Teknomekanik, Vol. 6, No. 2, pp. 67-81, December 2023
e-ISSN: 2621-8720 p-ISSN: 2621-9980

Experimental investigation of effect of extent and position of bypass openings on performance of a single unit liquid desiccant based indirect evaporative cooler

Pamu Raja Naveen[1,2*], Srinivas Kishore Pisipaty[2] and Siva Subramanyam Mendu[3]

[1] Department of Mechanical Engineering, Raghu Engineering College(A), Visakhapatnam, **INDIA**
[2] Department of Mechanical Engineering, Andhra University College of Engineering, Andhra University, Visakhapatnam, **INDIA**
[3] Department of Mechanical Engineering, MVGR College of Engineering (A), Vizianagaram, Andhra Pradesh, **INDIA**

* Corresponding Author: rjnvn15@gmail.com
Received Sep 02nd 2023; Revised Oct 21st 2023; Accepted Nov 03rd 2023

Cite this https://doi.org/10.24036/teknomekanik.v6i2.25172

Abstract: In high temperature and high humidity zones, evaporative cooling is ineffective and vapour compression systems are less energy efficient. Therefore, an alternative system is highly desirable which is effective, energy efficient and enables the use of cheap and sustainable energy sources. Indirect evaporative cooling helps in retaining humidity level of air, but is less effective in attaining lower air temperatures. To mitigate this challenge, M-cycle indirect evaporative cooling system helps in achieving sub-wet bulb temperatures. In this work, performance of a novel modified indirect evaporative M-cycle cooling system assisted by 40% aqueous Li-Cl liquid desiccant is experimentally investigated against various parameters. The cooling system used in this study is a single unit system which can perform indirect evaporative cooling, liquid desiccant dehumidification and internal cooling to the liquid desiccant. With an air velocity of 1 m/s at the inlet, the introduction of openings in between inlet and exit of the cooling system has shown a maximum improvement of 19.2% in its dew point effectiveness, with unaffected dehumidification effectiveness. Furthermore, it is observed that the dew point effectiveness is decreased with the increasing distance of openings from the inlet. The investigated cooling and dehumidification system is useful as a pre-air-conditioner to conventional air-conditioning systems and also as a stand-alone air-conditioning system.

Keywords: Liquid desiccant dehumidification; M-cycle; dew point evaporative cooler

1. Introduction

Air conditioning is an essential requirement these days rather than a luxury. Apart from the industrial purposes, it can be treated as a major energy consumption in domestic applications. Conventional Vapour compression systems (VCR) are highly energy consuming and researchers are trying to replace it with a less energy consuming alternative. In high temperature and low humidity climates evaporative cooling is useful to some extent with reasonable comfort. Evaporative cooling raises humidity of air which causes discomfort to the people in high temperature and high humidity climates. To compensate for this drawback indirect evaporative cooling is an alternative method. Although indirect evaporative cooling does not increase humidity of the air, however, this method is not as efficient as direct evaporative systems in cooling the air. In this regard, Hafiz M.U. Raza et al [1] fabricated 3 types of evaporative cooling systems viz., direct evaporative cooling (DEC), indirect evaporative cooling (IEC) and regenerative cycle (M Cycle) evaporative cooling for testing lab testing in an experimental work in Pakistan. Performances were compared and assessed. Direct evaporative cooling can achieve a lowest temperature of wet bulb temperature (WBT) of air. Temperatures achieved by indirect evaporative cooling are higher than those achieved by direct evaporative cooling. Many researchers worked on the practicability of indirect evaporative cooling in varied climatic conditions. A research done by Yang Y et al. [2] fully analyzes and compares indirect evaporative coolers with various designs and inlet air conditions. According to a review by Uzair Sajjad [3], The mechanical vapor compression technology that is now used in the air conditioning business can potentially be replaced with an M cycle cooling system.

An improvement of the indirect evaporative cooler is Maisotsenko cycle (M-cycle) that works on latent heat of vaporization of water and combines heat transfer with evaporative cooling. While with an indirect evaporative cooler, temperatures higher than wet bulb temperature are achieved, M-cycle can produce sub-wet bulb temperatures near to dew point temperature. Like in an indirect evaporative cooler, there are two channels, product air channel and working air channel, separated by a good heat conducting wall. In M-cycle systems, a part of the product air is diverted into the working air channel where it undergoes evaporative cooling, absorbing heat from air flowing through the product air channel. Sub-wet bulb temperatures near to dew point temperature are achieved due to the reason that the air is precooled before it enters the working channel. M-cycle systems find applications in HVAC, cooling towers, gas turbines and cooling of electronic equipment. In their study, Ragheb Raad et al. [4] simulated and evaluated the use of a back-worn cooling dress which employs regenerative evaporative cooling technique and discovered this method managed to dry air from this cloth than the single-channel design, resulting in a lower fabric temperature. Ali Mohammad Ez Abadi et al. [5] modeled and experimentally validated five different configurations of cooling systems based on regenerative cycle and concluded that these variations may be useful in different climatic conditions in order to achieve comfort conditions. In their work, Rubeena Kousar et al. [6] used experimentation to examine how a wide range of operational factors might affect the efficiency of counter flow and cross flow M cycle designs in view of all aspects. Imtiyaz Hussain et al [7] proposed a standby system in examining the effects of various climatic and method related factors on the working of direct, indirect, and Maisotsenko systems for temperature control in a variety of uses, including cultivation, animal care, electronic appliances, biotech, and agri-business. With a recommendation for vertical tube arrangement over horizontal arrangement, Xuchen Fan et al [8] constructed an innovative M-cycle based cooling tower and experimentally investigated its cooling efficiency. Sergey Anisimov et al [9] conducted a theoretical simulation on different variations in M-cycle cooling systems. Out of the variations, variation 2 - regenerative M-cycle showed lower outlet temperatures compared variation 3 – regenerative cooling system with perforations. In the perforated system, small multiple holes are made in the wall separating the product air and working air channels.

