EXPERIMENTAL INVESTIGATION OF AN INTEGRATED WIND INDUCED VENTILATION AND EVAPORATIVE COOLING SYSTEM

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Abstract

Low energy systems such as earth-air tunnels (EAT) and evaporative coolers are effective ways of reducing the energy demand for building ventilation and cooling. Direct evaporative coolers are effective in regions of low humidity. Their performance can be enhanced by coupling them to an EAT as pre-coolers. The reliance on electrical fan power can be reduced by using low pressure evaporative coolers and capturing wind when available. The paper reports on an experimental system developed to simulate a wind induced low pressure evaporative cooler, using mains pressure misting nozzles which is open to a 10.1 m^2 room with window opening. A section is included in the duct to simulate the pressure drop in EAT. The system performance is evaluated for different wind speeds, ambient conditions, and pressure drops. The results show that significant cooling and ventilation can be achieved with this combination. They also demonstrate that the prevailing wind velocity can play a major role in inducing air flow in an EAT and hence reducing the reliance on mechanical fan usage.

1. Introduction

Evaporative coolers provide a viable alternative to refrigeration air-conditioning systems in relatively dry and hot areas because of their lower installation and energy costs (El-Refaie & Kaseb 2009). Sensible pre-cooling in conjunction with direct evaporative cooling is one way of extending the climatic regions for evaporative cooling suitability. Earth air tunnels (EAT) make use of the ground as a heat sink and provide effective pre-cooling of the incoming air without adding any moisture, hence reducing both the dry bulb and wet bulb temperatures of the outside air . Such combination may result in a significant energy saving potential (Bansal & Mathur 2009; Heidarinejad et al. 2010). The major energy consuming element in EAT is a fan. Still EAT can be coupled to a wind tower (wind catcher), and whenever sufficient wind is available the system works purely in a passive mode as long far as the pressure drop in the whole system is kept low. Direct evaporative coolers require a significant fan power because of the associated pressure drops in the wet pads and it is practically difficult to couple them to a wind induced EAT. Misting is a low pressure alternative for evaporative cooling. In the air-conditioning industry, misting is gaining attention in the application of outdoor air cooling using misting fans, passive downdraught evaporative cooling, and pre-cooling in air cooled chillers (Ford et al. 1998; Wong & Chong 2010; Yu & Chan 2011). This paper presents an experimental investigation of wind induced EAT and evaporative cooling using low pressure misting. An experimental arrangement was developed for a single room building exposed to a wind driven EAT coupled to spray driven evaporative cooling. The objective of this study is to investigate and quantify the cooling capability of this combination of passive cooling elements during summer conditions.

2. Experimental arrangement

The layout for the indoor experimental set up is presented in fig. 1. A 3.6 m insulated duct with square cross section of 0.59 m side length is connected to a room with a floor area of 10.1 m^2 (width 2.19 m, length 4.65 m) and ceiling height of 2.4 m. The duct inlet to the room is positioned at 0.72 m from the floor. The square window opening with the same area to the duct cross section is located on the opposite wall of the room and its bottom is positioned at 1.56 m from the floor. The airflow rate is measured at the window with an Accubalance air capture hood with +/- 2.5 l/s accuracy. 12 T-type thermocouples with an error of +/- 1 °C were placed in the rig at various locations as shown in fig. 1. Four wire RTD sensors with an error of +/- 0.1 °C were placed at the mid inlet and outlet opening of the duct, at the mid height of the room and the center of the window. In total 4 thermocouples and 1 RTD are equally spaced vertically at the duct inlet and outlet section and the window at the mid lateral position. The temperature sensors were protected from possible water drift at the outlet of the duct. Three HIH series relative humidity sensors with accuracy of 3.5 % were used at the duct inlet, outlet and the window near the RTD sensors. A commercial software and data acquisition system is used to collect the temperature and relative humidity data.

We used a variable speed 300 mm axial fan to produce the dynamic head at the inlet for simulating wind. The fan is located at 1.1 m from the inlet to the duct. A Diffusion chamber and flow straighter (0.59 m square cross section) were attached to the axial fan. The inlet air temperature to the duct is controlled by a 15 kW heater with variable thermostat setting. Care was taken so that the heater fan does not affect the wind generated by the main stream fan. The space where the duct is located is widely open to the outside so that the static pressure developed in the space is negligible and resembles that of the atmosphere. The experiment included three phases: determining the stagnation pressure on the inlet wall for sealed building for calculation of the wind pressure coefficient, pressure drop test for different profiles to simulate the earth air tunnel (EAT), and misting evaporation performance tests.

