Cooling tower, and (inset) condensate line supplying to cooling tower sump

Condensate Recovery System for Large AHU with Enthalpy Wheel

By Raji Panicker
Senior Design Lead - Mechanical
CH2M, Mumbai

Abstract

This article presents the study carried out to identify and implement water conservation measures in the Building HVAC Systems in hot and humid climates. The study is validated based on the Mumbai International Airport Ltd. project, and summarizes the calculations and analysis of the re-use of gross annual condensate recovered from recirculating AHUs with enthalpy wheel. The details are being presented to highlight the benefit of a condensate recovery system.

Introduction

It is well known that more than 70% of Earth is covered with water including the oceans. But few know that less than 1% of it is potable and 90% of that 1% is unavailable, being in the polar ice caps. That leaves us with 0.1% of water as potable. Sea water is not even usable otherwise, let alone potable. Sea water desalination is not a commercially viable route, at least today. Population is rising, but water sources are not. To add to the problem, more and more sources are getting polluted and yield water unfit for drinking. Our sustainability goal is to save more drinking water and reuse water for flushing, irrigation or other non-potable use. The scarcity is already acute and in years to come it has the potential of causing wars.

HVAC is energy and water intensive. So, we need to help conserve water and power in whatever way we can. The focus of this article is to establish the feasibility of harvesting air conditioning condensate for non-potable use in large commercial

About the Author

Raji Panicker has 16 years of experience in the HVAC industry, and is currently with CH2M, the program managers for Mumbai International Airport Pvt. Ltd. He is a LEED AP BD+C professional from Green Business Certification Institute (GBCI), and member of USGBC and MIET (UK).

continued on page 54

continued from page 52

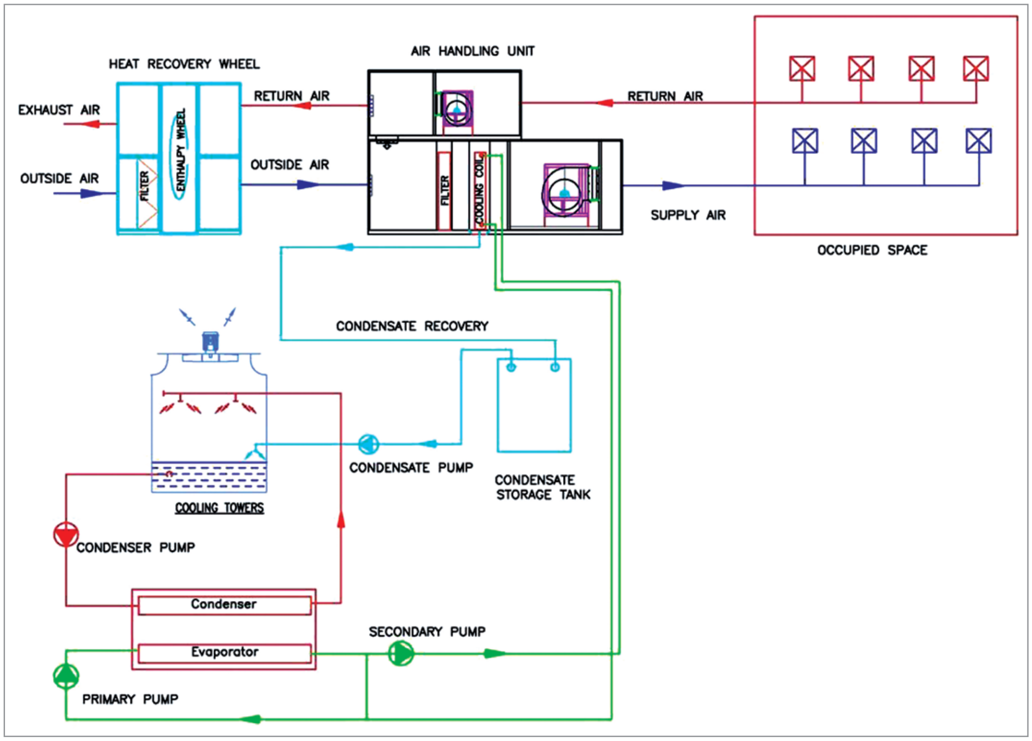

Figure 1: Condensate recovery schematic

installations, where we use large volumes of outside air. This study was done year round to take care of variations in the climatic conditions of Mumbai. It includes:

- Average outside air humidity ratio as per Mumbai conditions.
- Difference in moisture content when passing through the energy wheel.
- Amount of condensate recovered from the recirculating air handling unit (AHU).

