

DEFINITION OF A NOVEL PERFORMANCE INDICATOR FOR DESICCANT WHEELS

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Abstract. Desiccant wheels (DW) are key components in the desiccant evaporative cooling systems used for air-conditioning in buildings. They dehumidify the air stream by adsorbing the water vapour contained in the air through sorption effect. The equations behind the physics of the DW can be complex to integrate into a system simulation tool. This paper offers a simplified approach to evaluate the performance of a DW by enhancing the similarities between the thermodynamic processes of evaporative cooling and desiccant dehumidification. The new performance indicator is defined similarly to the wet bulb effectiveness of direct evaporative coolers. It is then applied to a DW in operation in a real system installation.

Keywords. Desiccant wheel, Dehumidification, Evaporative cooling, Performance indicator.

Nomenclature

c_p	Specific heat capacity (J/kg.K)
h	Specific enthalpy (J/kg)
$h_{fg,0}$	Water vaporisation enthalpy at 0°C (J/kg)
\dot{m}	Mass flow rate (kg/s)
P	Pressure (Pa)
q	Specific heat flux (J/kg)
\dot{Q}	Heat flux (W)
T	Temperature (°C)
<i>Special characters</i>	
ε	Effectiveness (-)
ω	Specific humidity (kg/kg)
<i>Subscripts</i>	
a	Humid air
in	Incoming
out	Outdoor air
reg	Regeneration air
v	Water vapour
w	Water
wb	Wet bulb

Alternative air-conditioning methods have become a significant area of interest to tackle the drawbacks of vapour-compression systems. Amongst them, desiccant evaporative cooling systems (DECS) couple desiccant dehumidification and evaporative cooling, offering a promising solution by primarily utilising low-grade energy sources such as solar energy, waste heat from industrial processes and district heating networks.

Desiccant wheels (DW) are key components of DECS as they dehumidify the incoming air stream through water adsorption. The inner matrix of the DW is covered with a desiccant material that adsorbs water through sorption effect. The water is then released on the secondary side during the regeneration process. The equations underlying the desiccant dehumidification process can rapidly become complex and semi-empirical models have been developed to describe the DW operation [2], [3].

This paper offers a simplified approach to evaluate the performance of a DW by enhancing the similarities between the evaporative cooling and desiccant dehumidification processes. Those processes are described based on the fundamental laws of thermodynamics and studied to establish the definitions for performance indicators. The newly defined performance indicator is then applied to an operating DW in a real system installation.

1 Introduction

Cooling accounts for nearly 20% of the total electricity demand of buildings worldwide and this share is expected to increase with global warming [1].

2 (De)humidification processes description

This section thoroughly describes the processes of evaporative cooling and desiccant dehumidification.

2.1 Evaporative cooling process

Evaporative cooling occurs through direct contact between an air stream and a water film. Heat is transferred from the air stream to the water film, hence evaporating water. The water vapour is then homogeneously mixed in the air stream. Figure 1 shows a schematic representation of the evaporative cooling process and the associated evolution of the air stream conditions in a psychrometric diagram.

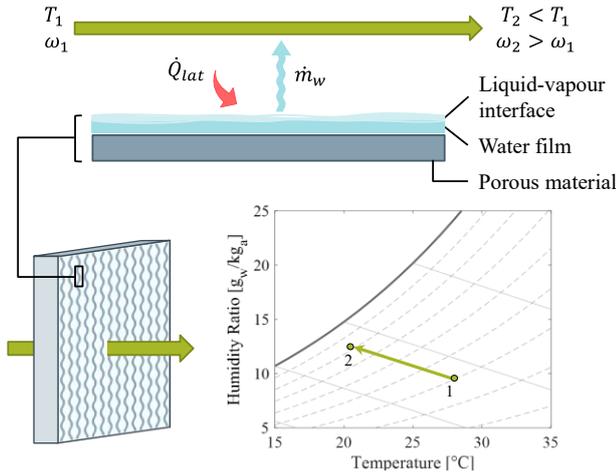


Figure 1: Schematic representation of the evaporative cooling process.

2.2 Desiccant dehumidification process

The desiccant dehumidification process occurs through an intermediate media, a desiccant material such as silica gel. As the air stream enters into contact with the desiccant material, moisture is adsorbed from the air and stored in liquid form at the surface of the desiccant. During the sorption process, latent and sorption heat is released to the primary airflow, which leaves the DW heated and dehumidified. Some additional heat transfer occurs due to sensible energy storage in the inner matrix of the DW during the regeneration process. Figure 2 shows a schematic representation of the desiccant dehumidification process and the associated evolution of the air stream conditions in a psychrometric diagram.

