PERFORMANCE EVALUATION OF INDIRECT EVAPORATIVE COOLING BY MEANS OF MEASUREMENTS AND DYNAMIC SIMULATIONS

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ABSTRACT

In an indirect evaporative cooling (IEC) installation the extracted air is cooled by means of adiabatic humidification. By passing over an air/air heat exchanger this air cools down the supply air. A clear interaction can be observed between the relative humidity of the extracted air and the thermal comfort realized in the building. To be able to predict the performances of this technique in an accurate way, a good knowledge of the indoor relative humidity is thus important.

This paper presents the results of measurements carried out in the summer of 2006 in a non-residential building at the Belgian coast which makes use of indirect evaporative cooling. An evaluation of the indoor summer comfort is made and the interaction between the thermal performance and the indoor humidity is investigated. Furthermore, dynamic simulations based on the system’s effectiveness were performed using the multizone building simulation program TRNSYS, in order to evaluate different parameters affecting the performance of IEC.

KEYWORDS
Indirect evaporative cooling, indoor humidity, thermal comfort, effectiveness, dynamic simulations

INTRODUCTION

The indoor relative humidity is influenced by moisture sources e.g. people, cooking …, ventilation with outdoor air, infiltration and the exchange of moisture with walls and furniture. Hygroscopic materials such as wood and textile are able to dampen out relative humidity variations (Simonson et al. 2002, Svennberg et al. 2004). Nevertheless the uncertainty of many parameters such as material data, available surface… makes it difficult to assess the humidity variations well. As a result most building simulation programs, e.g. TRNSYS, predict the relative humidity in a simplified way (SEL et al. 2004, Janssens and De Paepe 2005).

Indirect evaporative cooling is an interesting passive cooling method in which a clear interaction can be observed between the indoor humidity and the thermal performance of the technique. In the installation the return air passes over the wet side of an air/air heat exchanger (Figure 1). By adiabatic humidification the air stream is cooled. At the same time fresh air flows over the other side of the heat exchanger and is cooled down (Figure 2). The technique differs from direct evaporative cooling in which the supply air stream is directly humidified. By humidifying the return air the dry bulb temperature of the supply air can be lowered without increasing its humidity ratio. In this way a more comfortable indoor climate can be obtained.

Typically, an IEC installation can operate at different stages. When the outdoor air is cold enough in summer, the air flow rate is increased and the air is used for free cooling. As soon as the outdoor air temperature is too high, the fresh air is adiabatically cooled by moistening the heat exchanger. If the desired indoor temperature is not yet reached, active cooling may contribute to lower the temperature.

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of the supply air. In winter the heat exchanger can be used for heat recovery thus preheating the outdoor air. Because of its moderate climate the Belgian climate is suitable for the use of this technique on condition that the building cooling loads are also moderate.

Figure 1  Indirect evaporative cooling       Figure 2  Physical state of airstreams in IEC

This paper focuses on the interaction between the thermal performance and the indoor moisture balance. The first part discusses the measured performance of an existing installation and uses the results to establish a correlation between the indoor humidity and the cooling performance. In the second part this correlation is used to perform dynamic simulations evaluating various parameters influencing the thermal performance of indirect evaporative cooling.

MEASURED PERFORMANCE OF IEC

Measurements were carried out in the cafeteria of a holiday centre located at the Belgian Coast (Oostende) in the summer of 2006. Temperature and relative humidity were measured in the installation at all stages of the process as well as in the cafeteria. Also the position of the valves, pumps and fans were registered. The cross flow heat exchanger in the installation has a total width of 900mm, a length of 1950mm and is made out of polypropylene. Both the supply and return fan can work in two stages with a maximum air flow rate of 7100m³/h.

Thermal summer comfort

Figure 3  Measurements of IEC in Oostende (18/07/06 - 22/07/06)
Figure 3 presents the measured temperatures from 18 until 22 July 2006. Measurements were carried out every five minutes. The periods in which the IEC is working are indicated in figure 3 while figure 1 shows where the temperatures of the airstreams are measured.

During the first two days the temperature of the supply air downstream from the heat exchanger section did not differ much from that of the outdoor air. The heat exchanger was clearly bypassed to allow free cooling. The temperature measured behind the evaporator was always lower than before, showing active cooling has been working during the whole measured period. The outdoor temperature varied from 20°C to 35°C during the measuring period. The average temperature measured in the cafeteria was 26°C. Making use of evaporative cooling, a maximum temperature drop of 13.9°C was noticed on 19/07 when the outdoor temperature reached 34.8°C. On average a cooling of 3.3°C could be achieved with this method within the measuring period. At night free cooling was used for some hours to cool down the cafeteria.

The indoor summer comfort was evaluated using the Adaptive Temperature Limits Indicator (ATG) which takes into account the thermal adaptation of occupants to the indoor climate (van der Linden 2006). The method differs between alpha and beta-buildings. The evaluation of the beta-building is more severe, and assumes occupants cannot control the indoor climate e.g. by opening windows. Furthermore, thermal comfort is divided into three levels. It is required that buildings meet the standard level B, corresponding to 80% thermal acceptability, to have a good indoor comfort.