The calculation is for the month of January, when we recover minimum condensate. The reader can calculate for other seasons on the same lines.

Detailed Study Procedure

The project considered in this study is a 13,700 TR system with 6 chillers of 2,500 TR each and 6 cooling towers of 3,125 TR each along with associated pumps. Condensate recovery was done for only 70% of the total cooling load, as it was implemented mid-project. The possible re-use of condensate is either for flushing, irrigation or cooling tower make-up. In this project we used it to replenish make-up water in cooling tower.

Condensate water collected from recirculating AHUs is non-potable, and can be used to supplement the make-up water requirement of cooling tower. There is a possibility of biological growth in condensate water, hence biocide treatment was done before supplying it to the cooling tower sump.

In warm and humid zones like Mumbai, the latent load is high enough to generate a substantial amount of condensate. The make-up water requirement for cooling tower was estimated at around 24,500,000 gallons/year (9.27 x 10^7 l/year), approximately 1.1% of the total usage, based on the data sheet from cooling tower manufacturer. *Figure 1* shows the schematic diagram of a re-circulating AHU along with enthalpy wheel and cooling tower.

Water is collected from individual Air Handling Unit from MEP rooms and directed into a main storage tank inside the building, from where it is pumped into the cooling tower sump. Condensate water is circulated through insulated pipes to avoid any heat loss during transmission. Out of the total make-up water requirement, 12% was supplied from the recovered condensate, and the balance 88% through a Reverse Osmosis plant. The total cost for setting up the condensate recovery system was Rs. 32 lakh (US$ 58,000).

Methodology and Results

The first step was to gather temperature and humidity data for Mumbai climatic conditions, which is available at the online weather site or US Department of Energy. *Equation 1* was used to determine the saturation pressure of the water vapor with respect to dry bulb or wet bulb temperature in absolute scale of either rankine or kelvin.

$$\ln(P_{ws}) = \frac{C_1}{T_{wb}} + C_2 + C_3.T_{wb} + C_4.T_{wb}^2 + C_5.T_{wb}^3 + C_6.\ln(T_{wb}) \quad \text{..... Equation 1}$$

Where

T_{wb} = Wet bulb temperature °R (K)
P_{ws} = Saturation pressure of water vapor, psia (Pa)
C_1 = -1.0440397 E+04
C_2 = -1.129 465 0 E+01
C_3 = -2.702 235 5 E-02
C_4 = 1.289 036 0 E-05
C_5 = -2.478 068 1 E-09
C_6 = 6.545 967

Referring *Table 1* and substituting wet bulb temperature of 63.45°F (17.47°C) in *Equation 1*, we get 0.289 psia.

The next step was to find humidity ratio of outside air dry bulb/wet bulb, which was done using *Equation 2*, wherein temperature is in °F.

$$W = \frac{(1093 - 0.556 t_{wb}) W_s - 0.240\,(t_{db} - t_{wb})}{1094 + 0.444\, t_{db} - t_{wb}} \quad \text{...... Equation 2}$$

$$W_s = 0.62198\left(\frac{P_{ws}}{P_{atm} - P_{ws}}\right) \quad \text{...... Equation 3}$$

continued on page 56

continued from page 54

Where

Ws = Humidity ratio of saturated air at wet bulb temperature

P_{atm} = Atmospheric pressure, 14.6 psia (Pa)

Refer *Figure 2*, which shows calculated average hourly moisture content per cubic feet of air for Mumbai conditions. It can be seen that maximum humidity ratio occurs in the month of June and minimum in January. Hence, minimum condition of January will be the basis for all analysis in this study.

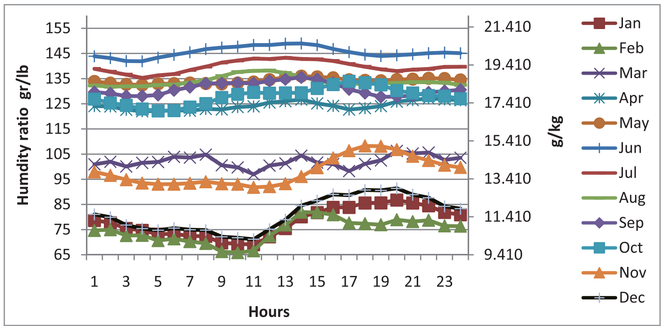

Figure 2: Average hourly statistics of outside air humidity ratio for Mumbai

Table 1: Average hourly statistics of weather condition for January

Hour	Dry Bulb °F (°C)	Wet Bulb °F (°C)	Humidity Ratio gr/lb (g/kg)
1	68.89 (20.50)	63.45 (17.47)	78.65 (11.24)
2	67.82 (19.90)	62.89 (17.16)	77.77 (11.11)
3	67.82 (19.90)	62.40 (16.89)	75.47 (10.78)
4	67.10 (19.50)	61.98 (16.66)	74.70 (10.67)
5	67.28 (19.60)	61.72 (16.51)	73.22 (10.46)

Once we had the humidity ratio of outside air, the next step was to calculate the humidity ratio of fresh air after the enthalpy wheel. Enthalpy wheel is used to recover latent and sensible energies, with an effectiveness of 75%. Enthalpy wheel used for the project was AHRI certified.