3 Methodology

In this section, the two previously described processes are analysed from a thermodynamic point of view to enhance the similarities between both processes and lay the foundations for defining a new performance indicator for the desiccant wheels.

3.1 Thermodynamic process analysis

Defining a control volume in which the considered process takes place is necessary to perform a thermodynamic analysis of the processes.

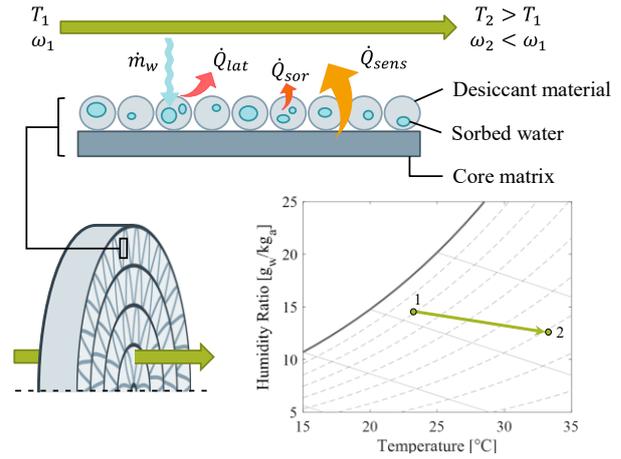


Figure 2: Schematic representation of the desiccant dehumidification process.

Figure 3 gives a general representation of the studied thermodynamic process which can be adapted to represent either the evaporative cooling process or the desiccant dehumidification process. The control volume consists of a box containing air and water. Heat and mass transfers occur through the system boundaries. In its most general form, energy conservation inside the system can be written as

$$\dot{m}_a \cdot h_{a,1} + \dot{m}_w \cdot h_w + \dot{Q}_{in} = \dot{m}_a \cdot h_{a,2} \quad (1)$$

and can be considered at steady-state under the following assumptions:

- The water flow rate should compensate exactly the amount of water added in (resp. removed from) the air, which can be written as

$$\dot{m}_w = \dot{m}_a (\omega_2 - \omega_1) \quad (2)$$

- The water that enters (resp. leaves) the reservoir does not carry sensible energy inside (resp. outside) the system. It should be at the average temperature of the water inside the reservoir.
- There is only latent heat transfer between air and water.

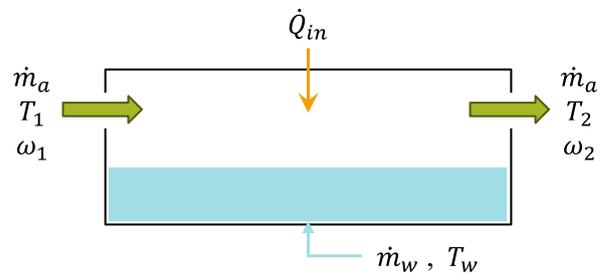


Figure 3: Schematic representation of the studied thermodynamic process.

Eq. (1) can be developed using equations (2)-(5), as shown in Eq. (6).

$$h_a = c_{p,a} \cdot T_a + \omega (c_{p,v} \cdot T_a + h_{fg,0}) \quad (3)$$

$$h_w = c_{p,w} \cdot T_w \quad (4)$$

$$h_{fg,2} = h_{fg,0} + (c_{p,v} - c_{p,w}) T_{a,2} \quad (5)$$

$$(c_{p,a} + \omega_1 \cdot c_{p,v}) (T_{a,2} - T_{a,1}) \quad (6)$$

$$= (\omega_2 - \omega_1) [-h_{fg,2} + c_{p,w} (T_w - T_{a,2})] + q_{in}$$

Eq. (6) can also be expressed as

$$q_{sens} = -q_{lat} + q_{sens,w} + q_{in} \quad (7)$$

- q_{sens} is the sensible energy gained by the air during the process. It is positive if the air is heated.
- q_{lat} is the latent heat of vaporisation of the water. It is positive if water evaporates in the air.
- $q_{sens,w}$ is the sensible energy brought by the water entering the system.
- q_{in} is the “parasitic” heat transfer occurring through the system boundaries.

3.2 Application to the evaporative cooling process

The thermodynamic analysis can be applied to the evaporative cooling process, considering some additional assumptions:

- There is no heat transfer across the system boundaries. The evaporative cooling process can be considered adiabatic.
- If the water enters the control volume at a temperature close to the outlet temperature of the air, the term of the sensible energy of the water can be neglected compared to the term related to the latent heat of vaporisation.

