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Catalyzing Cooling Tower Efficiency: A Novel Energy Performance Indicator and Functional Unit including Climate and Cooling Demand Normalization

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Abstract: Energy and climate targets necessitate efficiency indicators to reflect resource-saving potentials. Prevailing indicators for cooling towers, however, often omit the effect of outside conditions. Hence, this study introduces an innovative indicator grounded in the energy efficiency ratio. Our proposed metric is the cost–benefit ratio between electricity demand and the thermodynamic *minimum airflow*. Thus, we call the novel indicator the *airflow performance indicator*. To validate its feasibility, we apply the indicator first to an extensive dataset encompassing 6575 cooling tower models and second to a year-long case study involving a data center's wet cooling system. As a result, the energy performance indicator demonstrates that dry cooling requires eight times more *minimum airflow* at the median than evaporative cooling would, directly correlating to the fan power. Furthermore, efficiency benchmarks derived from the dataset of 6575 cooling tower models provide a comparative assessment of the case study. Defining the *quantified benefit* as minimum airflow additionally underscores the limitations of free cooling as the wet cooling system only partly covers the cooling demand, requiring chillers additionally. In conclusion, the indicator empowers the identification of energy-saving potentials in the selection, design, and operation of cooling towers. Moreover, the functional unit definition provides a foundation for future life cycle assessments of cooling towers, enhancing cooling tower efficiency and sustainability.

Keywords: key performance indicator; energy performance indicator; energy efficiency; environmental management; environmental impact; life cycle assessment; functional unit; physical optimum; free cooling; data center

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1. Introduction

The Sustainable Development Goals (SDGs) aim to 'double the global rate of improvement in energy efficiency' by 2030 [1] (p. 19). Regarding climate change and resource depletion, energy efficiency and environmental assessment methods generally reveal the most advantageous options. However, existing methods for cooling towers still need refinement, especially considering the increase in global cooling demand [2,3] due to factors such as the rise in data centers [4–6] and ambient temperatures [7].

The term 'efficiency' generally refers to the ratio of benefit to cost. Energy efficiency is mostly the 'ratio or other quantitative relationship between an output of performance [. . .], service, goods, commodities, or energy [. . .], and an input of energy' [8] (p. 7). The electricity input to a cooling tower is precisely measurable. However, quantifying the valuable output of the cooling tower is challenging.

Two main challenges arise in this context. Firstly, the key function of a cooling tower is dissipating heat into the environment by consuming electricity. Cooling towers consume exergy (electricity) to destruct exergy rather than producing useful exergy outputs, which is the opposite of how resource efficiency is typically defined. Exergy efficiency is typically the ratio between useful exergy outputs and total exergy inputs [9] (p. 135). Secondly, the heat transfer depends on the required cooling temperatures and the dynamic ambient humidity and temperature, which must be considered by an efficiency indicator. Therefore, a practical concept for evaluating cooling towers with a consistent quantification of the cooling tower benefit is needed to reveal and unlock efficiency potentials.

1.1. Previous Literature and Indicators

Indicators that assess the thermal effectiveness of cooling towers have been widely applied for decades [10–13]. However, these indicators, such as the Merkel number [14], the effectiveness Number of Transfer Units, 'e-NTU' [15], and the Merkel number refined by Poppe [16,17], quantify heat transfer effectiveness, independent of resource consumption and the required cooling service. Hence, these indicators cannot evaluate efficiency within an energy or environmental management system. Hitherto, efficiency indicators for cooling towers are inconsistently applied in typical energy and environmental assessment methods.

The efficiency of cooling towers is mostly represented by the ratio between consumed electricity and dissipated heat load in kW_{el}/kW_{th}. This ratio is called 'specific direct energy consumption' [18] (p. v) or specific electricity consumption [19] (p. 64), including the electrical final energy consumption, the nominal heat flow, the utilization ratio, and the operation hours. Nevertheless, these two parameters do not reflect the effect of temperature and ambient humidity.

As the inverse ratio, the energy efficiency ratio (EER) and the coefficient of performance (COP) generally represent the ratio between equipment (cooling or heating) capacity and the total power input 'at any given set of rating conditions' [20–22]. These indicators also do not account for ambient conditions.

Life cycle assessment (LCA) quantifies the 'eco-efficiency' as the environmental impacts relative to a defined 'functional unit'. The functional unit is the 'quantified performance of a product system for use as a reference unit' [23] (p. 4). As the functional unit definition is required per standard, previous studies have incorporated it in the LCA of cooling towers. However, these definitions are inconsistent, hampering the comparison between different cooling tower systems (cf. Table 1).

Furthermore, previous studies inconsistently defined the exergy efficiency of cooling towers [24–26]. For example, some studies ([24] (p. 190), [27] (p. 504)) use the exergy input–output ratio as an efficiency indicator for cooling towers based on Bejan [28] (p. 104).

Similar to the exergy analysis, the 'Physical Optimum' method introduced by Volta and Weber [29] serves to evaluate processes against a thermodynamically ideal reference process. The 'physical optimum factor' represents the ratio of 'the real amount of energetic expense [. . .] to the corresponding physical minimum expense' [29] (p. 3). No studies on the physical optimum of cooling towers have been conducted to date, except for investigating water consumption in the physical optimum of wet cooling towers [30] (p. 55). Table 1 outlines the various interpretations of cooling tower efficiency as a cost–benefit or benefit–cost ratio.

Table 1. Previous and related definitions of efficiency and quantified benefit of cooling towers in previous studies and standards. As efficiency is a cost–benefit ratio, the cost definition addresses the effort or valuable input, and the benefit definition represents the valuable output or service.

	Efficiency Indicator	Cost Definition (Effort)	Benefit Definition (Use)	Reference
Energy Management	Energy Efficiency	input of energy	output of performance, service, goods, commodities, or energy	ISO 50001 [8] (p. 7)
Best Available Techniques	Specific Direct Energy Consumption	energy consumed by all energy-consuming equipment	dissipated MW_{th}	[18] (p. 4)

Table 1. *Cont.*

	Efficiency Indicator	Cost Definition (Effort)	Benefit Definition (Use)	Reference
Efficiency of Buildings	Specific Electricity Consumption	electricity consumption (kWh_{el})	thermal load (MW_{th}) 2219 operation hours [h]	DIN V 18599-7 [19] (p. 64)
EER	Energy Efficiency Ratio	'effective power input [. . .] to the device' (W)	'total cooling capacity', 'at any given set of rating conditions' (W)	ISO 13253 [21]
LCA Eco-Efficiency	Environmental Impacts Per Functional Unit	environmental impacts	functional unit: '1 kWh of electricity produced by the plant, [. . .] referred to the operating period of 1 year'	ISO 14040 [23] ISO 14044 [31], [32] (p. 1079)
		environmental impacts	'provision of 1-megawatt heat rejection (cooling) capacity (MW_{th}) for a period of 1 year'	[33] (p. 50)
		environmental impacts	'cooling of 1 kg water from 35 to 28 °C in Germany for the overall usage time'	[34] (p. 140)
		environmental impacts	'cooling throughout 2019 with 2,450,000 m^3 of circulating water cooled from 21.21 to 14.87 °C [. . .] in Stuttgart-Vaihingen'	[35] (p. 3)
Exergy Analysis	Exergy Efficiency η_B	η_B = exergy output / exergy input = 1 − exergy destruction / exergy input		[24] (p. 190), [27] (p. 504)
		η_B = change in product exergy / change in supply exergy		[24] (p. 191)
		η_B = (air exergy output + input) / (water exergy input − output)		[26] (p. 2797)

The cost quantification is mainly consistent, representing the electricity consumption or LCA's environmental impacts. In contrast, the definition of benefit is inconsistent: either the amount of heat removal, omitting outside conditions, or specific definitions of heat load, cooling temperatures, and location, which are likewise incomparable. Definitions for the exergy efficiency of cooling towers are different from the cost–benefit concept but refer to the thermodynamic input–output balance, and the electricity consumption is mostly omitted. Beyond that, several studies outline the landscape of methods and indicators for environmental and energy efficiency evaluation [36–38].

The specific research gap that our study aims to address is the inconsistency of the *quantified benefit* of cooling towers. Hitherto, this research gap hampers the efficiency evaluation and comparability between cooling towers.

1.2. Objectives of This Work

This study aims to introduce an expedient indicator for quantifying the energy efficiency of cooling towers, considering that heat removal processes differ in varying ambient conditions and technical requirements. For example, the ambient temperature and humidity are crucial factors influencing the heat removal. Building upon the concept of the EER and specific electricity consumption, we implement the thermodynamic ideal as the efficiency reference point, allowing for targeted comparability between different cooling tower processes. For validation, we apply the novel indicator to a comprehensive dataset, including 6575 cooling tower models by Wenzel et al. [39]. Moreover, as a case study, we evaluate the efficiency of a data center's wet cooling system by analyzing the logged data for 2019. Subsequently, discussing the indicator's usefulness in interpreting the results is crucial to this study.

2. Materials and Methods

This section introduces how we quantify the cooling tower's benefit, effectiveness, and efficiency (Section 2.1). Subsequently, we present the system definition, the dataset comprising various cooling tower models, and the case study data in Section 2.2.

2.1. Efficiency Evaluation

As a starting point, we look at established energy efficiency indicators for other technical processes, which often refer to product mass. The Energy Performance Indicator (EnPI) used within energy management systems can be defined as [40] (p. 22)

$$EnPI = \frac{energy\ input}{product\ mass}. \tag{1}$$

However, cooling towers do not produce a valuable output measurable in mass or energy. To still assess the efficiency of cooling towers, we begin by investigating their fundamental thermodynamic function.

2.1.1. The Quantified Benefit of Cooling Towers

As discussed in Section 1, a cooling tower primarily serves to lower the temperature of a coolant flow from a specific inlet temperature to a required outlet temperature. The removed heat dissipates into the surrounding environment, involving exergy destruction.

Fundamentally, the cooling tower transfers heat from the coolant to an ambient airflow. Thus, the predominant task of cooling towers is to ensure heat transfer between the coolant and ambient air. For this purpose, the cooling tower must induce airflow movement, which they achieve with fans or natural draft. Figure 1 illustrates the fundamental principle of the cooling tower. In the case of evaporative cooling, mass is transferred to the airflow. Wet cooling towers have either an open or a closed coolant circuit. For the latter, additional spray water enters and leaves the system boundary, while the coolant circuit is closed with no mass transfer to the airflow. For simplification, the Figure summarizes the coolant and additional spray water in the case of closed wet cooling, although the masses and temperatures of both fluids differ.

