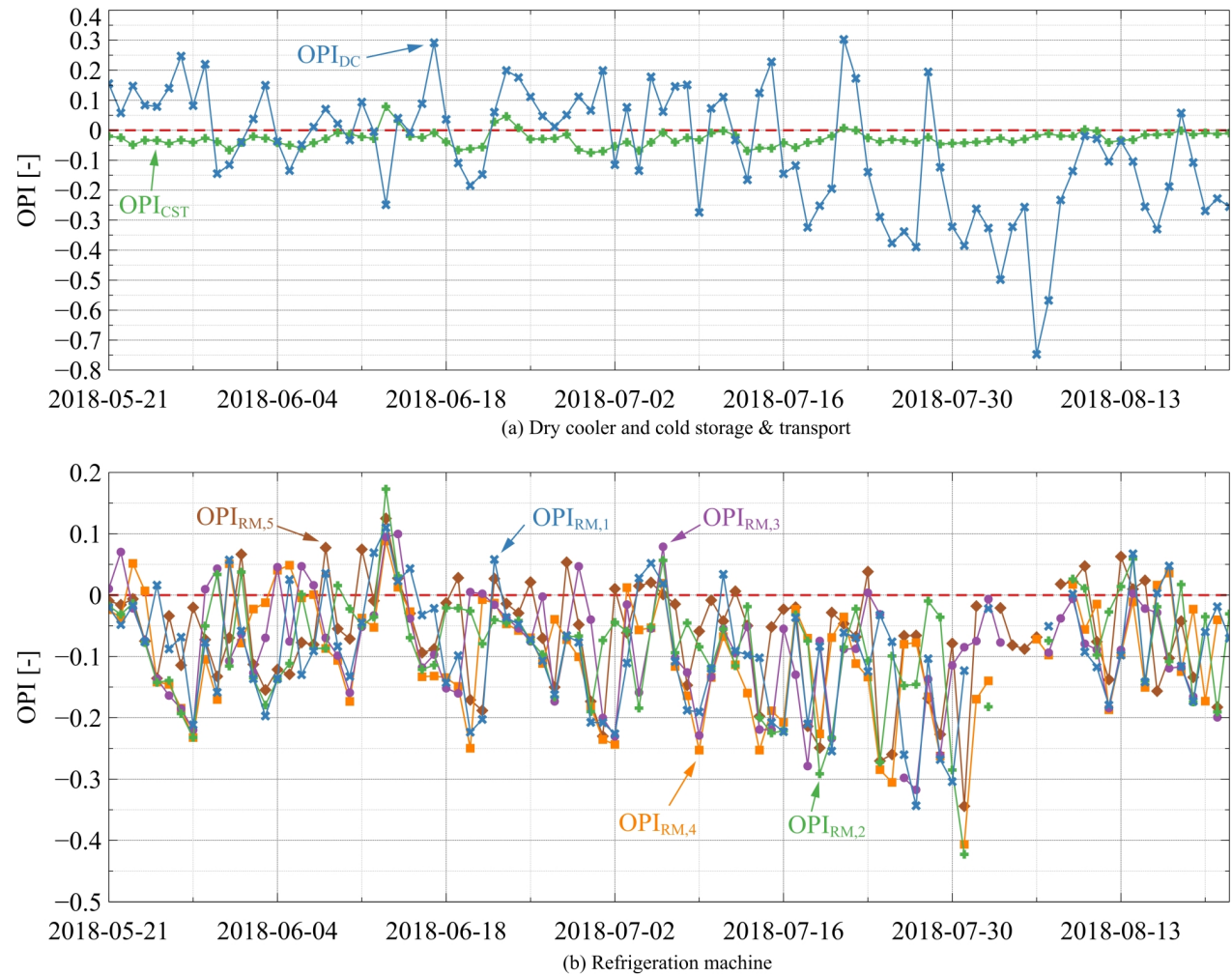


Figure 4: Optimization potential index of the subsystems (a) dry cooler, cold storage & transport and (b) refrigeration machine.

Fig. 5 illustrates the daily exergy efficiency η_{ex} (y-axis) of the subsystem RM and CST in the field plant under investigation in function of the date (x-axis) from mid of May to mid of July. As no field measurement data for the calculation of the thermal exergy of the dry cooler and cooling location is available, the respective exergy efficiency of the subsystems DC and CL cannot be determined.