Equation 1 is used to calculate the humidity ratio of fresh air leaving the enthalpy wheel.

$$w_{fa} = w_{oa} + \varepsilon\,(w_{ra} - w_{oa}) \qquad \text{..... Equation 4}$$

Where

W_{fa} = Humidity ratio of fresh air leaving enthalpy wheel, gr/lb (g/kg)

E = Energy wheel total effectiveness

W_{ra} = Humidity ratio of return air entering enthalpy wheel, gr/lb (g/kg)

W_{oa} = Humidity ratio of outside air condition, gr/lb (g/kg)

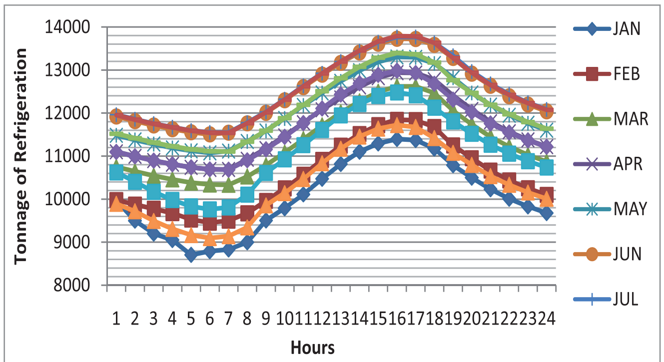

Figure 3: Cooling load profile

Air conditioning and ventilation for the building was done with multiple floor-mounted AHUs, treated fresh air units (TFUs) and fan coil units (FCUs), with chilled water being supplied from a district cooling plant with a ΔT of 14°F (7.7°C). Fresh air-flow requirement considered in the project was as per National Building Code (NBC) 2005 (India), which is 22% more than the requirement of ASHRAE 62.1.

Based on the load profile generated (*Figure 3*) from hourly analysis program (HAP) software, average hourly fresh air-flow range is considered in the calculation.

Using outside air condition from *Table 1* for January,

Dry bulb = 68.89°F (20.50°C)

Wet bulb = 63.45°F (17.47°C)

Humidity ratio = 79.03 gr/lb (11.29 g/kg)

To achieve better dehumidification, we need to keep the supply air leaving at 55°F or below. Exhaust air condition through enthalpy wheel is the return coming from AHUs and assumed constant round the year as the building is designed for comfort conditions. Data assumed are as below.

Return air condition:

Dry bulb = 77.0°F (25.00°C)

Wet bulb = 65.7°F (18.33°C)

Humidity ratio = 76.86 gr/lb (10.98 g/kg)

Substituting the above data in *Equation 1*, we get the humidity ratio of air per cubic feet leaving the enthalpy wheel.

W_{fa} = 78.65 + 0.75 (76.86 - 78.65) (I-P)

W_{fa} = 77.31 gr/lb.

W_{fa} = 11.24 + 0.75 (10.98 -11.24) (S-I)

W_{fa} = 11.04 g/kg

The output of the calculation is projected in *Figure 4*, which is an average hourly humidity ratio of air leaving the enthalpy wheel. If we compare the graphs of *Figure 2* and *4*, it can seen that there is a drop in humidity ratio after the enthalpy wheel.

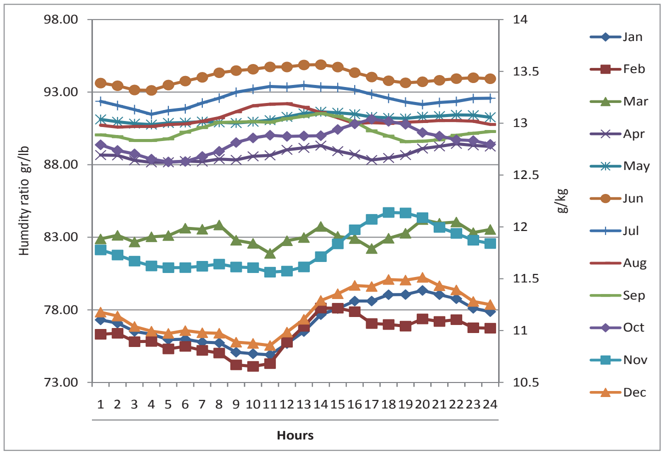

Figure 4: Average hourly statistics for moisture content of outside air leaving enthalpy wheel

