

Exergy performance and optimization potential of refrigeration plants in free cooling operation



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ABSTRACT

Exergy analysis has been widely used to assess refrigeration systems by evaluating exergy losses or exergy efficiency. The latter is mostly used as an indicator to determine the system performance, which requires the comparison of the actual system with its idealized reversible version, but not the practical achievable efficiency. Therefore, a practice-oriented evaluation method for refrigeration plants in free cooling operation is proposed, based on exergy analysis and technical standards as baseline. By considering the exergy input of auxiliary devices, the overall design of hydraulic circuits can be assessed on subsystem level. The achievable optimization potential compared to the state of the art in technology and the performance is revealed with the introduced optimization potential index (OPI). The application is demonstrated on a case study, where the analysis reveals an adequate operation of the field plant in general. Most cooling locations show potential for improvement, which is indicated by an OPI superior to zero. Moreover, the auxiliary electrical exergy input shows the same magnitude as the thermal exergy input, which emphasizes the importance of reducing the electrical energy usage of auxiliary devices in refrigeration plants to increase the performance.

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1. Introduction

In order to evaluate and reduce the energy consumption in buildings, e.g. heating, ventilation and air-conditioning (HVAC) systems, different approaches were applied. One suitable method is the concept of exergy. In the last couple of years, several exergy-based investigations were carried out in the field of building technologies, namely heat pumps [1,2], HVAC [3,4], photovoltaics [5,6], building energy systems [7,8] and refrigeration systems [9,10].

From a thermodynamic viewpoint, exergy can be seen as the unrestrictedly convertible portion of energy (maximum mechanical energy), which can be obtained from any form of energy by reversible interaction with the environment [11]. Exergy can also be identified as the quality of energy or available energy. In comparison to energy, exergy is destroyed in real (irreversible) thermodynamic processes, such as heat transfer in a heat exchanger. This represents a loss in energy potential to produce work. Such losses are the key for optimization, as their location and magnitude can be

identified throughout systems and subsequently countermeasures may be initialized. Furthermore, thermal and electrical energy are comparable with the exergy method, as the same quantity is used. Thus, exergy analysis allows a deep insight in the different energy flows and the corresponding losses of thermodynamic systems which is not feasible with a purely energetic analysis.

Refrigeration systems for air-conditioning applications are one key component to generate the desired cooling in buildings. With the goal to increase their overall efficiency, different concepts were introduced and applied to assess the energy and exergy utilization. Shan et al. proposed an improved chiller sequence control strategy for refrigeration plants with centrifugal chillers, where the authors showed an energy saving potential of 3% in comparison to the original control strategy [12]. In another study, free cooling for a data center in a hot summer and warm winter region with a cooling load of approximately 10 MW was evaluated [13]. An annual system coefficient of performance (COP) increase from 5.9 to 7.3 and an energy saving rate of 19.2% was observed if free cooling is fully utilized. Sorrentino et al. proposed a model-based key performance indicator (KPI), based on a reference energy efficiency index, to assess the performance of telecommunication cooling systems [14].

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Nomenclature			
<i>Abbreviations</i>		W	electrical energy [J]
AHU	air handling unit	\dot{W}	electrical power [W]
CL	subsystem cooling location	<i>Subscripts</i>	
COP	coefficient of performance	O	reference state for exergy analysis
CPI	control perfect index	amb	ambient
CST	subsystem cold water storage & transport	CL	cooling location
DC	subsystem dry cooler	CP	circulating pump
FC	subsystem free cooling	CPR	compressor
HVAC	heating, ventilation and air-conditioning	CST	cold water storage & transport
KPI	key performance indicator	D	distribution
OPI	optimization potential index	DC	dry cooler
RM	subsystem refrigeration machine	el	electrical
SIA	Swiss Society of Engineers and Architects	ex	exergetic
VDMA	Verband Deutscher Maschinen- und Anlagenbau	FC	free cooling
<i>Variables</i>		HE	heat exchanger
B	exergy [J]	in	component/system input
\dot{B}	exergy flow rate [W]	kin	kinetic
b	specific exergy [J/kg]	L	loss/destroyed
F	amplification factor [–]	m	mass
\dot{m}	mass flow rate [kg/s]	out	component/system output
P	pressure [Pa]	th	thermal
Q	thermal energy [J]	w	work
\dot{Q}	heat flow rate (thermal power) [W]	<i>Superscripts</i>	
\bar{T}	logarithmic mean temperature	*	reference value according to technical standards
T	temperature [K]	<i>Greek letters</i>	
t	time [s]	ΔT	temperature difference [K]
V	volume [m ³]	η	efficiency [–]

By comparing the measured (actual) cooling system power consumption to a simulated (reference) one, the key figure may be used to assess the cooling efficiency of the system or to realize a model-based fault detection, which reduces the risk of cooling device failure. Fan et al. evaluated the performance of an HVAC system in an airport terminal building and determined the influence of different control strategies on the system performance [4]. By means of a simulation model of the building and HVAC system and exergy analysis, it was found that the air handling unit (AHU) has the highest optimization priority, followed by the chillers. In order to evaluate the system performance over a certain time period in the setting of different control strategies with respect to an ideal operating level, the control perfect index (CPI) was introduced. Three different control strategies were assessed, where the highest yearly CPI value of 0.82 was reached with the optimal load allocation strategy in comparison to 0.77 with the original one.

Regarding vapor compression refrigeration plants, the interest extends from the refrigeration machine itself to the neighboring system structures. In practice, a well-suited hydraulic integration of the chiller or free cooling module is crucial in order to achieve a highly performing plant. Furthermore, it can be beneficial to partition the refrigeration plant into different subsystems, where each of them can be assessed individually. The influence of auxiliary devices on the exergy destruction was considered in a study of Kazanci et al., where different cooling systems were compared [15]. The authors showed that the electrical exergy input of auxiliary devices can be significant, which reveals the importance of reducing the latter to increase the system performance. Another study investigated the energy savings if variable frequency drives

for cooling tower fans are implemented [16]. Results showed that with a variable frequency drive, the water consumption was shortened by 13% and for the same cooling load, the overall energy consumption of the refrigeration plant was reduced by 5.8%. The approach to partition the investigated system into subsystems was applied by Harrell et al. to assess the performance of a chilled water system of a campus [9]. The authors found the highest system exergy efficiency of 13% at an evaporator cooling load of 70%. By dividing the refrigeration plant into four different subsystems, the most significant exergy losses of 42 and 31% were discovered in the stream turbine and refrigeration subsystem, respectively. A detailed exergy analysis of a ground-source heat pump was carried out by Menberg et al. [2]. The system was evaluated with 8 subsystems in heating as well as 6 subsystems in cooling operation. The exergy flow path was examined from the input to the output, where the exergy consumption in cooling mode is mentioned to be approximately 2.71 kW compared to 7.62 kW in heating mode. The authors state the need of additional research regarding variable boundary and system conditions, the investigation of different reference environment definitions and dynamic exergy evaluation. Another study focused on measuring concepts in order to evaluate the efficiency of refrigeration plants with an exergy-based approach [17]. Typically, an exergy analysis requires an increased amount of measured quantities. Therefore, they suggest using adequate system boundaries for the calculation and to use thermodynamic relations to compute missing variables where possible. Further, they state the need of achievable reference values in order to allow an absolute comparison, as they can vary across different refrigeration plants.

