Preprints of the 19th World Congress
The International Federation of Automatic Control
Cape Town, South Africa. August 24-29, 2014

Dynamic modelling of desiccant wheels for the design of energy-efficient air handling units

Alberto Leva[a], Erica Zavaglio[a], Marco Bonvini[b]

[a] *Politecnico di Milano, Dipartimento di Elettronica, Informazione e Bioingegneria Piazza Leonardo da Vinci, 32 - 20133 Milano, Italy*
[b] *Lawrence Berkeley National Laboratory 1 Cyclotron Road, 94704 Berkeley, CA*
(e-mail: {alberto.leva,erica.zavaglio}@polimi.it, mbonvini@lbl.gov)

Abstract: Desiccant wheels can improve the efficiency of air handling units, as their inherent heat recovery capabilities reduce the external energy needs. On the other hand, however, desiccant wheels introduce state variable interactions that are not present in traditional units. Hence, to actually yield the possible advantages, air handling units with desiccant wheels require accurate control, which in turn calls for dynamic wheel models that can be parametrised from design data, so as to allow for virtual prototyping, and couple reliability with numerical efficiency, to deliver the numerous system-level simulations required with an acceptable computational effort. This paper proposes a dynamic model for desiccant wheels fulfilling the above requirements. To support the proposal, examples ere reported on the use of the proposed model for the design and control of innovative unit layouts.

Keywords: Dynamic modelling, desiccant wheels, energy efficiency, air handling units.

1. INTRODUCTION

Air Handling Units (hereinafter AHUs for short) are a primary component of air conditioning systems. Their purpose is to provide air renovation, together with maintaining the temperature and humidity of the conditioned ambient at prescribed reference values. Achieving such a goal obviously requires a significant amount of energy, thus making research on AHU layouts and controls important for the attainment of efficiency objectives that are nowedays becoming more and more stringent.

Focusing as an example on summer AHU operation, and simplifying the discussion for introductory purposes, air needs de-humidifying and cooling. In traditional AHUs, these two tasks are respectively carried out by a cooling coil, which takes care of reaching the desired humidity, and by a heating one located downstream, that restores the air cooled down by de-humidification to the required temperature. Two energy inputs are thus required, and most important, the (essentially latent) heat drawn from air in the de-humidification stage, is lost. On the other hand, such AHUs are substantially triangular systems, as temperature control does not affect humidity control.

Desiccant Wheels (DWs) are composed of a rotary element with air channels in the axial direction. Due to rotation, each channel is put in contact alternatively with the air being conditioned, termed the "process" air, and with another warmer air flow, called the "regeneration" one. Channel walls are covered with a material that absorbs humidity from the process air, and releases it to the regeneration one. As a result, the process air is de-humidified by absorption toward a warm surface, not by condensation onto a typically too cold one, which reduces or eliminates the need for the downstream heating coil. Moreover, to obtain regeneration air, low-enthalpy sources are suitable, like for example solar thermal collectors or engine exhausts, with apparent additional advantages. Finally, also the power for the cooling coil is reduced, as part of the de-humidification process is carried out by the DW. This permits for example to operate with higher evaporation temperatures of the coolant, thus increasing the Coefficient of Performance (COP) of the required refrigerating machine. However, the phenomena taking place in a DW inherently introduce a significant temperature-humidity cross-coupling, thus making the control of an AHU with a DW more complex than that of a traditional AHU.

Summarising, DWs are a promising addition to AHUs, but their effective use requires a thorough unit and control design, thus adequate dynamic simulation models. In this respect, the contributions of this paper can be summarised as follows.

- A DW dynamic model is proposed that fills a gap, observed in the literature, between three-dimensional fine-scale models, and phenomenological data-based ones. Models of the former type, owing to their inherent complexity, are not well suited for control-oriented system-level studies, that may require a large number of simulations. Models of the latter type, conversely, are intrinsically tied to the *scenarii* that provided the data: as such, they cannot be parametrised from design information only, nor can be guaranteed to be valid if one has to consider operating conditions that are significantly different from said *scenarii*.
- Examples of design-oriented simulation studies are presented, compatibly with space limitations, to witness that the accuracy and computational efficiency of DW models obtained with the proposed approach actually fit the mentioned design needs.

2. BRIEF LITERATURE REVIEW

A good review on DW-related research is La et al. [2010], to which the interested reader is referred. Here, for space reasons, we just point out some facts that are relevant to the presented work, and provide its motivation.

