Predicting the waterside temperature difference of a cooling coil in part load

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Abstract. Chilled water plants must operate in the most energy efficient way possible. One of the threats to a good energy performance in part load is the 'Low ΔT syndrome', several causes of which are related to the cooling coils. In order to detect deviation behaviour of the waterside temperature difference (ΔT), one needs to know the normal return water temperature during part load operation of the cooling coils. However, until now it has not been possible to predict and quantify the waterside temperature in part load. This paper provides a mathematical derivation to predict the waterside temperature difference in part load. For a constant air and water flow, the performance is only determined by the nominal water leaving temperature and leaving air temperature. Both temperatures determine if the chilled water temperature decreases or increases in part load and the shape of the heat exchanger characteristic of the cooling coil. The result is tested for two known causes of the Low ΔT syndrome, namely 100% outdoor air handling units and economizers. The results show that placing the cooling coil behind (blow through), instead of in front of (draw through), the fan will result in an increased chilled water temperature difference in part load.

Keywords. Cooling coil, part load, low-ΔT syndrome, heat exchanger characteristic
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1. Introduction

Cooling is required for indoor climate control in many buildings, and buildings are equipped with Chilled Water (CHW) plants for the purpose of cooling. The cold water can be supplied by chillers or a sustainable source, such as a seasonal thermal energy storage or, a typical Dutch variant, the Aquifer Thermal Energy Storage (ATES). In order to minimize the energy consumption, chillers should operate in the most efficient way. The CHW-plant is designed for full load though generally operates in part load (PL) condition; a good PL efficiency is therefore crucial for a good performance. In PL the Chilled Water Temperature Difference (CHWdT) often differs from the full load condition. One of the causes that threatens a good performance of the CHW-plant is a reduced Chilled Water Return Temperature (CHWR) in PL. This can result in increased energy consumption of circulation pumps and chillers [1–3]. A reduced CHWR with a decreased CHWdT will, for the same capacity, result in increased water flow and pump energy usage. This is especially true for (A)TES systems, because here the pump energy usage is the main source of energy consumption. Many large CHW-plants suffer from a reduced CHWR [2,4] and a series of causes are known as the Low ΔT syndrome: *"A condition whereby a low chilled water return temperature (CHWR) causes an excessive amount of chilled water to circulate to meet system cooling loads and chillers receiving the low temperature CHWR cannot be loaded to their design capacity*"[3,5].

A selection of causes are as follows:

- poorly selected cooling coils [2];
- a reduced set point of the leaving air temperature (LAT) [6];
- economisers and 100% outside air unit [2];
- laminar flow in the cooling coil [7];
- increased chilled water supply temperature (CHWS) in PL [2].

In order to detect the Low ΔT syndrome, the normal CHWdT for the PL of a cooling coil must be known and compared with the actual CHWdT. The definition of the Low ΔT syndrome uses the words "*excessive amount*" and "*low chilled water return temperature*", but in reality, the reduced CHWR can occur without the "excessive amount" of chilled water circulation. This detection is difficult because, to the best of our knowledge, it is for example not possible to accurately predict or quantify a decreasing (Unfavourable) or increasing (Favourable) CHWdT of the cooling coil in PL, see **Fig. 1**. Therefore, it is not known how the selection of a cooling coil can be improved in order to achieve an increased CHWdT in PL to meet the desired dT condition.

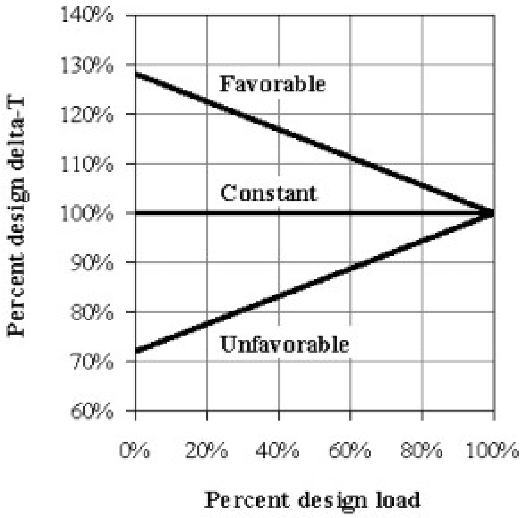

Fig. 1 - Favourable, Constant and Unfavourable CHWdT based on actual data [8]

The main question addressed in this paper is: How will the CHWdT develop – decreasing, constant or increasing – during PL operation?

