Experimental Verification of Indirect Adiabatic Cooling by Ventilation Air

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Abstract. Current efforts to reduce the energy consumption of buildings leads to savings in all categories of their operation. A traditional energy saving tool is low-energy cooling, based on the alternation of day/night temperatures, cold storage, or the use of high-temperature cold sources such as Earth's semi-solid or groundwater. This paper deals with the less widespread method for cooling buildings, based on latent heat removal for converting water to vapour.

Within three years, approximately 150 operating states were measured for different temperatures and airflows and various types and numbers of humidifiers, as well as nozzle pressures and types of heat exchangers. The paper outlines the results of these experiments and shows the available cooling capacity coupled to the ventilation air. The fact is that cooling power increases with increasing outdoor temperature, which makes indirect evaporative cooling an interesting method of providing cold during the times when it is under greatest demand.

1. Introduction

Cooling is an important issue for contemporary buildings, and its importance is growing with the increasing quality of the building envelope, as well as indoor heat gains due to the increasing amount of electronic equipment, and the tightening of requirements on indoor environments and climatic change [1]. Cooling devices came into the market in the 1930s as rare luxury devices, which first became affordable in 1947 [1]. In 2015, they represented a rapidly developing industry sector, approaching a total annual turnover of close to 100 billion dollars, and still growing.

The cooling of buildings represents a considerable percentage of the total energy consumption in the world. The world annual energy consumption for cooling in 2010 was close to 1.25 PWh. More than 45 % of that energy was consumed by commercial buildings. The future average cooling energy demand for commercial buildings is calculated to rise 275 % by the year 2050 [2], despite efforts to achieve energy efficient buildings [3].

However, unlike heating, cooling can often be provided by natural and economic systems of low-energy cooling techniques, with no compressor-based sources. Most of those techniques are based on the alternation of day/night temperatures, temperature differences between outdoor and indoor air, and high temperature sources of natural coolness, such as ground or water mass. All of these techniques have their own individual limitations, especially regarding available power or total amount of available coolness [4].

One of the most promising methods is Indirect Evaporative Cooling (IEC), based on sensible heat transfer into latent heat fraction by the evaporation of water. Rather than Direct Evaporative Cooling
(DEC), which increases the moisture content of cooled air, IEC uses secondary air stream (working air) which is cooled by evaporation and humidified, to cool the primary air stream on a heat exchanger, with no moisture exchange.

This principle has been known since 1930 and it was already known then that it consumes less energy than other cooling methods [5]. That technology has been pushed into the background by a more powerful but more demanding compressor cycle.

In the 21st century, the world is looking for technical solutions for near-zero energy buildings and IEC has returned to the focus of many people. It has been proven that, between 2000 and 2016, IEC provided extremely cheap cooling, but always above the wet-bulb temperature, and that the best climatic conditions for this principle are a hot and dry climate [6]–[10]. IEC is also well able to precool outdoor air in humid tropical climates before traditional cooling [11].

Later on, the novel construction of heat exchangers was developed, allowing indirect evaporative cooling to sub-wet bulb temperatures, called M-Cycle [12]–[14]. These special exchangers are theoretically able to cool fresh air to dew-point temperatures by evaporation, but the same size of exchanger provides less cooling power than the usual IEC [15][16]. Even now, in 2018, the IEC finds it difficult to work its way into the market. One of the barriers to market expansion is the difficulty of predicting cooling performance.

IEC cooling performance of is a result of extremely complex simultaneous heat and mass transfer in the exchanger. Sensible heat is transferred between two streams, while sensible and latent heat is simultaneously transferred within one of those non-homogeneous streams, which makes that action difficult to resolve and calculate. In the past, multiple models were created to describe that phenomenon.

Erns and Dreyer found the first actual mathematical model in 1991 [17]. Their model was based on the finite volumes method and assumed that water is recirculated and evenly distributed over the plate surface and that the air/water interface is approximately equal to the plate area. Those assumptions are impossible to meet under actual circumstances.

An interesting model was presented by G. Heidarinejad in 2015 [18], taking into account longitudinal heat conduction along the plate. In the same year, S. Lowrey showed a model of IEC [19] that described energy fluxes within the stream. But the Nusselt number used for actual heat transfer coefficients was drawn from very old publications.

In 2016, Y. Wan presented a very abstract model, where the heat transfer coefficient was calculated for several cases via CFD simulation [20]. Anisimov also offered a detailed description of heat and mass transfer processes on crossflow wetted exchangers [21]. Similar model were shown by Chua in 2016 [22], that included heat and mass transfer coefficients.

