PARAMETERS AFFECTING THE PERFORMANCE OF A DEWPOINT COOLER CONSISTING OF A COUNTER FLOW HEAT EXCHANGER USING WATER AS REFRIGERANT

M. JANSSEN(a), P. UGES(b)
[(a)] Re/genT BV, Lagedijk 22,
5705BZ Helmond, The Netherlands
martien.janssen@re-gent.nl
[(b)] StatiqCooling BV,
Rijssen, The Netherlands,
peteruges@statiqcooling.nl

ABSTRACT

A dewpoint cooler is defined as an indirect evaporative cooler with a counter flow arrangement of the primary air to be cooled and the process air to be humidified and ventilated to the environment. Due to the counter flow arrangement the air exit temperature is not limited theoretically to the wet bulb temperature. The construction consists of polypropylene channel plates which guide the primary flow through its channels. The plates are covered with a hygroscopic foil and the return process flow is forced to flow between the plates, and is therefore in direct contact with the foils. Water is supplied to the foils via a distribution system.

A number of such constructions have been build and experimentally investigated. The paper presents the experimental set-up with some results. A model has been developed based on simultaneous heat and mass transfer at the water film developed on the hygroscopic foil. The model is used for the evaluation of the experimental results and forms a theoretical base for maximum achievable performance.

The model is subsequently used for optimising the heat exchanger geometry with respect to plate sizing, distances, etc. The prime parameters are here the exit temperature desired, the cooling capacity of the unit and the friction losses (pressure losses) within the unit. It is shown that the energy efficiency of the unit strongly depends on the design and that there is a trade off between minimum temperature achievable and energy efficiency. Finally some comparison with mechanical air conditioning systems is made with respect to the dependence of the capacity and efficiency of the ambient temperature and humidity. It is shown that even in moderate climates a dewpoint cooler can be effectively operated.

1. INTRODUCTION

Indirect evaporative coolers are in general used for air conditioning purposes. In contrast to direct evaporative coolers (adiabatic coolers) such units are also applicable in moderate humidity environments since the air entering the space to be cooled is not humidified but cooled by a secondary air stream which is adiabatically cooled. The unit under investigation is a special version of an indirect evaporative cooler as the secondary air stream (process air) is split off immediately at the exit of the heat exchanger and let back in counter flow arrangement with the primary air flow as shown in Figure 1. Due to this arrangement, the exit temperature is not limited to the wet bulb temperature at in stream conditions but theoretically at the dewpoint temperature, hence the unit is referred to as a dewpoint cooler. The prime advantage of this design is that contaminations present in the primary air stream will not enter into the process channel due to the centrifugal forces.

Approximately one third of the air is recirculated as process air. The system contains typically only one fan placed upstream of the unit, though a second fan downstream in the process air stream is also feasible. The primary air duct is in fact made up by a series of channel plates of polypropylene as shown in Figure 1. A hygroscopic foil is attached to each external surface of the channel plates by a spot-welding process. The plates are assembled together using spacers in such a way that the spaces between the plates form the process air duct. Water is fed at the top of the heat exchanger block and moves down by gravity and hygroscopic effects. As the process air passes along the wetted surface of the plate, the water evaporates and heat is extracted from the plate and subsequently from the primary air stream.