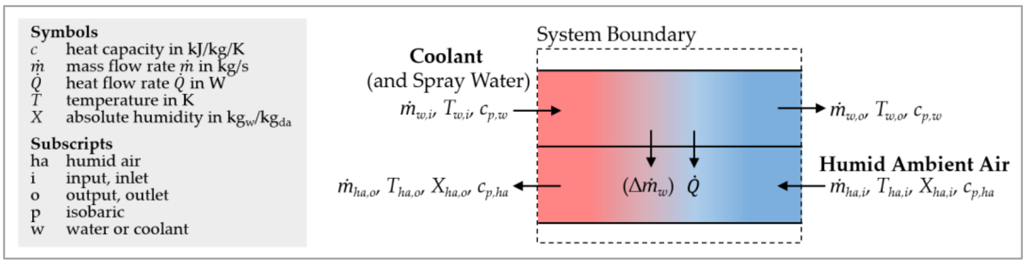

Figure 1. Energy balance of a cooling tower, lowering the coolant inlet temperature $T_{w,i}$ by heat transfer $\dot{Q}$ (and mass transfer $\Delta\dot{m}_w$ in case of evaporative cooling) to the ambient air. The coolant mass flow $\dot{m}_{w,i}$ with isobaric heat capacity $c_{p,w}$ cools down from the inlet temperature $T_{w,i}$ to the outlet temperature $T_{w,o}$. The inlet mass flow of ambient air $\dot{m}_{ha,i}$ has ambient temperature $T_{ha,I}$ absolute moisture $X_{ha,i}$, and the isobaric heat capacity $c_{p,ha}$.

The energy balance of the system comprises the enthalpy difference between inputs (i) and outputs (o) of humid air (ha) and the coolant (w). The enthalpy H depends on the mass flow $\dot{m}_x$ in kg/s, the isobaric heat capacities $c_{p,x}$ in kJ/kg/K, the absolute moisture of humid air X in $\mathrm{kg_w/kg_{da}}$, and the temperatures T_x in K. For the fluid properties, we use the open-source thermophysical property library *CoolProp* [41]. Based on the dry air mass flow $\dot{m}_{da}$, the coolant inlet mass flow $\dot{m}_{w,i}$, the enthalpy values per mass h_x, the coolant's isobaric heat capacity $c_{p,w}$, the coolant outlet temperature $T_{w,o}$, and the absolute moisture X, the complete energy balance between air and coolant of open wet cooling in counterflow is [42] (p. 1492)

$$\dot{m}_{da}[h_{ha,o} - h_{ha,i}] = \dot{m}_{w,i}(h_{w,i} - h_{w,o}) + \dot{m}_{da}c_{p,w}T_{w,o}(X_o - X_i). \tag{2}$$

For dry cooling, the absolute humidity X in kg_w/kg_{da} of inlet and outlet air is constant, meaning no mass transfer. Based on Equation (2) and the removed heat load $\dot{Q}$ in W, the mass flow rate of dry air $\dot{m}_{da}$ in kg/s through an open cooling tower amounts to

$$\dot{m}_{da} = \frac{\dot{m}_{w,i}(h_{w,i}(T_{w,i}) - h_{w,o}(T_{w,o}))}{h_{ha,o} - h_{ha,i} - c_{p,w}T_{w,o}(X_o - X_i)} = \frac{\dot{Q}}{h_{ha,o} - h_{ha,i}}. \tag{3}$$

Based on the idea of the physically ideal reference process in exergy analysis and the physical optimum method (cf. Section 1.2), there is a physically ideal situation in which the cooling tower serves its air-moving purpose with minimum effort: The heat load is theoretically removable with a specific minimum airflow rate in the idealized case, where the following applies:

- The cooling tower operates at pure counterflow.
- The air outlet temperature equals the coolant inlet temperature: $T_{ha,o} = T_{w,i}$.
- The cooling tower uses evaporative cooling.
- The outlet air is 100% saturated: $X_o = X''(T_{w,i})$.
- Nevertheless, the ambient wet-bulb temperature (WBT) must be less than the required cooling temperature, $WBT_{ha,i} \leq T_{w,o}$.

In this ideal situation, the minimum mass flow rate of counter-flowing dry air $\dot{m}_{min}$ in kg/s results from Equation (3) with an outlet air temperature that equals the coolant inlet temperature $T_{ha,o} = T_{w,i}$ and saturated outlet air and absolute moisture $X_o = X''(T_{w,i})$, and

$$\dot{m}_{min} = \frac{\dot{m}_{w,i}(h_{w,i}(T_{w,i}) - h_{w,o}(T_{w,o}))}{h_{ha,o}(T_{w,i}, X'') - h_{ha,i} - c_{p,w}T_{w,i}(X'' - X_i)} = \frac{\dot{Q}}{h_{ha,o}(T_{w,i}, X'') - h_{ha,i}}. \tag{4}$$

This *minimum airflow* is the *quantified benefit* of a cooling tower and serves as an efficiency reference or functional unit, marking the physical ideal. No minimum energy demand is derivable because the mechanism for achieving the mass flow in cooling towers, whether through fans or natural draft, depends on the specific technical setup. Hence, as the minimum power demand can potentially be zero with a natural draft, the physical minimum effort cannot be quantified in energy units but can only be defined as the required *minimum airflow*. Furthermore, the *minimum airflow* meets the requirements for the functional unit definition in LCA, which, however, is not part of this study.

2.1.2. The Effectiveness Indicator

As part of the efficiency evaluation, we must regard how effectively the cooling tower achieves the desired heat transfer because the actual values differ from the nominal values. Regarding the operation in practice with varying ambient conditions, we examine to what extent the cooling tower matches the manufacturer's nominal (n) heat transfer values (nominal heat load Q_n, nominal volume flow rate $V_{w,n}$, nominal coolant inlet temperature $T_{w,i,n}$, and nominal coolant outlet temperature $T_{w,o,n}$). If the ambient WBT approaches the required coolant outlet temperature $T_{w,o,n}$, the targeted cooling temperature is hardly achievable—or not at all if the WBT exceeds the required cooling temperature. As a result, the real (r) coolant outlet temperature $T_{w,o,r}$ increases, and the targeted cooling service remains unachieved. In this case, the effectiveness, generally defined as the actual–nominal ratio of actual to targeted benefit, is below 100%.

Figure 2 illustrates the discrepancy between the nominal *quantified benefit* with nominal technical and environmental conditions against the actual *quantified benefit*, which results from the actual heat transfer and ambient conditions.

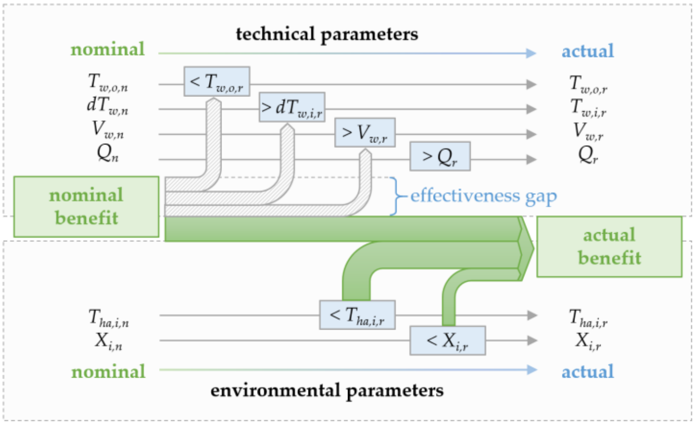

Figure 2. Difference between the nominal (n) and the actual (r) benefit, quantified as *minimum airflow*, due to the actual technical and environmental parameters. Environmental parameters include ambient temperature $T_{ha,i}$ and absolute humidity X_i. Lower heat transfer values imply an effectiveness gap. The technical parameters include the nominal coolant inlet and outlet temperatures $T_{w,i,n}$ and $T_{w,o,n}$, the volume flow rate $V_{w,n}$, the resulting nominal heat load Q_n, and the corresponding actual values (r).

Regarding the environmental parameters, the *quantified benefit* increases with higher ambient temperature $T_{ha,r}$ and absolute humidity $X_{i,r}$ compared to the design values (n). Conversely, the *quantified benefit* decreases if the ambient temperature and humidity are lower than the nominal values. We omit ambient pressure because it barely affects the mass flow rate of air, unlike the volume flow rate.

Less heat transfer than the target values results in a decline of actual *quantified benefit* (for example, the real coolant outlet temperature is less than the nominal, $T_{w,o,r} < T_{w,o,n}$). However, the *quantified benefit* would stay the same if the required technical values were exceeded because more benefit than targeted would be useless, as in general. In that case, the *minimum airflow* would be calculated from the nominal technical parameters with the actual ambient parameters.

If the technical parameters remain unachieved, an 'effectiveness gap' is implied. In contrast, changes in the environmental parameters are considered in the *quantified benefit*. If technical and environmental parameters vary in opposite directions, they will offset each other accordingly.

Regarding the cooling tower efficiency, the *minimum airflow* is calculated from the actual technical and environmental values, although the actual benefit might be less effective than the target values, leaving an *effectiveness gap*. Consequently, an indicator of the effectiveness must also demonstrate the need for additional equipment, such as chillers.

For this purpose, we calculate the effectiveness as the ratio of actual (r) *quantified benefit* to the airflow that would be required for the target values with actual environmental parameters. As long as the *WBT* is smaller than the nominal coolant outlet temperature $T_{w,o,n}$, the ratio is

$$Effectiveness = \frac{\dot{m}_{da,min}\left(\dot{Q}_r, T_{w,i,r}, T_{w,o,r}, T_{amb,r}, X_{amb,r}\right)}{\dot{m}_{da,min}\left(\dot{Q}_n, T_{w,i,n}, T_{w,o,n}, T_{amb,r}, X_{amb,r}\right)}. \tag{5}$$

The nominal cooling temperature will be physically unachievable if the WBT exceeds the required cooling temperature. In this case, calculating the *minimum airflow* with Equation (4) is inapplicable and would deliver invalid values. Thus, the effectiveness is zero as the cooling target is unachievable if *WBT* is higher than the nominal coolant outlet temperature $T_{w,o,n}$.