All the above models are very complex and rely on finite elements, finite differences or finite volume methods. Their use is possible for scientists, but not for field engineers in the industry. In common technical practice, engineers need proper recommendations and simple equations to determine the cooling performance of devices.

A step in this direction was published by Kim [23], but his regress model is only supported by calculated, non-experimental data and also carries some assumptions and boundary conditions that are unacceptable. Finding equations and recommendations usable in the industrial field are the objectives of the authors of this paper.

To cover the large uncertainties of the calculated data, experimental data must be used. This paper presents the authors’ acquired experimental dataset on actual-size air-handling units, and baselines for conducting simple descriptions of the IEC process.

2. Methods

To obtain such experimental data, an air-handling unit with heat recovery was installed in the lab and equipped with a water humidification chamber. Nominal air flow rate was 1000 m³/h in both fresh air and wasted air channels. Two fans were equipped with inverters to adjust proper volumetric flow rates. The unit was connected via a special plug to an open window in order to operate with outdoor air. The picture and scheme of the unit is seen in Figure 1 and Figure 2.
The unit was equipped with a large number of laboratory sensors for temperature, relative humidity, and air velocity to capture the state and volumetric flow rates of the air streams in each of four air-ducts marked as follows: ODA (Outdoor air), SUP (Supply air), ETA (Extract air) and EHA (Exhaust air).

More temperature sensors were directly installed between the plates of heat exchanger. In addition, the temperature, pressure and flowrate of the water sprayed into the humidification chamber was measured. Therefore, all main inputs and outputs of the machine were controlled.

To acquire cooling performance data including sensitivity to practical boundary conditions, further operational states were considered and measured. Additional types of water-spray nozzles were used, including their numbers and water pressures. Two types of heat exchanger were tested, under two variable volumetric flow-rates of air. The list of test setups may be seen in Table 1.

Combinations of all these parameters lead to many operational states, while in each of them there was an effort to measure at least twice, under two different outdoor air temperatures. For this purpose, an auxiliary heater was installed in ODA to increase temperature in this channel. Each operational state was measured in steady state, after 20–50 minute periods of stabilisation. Steady state parameters were
calculated at 10 minute averages. Over a three year period, 208 measurements was completed, excluding test operations prior to measurement.

Table 1. Experimental setups of measurements.

| Heat recovery exchanger | plates spacing 6 mm |
|-------------------------|---------------------|
|                         | plates spacing 2.5 mm |
| Volumetric air-flow rate | 750 m$^3$/h |
|                         | 1000 m$^3$/h |
| Nozzle type             | Hollow cone, axial nozzle 214.184 |
|                         | Full cone, axial nozzle 490.406 |
| Number of nozzles       | 1–9, step 1 piece (Hollow cone) |
|                         | 1–6, step 1 piece (Full cone) |
| Water pressure at nozzles | 100–600 kPa, step 100 kPa |

3. Results
During each measurement, stated variables of moist air were measured on both primary and working air streams. To assess the validity of results, energy income to working air and effective cooling of primary air was calculated and compared. Measurement was considered valid when the difference between energy fluxes computed on primary and working air were below 20% of this flux. This comparison may be seen in Figure 3.

In small energy fluxes, only a minority of measurements met the precision requirement. But with measurements of heat transfers higher than 2.5 kW, the precision was sufficient. For further evaluation, data from 2016 was excluded due to a methodological error in their measurement. From the remainder of the dataset, 75 measurements with proven relevance were chosen for evaluation.
For the purpose of cooling, the temperature of fresh air inlet to the ventilated space (SUP) is the most important parameter. Measured temperature is presented in Figure 4, related to outdoor air temperatures, types of exchangers and nozzles.

Resulting temperatures are also strongly related to the inlet temperatures of working air $t_{ETA}$, which is not represented in the picture. However, this temperature varied within a range 24–28 °C for over 90 % of the time.

It can be concluded that the relation to outdoor air temperature is not very strong, while primary air can be cooled to 16 °C from an initial temperature of 21 °C, and to 19 °C from an initial temperature of 40 °C.

Furthermore, we can confirm that the size of the heat exchange surface has a significant effect on the final temperature. Heat exchanger with 6 mm plate spacing have approximately a 42 % heat exchange surface compared to exchanger with 2.5 mm spacing, but approximately 60 % temperature effectiveness of the 2.5 mm exchanger. The heat flow density is higher, but a temperature of 18 °C in the case of 2.55 exchanger spacing is far more usable, than temperatures higher than 22 °C for the latter.