As the experiment is not done in the wind tunnel, it is difficult to evaluate the free stream wind velocity. We rely on measured stagnation pressure on inlet opening of the duct P_{sin} for sealed building and the wind pressure coefficient on the inlet wall is obtained with the help of 3D CFD simulation using FLUENT considering the boundary layer of the free stream velocity for open tertian. 8 pressure taps were placed on the closed inlet wall of the duct (fig. 2) and the average pressure P_{sin} is measured by TT500s micro-manometer (0.04 Pa accuracy) for different fan speeds.

A pressure drop test was performed by placing different profiles in the duct. Airflow is induced by the variable speed mixed fan located on the ceiling of the room and the window is sealed. A flow straighter is located at the inlet to the duct, and the profiles are placed 2.4 m away from the duct inlet. Pressure taps are located at 10 cm before and after the profiles on each wall of the duct as show in fig. 1. The pressure differential is measured using a TT500s micro-manometer and the airflow rate through the duct is measured using the air capture hood.

The following profiles were used to simulate the pressure drop due to friction in EAT pipes

1. Fly screen (1 mm x 2 mm, 0.15 mm wire thickness)
2. Single layer perforated steel plate (staggered, 4.5 mm hole, 6.4 mm pitch, 1.2 mm thick)
3. Single layer perforated steel plate(staggered, 4.5 mm hole, 7.9 mm pitch, 1.6 mm thick) and
4. Double layer perforated steel plate (similar to type 3)

In the evaporator section, low pressure acetal plastic misting nozzles were used for this experiment. Nozzle with low flow rate and wide angle were preferred for the experiment. We used a nozzle with 115 degree head angleand measured the water flow rate of the nozzle in a separate test setup. At 5 bar, which is typical of the main pressure in Adelaide, Australia. The flow rate of the nozzle was found to be 2.85 l/hr. The location of the nozzles and the drift protectors in the evaporator section is shown in fig. 1. Two nozzles in parallel gave good moisture distribution. Hence we used such arrangement for the rest of the experiment. The nozzles faced the downstream of the airflow so as to minimize the pressure drop due to drag on the airflow. The nozzles were given a slight upward inclination so that the mist is distributed uniformly in the evaporator section. The water pressure in the nozzles was controlled by a manual valve. The main problem with misting application is the possibility of moisture droplets drift to the conditioned space. A number of profiles were trialed to prevent the carryover . Flyscreen, louver and polyester fabric were used (fig. 3) .

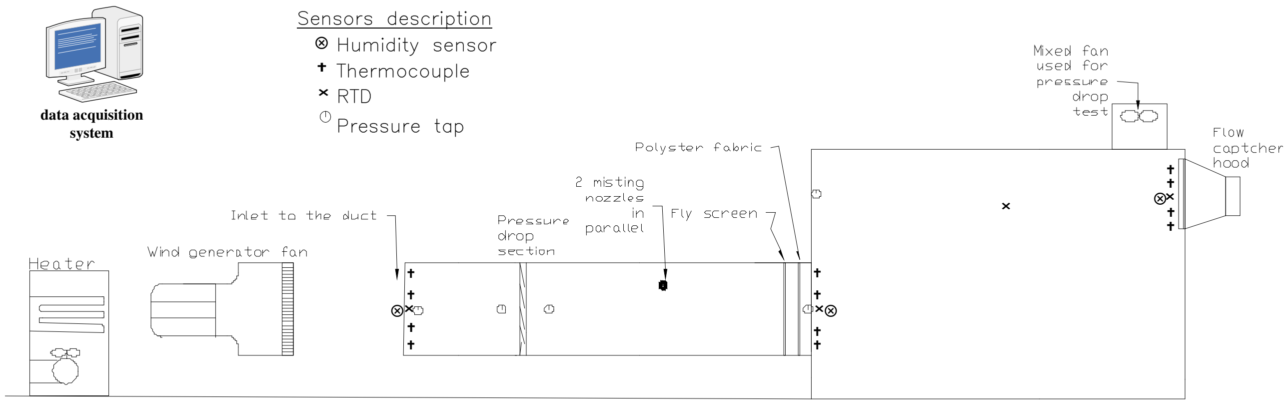

Fig. 1 : Schematic diagram of the experimental layout

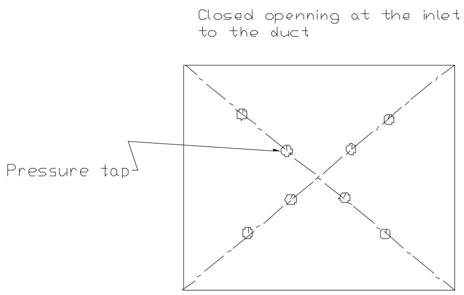

Fig. 2: Pressure taps arrangment on the closed opening at the inlet duct for measurement of wind pressure