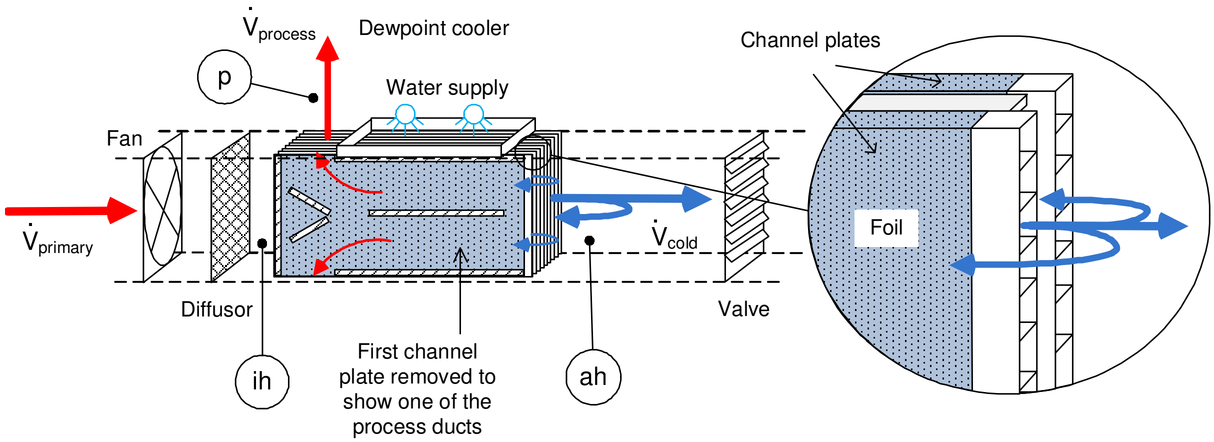

Figure 1 : Principle layout of the dewpoint cooler

Typical dimensions of the heat exchanger is 0.75 to 1.5 m in height, app. 1.3 m in length and a number of 60 to 100 plates stacked to each other. In this paper a demonstration, smaller version of such heat exchanger is experimentally and theoretically investigated. This demonstration unit has a plate length of 1.38 m, a height of 0.5 m and 74 plates giving a width of 0.53 m.

2. EXPERIMENTAL SET UP

A series of experiments has been performed on the small demo dewpoint cooler. Next to a set of geometrical parameters, there are four operating parameters influencing the performance of the cooler, namely inlet air humidity and temperature and primary and process air flow. For convenience, the process air flow is expressed as a fraction of the primary air flow:

$$r = \frac{\dot{V}_{process}}{\dot{V}_{primary}} \tag{1}$$

Typical test conditions for the inlet were 27 °C / 47.5 % RH, 32 °C / 30 % RH and 35 °C / 25 % RH. The primary flow rate was varied from 1000 to 2200 m3/h approximately and the process air flow fraction from 0.2 to 0.4.

As performance data one can distinguish between the exit temperature achieved and the cooling capacity which is defined as:

$$\dot{Q}_c = \dot{m}_{cold} \cdot (h_{ih} - h_{ah}) \tag{2}$$

Where the indices refer to the positions shown in Figure 1. Rather than using the exit temperature as an output parameter, it is also possible to express it in non-dimensional form by defining a dewpoint effectiveness as follows:

$$\varepsilon_{dp} = \frac{(t_{ih} - t_{ah})}{(t_{ih} - t_{d,ih})} \cdot 100 \tag{3}$$

This is in analogy with the direct saturation effectiveness (or wet bulb effectiveness) defined for adiabatic coolers (ratio of actual temperature drop with the maximum possible) where maximally the wet-bulb temperature at primary air inlet conditions can be achieved (ASHRAE 2004).

The experiments have been performed under laboratory conditions by controlling primary air inlet temperature and humidity (at point *ih* in Figure 1) using a heating and humidification process prior to the fan. Temperatures are registered just upstream (point *ih*) and downstream (point *ah*) of the dewpoint cooler block and in the process air exit channel (point *p*) with thermocouples and proper data acquisition equipment. Humidity has been determined by humidity sensors at point *ih* and *p*. Flow measurements have been performed by anemometry at the cold air and process air outlet streams. Using density relations for moist air the volume flows were converted to mass flows. Further pressure drop measurements have been performed using piezoelectric pressure sensors. Enthalpies are determined using appropriate moist air relations and using temperature and relative humidity as input data.

In all experiments an energy balance has been made using the air mass flows and the enthalpy differences over the primary and process air flow. The residual error in the energy balance proved to be typically below 20 %. This fairly high residual error results from the fact that the energy measurement on the process air side suffers a high uncertainty as it is very sensitive to the final humidity reached at the exit, which often is very close to 100 % RH. In such case a typical uncertainty of the measured humidity of 2 % RH results already in a large uncertainty in the energy flow. Note that the measured cooling capacity of the unit and the temperature efficiency are not affected by this uncertainty so these results are generally significantly more accurate.