![Figure 4: Supply air temperature in relation to exchanger type, nozzle type and outdoor air temperature.](image)

![Figure 5: Normalised cooling power in kJ of removed heat per kg of air, in relation to exchanger type, nozzle type and outdoor air temperature.](image)

The lowest temperature $t_{SUP}$ is reached while lowest initial temperature $t_{ODA}$ was measured. In fact, this is the case when cooling demand is lowest. However, overall cooling power is also lowest, because of the lower differences between $t_{ODA}$ and $t_{SUP}$. Slight variations are observed in higher outdoor temperatures. This can be caused by the fact that temperature scatter-plots do not contain information concerning the volumetric flow rates of air.

In Figure 5, normalized cooling power is shown, in kJ of heat removed from each kilogram of primary air. It is obvious that cooling power increases with outdoor air temperatures and cooling demand. Nozzles with hollow cones appeared to be a more promising than nozzles with full cones. This is probably caused by the fact that, for the same water flow rate, more nozzles or higher pressures are needed in the case of hollow cones, so that water distribution is more even and drops are smaller.

As is shown in Figure 4 and Figure 5, both the supply air temperature and normalized cooling power are important, but the best results are on the opposite sides of working areas. Another quality quantifier is to be found, to describe overall efficiency of the process. Anisimov [24] offered a function of wet-bulb efficiency, that takes into account the lowest possible supply air temperature and the fact that overall cooling power is greater with rising outdoor temperature. Wet-bulb efficiency (1) is expressed
as the ratio between actual temperature differences in the primary air, and theoretical maximum temperature difference on that stream:

\[ e_{wb} = \frac{t_{ODA} - t_{SUP}}{t_{ODA} - t_{WB,ETA} - t_{WB,ETA}} \times 100 \quad [\%] \quad (1) \]

Focused on the most promising combination of exchanger and nozzle type, wet-bulb efficiency was calculated and related to water flow rate. As is shown in Figure 6, very good efficiency of over 80 % can be reached with relative amounts of water from 0.025 litre sprayed per cubic meter of air. Lesser amounts of water lead to significant decreases of efficiency which, even with no moisture added, cannot decrease to zero because of remaining sensible heat transfer.

Around a water flow rate of 0.04 l/m³, the same amount of water supplied by fewer nozzles with higher pressure can be observed, leading to far higher efficiency than when using more nozzles with lower pressure. In objective practice, it appears important to operate nozzles at the highest possible pressure and, by using fewer nozzles, set a normalized water flow rate between 0.03 and 0.09 l/m³.

In Figure 7 the same data are shown, but related to outdoor air temperature. It can be considered as proven that there is no clear relationship between wet-bulb efficiency and outdoor air temperature.

![Figure 6: Wet-bulb efficiency of cooling on 2.5 mm spacing plate exchanger and hollow cone nozzles, related to water flow rate and water pressure.](image1)

![Figure 7: Wet-bulb efficiency of cooling on 2.5 mm spacing plate exchanger and hollow cone nozzles, related to outdoor air temperature and water pressure.](image2)

4. Conclusion
In this paper the experimental verification of the process of indirect evaporative cooling was analyzed for the final purpose of building an equation predicting the cooling power of IEC.

It has been proven that cooling power of indirect evaporative cooling increases with outdoor air temperature, as well as usually increasing the cooling load. With proper type nozzles and heat exchangers, a 16–19 °C supply air temperature is reachable when outdoor air temperatures vary from 21 °C to 36 °C.

For the IEC, the same qualities of heat exchangers are required for its winter operation: large heat transfer surfaces and good heat transfer coefficients. As a good indicator of overall system efficiency, wet-bulb efficiency appeared to best express the degree of approximation of supply air temperature to the theoretical minimum.

Experiment significance is considered as sufficient, so the IEC is confirmed as a very good way to reduce energy consumption in buildings. IEC is not suitable for removing relatively large heat loads from the interior because of low temperature differences and low heat capacity of the air, which would result in the need to change large quantities of air. Cooling power has proven to be sufficient to remove heat load caused by ventilation, and even some additional heat loads.

It is relevant that the cooling power obtained by the evaporation of water is extremely cheap, especially when service-water is used. Wet/bulb efficiency higher than 80 % is reached with water
consumption around 0.03 liters of water per cubic meter of air. It is to mention, that only small part of water is evaporated, and the rest can be recirculated, if construction of the apparatus allows it. Due to the fact, that IEC provides cooled air not only to interior, but also to the exterior and also that IEC adds moisture content to the outer atmosphere where our cities seriously need it, IEC appears to be an excellent low-energy cooling technique for contemporary buildings.

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