According to the literature search, the investigated refrigeration systems are mostly split into reasonable subsystems with respect to the available measurement data in order to allow a detailed assessment. The performance and optimization potential is then evaluated in terms of key figures. Typically, energy or exergy performance indicators are referred to a thermodynamic ideal, which is not achievable in practice. However, it is relevant for plant operators to identify the realizable improvements and to have reference values for an absolute comparison. Optimization potentials should be revealed, as measures which improve the system effectiveness most likely prevent frequent shortcomings during refrigeration plant operation. It should not require specialists in order to determine whether there is a need to take action on an installation. Therefore, a practice-oriented assessment method with technical standards as baseline is applied [18]. The objective of the present work is to introduce the method for refrigeration plants in free cooling operation, which, to the best of our knowledge, has never been applied before to such systems. To ensure a broad applicability in practice and to avoid high retrofitting costs by installing additional measurement equipment, the presented evaluation method aims to consider the most common measuring concepts of real refrigeration plants. First, the plant is divided into four different subsystems: dry cooler, free cooling, cold water storage & transport and cooling location. An exergy analysis is then carried out and the optimization potential index (OPI) is proposed for the free cooling configuration. This allows a straightforward determination of the performance and optimization potential of each subsystem in the refrigeration plant. With the OPI, a comparison between different refrigeration systems is possible and the reference can vary depending on the technological requirements, which is of great importance in practice. The assessment system is applied and its functionality demonstrated on a field plant as a case study.

2. Refrigeration plant

Fig. 1 shows schematically a typical refrigeration plant with cold water distribution and free cooling. Usually, multiple refrigeration

machines are set up in parallel (e.g. redundancy) and multiple cooling locations are present. The piping & instrumentation diagram is not intended to show the ideal configuration, but to reveal the assignment of the auxiliary devices to the different subsystems. The definition of the latter is based on investigations regarding existing systems but can be chosen differently according to the needed degree of detail. The present study focusses only on the refrigeration system, meaning other building components such as ventilation systems or the building envelope are not considered.

In refrigeration machine operating mode, the cooling is realized by a refrigeration cycle (subsystem refrigeration machine RM), where the chilled water is distributed over a secondary hydraulic circuit (subsystem cold water storage & transport CST) to the cooling locations, e.g. air-handling units (subsystem cooling location CL). The heat from the cycle process is discharged over another secondary hydraulic circuit (subsystem dry cooler DC) with the aid of dry coolers. If the surrounding air temperature is substantially lower than the temperature in the refrigeration system, the use of free cooling is feasible. Typically, a free cooling module (subsystem free cooling FC), in the simplest form consisting of a heat exchanger, is arranged in parallel to the chillers. In free cooling operation, the coupling of the subsystem dry cooler (DC) and cold water storage & transport (CST) allows an indirect cooling of the cooling location (CL) without the use of the refrigeration machines (RM). Thus, the electrical energy of the compressors can be economized. This free cooling operating mode is mostly present in the colder months over the year and is subject of the present analysis.

Moreover, the exergies of every subsystem in free cooling operation as well as the measured quantities for the computation are depicted in Fig. 1 (for details see Tables 1 and 2). The arrows indicate the exergy flows and B_L describes the exergy loss in every subsystem. The cooling location and dry cooler thermal exergy, B_{CL} and B_{DC} , are not needed for the assessment but shown for completeness. T_{amb} represents the ambient air temperature.

For the present study, a field plant is examined, from which measurements were collected during the year 2018. The refrigeration plant is located in Winterthur, Switzerland and incorporates

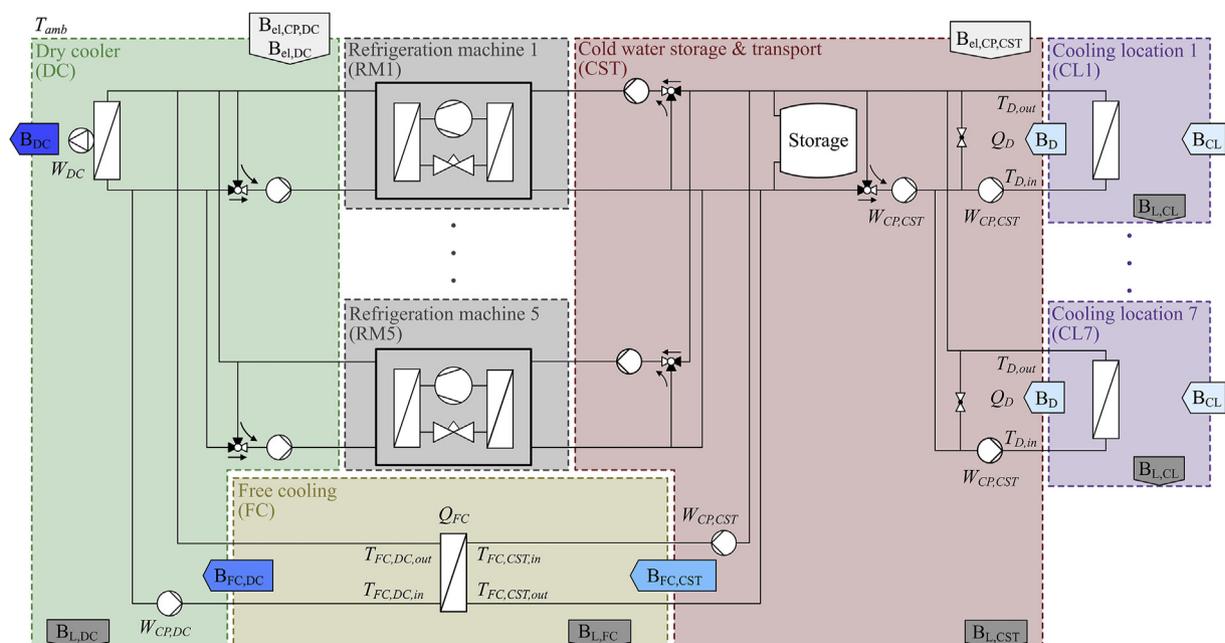


Fig. 1. Simplified piping & instrumentation diagram of a typical refrigeration plant with cold water distribution and free cooling divided into different subsystems. The exergy inputs and outputs in free cooling operation are depicted with arrows and measured variables shown in italic (for details see Tables 1 and 2).

