A novel hybrid dew point cooling system for mobile applications

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Abstract

A novel dew point cooling system for truck cabin cooling has been designed and tested, which shows excellent performance in environments of low to medium relative humidity. Experimental results showed that cooling capacity of over 1kW could be achieved with up to 12K of sensible cooling of the supply air and a COP of up to 7. The DPC unit can provide comfort cooling to a vehicle using a rooftop unit whose DPC core is no larger than 0.5m x 0.5m x 0.25m.

Keywords: indirect evaporative cooler, dew point cooler, mobile air conditioning, natural refrigerants

Introduction

In 2010, the global transport sector consumed over 2,000 million tons oil equivalent, representing about 20% of global energy supplies WEC[1], whilst more than 60% of oil consumed was for the transport sector. Of this, around 75% was consumed by road transport. Transport energy consumption is largely based on oil, so contributes to pollution and carbon emissions. Mobile air conditioning (MAC) is now a standard facility in road vehicles such as automobiles, trucks and buses, but can consume over a quarter of all the power available from the engine. Most MAC systems utilise vapour compression cycles whose working fluids are high global warming potential refrigerants, such as HFC134a (GWP~3500 (Forster *et al* [2])). MAC systems contribute to global warming both directly due to leakage of refrigerant into the atmosphere and indirectly due to the carbon dioxide released in combusting fossil fuels (Grof, [3]). The significant leakage associated with MAC systems has lead to many authorities prohibiting the use of HFC refrigerants such as HFC134a, and so alternatives are being developed. However this does not address the power consumption and associated global warming effects of operating a MAC system. If natural or low carbon cooling systems could be integrated into MAC systems, then carbon emissions and engine power consumption could be reduced. In many climates around the world where cooling is required, evaporative cooling is an effective way of providing comfort cooling if the relative humidity is medium to low and water is available.

Surveys were carried out to investigate whether evaporative cooling would be suitable for MAC systems, and it was found that the most appropriate application would be truck cabin parking coolers. Many of these products are mounted on roofs and provide cooling to occupants when the truck is not being driven. These are mainly based on MAC vapour compression technology, but some use direct evaporative cooling. There are two main disadvantages to these systems; MAC systems consume battery power that is ultimately derived from the truck engine, whilst direct evaporative coolers increase the moisture in the air that is being conditioned and therefore can reduce comfort. An alternative is indirect evaporative cooling based on the

Maisotsenko cycle (M-cycle) (Muhammad *et al* [4]), which we also term the dew point cooler, because it can potentially cool to the dew point temperature.

Dew point cooler

Figure 1a shows a simple schematic diagram to illustrate the cooling process and Figure 1b shows the process on a psychrometric chart. The dew point cooler consists of a number of paired channels separated by an impermiable wall. One channel is termed the dry channel (product channel in Fig 1a) and the other is termed the wet channel (working channel in Fig 1a). A number of openings are provided in the wall to allow the flow of air from one side to the other. The wall is usually provided with a wick type structure to draw water into the channel and provide a wetted surface for the wet channel.