There has been research done on improving the CHWdT of cooling coils in PL. A summary of this literature is given in **Tab. *1***.

Considering the above-mentioned aspects, the objective of this work is to contribute to the knowledge of the waterside performance of the cooling coil in PL. This enables the evaluation of the PL behaviour of cooling coils during the design and analysis of CHW systems and allows proper measures to be taken, if necessary. As a result, the performance of the whole CHW-plant can be improved. The result is a mathematical formulation to predict an increasing, decreasing or constant CHWdT development in PL. The results are presented in a normalised form based on the nominal conditions.

Tab. 1 - Investigations of part load behaviour of a cooling coil

Author	Investigation	Comment
Yamaguchi et al. [9]	Investigated the PL performance of a cooling coil of six types of engineering models.	The use of the model depends on fields of investigation because waterside results can vary. PL behaviour is not mentioned.
Lu et al. [10]	Presented a performance graph of a cooling coil based on the selection data of the supplier.	Waterside temperatures are not mentioned.
Morisot et al. [11]	A simplified engineering model is discussed to determine the results of a cooling coil in non-design conditions.	Waterside PL behaviour is not studied.
Fiorino [12]	Gave 25 “best practices” to increase the CHWdT in PL.	It recommends selecting an oversized cooling coil. However, it provides no additional knowledge on the CHWdT in PL.
Sekhar & Tan [13]	Studied the dehumidification performance of an oversized (200%) cooling coil in a humid climate.	The waterside results are given but not evaluated.
Zhang et al. [14]	Studied the CHWdT characteristics of chilled water-cooling coils for different geometric configurations and various waterside and airside conditions by means of computer simulation.	It shows that the CHWdT could be higher, equal or lower in PL. A prediction of the CHWdT in PL was not given.

2. Research method

This paper provides a mathematical, normalized model, calculating the CHWdT of a cooling coil as a function of the PL ratio of a cooling coil. This model was obtained by derivation, using three equations describing the heat transfers in a cooling coil. These equations are the waterside capacity, the airside capacity, and the heat transfer capacity using the ε-NTU method [15]. In the derivation, the LAT and the entering chilled water temperature were kept constant. The condition of the cooling coil is considered dry with constant heat transfer at the airside and the waterside. The normalized form was obtained by using characteristics of the design or nominal condition as used by the authors of [8]. To verify the mathematical model, it is used to create the heat exchanger characteristic, and this is compared with the results from reference [16].

3. Results

The results are presented in two sections. The first section presents the result of the derivation of the CHWdT in PL of a cooling coil. The second section provides the heat exchanger characteristic with verification.

3.1. Derivation of a waterside temperature difference in part load for a constant water flow and a constant air flow

The matching hydraulics for the cooling coil is shown in **Fig. 2**. The heat transfer is kept constant on the waterside by means of a circulation pump (CP) and the capacity is controlled by the two-way control valve (CV). The temperatures entering and leaving the cooling coil are the secondary temperatures, denoted *sec*; the temperatures connected to the main distribution net are called the primary temperatures, denoted *prim*. The nominal waterside temperature difference on the primary side and the secondary side are the same.

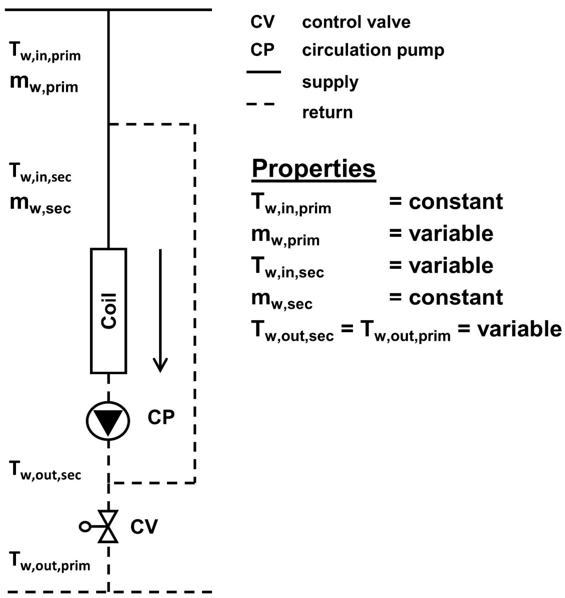

Fig. 2 - Hydraulic configuration of a cooling coil, mixing circuit (module 5) [17]

