Analysis of Suitable Thermodynamic Properties and Conditions for Dew Point Cooling system

N N A Noramzil1, Z M Razlan1, I Zunaidi1, M Izham1, Shahriman A B1, A Harun2, M S M Hashim1, M K Faiz1, I Ibrahim1, N S Kamarrudin1, W K Wan1 and M F H Rani1

1School of Mechatronic Engineering, Universiti Malaysia Perlis, Pauh Putra Campus, 02600, Arau, Perlis, Malaysia
2School of Microelectronic Engineering, Universiti Malaysia Perlis, Pauh Putra Campus, 02600, Arau, Perlis, Malaysia

*Corresponding author: zuradzman@unimap.edu.my

Abstract. This study introduced dew point cooling system based on Maisotsenko Cycle Air-Conditioning (MAC/M-Cycle) for hot and humid climate. The purpose of this study to analyze the air thermodynamic properties and conditions supplied to the conditioned space. The room model has been developed to investigate the dry bulb temperature, specific humidity, and velocity distribution in the room. The simulation results showed a significance difference cooling capacity between these two systems. Consequently, this integrated system can realize significant energy saving and indoor air quality.

1. Introduction

Nowadays, clean and environment friendly energy is one of the fundamental human needs. Currently, usage of existing energy are mostly inefficient and/or involved in technology that harmful to environment. Such as, heating, ventilation, and air-conditioning (HVAC) are the biggest energy consumer today [1] due most air-conditioning systems are based on the vapor compression cycles which are consuming primary energy [2] and causing ozone layer depletion, greenhouse effect and global warming [3].

Besides, fossil fuels are diminishing rapidly and these reserves (oil, coal, and natural gas) contributing almost 80% of the world’s primary energy demand [4]. In addition, the existing renewable energy such as hydro, geothermal, solar, and the wind are contributing 13.1% only while nuclear power is contributing 6.5% of the total energy supply [4,5].

So, make environmentally friendly cooling techniques and low-cost air-conditioning will reduce the energy consumption. Direct and/or indirect evaporative cooling systems are environmental friendly and low-cost but due to thermodynamic limitation, it couldn’t maximize the utilization for this system in some conditions or situations such as at Oman, Turkey and Saud Arabia [2].

The dew point cooling system utilized the psychrometric renewable energy in the indirect evaporative cooling process by combining it with the thermodynamic processes of heat transfer with a much different airflow [6]. These system are based on Maisotsenko Cycle (M-Cycle) which has been discovered and patterned by Valeriy Maisotsenko at 1976 with patent number SU979796 and SU620745 [7]. This system combines the heat exchanger and advance indirect evaporative cooling (IEC) process to overcome the cooling limit which can reject heat of product air to the ambient dew point temperature theoretically compared to IEC process which can cool the air only to the incoming wet-bulb temperature [7].
Different from conventional systems which are used chlorofluorocarbon-based refrigerants (CFCs), dew point cooling system just uses water by utilizing water evaporation and latent heat [8], so it is more an environmental friendly. Besides, just using some fans and without using any compressor or pump, electricity demand is very low [8]. But, due to thermodynamic limitations, it is unfeasible air conditioning system in many situations such as in humid environment.

2. Methodology

The mathematical description and the cooling system configuration will be discussed in this section.

2.1. Mathematical description of the system

The governing equation of this study can be described as follow. The conservation of mass or also known as the continuity equation is expressed as:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$$  

(1)

The conservation of momentum equations is given as:

$$\rho \left( \mu \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = \rho \frac{\partial e}{\partial x} - \frac{\partial P}{\partial x}$$  

(2)

Mass flow rate of a fluid, need most on this analysis where can be described as below:

$$\dot{m} = \rho V A$$  

(3)

where $\rho$ is the density of the fluid, $V$ is the average fluid velocity, and $A$ is the cross-sectional area.

2.1.1. Cooling load calculation.

Based on ASHRAE, there are many methods available to calculate the heat load in the conditioned space such as total equivalent temperature differential/time-averaging method, a method of the transfer function and the heat balance method [9]. In this study, the cooling load temperature difference (CLTD)/cooling load factor (CLF) method employed to calculate the heat gain and cooling load of the conditioned space.