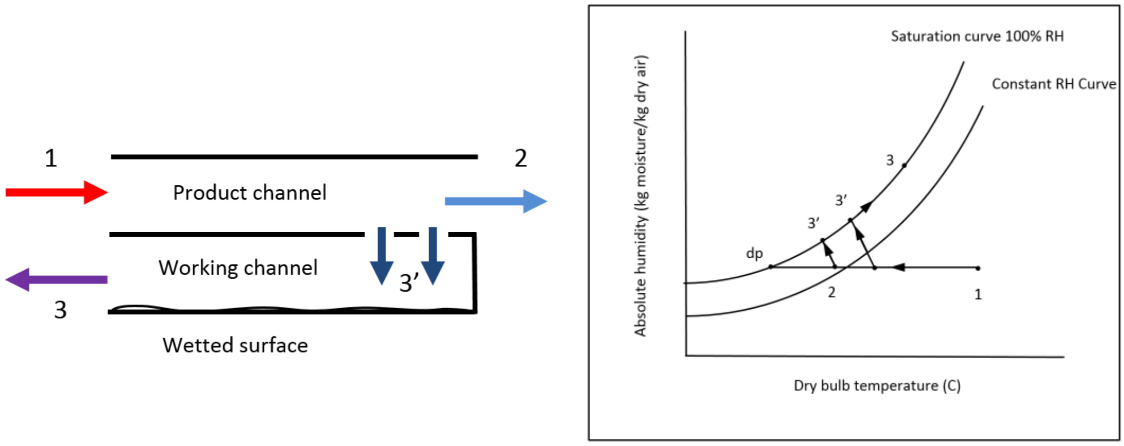


Figure 1. (a) Diagram of dew point cooler (b) indirect evaporative cooling process illustrated on psychrometric chart

In figure 1b, the x-axis represents the dry bub temperature and the y-axis represents humidity ratio (ratio of the mass of water vapour to the mass of dry air). Air is introduced into the exchanger at (1) at a particular temperature and humidity. As the air flows through the dry channel, some of the flow is directed to the wet channel (3') through the openings. The air in the wet channel flows counter to the flow in the dry channel. The difference in vapour pressure between the water vapour in air and water at the wetted surface drives evaporation from the water to the air. The latent heat transferred cools the dry channel and therefore provides sensible cooling of the product air. The process continues along the exchanger until the product air leaves the dry channel at (2), and the working air leaves the wet channel at (3). In theory, the dry air could be cooled to its dew point temperature from 1 to dp.

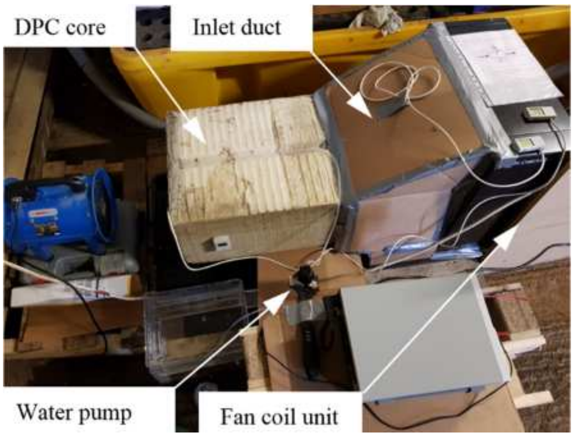


Figure 2. Image of DPC core preliminary test rig@ EPS Ltd.

This paper reports on a novel vehicle cooling system that uses natural refrigerants to provide cooling for a truck cabin. A compact and lightweight dew point cooling system is being developed that will either reduce the need for conventional vapour compression systems or replace them altogether.

The smallest standard core available to us was approximately 0.5m x 0.5m x 0.25m (L x W x H), so we obtained a core and designed a system around it that would minimise the weight and envelope whilst maximising its cooling performance.

Our consortium partners EPS Ltd carried out some preliminary testing of a single dew point cooler core to assess its potential for cooling prior to integration into the dew point cooling air conditioning system. Figure 2 shows an image of the test rig. The rig consists of a heat pump and fan coil unit that controls inlet temperature and air flow with 5 speed variability, a sealed ducting system that directs the inlet flow to the DPC core, a 12VDC peristaltic pump that can supply water flows of up to 5kg/min, temperature and humidity sensors for the inlet, supply and outlet streams, rotary vane type anemometer to measure velocities, weighing machine to determine mass flow of water pumped, and fan coil power measurement devices.

Table 1. Air flows and power consumption for DPC core testing

Speed	Inlet (m^3/hr)	Product (m^3/hr)	Working (m^3/hr)	*AR*	Fan Power (W)
1	721	152	570	3.75	40
2	941	231	711	3.08	57
3	1390	371	1019	2.75	83

Table 1 shows average flows and fan power consumption for three fan speed settings. It can be seen that AR decreases slightly from 3.75 at speed 1 to 2.75 at speed 3. Figure 3 shows the variation in a) cooling capacity and b) dew point effectiveness with inlet temperature. Inlet relative humidity was approximately constant at 20% during the tests. Cooling capacity inceases from approximately 300W at 20°C to 1200W at 40°C. Dew point effectiveness increased from about 70% at 20°C to approaching 100% around 35-40°C.