Since designing a DW involves both the geometry of the rotary element and the characteristics of the desiccant material, numerous studies were performed with very detailed models, typically accounting for the three-dimensional nature of the device, see for example Antonellis et al. [2010]. Also, to reduce complexity, the attempt was made to use empirical correlations drawn from manufacturer's data [Butera et al., 2002], and to exploit analogies with rotating sensible heat exchangers [Stabat and Marchio, 2009], most frequently using the so called "characteristic potentials" method to predict the coupled heat and mass transfer. Moving up from the design of the DW to that of the AHU, models were proposed for optimising the unit layout, and choosing the best operating point for the DW, see e.g. Antonellis et al. [2010], Panaras et al. [2011a]. As for AHU control finally, numerous works can be found, ranging from PID-based solutions Tashtoush et al. [2005], up to neural networks [Guo et al., 2007] and model-based schemes [He et al., 2005]. However, at least to the best of the authors' knowledge, a minority of said works deal with control of DW-endowed AHUs, examples being Panaras et al. [2011b], Vitte et al. [2008].

Summarising, to date the literature offers basically two types of DW models. Complex ones often require three spatial dimensions, and are poorly suited for studies that may involve many simulation runs. Simple ones – e.g., based on characteristic potentials – are more efficient, but have two main drawbacks. First, they are natively steady-state. When dynamics needs introducing, which is a necessity for control studies, heuristic and often quite simplistic solutions are adopted, based e,g. on residence times or similar ideas; however, such solutions may fall short of perfection in catching the real physics, especially when demanding control specifications require to reliably know the system dynamics at quite fast time scales. Second, the parameters of the mentioned simple models have typically no direct physical meaning, making it difficult to describe a machine that is being designed together with its control. Indeed, models suited for effective system-level studies should in fact fill the so observed complexity gap.

The DW modelling approach presented herein is an attempt to solve the issue just mentioned. It is based on first-principle equations, thus with physically meaningful parameters, includes spatial discretisation at a detail level suitable for system and control studies, and takes care of achieving a reasonably low computational complexity.

3. THE PROPOSED DW MODEL

As exemplified in Figure 1, a DW is a rotating cylinder made by a matrix of support material, that realises numerous small axial channels, and is covered by a layer of desiccant. The wheel is exposed to two two air flows – the mentioned "process" and "regeneration" one – with opposite directions. Each flow impacts a portion of the wheel, thus defining a regeneration and a process area. By its rotation, the wheel makes each channel take part alternatively in both flows, thereby providing heat and humidity transfer.

The main difficulties in structuring a model for the relevant phenomena come thus from their native three-dimensional nature, including the wheel motion. To achieve a complexity level suitable for this work, thus, we propose a *two*-dimensional spatial discretisation, assuming that airflows are distributed uniformly enough in the radial direction, which is quite reasonable in practice, and we also avoid describing the wheel rotation explicitly.

Figure 1 summarises the proposed spatial discretisation: the wheel, of length L and radius R, is divided in $N \times M$ volumes or slices, that correspond to a fixed region of the space (i.e., they do not rotate with the wheel). Each slice covers an angle $\theta = 360°/N$, and has length $dl = L/M$. The structure of the channels that are part of the support matrix is described by a and b, that define respectively its height and the length. Furthermore, the proposed modelling framework is based on the following assumptions:

- axial heat conduction and water vapour diffusion in the air are negligible;
- there are no radial temperature or moisture content gradients in the matrix;
- hysteresis in the sorption isotherm for the desiccant coating is neglected and the heat of sorption is constant;
- all channels are identical, with constant heat and mass transfer surface areas, adiabatic, and impermeable
- the matrix thermal and moisture properties are constant, as are the mass and heat transfer coefficients, and the adsorption heat per unit mass of adsorbed water;
- mass carry over between two air flows is neglected.

Each slice is characterised by the effective efflux area A_{eff}, defined as

$$A_{eff} = \frac{\theta \pi R^2}{360°} - n\left(\pi\frac{b}{2} + 2\left(a - \frac{b}{2}\right)\right)dx \tag{1}$$

where dx is the thickness of the matrix structure,and

$$n = \frac{\theta \pi R^2}{ab \cdot 360°} \tag{2}$$

is the number of channels contained in the slice. The support material has a lateral area, A_{xcg}, defined as

$$A_{xcg} = 2n\left(\pi\frac{b}{2} + 2\left(a - \frac{b}{2}\right)\right)dl. \tag{3}$$

and the adsorption/desorption rate, like the convective heat exchange between wheel and air, are proportional to this area.