![Image](image.png)

Figure 4  Thermal comfort in July using the Atg evaluation method (beta–building)

The evaluation of the indoor climate in July 2006 is given in figure 4. On the horizontal axis the running mean outdoor temperature $T_{e,ref}$ is presented, which is a weighted average of the outdoor temperature from the current day and that of the three preceding days. The vertical axis shows the indoor operative temperature $T_{i,o}$. In the graphs only the occupancy hours are taken into account (08-22h). During 79.3% of the occupancy hours in July the indoor climate was situated in class B or better. We can conclude that although active cooling was working most of the time, the cafeteria did not always meet the criterion for good thermal comfort.

Regarding humidity comfort criteria the indoor air humidity should be, according to recommendations in EN 13779 between 30 and 70% (CEN 2004). Measurements have shown that this requirement was satisfied in the cafeteria during the whole measuring period.

Hygrothermal interaction

The relative humidity of the return air plays an important role in the performance of IEC as it is moistened in the installation. From the measurement data we derived a linear correlation between the
amount of cooling of the supply air and the difference between the dry bulb temperature of the supply air entering and the wet bulb temperature of the return air leaving the heat exchanger (Figure 5). The smaller this latter temperature difference, the less the fresh air can be cooled because saturation is reached faster when humidifying the return air. This can as also be seen in Figure 2.

\[ y = 0.825x - 0.5338 \]
\[ R^2 = 0.996 \]

Figure 5  Correlation between the thermal performance and the indoor humidity
(Oostende, 18/07/06-22/07/06)

The effectiveness \( \varepsilon \) of an IEC system describes its thermal performance and is given by:

\[ \varepsilon = \frac{\theta_{s,1} - \theta_{r,1}'}{\theta_{s,1} - \theta_{r,1}'} \quad (1) \]

The effectiveness thus compares the actual temperature difference realized by IEC with the maximum possible difference. Calculated from the measured data the effectiveness of the installation in Oostende is on average 82.5 %. The constant value shows that the performance of an IEC installation is independent on the conditions of temperature and relative humidity, of both outdoor air and return air.

It should be noted that the measured points on figure 5 do not intersect the origin: a small available temperature change \( \theta_{s,1} - \theta_{r,1}' \) will not be perceived by the supply air flow. This can be explained by the fact that a small amount of energy is used to heat up the cold sprayed water to the average temperature in the heat exchanger. Further research on the importance of this phenomenon will be undertaken.

**DYNAMIC SIMULATIONS**

**Simulation model**

The definition of the effectiveness can be used as a first step in studying the performance of an IEC system. As the performance of the installation depends on wet bulb temperature of the return air, the indoor temperature realized using IEC is defined both by the room heat and moisture balance. Therefore in TRNSYS (SEL et al. 2004) a simple model assuming a constant effectiveness was built, which calculates the temperature of the supply air using Eq. (1) in every time step. The model has the advantage to be able to predict the indoor thermal comfort, without the need of numerical models that
require a large calculation time.

As a case study, a small office model was designed. The office has a floor surface of 15m² and a height of 2.8m with one window of 3m² facing west, being the most disadvantageous orientation. No shading is present. The office is occupied during the office hours (08-16h) by one person having a sensible heat production of 65W and a moisture production of 0.07kg/h (Harriman et al. 2001). Apart from these, internal gains due to one pc (140W) and lights (10W/m²) are introduced. The office is built up from a traditional heavy construction. Only the west oriented wall is an external wall, all the other boundaries are assumed to be adiabatic. For the floors and ceilings only the layers that are most important for the storage and conductance of heat, are modelled. There is no lowered ceiling or raised floor taken into account. Due to symmetry, only half of the construction is modelled for the interior walls (Table 1). Moisture buffering in the plastered walls and ceilings and in some basic furniture was taken into account using the lumped model in TRNSYS, assuming the capacity of the room was 5 times the capacity of the indoor air.

In the office model also a control system was introduced:
Day control: Heating set point 21°C (07-17h) + hygienic ventilation air change rate (07-18h)
Cooling set point IEC 25°C (07-18h) until \( \theta_i < 22°C \) + cooling air change rate
Night control: Heating set point 16°C (17-07h)

The airflow rate for cooling is derived from cooling load calculations (VDI 1996) and the hygienic ventilation rate is calculated for one person to be 36m³/h to meet the minimal ventilation requirements (IDA 2) in non-residential buildings (CEN 2004). Simulations were run for an entire year using a 15-minute time step. The outdoor climate from Uccle was applied for which temperatures have a return period of ten years (Meteotest 2003). The summer period extends from 01May-30Sept. To include all days, weekends are not taken into account.

