Feasibility of Controlling Heat and Enthalpy Wheel Effectiveness to Achieve Optimal Closed DOAS Operation

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ABSTRACT

Dedicated outdoor air systems (DOAS) make use of the sensible and latent capacity of the return (exhaust) air to cool and dehumidify the supply (outside) air. “Closed DAOS” is an advanced form of typical DOAS with desuperheater and evaporatively cooled condenser in the return air stream. The air entering the condenser in Closed DOAS is in the fully saturated state. In order to fully saturate the return air, it needs to be sprayed by water. Sufficient amounts of water is required for fully saturating the air and keeping the coils of the condenser wet during the process. This water is obtained internally as a result of condensation on the evaporator fins, not requiring additional water use at the buildings site.

Closed packaged configuration requires to have a water balance of water obtained in evaporator to that required for saturating the air and wetting the condenser coil. This water balance can be achieved under most conditions by an optimal control of enthalpy wheel and heat wheel (run around heat exchanger) effectiveness to control the amount of water obtained in the evaporator. The optimal control of heat and enthalpy wheel effectiveness is not only important for making the necessary water balance but also to keep the latent load as low as possible. This paper discusses the Closed DAOS configuration with psychrometrics and water balance analyses.

1. INTRODUCTION

Energy use in buildings has become a topic of serious consideration due to improved living standards coupled with increased fuel and electricity price. Heating, ventilation, and air-conditioning (HVAC) is typically the largest load component in buildings. Indoor air quality is important for health and thermal comfort inside the building and has recently gained much attention (Mo JH et al, 2005). By some estimate, up to 50% of the building’s total energy demand is utilized in ensuring thermal comfort for its occupants (Enteria, 2011).

In United Arab Emirates (UAE), weather conditions are extremely hot and humid for the majority of the (year round) cooling season. During peak summer days, about 60-75% of the building’s electrical energy is consumed by the cooling systems and these systems constitute around 40-55% of UAE’s annual energy consumption (Ali, 2011). The development of energy efficient systems can result in considerable energy reduction. Ventilation air dehumidification plays a growing role in HVAC industry. It satisfies the minimum indoor air quality while minimizing the energy consumption by separating sensible and latent loads.

A dedicated outdoor air system (DOAS) handles ventilation and humidity load and part of sensible heating/cooling load. The remaining part of sensible load is met by a parallel system such as (passive) chilled beams that operate at a
higher temperature to avoid condensation. DOAS are typically employed in buildings that do not use a constant or variable air volume systems. Improving DOAS efficiency can directly impact the energy consumption of cooling systems and can result in considerable energy savings.

S. Mumma et al (2002) analyzed DOAS in combination with ceiling radiant cooling panels (CRCP). They found that if DOAS fully manages the latent loads then the sensible loads can be managed by operating the chilled water at higher temperature than used for the DOAS. This can result in the reduction of energy consumption and increased COP due to reduced compressor pressure ratio since chilled water is now supplied at higher temperature.

In this paper, different established DOAS configurations are discussed and a new DOAS configuration, closed packaged DOAS is introduced and analyzed.

2. DOAS CONFIGURATIONS

Flexible DOAS models allow to investigate a range of configurations, any of which might give an optimal performance for a set of air and refrigerant side boundary conditions. Most of the work on these configurations was established by Zakula (2013). Three different DOAS configurations are shown in Figure 1.

![Figure 1: Various DOAS configurations](image)

System 1 is a typical DOAS consisting of an enthalpy wheel, an evaporator in the supply stream, and an outdoor condenser. System 2 is a modification of system 1 with a run around heat exchanger around the evaporator to precool and reheat the air as it enters and leaves the evaporator. Wallin et al. (2012) has shown that annual heat recovery of up to 47% is possible using a run around heat exchanger around evaporator. The condenser in system 1 is split into condensing, desuperheating, and subcooling sections in system 3 to obtain better performance under high ambient conditions, specifically achieving entering air temperatures below the ambient for the condensing and subcooling section. The desuperheater can be operated using ambient air to reduce the load on the condensing section. According to Mumma et al (2010), it is important to have a balanced flow between the supply and return stream, otherwise with an imbalanced flow, the energy recovery rate is considerably decreased.