The water mass flow rate w_{ad} adsorbed by the desiccant material (positive if adsorbed) is a function of the adsorption isotherm of the desiccant material Kodama et al. [2001], i.e.,

$$w_{ad} = \frac{M_d}{\tau}\left(0.24\phi^{\frac{2}{3}} - X_d\right) \tag{4}$$

where M_d is the mass of the desiccant material, ϕ is the relative humidity, τ is a discharge time constant, and

$$X_d = \frac{M_{wd}}{M_d} \tag{5}$$

is the desiccant water content defined, i.e., the ratio between the mass of water inside the desiccant material (M_{wd}) and that of the same material. The discharge time constant is

$$\tau = \frac{M_d}{A_{xcg} h_m} \tag{6}$$

where h_m is a mass transfer coefficient. Such an adsorption/desorption process then also brings in a convective heat rate q_{cam} from air to desiccant material, obtained as

$$q_{cam} = h_t A_{xcg}\left(T_a - T_m\right) \tag{7}$$

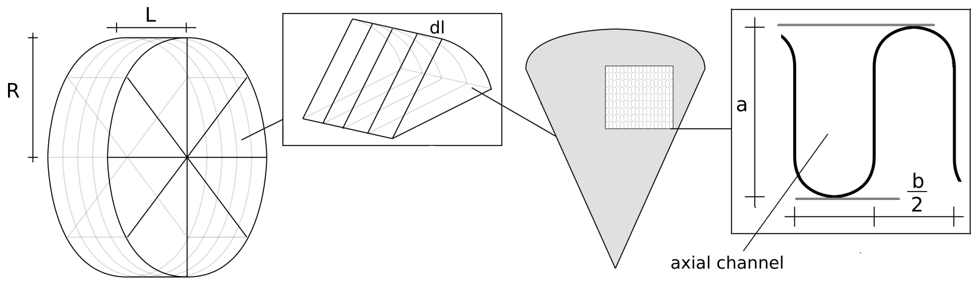

Fig. 1. Example of desiccant wheel structure illustrating the used spatial discretisation.

where h_t is the convective heat transfer coefficient, and T_a, T_m are respectively the air and matrix temperatures. There is also a heat rate q_{wam} associated to the water mass flow rate adsorbed or released by the desiccant material, i.e.,

$$q_{wam} = w_{ad} \begin{cases} h_{wv}, & \text{if } w_{ad} \geq 0 \\ h_{wm}, & \text{if } w_{ad} < 0 \end{cases} \quad (8)$$

where h_{wv} is the water vapour specific enthalpy, and h_{wm} the water specific enthalpy in the desiccant material. Once mass and heat flows are defined, the mass and energy balance equations can be written as

$$\dot{M_{wd}} = w_{ad} \quad (9)$$
$$\dot{M_a} = m_{in} - m_{out} - w_{ad} \quad (10)$$
$$\dot{M_{wa}} = m_{in}X_{in} - m_{out}X_{out} - w_{ad} \quad (11)$$
$$\dot{E_m} = q_{wam} + q_{cam} \quad (12)$$
$$\dot{E_a} = m_{in}h_{in} - m_{out}h_{out} - q_{wam} - q_{cam} \quad (13)$$

where M_{wd} is the mass of water in the desiccant, M_a, M_{wa} the masses of moist air and vapour in the volume, E_m, E_a the energies of the matrix and desiccant material, and of the moist air contained inside the volume; m_{in} and m_{out} are respectively the incoming and outgoing moist air mass flow rates, while X_{in}, h_{in}, X_{out} and h_{out} are the water vapour mass fraction and specific enthalpies respectively associated to said flow rates.

Each slice is modelled as an equivalent duct, made of control volumes connected by elements that define a pressure/flow relationship, see Figure 2. The relationship between the mass flow rate w and the pressure drop dp is assumed linear, thus

$$dp = k\,dl\,w \quad (14)$$

where dl is the element length, and k the friction factor per unit length. All elements are have the same friction factor; the first and last pressure drop elements have a length of $dl = L/(2M)$, the others of $dl = L/M$.

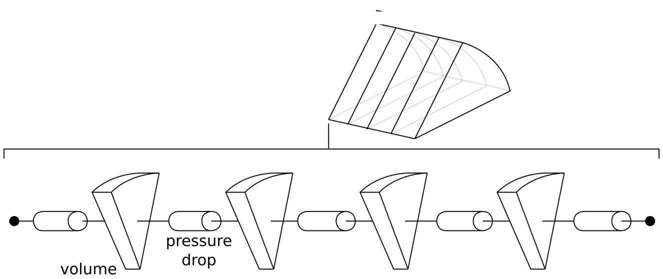

Fig. 2. A wheel slice modelled as an equivalent duct.