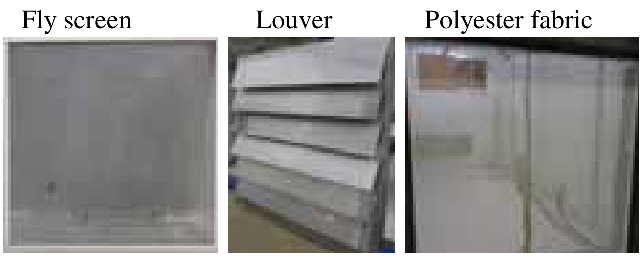

Fig. 3: Different carryover preventors investigated in the experiment

3. Test results and discussion

3.1. Wind speed simulation

The wind pressure coefficient (Cp) on the inlet duct with reference to the free stream wind on the building ceiling height ($H_{ref} = 2.4\, m$) is found to be 0.72 for open tertian (roughness n=0.143, (Liddament 1996)) based on FLUENT's simulation finding. Hence the corresponding wind velocity at 10 m height (the standard measuring height (H_{met}) for meteorological wind data) is predicted using power law

$$U = \left|\frac{2P_{sin}}{C_p \rho}\right|^{0.5} \left(\frac{H_{met}}{H_{ref}}\right)^n \quad (1)$$

Where ρ is the density of air

Fig. 4 shows the measured stagnation pressures (P_{sin}) and the corresponding wind speed (U) at 10m.

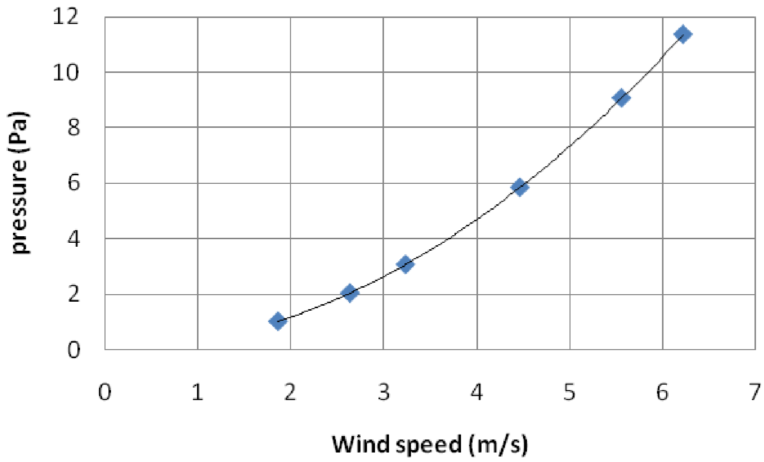

Fig. 4: The available wind pressure at the inlet wall and the corresponding wind speed at 10 m height

3.2. Pressure drop test for EAT simulation

The resistance coefficient (k value) for each component used is found using regression analyses for the measured pressure drop and air speed in the range of 0-1.5 m/s. Tab. 1 shows the k values for each profile in the measured airflow rate range.

Tab. 1: k_values for different profiles in the pressure drop section

Type number	**description**	**k_value**	**Regression R^2**
1	Fly screen	1.3	0.988
2	Single layer perforated steel plate	1.88	0.998
3	High pitch Single layer perforated steel	5.97	0.998
4	Double layer perforated steel plate	9.46	0.998

A 0.5 m diameter polyethylene pipe is assumed to be used in EAT and the corresponding length which gives similar pressure drops to the profiles is calculated. It is assumed that besides friction in the pipe, a 90^0 smooth bend at the inlet and outlet and in every 12 m length is used. Fig. 5 shows the pressure drop in each profile and in the corresponding equivalent pipe length with respect to the wind velocity in the tunnel. Pipe lengths of 10 m, 12 m, 60 m, and 100 m can represente the pressure drop of the components used with average percentage error of less than 10% for each types in the duct air speed range 0-1.5 m/s as the pipe friction factor doest not vary significantly in this range.