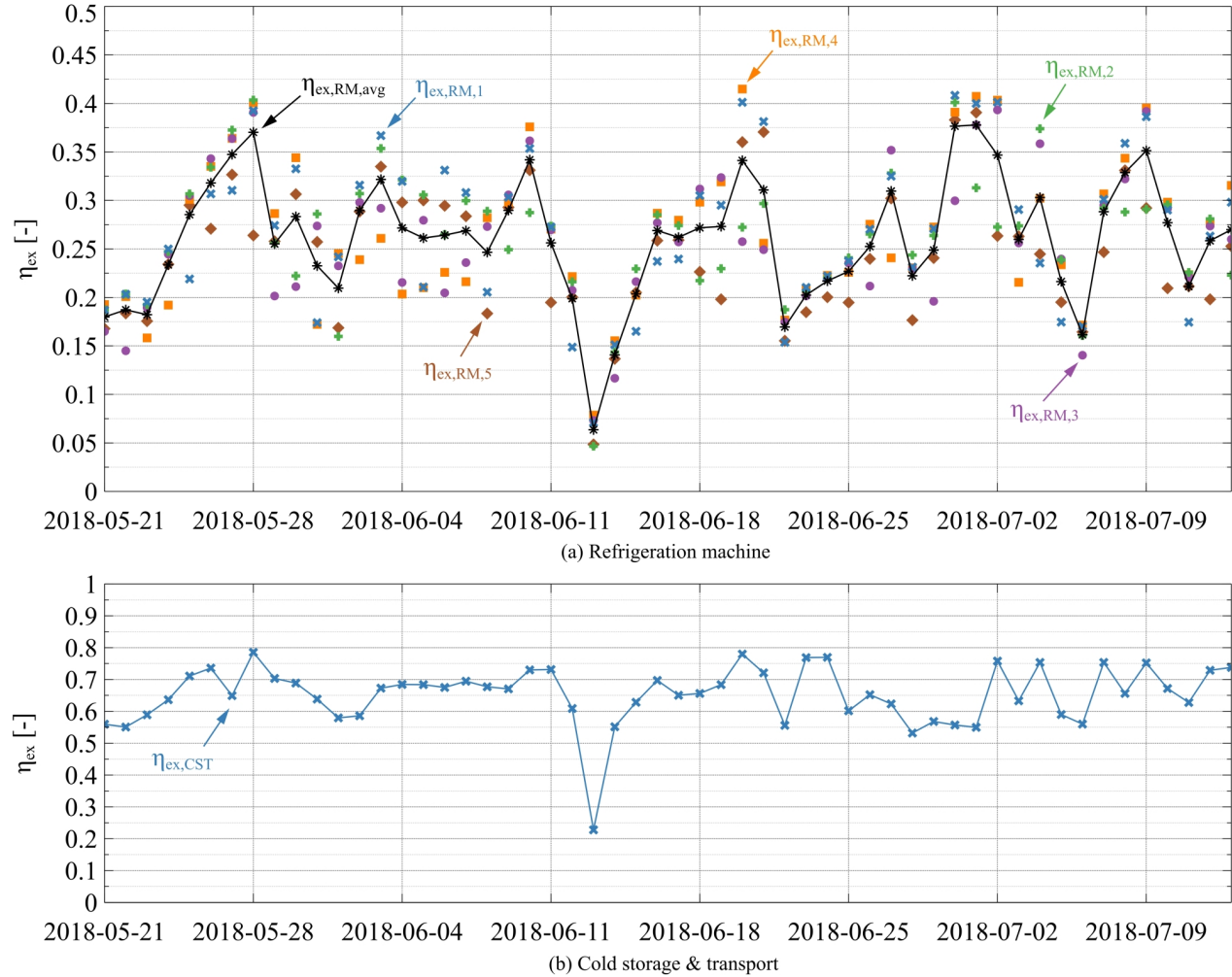


Figure 5: Exergy efficiency of the subsystems (a) refrigeration machine and (b) cold storage & transport.

5.2.1. Subsystem dry cooler

To begin with, the subsystem dry cooler has an average OPI_{DC} of approximately -0.06 and fluctuates strongly within a range of -0.75 to 0.3 (see Fig. 4a). In 55 % of the time, the optimization potential index is inferior to 0 and therefore the technical requirements are fulfilled. Nevertheless, in the time period from end of May to mid of July OPI_{DC} is mostly superior to 0, indicating potential for improvement. Similarly to the investigated test case 2 in section 5.1 a high key figure is an indicator for an elevated condenser secondary side temperature or an elevated electrical power consumption of the auxiliary devices, compared to the technical standards. Possible reasons are a temperature rise through mixing circuits, ambient air recirculation at the

dry cooler which compromises the heat transfer or an unfavorable design of the dry cooler for hot weather conditions (eg. an elevated ambient air temperature is present and thus an increased dry cooler fan speed is necessary to ensure the needed heat transfer). A fouled or defective heat exchanger can be excluded as a reason for the unfavorable operating condition, because otherwise OPI_{DC} would generally be higher.