Table 1
Measured variables in each subsystem for the exergy calculation.

Subsystem	Measured variables	
DC	$W_{CP,DC}$	circulating pump electrical energy
	W_{DC}	dry cooler fan electrical energy
FC	Q_{FC}	free cooling thermal energy
	$T_{FC,DC,in}$	free cooling inlet temperature DC side
	$T_{FC,DC,out}$	free cooling outlet temperature DC side
	$T_{FC,CST,in}$	free cooling inlet temperature CST side
CST	$T_{FC,DC,out}$	free cooling outlet temperature CST side
	$W_{CP,CST}$	circulating pump electrical energy
CL	Q_D	cold water distribution thermal energy
	$T_{D,in}$	cold water distribution inlet temperature
	$T_{D,out}$	cold water distribution outlet temperature
All	T_{amb}	ambient air temperature

five refrigeration machines with a total cooling power of approximately 5 MW and ammonia (R717) as refrigerant. The plant supplies seven buildings with chilled water, where the cooling locations represent air-handling units of ventilation systems in the different buildings, three of which were integrated into the system in April 2018. The temperature requirement of all cooling locations is an average value of ≈ 13 °C between the cold water supply and return flow according to the design. Additionally, 12 circulating pumps are present, from which three are winter pumps for the free cooling operation, in order to transport the cold water through the underground distribution network. One free cooling heat exchanger is integrated to the system, while three dry coolers discharge the heat to the environment. As a result, 10 different subsystems are defined to analyze the free cooling operation. No data was registered of the circulating pumps in free cooling operation (winter pumps). As an approach, the electrical energy input is calculated under the assumption that they behave similarly as the circulating pumps in refrigeration machine operation.

3. Assessment approach

3.1. Exergy balance and key figures

The rate of change of exergy dB/dt over a control volume, e.g. over a subsystem in the refrigeration plant, can be written as [19,20]:

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

The three underlined terms correspond to the exergy transfer by heat \dot{B}_{th} , work \dot{B}_w and mass flow \dot{B}_m , where

- \dot{Q}_i represents a heat flow rate in or out of the control volume,
- T_i is the temperature at which the heat transfer takes place,
- T_0 corresponds to the reference temperature,
- \dot{W} represents mechanical or electrical power in or out of the control volume,
- p_0 is the reference pressure,
- dV/dt corresponds to the rate of change of the control volume,
- $\dot{m}_{in,j}$ and $\dot{m}_{out,k}$ represent the mass flow rates in and out of the control volume,
- $b_{in,j}$ and $b_{out,k}$ are the incoming and outgoing specific flow exergies, and,
- \dot{B}_L corresponds to the exergy flow rate losses in the control volume due to irreversibilities.

By definition, mechanical energy is pure exergy, as it can be fully transformed into any other form of energy (e.g. heat). In contrast, thermal energy contains only a portion of exergy and cannot be fully transformed. By assuming steady-state, incompressible flow (liquid in the hydraulic circuits) and neglecting pressure losses over the system boundaries, Eq. (1) can be expressed as follows [8,21]:

$$0 = \sum_i \left(1 - \frac{T_0}{T_i}\right) \dot{Q}_i - \dot{W} + \sum_l \left(1 - \frac{T_0}{\bar{T}_l}\right) \dot{Q}_l - \dot{B}_L \quad (2)$$

where the exergy transfer by mass flow is now expressed by the net heat \dot{Q}_l transported over the system boundary with the incoming and outgoing mass flow and the logarithmic mean temperature \bar{T}_l defined as:

$$\bar{T}_l = \frac{T_{in,l} - T_{out,l}}{\ln\left(\frac{T_{in,l}}{T_{out,l}}\right)} \quad (3)$$

with $T_{in,l}$ the temperature of the incoming and $T_{out,l}$ the temperature of the outgoing mass flow, respectively. With the intention of ensuring a broad applicability of the presented method, the simplifications mentioned above are considered tolerable. While the pressure losses are not explicitly treated for the exergy transfer between the subsystems, they are indirectly considered on subsystem level by assessing the exergy input of the circulating pumps

Table 2
Exergy inputs and outputs of each subsystem.

Subsystem	Exergy input		Exergy output	
DC	$B_{el,CP,DC}$	circulating pump exergy	B_{DC}	dry cooler exergy
	$B_{el,DC}$	dry cooler fan exergy		
	$B_{FC,DC}$	free cooling exergy DC side		
FC	$B_{FC,CST}$	free cooling exergy CST side	$B_{FC,DC}$	free cooling exergy DC side
	$B_{FC,CST}$	free cooling exergy CST side		
CST	$B_{el,CP,CST}$	circulating pump exergy	$B_{FC,CST}$	free cooling exergy CST side
	B_D	cold water distribution exergy		
CL	B_{CL}	cooling location exergy	B_D	cold water distribution exergy

in the exergy analysis. A detailed hydraulic evaluation could be carried out in a further evaluation step, assuming that the required technical data is available. The exergy efficiency η_{ex} is mostly used to assess the exergy performance of thermodynamic systems. The key figure can be evaluated by balancing the total exergy output \dot{B}_{out} to the input \dot{B}_{in} [22]:

$$\eta_{ex} = \frac{\dot{B}_{out}}{\dot{B}_{in}} = 1 - \frac{\dot{B}_L}{\dot{B}_{in}} \quad (4)$$

which indicates how much exergy leaves the system compared to the input. The key figure reaches a value of 0 if the energy is completely degraded (only losses present) and 1 if a thermodynamic ideal process (no exergy losses) is considered. Since every real process is irreversible, exergy losses are to be expected and the need to know the best process possible is appearing. This indicator allows a relative comparison between different processes with regard to the thermodynamic ideal, but there are no reference values available stating which exergy efficiency should be achieved with a certain process. Thus, the quantification of the performance and possible optimization potentials according to the state of the art in technology is problematic, which is of great importance in practice. For this reason, the optimization potential index (OPI) is applied which has the following general form [18]:

$$OPI = 1 - \frac{\eta_{ex}}{\eta_{ex}^*} = 1 - \frac{\frac{\dot{B}_{out}}{\dot{B}_{in}}}{\frac{\dot{B}_{out}^*}{\dot{B}_{in}^*}} = 1 - \frac{\dot{B}_{in}^*}{\dot{B}_{in}} \quad (5)$$

The OPI balances the effective exergy efficiency η_{ex} according to measurement data with a reference efficiency η_{ex}^* . The key figure describes the behavior of the real system in comparison with a reference system under identical operation conditions while the same exergy output \dot{B}_{out} is achieved. Thus, the evaluation of a subsystem is carried out with the assumption that the adjacent subsystems perform identically. \dot{B}_{in} and \dot{B}_{in}^* describe the actual exergy input according to measurements and the reference exergy input, respectively. As a first approach, the reference values are defined according to technical standards (see subsection 3.3). This represents an achievable technological baseline, where the specified conditions can be fulfilled or exceeded if the refrigeration plant is properly engineered and maintained. Typically, such standards are specified in tenders or contracts and should be fulfilled at the stage of commissioning. Any other appropriate standards or limits (e.g. performance parameters) could find application, i.e. the baseline can depend on the technological requirements in different countries or regions.