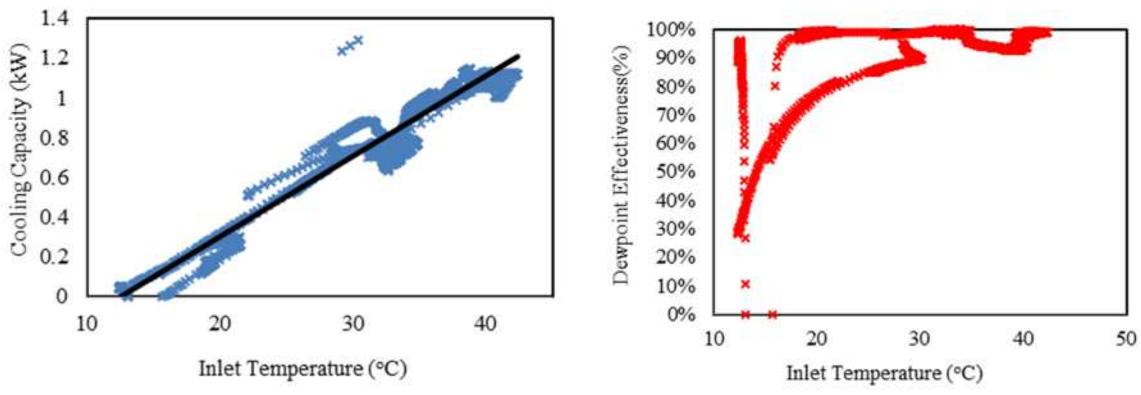


Figure 3. Variation in a) cooling capacity and b) dew point effectiveness with inlet temperature

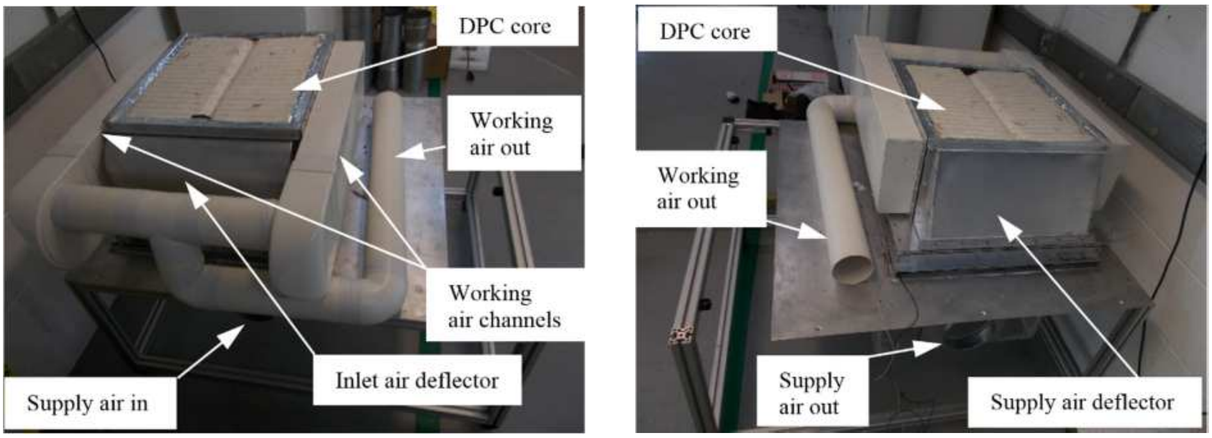


Figure 4 (a) front view of dew point cooler system, (b) rear view of dew point cooler system.