In high temperature and high humidity areas, dehumidifying the air is necessary to meet the comfort needs of the occupants. Liquid desiccants (LD) are promising alternatives to conventional Vapour compression systems in dehumidifying the air. Complete wetting of the surfaces has been a problem in liquid desiccant systems. In their study, Ronghui Qi et al. [10] proposed a silica super-hydrophilic blanket by the sol-gel method on pre-etched plastic surfaces to improve wetting performance, and they observed that this innovative coating could significantly improve the wetting performance on plastic surfaces. Because the way liquid flows, influences heat and mass transfer, Ronghui Qi et al. [11] suggested a mathematical model to assess the disperse patterns of air-liquid flow down convective absorption, for dehumidification, and they addressed the issue of completely wetting the surfaces.

Hybrid systems that combine Liquid desiccant systems with conventional VCR systems are also feasible and their performance evaluation is found in the literature. Experimental research has been done by Kumar S et al.[12] on a liquid desiccant dehumidifier coupled with a vapour compression refrigeration system. When compared to VCR alone, the highest gains of the system in COP and MRR are 8.09% and 26.75%, respectively. Kashish Kumar and Alok Singh [13] suggest a coalescence of a liquid desiccant air-conditioning system with a VCR machine as a good alternative, in hot and humid areas. Using an absorption system and vapor compression refrigeration system with subcooling and a cascade mode for downsizing, Erjian Chen et al. [14] suggested a solar-powered hybrid system with air cooling in their work.

A different category of hybrid systems that combine direct and indirect evaporative cooling with liquid desiccant dehumidification systems offer significant energy savings. Rubeena Kousar et al. [15] conducted a thorough analysis on economics and effectiveness of different configurations of a multistage dew point indirect evaporative cooler with solar desiccant system. Yanling Zhang et al. [16] prepared and tested a theoretical heat and mass transfer model and performed a thorough parametric analysis on an indirect evaporative cooled liquid desiccant air conditioning system with a hexagonal plate heat exchanger that features both counter flow and cross flow. Some researchers focused on environmental issues raised in air-conditioning. Reza Andika Setyadi and Rudy Setia budy [17] addressed practical issues in using variable frequency drives of heating, ventilating and air-conditioning in medical equipped health care centers. Fadhillah Ikhsan Dinul et al [18] investigated the effectiveness of using NaOH and Na_2CO_3 solutions as CO_2 absorbents.

One of the factors that deteriorates the cooling and dehumidification performance of liquid desiccant systems is rise of liquid temperature due to latent heat effects. Internal cooling is often provided by some means to overcome this problem. One of the arrangements for providing internal cooling to liquid desiccant is to use cooling water tubes over which liquid desiccant flows. Richard Jayson Varela et al. [19] analyzed the working of a novel adiabatic packed bed 3 fluid system with fins, employing a unique ionic liquid desiccant solution. Cui et al. [20] designed and simulated a system that integrates evaporative and desiccant systems. It combines the benefits of M-cycle systems and desiccant liquid systems.

All the above research was carried-out with a common aim of replacing the conventional vapour compression systems with low energy consuming systems, for air-conditioning applications. Most of the work was done on evaporative cooling and liquid desiccant dehumidification systems owing to their dependence on cheap and sustainable energy sources. Out of them, the systems which incorporated evaporative cooling and liquid desiccant dehumidification in a single unit are attractive because of their compact design. These systems suffer from inferior performance. Moreover, most of the work done on such systems is theoretical and limited experimental research was carried out. This shows the necessity of exploring and examining possible ways and modifications that can improve the performance of liquid desiccant assisted M-cycle cooling systems. Furthermore, experimental studies are much more needed to support the findings of theoretical and numerical studies. To fill this gap, experimental investigation was carried out on such a system in the present work. Many modifications in the previous designs were attempted to improve the cooling performance and the results are presented. The effects of various parameters on the cooling and dehumidification performance were observed and presented. It is possible to design liquid desiccant dehumidification systems in such a manner that the liquid desiccant used can be provided internal cooling with the help of an evaporative cooling in a regenerative way which is a more effective way of evaporative cooling. Such a system is designed, fabricated and tested for its performance under the effects of various parameters. Investigations were carried out by some researchers to assess the performance of M-cycle systems by providing multiple holes in the wall separating the two channels [9] . As a modification to these provisions, the influence of an intermediate opening provided at various locations and with various dimensions, is experimentally observed along with many other parameters that influenced its performance.

2. Material dan methods

2.1 Experimental setup

The experimental setup for the present investigation is basically an M-cycle based parallel plate evaporative cooling system assisted by liquid desiccant dehumidification showcased by the schematic diagram (Figure.1&2). Figures depict the whole experimental setup, which includes an air conditioning system and a liquid desiccant assisted indirect evaporative cooling system. The cooling system for the current study is fabricated by stacking together thin rectangular aluminium sheets of dimensions 1m X 0.2m, parallel to each other and 5mm apart, thereby making a number of parallel channels meant for flow of air through them. Aluminium sheets are chosen because they are good conductors of heat, available in very thin sheets and less costly. Alternate channels are meant for the flow of product air and working air.

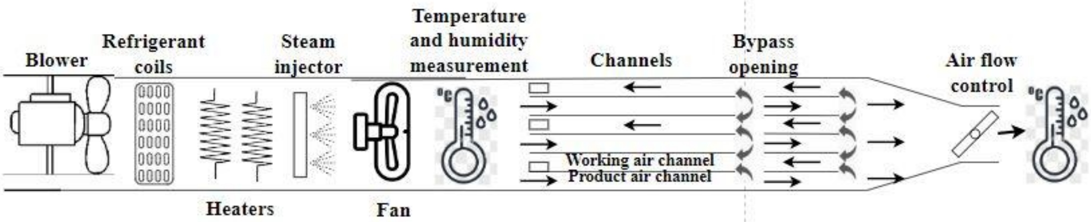

Figure 1 : Schematic representation of experimental unit

Air enters the product air channels at the front end of the setup and passes through the product air channels. The air that enters the cooling system at the entrance is controlled by a fan speed regulator. At the other end of the setup, by placing a partial restriction to the flow of air, a part of the air leaves the system as product (supply) air and the remaining air enters the adjacent working air channels. Air after flowing through the working air channels, leaves the cooling system at the front end through the openings provided at the top cover as exhaust air. The portion of the air diverted into the working channels is controlled by the sliding plate opening at the rear end. As the aluminium sheets are very thin, there is a very little resistance to heat transfer across them [21] . In the aluminium sheets, bypass openings are cut in various sizes and at various locations. Part of product air passes into working air channels through these intermediate openings before reaching the rear end. The effectiveness of the cooling system in reducing the temperature and humidity is examined in relation to the size and location of these intermediate bypass openings.