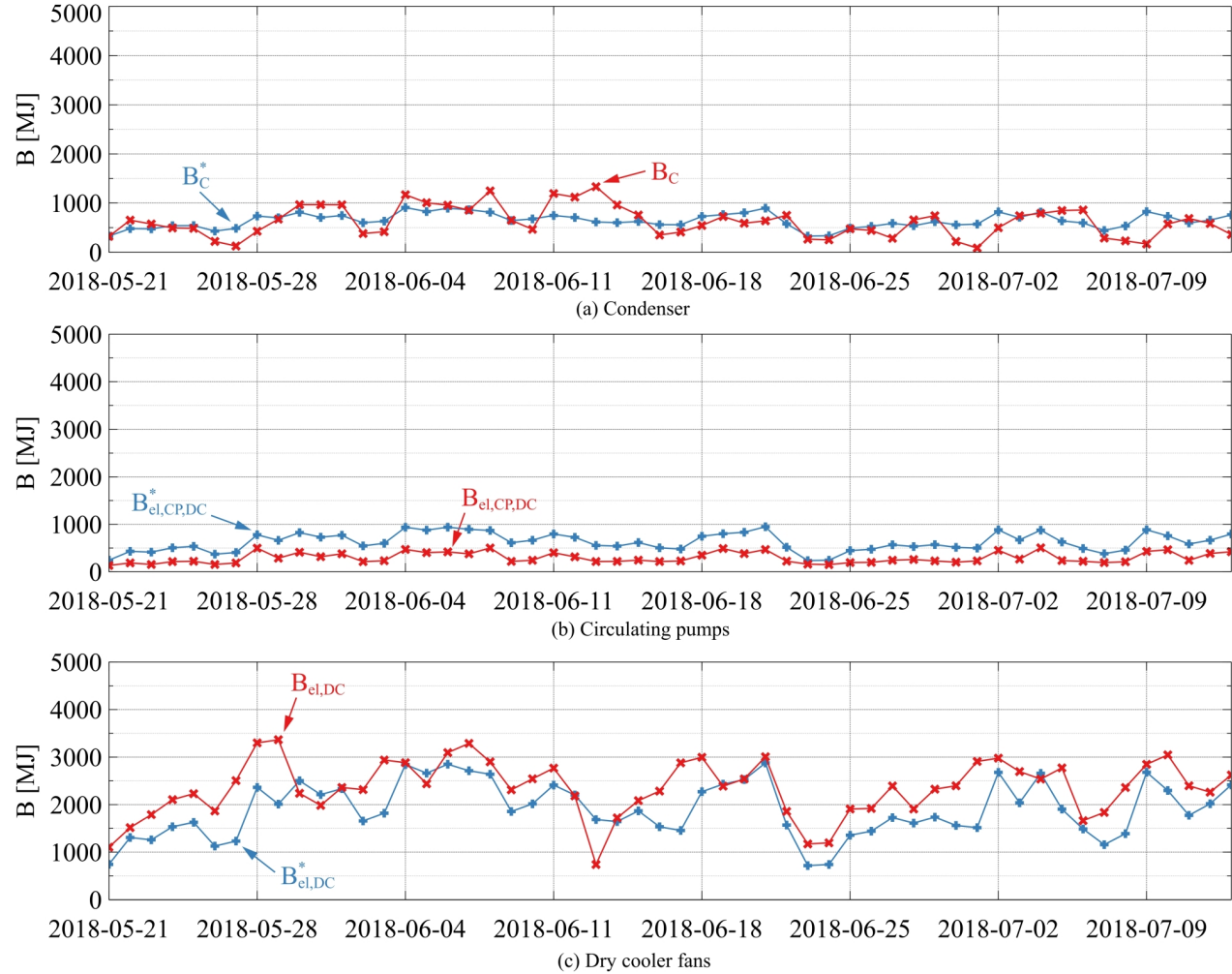


Figure 6: Actual and reference daily exergy sum of the different components in the subsystem DC: (a) condenser, (b) circulating pumps and (c) dry cooler fans.

Furthermore, to investigate these unfavorable operating conditions, [Fig. 6](#) depicts the actual and reference daily exergy sum (y-axis) of the components in the subsystem DC in function of the date (x-axis) in the mentioned time period. By examining the exergy of the condenser (see [Fig. 6a](#)) and the circulating pumps (see [Fig. 6b](#)) it is apparent, that the actual exergy input is generally lower or exhibits the same magnitude as the reference, which is favorable for the operating point of the refrigeration plant. **However, the actual dry cooler fans exergy (see [Fig. 6c](#)) is mostly elevated compared to the reference. This results in an increased value of the OPI, indicating potential for improvement. A similar behaviour was shown by the circulating pumps exergy in test case 2 (see section [5.1](#)).** Moreover, the dry cooler fans exergy, both the actual and

reference, is approximately up to 4.5 times higher as the condenser and circulating pumps exergy. This correlation underlines the fact, that the dry cooler fans electrical exergy is the driving quantity and has the most impact on OPI_{DC} . Therefore, the key figure is at times higher than 0, regardless how well the condenser and the circulating pumps perform. Taking the 1st of July as an example, the actual dry cooler fans exergy effort is twice as high as determined by the technical standards, which corresponds to an exergy difference of approximately 1400 MJ. Latter is 1.3 times the reference exergy sum of the condenser and circulating pumps, and thus determines the outcome.

Interestingly, in exergetic considerations, the electrical exergy plays a substantial role in comparison to the thermal exergy input. In energetic considerations, the electrical energy input only accounts for 1.2 to 3.5 % of the thermal energy according to the technical standards. This relationship underlines the importance of minimizing the electricity consumption of auxiliary devices where possible, in order to achieve an adequate system performance, which was also stated in other research [35].

5.2.2. Subsystem refrigeration machine

The optimization potential index of the five different refrigeration machines lies in the range of -0.42 to 0.17 , whereas the maximum difference of 0.34 between two chillers is reached on July 31st (see Fig. 4b). As the cooling load varies depending on the weather condition and building occupancy, not all chillers were operating daily and thus in some situations, e.g. on August 4th, not all key figures are present. Also, the values fluctuate with a lower magnitude compared to the subsystem DC and are closer to 0. All OPI_{RM} are 73 % and 87 % of the time inferior to 0 and 0.05, respectively. Refrigeration machine 4 performs best with an average optimization potential index of -0.1 , while refrigeration machine 5 performs worst with an average key figure of -0.07 . Therefore, little to no optimization potential compared to the technical standards is present. Moreover, daily outliers are not necessarily an indication of a malfunction, but if the key figure continuously and significantly increases, the plant would need inspection.