3. DOAS WITH ENTHALPY WHEEL AND SENSIBLE HEAT WHEEL

As discussed in the preceding section, system 2 has shown energy savings due to run around heat exchanger around
the evaporator. This is because in system 1, the evaporator is bringing the temperature of the air below its dew point temperature to ensure complete latent cooling. However, reheating is required in order to bring the supply air temperature according to ASHRAE Standard 62-1999 for minimum air temperature inside the building. The reheating in case of system 2 is provided by the run around heat exchanger, which at the same time precools the air entering the evaporator, thus simultaneously decreasing the evaporator load. The run around heat exchanger is modeled using sensible heat wheel. Figure 2 shows the schematic of DOAS with heat wheel (HW) and enthalpy wheel (EW) and includes state point locations. To reheat the air after the evaporator, heat wheel heats the air from point 4 to 5 by cooling the air entering the evaporator from point 2 to 3 as shown in Figure 2.

![Figure 2: Balanced Flow DOAS with heat wheel and enthalpy wheel](image)

Analysis of the DOAS with heat wheel and enthalpy wheel is shown on the psychrometric chart in Figure 3. It can be seen from the psychrometric analysis that the heat wheel reduces the temperature of the outdoor air entering the evaporator close to its dew point temperature by transfer the heat to the air leaving the evaporator. This way the evaporator load is reduced along with the energy required to reheat the supply air.

![Figure 3: Balanced flow DOAS with heat wheel and enthalpy wheel](image)

(EW effectiveness = 90%, HW effectiveness = 90%)
4. CLOSED Packaged DOAS

The dehumidification of humid outdoor air in the evaporator produces condensed water. The amount of condensed water obtained in the evaporator depends on the specific humidity of the outdoor air. This water can be used to improve heat pump performance. Condenser water from the evaporator can be used to remove heat from refrigerant in the condenser. For humid climates, it is likely that the amount of condensed water could be sufficient to remove the heat of condensation of the refrigerant.

Closed DOAS. The schematic of a closed DOAS is shown in figure 4. The condensed water from the evaporator is sprayed on to the air entering the condenser to bring it near its saturation point. Part of the condensed water is sprayed directly on to the condenser coils to increase the heat transfer coefficient that improves the heat transfer process in condenser. The air after the condenser is still cooler than the outdoor air. As a result, this air is used for the desuperheating of the refrigerant. The above process result in lower condensing temperatures. The lower condensing temperatures results in lowering compressor pressure ratio which in turn decreases compressor work and increases system COP.

![Figure 4: Schematic for Closed DOAS](image)

4.1. First Law Analysis of Closed DOAS

The feasibility analysis is conducted using different values of heat wheel and enthalpy wheel effectiveness. The outdoor air stream is passed through the enthalpy wheel where it exchanges heat and moisture with the return air stream (1-2 in Figure 4). After passing through enthalpy wheel, air is passed though heat wheel (2-3) that act as a run around heat exchanger to get air sensibly cooled to a near saturation condition. The saturated or near saturated air is then sensibly and latently cooled by the evaporator (3-4) such that air leaving the evaporator is at a cooler and dryer point on the saturated line. The air leaving the evaporator is controlled to a specific humidity of 9 g/kg which corresponds to 24°C dry bulb temperature and 50% relative humidity (ASHRAE Standard 62-1999). To achieve these conditions, air needs to be heated after the evaporator. This is achieved by passing part of the air after the evaporator through the heat wheel (4-5) where it cools the air entering the evaporator and rest of the air is passed through subcooler (4-5') to lower the liquid refrigerant temperature. For feasibility analysis, the evaporator surface temperature is assumed to have a constant value of 12°C.

Using evaporator heat transfer rate, \( Q_e \), the mass flow rate of refrigerant, \( \dot{m}_r \), is calculated using equation 1.

\[
Q_e = \dot{m}_r (h_{r,e, out} - h_{r,e, in})
\]
Water obtained due to the condensation of moisture in air in the evaporator is calculated using equation 2.

\[ m_{w,\text{avail}} = m_a (\omega_{e, in} - \omega_{e, out}) \]  
\[ (2) \]

where air leaving the evaporator is assumed to have a constant specific humidity of \( \omega_{e, out} = 9 \text{ g/kg} \).

The return air after passing through the enthalpy wheel (7-8) is adiabatically cooled (8-9) by part of water obtained from evaporator and rest of water is used to wet the condenser coil to complete the condensation process. Checking whether water obtained from the evaporator is sufficient for adiabatic cooling of air and refrigerant condensation process proceeds as follows:

The air after adiabatic cooling is assumed to be fully saturated with wet bulb temperature same before and after the process (Equation 3).

\[ T_{wb,\text{adiab}} = T_{wb, EW, out} \]
\[ RH_{wb,\text{adiab}} = 100\% \]  
\[ (3) \]

Using the above mentioned conditions in equation 3, dry bulb temperature, absolute humidity and enthalpy of air after the adiabatic process are calculated. The water required for adiabatic cooling is calculated using equation 4.

\[ m_{w,\text{adiab}} = m_a (\omega_{adiab} - \omega_{EW, out}) \]  
\[ (4) \]

The condenser heat transfer rate, \( Q_{\text{cond}} \) is given by equation 5.

\[ Q_{\text{cond}} = m_r (h_{r,c, in} - h_{r,c, out}) = m_a (h_{a,c, out} - h_{adiab}) \]  
\[ (5) \]

Air dry bulb temperature after condensation, \( T_{a,c, out} \) is obtained by solving equation 5.