For that case, we additionally calculate the difference between actual and nominal heat load to highlight the heat removal, which reduces the cooling demand on other cooling equipment:

$$Effectiveness\ gap = \dot{Q}_n - \dot{Q}_r. \tag{6}$$

In the following, we include both the 'effectiveness' as the minimum airflow ratio (Equation (5)) and the 'effectiveness gap' as the difference between the actual and nominal heat load (Equation (6)).

2.1.3. The *Airflow Performance Indicator* (*AirPI*)

The *minimum airflow* as the *quantified benefit* fills the gap in the cost–benefit equation Equation (1) as EnPI to quantify the cooling tower efficiency. The *minimum airflow* includes the actual technical and ambient parameters (cf. Section 2.1.2). We call the novel EnPI the *airflow performance indicator (AirPI)* for cooling towers, calculated as the ratio between electricity demand and the *minimum airflow*:

$$AirPI = \frac{cost}{benefit} = \frac{P}{\dot{m}_{min}}. \tag{7}$$

Besides the instantaneous efficiency assessment, integrating electricity demand and *minimum airflow* over time serves, for example, as an annual benchmark, similar to the seasonal energy efficiency ratio.

2.2. *System and Data*

The efficiency evaluation method applies not only to the typical hyperboloid-shaped cooling towers with natural draft but also predominantly to the significantly smaller serial package-type recirculating systems with nominal powers below 200 MW_{th}. These cooling towers are smaller than 20 m in height and stand on an area of less than 400 m^2 [43] (p. 26). Furthermore, unlike natural-draft cooling towers, package-type cooling towers require forced ventilation.

Cooling towers have different designs to serve diverse requirements across sectors. We classify the following six types of cooling towers in line with Wenzel et al. [39] (cf. Table A1 in the Annex for the detailed definitions):

1. Dry cooling towers;
2. Wet cooling towers with open circuits (direct);
3. Wet cooling towers with closed circuits (indirect);
4. Hybrid cooling towers with direct wetting;
5. Hybrid cooling towers with spraying devices;
6. Hybrid cooling towers with wetting mats.

2.2.1. System Definition

Figure 3 illustrates the system generalized for the six types of cooling towers. Accordingly, the pumps are outside the system to investigate the cooling tower system independently of the hydraulic integration because inefficiencies of the coolant pipe system cannot be assigned to the cooling tower but must be addressed separately.

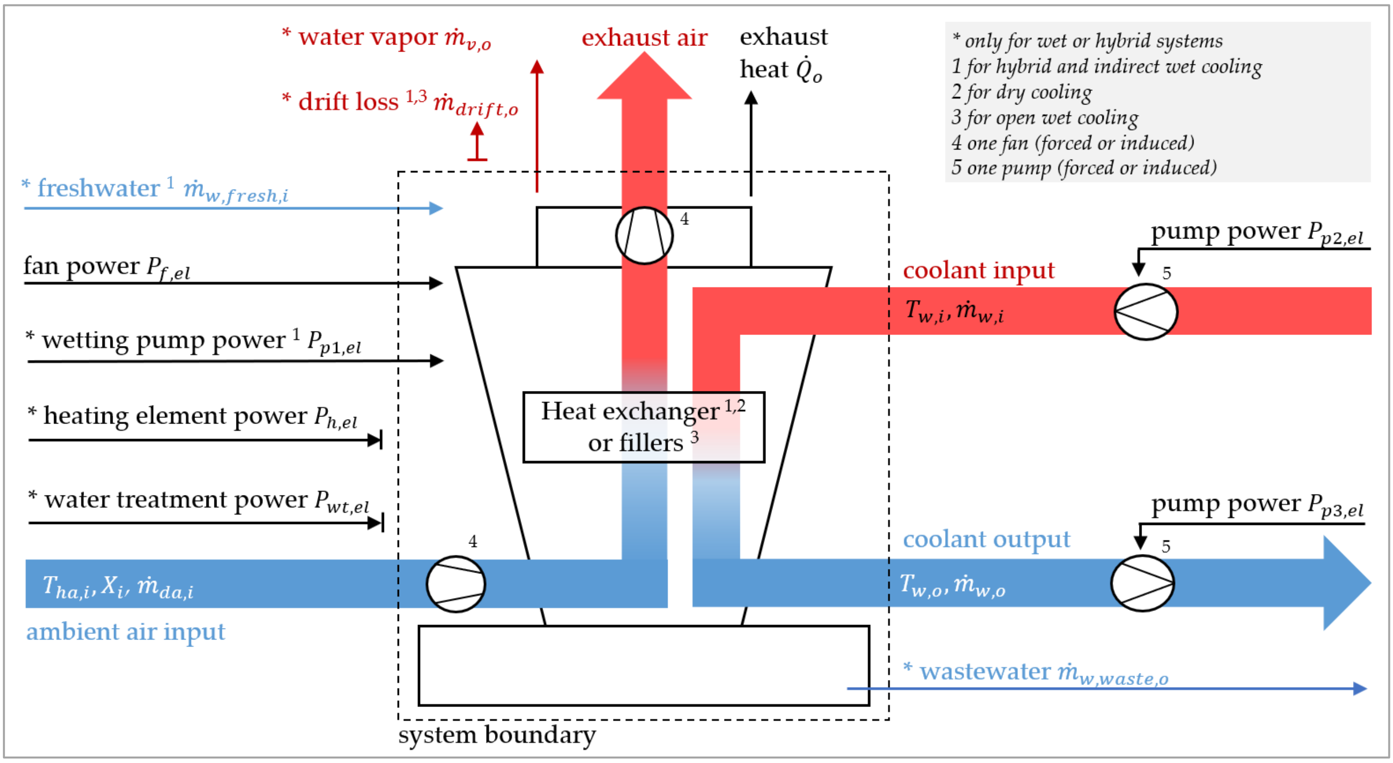

Figure 3. Generalized system boundary of different cooling tower types, based on Wenzel et al. [39]. The cooling tower transfers heat from the coolant flow to the ambient airflow.

2.2.2. Dataset of Cooling Tower Models

Firstly, we use the *AirPI* to assess the cooling towers in the dataset by Wenzel et al. [39] to cover a broad range of different cooling tower models. The data include the six types of cooling towers described in Section 2.2. The dataset comprises 11,338 data points of 6730 cooling tower models by 27 manufacturers. Figure 4 provides an overview of the data availability and gaps.

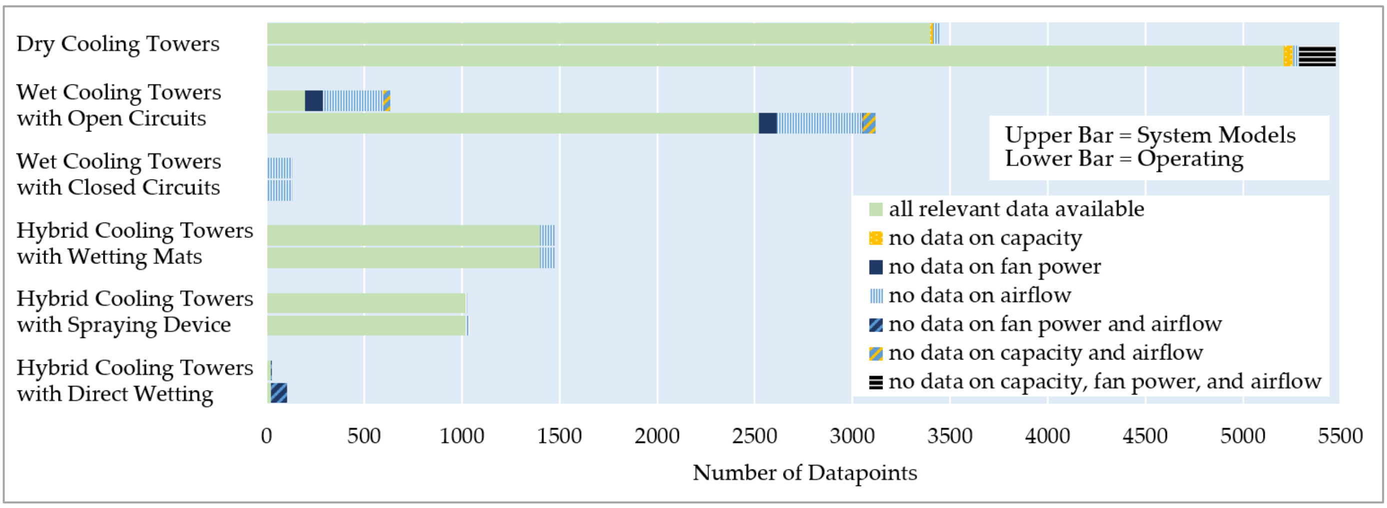

Figure 4. Dataset of cooling tower models based on manufacturer data [39] (p. 5). Table S1 in the Supporting Material provides the underlying data.

This study focuses on dry cooling towers, open wet cooling towers, and hybrid cooling towers with either wetting mats or spraying devices due to the lack of data for closed wet and directly wetted hybrid cooling towers. The data for these four cooling tower types comprise 11,101 data points and 6575 cooling tower models. We calculate the *minimum airflow* for all data points using Equation (3) and compare the *minimum airflow* to the nominal and theoretical airflow without evaporation. Subsequently, we calculate the *AirPI* for all cooling tower models based on the *minimum airflow*. Table S1 in the Supporting Material provides all corresponding data.

For the statistical evaluation, we show the data distribution through boxplot diagrams. These diagrams illustrate a specific parameter's median, the range between minimum and maximum values (whiskers), and the interquartile range (IQR). The IQR spans the data between the quartiles of 25% and 75%. Outliers are those values that are further away from the IQR than 1.5 times the IQR. The whiskers reach the farthest data points that are not yet outliers.

2.2.3. Case Study Data

Besides the dataset presented in Section 2.2.2., we use the *AirPI* to assess a wet cooling system throughout one year, including varying ambient and heat transfer parameters. This case example is part of the cooling system of the High-Performance Computing Center (HLRS) of the University of Stuttgart. Accordingly, the wet cooling system is located in Stuttgart-Vaihingen in Germany, at latitude 48.739, longitude 9.097, and 449 m above sea level.