For any cooling coil, the capacity $\dot{Q}$ [W] can be written for the airside, for the waterside and in terms of heat transfer. The latter is defined by the ε-NTU method [18]. The lumped element model of the cooling coil and the energy balance are given in **Fig. 3**.

First, the airside heat transfer (capacity) equation is written as:

$$\dot{Q} = \dot{m}_a . c_a . \left(T_{a,in} - T_{a,out}\right) \quad (1)$$

Here, $\dot{m}_a$ [kg/s] is the mass flow of the air, c_a [J/kg.K] is the heat capacity of the air, $T_{a,in}$ [°C] the air entering temperature and $T_{a,out}$ [°C] the LAT.

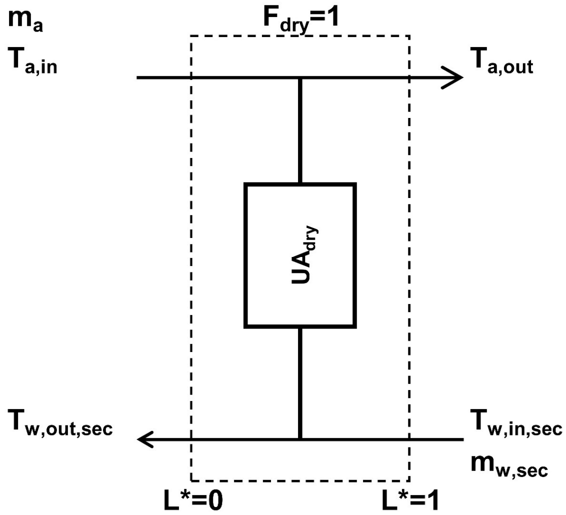

Fig. 3 - Model and energy balance of a dry cooling coil in counter flow

Second, for the waterside:

$$\Delta T_{w,sec} = T_{w,out,sec} - T_{w,in,sec} = \frac{\dot{Q}}{\dot{m}_{w,sec} . c_w} \quad (2)$$

Here, $\Delta T_{w,sec}$ [K] is the secondary CHWdT, $T_{w,out,sec}$ [°C] the secondary CHWR, $T_{w,in,sec}$ [°C] the secondary CHWS, $\dot{m}_{w,sec}$ [kg/s] the secondary water flow and c_w [J/kg.K] the heat capacity of water. Third, from the ε-NTU method we obtain:

$$T_{w,in,sec} = T_{a,in} - \frac{\dot{Q}}{\varepsilon . C_{min}} \quad (3)$$

Here, ε [-] is the effectiveness of the heat exchanger and C_{min} [J/kg] is the minimum capacity flow. $T_{w,out,sec}$ is $T_{w,in,sec}$ (3) increased with $\Delta T_{w,sec}$ (2), where the denominator of the latter is rewritten as the secondary waterside capacity flow $C_{w,sec}$ [J/kg]. Since $T_{w,out,sec}$ [°C] is equal to $T_{w,out,prim}$ [°C], it is written as $T_{w,out}$:

$$T_{w,out} = T_{a,in} - \frac{\dot{Q}}{\varepsilon . C_{min}} + \frac{\dot{Q}}{C_{w,sec}} \quad (4)$$

In order to calculate $\Delta T_{w,prim}$, $T_{w,in,prim,nom}$ [°C] is subtracted on the left and right-hand side of the equation and both sides are divided by the primary ΔT_w of the nominal condition [8]:

$$\frac{T_{w,out} - T_{w,in,prim,nom}}{T_{w,out,nom} - T_{w,in,prim,nom}} = \quad (5)$$

$$\frac{T_{a,in} - T_{w,in,prim,nom} + \dot{Q}\left(\frac{1}{C_{w,sec}} - \frac{1}{\varepsilon . C_{min}}\right)}{T_{w,out,nom} - T_{w,in,prim,nom}}$$