2.1.2. Performance evaluation.

Since this M-Cycle system could be cooled without dehumidification, there is only sensible cooling capacity described as:

$$\dot{Q_s} = \dot{m}_1 c_p \Delta T_1 = \dot{m}_1 c_p (T_{1,in} - T_{1,out})$$  

(4)

where $\dot{m}_1$ is the mass flow rate of the product air, $c_p$ is the specific heat of air, $(\Delta T_1 = T_{1,in} - T_{1,out})$ is the temperature difference between the inlet and outlet of product air, and $(\Delta h = h_{1,in} - h_{1,out})$ is the enthalpy difference of product air.

$$\dot{Q_s} = \dot{m}_1 h_{fg} \Delta \omega_1 = \dot{m}_1 h_{fg} (\omega_{1,in} - \omega_{1,out})$$  

(5)

$$\dot{Q}_{\text{tot,cooling}} = \dot{Q}_s = \dot{m}_1 (h_{1,in} - h_{1,out})$$  

(6)

2.2. Cooling system configuration

The conditions of the system both system illustrated as in Fig. 1. As in Fig. 1(a), where M is a mixture air of O the outside air and R the return air. While S is the supply air after the air goes through the cooling coil and may reach the dew point temperature of cooling coil.

Vapor Compression Air Conditioning (VAC) system analyzed in general with recirculation air and using a cooling coil to cool the supply air. While dew point cooling system based on M-Cycle (MAC) analyzed integrated with desiccant wheel where the sensible cooling provided by dew point cooling system and the latent cooling is provided by desiccant wheel.
3. Results
The total heat gain of the conditioned space is 2.0763kW with 1.9851kW and 0.0912kW for sensible and latent heat gain, respectively with room condition 27 °C dry bulb temperature and 19 °C wet bulb temperature. The temperature, specific humidity and velocity distribution discussed as follow section.

Figure 1 Schematic diagram for: (a) VAC system, (b) Dew point cooling system

Figure 2 Contour distribution for: (a) temperature, (b) specific humidity, (c) velocity.
3.1. Temperature distribution
Initial inlet temperature range 16-17°C, it is shown that the difference temperature distribution for VAC and dew point cooling system where the supply air temperature is in the range from 15.27°C to 17.94°C. Where in general the VAC system can provide lower temperature for both cases. This can explain due to the temperature blow to the exhaust fan and fast infiltration and 100% return air which is cool air will become exhausted air while the VAC system recirculated that cool air.

3.2. Specific humidity distribution
With initial inlet range of specific humidity is 10.2-10.4 kg/kg, result shows that the humidity increased in small range from 0.1 to 0.3 g/kg only. This is due to have continuously gain humidity from other factor which is from occupant and infiltration.

3.3. Velocity distribution
With initial inlet velocity range 1.5-2.5 m/s, it’s shown that the velocity supply air for M-Cycle case 1 not distributed much due to high velocity from the blower. The infiltration of VAC system shows contribution in velocity but not for M-Cycle system.

3.4. Cooling capacity and thermal comfort
Based on the cooling capacity as shown in Table 1, generally, the type 2 supply air is higher than type 1 supply air with discrepancy range from 0.38 to 0.68 kW which is M-Cycle case 2 can provide the higher cooling capacity than the others. The cooling capacity does not reach the heat load due to heat loss to surrounding and latent load does not remove totally in the room while latent heat gain is high 0.0912 kW.

Based on thermal comfort [10], the supply air after 1-hour operation still in the range of the thermal comfort. The mean velocity suggested from [11] which is below than 0.82 m/s for light sedentary activity while the mean velocity provided in the room is in range 0.12-0.14 m/s.

The best type and case is Type 2 with condition 2, it is shown in the Fig. 2 with contour of distribution for temperature, specific humidity and velocity. The location of contour located effectively in the middle of the room where occupant will sit most of the time.