The walls of the working air channels are maintained wet by pasting filter paper on the walls and a continuous supply of water. Liquid desiccant is made to flow on the walls of the product air channels as a thin film, over a part of the total length of the cooling system. Two containers are arranged at an elevation to hold liquid desiccant and water from where these liquids flow into the cooling system by gravity. Ball valves are provided to regulate the flow in the pipelines that carry liquids into the system. Water flow is kept very low just to compensate for the evaporative loss. Water flow is kept very low just to compensate for the evaporative loss. The desiccant liquid, after flowing through the cooling system, is collected at the bottom which can be regenerated by some means and reused. The air flowing through product air channels undergoes dehumidification by coming into contact with the liquid desiccant flowing down over the walls by gravity. A 40% Lithium chloride aqueous solution is used as liquid desiccant solution. The air flowing through the working channels undergoes evaporative cooling, thus taking the heat of condensation of dehumidification and cools the product air [22].

For measuring air flow rates, a fan type anemometer is used. To keep track of the condition of the air, temperature and humidity monitoring equipment are kept at the entrance and exit of the

Teknomekanik, Vol. 6, No. 2, pp. 67-81, December 2023
e-ISSN: 2621-8720 p-ISSN: 2621-9980

cooling system. A vapor compression system-based air conditioner is used to maintain different entry air conditions to the cooling system. The air-conditioning unit consists of refrigerant evaporator coils, heaters, steam injector and a blower. The ambient air first enters the air conditioner. Its condition is altered to the desired inlet condition by changing its temperature and humidity ratio with the aid of the heater regulator, steam

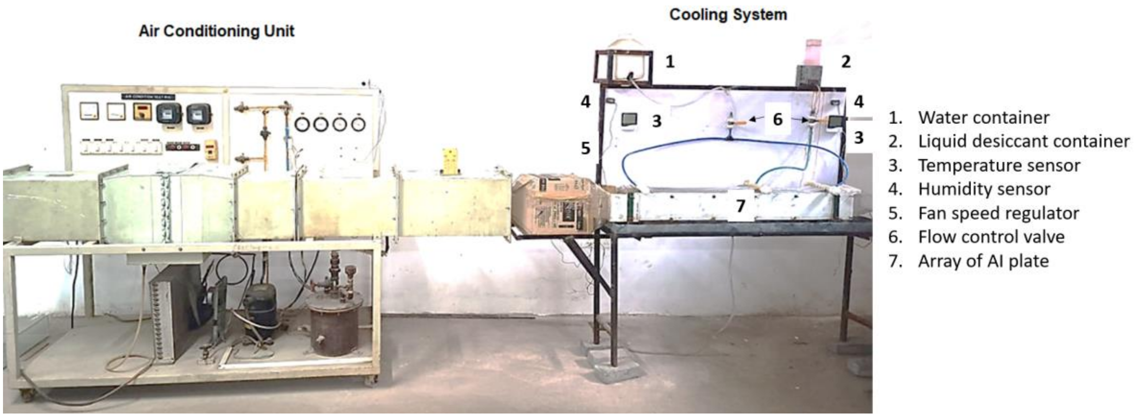

Figure 2 : Experimental setup

2.2 Calculations involved

Performance is evaluated in terms of air exit temperature, air exit humidity ratio, Dew point effectiveness and Dehumidification effectiveness. Dew point effectiveness and dehumidification effectiveness given by the following equations 1 [23] & 2 [24] .

$$Dew\ point\ \ effectiveness\ \varepsilon_{dew} = \frac{T_{in}-T_{out}}{T_{in}-T_{dew}} \tag{1}$$

$$Dehumidification\ \ effectiveness\ \varepsilon_{deh} = \frac{\omega_{in}-\omega_{out}}{\omega_{in}-\omega_{sat,T_{sol,in}}} \tag{2}$$

T stands for temperature and ω stands for humidity ratio of air, $\omega_{sat,T_{sol,in}}$ is the specific humidity of air contacting with liquid desiccant at a given liquid inlet temperature under saturation condition.

T_{in} and T_{out} are air temperatures of entering air and leaving of the cooling system. The instrument displays Relative humidity ($\emptyset$) of air.

Vapour pressure in air, P_v is calculated by $P_v = P_{vs} \times \emptyset$ (3)

where P_{vs} is the vapour pressure in air under saturated conditions which is read from steam tables corresponding to air temperature.

Humidity ratio can be calculated from $\omega = 0.622 \times \frac{P_v}{P_t - P_v}$ (4)

Where P_t is barometric pressure.

$\omega_{sat,T_{sol,in}}$ can be calculated by $\omega = 0.622 \times \frac{P_v}{P_t - P_v}$ (5)

where P_v in this equation is the vapour pressure in air that gets into contact with liquid desiccant and comes to the liquid inlet temperature and becomes saturated. This is taken from the data available in the literature [25].

3. Results and discussion

Several factors affecting the performance of the cooling system are observed, and their effects are tested experimentally. Performance is evaluated in terms of air exit temperature, air exit humidity ratio, Dew point effectiveness and Dehumidification effectiveness. Inlet air velocity is kept at 1 m/s throughout the experimental study.

3.1 Effect of return air to supply air ratio

Air enters the supply air channels in the examined system and after passing through the channels, a portion of the supply air is made to pass through return air channels and rest of the air leaves the system, as outcome. This ratio of working air flow to product air flow 'r' influences the state of the supply air and is varied in the range of 0.1 and 0.8. The relationship between working air to product air ratio and exit air temperature and humidity ratio is shown in Figure 3.