Furthermore, by examining the exergy efficiency of the chillers (see Fig. 5a), one can see, that there is a variation of the exergetic efficiency from 0.05 on June 13th up to 0.58 on July 31st. The daily fluctuation of the exergy efficiency between the different chillers is approximately 0.1. Refrigeration machine 4 has the highest average exergy efficiency of 0.29 and refrigeration machine 5 the lowest with a value of 0.27, where this behaviour is identical with the findings from the evaluation of OPI_{RM} . This shows, that the optimization potential index also delivers information about the performance and effectiveness of the plant. Moreover, the daily average exergy efficiency of all refrigeration machines $\eta_{ex, RM, avg}$ is depicted in Fig. 5a. The overall average exergetic efficiency is approximately 0.29, which is close to findings in literature for vapor compression refrigeration machines [13, 14, 38].

5.2.3. Subsystem cold storage & transport

The optimization potential index of the subsystem CST is almost in steady-state over the whole time period (see Fig. 4a). The index reaches its minimum of -0.07 on June 30th and its maximum of 0.08 on June

13th. OPI_{CST} is 93 % of the time smaller than 0 and has an average value of -0.03 . Therefore, the subsystem CST performs according to the technical standards. This outcome leads to the assumption that the hydraulic circuit is well designed and the circulating pumps well controlled, so that the electrical energy consumption of the circulating pumps does not exceed the limit of 1.5 % of Q_E .

In addition, the exergy efficiency ranges from 0.53 to 0.83, where the average exergy efficiency is 0.69 (see Fig. 5b). An outlier with a value of 0.23 is present on June 13th. The cause is a sudden decrease in the daily averaged ambient air temperature from 18.05 °C on June 12th to 13.61 °C on June 13th. Consequently, the thermal exergy in- and output (B_E and B_D) decreases while the electrical exergy input ($B_{el,CP,CST}$) remains high to assure the desired cooling load, which results in a low exergy efficiency. This correlation also influences the optimization potential index of the subsystems RM and CL (see Fig. 4b and 7) as well as the exergy efficiency of the subsystem RM on the mentioned day (see Fig. 5a). Moreover, the exergy efficiency of the subsystem CST is generally higher as compared to the refrigeration machines. This leads to the conclusion, that the exergy losses in the subsystem CST are correspondingly small. It is assumed that processes in the subsystem CST, which have a small temperature difference (e.g. mixing in the hydraulic circuit), exhibit a minor exergy destruction, which is favorable for the effectiveness. Latter cannot be confirmed through a calculation due to missing measured quantities, but this correlation was also stated in literature [23].

5.2.4. Subsystem cooling location

The optimization potential index in the subsystem CL exhibits a differentiated behavior depending on the cooling location (see Fig. 7). All locations have a reference average temperature of the cold distribution between the in- and outlet of ≈ 13 °C according to the design. Cooling location 4 performs best followed by location 1 with an average $OPI_{CL,4}$ of -0.88 and $OPI_{CL,1}$ of -0.11 (see Fig. 7a). Both yield 99 % of the time a value smaller than 0 and thus no optimization potential is present. The mean cold distribution temperature reaches $T_{D,1} \approx 14$ °C and $T_{D,4} \approx 14.3$ °C and is elevated compared to the design point. Therefore, the actual supplied exergy is lower as the reference, which is favorable for the plant operation. The peaks in $OPI_{CL,4}$ are assumed to be caused by a decrease of the temperature difference between $T_{D,4}$ and the reference temperature T_0 . The cooling location 2 and 7 operate close to the norm with an averaged value of 0.031 and 0.032, respectively, while achieving at least 84 % of the time a key figure inferior to 0.1 (see Fig. 7b).

Conversely, the optimization potential index of the other cooling locations is most of the time above 0, while staying in a bound of approximately 0.2 (see Fig. 7b). The cooling location 6 performs worst with a mean optimization potential index of 0.12, while the locations 3 and 5 achieve an averaged value of 0.1 and 0.07, respectively. As suspected, the reason is a decreased mean cold distribution temperature, yielding an increased exergy expense compared to the reference and thus a high optimization potential index is present. Accordingly, a performance increase is possible by investigating and resolving the issues at the concerned cooling locations. A possible reason is a fouled heat exchanger, which results in an elevated temperature difference in order to ensure the desired cooling load.

As mentioned in section 5.2.3, outliers are present on June 13th due to a decrease in daily averaged ambient