The wet cooling system comprises four wet cooling towers. Each cooling tower has a nominal capacity of 1.2 MW_{th}, resulting in a total capacity of 4.8 MW_{th} [44] (p. 32). The heat transfer is in counterflow with forced-draft ventilation. Each cooling tower has two 22 kW fans, which are speed-controlled based on temperature as the control variable. The entire wet cooling system is connected to two 22 kW pumps located outside our system boundary (cf. Figure 3).

The data on the cooling system are provided by Bayer et al. [45]. The temporal resolution of the data is 5 min intervals. The relative humidity measured at the HLRS varies between 16% and 100%, with an average of 78%; the DBT amounts to –9 °C minimum, +34.9 °C maximum, and +10.0 °C on average [45]. Table 2 presents the nominal and actual heat transfer data.

Table 2. Cooling tower data at nominal condition with 5.5 °C of WBT and the minimum, maximum, and average of the operating data measured in 2019.

Parameter	Unit	Nominal Values	Actual Values (Min–Max; Avg)
coolant inlet temperature	K	20	7.9–29.2; 21.2
coolant outlet temperature	K	10	8.0–28.1; 14.9
coolant volume flow rate	m^3/h	412.8	0–413.9; 279.6
cooling capacity	kW_{th}	4 800	0–3 672; 2 042
		Ref: [46] (p. 4)	Ref: [45]

3. Results and Discussion

Before assessing the efficiency of the dataset's cooling towers and the wet cooling system as a case study, we focus on the general factors influencing the benefit, quantified as the theoretical *minimum airflow*. This *minimum airflow* is proportional to the removed heat load cf. Equation (3). Nevertheless, the *minimum airflow* also depends on the required cooling temperatures and the outside conditions. For example, Figure 5 plots the *minimum airflow* for a heat removal of 4.8 MW_{th} with a 20 °C inlet and 10 °C outlet temperature of the coolant.

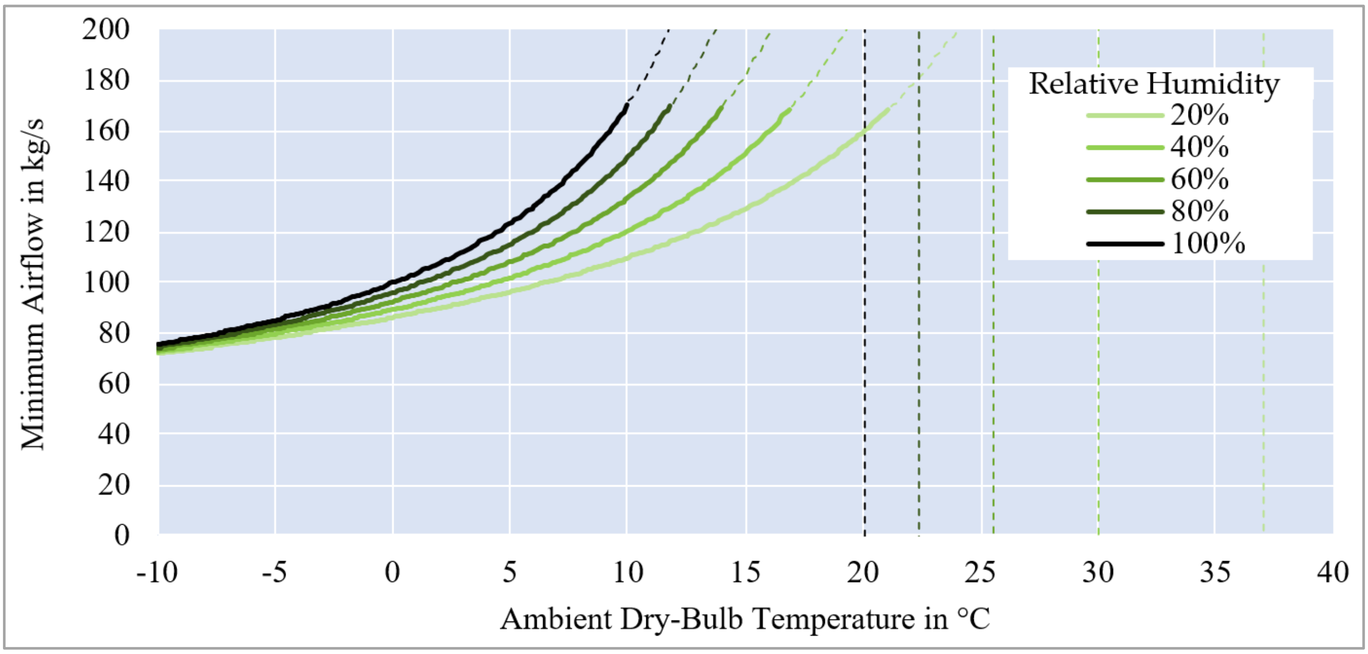

Figure 5. Theoretical *minimum airflow* for 4.8 MW_{th} heat removal from 20 °C to 10 °C required coolant temperature depending on the outside temperature and humidity. Table S2 in the Supporting Material provides the underlying data.

Figure 5 demonstrates that the required *minimum airflow* increases with rising ambient temperature or humidity. If the ambient WBT exceeds the required cooling temperature of 10 °C, the required cooling temperature and the defined benefit will be unreachable against the temperature gradient. The graph highlights the impossibility of benefit achievement by dashed lines above 10.0, 11.8, 14.1, 17.0, and 21.1 °C DBT, respectively, for each line of constant humidity. For example, with 100% saturated ambient air, where the WBT equals the DBT, the required cooling temperature is unachievable above 10 °C DBT. In contrast, 20% saturated air can have up to 21.1 °C DBT, which equals 10 °C WBT, to theoretically still reach the required cooling temperature. In practice, the cooling approach is the additional temperature difference necessary to reach the required cooling temperature.

Moreover, each asymptote at 20.0, 22.5, 25.6, 29.9, and 36.8 °C is the DBT of 20 °C WBT for each humidity content. For example, 20%-saturated air with a WBT of 20 °C has 36.8 °C DBT. Mathematically, the curves of *minimum airflow* increase to infinite with an asymptote, where the ambient WBT exceeds 20 °C of coolant outlet temperature and delivers negative values above these asymptotes.

As this example shows, quantifying the benefit as *minimum airflow* includes the heat load, cooling temperatures, and the varying ambient parameters. At the same time, this *benefit quantification* highlights the ambient states where it is impossible to achieve the targeted benefit. Hence, the *minimum airflow* indicates these additional aspects, whereas the heat amount or flow as the standard efficiency reference does not. In the following, we apply this *quantified benefit* to test its feasibility for the efficiency assessment.

3.1. Efficiency Evaluation of Different Cooling Tower Models

In the first step, we focus again on the *minimum airflow* as the *quantified benefit*. Figure 6 displays the theoretical *minimum airflow* (in green) for the dataset of cooling tower models introduced in Section 2.2.2. In addition, the graph demonstrates the theoretical minimum airflow without evaporation (in yellow) and the nominal airflow (in orange) stated by the manufacturer for the nominal technical parameters.

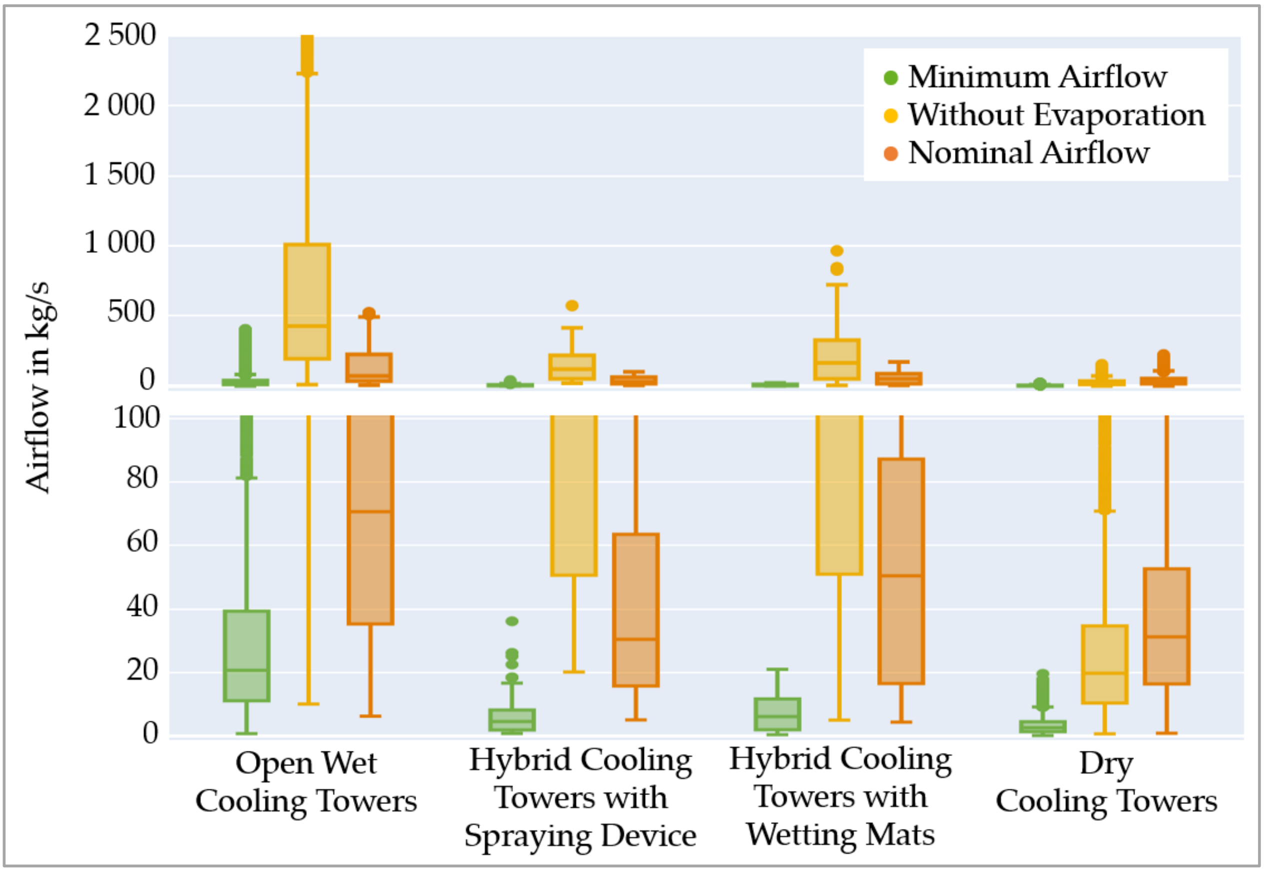

Figure 6. *Minimum airflow* (green) compared to the theoretical minimum airflow without evaporation (yellow) and the nominal airflow (orange) of each operating point in the cooling tower dataset. Each type's data consist of 1035 to 5475 operating points (cf. Figure 4). Table S3 in the Supporting Material provides the underlying data.