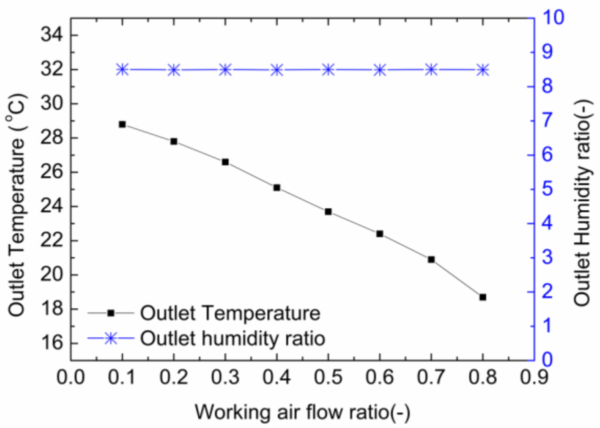

Figure 3 : Outcome of working air to product air ratio

With higher values of the ratio 'r', outlet temperature is reduced. This is because greater quantity of working air flowing through the working air channels, makes the evaporative cooling more effective [26] . Greater working air flow can absorb most of the heat of condensation resulting from the dehumidification process and significantly cool the product air. Dehumidification effectiveness does not get affected significantly. This is due to the unvaried liquid desiccant film length. Entering air temperature and specific humidity are kept at 30 °C and 16 g/kg. Extent of the desiccant solution film is kept 0.5 m out of the total length of 1 m. These results are compared to theoretical results of a more or less similar system in the literature [20] . Theoretical analysis was done using COMSOL MultiPhysics. Average deviations of 16.6% in outlet temperatures, 42.6% in dew point effectiveness, and 13.7% in dehumidification effectiveness are observed. These deviations can be attributed to the assumptions taken in the theoretical analysis or to physical losses in experimentation.

3.2 Effect of liquid desiccant flow length ratio

The relationship between exit air condition and liquid desiccant flow length is depicted in Figure 4. Outlet specific humidity is directly affected by the length of liquid desiccant film, showing a continuous reduction in humidity ratio. It is because of a greater area of desiccant film exposed to air flow [27] . The intensity of the dehumidification process is directly proportional to area of desiccant film. This is also true in case of cross flow arrangement of air and desiccant liquid flows, as in the examined system.

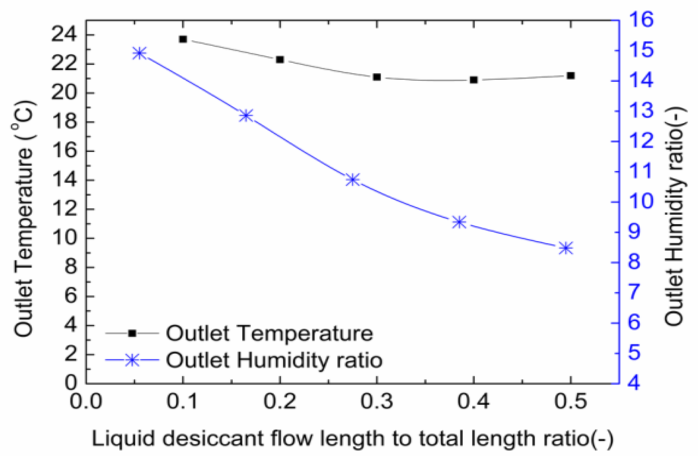

Figure 4 : Outcome of liquid desiccant flow length ratio

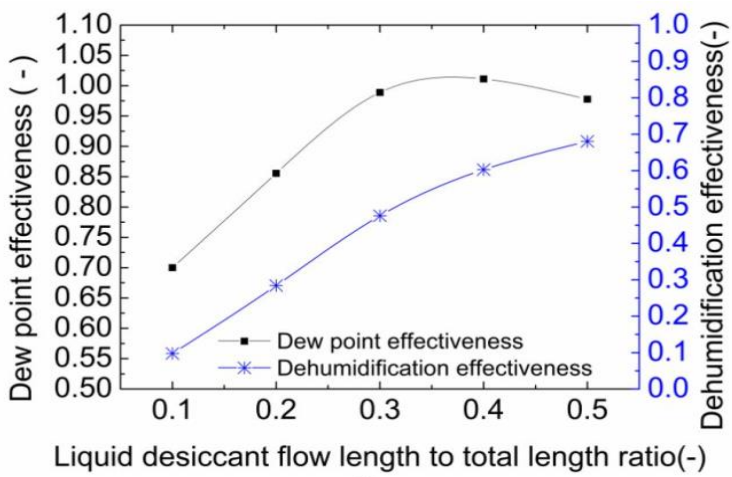

Figure 5 : Outcome of Liquid desiccant flow length

Outlet air temperature is lowered with an increase in desiccant flow length up to some length and shows a rise in temperature beyond that. This is because greater exposure of air to desiccant enhances the dehumidification process making air to become drier. Higher degree of dryness of working air results in better evaporative cooling. Evaporative cooling is more effective with drier air, to take away the heat condensation and to cool the product air. This is due to lower wet bulb temperature of drier air. Later with further increase in desiccant flow length, condensation heat in desiccant dehumidification dominates the enhanced evaporative cooling and results in a raise in temperature. This change in the tendency is seen at 30% of the total length. Return air to supply air flow rate ratio is maintained at 0.7. Inlet temperature and specific humidity are maintained as 30^{o} C and 16 g/kg. Figure 5 depicts the effect of liquid desiccant flow length on dew point effectiveness and dehumidification effectiveness. Dehumidification effectiveness is enhanced with the increased length of desiccant flow. Dew point effectiveness is raised with increase in liquid desiccant flow length up to some length and shows a fall beyond that. This is because dew point temperature is inversely proportional to outlet air temperature if the inlet air condition is maintained constant, according to the definition of dew point effectiveness.

3.3 Effect of inlet air temperature and humidity ratio

Working of the cooling system in varied conditions of ambient air is examined by providing inlet air to the system through an air-conditioning unit. Refrigerant coils, heaters and steam injection units bring the air to the test condition. Air is supplied at three different humidity ratios 12, 16 and 20 g/kg of dry air and at varied temperatures of 27.5, 30, 32.5, 35 and 37.5^{o} C. Return air to supply air flow rate ratio is maintained as 0.7. Length of the liquid film is kept 0.5 m.