Knowing the humidity ratio of the outside air leaving the enthalpy wheel, next step is the calculation of parameters of AHU mixed air entering the cooling coil. *Equation 5* determines the mixing air condition.

continued on page 58

continued from page 56

$$w_{ce} = \frac{(R_a \times w_{ra} + O_a \times w_{fa})}{(R_a + O_a)} \, (IP \text{ and } SI) \qquad \text{..... Equation 5}$$

Where

W_{ce} = Humidity ratio of cooling coil entering condition, gr/lb (g/kg)
R_a = Return air flow, ft³/min, (m³ /s)
W_{ra} = Humidity ratio of return air-flow, gr/lb, (g/kg)
O_a = Leaving enthalpy wheel fresh air air-flow, ft³/min (m³/s)

Using the above equation and referring *Table 2* data for air-flow, we get humidity ratio of coil entering condition.

$$w_{CE} = \frac{(1562282 \times 76.86 + 401358 \times 77.31)}{(1562282 + 401358)} \text{ (IP and SI)}$$

W_{CE} = 76.95 gr/lb (10.99 g/kg)

After we got the humidity ratio of air entering the cooling coil, the final step was calculation of the difference in humidity ratio of the air entering and leaving the coil as per *Equation 6.*

Figure 4 shows the difference in humidity ratio; and the average condensate generation potential per month for recirculating AHU is shown in *Figure 5*.

Equation 7 and *8* are used to calculate the total amount of condensate recovered from air handling unit.

$$\Delta\omega = w_{CE} - w_{SA} \qquad \text{..... Equation 6}$$

$$H = M \times \varrho \times 60 \times \frac{\Delta\omega}{7000} \times \frac{1}{8.33} \quad \text{I-P} \qquad \text{..... Equation 7}$$

$$H = M \times \varrho \times 3600 \times \left(\frac{\Delta\omega}{1000}\right) \quad \text{S-I} \qquad \text{..... Equation 8}$$

Where,

H = Condensate generated, gal/hr (l/hr)
M = Supply air flow, ft³/min (m³/s)
e = Air density, lb/ft³ (kg/m³)
$\Delta\omega$ = Difference in humidity ratio across cooling coil, gr/lb (g/kg)
W_{SA} = Humidity ratio of supply air leaving cooling coil, gr/l, (g/kg)

Table 2 shows the average total supply air, fresh air and return air from all AHUs in the building for the month of January, which has been derived from the cooling load software output sheet. For the purpose of calculation, air flow quantities have been added from all the individual AHUs.

Table 2: Average air flow and humidity ratio for the month of January

Hours	Supply air flow	Return air flow and humidity ratio		Outside airflow after recovery and humidity ratio		Mixing condition
	ft³/min (m³/sec)	ft³/min (m³/sec)	gr/lb (g/kg)	ft³/min (m³/sec)	gr/lb (g/kg)	gr/lb (g/kg)
1	1963639 (926)	1562282 (737)	76.86 (10.49)	401358 (189)	78.65 (11.24)	76.95 (10.99)
2	1873137 (884)	1490278 (703)	76.86 (10.49	382859 (180)	77.77 (11.11)	76.91 (10.99)
3	1813985 (856)	1443216 (681)	76.86 (10.49)	370769 (174)	75.47 (10.78)	76.79 (10.97)
4	1784410 (842)	1365565 (644)	76.86 (10.49)	364724 (172)	74.70 (10.67)	76.75 (10.96)
5	1716385 (810)	1379089 (650)	76.86 (10.49)	350820 (165)	73.22 (10.46)	76.67 (10.95)

To achieve the total condensate generated from AHUs, the difference in humidity ratio has to be multiplied with total supply air flow quantity. The density of the air is considered as 0.076 lb/ft³, (1.22 kg/m³). Below is the supply air temperature which is maintained at constant leaving condition throughout the year as the building is designed for comfort condition.