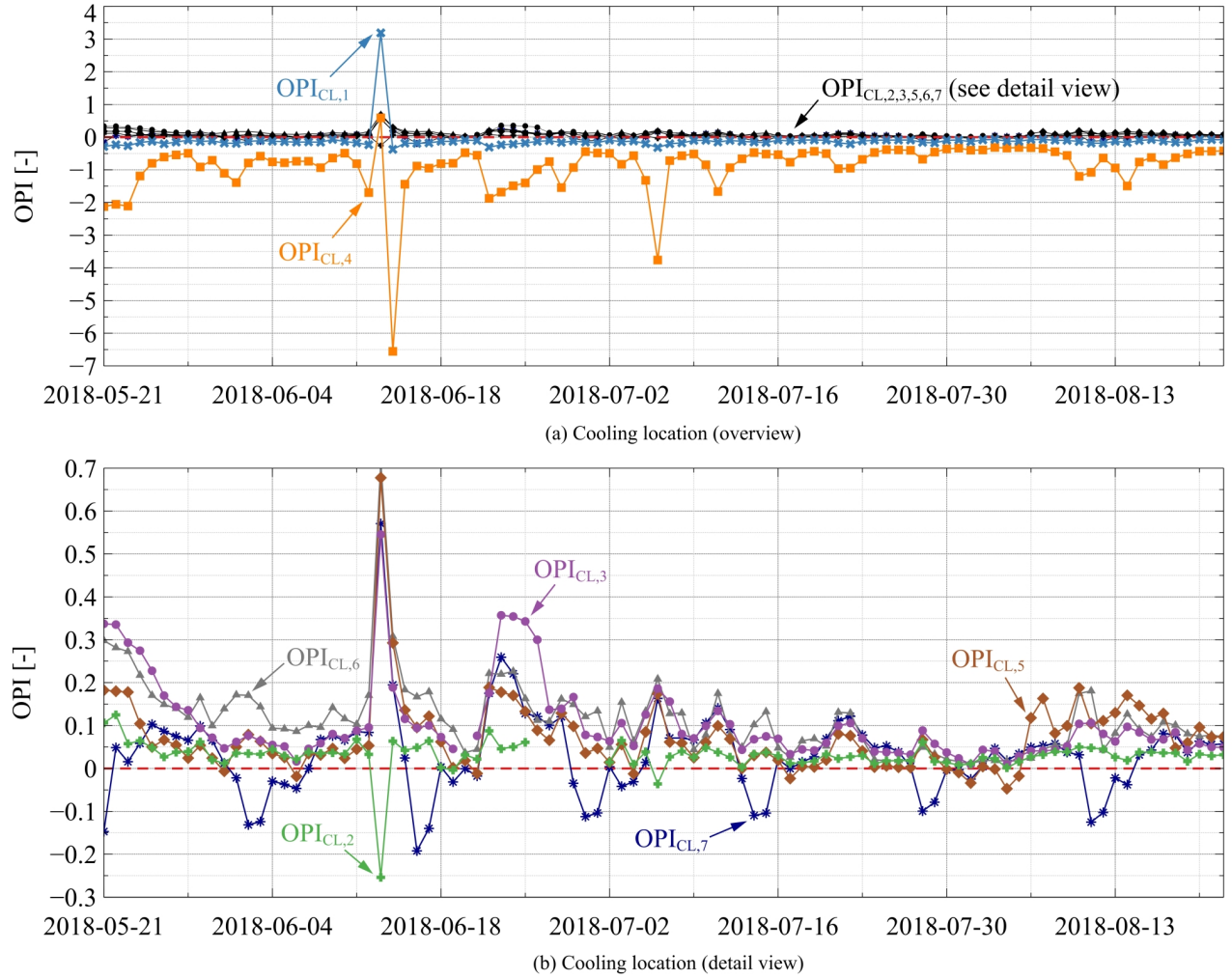


Figure 7: Optimization potential index of the subsystem cooling location: (a) overview and (b) detail view representation.

air temperature without a variation of the cooling load. $OPI_{CL,1}$ exhibits a value of 3.18, caused by a higher temperature of the cold distribution compared to the environment and the change of the thermal exergy transfer direction. In this situation, a potential for free cooling is present, meaning the chilling could be realized with the surrounding air, substituting the refrigeration machine. However, such individual outliers should not be overly considered, as solely a constantly poor key figure indicates a malfunction.

5.2.5. Comparison of OPI and COP

In order to demonstrate the practical usability of the proposed method, the OPI is compared with the COP of each subsystem. To facilitate the comparison, the key figures are averaged over all refrigeration machines and cooling locations, resulting in one key figure per subsystem. The COP_{RM} is calculated according to Eq. 6, while for the subsystems DC and CST the respective electrical energy consumption of the auxiliary devices is included as effort. For the COP_{CL} , the cold distribution thermal energy Q_D and all the electrical energy inputs are considered. With this definitions, the COP should deliver information about the performance with

respect to the defined subsystems as close as possible to the OPI. For the comparison, measurement data of June 10th and July 1st is used. The OPI and COP values are listed in Table 6, where an adequate operation of the refrigeration plant is present on June 10th. Conversely, on July 1st, the dry cooler fans electrical energy consumption is increased (compare Fig. 6), which represents a faulty operation of the subsystem DC with potential for improvement.

Table 6: OPI and COP values in the different subsystems on June 10th (adequate operation) and July 1st (faulty operation of the dry cooler fans).

Subsystem	June 10 th		July 1 st	
	OPI [-]	COP [-]	OPI [-]	COP [-]
DC	-0.03	5.65	0.20	5.31
RM	-0.11	8.21	-0.19	9.39
CST	-0.01	7.70	-0.07	9.00
CL	-0.04	5.17	-0.08	3.68

First of all, it is apparent that the refrigeration machines perform well on both days, which is revealed with a low OPI and a high COP, respectively. The same behavior is observed in the subsystem CST and is consistent with the general findings of the field plant analysis. Moreover, the COP_{CST} and COP_{DC} values are lower compared to the COP_{RM} , which is due to the additionally included effort of the auxiliary devices. Overall, by analyzing the values in Table 6 on both days, the OPI delivers similar information as the COP when it comes to the performance assessment. A performance increase is revealed with a decrease in OPI value or an increase in COP value and vice versa. An exception is present in the subsystem CL, where the OPI predicts a performance increase from June 10th to July 1st, while the COP reveals a performance decrease. However, this discrepancy should not be overly considered, as the OPI_{DC} only accounts for the cold distribution thermal exergy for a local assessment of the defined subsystem. In comparison, the COP_{CL} includes additionally all the electrical energy inputs of the refrigeration plant. Therefore, this key figure should rather be considered as an overall system performance indicator.