The refrigerant condensing temperature is calculated from the air temperature at the condenser outlet using an approach temperature of 3 K as given by equation 6.

\[ T_{r,c} = T_{a,c, out} + 3 \text{ K} \]  
\[ (6) \]

The required mass flow rate of water, \( m_{w,c} \) during the condensation process using \( Q_c \) is given by equation 7 while total water mass flow rate required \( m_{\text{req}} \) is given by equation 8.

\[ Q_c = m_{w,c} h_{f,g,w} \]  
\[ (7) \]

\[ m_{\text{req}} = m_{w,\text{adiab}} + m_{w,c} \]  
\[ (8) \]

Application of the above procedure is shown in Figures 5, 6, 7, 8, and 9 where the enthalpy wheel sensible and latent effectiveness and heat wheel effectiveness are changed. Latent and sensible heat effectiveness are considered separately for enthalpy wheel for the purpose of analysis.

For first case in Figure 5, no run around heat exchanger is considered. This is done by setting heat wheel effectiveness equal to 0. The air entering the evaporator is far from the saturation point (point 3 in Figure 5). Air in the evaporator is first sensibly cooled to its saturation point and then it is latently cooled to remove its moisture and decrease its specific humidity.
In second case, the run around heat exchanger is included by increasing the heat wheel effectiveness from 0 to 0.5 while enthalpy wheel sensible and latent effectiveness is kept same as for the first case (i.e. 0.5). It can be seen from Figure 6 that air entering the evaporator is precooled in the heat wheel by the air leaving the evaporator and as a result, sensible load of the evaporator is reduced. Therefore, in this case evaporator will handle the reduced sensible load compared to first case and the latent load.

In third case, heat wheel effectiveness is increased to 0.9 while enthalpy wheel sensible and latent effectiveness are kept same as in the first two cases. In this case heat wheel removes sensible load completely and a part of latent load (see Figure 7). The evaporator load in this case is part of latent load only. Therefore, for a given enthalpy wheel
effectiveness, it can be observed that evaporator load is inversely proportional to heat wheel effectiveness. However, a mechanism needs to be established to remove the water from the heat wheel for effective heat transfer and also to get enough water to make the necessary water balance to achieve closed DOAS.

Figure 7: Third case: EW sensible effectiveness = 50%, EW latent effectiveness = 50%, HW effectiveness = 90%, outdoor air at 47 °C, 15 g/kg

In fourth case, the heat wheel effectiveness and enthalpy wheel latent effectiveness is kept same as second case while enthalpy wheel sensible effectiveness is changed from 0.5 to 0.9 (see Figure 8). It can be observed that the sensible cooling required by the evaporator to bring the air close to its dew point temperature is decreased with an increase in the enthalpy wheel sensible effectiveness i.e. heat of evaporation is inversely related to enthalpy wheel sensible effectiveness.

Figure 8: Fourth case: EW sensible effectiveness = 90%, EW latent effectiveness = 50%, HW effectiveness = 50%, outdoor air at 47 °C, 15 g/kg
In fifth case, the heat wheel effectiveness and enthalpy wheel sensible effectiveness is kept same as second case while enthalpy wheel latent effectiveness is changed from 0.5 to 0.9 (see Figure 9). It can be observed that the latent cooling required by the evaporator is decreased with an increase in the enthalpy wheel sensible effectiveness due to the removal of moisture in the enthalpy wheel. However, the water balance required for closed DOAS is not possible in this configuration due to the removal of moisture mainly in the enthalpy wheel. Therefore, for this case, condensed water from the evaporator is not adequate enough for adiabatic cooling of air and keeping the condenser coils wet all the time.