With the OPI, the results are straightforward to interpret (see Fig. 2). No specialists are required to reveal eventual issues or improvement potentials, which is of great importance in practice.

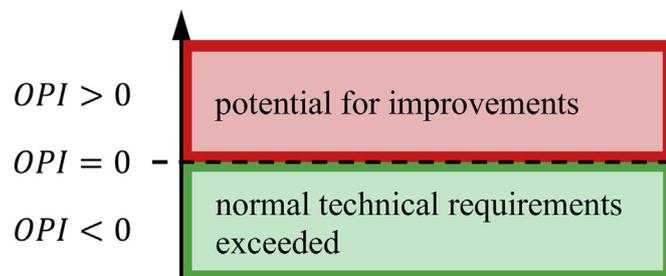


Fig. 2. OPI scale with the corresponding operating conditions.

For example, plant operators can easily evaluate the system operation by tracking the key figure over time. A colored indicator as depicted in Fig. 2 could be realized and implemented in monitoring systems, which helps also to raise awareness for an efficient refrigeration plant operation. If the actual exergy input of the subsystem is larger than the reference (meaning an increased input is required to achieve the same output), the OPI is larger than 0 and points out improvement potential. For example, assuming a fouled heat exchanger of the dry cooler. As a consequence, the cooler fans need to operate at a higher frequency to ensure the needed heat transfer, while consuming an increased amount of electrical energy. The actual exergy input of the subsystem DC will be larger compared to the reference and the OPI reveals the issue with a value superior to 0. Conversely, if the dry cooler is correctly maintained and operated, the actual exergy input is lower than the reference, where the OPI delivers a value inferior to 0. Thus, no optimization potential according to the state of the art in technology is present. An OPI of 0 indicates that the technical requirements are met. Therefore, the OPI delivers a first localization of eventual issues and optimization potentials on subsystem level. In a second evaluation step, the malfunction can be further narrowed down with a more detailed analysis. Eventually, by including a cost analysis, it is revealed if adjustments to the system are worthwhile, not only from a technical but also an economical aspect. Plant operators can then initialize the appropriate corrective measures by specialists.

As for technical reasons, typically no physical data is available in real systems for the calculation of B_{CL} (see Fig. 1) of the subsystem CL. Therefore, an alternative approach is applied for the latter in free cooling operation. Namely, the exergy input is set to be the same in the actual and in the reference condition, while the exergy output varies according to the technical standards. The latter is plausible from a thermodynamic viewpoint, as it is favorable for the subsystem, if more exergy is extracted compared to the reference operating condition. Thus, the interpretation of the OPI remains the same for the subsystem CL.

3.2. Reference environment

Exergy describes the energy potential with respect to an environmental reference. Therefore, exergy evaluations require a proper definition of the latter. Torío et al. investigated different reference states for the building environment, where they recommend using the surrounding ambient air as reference [23]. The variable outdoor temperature was applied in other studies [24,25]. However, other research stated to use a fixed and constant reference in order to carry out a consistent and reliable exergy analysis [26,27]. Therefore, as a first approach, a constant reference temperature T_0 of 5 °C is used in the present study, which represents the averaged outdoor temperature of the months January to April and November to December 2018, according to measurements from the investigated field plant. While the definition of the reference environment may influence the absolute values of the exergy, the effect on the OPI is assumed to be small, as the same reference temperature is used for the actual and reference exergy.

3.3. Subsystem evaluation

With the goal to achieve a daily assessment of refrigeration plants in free cooling operation, as the cooling load in air-conditioning applications typically exhibits a daily rhythm, a quasi-stationary approach is chosen. The exergy flow rates are determined and balanced over each subsystem. The needed quantities are calculated from measurement data, obtained with reference values from technical standards or determined with an

Table 3
Proposed OPI definitions of every subsystem in free cooling operation of the refrigeration plant.

Subsystem	Optimization potential index
DC	$OPI_{DC} = 1 - \frac{\sum_{t=0}^{24h} [B_{FC,DC}^* + \sum_i B_{el,CP,DC,i}^* + \sum_j B_{el,DC,j}^*]}{\sum_{t=0}^{24h} [B_{FC,DC} + \sum_i B_{el,CP,DC,i} + \sum_j B_{el,DC,j}]} \quad (6)$
FC	$OPI_{FC} = 1 - \frac{\sum_{t=0}^{24h} B_{FC,CST}^*}{\sum_{t=0}^{24h} B_{FC,CST}} \quad (7)$
CST	$OPI_{CST} = 1 - \frac{\sum_{i=0}^{24h} [\sum_i B_{D,i}^* + \sum_j B_{el,CP,CST,j}^*]}{\sum_{i=0}^{24h} [\sum_i B_{D,i} + \sum_j B_{el,CP,CST,j}]} \quad (8)$
CL	$OPI_{CL} = 1 - \frac{\sum_{t=0}^{24h} B_D}{\sum_{t=0}^{24h} B_D^*} \quad (9)$

exergy balance (see Eq. (2)). Subsequently, the exergy flow rates are numerically integrated with the trapezoidal rule over the present measurement interval of 5 min. The corresponding exergies of each interval are then summed up over 24 h and the optimization potential index is evaluated on a daily basis.

Each subsystem of the field plant in free cooling operation is assessed separately with the described procedure. Table 3 lists the proposed definitions of the optimization potential index of each subsystem under consideration, with the exergies according to Table 2. The exergies marked with an asterisk represent reference values computed with technical standards. The detailed approach is introduced below using the subsystem dry cooler (DC) as an example, whereby the OPI for the other subsystems is determined similarly using the corresponding measured variables from Table 1.