The working air is collected on both sides and leaves in a single duct. Figure 3b) shows that we almost reached saturation conditions in the supply air stream. The working air is collected on both sides and leaves in a single duct. meaning that almost all of the potential for evaporation was realised. This was partly due to the low RH, thus providing a large driving mechanism for evaporation in the wet channels, and low inlet temperatures prior to processing by the fan coil (tests were carried out during winter UK conditions in an outbuilding). The results gave us confidence that the process could work successfully and effectively, so in the next section we report results from controlled experiments on an integrated dew point cooler air conditioner.

Dew point cooler rooftop cabin cooler

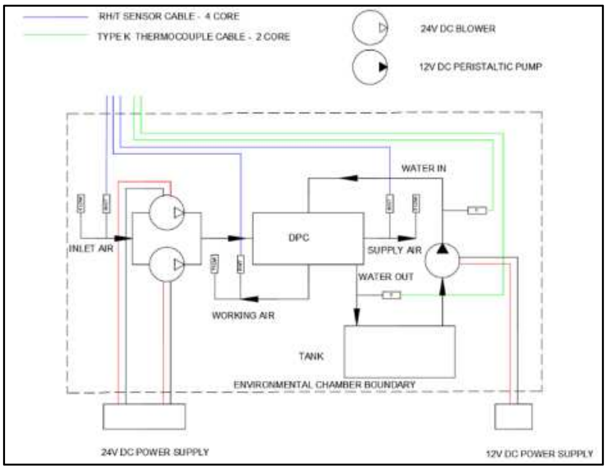


Figure 5. Dew point cooler schematic diagram

In order to draw air from the cabin (air inlet), direct it through the core and back to the cabin, whilst discharging the working air, we needed to design a compact air distribution system. In comparable rooftop air conditioning systems, the air being drawn from the cabin and delivered back to the cabin is usually located within the roof hatch. In a conventional rooftop A/C unit, the air is drawn from the cabin by a blower and directed a short distance across the evaporator of the A/C unit. In most cases these consist of narrow double row heat exchangers, and so the width (in the direction of air flow) can be as low as 50mm. Unfortunately, the width of the dew point core is around 500mm, and so we needed to design a distribution system that could draw in the air, direct it to the inlet to the dew point core, channel the supply air back to the cabin whilst distributing the working air away. We decided to site the blowers directly above the air inlet, direct the air in channels formed from aluminium channel section underneath the core, deflect the air 180° using an inlet air deflector, so that it flowed in line with the product channels, directed the supply air back to the cabin through a supply air deflector, whilst also directing working air using two 90mm x 200mm working air channels and various transition pieces, elbows, etc. Figures 4a and 4b show front and rear views of the dew point cooler with labels showing various features.

A simple schematic diagram of the system is shown in figure 5. Two 24V DC compact blowers 220 x 220 x 56 were provided to give the required volume flow. Vaisala HMP110 humidity/temperature sensors were used at the inlet, supply and working air outlets. K Type thermocouples were used to measure temperatures at various locations. Air flows were determined by measuring air velocity at various locations using a Testo 416 vane type anemometer. Descriptions of the measuring instruments used, their range and accuracy can be found in Table 2. DC power was supplied to the 24V DC blowers by a TTI CPX200 duel channel power supply unit and DC power was supplied to the 12V DC perastaltic pump by a Extech 37220 Quad power supply unit. A 10 litre water tank was provided to supply the system with water. Water flow was determined by placing the water tank on a digital weighing platform (CPS 66220) and recording the difference in mass over a given time period. Testing was carried out in a climate chamber which allowed us to set and control temperature and relative humidity.

Table 2. Instumentation

Sensor description	Parameter	Range	Accuracy
K Type Thermocouple	Temperature	0-1100°C	±1.5°C
Vaisala HMP110 humidity/temperature probe	Relative humidity	0-90%RH	±1.5%RH
	Temperature	0-40°C	±0.2°C
	Temperature	40-80°C	±0.4°C
Testo 416 Vane Anemometer	Air speed	0-40m/s	±0.2m/s (0-60°C)
CPS 66220 weighing platform	mass	0-100kg	±5g

Test procedure

Tests were carried out over a range of temperatures and relative humidities from 25°C to 45°C and from 40% to 80%RH. During individual tests, monitoring and logging was carried out for at least 60 minutes at steady state for each test point. At various intervals, but for only short periods of time to minimise changes to the conditions, the chamber was accessed to record the mass of the water tank and air speeds, so that the mass flow of water and the air volume flow rates could be determined.