![Figure 9: Fifth case: EW sensible effectiveness = 0.5, EW latent effectiveness = 0.9, HW effectiveness = 0.5, outdoor air at 47 °C, 15 g/kg](image)

4.2. Results for all Cases

The comparison of amount of condensed water from evaporator is made with the amount of water required for adiabatic and condensation process is made using different outdoor air conditions and various combinations of heat and enthalpy wheel effectiveness. For all different cases, return air enters the enthalpy wheel at temperature of 24 °C and temperatures of 32°C and 47°C (based on Abu Dhabi, UAE summer weather conditions). Three values of design effectiveness are considered for each wheel: 0, 0.5 and 0.9, leading to 27 possible combinations of EW and HW effectiveness. Selected cases where the ratio of required water, \( \frac{m_{\text{req}}}{m_{\text{avail}}} \), to the available water, \( m_{\text{avail}} \), is less than or equal to 1.5 and few selected cases where ratio is greater than 10 are shown in Table 1.

<table>
<thead>
<tr>
<th>HW ( \varepsilon )</th>
<th>EW ( \varepsilon )</th>
<th>Outdoor air conditions</th>
<th>( \frac{m_{\text{req}}}{m_{\text{avail}}} )</th>
<th>( Q_{\text{evap}} )</th>
<th>Air condition after HW</th>
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<tr>
<td>Sensible Latent</td>
<td>T (°C) w (g/Kg)</td>
<td></td>
<td>(kW)</td>
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<td></td>
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<tr>
<td>0.0 0.0 0.0</td>
<td>32 30</td>
<td>1.5</td>
<td>22.3</td>
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<td>0.0 0.9 0.0</td>
<td>32 30</td>
<td>1.5</td>
<td>21.3</td>
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<td>0.5 0.0 0.0</td>
<td>32 30</td>
<td>1.3</td>
<td>20.4</td>
<td>Saturated</td>
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<tr>
<td>0.5 0.0 0.0</td>
<td>47 30</td>
<td>1.5</td>
<td>22.9</td>
<td>Saturated</td>
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<td>0.5 0.5 0.0</td>
<td>32 30</td>
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<td>19.7</td>
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<td>1.5</td>
<td>20.9</td>
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<tr>
<td>0.5 0.9 0.0</td>
<td>32 30</td>
<td>1.4</td>
<td>19.1</td>
<td>Saturated</td>
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</tbody>
</table>
5. CONCLUSION

- The amount of water condensed in the evaporator is not affected by changing heat wheel effectiveness while keeping enthalpy wheel latent and sensible effectiveness constant. This is because specific humidity doesn’t change in heat wheel and evaporator leaving conditions are fixed (24 °C and 9 g/kg). Heat wheel effectiveness reduces the sensible cooling required from evaporator, $Q_{\text{fH}}$. By choosing a reasonable value of heat wheel design effectiveness i.e. around 0.5, we can ensure that air leaving the heat wheel is almost saturated, leading to the evaporator operation primarily in fully wet fin condition (e.g. large latent load).

- As a corollary to the previous point, note that of all cases with insufficient water supply, the shortages are generally least when the evaporator inlet conditions are near or at saturation.

- The amount of water condensed in the evaporator and amount of water required in condenser is affected by changing the enthalpy wheel latent effectiveness while keeping enthalpy wheel sensible and heat wheel effectiveness constant. For higher values of enthalpy wheel latent effectiveness (>0.5), the air entering the evaporator has lower specific humidity because of moisture removal in enthalpy wheel. The difference in the amount of water required by condenser and water available from evaporator increases with an increase in the effectiveness of enthalpy wheel latent effectiveness.

- For cases, where air specific humidity is high i.e. 30 g/kg, the amount of water produced by evaporator is close to the amount of water required for condensation except where enthalpy wheel latent effectiveness is high (>0.5). However, for all the cases, at least some amount of additional water is required to complete the condensation process.

It is observed that as enthalpy wheel and heat wheel effectiveness increased, the evaporator load is greatly reduced. Thus, although the water balance left a shortage, the amount of additional water needed to evaporatively cool the condenser becomes quite small.

Therefore, in future, the evaluation of performance of a DOAS with evaporatively-cooled condenser in the return air stream is required.
1. NOMENCLATURE

<table>
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<td>$\dot{m}$</td>
<td>mass flow rate, kg/s</td>
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<td>$h_{fb}$</td>
<td>latent heat of vaporization, kJ/kg</td>
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<tr>
<td>$T$</td>
<td>Temperature, K</td>
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<tr>
<td>$w$</td>
<td>Specific humidity, g/Kg</td>
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<tr>
<td>$h$</td>
<td>Enthalpy, kJ/kg</td>
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<td>Effectiveness</td>
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<td>Enthalpy Wheel</td>
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Subscripts

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<td>wb</td>
<td>wet bulb</td>
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REFERENCES