Figure 6 addresses multiple aspects. Firstly, we compare the *minimum airflow* (green) across the four types. Accordingly, the median of wet cooling with 20.6 kg/s *minimum airflow* is significantly larger than that of dry cooling with 2.5 kg/s *minimum airflow*. Hybrid cooling with wetting mats or spraying devices provides 6.1 kg/s or 4.5 kg/s *minimum airflow* at the median, respectively. This comparison by *minimum airflow* correlates to the heat loads but also includes the cooling temperatures and environmental conditions. In conclusion, higher cooling demands in terms of greater heat loads, lower cooling temperatures, or higher ambient temperatures necessitate higher *minimum airflow*, which can be supplied by wet cooling towers or multiple units of models with a smaller *minimum airflow*.

Secondly, the graph underscores the energy-saving potential of evaporative cooling, comparing the *minimum airflow* (green) and the minimum airflow without evaporation (yellow). The difference is the additional flow required if no evaporation is applied. On the one hand, we notice that the cooling service of wet cooling would require significantly more airflow if evaporation was not applied: 21 times more at the median. However, some operating points would be inaccessible to dry cooling because the given operating points of wet and hybrid cooling partly include cooling temperatures below the DBT. On the other hand, we find the energy-saving potential of dry cooling towers if evaporative cooling was applied to these operating points. Regarding the medians, the minimum airflow without evaporation (yellow) of dry cooling towers is 7.8 times the *minimum airflow* (green). In conclusion, evaporative cooling would require significantly less airflow by a factor of 7.8 for these operating points, which directly correlates to the fan power. Moreover, wet and hybrid cooling towers operate closer to the WBT than dry cooling towers, as evaporation is more significant to the *minimum airflow* at smaller cooling approaches (cf. Figure 5).

Thirdly, we compare the nominal airflow (orange) to both *minimum airflow* (green) and minimum airflow without evaporation (yellow). The nominal airflow is always greater than the *minimum airflow* (green), marking the physical ideal. Furthermore, as expected, the nominal airflow of dry cooling is larger than the theoretical minimum airflow without evaporation (yellow), marking the physical ideal without evaporation. In contrast, wet cooling towers apply evaporative cooling so that the nominal airflow is smaller than the theoretical minimum airflow without evaporation (yellow) because the heat transfer is based not only on convection but also on evaporation. Hybrid cooling applies evaporation in the case of higher environmental temperatures, so the nominal airflow is partly smaller than the minimum airflow without evaporation.

The ratio of nominal to *minimum airflow* already indicates the efficiency as part of the *AirPI*. Inefficiencies result from imperfect heat transfer or the absence of evaporation. For example, the outlet temperature of air is smaller than the coolant inlet temperature in practice. This effect increases in the case of cross-flow arrangement compared to counterflow heat transfer.

In the next step, we examine the efficiency of the cooling tower models with the *AirPI* introduced in Section 2.1.3., which is the ratio of fan power to *minimum airflow* (with evaporation). Figure 7 visualizes the *AirPI* of each type as boxplots and over the nominal heat load of each cooling tower type.

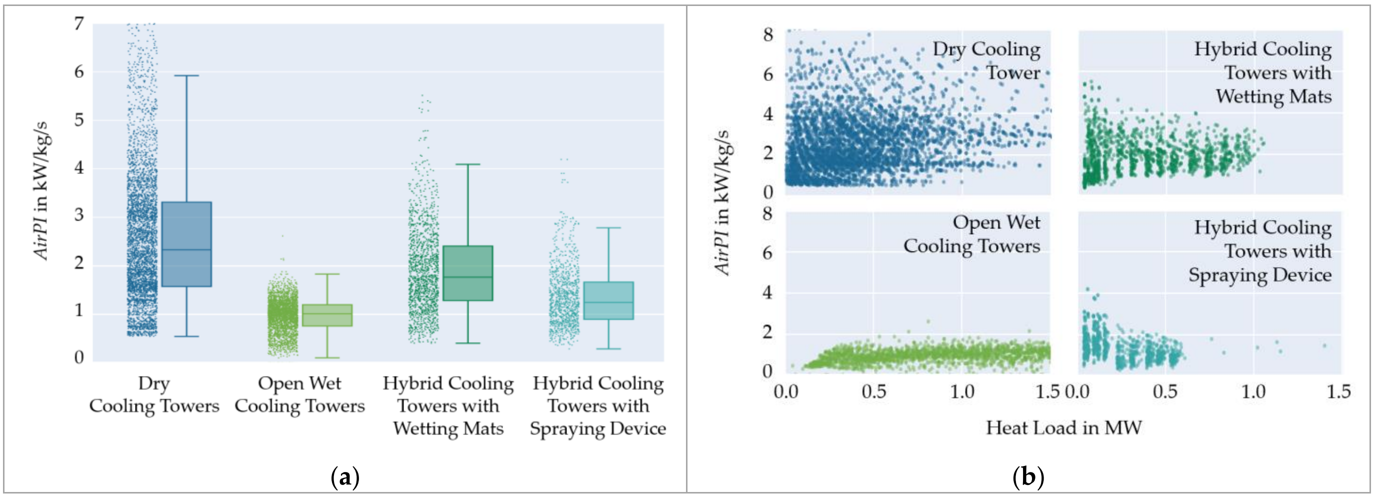

Figure 7. *AirPI* results for each cooling tower type (**a**) as boxplot diagram and (**b**) over the thermal capacity below 1.5 MW_{th}. Each type's data consist of 1035 to 5475 operating points (cf. Figure 4). Table S4 in the Supporting Material provides the underlying data.

As Figure 7a visualizes, the *AirPI* values of the four cooling tower models are approximately between 0.1 and 6.0 kW/kg/s, except for outliers. Dry cooling towers have a higher median of 2.3 kW/kg/s than open wet cooling towers with 1.0 kW/kg/s, hybrid cooling towers with wetting mats with 1.8 kW/kg/s, and hybrid cooling towers with spraying devices with 1.3 kW/kg/s. These values apply only to the specific operating points of the dataset. Nevertheless, the *AirPI* reflects the expected efficiency proportions between the cooling tower types. Dry cooling is the least efficient due to no evaporation. In contrast, wet cooling towers are most efficient due to continuous evaporation. Hybrid cooling lies between wet and dry cooling efficiency because evaporation is applied for part of the year. Hybrid cooling towers with spraying devices are more efficient than the ones with wetting mats because the mats cause higher pressure losses.

Furthermore, regarding *AirPI* values over the nominal heat load, the *AirPI* values of dry and hybrid cooling towers slightly decrease with increasing capacity. For wet cooling, no correlation is observed.

For comparison, Figure 8 shows the specific electricity consumption by DIN V 18599-7 [19] (p. 64) to discuss the added value of the *AirPI* as a novel EnPI.

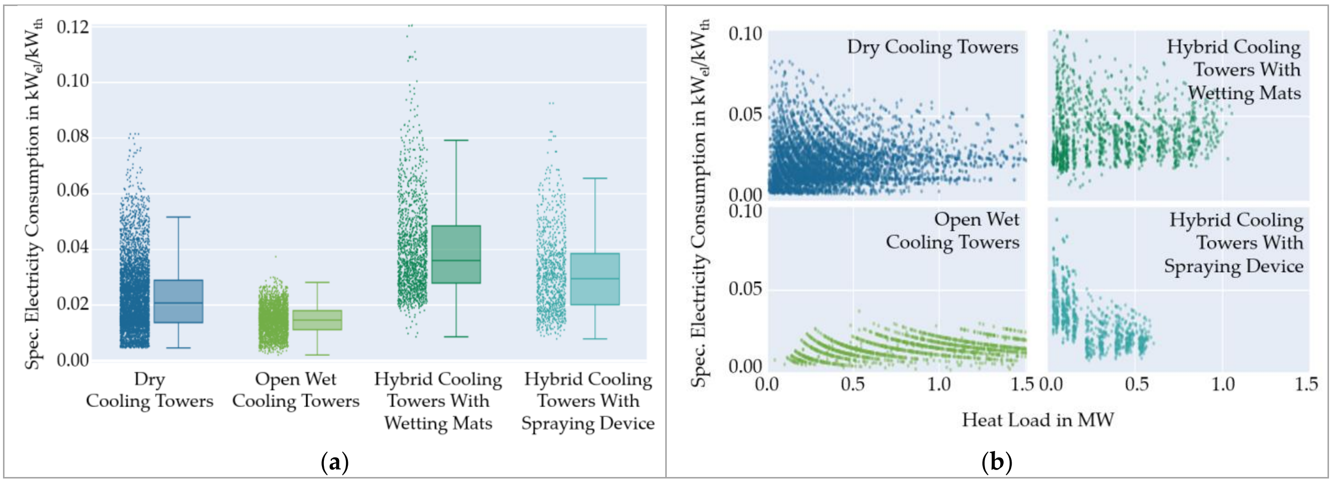

Figure 8. Results for the specific electricity consumption by Wenzel et al. [39] (**a**) as a boxplot diagram and (**b**) over the heat load.

Comparing the *AirPI* in Figure 7a to the specific electricity consumption in Figure 8a shows some similarities. Both cost–benefit ratios indicate that open wet cooling is more energy-efficient than dry and hybrid cooling, which meets the expectations. Furthermore, the indicator values over the heat load show the same tendencies. However, contrary to the specific electricity consumption, the *AirPI* demonstrates that hybrid cooling is more energy efficient than dry cooling due to the normalization of climate and cooling demand. Differences between both indicators generally result from including the ambient and cooling temperatures in the calculation of the *AirPI*. Furthermore, the higher scattering of dry cooling may result from a higher scattering of the ambient temperature of operating points in the dataset compared to wet and hybrid cooling [39] (p. 8).

Regarding the wet and hybrid cooling values, the operating points are comparably more exacerbating for the heat transfer than the operating points of dry cooling. Exacerbating conditions could be comparably high ambient temperature and humidity or low cooling temperatures. This effect is not considered by the specific electricity consumption but by the *AirPI* definition, demonstrating the added value of the *AirPI* as a novel EnPI.