Here, $T_{w,out,nom}$ [°C] is the CHWR in nominal condition. The dimensionless waterside temperature difference T^*_w [-] is defined as equation (6):

$$T_w^* \equiv \frac{T_{w,out} - T_{w,in,prim,nom}}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (6)$$

The dimensionless capacity $\dot{Q}^*$ [-] is defined as:

$$\dot{Q}^* \equiv \frac{\dot{Q}}{\dot{Q}_{nom}} \qquad (7)$$

Here, $\dot{Q}_{nom}$ [W] is the capacity in nominal condition. Now, the left-hand side of (5) is replaced by (6) and $\dot{Q}$ is replaced with (7), and (5) is written as (8):

$$T_w^* = \frac{T_{a,in} - T_{w,in,prim,nom} + \dot{Q}^*\left(\frac{\dot{Q}_{nom}}{C_{w,sec}} - \frac{\dot{Q}_{nom}}{\varepsilon . C_{min}}\right)}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (8)$$

Both expressions between brackets in the numerator can be rewritten in a CHWdT for the nominal conditions. The left expression between brackets is based on ΔT_w (2) and the right expression between brackets is based on the definition of the ε-NTU method (3):

$$T_w^* = \frac{T_{a,in} - T_{w,in,prim,nom} + \dot{Q}^*\left(\left[T_{w,out,nom} - T_{w,in,nom}\right] - \left[T_{a,in,nom} - T_{w,in,nom}\right]\right)}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (9)$$

After eliminating $T_{w,in,nom}$, this results in:

$$T_w^* = \frac{T_{a,in} - T_{w,in,prim,nom} - \dot{Q}^*\left(T_{a,in,nom} - T_{w,out,nom}\right)}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (10)$$

The Entering Air Temperature (EAT) $T_{a,in}$ can be calculated, similar to the nominal condition $T_{a,out,nom}$ [°C] from equation (1), from a known capacity $\dot{Q}$ and with a constant LAT:

$$T_{a,in} = \frac{\dot{Q}}{\dot{m}_a . c_a .} + T_{a,out,nom} \qquad (11)$$

The capacity $\dot{Q}$ is rewritten with (7) in the dimensionless form:

$$T_{a,in} = \dot{Q}^*\left(\frac{\dot{Q}_{nom}}{\dot{m}_a . c_a .}\right) + T_{a,out,nom} \qquad (12)$$

The term between brackets is the airside temperature difference for the nominal condition, resulting in:

$$T_{a,in} = Q^*\left(T_{a,in,nom} - T_{a,out,nom}\right) + T_{a,out,nom} \qquad (13)$$

This expression for $T_{a,in}$ (13) is substituted in (10):

$$T_w^* = \frac{\dot{Q}^*\left(T_{a,in,nom} - T_{a,out,nom}\right) + T_{a,out,nom} - T_{w,in,prim,nom} - \dot{Q}^*\left(T_{a,in,nom} - T_{w,out,nom}\right)}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (14)$$

Rearranging this equation results in a waterside temperature difference T^*_w, depending only on the capacity $\dot{Q}^*$ and constants of the nominal or design condition:

$$T_w^*\left(\dot{Q}^*\right) = \left[\frac{T_{w,out,nom} - T_{a,out,nom}}{T_{w,out,nom} - T_{w,in,prim,nom}}\right] Q^* + \frac{T_{a,out,nom} - T_{w,in,prim,nom}}{T_{w,out,nom} - T_{w,in,prim,nom}} \qquad (15)$$