Rather than analysing the dewpoint cooler on the basis of experimental results, a theoretical model was developed which was fitted to the, limited, set of experiments. Next the model is used to predict performance over a wider range of operating conditions and geometrical input parameters.

3. THEORETICAL MODEL

A detailed theoretical model has been constructed to simulate the behaviour of the dewpoint cooler. The model contains mass and energy balances of both the primary and process air stream. The water absorbed in the hygroscopic foil is treated as a water film. At the interface between water and air the energy balance can be given as:

$$\alpha_p (t_p - t_w) + h_d H_{fg} (x_p - x_w) + \frac{t_i - t_w}{R_{th}} = 0 \tag{4}$$

Where the diffusion coefficient and the heat transfer coefficient at the water surface can be interrelated using the classical Lewis relation and assumed to be unity:

$$\frac{\alpha_p}{h_d C_p} = 1 \tag{5}$$

For the heat transfer on both the primary and process side classical relations for the Nusselt numbers have been employed for heat transfer between parallel plates for ducts of various cross-sections (Mills, 1992). Note that in all tested circumstances, the flow remains laminar through the plates.

Pressure losses are calculated using classical relations for flows in ducts using the appropriate relations for the hydraulic diameter. Friction factors for laminar flow have been applied. The energy associated with all friction losses can hereafter be calculated as follows:

$$P_{friction} = \Delta p_{primary} \dot{V}_{primary} + \Delta p_{process} \dot{V}_{process} + \Delta p_{distribution} \dot{V}_{cold} \tag{6}$$

where the last part is in fact the pressure losses over the distribution system of the refrigerated air or over a regulating valve at the exhaust of the cooler as shown in Figure 1. In general this pressure forces the air to flow through the process ducts and is therefore by definition equal to the pressure drop over the process channels. The friction losses inside the heat exchanger are also taken into account in the earlier mentioned energy balances.

When the net cooling capacity of the unit is divided by the power input needed for the fan, a coefficient of performance can be defined as follows:

$$COP = \frac{Q_{net}}{P_{fan}} = \frac{\dot{Q}_c - \Delta p_{distribution} \dot{V}_{cold}}{P_{friction} / \eta_{fan}} \tag{7}$$

Within the model it is assumed that air flows are all uniformly distributed between the channel plates and the process channels. It is further assumed that the water film is in perfect contact with the channel plates and no water transfer through the plate takes place. In fact with these assumptions it is sufficient to evaluate only a single plate with its process channel.

A finite difference scheme was set-up including all mass and energy equations plus a set of equations for moist air properties. The calculation is performed in downward direction of the primary air stream. Due to the counter flow arrangement with the process air this means that estimates for the process air exit conditions are required. Hereafter an iterative procedure is required to resolve these exit conditions. The scheme is programmed in MS Excel using appropriate VBA code and the circular calculation option for the necessary iteration process. The cooler is typically divided in 60 slices over the length of the heat exchanger.

4. COMPARISON EXPERIMENTS AND MODEL

The experiments performed on the demo cooler were also simulated. From comparison it was learned that the model generally overestimated the performance. Hence a corrective factor was added to the model which assumed that only a part of the cooler was effectively used for the water evaporation process (effective area parameter). Using such a factor of 0.6 (meaning that 60 % of the heat transfer surface area is effectively used) experiments generally correlate with the model within 5 %, this both for the cooling capacity and the dewpoint efficiency. It needs to be mentioned that the base case cooler is already relatively long in size, meaning that a substantial increase in heat exchanger surface area only leads to a small increase in cooling capacity and dewpoint efficiency. The difference between a surface area effectiveness of 0.6 and 1.0 is generally less then 10 % in cooling capacity for most tested conditions.

5. COOLER CHARACTERISTICS

As the model predicted the experiments reasonably accurately, it is now used to discuss the basic behaviour of the dewpoint cooler using again the same demo cooler block as the base case. It is of interest to study the influence of the primary and process air flow on the performance parameters, as well as the influence of inlet temperature and humidity. First a principle simulation is run at nominal conditions, from which the results as well as the cooler block main dimensions are presented in Figure 2. The calculation is shown at the reference condition for AC unit testing (see EN 14511), namely 27 C / 47.5 % RH. The nominal air flow rate for this unit is 1900 m3/h. In this simulation as well as in all simulations following an area effectiveness of 60 % has been applied. The diagram drawn in the middle of the figure represents the primary and process dry bulb temperature and the surface temperature of the water film in the process channels, all as a function of the position within the channel.