The OPI_{DC} is proposed according to Eq. (6) (see Table 3), where the electrical exergy input of each present circulating pump and dry cooler fan is summed up. The free cooling exergy on the DC side $B_{FC,DC}$ is defined as:

$$B_{FC,DC} = Q_{FC} \left(1 - \frac{T_0}{\bar{T}_{FC,DC}} \right) \quad (10)$$

where Q_{FC} represents the free cooling thermal energy, T_0 the reference temperature and $\bar{T}_{FC,DC}$ the logarithmic mean temperature of the secondary circuit of the heat exchanger according to:

$$\bar{T}_{FC,DC} = \frac{T_{FC,DC,in} - T_{FC,DC,out}}{\ln \left(\frac{T_{FC,DC,in}}{T_{FC,DC,out}} \right)} \quad (11)$$

with the inlet and outlet temperature $T_{FC,DC,in}$ and $T_{FC,DC,out}$ of the heat exchanger on the dry cooler side. By definition, electrical energy is pure exergy, and thus, the exergy of the dry cooler fans $B_{el,DC}$ is given by:

$$B_{el,DC} = W_{DC} \quad (12)$$

with the dry cooler fan electrical energy W_{DC} . Correspondingly, the exergy of circulating pumps $B_{el,CP,DC}$ is defined by:

$$B_{el,CP,DC} = W_{CP,DC} \quad (13)$$

with $W_{CP,DC}$, the respective electrical energy input of the circulating pumps. The reference exergy of the free cooling on the DC side $B_{FC,DC}^*$ is given by:

$$B_{FC,DC}^* = Q_{FC} \left(1 - \frac{T_0}{T_{FC,DC}^*} \right) \quad (14)$$

where $T_{FC,DC}^*$ represents the reference temperature of the free cooling module. As no reference temperatures for the inlet and outlet of the free cooling module according to technical standards are available, a definition similarly to Eq. (11) is not possible. Therefore, a calculation with the ambient air temperature as a basis and temperature differences according to technical standards as well as available measurements is proposed as a first approach:

$$T_{FC,DC}^* = T_{amb} + \Delta T_{HE} + \frac{T_{FC,DC,out} - T_{FC,DC,in}}{2} \quad (15)$$

where T_{amb} is the ambient air temperature and ΔT_{HE} the temperature difference between the secondary hydraulic circuit medium and the ambient air in the dry cooler heat exchanger. According to the technical standard VDMA 24247–8 [28], a desirable temperature difference ΔT_{HE} of 6 K is defined. The reference exergy of the dry cooler fans $B_{el,DC}^*$ is proposed as follows:

$$B_{el,DC}^* = \frac{1}{f_{el,th,DC}} Q_{FC} \quad (16)$$

with the electro-thermo amplification factor for dry cooler fans $f_{el,th,DC}$ (see Table 4). This factor determines the desirable amount of electrical energy consumed by auxiliary devices with respect to the transferred thermal energy. Similarly, the reference exergy of the circulating pumps $B_{el,CP,DC}^*$ is given by:

$$B_{el,CP,DC}^* = \frac{1}{f_{el,th,CP,DC}} Q_{FC} \quad (17)$$

where $f_{el,th,CP,DC}$ represents the electro-thermo amplification factor for circulating pumps (see Table 4). The amplification factors are defined in the technical standard SIA 382/1 [29] with respect to the thermal energy of the condenser and evaporator of the integrated refrigeration machine. As an approach, the thermal energy of the free cooling module is used for the computation of the reference exergies of the auxiliary devices, as the refrigeration machines are turned off in free cooling operation.

Concerning the other subsystems, the reference exergy of the circulating pumps $B_{el,CP,CST}^*$ in the subsystem CST is computed similarly to Eq. (17) with the electro-thermo amplification factor $f_{el,th,CP,CST}$ according to Table 4. The reference exergy of the free cooling module $B_{FC,CST}^*$ on the CST side is determined by an exergy balance of the reference exergies in the subsystem CST. The cold

Table 4
Amplification factors for auxiliary devices according to the technical standard SIA 382/1 [29].

Subsystem	Electro-thermo amplification factor	Threshold value
DC	dry cooler fan $f_{el,th,DC}$	≥ 28
	circulating pump $f_{el,th,CP,DC}$	≥ 85
CST	circulating pump $f_{el,th,CP,CST}$	≥ 65

water distribution reference exergy B_D^* of the subsystem CL is calculated according to Eq. (14), but with the cold water distribution thermal energy Q_D . The cold water distribution reference temperature T_D^* is determined similarly to Eq. (11) with temperatures of the secondary side hydraulic circuit according to the technical standard SIA 382/1 [29]. A cold water distribution reference inlet temperature $T_{D,in}^*$ of 10 °C and a cold water distribution reference outlet temperature $T_{D,out}^*$ of 16 °C is specified, representing the temperature level for an air-conditioning application with partial dehumidification.

4. Results and discussion

The investigated field plant exhibits free cooling operation if the measured compressor electrical energy W_{CPR} is equal to and the free cooling thermal energy Q_{FC} larger than 0 J, respectively. This is the case from January to mid-March as well as from end of October to December 2018. Mixed operation of the refrigeration machines and the free cooling module, e.g. precooling overnight, is not considered in the present work. The daily OPI (y-axis) of every subsystem in the refrigeration plant in function of the date (x-axis) is depicted in Fig. 3.

4.1. Subsystem FC

To begin with, the subsystem FC shows an average OPI_{FC} of -0.30 and exhibits almost a steady-state behavior over the investigated time period (see Fig. 3a). The minimum of -0.45 is achieved on December 25th and the maximum of -0.21 on November 21st. The key figure is 100% of the time lower than 0 and thus, the technical requirements according to the applied technical standards are exceeded. This leads to the assumption that the implemented heat exchanger is well designed according to the specifications, which results in an adequate operation and performance.

4.2. Subsystem CST

The subsystem CST reveals a similar behavior, where the technical requirements are fulfilled. The OPI_{CST} is 100% of the investigated time period lower than 0 and shows an average value of -0.30 (see Fig. 3a). The hydraulic circuit is apparently well designed and the circulating pumps correctly operated, resulting in a low electrical power consumption and a low exergy input compared to the reference, respectively. Interestingly, the performance indicator reveals an increase of approximately 0.2 from mid-March to end of October. A possible reason are the three additional cooling locations which were integrated at the end of April, and