Dry bulb = 55.60°F (13.11°C)
Wet bulb = 55.00°F (12.77°C)
Humidity ratio = 63.63 gr/lb (9.09 g/kg)

$$H = 1963639 \times 0.076 \times 60 \times \left(\frac{76.95 - 63.63}{7000}\right) \times \frac{1}{8.22}$$

H = 2072 gal /hr (I-P)

$$H = 927 \times 1.227 \times 3600 \times \left(\frac{11.05 - 9.09}{1000}\right)$$

H = 8025 l/hr (S-I)

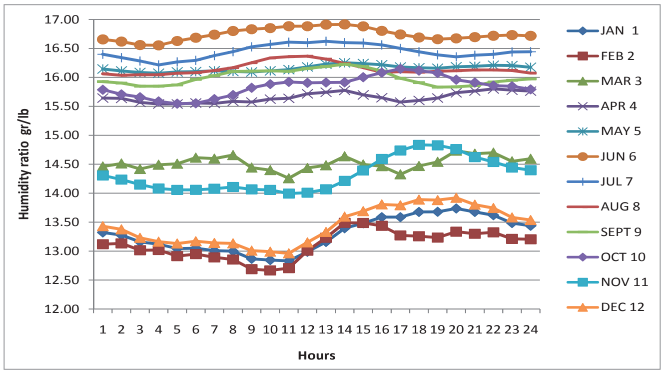

Figure 4: Difference in humidity ratio of cooling coil air entering/leaving per cubic feet of air

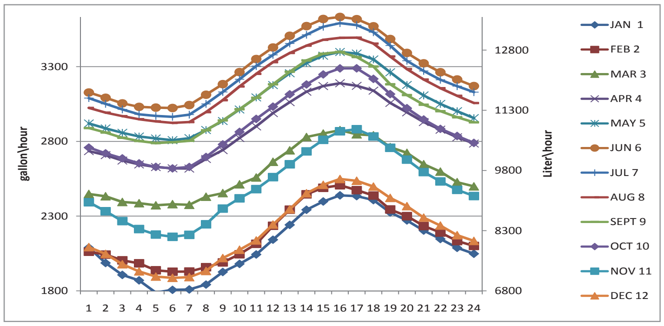

Figure 5: Condensate generation potential per month

Condensate generated annually is shown in *Figure 6*. Based on the amount of fresh air flow the condensate generation potential will change.

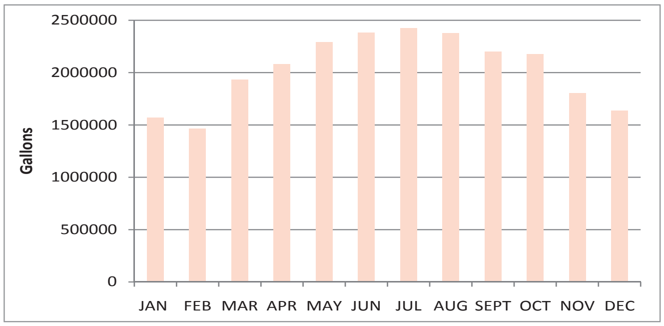

Figure 6: Average condensate generated monthly

continued on page 60

continued from page 58

The condensate that is collected is diverted into a storage tank of 16,000 gallons (6.0 x 10^4 l) capacity, with an annual condensate collection of 24,500,000 gallon/year (9.27 x 10^7 l/year). As stated earlier, the estimated cooling tower make up requirement annually is around 200,000,000 gallons (7.57 x 10^8 l), which is approximately 1.1% of total condenser water circulation rate. The captured condensate water supplemented around 12% of make-up requirement.

The cooling tower for this project consisted of a common sump, with total sump storage capacity of 1,52,380 gallon (5.76 x 10^5 l). Condensate water stored in the tank is pumped at 130 gallon per min (495 liters) into the cooling tower sump.

Our validation is ongoing for the amount of condensate generated by installing a water meter, the average water collection daily was recorded at around 45,000 USGPM (1,73,000 l). There are a few phases of the building that are still to be operational, and we expect the condensate generation rate to increase.

Figure 7: Water meter to monitor condensate flow rate

Conclusion

The above analysis shows that the gross annual condensate generation can reduce the make-up water demand of cooling tower by 12% annually. Considering the project scale, the percentage of water saved would significantly reduce the water bill. We can conclude that condensate water recovery is a way forward towards sustainable growth and could be adopted in other similar projects

References

2000 ASHRAE Handbook—HVAC Systems and Equipment, Chapter 44. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1997 *ASHRAE Handbook—Fundamentals*, Chapter 6, Psychrometrics. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE Green Guide, Third Edition—The design, construction and operation of sustainable buiding, Chapter 14. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Condensate water collection for an institutional building in Doha, Qatar: An opportunity for water sustainability, John A. Bryant and Tausif Ahmed, Texas A&M University, Qatar.