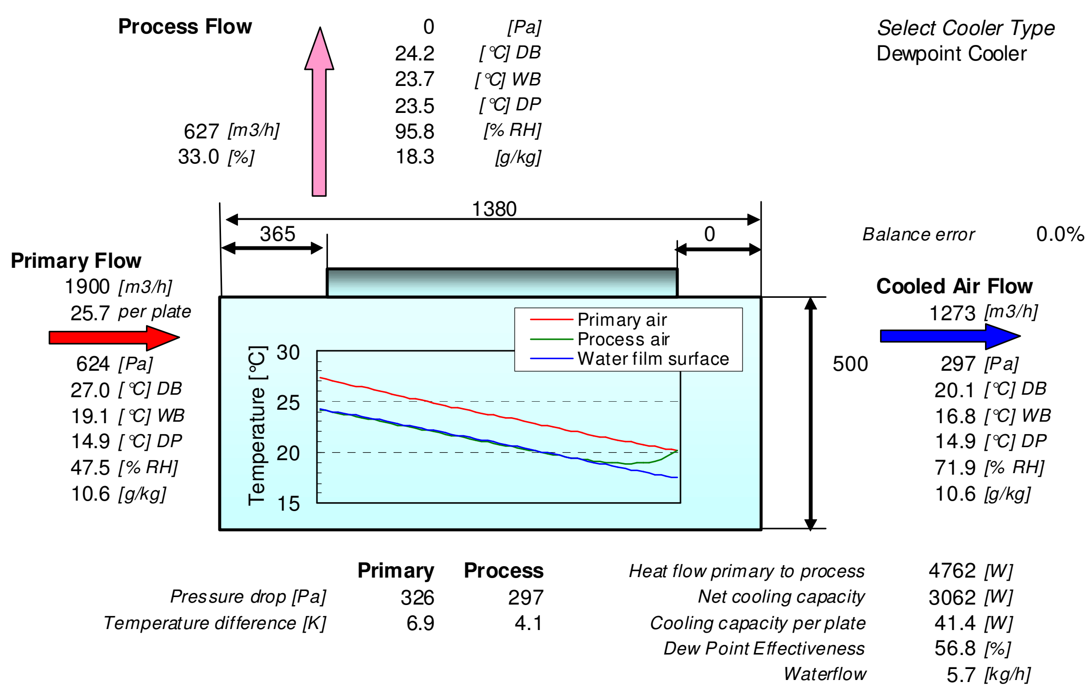

Figure 2 : Base case input data and performance of the cooler block

The influence of the primary and process air flow is shown in the diagrams in Figure 3 where it can be seen that the exit temperature drops with increasing process flow ratio. This initially also results in an increase in cooling capacity (from 20 to 30 % process flow fraction). However, an increase in process air flow at constant primary air flow means that the net air flow usable for the space to be conditioned reduces, which effect starts to be become dominant after app. 40 %. It can be seen that the cooling capacity shows a maximum around 25 to 35 % process air flow. The effect of an increased primary air flow is that the cooling capacity indeed increases but the exit temperature as well.

As the friction losses increase with the square of the air flow rate the COP of the unit is very sensitive to the flow conditions. At a process flow fraction of 33 % the COP can be as high as 14 for 1000 m3/h down to a value of 3 at 3000 m3/h. In the calculation of this COP it has been assumed that the fan efficiency is 60 %, which is feasible for e.g. brushless DC motor fans. Comparing these values with a well designed mechanical air conditioning unit which achieves typical a value of 4 at these input temperatures/humidity's, it is clear that the unit should not be overloaded with respect to the flow rate. It is also clear that a higher process air fraction than app 35 % is not useful. It also shows that capacity control by means of fan speed control leads to very efficient operation at part load conditions.