3.3. Carryover protectors

Fig. 6 shows the pressure drop in the different drift separators considered while the mister is on. The louver and the fabric with the flyscreen were able to capture the carryover, however the pressure drop in the louver is too high. The pressure drop across the louver is over 50% higher than the pressure drop across combined flyscreen and cloth at a duct air speed of 0.5 m/s. Hence the combined fabric and flyscreen was used as a carryover protector for the experiment.

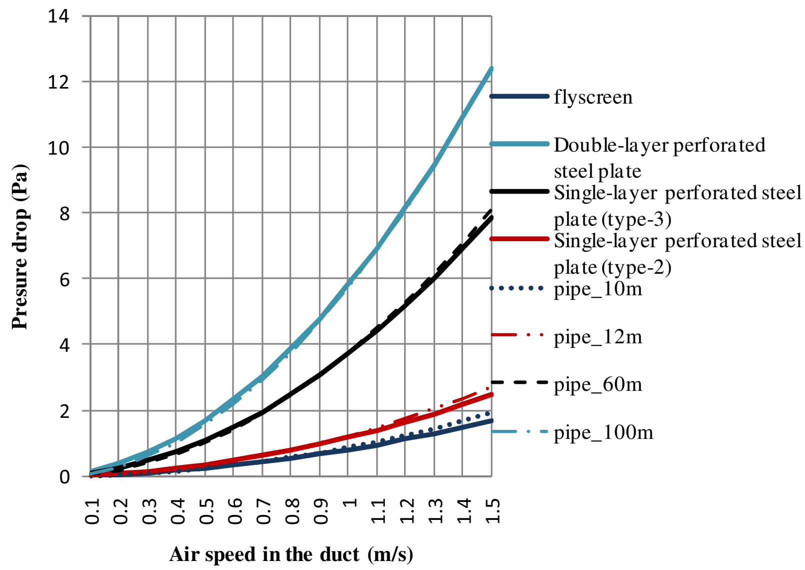

Fig. 5 : pressure drop in the profiles and equavalent EAT tube length for different air speeds in the duct

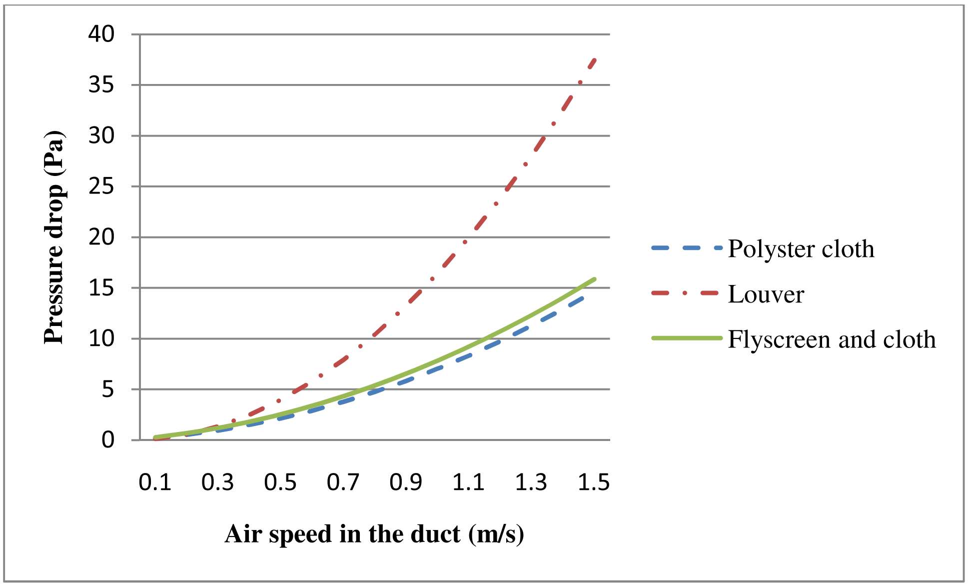

Figure 6: pressure drop in different drift protectors for different air speed in the duct

3.4. Airflow rate in Earth air tunnel

Fig. 7 shows the airflow rates in different EATs for different wind speeds, and same inlet conditions while the mister is running. Increasing the length of the EAT from 10 m to 100 m, results in only 34% reduction in airflow rate for typical wind speed of 3.2 m/s. With reasonably long EATS having large cross section, significant amount of pre-cooling can be obtained from wind. For EAT of 60m length, airflow rate of 172 l/s can be obtained with the corresponding wind speed of 3.2 m/s. This translates into 3 air changes per hour for a 70 m^2 building with ceiling height of 3 m.