This efficiency analysis of different types of cooling towers in a nominal state can benchmark plants in practice, as follows in the next section.

3.2. Efficiency Evaluation across Varying Environmental Conditions

We analyze the wet cooling system of the HLRS as a case study to plot the time curve of technical requirements and ambient parameters. As before, the *AirPI* is the ratio of fan power to *minimum airflow* for the cooling service.

In terms of cooling temperatures and heat load, the targeted cooling service remains partly unmet at some intervals of the year. Therefore, we quantify the 'effectiveness gap' as the difference between the actual heat load and nominal heat. Furthermore, we calculate the 'efficiency' as the ratio between the *minimum airflow* for the actual benefit and the *minimum airflow* theoretically required for the nominal benefit (cf. Section 2.1.2). This metric is zero if the ambient WBT exceeds the target cooling temperature.

Figure 9 plots the *minimum airflow* of the actual benefit (dots) while highlighting the function of *minimum airflow* theoretically required to achieve the nominal heat load and cooling temperatures (continuous lines).

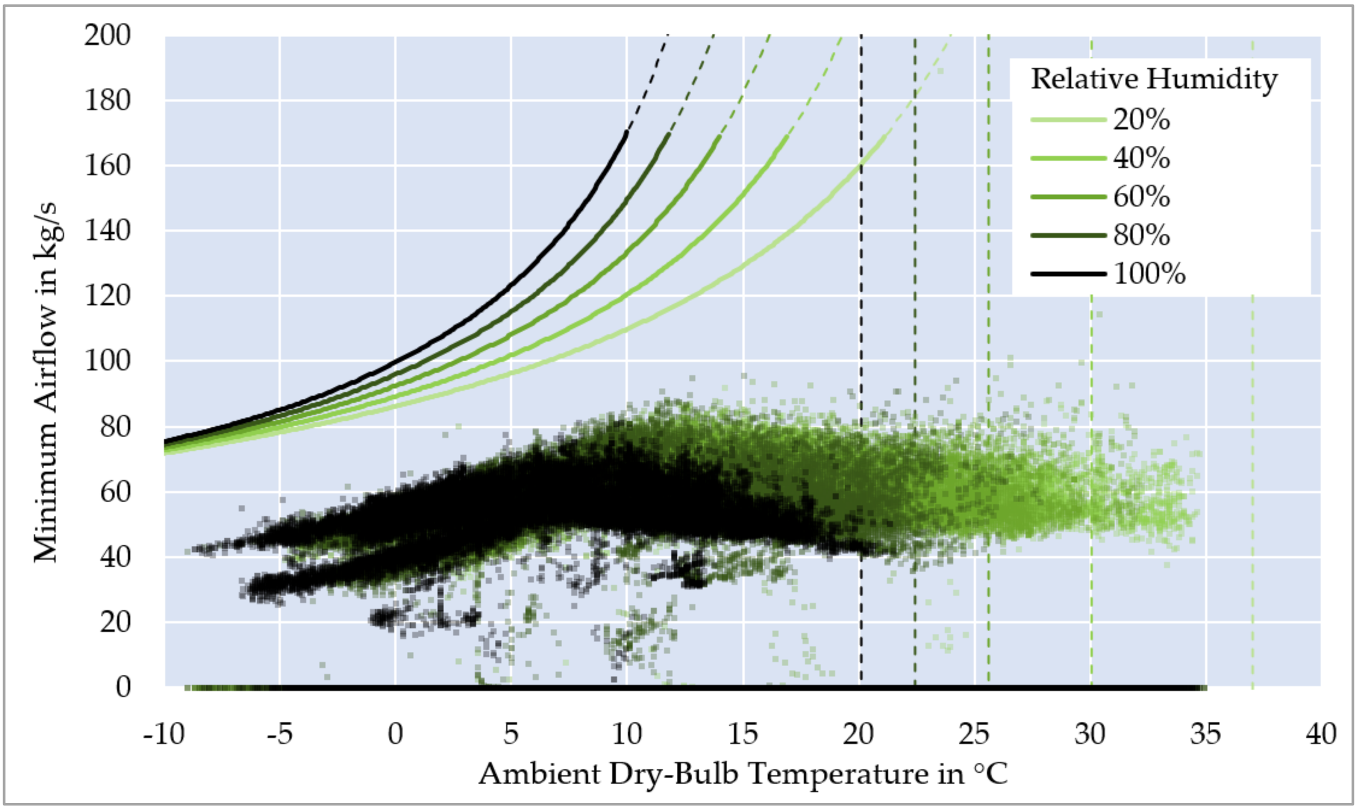

Figure 9. Theoretical *minimum airflow* of the HLRS wet cooling system depending on the ambient temperature and humidity for the nominal state of 4.8 MW_{th} from 20 °C to 10 °C, which the continuous lines represent, and the *minimum airflow* for the actual heat removal and cooling temperatures for the measured 5 min interval values in 2019, which is graphed by 105,108 points. Table S2 in the Supporting Material provides the underlying data.

With increasing temperature, the actual benefit as *minimum airflow* values (dots) increasingly deviates from the *minimum airflow* theoretically necessary for the nominal benefit (lines). For example, in the range of 20 to 30 °C DBT, the actual benefit remains around 60 kg/s *minimum airflow*, although the *minimum airflow* to achieve the nominal benefit delivers invalid values (dashed lines). In this case, achieving the nominal benefit is physically impossible. In conclusion, the increasing deviation of the actual benefit from the nominal implies decreasing effectiveness as the cooling demand remains unmet. Furthermore, Figure 9 demonstrates that the oversizing of the wet cooling system as the actual benefit (dots) significantly differs from the nominal values (lines) not only at high ambient temperature but also in the range of dimensioning outside conditions.

Furthermore, Figure 10 graphs several parameters: the water and ambient temperatures, the heat flow rate, the resulting *minimum airflow*, the fan power, the resulting *AirPI*, and the resulting effectiveness indicator. Additionally, Figure 10 includes the specific electricity consumption to compare the indicators. The parameters are plotted over the WBT (left) and in the time curve of the year (right). The effectiveness values increasingly overlap at zero with rising WBT. For example, between 22.5 and 23.5 °C, all effectiveness values are 0%, and no value is positive. Therefore, the red dots additionally demonstrate in 1 °C intervals the percentage of positive values related to the total value number. For example, a red dot at 40% means 60% of the effectiveness values (black or gray dots) overlap at 0%. Reasons for low effectiveness can be high ambient humidity, high ambient temperature, and less heat transfer than in the nominal condition (cf. Section 2.1.2). Table S5 in the Supporting Material provides the underlying data.

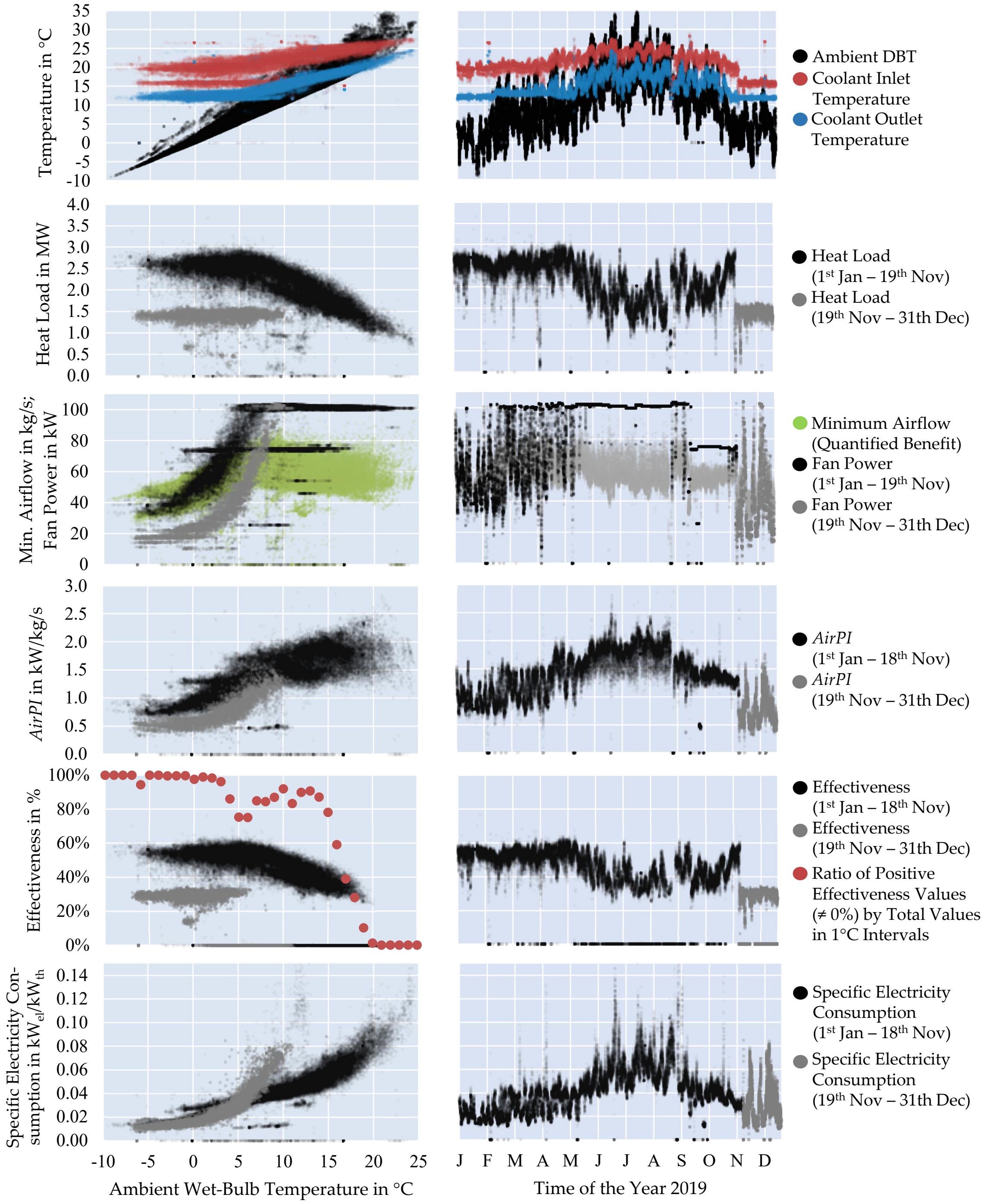

Figure 10. Parameters of the HLRS wet cooling system over the ambient WBT and over time: coolant temperatures, ambient DBT, heat load, *minimum airflow*, fan power, *AirPI*, effectiveness referring to the *minimum airflow*, and the specific electricity consumption. Table S5 in the Supporting Material provides the underlying data.