When applied to the evaporative cooling process, Eq. (7) gives:

$$q_{sens} = -q_{lat} \quad (8)$$

indicating that the airflow loses sensible energy that is converted into latent heat for the vaporisation of water. As illustrated in the psychrometric diagram in Figure 4, the evaporative cooling process is isenthalpic and the air saturation limits the maximum humidification rate. The evaporative cooling process performance can be measured by comparing the actual to ideal latent heat fluxes, corresponding to the ratio of actual to ideal sensible heat fluxes. The lowest temperature reached by the air through the evaporative cooling process is the wet bulb temperature, which leads to the definition of the *wet bulb effectiveness*, an indicator widely used in the literature [4].

$$\varepsilon_{wb} = \frac{q_{lat}}{q_{lat,max}} = \frac{q_{sens}}{q_{sens,max}} = \frac{T_1 - T_2}{T_1 - T_{2b}} \quad (9)$$

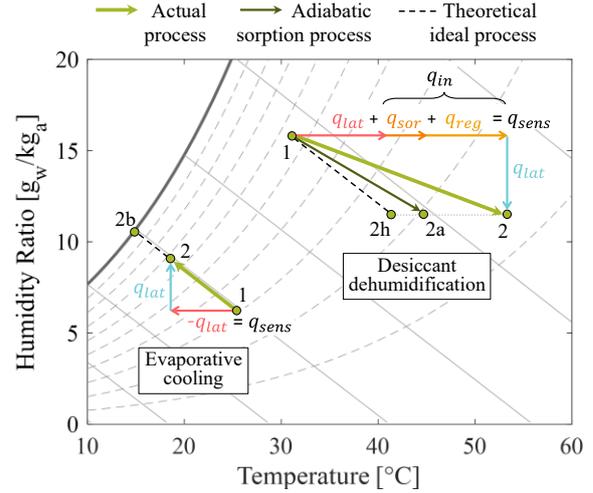


Figure 4: Representation of the actual and ideal evolutions for the evaporative cooling and desiccant dehumidification processes and decomposition into simplified heat fluxes.

3.3 Application to the desiccant dehumidification process and definition of a new KPI

The same methodology can be applied to the desiccant dehumidification process by considering two additional heat fluxes:

- The sorption process is an exothermic reaction. Moisture is removed from the air and bonds with the sorbent, releasing heat, called *heat of sorption* (q_{sor}), into the air stream [5].
- For the desiccant wheel to work continuously, the desiccant material should be regenerated, *i.e.* the moisture adsorbed from the airflow should be removed from the desiccant through heating. As the desiccant wheel rotates, the newly regenerated desiccant material reaches the primary side. The inner matrix of the DW is still at a temperature close to the regeneration temperature and heat is subsequently transferred to the primary air. The heat transfer due to the regeneration process (q_{reg}) is proportional to the temperature difference between the process and regeneration sides.

The thermodynamic analysis can be applied to the desiccant dehumidification process under the following assumptions:

- The aforementioned heat transfers are accounted for in the q_{in} term of Eq. (6).
- If the water leaves the control volume at a temperature close to the outlet temperature of the air, the term of the sensible energy of the water can be neglected compared to the term related to the latent heat of vaporisation

Finally, Eq. (7) gives:

$$q_{sens} = -q_{lat} + q_{sor} + q_{reg} = -q_{lat} + q_{in} \quad (10)$$

In this case, the water condensation into the desiccant material releases heat absorbed by the airstream in the form of sensible heat. As illustrated in Figure 4, the additional heat transfer q_{in} causes a deviation of the actual process from the ideal theoretical process. The heat of sorption causes a deviation from the *isenthalpic* dehumidification, while the heat transfer due to the regeneration process causes a deviation from the *adiabatic* dehumidification. In other words, q_{reg} reflects the irreversibility linked to the regeneration process, while q_{sor} characterises the desiccant material's ability to bond with water.

Similarly to the wet bulb effectiveness, we can define two new indicators for the DW to quantify the closeness to the ideal process under a similar dehumidification rate. The *isenthalpic effectiveness* expresses the closeness of the actual process to an isenthalpic dehumidification:

$$\varepsilon_h = \frac{q_{sens,s}}{q_{sens}} = \frac{-q_{lat}}{-q_{lat} + q_{in}} = \frac{T_{2h} - T_1}{T_2 - T_1} \quad (11)$$

The *adiabatic effectiveness* expresses the closeness of the actual process to adiabatic dehumidification:

$$\varepsilon_a = \frac{q_{sens,a}}{q_{sens}} = \frac{-q_{lat} + q_{sor}}{-q_{lat} + q_{in}} = \frac{T_{2a} - T_1}{T_2 - T_1} \quad (12)$$

The adiabatic and isenthalpic effectiveness are both defined for a constant dehumidification rate because we are interested in characterising the temperature deviation from the ideal evolution when the dehumidification rate is fixed.