A key point to achieve the desired simplicity is the adopted (and novel) way of representing the wheel rotation. We do that by introducing an equivalent flow of material in the angular direction, according to the wheel speed. Said flow is thought of as composed by the desiccant (with its water content) and the matrix material. Each slice, with all its control volumes as per Figure 2 is thus spatially fixed (i.e., it does not move with the wheel), and is traversed in the axial direction by either the process or the regeneration airflow, and in the angular direction by the desiccant flow, as summarised in Figure 3.

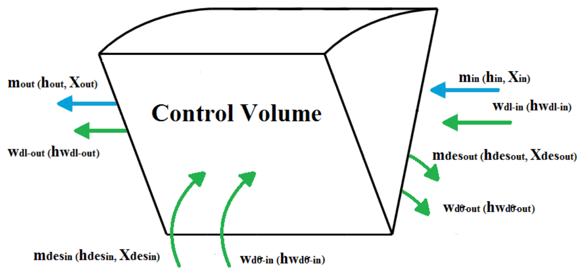

Fig. 3. Air, water and desiccant mass flows in a slice control volume.

The phenomena in each control volume are structurally the same as those of a single channel, which is rigorous as long as one considers the number of channel in each volume to be

$$n = \frac{\theta \pi R^2}{a\,b\,360^\circ} \quad (15)$$

where θ is the angle of the slice, and the other quantities have the meaning of Figure 1. The mass and energy balance for each volume are written trivially, with the sole exception of the desiccant mass one, which reads, with the notation of Figure 3,

$$mdes_{out} = \left(\frac{R^2}{2}\,dl\,\omega_w\,(\rho_D + \rho_M)\right)\left(1 + \frac{Mwm}{M_D + M_M}\right). \quad (16)$$

where ρ_D, ρ_M, M_D and M_M are the densities and masses of the desiccant and the matrix material, and ω_w is the DW angular velocity. We omit further details for brevity, but it is important to stress that the introduced simplification for describing the wheel motion, peculiar to the proposed approach, greatly enhances simulation efficiency.

The proposed DW model was validated versus experimental data from Kodama et al. [1993], exhibiting good precision and performance. Given the scope of this paper, however, we omit details on this matter, and just report two figures (reference data are in the paper quoted above). Figure 4 shows the process air outlet humidity for various inlet process air conditions and rotating speeds, thus backing up the proposed modelling approach. Figure 5 shows the spatial distribution of humidity in the matrix, thereby specifically supporting the adopted equivalent motion representation.

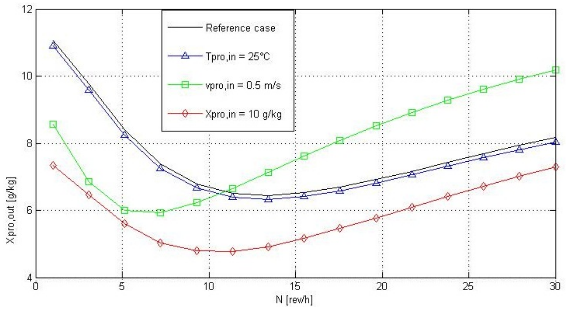

Fig. 4. Example of simulated operation with different inlet conditions and speed.

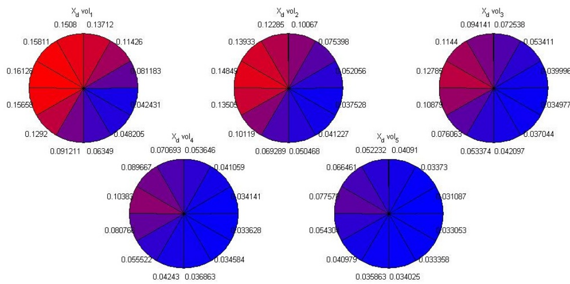

Fig. 5. Example of simulated spatial distribution of matrix humidity with five volumes per slice.