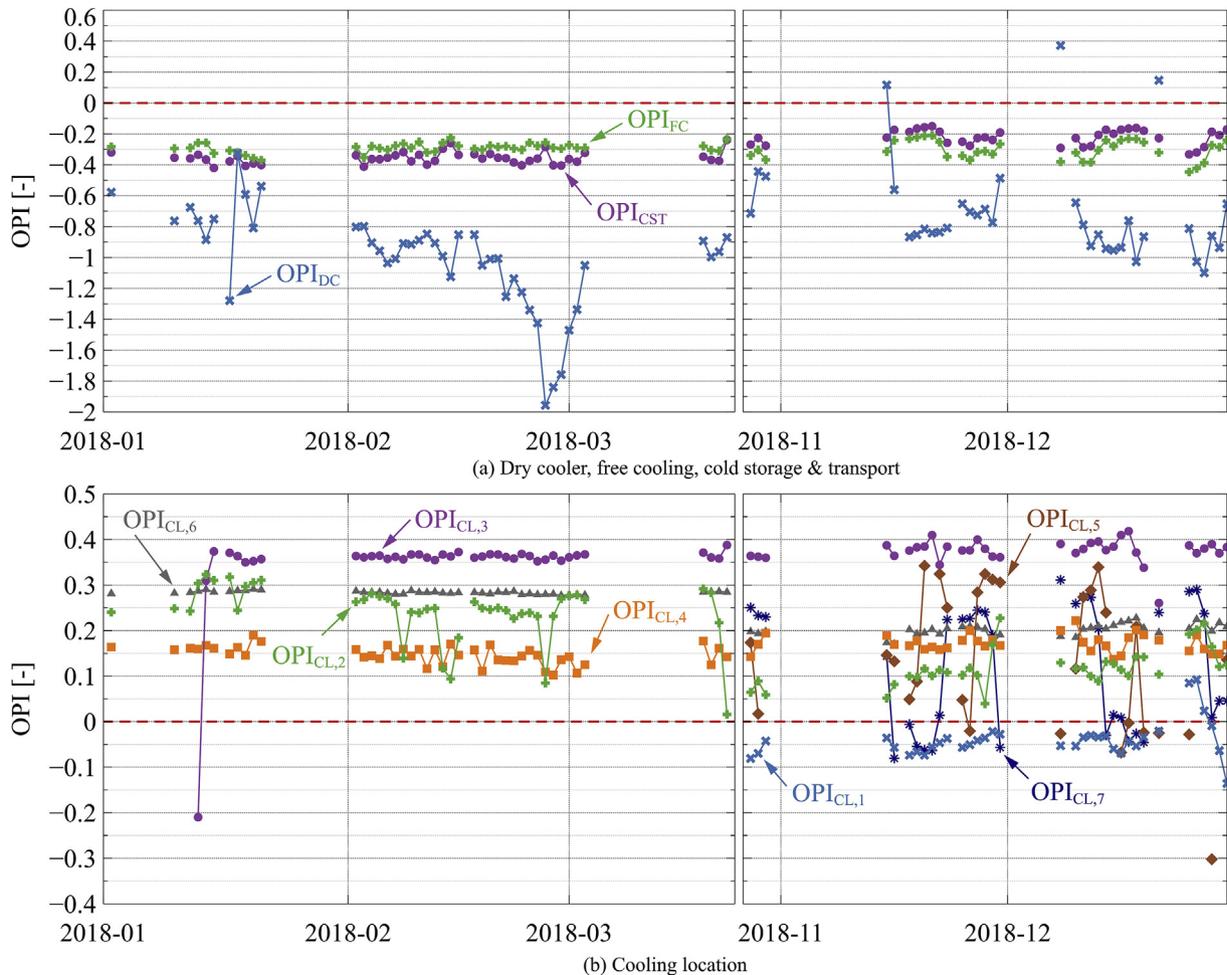


Fig. 3. Optimization potential index (OPI) of the subsystem DC, FC and CST (a) as well as subsystems CL,1 to CL, 7 (b).

thus, the actual exergy input of the subsystem is increased. Most likely, the circulating pumps exhibit an increased electrical energy usage, due to the additional piping installed in the hydraulic circuit.

4.3. Subsystem CL

The cooling locations CL reveal a differentiated behavior (see Fig. 3b). Cooling location 1 performs best, has an averaged $OPI_{CL,1}$ of approximately -0.04 and is 92% of the time lower than 0. Cooling locations 2, 4, 5 and 7 operate close to the technical requirements with an average OPI of 0.19, 0.16, 0.13 and 0.12, respectively. Accordingly, these cooling locations should be observed closely during further operation. An increase of the key figure would indicate a possible malfunction. In contrast, the cooling location 3 performs worst with an average $OPI_{CL,3}$ of 0.36 and cooling location 6 reveals an averaged $OPI_{CL,6}$ of 0.25. Therefore, both cooling locations show an increased potential for improvement. A possible reason can be a fouled or suboptimal heat exchanger, which results in a decreased cold water distribution temperature to achieve the needed heat transfer, compared to the chosen reference (see subsection 3.3). Also, a different dimensioning might be present in the refrigeration plant compared to the initial design, meaning that the cooling locations supply an air-conditioning application with a different temperature level as assumed in the reference conditions (e.g. an air-conditioning application without dehumidification). As a consequence, the mentioned cooling locations should be

inspected to determine eventual issues and to initiate countermeasures in order to achieve an increased performance.

4.4. Subsystem DC

The subsystem DC operates according to the technical requirements, similarly to the subsystems FC and CST, and exhibits an average OPI_{DC} of -0.87 (see Fig. 3a). Three outliers with a value superior to 0 are present on November 15th as well as December 8th and 21st. However, such outliers should not be overly considered, as only a constantly high or rising key figure is a possible indication of a faulty operation. By investigating the daily exergy sums, the reason for the adequate performance of the subsystem DC is revealed (see Fig. 4).

The exergy input of the circulating pumps (see Fig. 4b) is all the time lower than the reference. Also, the exergy input of the dry cooler fans (see Fig. 4c) is substantially, around factor 2, lower than the reference. This is favorable for the refrigeration plant operation. The actual dry cooler fan input is increased by approximately a factor of 6 the same days the OPI reveals a potential for improvement, and thus, the behavior of the dry cooler fans determines the outcome. The thermal exergy input of the free cooling module (see Fig. 4a) exhibits values according to the technical standards. It is revealed, that a decreased electrical exergy input is more favorable for the subsystem than an adequate temperature level. The reference electrical exergy is roughly factor 2 higher than the reference