Furthermore, by comparing the COP_{DC} on June 10th with July 1st, a decrease of 0.34 is present. This reveals a performance reduction of the subsystem DC due to the present increase in electrical energy consumption of the dry cooler fans. However, by only considering the COP, in this case a value of 5.31, it is not apparent if any optimization potential is present. By analyzing the OPI_{DC} , an increase of 0.23 is observed. Therefore, the inadequate operation is indicated with a positive OPI value, which represents a potential for improvement. This relationship reveals the purpose and the significance of the optimization potential index. While the COP and OPI deliver adequate information about the refrigeration plant performance, the OPI reveals additionally any eventual optimization potential with respect to the state of the art in technology. Furthermore, the OPI allows an individual assessment of the different subsystems. One possible drawback of the proposed method is the additional need of temperature measurements for the exergy computation. However,

a heat meter typically measures the volume flow rate together with the present temperatures, and thus, the needed data for the exergy analysis is usually available in field plants.

6. Conclusions and outlook

The introduction of a novel key figure based on technical standards, the optimization potential index, enables the determination of the performance and corresponding improvement potential of vapor compression refrigeration plants with cold water distribution. By dividing the plant into different subsystems, each one can be assessed individually. To the best of our knowledge, the proposed exergy based method enables an absolute comparison of different refrigeration plants with defined reference values according to the state of the art in technology. Additionally, the optimization potential is revealed at a glance, regardless of the complexity of the system. Therefore, the results can also be interpreted by laypersons who can, if needed, initiate corrective measures by specialists. The functionality of the assessment method has been successfully proved by applying it to two theoretical test cases, to a real field installation and by comparing the optimization potential index with the coefficient of performance.

The present analysis of the field plant leads to the conclusion that the subsystem dry cooler performs well, except for the warmer days in the period end of May to mid of July, where an optimization potential is present. The subsystem should therefore be inspected for defects and corrective measures be taken if necessary (e.g. optimize the temperature level of the secondary hydraulic circuit). Furthermore, all refrigeration machines perform according to the technical standards while the optimization potential index is at least 87 % of the time below 0.05. The subsystem refrigeration machine exhibits elevated exergy losses compared to the subsystem cold storage & transport and therefore showing an average exergy efficiency of approximately 29 %. In addition, the subsystem cold storage & transport performs according to the technical norms while being in steady-state over the investigated time period. The mean exergy efficiency of 69 % is elevated compared to the subsystem refrigeration machine. Moreover, four out of seven cooling locations exhibit no to little optimization potential, while the remaining are not performing accordingly. Cooling location 3, 5 and 6 should therefore be inspected for defects and corrective measures be taken if necessary (e.g. optimize the heat transfer in the heat exchanger). Overall, the electrical exergy input has typically a higher magnitude as the thermal exergy and therefore determines the outcome of the key figures. This fact underlines the importance of minimizing the electrical energy input, e.g. with the use of frequency converters to regulate the rotational speed of pumps in partial load, in order to reduce the operating costs.

As an outlook, the presented method could be implemented in a monitoring system, which delivers daily information about the operating state of the plant. While a basic evaluation was successfully carried out and the functionality of the evaluation system was demonstrated, the interpretation becomes clearer by defining further limits. Hereby, an evaluation in form of e.g. a colored indicator could be realized, which defines the different operating states (see Fig. 2b). However, these limits must be determined with statistically sound, representative measurements and with the help of experts. Alternatively, thermodynamic models may be

used, in order to simulate different limiting operation conditions. Moreover, an optimization potential index for the overall refrigeration plant performance could be conceived. Also, additional system components of refrigeration plants (e.g. free cooling and heat utilization) or the air conditioning system with the cooled room and building envelope could be integrated in the evaluation. At the same time, a more detailed analysis of the plant would be possible by defining additional subsystems, with the disadvantage of requiring an increased number of measuring locations. Moreover, while the assessment runs well during the summer months, when a sufficient temperature difference between the system and the environment is present, further research is needed regarding the selection of the reference temperature. Its influence in transition periods such as spring and autumn needs to be further clarified, as the direction of exergy transfer is inverted. Furthermore, the influence of the air humidity must be included, when the cooled room itself or hybrid coolers, e.g. cooling towers, are considered.