The experimental results were used to determine the change in temperature between inlet and supply ΔT, cooling capacity Q_{supply}, dew point effectiveness ε_{dp}, and coefficient of performance *COP*.

$$\Delta T = T_{in} - T_{supply} \quad [1]$$

$$Q_{supply} = V_{supply} . cp_{supply} . \rho_{supply} . \Delta T \quad [2]$$

$$\varepsilon_{dp} = \frac{T_{in} - T_{supply}}{T_{in} - T_{dp}} \quad [3]$$

$$COP = \frac{Q_{supply}}{W} \quad [4]$$

$$W = W_{fans} + W_{pump} \quad [5]$$

Where T_{in} (°C) is inlet temperature, T_{supply} (°C) is supply air temperature, V_{supply} (m^3/s) is supply volume flow, cp_{supply} (J/kg/K) is specific heat capacity, ρ_{supply} (kg/m^3) is density of supply air, T_{dp} (°C) is the dew point temperature, W (W) is the total work input to the system, W_{fans} (W) is the fan work input and W_{pump} (W) is the pump work input.

Results

Figures 6a, 6b and 6c show the variation in temperature difference from inlet to supply with inlet temperature and relative humidity. They represent results at approximately 40%, 60% and 80% RH, respectively. The data were reduced by filtering temperature and RH readings within ±1°C and ±1%RH of the values of interest. The results were then statistically analysed by obtaining the sample mean and sample standard deviation for values 25°C, 30°C, 35°C, 40°C and 45°C, and RH values of 40%, 60% and 80%.

The sample mean $\bar{X}$ and the sample standard s deviation of a given variable X for a sample size n are determined in the usual way (Evans and Rosenthal [4]).

Table 3 shows the results of reducing the data and applying the statistical analysis. It should be noted that there were not enough data points at 25°C/80%RH and 45°C/40%RH, so we propose to interpolate from the data obtained.

Figure 6d shows the variation of temperature difference with inlet temperature and for RH at 40%, 60% and 80%, figure 7a shows the variation of cooling capacity with inlet temperature and for RH at 40%, 60% and 80% and figure 7b shows the variation of COP with inlet temperature and for RH at 40%, 60% and 80%.

Table 3. Reduced data

T_{in} (°C)	RH_{in} (%)	$\overline{\Delta T}$ (K)	n	s (K)
25	40	5.18	29	0.23
25	60	3.16	34	0.44
25	80	-	-	-
30	40	7.34	22	0.09
30	60	3.55	7	0.41
30	80	2.17	23	0.34
35	40	8.36	104	0.29
35	60	4.86	23	0.49
35	80	2.33	59	0.23
40	40	9.44	119	0.23
40	60	5.31	16	0.63
40	80	2.67	87	0.25
45	40	-	-	-
45	60	6.41	17	0.51
45	80	3.37	84	0.25