4 Results and discussion

4.1 Application of KPI to a real test case

The dehumidification rate and effectiveness of a DW depend on the air conditions at the inlets. Lower temperatures on the primary side drive the sorption process, resulting in larger dehumidification rates [6]. Conversely, the regeneration process is driven by higher temperatures on the secondary side, resulting in an enhanced sensible heat transfer from the core matrix of the DW and the primary air stream.

The effect of the DW inlet air conditions on the dehumidification rate and DW effectiveness has been studied on a real desiccant wheel, rotating at a constant speed, in a desiccant evaporative cooling system test case in Denmark [7], [8]. It has been decided to compute the isenthalpic effectiveness rather than the adiabatic effectiveness because the

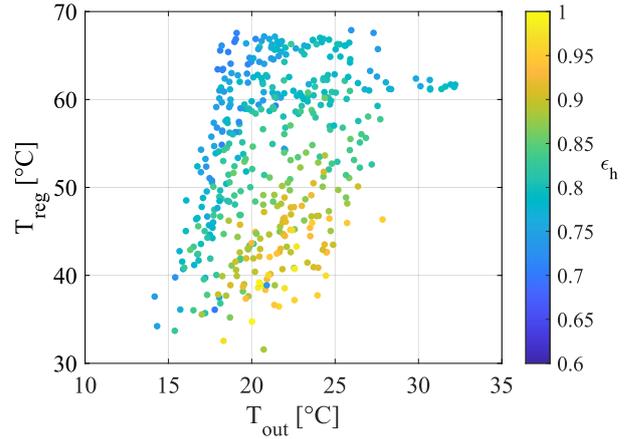


Figure 5: Evolution of the isenthalpic effectiveness depending on the outdoor and regeneration temperatures.

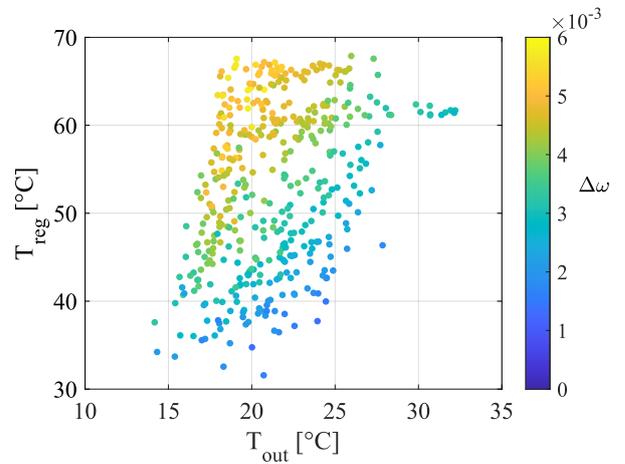


Figure 6: Evolution of the dehumidification rate depending on the outdoor and regeneration temperatures.

latter requires information about the heat of sorption, which computation is not straightforward [9].

Figures 5 and 6 show the evolution of the dehumidification rate and the computed values of the isenthalpic effectiveness for the tested outdoor and regeneration temperatures. The isenthalpic effectiveness mostly varies between 0.75 and 0.95, the highest values being reached for higher outdoor temperatures and lower regeneration temperatures. Since the extra sensible heat transfer is proportional to the difference between the regeneration and outdoor temperatures, those conditions result in the dehumidification process closest to the ideal one.

4.2 Empirical correlations

It has been established that the dehumidification rate and the isenthalpic effectiveness at a fixed rotation speed depend only on the outdoor and regeneration temperatures. Empirical correlations can be developed to ease the modelling of constant rotation

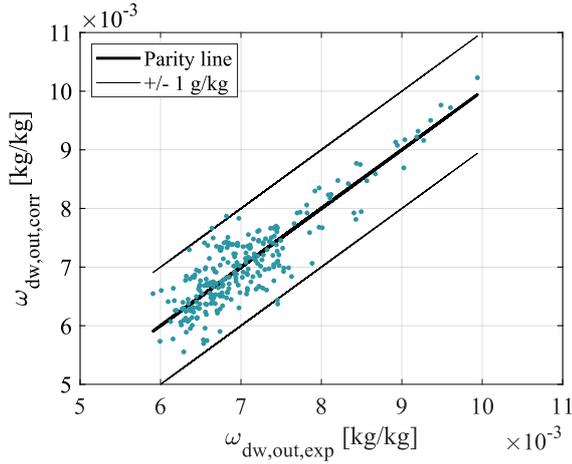


Figure 7: Measured and calculated specific humidity at the desiccant wheel outlet.