This equation (15) shows that T^*_w of a cooling coil in PL can be written as a linear function of $\dot{Q}^*$ and the temperatures of the nominal condition. The slope represents the favourable, unfavourable and constant T^*_w in PL of **Fig. 1**, which only depends on $T_{w,out,nom}$ and $T_{a,out,nom}$. An unfavourable T^*_w in PL cannot always be avoided. As a result, for two known causes of the low ΔT syndrome (100% outdoor air handling unit & economizer), it is found that a commonly used definition from the literature is not suitable. Typical, with an economiser in the air handling unit, at a decreasing EAT the recirculation damper closes while the outdoor damper opens to maximise the free cooling capability. Eventually, only outdoor air is processed and the air handling unit is operating as a 100% outdoor air unit. If the EAT drops below the CHWR of the nominal condition, the CHWdT will decrease. The numbers of

Tab. 2 are used to illustrate this for the increasing, constant and decreasing CHWdT, and the accompanying EAT profiles in **Fig 4**. The EAT is normalized using equation (6).

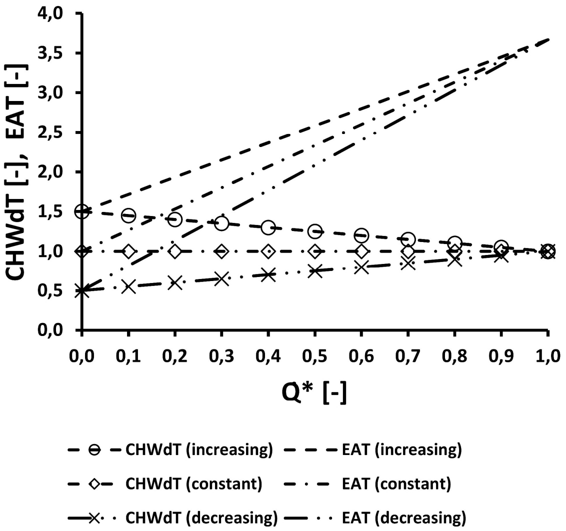

Fig 4 - Increasing, constant and decreasing CHWdT for accompanying EAT profiles

For these cases, the name low ΔT syndrome is misleading since a reduced CHWR can occur during normal conditions without the increased waterflow and therefore during fault-free operation.

3.2. Heat exchanger characteristic of a cooling coil with a constant water flow and a constant air flow

The previous result can be used to create heat exchanger characteristics (HXC), also known as the non-linearity of the cooling coil. The HXC is defined as the capacity as a function of the primary mass flow. It should be noted that the primary mass flow is not constant but variable, and is controlled by the CV (see **Fig. 2**). This HXC can be compared with the characteristics found in the literature for verification. In order to calculate the dimensionless heat characteristic, the primary mass flow $\dot{m}^*_w$ [-] is defined as:

$$\dot{m}^*_w \equiv \frac{\dot{m}_w}{\dot{m}_{w,nom}} \qquad (16)$$

Here, $\dot{m}_{w,nom}$ [kg/s] is the primary mass flow at the nominal condition. As a result, $\dot{Q}^*$ can be calculated in the following way:

$$\dot{Q}^* = \dot{m}^*_w . T^*_w \qquad (17)$$

With (15) substituted in (17), the full expression for the HXC yields:

$$\dot{Q}^*(\dot{m}^*_w) = \frac{\dot{m}^*_w . \left(\frac{T_{a,out,nom} - T_{w,in,prim,nom}}{T_{w,uit,nom} - T_{w,in,prim,nom}}\right)}{1 - \dot{m}^*_w . \left(\frac{T_{w,out,nom} - T_{a,out,nom}}{T_{w,uit,nom} - T_{w,in,prim,nom}}\right)} \qquad (18)$$

Like the CHWdT in PL, the form of the HXC is determined by the same variables. The three different forms are shown in **Fig. 5**, **Fig. 6**, and **Fig. 7** and are based on the numbers of

Tab. 2. These numbers are used to indicate the three different presentations for an increasing, constant, and decreasing waterside temperature difference in PL.

The convex form of the figure of **Fig. 5** is often encountered in literature and is often drawn as a "rule of thumb" and is applicable for a favourable (increasing) CHWdT in PL.