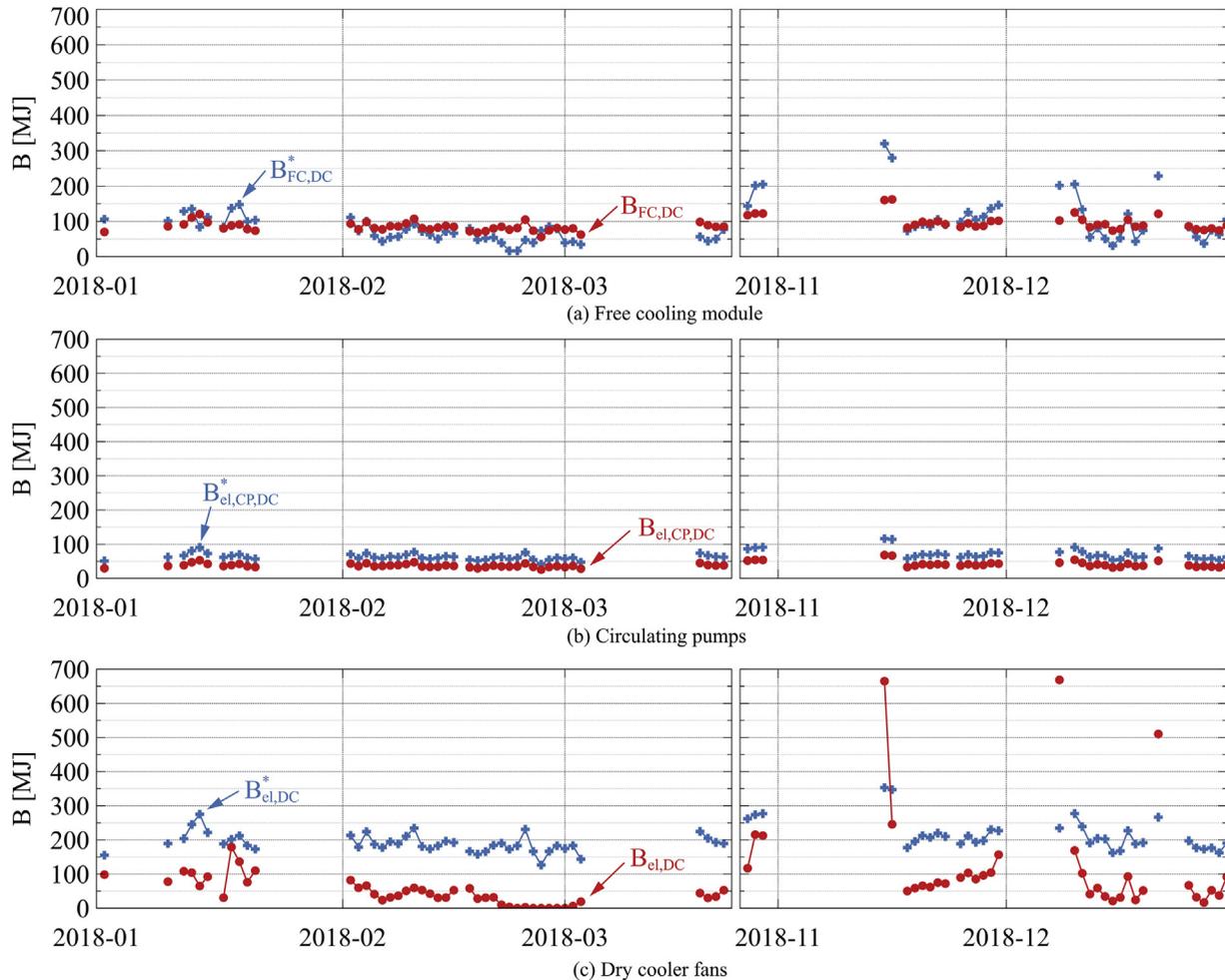


Fig. 4. Daily exergy sum of the subsystem DC input from the free cooling module (a), circulating pumps (b) and dry cooler fans (c).

thermal exergy and, interestingly, the actual electrical exergy inputs of the auxiliary devices exhibit a similar magnitude as the actual thermal exergy input. In energetic considerations, their input would account only for roughly 4.7% of the thermal load according to technical standards. This outcome emphasizes to reduce the auxiliary electrical energy input where possible, e.g. with an adaptive speed control of the circulating pumps and dry cooler fans, in order to achieve a highly performing refrigeration plant.

5. Conclusions and outlook

The introduced optimization potential indices for free cooling operation of refrigeration plants allows an assessment of the performance and the corresponding optimization potential in every subsystem at a glance. The identification of possible optimization potentials is relevant, as measures which improve the system effectiveness most likely prevent frequent shortcomings during refrigeration plant operation. The functionality of the methodology was shown by applying it with experimental data from an existing field plant as a case study. With defined reference values according to the state of the art in technology, an absolute comparison between different refrigeration plants is feasible. The interpretation of the results is straightforward, which is of great importance in practice. Laypersons can easily determine the system operating state and initiate countermeasures by specialist if needed. A simple red/green indicator could be realized to facilitate the monitoring of the plant operation. Furthermore, the exergy-based assessment system helps to sensitize all involved actors, e.g. plant operators, regarding an efficient operation of refrigeration plants. The presented method finds possibly also application in other building energy systems, such as heat pumps.

The investigated field plant reveals a performance according to the technical requirements in general, whereby it can be concluded that the overall design of the hydraulic circuits is adequate and the auxiliary devices properly controlled. Two out of seven cooling locations show significant potential for improvement. The latter could be realized by e.g. ameliorating the heat transfer or including another hydraulic circuit for the cold water distribution on the desired temperature level according to the air-conditioning application. Moreover, the analysis shows the importance of reducing the energy usage of auxiliary devices to achieve an adequate performance of the plant.

Further investigations are needed regarding the determination of the cooling location exergy. As no measurements were available for the present study, an alternative approach was used for the subsystem cooling location to determine the optimization potential index. To allow a consistent definition of the latter, the temperature and the corresponding air humidity of the cooling location circuit (e.g. ventilation system) needs to be measured in the future. This includes the determination of the exergy of humid air, which is typically present in air-conditioning applications or evaporative coolers, e.g. cooling towers. Likewise, if further technical information would be available, a hydraulic analysis could be integrated to the proposed method in order to evaluate the specific flow exergies and to consider occurring pressure losses in detail. While no cost analysis was considered in the present work, it could be carried out in a subsequent evaluation step in order to determine if the found optimization potential is also worthwhile from an economic point of view. Furthermore, the exergy-based assessment system should be applied to other field plants, preferably with a different cooling power and other subsystems such as heat utilization. This helps to identify interrelationships between various refrigeration systems and to use them for future evaluations. Further technical reference values should be determined, as it is currently not possible to estimate which values of the optimization potential index close to the

limit of 0 still represent a permissible operation. Ideally, these limits are specified with statistically sound, representative measurements and with the help of experts. Not only technical and methodical but also strategic aspects have to be considered together with all actors in the industry. This involves long term investigations of different real systems. In addition, future investigations should analyze the influence of the reference environment on the optimization potential index.

CRedit authorship contribution statement

Lorenz Brenner: Methodology, Formal analysis, Investigation, Writing - original draft, Visualization. **Frank Tillenkamp:** Conceptualization, Validation, Writing - review & editing, Supervision, Funding acquisition. **Christian Ghiaus:** Validation, Writing - review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.energy.2020.118464>.