The water flow rate does not correlate to the WBT but varies between 200 and 400 m^3/h due to the control system. Table S5 in the Supporting Material provides detailed data. Consequently, the graphs display a data scattering of all relevant parameters except for the ambient DBT, which scatters independently of the water flow rate.

Besides the data scattering, Figure 10 confirms several expected correlations with the WBT. Firstly, the coolant temperatures rise significantly above a specific outside WBT threshold. Specifically, both temperatures begin to rise from approximately 13 and 18 °C at 7 °C WBT to 29.1 and 28.2 °C at 25 °C WBT, respectively, as detailed in Table S5 in the Supporting Material. Secondly, the heat load remains almost constant below 5 °C WBT. Above 5 °C, the heat load decreases to values below 1 MW_{th} at 25 °C WBT because the coolant temperature difference decreases. Thirdly, the fan power exhibits an exponential increase, reaching 100 kW_{el} at around 10 °C WBT and remaining nearly constant above that temperature due to using a variable frequency drive and the nominal power.

The *quantified benefit*, expressed as the *minimum airflow*, comprises the coolant temperatures, heat load, and outside conditions. Thus, this *quantified benefit* will increase if the outside temperature rises despite constant coolant temperatures and heat load. A shrinking temperature difference between the coolant and the ambient WBT hampers heat transfer, necessitating an increase in *minimum airflow* to maintain constant coolant temperatures and heat load. Consequently, the *minimum airflow* slightly increases from around 50 kg/s at –5 °C to around 65 kg/s at 5 °C WBT despite nearly constant coolant temperatures and heat load below 5 °C WBT. However, above 10 °C WBT, the *minimum airflow* slightly decreases due to rising coolant temperatures and a decrease in heat load, which outweighs the contrary effect of the increasing WBT on the *minimum airflow*.

As the WBT rises to 5 °C, more *quantified benefit* is generated. However, the costs quantified as fan power increase proportionally more than the *quantified benefit*, as Figure 10 shows. Consequently, the *AirPI* values increase with rising WBT, indicating decreasing efficiency. This means the cooling system operates more efficiently at a partial load of the fans. Typically, fans operate most efficiently at nominal loads, as standard fan characteristics of efficiency over the air volume demonstrate. However, fan efficiency refers to the generated airflow but not the *minimum airflow* as the *quantified benefit* of cooling towers. While the airflow generated by the fan increases as the WBT rises, the *minimum airflow* increases less than the actual airflow through the fan and decreases above 10 °C. Overall, the cooling tower efficiency quantified as the *AirPI* consequently decreases with increasing WBT, despite an increase in the fan efficiency (cf. Section 3.3).

Furthermore, the decreasing heat load with rising WBT indicates the decreasing effectiveness. The increasing *effectiveness gap* is quantified as the difference between the actual heat load and the design capacity of 4.8 MW_{th}. However, the removed heat load can have different qualities in terms of temperature levels. For example, higher cooling temperatures can be additionally disadvantageous for the cooling purpose. Quantifying the benefit of the removed heat load does not capture the challenge of meeting the cooling demand. Hence, we additionally calculate the 'effectiveness' as the ratio of *minimum airflow* for the actual benefit to the airflow required for the nominal benefit (cf. Section 2.1.2.). The effectiveness underscores the increasing difficulty in meeting cooling demands. If the outside WBT exceeds the required cooling temperature, the effectiveness will amount to 0% (cf. dashed lines in Figure 9). Moreover, Figure 10 shows that 100% effectiveness is never achieved. The maximum effectiveness is approximately 60% due to the system's dimensioning and design.

Comparing the *AirPI* to the specific electricity consumption demonstrates that both parameters increase as the WBT rises. However, the specific electricity consumption values tend to have a steeper gradient than the *AirPI* values. The reason is that the AirPI accounts for more quantified benefit for the same heat transfer if ambient temperature and humidity are higher. In contrast, the specific electricity consumption does not recognize the increased benefit of cooling at high outside temperature or humidity levels.

The plots of the time curve over one year show the exact correlation of the parameters to the WBT. For example, in the hot summer months, from July until September, the coolant temperatures are comparably high, the fan power amounts to 100 kW_{el}, and the heat load, efficiency, and effectiveness are comparably low. In November and December 2019, the heat load and fan power were reduced due to the deconstruction of one high-performance computer, so less cooling was needed. The before observed and discussed effects of partial load arise: less fan power, higher efficiency, and lower effectiveness. Furthermore, the high-performance computer was out of operation for maintenance in the first week of September, calendar week 36 [44] (pp. 29–30).

Direct conclusions to efficiency measures are limited without further studies for comparison. For example, if the fans were not speed-controlled and the fan power was constant, the *AirPI* would deliver higher values for lower WBT. We conclude that the *AirPI* confirms the energy saving of a variable frequency drive. Future studies should investigate systems with and without variable frequency drives for comparison. Moreover, the inefficiency in summer months could result, among others, from insects accumulating at the air filter, leading to pressure losses. This is a well-known impact at the site. Assessing the need for system cleaning or maintenance requires an efficiency assessment over a more extended period.

The seasonal *AirPI* calculated as the ratio of integrated fan power and *minimum airflow* over the year amounts to 1.35 kJ/kg. The AirPI allows including November and December despite reduced cooling demand without further normalization. Applying the *AirPI* as EnPI in the future makes different years comparable despite differing ambient conditions and intervals of reduced cooling demand, in contrast to the EER or seasonal EER.

In the next step, we benchmark these results with the dataset of cooling tower models (Section 3.1). The efficiency of open wet cooling towers in nominal states ranges approximately between 0.1 and 1.8 kW/kg/s. The seasonal *AirPI* of 1.35 kJ/kg is within that range. The instantaneous *AirPI* of the HLRS cooling system ranges from approximately 0.4 to 2.5 kW/kg/s. At low WBT, the wet cooling system is more efficient than the dataset's median of 1.0 kW/kg/s for wet cooling towers. At higher WBT, the wet cooling system becomes less efficient with up to 2.5 kW/kg/s. However, as expected, systems in operation are less efficient at some operating points than in the nominal states given by the manufacturer because the nominal states mostly encompass low ambient humidity and temperature.

Nevertheless, this study does not include the refrigeration system of the HLRS cooling system. Hence, future investigations should consider the entire cooling system for further comparison and conclusions. Furthermore, this study excludes the efficiency potentials outside the system boundary, such as in the pump system or the increase in required cooling temperatures. For example, the high-performance computer installed in November 2019 significantly reduced the electricity demand for cooling due to its higher cooling water temperatures. The *AirPI* does not display efficiency potentials in such modification in the cooling demand but independently reflects the efficiency.

The case study underscores the advantages of the *AirPI* as EnPI due to its incorporation of ambient and technical parameters. The HLRS case study confirms this aspect by comparing the *AirPI* to the specific electricity consumption, which refers only to the heat transfer that, however, can have different qualities in terms of temperatures and ambient conditions. For future studies, the novel indicator will enable comparability across different configurations, weather conditions, and heat loads. In this way, high ambient temperatures do not mistakenly indicate inefficiency. Common indicators, such as seasonal EER, that refer to the dissipated heat load still need normalization. Future studies should further test the *AirPI* by comparing the indicators over several years.

3.3. Discussion on Indicator Feasibility

Comparing the *AirPI* as EnPI to common efficiency assessment methods and indicators for processes in general highlights one significant difference: the effectiveness of cooling towers. Although other indicators embody the cost–benefit ratio, such as the EER, these indicators do not directly suggest the limits of free cooling. Thus, the reference to the thermodynamic ideal is crucial, represented by quantifying the benefit as *minimum airflow* for the *AirPI*. We conclude that the *AirPI* as EnPI is predominantly feasible for systems with higher effectiveness over the year or that the additional cooling device must be included to optimize the energy demand of the total system.

Examining the thermodynamic benefit of cooling towers may instead lead to considering the amount of destructed exergy because it is quantifiable. Destructed exergy would be an unusual benefit of a technical process. The issue, however, lies not in its queerness but rather in the fact that focusing only on the amount of destructed exergy omits the temperature differentials. For example, a cooling tower that operates at higher coolant temperatures relative to ambient temperature has a comparatively easier task than another cooling tower that destructs the same amount of exergy closer to the ambient temperature. The temperature levels are crucial because heat transfer increases with increasing temperature gradient. For example, the heat load of a coolant that is only slightly warmer than the WBT requires more *minimum airflow* to facilitate the heat transfer.

The *AirPI* in the unit of kW/kg/s may still look like an unusual indicator. However, our concept is similar to indicators such as the specific energy consumption for products, for example, in MWh electricity per ton of steel. These indicators must, however, be distinct from percental efficiency indicators, such as electrical efficiency, which state the inverse benefit–cost ratio, where cost and benefit have the same unit.

Moreover, the fan electricity demand can be examined more thoroughly to further distinguish between different inefficiency sources with the *AirPI*. The electricity demand of fans generally depends on the volume flow rate of humid air $\dot{V}_{ha}$, total pressure losses Δp, and total fan efficiency, including motor efficiency η_{fan}:

$$\eta_{fan} = \frac{benefit}{cost} = \frac{\dot{V}_{ha} \cdot \Delta p}{P_{fan}}. \quad (8)$$

Thus, the EnPI evaluates different inefficiency causes simultaneously: the pressure drop, the fan efficiency, and the actual airflow compared to the ideal *minimum airflow.* These factors jointly contribute to the energy demand. The EnPI can be separated into these influencing factors to find the efficiency-saving potentials:

$$AER = \frac{P}{\dot{m}_{a,min}} = \frac{\dot{V}_{ha} \cdot \Delta p}{\eta_{fan} \cdot \dot{m}_{a,min}} = \frac{\dot{V}_{ha}}{\dot{m}_{a,min}} \cdot \Delta p \cdot \frac{1}{\eta_{fan}}. \quad (9)$$

Besides the ideal situation of the *minimum airflow*, the pressure drop and fan efficiency have one physically ideal reference point: the pressure drop would amount to zero, or a negative value for the natural draft, and the fan efficiency is theoretically 100% without technical imperfections. Figure 11 illustrates the cooling tower’s losses and inefficiencies assigned to these three components.