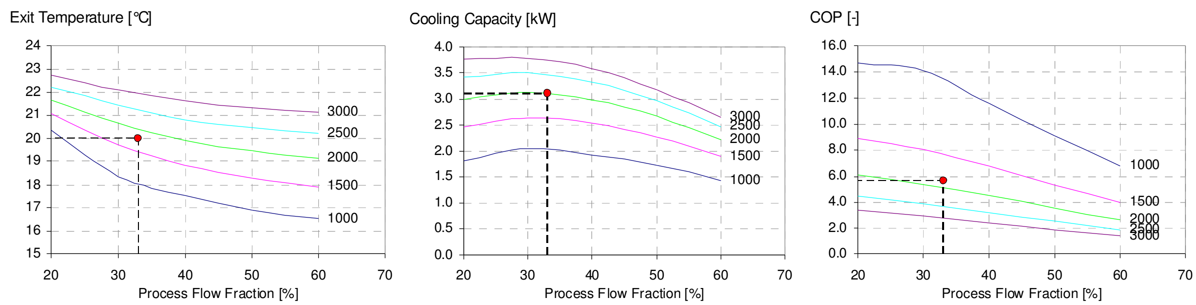

Figure 3 : Influence primary air flow (given next to the curves in m3/h) and process air flow ratio, the red dot represents the nominal condition.

The next parameters to evaluate are the effect of the input temperature and humidity on the cooler performance. Again using the same base line cooler, the following diagrams are constructed.

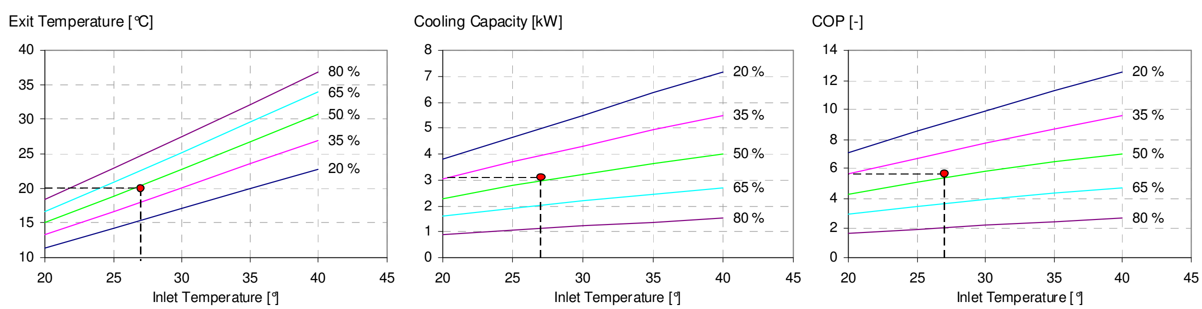

Figure 4 : Influence of relative humidity (given next to the curves in % RH) and temperature at the primary air inlet, the red dot represents the nominal condition.

Here it is shown that an increase in inlet temperature (at constant % RH) results in an increase in exit temperature. This is due to the heat transfer limitation of the cooler itself. Interesting to note is that the cooling capacity increases at higher ambient temperatures, hereby following the thermal load of the space being cooled. This is in clear contrast with mechanical refrigeration. Since the friction losses are virtually independent on the inlet conditions, the COP increases proportionally with the cooling capacity.

The largest drawback of, any, evaporative cooler is also shown in these diagrams. In case the humidity level increases the cooling capacity drops proportionally finally reducing to zero at saturated inlet conditions. For this specific reason, the evaporative cooler must be seen as a "top-cooler" device with no guaranteed performance all year around.

For moderate inlet temperatures the cooling capacity also drops compared to the nominal condition, meaning that at temperatures of e.g. 22 °C ambient, the efficiency of the cooler becomes critically low, unless the air flow is reduced which is typically feasible as the thermal load of the space being cooled also reduces.