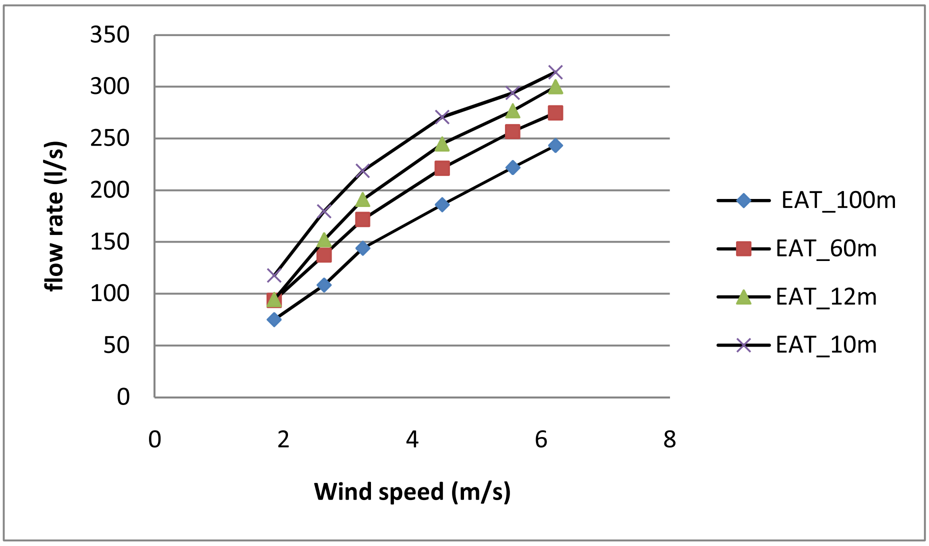

Fig. 7: Airflow rate in EATs simulated for different wind speed at 10 m height

3.5. Evaporator

Accounting for the EATs pre-cooling effect, the inlet condition to the evaporator section was set at moderate temperatures of 29 °C, and 25 °C to characterize the effect of mist evaporation. South Australia is a good example of a dry and hot summer climate with the annual ground temperature at 2 m depth varying from 14-22 °C (Juan & Baggs 2009). Hence a significant pre-cooing in the EATs could be expected from such locations. Tab. 2 provides the test results for 100 m EAT for different wind speeds. The air temperature drops by about 10 °C for all wind speeds in the evaporator section. Fig. 8 shows the saturation efficiency of the misting evaporation and the corresponding cooling load offset based on room comfort temperature of 26 °C for inlet air temperatures of 29 °C and 25 °C. The effectiveness of the evaporator is high at moderate flow rates ranging from 140-200 l/s. The average effectiveness of the evaporator for the higher inlet temperature is 75 % and for inlet temperature of 25 °C, the average effectiveness is found to be 80 %. For a typical wind speed of 3.2 m/s and pre-cooled air temperature of 29 °C, the cooling load which can be offset is about 1.4 kW based on the experimental conditions, which is sufficient for 20 m^2 building space with design load of 70 W/m^2. In general, the misting evaporation performed better in low flow rates which can easily be achieved for moderate wind speeds attainable on summer day in most dry and hot regions.

Tab. 2: Evaporation test results of the simulated EAT of 100m for different wind speeds

Wind speed (m/s)	Duct inlet temperature (°C)	Duct inlet RH (%)	Duct outlet temperature (°C)	Temperature at window (°C)	Room Temperature (°C)	Window RH (%)	Volume flow rate (l/s)
1.9	28.9	17.5	18.1	19.0	19.1	69.5	75.0
2.6	29.1	18.1	17.6	19.4	19.5	68.6	108.3
3.2	29.0	18.4	17.7	19.6	20.0	64.3	144.0
4.5	28.9	18.7	18.1	20.3	20.3	58.7	186.0
5.6	28.9	19.1	18.6	20.9	21.1	55.4	222.0
6.2	28.7	19.2	18.8	21.2	21.7	53.7	243.3