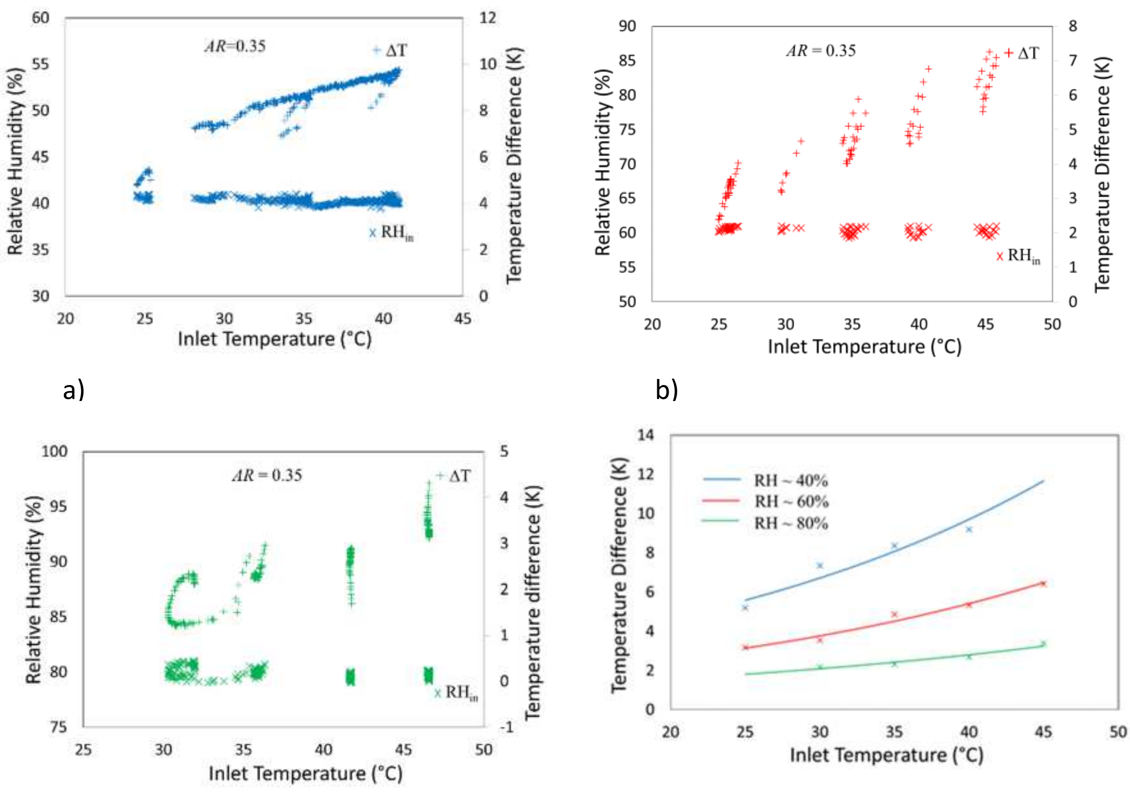
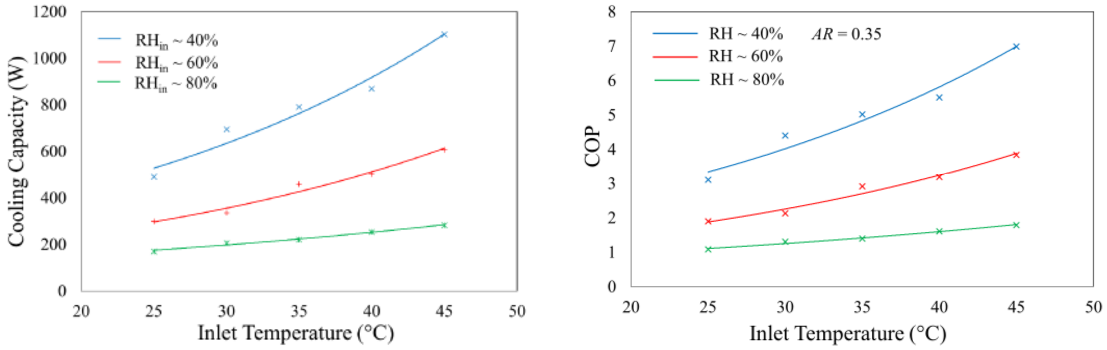


Figure 6 a) ΔT data at 40% RH, b) ΔT data at 60% RH, c) ΔT data at 80% RH, and d) variation in ΔT with T_{in} and constant RH.



a) b)

Figure 7. Variation in a) cooling capacity with T_{in} and constant RH, f) and b) COP with T_{in} and constant RH.