4. DESIGN AND CONTROL STUDIES

We now present two simulation studies aimed at designing an AHU with a DW, and the corresponding control system. The first study refers to an AHU of conventional configuration, and aims at quantifying the energy efficiency improvement that can be achieved by adding a DW. The second one conversely explores, albeit at a still preliminary level, the novel idea of using the DW rotation speed as a control variable. The main goal of this section, with particular reference to the second of the presented cases, is to evidence that similar studies are greatly facilitated – not to say, in practice, enabled – by the approach on which the presented model is grounded.

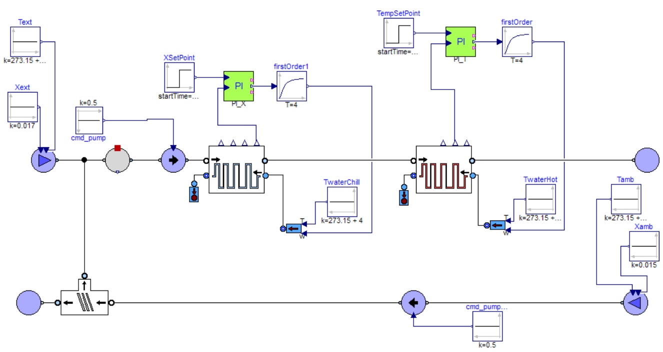

Fig. 6. Modelica model of a traditional AHU with temperature/humidity control.

The DW model of Section 3 was implemented in the Modelica language, and used to construct two models of complete AHUs, one with a very classical layout and one with the addition of a DW. The corresponding Modelica schemes, including control, are shown in Figures 6 and 7, respectively.

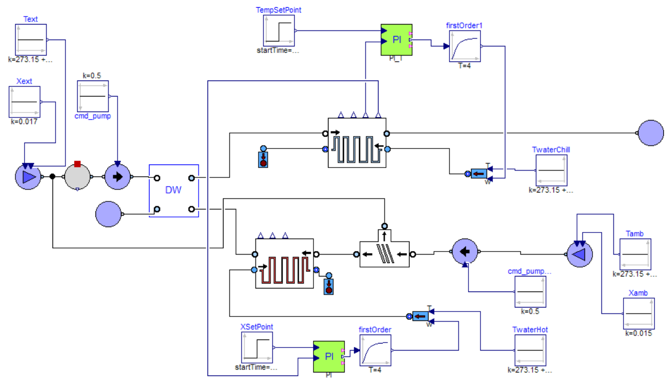

Fig. 7. Modelica model of an AHU with DW and temperature/humidity control.

4.1 *Endowing an AHU with a DW to improve energy efficiency*

In this example, the PI-based control schemes shown in Figures 6 and 7 were tuned based on simulated open-loop step response for "quite similar" closed-loop transients in the time domain. As shown by Figures 8 and 9, the slower dynamics introduced by the significant heat capacity of the wheel have a visible effect, but the obtained performance is acceptable in both cases.

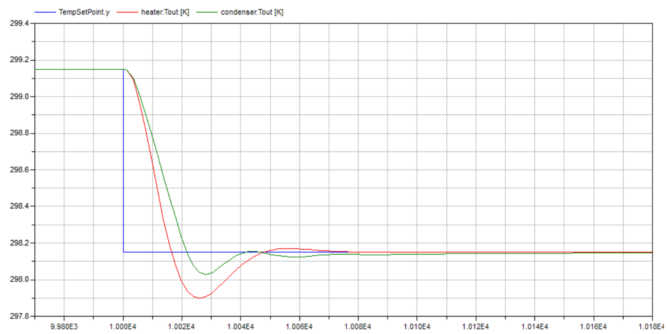

Fig. 8. Response to a temperature set point step with the conventional AHU of Figure 6 (red) and the DW-endowed one of Figure 7 (green).

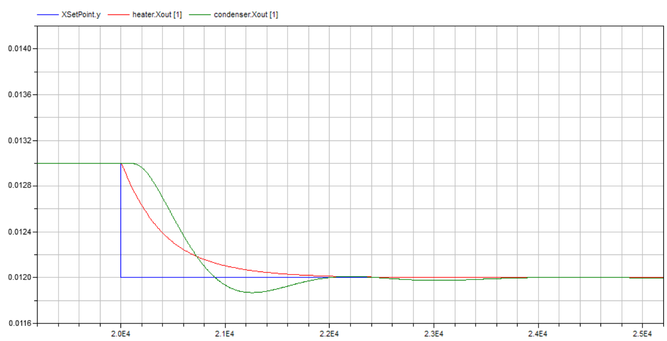

Fig. 9. Response to a humidity set point step with the conventional AHU of Figure 6 (red) and the DW-endowed one of Figure 7 (green).