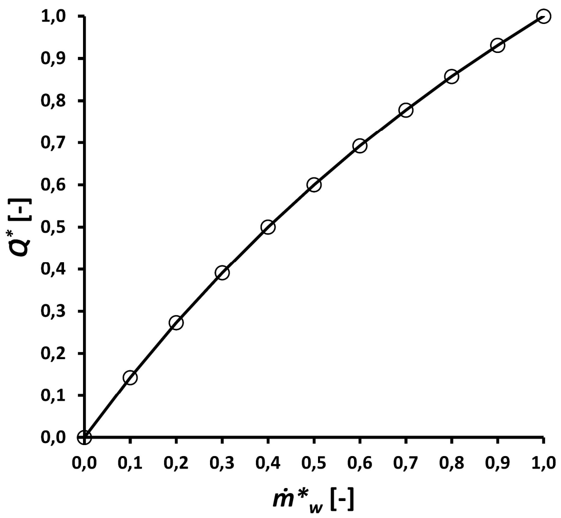

Fig. 5 - Heat exchanger characteristic for a cooling coil with an increasing CHWdT in part load

Fig. 6 shows the linear HXC for a cooling coil with a constant CHWdT in PL. **Fig. 7** shows the HXC of a cooling coil with a decreasing CHWdT in PL, resulting in a concave form. Moreover, the relationship between the T^*_w in PL and the form of the HXCs is, to the best of our knowledge, not available in the literature.

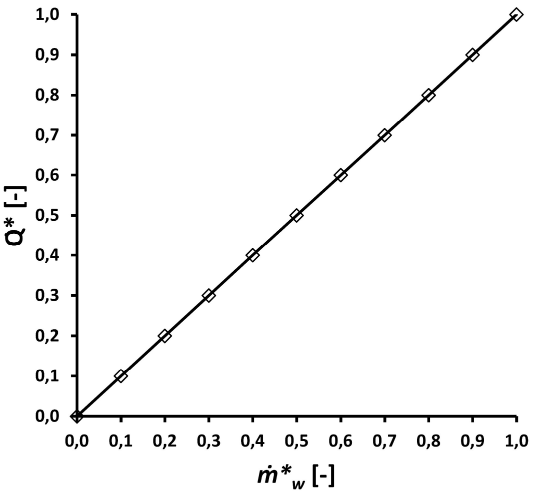

Fig. 6 - Heat exchanger characteristic for a cooling coil with a constant CHWdT in part load

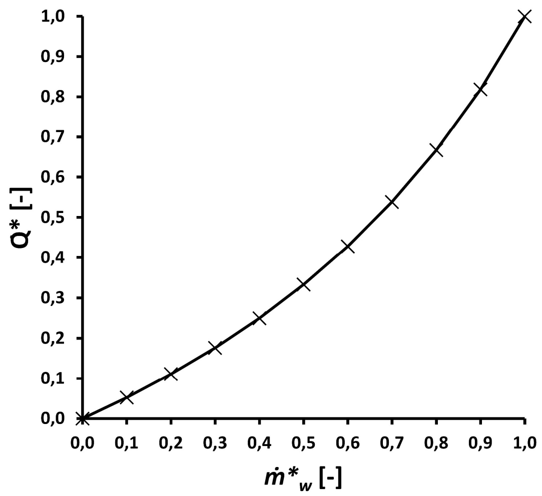

Fig. 7 - Heat exchanger characteristic for a cooling coil with a decreasing CHWdT in part load

For verification, based on the data from **Tab. 2**, all HXCs are created with the same numbers and the equation found in a handbook for a mixing circuit (**Fig. 2**) and a constant LAT [16]. As a result, the current work is successfully verified as there was no difference, with both HXCs being identical.

Tab. 2 - Nominal conditions for three different cooling coils

	Increasing CHWdT	Constant CHWdT	Decreasing CHWdT
EAT	28°C	28°C	28°C
LAT	15°C	12°C	9°C
CHWS	6°C	6°C	6°C
CHWR	12°C	12°C	12°C

The results are summarized in **Tab. 3**.

4. Discussion

By means of a better understanding of the waterside PL behaviour of a cooling coil, the design of a CHW plant can be improved and known causes of the Low ΔT syndrome can be challenged. It is shown that the CHWdT of a dry cooling coil with constant air flow and constant water flow in PL can be written as a linear function based only on the capacity and temperatures of the nominal condition. As a result, during the design phase a decreasing (Unfavourable), constant and increasing (Favourable) CHWdT can be distinguished, and appropriate measures can be considered and possibly implied.