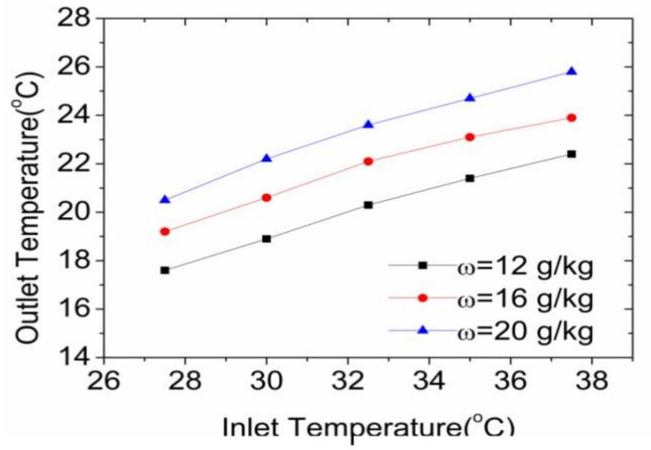

Figure 6 : Outcome of inlet air temperature and humidity ratio on outlet temperature

Figure 6 indicates the change of outlet temperature with inlet air temperature and humidity ratio. Exit air temperature is proportional to inlet air temperature. Lower outlet air temperatures are seen with low entering air humidity ratios. It is because with low humidity ratios, evaporative cooling is more effective. Figure 7 shows the effect of inlet air temperature and specific humidity on outlet humidity ratio and dew point effectiveness. At lower inlet air temperatures, dew point effectiveness is higher because the condition of air is nearer to dew point. Dew point temperature depends only on the humidity ratio and is independent of the dry bulb temperature of air. Any decrease in temperature brings the condition of air close to its dew point at lower inlet air temperatures, resulting in higher dew point effectiveness. This effect is more at higher entering air specific humidity. At higher inlet humidity ratios, dew point effectiveness is obviously greater because the entry condition of air is nearer to its dew point condition at a given temperature [28]. This can be observed from the psychrometric chart. Due to the positive slope of the saturation curve, the inlet air condition is nearer to its dew point at higher humidity ratios. This effect is less prominent at higher inlet temperatures. This because the closeness of the inlet air condition to its dew point temperature due to the positive slope of the saturation curve, is proportionally small in comparison to the large distance of

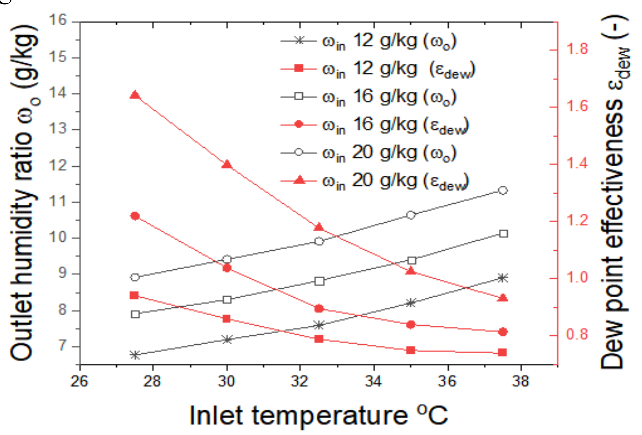

Figure 7 : Effect of inlet air condition on outlet humidity ratio and dew point effectiveness

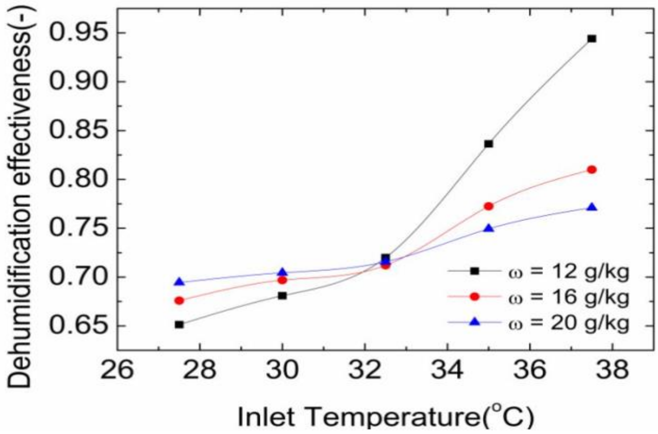

Figure 8 : Outcome of inlet air temperature and on dehumidification effectiveness

inlet condition from saturation curve at higher temperatures in the psychrometric chart. The influence of inlet temperature of air on outlet air humidity ratio seen in Figure 7 depicts that, at higher inlet air temperatures, dehumidification is not much effective resulting in higher outlet air humidity ratios. Liquid desiccants are much more effective in dehumidification at lower temperatures owing lower vapor pressures at the surface. Outlet air humidity ratios are obviously higher at higher entry air humidity ratios. With reference to Figure 8, Dehumidification effectiveness is higher at elevated inlet air temperatures. This is because of the fact that, at higher temperatures, vapor pressures at liquid desiccant surfaces are higher and nearer to the vapor pressure in inlet air. Any reduction in humidity brings the condition near to vapor pressure of the liquid resulting in higher dehumidification effectiveness. Higher temperatures are favorable for lower inlet air specific humidity with respect to dehumidification effectiveness. At higher temperatures, vapor pressures are higher at the desiccant liquid surface and at lower humidity ratios, vapor pressure in air is low, thus the vapor pressure difference is small and higher is the dehumidification. For instance, with very low vapor difference in between inlet air and desiccant liquid surface, dehumidification effectiveness is nearly 100% with little dehumidification, as per the definition of dehumidification effectiveness.

3.4 Effect of bypass air flow rate

As an alteration to improve the performance of the plate type liquid desiccant assisted M-cycle cooler, an opening is made in the wall separating the product air and working channels. Opening is provided in various extents and at different locations in order to observe the effect of these variations on dew point effectiveness and dehumidification effectiveness of the system. This intermediate opening bypasses a fraction of supply air into the evaporative air channels before reaching the rear end of the cooling system as seen in Figure 1. These bypass openings are kept at 0.6m from the front end of the cooling system. Rectangular openings are made through the entire height of the channel and of different lengths so that varied quantities of bypass air are diverted into working channels. The effect of bypass ratio, that is the ratio of quantity of air passed through the bypass openings to the total quantity of working air, is observed. Figure 9 describes the outcome of bypass air ratio on the leaving air condition at two varied working air ratios r=0.6 and r=0.7. A drop in outlet air temperatures is observed with increasing bypass air ratio up to some value around 0.65, beyond which temperatures rise. The air passing through the working air channel to the left of the bypass opening is a mixture of bypassed air and air coming from the upstream side. The bypassed air is drier and at higher temperature as it passes over the liquid desiccant film. The air approaching from upstream is relatively cooler and humid as it passes through product air channels without liquid desiccant film and over the water film in the working air channel. Dryness results in better evaporative cooling in the working air channel downstream of bypass opening and lower temperature of air results in better heat transfer across the channels. The air flowing through the working air channel downstream of bypass opening is a mixture of these two streams and its condition depends on the relative proportions of the constituent streams, resulting in the observed variation of outlet air temperature. Outlet air humidity ratio is unaffected with varied bypass air ratios as the length of the liquid desiccant film as well as working air to product air ratio are constant.