In figure 6d, temperature difference increases with inlet temperature and RH. For instance, at 25°C and 40% RH, the temperature difference is 5.5K and increases to 11.5K at 45°C. The system can therefore deliver a supply air temperature of approximately 20°C for an inlet temperature of 25°C and just under 33.5°C for an inlet temperature of 45°C. At 60% RH, temperature difference increases from approximately 3K at 25°C to 6.5K at 45°C, delivering processed air at 22°C and 38.5°C respectively. At 80% RH, temperature difference increases from just under 2K at 25°C to just over 3K at 45°C. This is to be expected as the driving mechanism for evaporative cooling is low.

Cooling capacity was determined based on the temperature difference and the average volume flow measured during the experiments. From a sample of n = 12 air speed measurements, the average volume flow for the supply and working air were determined and the working to supply air volume flow ratio (AR). Table 4 shows the results.

Figure 7a shows that the maximum cooling capacity was just over 1100W at an inlet temperature of 45°C and an RH of 40%, and the minimum was just over 130W at an inlet temperature of 25°C and an RH of 80%. These results show that substantial cooling can be achieved together with a decent sensible cooling effect.

The high temperature, low RH environment gives maximum performance because of the large driving mechanism, but it is applicable in only a few locations around the world, such as arid and dry regions, for example Houston, Arizona and Dubai, UAE. The largest potential market for such a system would be temperate, but hot climates around North America and the Mediterranean.

Table 4. Air speed and volume flows n = 12

	Air speed $\bar{U}$ (m/s)	Duct diameter (mm)	Volume flow $\bar{V}$ (m³/h)
Supply air	6.39±0.18	125	282.42±8.07
Working air	3.51±0.51	100	99.22±4.09
Working to supply air ratio AR			0.35±0.02

Table 5. Electrical power consumption n = 15

	Voltage $\bar{E}$(V)	Current $\bar{I}$(A)	Power $\bar{W}$ (W)
Blower 1	24.00	3.41	81.91±1.48
Blower 2	23.99	3.07	73.66±1.03
Pump	9.00	1.48	2.21±0.21
Total			157.78±2.13

These climates experience moderate and high cooling demands ranging from 25°C and 40% to 60%RH to 35°C and 40% to 60%RH, so we can see from the results that the system could provide a supply air temperature of 19°C and 22°C, with cooling capacities of 500W and 300W at an inlet temperature of 25°C and 27°C and 30°C with cooling capacities of 750W and

600W, respectively at an inlet temperature of 35°C. The voltage and current consumed by the blowers and the pump were recorded during testing and Table 5 shows the mean of the voltages and currents from the individual components and the average overall power consumption. The power consumed by the system was on average approximately 158W. This was used to determine the COP of the system. Figure 7b shows how COP varies with inlet temperature and relative humidity. At 40%RH, COP increases from 3.28 at 25°C to 6.85 at 45°C. As RH increases, COP decreases. At 60%RH, COP increases from 1.87 at 25°C to 3.78 at 45°C and at 80%RH, COP increases from 0.84 at 25°C to just under 2 at 45°C. At low to medium conditions the system provides cooling with a COP from 2 to 5, and can achieve a COP up to 7.

Discussion and conclusions

Preliminary testing showed great potential for the dew point cooler to provide cooling at high effectiveness. The product and working air flows discharged directly to the surroundings and so the pressure losses were low, resulting in an *AR* of around 4. When the core was integrated into a unit designed to provide cooling to a truck cabin and exhaust the working air, losses due to friction were introduced thus reducing flows. The working air flows were collected in side channels and ducted into one outlet and so there were large restrictions on that side, resulting in an *AR* of around 0.3. The restriction on working air flow meant that the system could not achieve high dew point effectiveness. However, we were able to achieve over 1kW of cooling and over 12K in temperature reduction in favourable operating conditions (low to medium RH and high ambient temperatures). The initial design utilised aluminium sections and plates to assist easy assembly and dismantling, but updates in its design will develop a one piece moulded plastic housing with a minimum of sharp turns, and restrictions. By improving the design, it is predicted that we can improve dew point effectiveness toward 80%, increase temperature reductions to 15K, cooling capacities of 1.5kW whilst reducing power consumption and thus increasing COP approaching 10.

Acknowledgements

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