Tab. 3 - Summary of results for CHWdT and HXC in part load

Slope [-]	**Key**	**T^*_w [-]**	**HXC $Q^*(m_w^*)$**
negative slope	$T_{w,out,nom} < T_{a,out,nom}$	Favourable	Convex
Slope = 0	$T_{w,out,nom} = T_{a,out,nom}$	Constant	Linear
positive slope	$T_{w,out,nom} > T_{a,out,nom}$	Unfavourable	Concave

By placing the cooling coil behind the fan (blow-through), the supply fan will add heat before the cooling coil and this must be accounted for during the selection of the cooling coil. The LAT remains the same in contrast with the draw-through configuration, where the fan is placed behind the cooling coil. As a result, to compensate for the fan heat and loss of net cooling capacity, the LAT of the design must be reduced in order to maintain the required cooling capacity of the zone. The effect of 1K fan heat on the favourable and unfavourable configuration of **Tab. 2** is quantified with equation (15) for the draw-through and blow-through configurations and shown in **Tab. 4**.

For both cases, the blow-through configuration leads to an increased CHWdT compared to the draw-through configuration; the slope is smaller. Also, in both cases the highest CHWdT is found for the blow-through configuration at the minimum load condition if $\dot{Q}^*$ approaches zero; the constant value is the highest. The results show that placing the cooling coil behind (blow through) instead of in front of (draw through) the fan will result in an increased CHWdT in PL. Although a comparison between the draw-through and blow-through configuration has been made for cooling and dehumidification performances, the effect on the CHWdT was not considered [19].

Tab. 4 - Change of the slope due to 1K fan heat for the favourable and unfavourable chilled water temperature difference for a draw through and a blow through configuration

	Favourable		Unfavourable	
	Draw-through	Blow-through	Draw-through	Blow-through
slope	-0.33	-0.5	+0.67	+0.5
constant	+1.33	+1.5	+0.33	+0.5

According to the ASHRAE handbook, a conventional building with a traditional all-air system is typically designed at a supply air temperature of 12.8°C [20]. The ASHRAE standard 90.1-2019 [21] states that the design CHWR should not be lower than 13.9°C with a minimum CHWdT of 8.3K (exceptions excluded). With $T_{w,out,nom} > T_{a,out,nom}$, the key of **Tab. 3** shows an unfavourable CHWdT resulting in a slope of +0.13 with equation (15). Circumstances where this can happen include, for example, 100% outdoor air handling units or air handling units equipped with economisers. Here, the EAT can drop below the design CHWR. As a result, the CHWR can never be higher than the EAT. In addition, the increase of the CHWdT during design can increase the risk of the emergence of the low ΔT syndrome. However, this can happen without the "excessive flow" mentioned in the definition of the Low ΔT syndrome. Also, for ATES systems with a high CHWS (~10°C) and large CHWdT up to 10K, this should be considered during PL.

The results can be used to give additional information on existing studies. For example, actual data are simplified to create the favourable or unfavourable CHWdT of **Fig. 1** [8] and these could be reproduced with equation (15). For the same cooling capacity, an unfavourable CHWdT will have a higher chilled water flow compared to the favourable CHWdT, however, in [8] it seems to lack the "excessive flow" from the definition of the Low ΔT syndrome.

5. Conclusions

The knowledge of waterside performance needs to be further extended to mitigate the negative effects of a reduced CHWdT at PL, and to challenge other causes of the Low ΔT syndrome. This work provides a verified mathematical model to qualify and quantify the waterside PL of a cooling coil. With the result of this work, a decreasing waterside temperature difference could be estimated, giving engineers the possibility to take proper measures. Based on this work, the connection between the PL behaviour and the heat exchanger characteristic has been made. The current results are limited to a constant waterside and airside heat transfer, a dry cooling coil with a constant leaving air temperature.

Data Statement

Data sharing not applicable to this article as no datasets were generated or analysed during the current study.

6. Acknowledgement

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