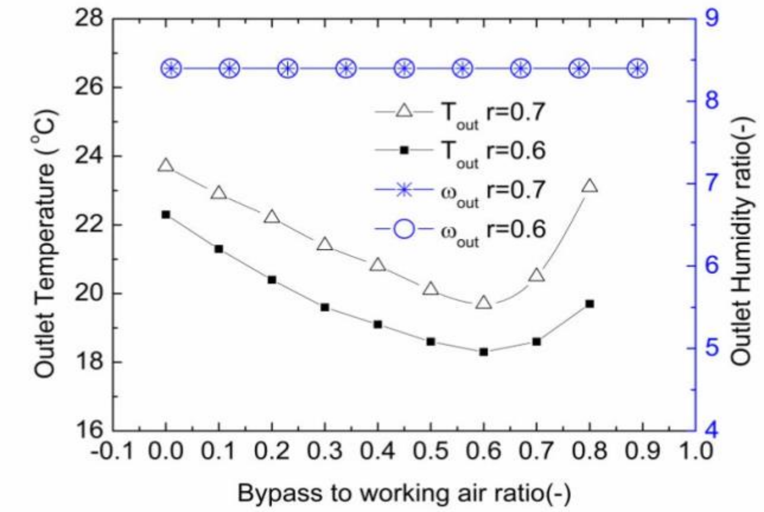

Figure 9 : Effect of bypass airflow rate

3.5 Effect of position of bypass opening

The bypass openings provided which allow a fraction of supply air to pass into evaporative air channels are made at different locations and the outcome of the position on the performance is observed. Bypass openings are made at various distances from the front end of the system and the ratio of this distance to the total length of the channels is taken as a parameter as seen in figure 10. Liquid desiccant flow is limited to 50% of the total length. Bypass openings are provided at various locations beyond desiccant liquid flow. Working air to product air flow rate is maintained 0.7 and bypass air flow ratio as 0.65. A continuous rise in outlet air temperature is observed with the location of bypass openings farther away towards the rear end. This is because the air flowing downstream of the bypass opening is drier when the bypass opening is nearer to the front end of the cooling system. Drier air undergoes evaporative cooling more effectively and absorbs most of the heat of condensation. However, leaving air specific humidity is unaffected by the location of bypass openings as the desiccant liquid film length is maintained constant.

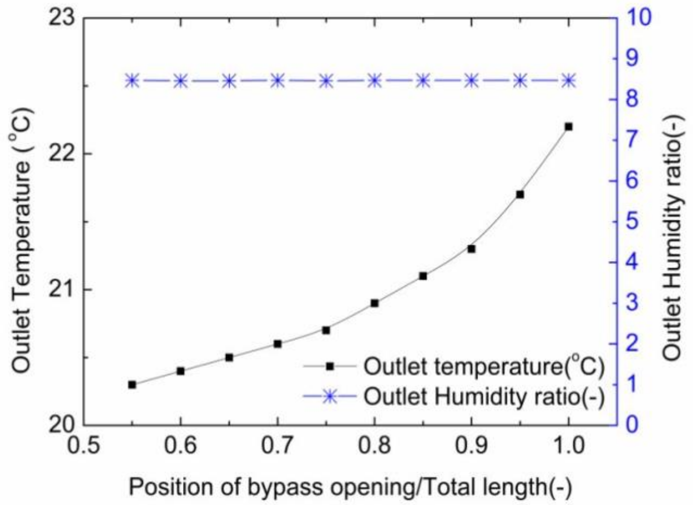

Figure 10 : Effect of position of bypass opening

4. Conclusion

A Li-Cl liquid desiccant assisted M-cycle indirect evaporative cooler is investigated experimentally. Performance of the cooler is examined against various parameters regarding air flow rates and ratios, extent of liquid desiccant exposure to air, inlet air condition in terms of temperature and humidity and also for the extent and location of intermediate openings provided in the plates separating product air and working air channels. Performance of the cooling system is evaluated in terms of dew point effectiveness and dehumidification effectiveness. Increased working air to product air ratio resulted in enhanced dew point effectiveness but does not influence its dehumidification effectiveness. Increased liquid desiccant flow length to total length ratio showed a raise in dew point effectiveness up to some value around 0.35L and a fall beyond that and showed a continued raise in dehumidification effectiveness. Higher inlet air temperatures resulted in a fall in dew point effectiveness. This effect is more significant at higher inlet air specific humidity. Higher entry air temperatures showed a rise in dehumidification effectiveness. This rise is steep at lower inlet air humidity ratios. Higher inlet air humidity ratios resulted in higher dew point effectiveness and lower dehumidification effectiveness. Wider bypass openings resulted in lower outlet air temperatures up to some value of bypass air to working air flow ratio around 0.6 and showed a rise in outlet air temperature beyond that. Dehumidification effectiveness is unaffected with bypass openings. Position of bypass openings nearer to the exit of product air resulted in higher outlet temperature continuously. These openings can only be provided beyond the length in which there is liquid desiccant flow. However, dehumidification effectiveness is unaffected with the provision of bypass openings. Most similar cooling systems combining evaporative cooling and liquid desiccant dehumidification have two or three units. The discussed cooling system is incorporated in a single unit in which air cooling, dehumidification and internal cooling to liquid desiccant are performed. The incorporation of intermediate openings in between product air and working air channels improved dew point effectiveness to a maximum of 19.2 %. However, dehumidification is unaffected. The investigated cooling and dehumidification system can be used as a pre-air-conditioner to conventional air-conditioning systems and also as a stand-alone air-conditioning system.

Author contribution

Raja Naveen Pamu: data collection, data analysis, experimentation, draft preparation, correspondence. Srinivas Kishore Pisipaty: supervision, reviewing and editing. Siva Subramanyam Mendu: conceptualization, methodology, validation.

Funding statement

This research received no specific grant from any funding agency in the public, commercial, or not-for-profit sectors.

Acknowledgements

Left Blank

Competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper and they have not received any research grants from finding agencies or financial support for attending symposia.