6. COOLER IMPROVEMENT

The demo cooler studied as a baseline case was mainly designed to achieve an as low as possible exit temperature. If the cooler is evaluated at the same nominal conditions as shown in Figure 2, but with varying plate length, a diagram can be constructed which is shown in Figure 5. This shows that in order to achieve a low exit temperature, a disproportional amount of heat transfer surface must be added. An interesting point to see in this diagram is that it is practically impossible to reach the dewpoint temperature (as it is for adiabatic coolers to reach the wet-bulb temperature). At a length of 3 m the exit temperature is 18 °C which is 1 K below the wet bulb temperature but still 3 K above the dewpoint temperature. This effect is caused by a much reduced heat transfer due to the increasingly reducing temperature differences between the primary air and the water film surface.

Figure 5 suggests that the optimal cooler length for efficiency is around 1200 mm which is close to the base line cooler length. However, this is only true if all the other parameters such as height and width of the cooler block remain the same.

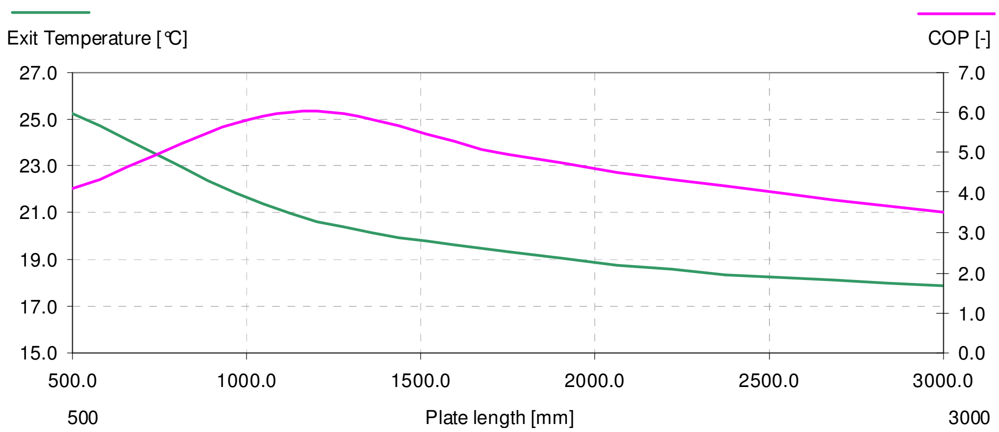

Figure 5 : Effect of plate length on cooler performance

Using the model two optimisation runs were made (using MS Excel Solver) both aiming at increasing the COP but with different constraints (both runs at 27 °C / 47.5 % RH inlet condition):

1. Highest efficiency using the same amount of plastic material and with the same cooling capacity as the base case (3 kW). The following parameters were left free: plate length, plate height, number of plates, plate distance, plate thickness (=width of primary air flow channel), primary air flow and process air flow fraction. This allows the cooler block to become quite different in height and width.
2. Highest possible cooling capacity with a COP of 11.2 (double of the base case), leaving the block height and width the same, so that the cooler block can still be placed within the same air handling duct. The following parameters were left free: plate length, number of plates, plate distance, plate thickness (=width of primary air flow channel), primary air flow and process air flow fraction.

The results of these optimisations are shown in Table 1, which shows a COP increase with a factor 3 for the first run and a COP increase with a factor 2 (by design) for the second run, both compared to the base line.

		Base Line	Optimised-1	Optimised-2
Plate length	*[m]*	1.38	0.64	1.21
Plate height	*[m]*	0.50	0.87	0.50
Number of plates	*[-]*	74	107	83
Total thickness of the channel plate	*[mm]*	4.0	2.7	3.2
Total width heat exchanger	*[m]*	0.53	0.64	0.53
Plate distance (total)	*[mm]*	3.0	3.1	3.0
Heat transfer surface area	*[m2]*	75.1	51.1	69.9
Compactness	*[m2/m3]*	206	143	219
Plastic volume	*[m3]*	0.035	0.035	0.031
Primary air flow	*[m3/h]*	1900	2021	1216
Velocity primary air inside the channel plate ducts	*[m/s]*	4.3	2.7	3.1
Pressure drop primary air flow	*[Pa]*	326	143	259
Heat transfer coefficient primary air side	*[W/m2K]*	31	47	39
Process air flow ratio	*[%]*	33.0	24.5	26.0
Velocity process air between the channel plates	*[m/s]*	1.7	0.5	0.7
Pressure drop process air	*[Pa]*	297	37	112
Heat transfer coefficient process air-water film	*[W/m2K]*	39	36	38
Dry bulb temperature primary air outlet	*[°C]*	20.1	21.2	19.3
Cooling capacity	*[W]*	3062	3009	2340
Dew Point Effectiveness	*[%]*	56.8	48.1	63.2
Water flow	*[kg/h]*	5.7	4.9	3.7
Friction losses total	*[W]*	329	101	125
Fan efficiency	*[-]*	0.6	0.6	0.6
COP	*[-]*	**5.6**	**17.9**	**11.2**