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References

- [1] V. Vakiloroyaya, B. Samali, A. Fakhar, K. Pishghadam, A review of different strategies for HVAC energy saving, *Energy Conversion and Management* 77 (2014) 738–754. [doi:10.1016/J.ENCONMAN.2013.10.023](https://doi.org/10.1016/J.ENCONMAN.2013.10.023).
- [2] C. Park, H. Lee, Y. Hwang, R. Radermacher, Recent advances in vapor compression cycle technologies, *International Journal of Refrigeration* 60 (2015) 118–134. [doi:10.1016/J.IJREFRIG.2015.08.005](https://doi.org/10.1016/J.IJREFRIG.2015.08.005).
- [3] G. Pottker, P. Hrnjak, Experimental investigation of the effect of condenser subcooling in R134a and R1234yf air-conditioning systems with and without internal heat exchanger, *International Journal of Refrigeration* 50 (2015) 104–113. [doi:10.1016/J.IJREFRIG.2014.10.023](https://doi.org/10.1016/J.IJREFRIG.2014.10.023).
- [4] H. Fritschi, F. Tillenkamp, R. Löhner, M. Brügger, Efficiency increase in carbon dioxide refrigeration technology with parallel compression, *International Journal of Low-Carbon Technologies* [doi:10.1093/ijlct/ctw002](https://doi.org/10.1093/ijlct/ctw002).
- [5] M. Schweiker, M. Shukuya, Comparative effects of building envelope improvements and occupant behavioural changes on the exergy consumption for heating and cooling, *Energy Policy* 38 (6) (2010) 2976–2986. [doi:10.1016/J.ENPOL.2010.01.035](https://doi.org/10.1016/J.ENPOL.2010.01.035).
- [6] V. Bonetti, G. Kokogiannakis, Dynamic Exergy Analysis for the Thermal Storage Optimization of the Building Envelope, *Energies* 10 (1) (2017) 95. [doi:10.3390/en10010095](https://doi.org/10.3390/en10010095).
- [7] S. Dubey, S. Solanki, A. Tiwari, Energy and exergy analysis of PV/T air collectors connected in series, *Energy and Buildings* 41 (8) (2009) 863–870. [doi:10.1016/J.ENBUILD.2009.03.010](https://doi.org/10.1016/J.ENBUILD.2009.03.010).

- [8] R. Mishra, G. Tiwari, Energy and exergy analysis of hybrid photovoltaic thermal water collector for constant collection temperature mode, *Solar Energy* 90 (2013) 58–67. [doi:10.1016/J.SOLENER.2012.12.022](https://doi.org/10.1016/J.SOLENER.2012.12.022).
- [9] M. Razmara, M. Maasoumy, M. Shahbakhti, R. Robinett, Optimal exergy control of building HVAC system, *Applied Energy* 156 (2015) 555–565. [doi:10.1016/J.APENERGY.2015.07.051](https://doi.org/10.1016/J.APENERGY.2015.07.051).
- [10] B. Fan, X. Jin, X. Fang, Z. Du, The method of evaluating operation performance of HVAC system based on exergy analysis, *Energy and Buildings* 77 (2014) 332–342. [doi:10.1016/j.enbuild.2014.03.059](https://doi.org/10.1016/j.enbuild.2014.03.059).
- [11] Y. Bi, X. Wang, Y. Liu, H. Zhang, L. Chen, Comprehensive exergy analysis of a ground-source heat pump system for both building heating and cooling modes, *Applied Energy* 86 (12) (2009) 2560–2565. [doi:10.1016/J.APENERGY.2009.04.005](https://doi.org/10.1016/J.APENERGY.2009.04.005).
- [12] E. Saloux, M. Sorin, A. Teyssedou, Assessing the exergy performance of heat pump systems without using refrigerant thermodynamic properties, *International Journal of Refrigeration* 93 (2018) 1–9. [doi:10.1016/J.IJREFRIG.2018.06.005](https://doi.org/10.1016/J.IJREFRIG.2018.06.005).
- [13] A. Yataganbaba, A. Kilicarslan, I. Kurtbaş, Exergy analysis of R1234yf and R1234ze as R134a replacements in a two evaporator vapour compression refrigeration system, *International Journal of Refrigeration* 60 (2015) 26–37. [doi:10.1016/J.IJREFRIG.2015.08.010](https://doi.org/10.1016/J.IJREFRIG.2015.08.010).
- [14] R. Roy, B. K. Mandal, Thermodynamic Analysis of Modified Vapour Compression Refrigeration System Using R-134a, *Energy Procedia* 109 (2017) 227–234. [doi:10.1016/J.EGYPRO.2017.03.050](https://doi.org/10.1016/J.EGYPRO.2017.03.050).
- [15] W. Fratzscher, V. M. Brodjanskij, K. Michalek, *Exergie - Theorie und Anwendung*, Deutscher Verlag für Grundstoffindustrie, Leipzig, 1986.
- [16] A. Bejan, G. G. Tsatsaronis, M. J. Moran, *Thermal design and optimization*, Wiley, 1996.
- [17] M. Becker, T. Köberle, D. Pfeiffer, D. Rettich, EMeBK_a : Entwicklung und exemplarische Anwendung von Methoden zur energetischen Bewertung von Kälteanlagen im laufenden Betrieb, Tech. rep., Hochschule Biberach, Institut für Gebäude- und Energiesysteme, Biberach (2016). [doi:10.2314/GBV:87199531X](https://doi.org/10.2314/GBV:87199531X).
- [18] Mechanical Engineering Industry Association (VDMA), *Energieeffizienz von Kälteanlagen. Teil 2: Anforderungen an das Anlagenkonzept und die Komponenten (VDMA 24247-2)*, 2011.
- [19] Mechanical Engineering Industry Association (VDMA), *Energieeffizienz von Kälteanlagen. Teil 7: Regelung, Energiemanagement und effiziente Betriebsführung (VDMA 24247-7)*, 2011.
- [20] M. Sorrentino, G. Rizzo, A. Trifiro, F. Bedogni, A Model-Based Key Performance Index for Energy Assessment and Monitoring of Telecommunication Cooling Systems, *IEEE Transactions on Sustainable Energy* 5 (4) (2014) 1126–1136. [doi:10.1109/TSTE.2014.2334365](https://doi.org/10.1109/TSTE.2014.2334365).