| Type | Case | $T_{in}$, (°C) | $\omega_{in}$, (kg/kg) | $V_{in}$, (m/s) | $T_{ro}$, (°C) | $\omega_{ro}$, (kg/kg) | $V_{ro}$, (m/s) | $\Delta h$, (kJ/kg) | $\Phi_{room}$, (kW) |
|------|------|----------------|------------------------|----------------|----------------|------------------------|----------------|------------------|-------------------|
| VAC  | 16   | 0.0104         | 1.5                    | 1.9            | 0.0103        | 0.1246                 | 8.48           | 0.9779           |
| MAC 1| 16   | 0.0104         | 1.5                    | 19.5           | 0.01025       | 0.1358                 | 8.12           | 0.9364           |
| MAC 2| 16   | 0.0104         | 1.5                    | 19.7           | 0.01035       | 0.1333                 | 7.74           | 0.8926           |
| VAC  | 16.98| 0.0102         | 2.436                  | 19.8           | 0.0101       | 0.1246                 | 7.70           | 1.4421           |
| MAC 1| 16.98| 0.0102         | 2.436                  | 19.5           | 0.0101       | 0.1358                 | 6.50           | 1.2173           |
| MAC 2| 16.98| 0.0102         | 2.436                  | 20.5           | 0.0101       | 0.1333                 | 7.50           | 1.4046           |

4. Conclusion and recommendation
Based on this study, dew point cooling system can provide less cooling capacity than VAC system with discrepancy 0.2 kW which is not much different with the conventional cooling system in term of the cooling load. The system also needs an additional system to implement the system in a humid climate, where in this study the dew point cooling system integrated with the solid desiccant system. The suitable thermodynamic properties for dew point cooling system based on this study is 16.98°C dry bulb temperature, 0.0102 kg/kg and the velocity of the inlet is 2.436 m/s.

Future study can compare the dew point cooling system integrated with other system such as Vapor-AC, Desiccant-AC, or Ejector-AC. Besides, M-Cycle alone have many configurations for future study to increase the efficiency such as perforated working channel and a different configuration of working
and product channel. Based on this study also, the return air has wasted to exhaust air where the air is already cooled. The future project can study the effectiveness of using return air as working air.

**Acknowledgments**

Thousands The authors acknowledge School of Mechatronic Engineering, Universiti Malaysia Perlis for the facilities. Special thanks to those who contributed to this project directly or indirectly.

**References**

[1] Caliskan H, I Dincer, and A Hepbasli 2011 Exergetic and sustainability performance comparison of novel and conventional air cooling systems for building applications *Energy Build.* 43 1461–72

[2] Sultan M, I I El-Sharkawy, T Miyazaki, B B Saha, and S Koyama 2015 An overview of solid desiccant dehumidification and air conditioning systems *Renew. Sustain. Energy Rev.* 46 16–29

[3] Heidarinejad G, M Heidarinejad, S Delfani, and J Esmaeelian 2008 Feasibility of using various kinds of cooling systems in a multi-climates country *Energy Build.* 40 1946–53

[4] Asif M 2009 Sustainable energy options for Pakistan *Renew. Sustain. Energy Rev.* 13 903–9

[5] International Energy Agency (IEA) 2007 *Renewables in Global Energy Supply : An IEA Fact Sheet* (France: International Energy Agency (IEA) Publications)

[6] Caliskan H, A Hepbasli, I Dincer, and V Maisotsenko 2011 Thermodynamic performance assessment of a novel air cooling cycle: Maisotsenko cycle *Int. J. Refrig.* 34 980–90

[7] Mahmood M H, M Sultan, T Miyazaki, S Koyama, and V S Maisotsenko 2016 Overview of the Maisotsenko cycle – A way towards dew point evaporative cooling *Renew. Sustain. Energy Rev.* 66 537–55

[8] Tertipis D and E Rogdakis 2015 Maisotsenko cycle: technology overview and energy-saving potential in cooling systems *Energy Emiss. Control Technol.* 3 15–22

[9] Gadalla M and M Saghaififar 2016 Performance assessment and transient optimization of air precooling in multi-stage solid desiccant air conditioning systems *Energy Convers. Manag.* 119 187–202

[10] Shaharon M N and J Jalaludin 2012 Thermal Comfort Assessment-A Study Toward Workers’ Satisfaction in a Low Energy Office Building Mohd Nafiz Shaharon and Juliana Jalaludin Department of Environmental and Occupational Health, Faculty of Medicine and Health Sciences, *Am. J. Appl. Sci.* 9 1037–45

[11] International Organization for Standardization (ISO) 2005 *ISO 7730:2005 (E) Ergonomics of the thermal environment Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria* vol 3 (Switzerland: International Organization for Standardization (ISO))