Table 1 : Comparison of base line and optimised unit

A number of remarks can be made here to explain the large performance increase in the first optimisation run:

- The length of the cooler has reduced by more than a factor 2. The surplus of material is used to increase the height and width.
- Due to these measures the air velocities drop significantly.
- The lower air velocities result in much lower pressure drops, resulting in a reduction of the total friction losses by a factor 3.
- The lower air velocities do not affect the heat transfer coefficients negatively since the flow is laminar.
- The heat transfer coefficient in the primary air stream has increased significantly due to the reduced hydraulic diameter by the reduced plate thickness.
- The primary air flow has remained very similar; the process air fraction was optimal at 24.5 %.

The temperature performance has slightly decreased as the exit temperature has gone up from 20.1 °C to 21.2 °C. It is clear that if there is space available in width and height direction, an optimal cooler will be quite short. However, this may not be practical and may increase also costs as the costs of an air handling unit scales with the duct sizes.

The second optimisation takes this effect into account and maintains the same height and width of the block. This leaves only the channel plate dimensions and the spacing to be optimised. The higher COP can only be achieved sacrificing some heat exchanger length and reducing the primary air flow. This resulted in a 24 % lower cooling capacity of the unit, but with a double efficiency compared to the base line.

It needs to be mentioned that generally a pressure drop over the distribution system of the refrigerated air is present. In the first optimised case the pressure drop over the process channel is so small that likely the pressure drop over the distribution system is larger, which means that more air then desired will be pushed through the process channel. To avoid this a regulating valve can be placed in the exhaust of the process channel. Obviously pressure losses in this valve need to be added to the above calculation in such case, hereby reducing the COP calculated.

7. CONCLUSION

Experiments and modelling has been performed on a small version of a dewpoint cooler and the influence of the operating conditions has been discussed. The results can be translated easily to full scale versions of cooler blocks if the same internal velocities in both primary and process air flow are maintained. Air friction losses increase with the square of the air stream velocities, while the cooling capacity increases less then linearly. This results in a fast degradation of the coefficient of performance of the unit with increased air flow rates.

Using modelling, it has been proven feasible to improve the energy efficiency of a unit considerably. If there are no specific space constraints, shortening the cooler will lead to a very high COP at the same material use and cooling capacity. If the duct size of the air handling unit is the limiting case, it is still possible to increase the efficiency of the unit, however, not without reducing its maximum cooling capacity. As the optimised unit designs have not yet been experimentally validated, additional experiments will be required to support these conclusions.

NOMENCLATURE

C_p	Air specific heat [J/kgK]
H_{fg}	Water latent heat [J/kg]
h	Enthalpy [J/kg]
h_d	Mass diffusion coefficient [kg/(m^{2}s)]
$\dot{m}$	Mass flow [kg/s]
$P_{friction}$	Friction losses [W]
$\dot{Q}_c$	Cooling capacity [W]
r	Process flow ratio [-]
R_{th}	Thermal resistance [Km^2/W]
t	Temperature [°C]
t_d	Dew point temperature [°C]
x	Absolute humidity [kg/kg dry air]
$\dot{V}$	Volumetric flow rate [m^3/s]
α	Heat transfer coefficient [W/m^{2}K]
COP	Coefficient of performance [-]
ε_{dp}	Temperature efficiency [-]
Δp	Pressure drop [Pa]
η_{fan}	Fan efficiency [-]

indices

i	Primary air
p	Process air
w	Water surface

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2. A.F. Mills 1992, *Heat Transfer*
3. EN14511 *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 2: Test conditions,* Table 4