- [21] J. Harrell, J. Mathias, Improving Efficiency in a Campus Chilled Water System Using Exergy Analysis, *ASHRAE Transactions* 115 (1) (2009) 507–522.
- [22] X. Fang, X. Jin, Z. Du, Y. Wang, The evaluation of operation performance of HVAC system based on the ideal operation level of system, *Energy and Buildings* 110 (2016) 330–344. [doi:10.1016/J.ENBUILD.2015.11.020](https://doi.org/10.1016/J.ENBUILD.2015.11.020).
- [23] K. Menberg, Y. Heo, W. Choi, R. Ooka, R. Choudhary, M. Shukuya, Exergy analysis of a hybrid ground-source heat pump system, *Applied Energy* 204 (2017) 31–46. [doi:10.1016/J.APENERGY.2017.06.076](https://doi.org/10.1016/J.APENERGY.2017.06.076).
- [24] D. Schmidt, Low exergy systems for high-performance buildings and communities, *Energy and Buildings* 41 (3) (2009) 331–336. [doi:10.1016/J.ENBUILD.2008.10.005](https://doi.org/10.1016/J.ENBUILD.2008.10.005).
- [25] S. Eisenhauer, T. Hauck, M. Arnemann, Systematische Erstellung und Anwendung messtechnischer Konzepte zur energetischen Untersuchung von Kälteanlagen, in: *DKV-Tagung*, Dresden, 2015.
- [26] M. J. Moran, H. N. Shapiro, D. D. Boettner, M. B. Bailey, *Fundamentals of engineering thermodynamics: SI version*, John Wiley, New York, 2010.
- [27] Y. A. Çengel, M. A. Boles, *Thermodynamics: an engineering approach*, McGraw-Hill, Boston, 2005.
- [28] H. D. Baehr, S. Kabelac, *Thermodynamik*, Springer, Berlin Heidelberg, 2012. [doi:10.1007/978-3-642-24161-1](https://doi.org/10.1007/978-3-642-24161-1).
- [29] M. Shukuya, *Exergy: theory and applications in the built environment*, Springer, London, 2013. [doi:10.1007/978-1-4471-4573-8](https://doi.org/10.1007/978-1-4471-4573-8).
- [30] H. Torío, D. Schmidt, Annex 49 final report: Energy Conservation in Buildings and Community Systems (ECBCS) - Low exergy systems for high-performance buildings and communities, Tech. rep., Fraunhofer IBP, Stuttgart (2011).
- [31] M. Pons, On the Reference State for Exergy when Ambient Temperature Fluctuates, *International Journal of Thermodynamics* 12 (3) (2009) 113–121. [doi:10.5541/IJOT.1034000246](https://doi.org/10.5541/IJOT.1034000246).
- [32] M. G. Baldi, L. Leoncini, Effect of Reference State Characteristics on the Thermal Exergy Analysis of a Building, *Energy Procedia* 83 (2015) 177–186. [doi:10.1016/J.EGYPRO.2015.12.208](https://doi.org/10.1016/J.EGYPRO.2015.12.208).
- [33] M. G. Alpuche, C. Heard, R. Best, J. Rojas, Exergy analysis of air cooling systems in buildings in hot humid climates, *Applied Thermal Engineering* 25 (4) (2005) 507–517. [doi:10.1016/J.APPLTHERMALENG.2004.07.006](https://doi.org/10.1016/J.APPLTHERMALENG.2004.07.006).
- [34] I. Dincer, M. Kanoğlu, *Refrigeration Systems and Applications*, John Wiley & Sons Ltd, Chichester UK, 2010. [doi:10.1002/9780470661093](https://doi.org/10.1002/9780470661093).

- [35] O. B. Kazanci, M. Shukuya, B. W. Olesen, Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy, *Building and Environment* 100 (2016) 102–113. [doi:10.1016/J.BUILDENV.2016.02.013](https://doi.org/10.1016/J.BUILDENV.2016.02.013).
- [36] Mechanical Engineering Industry Association (VDMA), *Energieeffizienz von Klimakälteanlagen. Teil 8: Komponenten - Wärmeübertrager* (Standard VDMA 24247-8), 2011.
- [37] Swiss Society of Engineers and Architects (SIA), *Lüftungs- und Klimaanlage - Allgemeine Grundlagen und Anforderungen* (Standard SIA 382/1), 2014.
- [38] R. Ben Jemaa, R. Mansouri, I. Boukholda, A. Bellagi, Energy and exergy investigation of R1234ze as R134a replacement in vapor compression chillers, *International Journal of Hydrogen Energy* 42 (17) (2017) 12877–12887. [doi:10.1016/J.IJHYDENE.2016.12.010](https://doi.org/10.1016/J.IJHYDENE.2016.12.010).