Of course improvements could be achieved with some fine tuning, but this is not the scope of this example, that is centred on energy saving. To this end, a 14-hour simulation was carried out in typical summer operating conditions (we omit details for brevity) for both the considered AHU configurations. The results are summarised in Table 1, where the advantages can be

Consumption [kWh]	Standard AHU	AHU with DW	Saving %
Cooling	119.7	53.5	55.3
Heating	49.1	43.7	11.0
Total	168.9	97.2	42.5

Table 1. Energy saving obtained with a DW.

appreciated particularly in terms of cooling energy, since the DW allows to de-humidify the conditioned air without the need for a temperature decrease as large as in the case without DW (in the latter case, the air temperature in fact needs to be brought below the wet bulb one). It is finally worth noticing that the average duration of a simulation run is about $2.5\,s$ for 14 hours of simulated time, proving the efficacy of the proposed models for the intended type of studies.

4.2 Using the DW rotation speed for control

In the control schemes used so far, the two control variables are the "classical" ones – i.e., the cooling and heating flowrates q_c and q_h – and control was designed (simplifying for brevity) by pairing the former to humidity, and the latter to temperature. When using a DW, its angular velocity ω_w was thus kept constant, the achieved energy efficiency improvement coming from a lower cooler load. However, since the main energy saving is from cooling, two questions arise. The first one, already considered in the literature, is how to determine the optimal ω_w given the operating condition, providing finer control than is provide by the classical "low" and "fast" speed setting for "de-humidification mode" and "heating mode", respectively. Of course, the proposed model can be helpful in the above respect, also addressing dynamic aspects if this is required, but given space limits we do not delve into details on this matter.

The second question, on which conversely we now concentrate, is whether or not, and in the affirmative case when, one may advantageously use ω_w instead of q_c as a control variable. More precisely, the idea is to try to control the process air temperature and humidity with ω_w and q_h—incidentally, making the cooling coil activation not mandatory anymore. We thus investigated how ω_w and the inlet regeneration air temperature $T_{reg,in}$ affect the process air outlet conditions. The temperature $T_{pro,out}$ of the process air at the DW outlet is clearly always above the inlet one, regardless of the chosen $(\omega_w, T_{reg,in})$ couple. The interest for the proposed configuration is thus at present limited to situations in which de-humidifying and heating is necessary, i.e., essentially to winter operation, and possibly sudden environmental changes like summer storms. In addition, it is worth noticing that the most straightforward way to include a DW in an AHU preserves the heater downstream the wheel in the process air path, to introduce additional heating when needed. In the opinion of the authors, by adopting the control scheme here preliminarily proposed and that employs ω_w as a control variable, the need for that heater could be removed.

The used scheme is shown in Figure 10. The wheel velocity control has to act only when $T_{pro,out}$ falls below the set point. In the opposite case, ω_w needs keeping at a minimum (saturation) level. This level must be chosen properly, and a possible method to do so will be discussed shortly. Obviously also a maximum saturation value has to be set in order to maintain the DW in the correct operating range, but this proved far less critical.

The obtained results are exemplified in Figures 11 and 12, that show the effect of a slowly sinusoidal variation of the external temperature and humidity. Without acting on ω_w, keeping

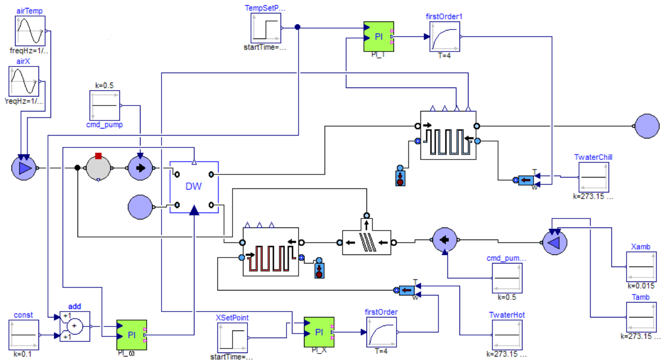

Fig. 10. Modelica model of the studied control configuration using ω_w.