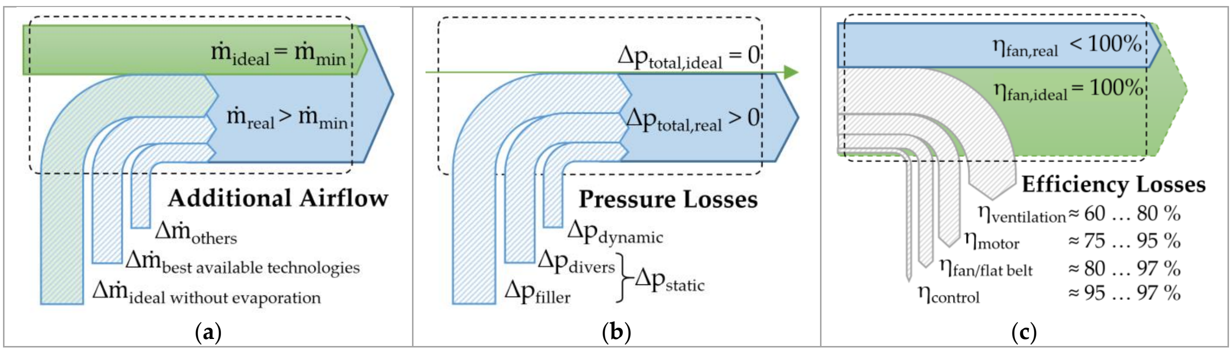

Figure 11. Sources of inefficiency in (**a**) additional airflow, (**b**) the pressure drop, and (**c**) the fan efficiency. All three have one physically ideal case (highlighted in green). In practice, technical imperfections cause additional effort or losses (in gray), resulting in the actual situation (in blue). The efficiency values of the ventilation, motor, fan, or flat belt and control system are approximate benchmarks by literature [47,48] (pp. 35–42), [49] (pp. 56–57), [50].

This *AirPI* definition comprising airflow, pressure loss, and fan efficiency provides valuable insights into inefficiencies, which can be technically avoidable or unavoidable.

3.4. Implications for the SDGs

Against the background of the SDGs by the United Nations [1], this paper predominantly serves the long-term goal of energy-efficient cooling towers. Improved energy efficiency directly contributes to SDG 7 (Affordable and Clean Energy). Moreover, reduced electricity consumption reduces all environmental impacts caused by electricity consumption in general; this indirectly affects several SDGs, such as SDG 13 (Climate Action).

Regarding SDGs related to human health and well-being, efficient cooling towers are required for the globally increasing cooling demand. Furthermore, cooling towers are omnipresent in industry, buildings, and other infrastructure. Thus, sustainable energy-efficient cooling towers contribute to SDG 9 (Industry, Innovation, and Infrastructure), SDG 11 (Responsible Consumption and Production), and SDG 12 (Responsible Consumption and Production).

Nevertheless, the most energy-efficient cooling towers use water for evaporation, exacerbating water scarcity and being contrary to SDG 6 (Clean Water and Sanitary). We recommend expanding the novel *AirPI* concept to at least water consumption. Beyond that, LCA evaluates the eco-efficiency, including energy and water efficiency and other impacts. However, previous LCA studies on cooling towers were incomparable due to inconsistent definitions of the functional unit (cf. Section 1.1). The *quantified benefit* introduced in this study solves this problem. Thus, we anticipate that defining the functional unit as the *minimum airflow* is expedient for LCA and inventory databases, enhancing comparability and model accuracy. Hence, we recommend including this definition in future LCA studies to test this hypothesis.

4. Conclusions

This paper introduces an EnPI that quantifies the energy efficiency of cooling towers by referring to the thermodynamic *minimum airflow* needed for the required heat removal with evaporation. Thus, we call the novel EnPI the *airflow performance indicator* or *AirPI*. Even dry cooling towers are assessed equally by referring to the ideal cooling process with evaporation to demonstrate the additional airflow needed if the effect of evaporation remains unused. The indicator's advantage is that it refers to the cooling tower's fundamental function, which serves to normalize the technical and ambient parameters.

Assessing 6575 cooling tower models at nominal operating points and one wet cooling system of a high-performance computing center as a case study leads to the following key findings:

- Regarding the model dataset, the *AirPI* quantitatively confirms that wet cooling towers are more energy-efficient than dry cooling towers: dry cooling towers consume 2.3 kW/kg/s at the median, whereas wet cooling has a median of 1.0 kW/kg/s and hybrid cooling with wetting mats or spraying devices 1.3 and 1.8 kW/kg/s, respectively.
- Furthermore, the *minimum airflow* underscores the energy-saving potential of evaporative cooling. Regarding the median of dry cooling towers, approximately 7.8 times more airflow is theoretically needed if evaporative cooling is not implemented, directly correlating to the required fan power.
- The case study demonstrates that the indicator serves to determine the efficiency of cooling towers in operation instantaneously or integrated over time, for example, as seasonal *AirPI*. However, the operation differs significantly from the nominal heat transfer. Thus, we recommend assessing the effectiveness simultaneously. The effectiveness, quantified as the ratio between *minimum airflow* of actual to nominal benefit, highlights the necessity of additional cooling devices.
- Thanks to the normalization of outside conditions, the case study results are comparable to the dataset of 6575 cooling tower models in nominal operating points. The

investigated wet cooling system turns out to be more efficient than the median of the manufacturer's nominal data of wet cooling towers. However, at high ambient temperatures, the efficiency decreases, and the effectiveness is low. Further investigations must include the entire cooling and chilling system.

Moreover, this study highlighted the physical limits of free cooling at high ambient temperatures. Future studies should investigate integrating the cooling tower's effectiveness into the efficiency concept. For example, if other processes yield one ton when two tons are demanded, doubling might suffice. For cooling towers, however, the benefit quantification is more complex, encompassing heat temperature levels, so a gap in effectiveness mostly requires other cooling devices, such as chillers.

To pursue the SDGs, quantifying energy efficiency is paramount to exploit the resource-saving potential by appropriate measures. Efficiency measures comprise optimizing the design, selection, and operation of cooling towers for maximum efficiency. The EnPI introduced in this paper proved to address these potentials. Accordingly, we recommend integrating the *AirPI* in cooling tower research and development, as well as corporate energy and environmental management systems to enhance resource efficiency and sustainability.

Supplementary Materials: The following supporting information can be downloaded at https://www.mdpi.com/article/10.3390/su152115454/s1, Table S1: Data from Figure 4 in the manuscript: Dataset of cooling tower models based on manufacturer data by Wenzel et al., Table S2: Data of Figures 5 and 9 in the manuscript: Theoretical minimum airflow for 4.8 MW_{th} heat removal from 20 °C to 10 °C depending on the outside temperature and moisture, Table S3: Data from Figure 6 in the manuscript: Minimum airflow compared to the theoretical minimum airflow without evaporation and the nominal airflow of each operating point of the cooler model dataset, Table S4: Data from Figure 7 in the manuscript: EnPI results of each cooling tower type data based on Wenzel et al., Table S5: Data from Figure 10 in the manuscript: Parameters of the HLRS wet cooler.

Author Contributions: P.M.W.: conceptualization, methodology, software, validation, formal analysis, investigation, data curation, writing—original draft preparation, visualization, funding acquisition; E.F.: software, formal analysis, writing—review and editing; P.R.: writing—review and editing, supervision. All authors have read and agreed to the published version of the manuscript.

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Data Availability Statement: The data presented in this study are either available within the article or the Supplementary Materials.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

Abbreviations

COP	coefficient of performance
DBT	dry-bulb temperature
EER	energy efficiency ratio
EnPI	energy performance indicator
HLRS	High-Performance Computing Centre of the University of Stuttgart
IQR	interquartile range
LCA	life cycle assessment

SDG	sustainable development goal
WBT	wet-bulb temperature

Symbols

$AirPI$	airflow performance indicator
c	heat capacity in kJ/kg/K
$EnPI$	energy performance indicator
H	enthalpy in J; enthalpy flow rate $\dot{H}$ in W
h	enthalpy per mass in J/kg (enthalpy of humid air refers to the mass of dry air)
m	mass in kg; mass flow rate $\dot{m}$ in kg/s
P	power in W
p	pressure in Pa
Q	heat in J; heat flow rate $\dot{Q}$ in W
T	temperature in K
V	volume in m^3; volume flow rate in m^3/s
X	absolute humidity in kg_w/kg_{da}
ϑ	temperature in °C
φ	relative humidity in %
η	efficiency in %
$\dot{}$	. . . flow rate [. . . /s]

Subscripts

″	saturated
0.25	25% quartile
0.75	75% quartile
a	air
ct	cooling tower
da	dry air
el	related to electricity
h	heating element
ha	humid air
i	input, inlet
min	minimum
n	nominal
o	output, outlet
p	pump or isobaric
r	real, actual
th	thermal
v	vapor
w	water or coolant
wt	water treatment

Appendix A

Table A1 provides the definitions of cooling tower types based on Wenzel [39].

Table A1. Overview of cooling tower types based on Wenzel et al. [39].

Type	Description
dry cooling towers	conductive heat transfer to dissipate heat to the surrounding air with a heat exchanger and forced ventilation
wet cooling towers with open circuit	heat dissipation by evaporation as the coolant trickles down fillers in direct contact with the ambient air
wet cooling towers with closed circuit	evaporative heat transfer but with a coolant-air heat exchanger and a separate circulating coolant system where the coolant trickles down to facilitate evaporation

Table A1. *Cont.*

Type	Description
hybrid cooling towers with direct wetting	hybrid cooling is a combination of dry and indirect wet cooling methods with seasonal shift, while directly wetted hybrid cooling towers involves a wetting circuit comprising an auxiliary water pump, water trickling down the heat exchanger, and a water collection tank
hybrid cooling towers with spraying device	the spraying system saturates the ambient air to enhance subsequent cooling in the heat exchanger
hybrid cooling towers with wetting mats	wetting-mat systems facilitate the water distribution by additional water trickling down the mats, enabling evaporation into the ambient air that subsequently passes the heat exchanger

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