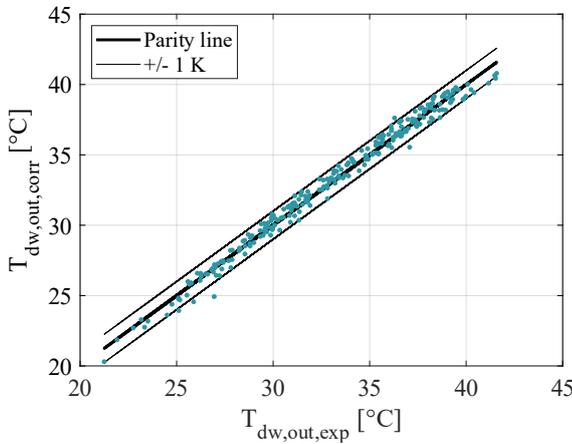


Figure 8: Measured and calculated temperature at desiccant wheel outlet.

speed desiccant wheels in practical applications. The correlation expressions take the following form:

$$X = A \cdot (T_{out})^m \cdot (T_{reg})^n \quad (13)$$

where X represents the parameters $\Delta\omega$ and ε_h ; and A , m and n are constant coefficients which depend on the operating and geometrical parameters of the DW. The computed coefficients for the studied DW are detailed in Table 1.

Table 1: Values of coefficients adopted in the $\Delta\omega$ and ε_h correlations.

X	A	m	n
$\Delta\omega$	$2.385 \cdot 10^{-4}$	-0.9966	1.4569
ε_h	1.6664	0.2630	-0.3825

The comparison between measured and calculated temperature and specific humidity at the DW outlet is shown in Figures 7 and 8. 40% of the dataset is used to fit the model parameters and the other 60% is used for model validation. With these simplified

correlations, the outlet air conditions can be computed with an accuracy of ± 1 K for the temperature and ± 1 g/kg for the humidity.

4.3 Limitations

The performance of a desiccant wheel strongly depends on its chemical and physical properties. In this work, the analysed desiccant wheel is made of aluminium foils covered with silica gel and channels with sinusoidal cross-sectional area. The correlations developed in Eq. (13) should be calibrated based on experimental data. It should be noted that one set of coefficients is valid only for one desiccant wheel geometry and for a constant rotation speed.

In this case, the air conditions at the DW outlet depend only on the inlet temperatures. Temperature ranges for the inlet conditions should be established to define the validity range of the correlations. Figure 9 represents the set of inlet air conditions that have been observed for the considered DW on the process and regeneration sides. On the regeneration side, the inlet specific humidity was rather constant since the regeneration air source was the indoor environment. For applications with inlet air conditions subject to wider variation ranges, the proposed empirical correlation could be modified to account for the inlet specific humidity on the regeneration side.

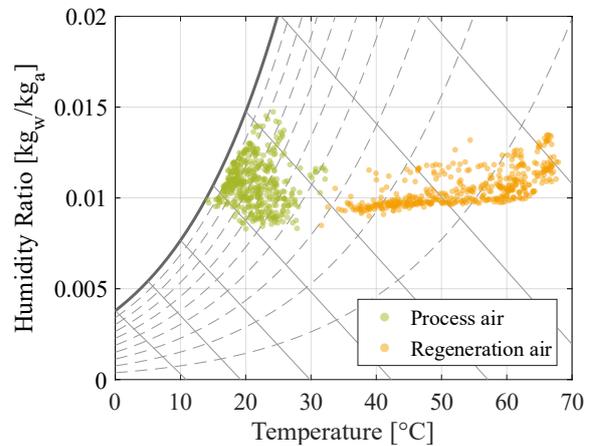


Figure 9: Operation range of the air conditions at the inlets of the DW on process and regeneration sides.

5 Conclusions

In this paper, a novel performance indicator for desiccant wheels has been developed from the thermodynamic analysis of the evaporative cooling and desiccant dehumidification processes. It is shown that the desiccant dehumidification process can be viewed as a combination of isenthalpic

dehumidification and sensible heating, leading to the definition of the isenthalpic effectiveness.

The application of the definition of the isenthalpic effectiveness to an actual operating DW shows that the isenthalpic effectiveness is a suitable indicator to characterise the performance of the DW. Empirical correlations are proposed for the computation of the dehumidification rate and the isenthalpic effectiveness depending on the outdoor and regeneration temperatures. With the proposed correlations, the outlet air conditions are predicted with an accuracy of ± 1 K for the temperature and ± 1 g/kg for the specific humidity.

Acknowledgement

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