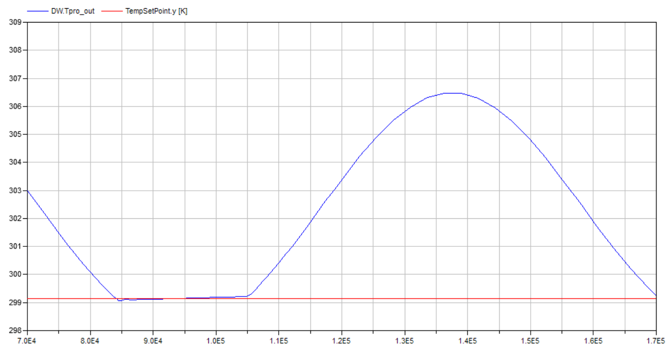

Fig. 11. Control via ω_w: $T_{pro,out}$ and its set point.

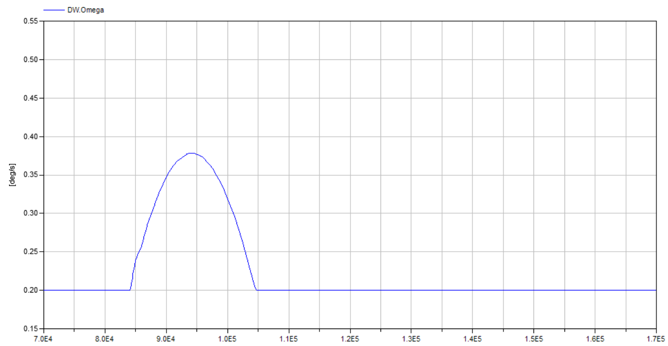

Fig. 12. Control via ω_w: wheel velocity.

$T_{pro,out}$ at the set point would require the intervention of the heating coil, while the control-driven increase of ω_w makes this unnecessary. Since the additional motor power is much lower than the required heater power, the advantage is apparent.

As for the selection of the minimum DW velocity, we can recall that the critical element is the cooler. Quite intuitively, thus, the higher its inlet air temperature is, the larger its energy consumption will be. The considered temperature depends on the wheel operation and corresponds to $T_{pro,out}$. Thus, if we succeed in reaching the desired humidity X_{pro}, out with the minimum $T_{pro,out}$, the energy consumption of the cooler will be the minimum possible.

To this end, Figure 13 reports X_{pro} versus $T_{pro,out}$ for various values of ω_w, one per curve. On each curve, dots mark different values of the regeneration air inlet temperature. By choosing the lowest wheel velocity that permits to reach the desired humidity, the outlet process air temperature is brought to reach the minimum value.

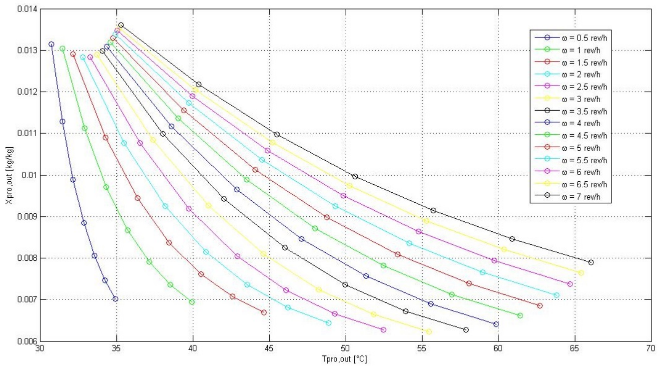

Fig. 13. Example of DW operational range.

Figure 13 shows another interesting result. The same value of X_{pro} is reached for a different temperature value, depending on the wheel velocity. Thus, the choice of the minimum ω_w results in the need of higher regeneration air temperature to reach the desired humidity value for the process air. The mode then evidences that in general (i.e., abstracting from the particular transients shown above) two opposite effects of reducing the wheel velocity need considering, namely a decrease in the cooling energy demand, but at the same time an increase in the heating energy one. By performing a set of simulations for the *scenarii* at hand, one can thus use the proposed model to effectively provide an answer.

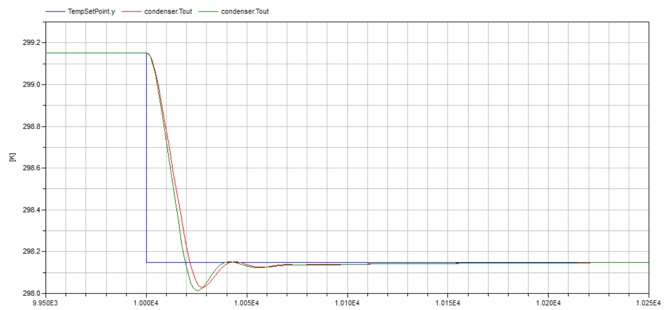

Fig. 14. Temperature set point step response with ω_w set to $3\,rev/hr$ (red) and $0.5\,rev/hr$ (green).