References

- [1] Wu D, Hu B, Wang R. Performance simulation and exergy analysis of a hybrid source heat pump system with low GWP refrigerants. *Renew Energy* 2018;116:775–85. <https://doi.org/10.1016/j.renene.2017.10.024>.
- [2] Menberg K, Heo Y, Choi W, Ooka R, Choudhary R, Shukuya M. Exergy analysis of a hybrid ground-source heat pump system. *Appl Energy* 2017;204:31–46. <https://doi.org/10.1016/j.apenergy.2017.06.076>.
- [3] Razmara M, Maasoumy M, Shahbakhti M, Robinett R. Optimal exergy control of building HVAC system. *Appl Energy* 2015;156:555–65. <https://doi.org/10.1016/j.apenergy.2015.07.051>.
- [4] Fan B, Jin X, Fang X, Du Z. The method of evaluating operation performance of HVAC system based on exergy analysis. *Energy Build* 2014;77:332–42. <https://doi.org/10.1016/j.enbuild.2014.03.059>.
- [5] Fudholi A, Zohri M, Jin GL, Ibrahim A, Yen CH, Othman MY, Ruslan MH, Sopian K. Energy and exergy analyses of photovoltaic thermal collector with ∇ -groove. *Sol Energy* 2018;159:742–50. <https://doi.org/10.1016/j.solener.2017.11.056>.
- [6] Agrawal S, Tiwari G. Energy and exergy analysis of hybrid micro-channel photovoltaic thermal module. *Sol Energy* 2011;85(2):356–70. <https://doi.org/10.1016/j.solener.2010.11.013>.
- [7] Sangi R, Müller D. Exergy-based approaches for performance evaluation of building energy systems. *Sustainable Cities and Society* 2016;25:25–32. <https://doi.org/10.1016/j.scs.2016.04.002>.
- [8] Saloux E, Teyssedou A, Sorin M. Development of an exergy-electrical analogy for visualizing and modeling building integrated energy systems. *Energy Convers Manag* 2015;89:907–18. <https://doi.org/10.1016/j.enconman.2014.10.056>.
- [9] Harrell J, Mathias J. Improving efficiency in a campus chilled water system using exergy analysis. *Build Eng* 2009;115(1):507–22.
- [10] Yang M-H, Yeh R-H. Performance and exergy destruction analyses of optimal subcooling for vapor-compression refrigeration systems. *Int J Heat Mass Tran* 2015;87:1–10. <https://doi.org/10.1016/j.jheatmasstransfer.2015.03.085>.
- [11] Fratzscher W, Brodjanskij VM, Michalek K. *Exergie - Theorie und Anwendung*. Leipzig: Deutscher Verlag für Grundstoffindustrie; 1986.
- [12] Shan K, Wang S, Gao DC, Xiao F. Development and validation of an effective and robust chiller sequence control strategy using data-driven models. *Autom ConStruct* 2016;65:78–85. <https://doi.org/10.1016/j.autcon.2016.01.005>.
- [13] Dong K, Li P, Huang Z, Su L, Sun Q. Research on free cooling of data centers by using indirect cooling of open cooling tower. *Procedia Engineering* 2017;205:2831–8. <https://doi.org/10.1016/j.proeng.2017.09.902>.

- [14] Sorrentino M, Rizzo G, Trifiro A, Bedogni F. A model-based key performance index for energy assessment and monitoring of telecommunication cooling systems. *IEEE Transactions on Sustainable Energy* 2014;5(4):1126–36. <https://doi.org/10.1109/TSTE.2014.2334365>.
- [15] Kazanci OB, Shukuya M, Olesen BW. Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy. *Build Environ* 2016;100:102–13. <https://doi.org/10.1016/j.buildenv.2016.02.013>.
- [16] Al-Bassam E, Alasseri R. Measurable energy savings of installing variable frequency drives for cooling towers' fans, compared to dual speed motors. *Energy Build* 2013;67:261–6. <https://doi.org/10.1016/j.enbuild.2013.07.081>.
- [17] Eisenhauer S, Hauck T, Arnemann M. *Systematische Erstellung und Anwendung messtechnischer Konzepte zur energetischen Untersuchung von Kälteanlagen*. Dresden: DKV-Tagung; 2015.
- [18] Brenner L, Tillenkamp F, Krütli M, Ghiaus C. Optimization potential index (OPI): an evaluation method for performance assessment and optimization potential of chillers in HVAC plants. *Appl Energy* 2020;259. <https://doi.org/10.1016/j.apenergy.2019.114111>.
- [19] Moran MJ, Shapiro HN, Boettner DD, Bailey MB. *Fundamentals of engineering thermodynamics: SI version*. New York: John Wiley; 2010.
- [20] Bejan A, Tsatsaronis GG, Moran MJ. *Thermal design and optimization*. Wiley; 1996.
- [21] Ducoulombier M, Sorin M, Teyssedou A. Thermodynamic bounds for food deep chilling tray tunnel operation. *Int J Therm Sci* 2007;46(2):172–9. <https://doi.org/10.1016/j.ijthermalsci.2006.05.001>.
- [22] Çengel YA, Boles MA. *Thermodynamics: an engineering approach*. Boston: McGraw-Hill; 2005.
- [23] Torío H, Schmidt D. *Annex 49 final report: energy Conservation in Buildings and Community Systems (ECBCS) - low exergy systems for high-performance buildings and communities*. Tech. rep. Stuttgart: Fraunhofer IBP; 2011.
- [24] Martinaitis V, Bielskus J, Januševičius K, Bareika P. Exergy efficiency of a ventilation heat recovery exchanger at a variable reference temperature. *Mechanics* 2017;23(1):70–7. <https://doi.org/10.5755/j01.mech.23.1.17678>.
- [25] Zhou Y, Gong G. Exergy analysis of the building heating and cooling system from the power plant to the building envelop with hourly variable reference state. *Energy Build* 2013;56:94–9. <https://doi.org/10.1016/j.enbuild.2012.09.041>.
- [26] Bonetti V. Dynamic exergy analysis for the built environment: fixed or variable reference?. *Proceedings of the 9th exergy, energy and environment symposium*. Split, Croatia: FESB, University of Split; 2017. p. 924–39.
- [27] Pons M. On the reference state for exergy when ambient temperature fluctuates. *Int J Therm* 2009;12(3):113–21. <https://doi.org/10.5541/IJOT.1034000246>.
- [28] Verein Deutscher Maschinen- und Anlagenbau (VDMA). *Energieeffizienz von Klimakälteanlagen. Teil 8: Komponenten - Wärmeübertrager (VDMA 24247-8)*. Beuth; 2011.
- [29] Swiss Society of Engineers and Architects (SIA). *Lüftungs- und Klimaanlageanlagen - Allgemeine Grundlagen und Anforderungen (SIA 382/1)*. SIA; 2014.