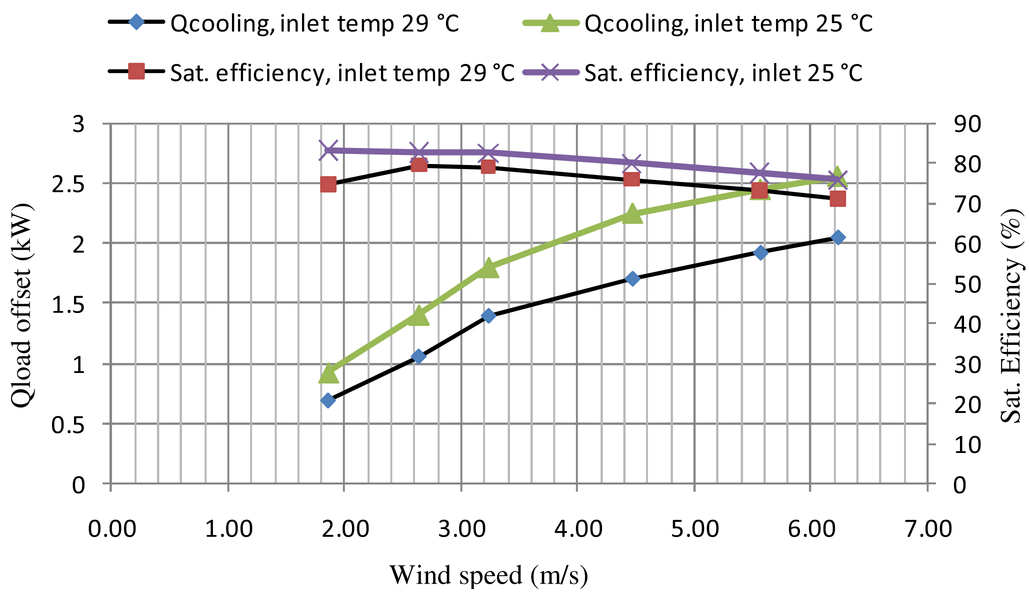

Fig. 8: The cooling performances of the misting evaporation for different wind speeds using 100m EAT as a pre-cooler

4. Application case study

A 50 m^2 school hall in Adelaide is being considered for application of this cooling system. The design load based on AIRAH (2007) is 96 W/m^2, this includes the ventilation load based on minimum outside air requirement. As EAT provides 100% outside air, the design load should be less than the stated value. For a well designed building it is estimated to be 50 W/m^2 based on comfort temp of 26 ^{o}C. Adelaide's design summer condition of 37 ^{o}C dry bulb and 21.4 ^{o}C wet bulb temperature (AIRAH 2007) were used.

Considering a 60 m long, 0.5 m diameter polyethylene pipe buried at 2 m depth with its inlet connected to a wind catcher. Based on fig. 6, we may have airflow rate of 200 l/s at the buried pipe for the corresponding wind velocity of 4m/s which is a typical day time wind speed in summer. In summer the average ground temperature at 2m depth is 19 ^{o}C (Juan & Baggs 2009). A sandy-clay soil with thermal conductivity of 1.3 W/m.K and thermal diffusivity of 0.75x10-7 m^2/s is assumed for the ground. Using the EAT mathematical model of Lee & Strand (2006), the outlet pre-cooled air temperature from the EAT is found to be 25 ^{o}C while the wet bulb temperature of the incoming air is reduced to 17.5 ^{o}C. The average effectiveness of the misting evaporation obtained from the experiment for low temperature is 80%. Thus the air temperature can be further cooled in the evaporator to a temperature of 19 ^{o}C. If we assume a thermal comfort temperature of 26 ^{o}C, the overall cooling load removed by the system would be 1.7 kW, which is about 70% of the design load. If the number of EAT tubes is increased to 2, then the system will remove the entire cooling load at the design summer conditions

5. Conclusion

The experimental results have demonstrated that significant cooling can be achieved with the integrated passive system of wind, EAT, and low pressure drop evaporator. Wind can generate sufficient airflows in long EATs with large cross section and hence the combined system provides significant amount of ventilation and pre-cooled air. Main pressure low flow rate misters can be used by preventing the carryover for direct evaporation with better effectiveness even at low to moderate airflow rates. The pressure drop in the system is very low and it is suitable for naturally driven ventilation. This passive system can be a substitute to an energy intensive mechanically driven refrigeration air conditioner for summer cooling and ventilation in locations with relatively dry and hot summers.

6. References

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