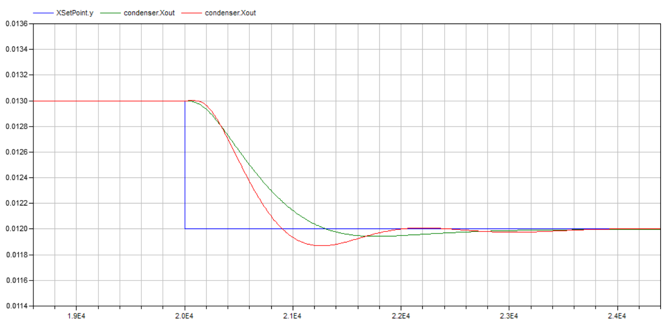

Fig. 15. Humidity set point step response with ω_w set to $3\,rev/hr$ (red) and $0.5\,rev/hr$ (green).

We omit this, however, and finally focus on investigating the *dynamic* effects of changing ω_w. Figures 14 and 15 show the effect of changing ω_w on closed-loop responses of interest. As can be seen, the result is acceptable, thereby further showing the usefulness of a *dynamic* DW model of good computational efficiency, as that proposed herein.

5. CONCLUSIONS AND FUTURE WORK

A model of a desiccant wheel was proposed to fill a complexity/efficiency tradeoff gap observed in the literature, thereby allowing for system-level design- and control-oriented studies on energy efficient air handling units. The proposed model is dynamic, thereby not limiting its use to component sizing, can be parametrised from design data, so as to allow for virtual prototyping, and couples good reliability with numerical efficiency, to withstand the potentially large number of required simulations with an acceptable computational effort.

Besides describing the model and the underlying approach, some selected uses of said model were briefly presented and commented on, to support both the model itself and its claimed usefulness for the targeted applications.

Future work will include further validations of the model, and its use to devise more energy efficient unit layouts, and their control.

REFERENCES

S. De Antonellis, .M. Joppolo, and L. Molinaroli. Simulation, performance analysis and optimization of desiccant wheels. *Energy and Buildings*, 42(9):1386 – 1393, 2010.

M. Butera, F. Guanella, R. Adhikari, and R.S. Beccali. Performance evaluation of rotary desiccant wheels using a simplified psychrometric model as design tool. In *The European Conference on Energy Performance and Indoor Climate in Building (EPIC 2002 AIVC)*, 2002.

C. Guo, Q. Song, and W. Cai. A neural network assisted cascade control system for air handling unit. *Industrial Electronics, IEEE Transactions on*, 54(1):620–628, 2007.

M. He, W.J. Cai, and S.Y. Li. Multiple fuzzy model-based temperature predictive control for hvac systems. *Information sciences*, 169(1):155–174, 2005.

A. Kodama, M. Goto, T. Hirose, and T. Kuma. Experimental study of optimal operation for a honeycomb adsorber operated with thermal swing. *Journal of Chemical engineering of Japan*, 26(5):530–535, 1993.

A. Kodama, T. Hirayama, M. Goto, T. Hirose, and R.E. Critoph. The use of psychrometric charts for the optimisation of a thermal swing desiccant wheel. *Applied Thermal Engineering*, 21(16):1657 – 1674, 2001.

D. La, Y.J. Dai, Y. Li, R.Z. Wang, and T.S. Ge. Technical development of rotary desiccant dehumidification and air conditioning: A review. *Renewable and Sustainable Energy Reviews*, 14(1):130 – 147, 2010.

G. Panaras, E. Mathioulakis, and V. Belessiotis. Solid desiccant air-conditioning systems design parameters. *Energy*, 36(5): 2399 – 2406, 2011a.

G. Panaras, E. Mathioulakis, and V. Belessiotis. Proposal of a control strategy for desiccant air-conditioning systems. *Energy*, 36(9):5666 – 5676, 2011b.

P. Stabat and D. Marchio. Heat and mass transfer modeling in rotary desiccant dehumidifiers. *Applied Energy*, 86(5):762 – 771, 2009.

B. Tashtoush, M. Molhim, and M. Al-Rousan. Dynamic model of an hvac system for control analysis. *Energy*, 30(10):1729–1745, 2005.

T. Vitte, J. Brau, N. Chatagnon, and M. Woloszyn. Proposal for a new hybrid control strategy of a solar desiccant evaporative cooling air handling unit. *Energy and Buildings*, 40(5):896 – 905, 2008.