References

[1] H. M. U. Raza, M. Sultan, M. Bahrami, and A. A. Khan, "Experimental investigation of evaporative cooling systems for agricultural storage and livestock air-conditioning in Pakistan," *Build Simul*, vol. 14, no. 3, pp. 617–631, Jun. 2021, https://doi.org/10.1007/s12273-020-0678-2

[2] Y. Yang, C. Ren, C. Yang, M. Tu, B. Luo, and J. Fu, "Energy and exergy performance comparison of conventional, dew point and new external-cooling indirect evaporative coolers," *Energy Convers Manag*, vol. 230, Feb. 2021, https://doi.org/10.1016/j.enconman.2021.113824

[3] U. Sajjad *et al.*, "A review of recent advances in indirect evaporative cooling technology," *International Communications in Heat and Mass Transfer*, vol. 122, Mar. 2021, https://doi.org/10.1016/j.icheatmasstransfer.2021.105140

[4] R. Raad, M. Itani, N. Ghaddar, and K. Ghali, "A novel M-cycle evaporative cooling vest for enhanced comfort of active human in hot environment," *International Journal of Thermal Sciences*, vol. 142, pp. 1–13, Aug. 2019, https://doi.org/10.1016/j.ijthermalsci.2019.04.010

[5] A. M. Ez Abadi, M. Sadi, M. Farzaneh-Gord, M. H. Ahmadi, R. Kumar, and K. wing Chau, "A numerical and experimental study on the energy efficiency of a regenerative Heat and Mass Exchanger utilizing the counter-flow Maisotsenko cycle," *Engineering Applications of Computational Fluid Mechanics*, vol. 14, no. 1, pp. 1–12, Jan. 2020, https://doi.org/10.1080/19942060.2019.1617193

[6] R. Kousar, M. Ali, M. K. Amjad, and W. Ahmad, "Energy, Exergy, Economic, Environmental (4Es) comparative performance evaluation of dewpoint evaporative cooler configurations," *Journal of Building Engineering*, vol. 45, p. 103466, Jan. 2022, https://doi.org/10.1016/j.jobe.2021.103466

[7] I. Hussain *et al.*, "Evaluating the parameters affecting the direct and indirect evaporative cooling systems," *Eng Anal Bound Elem*, vol. 145, pp. 211–223, Dec. 2022, https://doi.org/10.1016/j.enganabound.2022.09.016

[8] X. Fan *et al.*, "An experimental study of a novel dew point evaporative cooling tower based on M-cycle," *Appl Therm Eng*, vol. 190, May 2021, https://doi.org/10.1016/j.applthermaleng.2021.116839

[9] S. Anisimov, D. Pandelidis, and J. Danielewicz, "Numerical analysis of selected evaporative exchangers with the Maisotsenko cycle," *Energy Convers Manag*, vol. 88, pp. 426–441, 2014, https://doi.org/10.1016/j.enconman.2014.08.055

[10] R. Qi, J. Zhi, and L. Z. Zhang, "Wetting improvement of plastic working plate for liquid desiccant dehumidification systems," in *Energy Procedia*, Elsevier Ltd, 2019, pp. 3076–3081. https://doi.org/10.1016/j.egypro.2019.01.994

[11] R. Qi, L. Lu, and Y. Huang, "Parameter analysis and optimization of the energy and economic performance of solar-assisted liquid desiccant cooling system under different climate conditions," *Energy Convers Manag*, vol. 106, pp. 1387–1395, Dec. 2015, https://doi.org/10.1016/j.enconman.2015.10.064

[12] S. Kumar, F. Ahmad, and M. Vohra, "Experimental Investigation on Hybrid Liquid Desiccant Cooling System for Hot and Humid Climatic Conditions of India," 2022, pp. 129–138. https://doi.org/10.1007/978-981-16-3132-0_13

[13] K. Kumar and A. Singh, "Assessment of dehumidification performance of non-corrosive desiccant in hybrid liquid desiccant-vapour compression system," *Energy Sources, Part A: Recovery, Utilization, and Environmental Effects*, vol. 44, no. 3, pp. 6909–6926, Sep. 2022, https://doi.org/10.1080/15567036.2022.2104966

[14] E. Chen, J. Chen, T. Jia, Y. Zhao, and Y. Dai, "A solar-assisted hybrid air-cooled adiabatic absorption and vapor compression air conditioning system," *Energy Convers Manag*, vol. 250, p. 114926, Dec. 2021, https://doi.org/10.1016/j.enconman.2021.114926

[15] R. Kousar, M. Ali, N. A. Sheikh, S. I. ul H. Gilani, and S. Khushnood, "Holistic integration of multi-stage dew point counter flow indirect evaporative cooler with the solar-assisted desiccant cooling system: A techno-economic evaluation," *Energy for Sustainable Development*, vol. 62, pp. 163–174, Jun. 2021, https://doi.org/10.1016/j.esd.2021.04.005

[16] Y. Zhang, H. Zhang, H. Yang, Y. Chen, and C. W. Leung, "Counter-crossflow indirect evaporative cooling-assisted liquid desiccant dehumidifier: Model development and parameter analysis," *Appl Therm Eng*, vol. 217, p. 119231, Nov. 2022, https://doi.org/10.1016/j.applthermaleng.2022.119231

[17] R. A. Setyadi and R. Setiabudy, "Harmonic Distortion Characteristics Generated by Heating Ventilation Air Conditioning System Case Study in PCR Laboratory," *Jurnal Pendidikan Teknologi Kejuruan*, vol. 6, no. 1, pp. 27–35, Mar. 2023, https://doi.org/10.24036/jptk.v6i1.31323

[18] F. I. Dinul, H. Nurdin, D. Rahmadiawan, Nasruddin, I. A. Laghari, and T. Elshaarani, "Comparison of NaOH and Na2CO3 as absorbents for CO2 absorption in carbon capture and storage technology," *Journal of Engineering Researcher and Lecturer*, vol. 2, no. 1, pp. 28–34, Apr. 2023, https://doi.org/10.58712/jerel.v2i1.23

[19] R. J. Varela, N. Giannetti, K. Saito, X. Wang, and H. Nakayama, "Experimental performance of a three-fluid desiccant contactor using a novel ionic liquid," *Appl Therm Eng*, vol. 210, p. 118343, Jun. 2022, https://doi.org/10.1016/j.applthermaleng.2022.118343

[20] X. Cui, M. R. Islam, B. Mohan, and K. J. Chua, "Theoretical analysis of a liquid desiccant based indirect evaporative cooling system," *Energy*, vol. 95, pp. 303–312, Jan. 2016, https://doi.org/10.1016/j.energy.2015.12.032

[21] X. Zhao, S. Liu, and S. B. Riffat, "Comparative study of heat and mass exchanging materials for indirect evaporative cooling systems," *Build Environ*, vol. 43, no. 11, pp. 1902–1911, Nov. 2008, https://doi.org/10.1016/j.buildenv.2007.11.009

[22] T. Zhang, X. Liu, J. Jiang, X. Chang, and Y. Jiang, "Experimental analysis of an internally-cooled liquid desiccant dehumidifier," *Build Environ*, vol. 63, pp. 1–10, May 2013, https://doi.org/10.1016/j.buildenv.2013.01.007

[23] S. S. Chauhan and S. P. S. Rajput, "Parametric analysis of a combined dew point evaporative-vapour compression based air conditioning system," *Alexandria Engineering Journal*, vol. 55, no. 3, pp. 2333–2344, Sep. 2016, https://doi.org/10.1016/j.aej.2016.05.005

[24] S. Bouzenada, C. McNevin, S. Harrison, and A. N. Kaabi, "An experimental study on the dehumidification performance of a low-flow falling-film liquid desiccant air-conditioner," in *Procedia Computer Science*, Elsevier B.V., 2015, pp. 796–803. https://doi.org/10.1016/j.procs.2015.05.135

[25] M. Sahlot and S. B. Riffat, "Desiccant cooling systems: A review," *International Journal of Low-Carbon Technologies*, vol. 11, no. 4, pp. 489–505, 2016, https://doi.org/10.1093/ijlct/ctv032

[26] Kalpana and S. Subudhi, "Developments in liquid desiccant dehumidification system integrated with evaporative cooling technology," *International Journal of Energy Research*, vol. 46, no. 1. John Wiley and Sons Ltd, pp. 61–88, Jan. 01, 2022. https://doi.org/10.1002/er.6713

[27] R. Qi, C. Dong, S. Yu, and L.-Z. Zhang, "Modelling and experiments of falling film break-up characteristics considering mass transfer for liquid desiccant dehumidification," *Int J Heat Mass Transf*, vol. 181, p. 122027, Dec. 2021, https://doi.org/10.1016/j.ijheatmasstransfer.2021.122027

[28] H. Wang, L. Liu, L. Liu, and Q. Cheng, "Performance analysis of different air conditioning systems in apartment buildings under different climates in China," *International Journal of Refrigeration*, vol. 139, pp. 192–203, Jul. 2022, https://doi.org/10.1016/j.ijrefrig.2022.04.007

Appendix

Uncertainty analysis

A methodical procedure described by Beckwith et al [32] is used to assess the flaws connected to experimental analysis. The degree of uncertainty surrounding the data obtained using various instruments is shown in Table 1. For each of the parameters utilized in the analysis, the maximum potential error is calculated and compiled in Table 2.

a) Specific humidity, ω

$$\omega = 0.622 \times \frac{\phi p_{vs}}{p_t - \phi p_{vs}},\ \frac{U_\omega}{\omega} = \sqrt{\left(\frac{U_\phi}{\phi}\right)^2 + \left(\frac{4U_T}{T}\right)^2} = \sqrt{(1.0101)^2 + (4 \times 0.1428)^2} = 1.1604\%$$

b) Air flow rate, Q

$$Q = l^2.C,\ \frac{U_Q}{Q} = \sqrt{\left(\frac{U_C}{C}\right)^2 + \left(\frac{2U_l}{l}\right)^2} = \sqrt{(0.3334)^2 + (2 \times 0.1666)^2} = 0.4702\%$$

c) Dew point effectiveness, ϵ_{dew} $\qquad \epsilon_{dew} = \frac{T_i - T_o}{T_i - T_{dew}}$

$$\frac{U_{\epsilon_{dew}}}{\epsilon_{dew}} = \sqrt{\left(\frac{U_{T_i - T_o}}{T_i - T_o}\right)^2 + \left(\frac{U_{T_i - T_{dew}}}{T_i - T_{dew}}\right)^2} = \sqrt{(0.1835)^2 + (-0.18348)^2} = 0.2595\%$$

d) Dehumidification effectiveness, ϵ_{deh} $\qquad \epsilon_{deh} = \frac{\omega_i - \omega_o}{\omega_i - \omega_{sat,T_{sol}}}$

$$\frac{U_{\epsilon_{deh}}}{\epsilon_{deh}} = \sqrt{\left(\frac{U_{\omega_i - \omega_o}}{\omega_i - \omega_o}\right)^2 + \left(\frac{U_{\omega_i - \omega_{sat,T_{sol}}}}{\omega_i - \omega_{sat,T_{sol}}}\right)^2} = \sqrt{(1.007)^2 + (-1.0502)^2} = 1.455\%$$

Table 1 : Uncertainties of instruments and properties

SNo	Instrument name	Instrument's range	Measured variable	Least count of the instrument	Least. and highest values measured in the experiment.	Uncertainty (%)
1	LCD Digital Thermometer 1	-50 – 70° C	Inlet air temperature	0.1° C	27.5 – 37.5° C	0.1428
2	LCD Digital Thermometer 2	-50 – 70° C	Leaving air temperature	0.1° C	17.6 – 28.8° C	0.1428
3	LCD Digital Humidity meter 1	10 – 99 % RH	Relative humidity of inlet air	1 % RH	30.4 – 87.2%	1.0101
4	LCD Digital Humidity meter 2	10 – 99 % RH	Relative humidity of leaving air	1 % RH	35.2 – 82.3%	1.0101
5	Fan anemometer	0 – 30 m/s	Velocity of air	0.1 m/s	0 – 3 m/s	0.3334
6	Steel rule	0 – 30 cm	Length	0.5 mm	0 – 300mm	0.1666

Table 2 : Uncertainties of parameters and variables

SNo	Variable	% Uncertainty error
1	Specific humidity	1.1604
2	Air flow rate	0.4702
3	Dew point effectiveness	0.2595
4	Dehumidification effectiveness	1.4549