![Figure 6  Scheme of the simulation model and office model in Trnsys](image)

<table>
<thead>
<tr>
<th>Construction</th>
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<tbody>
<tr>
<td>Exterior wall (heavy)</td>
<td>Exterior bricks 0.09m</td>
<td>Exterior wall (light)</td>
<td>Aluminium 0.005m</td>
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<td>EPS 0.08m</td>
<td>Plasterboard 0.1m</td>
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<td>Plaster 0.01m</td>
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<tr>
<td>Interior wall (heavy)</td>
<td>Plaster 0.01m</td>
<td>Ceiling (Heavy) concrete 0.10m</td>
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<tr>
<td>Brick 0.09m</td>
<td>Plaster 0.01m</td>
<td>Plaster 0.01m</td>
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<tr>
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<td>Window</td>
<td>4/16/4 Ar - ( U_{\text{window}}) 2.27W/m²K</td>
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<tr>
<td>Light concrete 0.07m</td>
<td>(Heavy) concrete 0.10m</td>
<td></td>
<td>( U_{\text{glass}}) 1.4W/m²K</td>
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<tr>
<td>(Heavy) concrete 0.10m</td>
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Table 1  Construction details
Results and discussion
In the dynamic model the IEC installation was assumed to have a constant effectiveness of 85%. First, simulations were carried out in which we assumed that beside the IEC there was no active cooling present. From these results the percentage of the office hours during which an indoor comfort class B or better is established making use of IEC, could be derived. Furthermore it showed the percentage of time during which the IEC mode is working. Secondly simulations were performed in which additionally an active cooling was working during the office hours. The setpoint of the active cooling installation was 26°C. Evaluating these results it is possible to calculate the yearly cooling demand $Q_c$ that should be delivered by the active cooling system.

As the heat exchanger can be bypassed in the installation allowing free cooling with outdoor air (Figure 1), the cooling demand that is saved in the active cooling system using IEC in comparison to free cooling can be calculated:

$$Q_{c,\text{saved}} = Q_{c,\text{Free Cooling}} - Q_{c,\text{IEC}}$$

As a first parameter the cooling ventilation rate is varied (Figure 7a). The cooling air change rate under given loads was calculated to be 9ach. High air flow rates cause more moisture to be removed from the room providing a lower supply temperature and therefore a more comfortable indoor climate: using 9ach as the cooling ventilation rate, the indoor climate is situated in class B or better during 97% of the time. Lowering this value to respectively 7ach and 4ach, this value drops with respectively 5% and 36%.

![Figure 7 Influence of (a) cooling air change rate, (b) indoor moisture production (ach7) and moisture buffering capacity (ach7) in respectively a heavy (c) and light (d) construction type, on the thermal performance of IEC.](image)

Furthermore from the results it was found that there is an optimum air change rate for which the active cooling demand is the smallest and the largest savings can be expected: due to the applied control
system, using larger air change rates, the office is cooled faster which makes the system to switch off sooner. As a result the indoor temperature rises again in the next time step. In case 7ach are applied, 83% can be saved, while the active cooling system would have to deliver 14kWh/year to further cool the indoor temperature to 26°C during all occupancy hours.

Increasing the indoor moisture production has a large influence on the room’s moisture balance and therefore on the thermal performance of IEC. Apart from the gains of one person, being 0.07kg/h at light activities e.g. writing, other gains from bathing, washing … can be introduced (e.g. in health care). With higher indoor humidity, the supply air can be cooled less, resulting in a less comfortable indoor climate (Figure 2b). In case 1.5kg/h is produced instead of 0.07kg/h, the amount of comfort hours drops from 92% to 71% and the time during which the IEC installation is working increases with 8%. The cooling demand that can be saved using IEC lowers with more than half, from 83% to 35%, while the energy that should be delivered by the active cooling is multiplied by four.

It is stated before by previous authors that moisture buffering is able to contribute to a more comfortable indoor climate since it dampens out humidity variations (Simonson et al. 2002). Nevertheless figure 7c shows that changing the room’s moisture buffering capacity has no influence on the system’s performance. Obviously this can be explained by the fact that moisture buffering does not have a large effect on the average relative humidity. This was also stated in previous research (Svennberg et al. 2004). The same conclusion can be drawn for the influence of moisture buffering on IEC in a light construction office. Compared to a more heavy construction, the amount of comfort hours decreases here with 5% (Figure 7d). The saved cooling demand applying this technique in a light construction building is approximately 70%, which is 13% lower than in case of a more heavy construction building. At the other hand, the IEC installation will be active during 5% of the time more. Construction details can be found in Table 1.

CONCLUSION

Indirect evaporative cooling is an interesting passive cooling technique in which the thermal performance depends mainly on the indoor humidity. Measurements carried out in July 2006 in a non-residential building show that by using this technique the supply temperature can be cooled up to 14°C during warm periods. Despite the fact that active cooling was present to further cool the supply air, an evaluation by the adaptive temperature limits indicator showed that a good indoor thermal comfort was not always satisfied during the whole measured period.

The evaluation of the measured data shows a linear correlation between the cooling amount of the supply air and the maximum cooling which is possible using IEC. Dynamic simulations based on this constant effectiveness affirm that the knowledge of the indoor moisture balance is important to predict the performance of indirect evaporative cooling in an accurate way.

The indoor moisture production plays the most important role in lowering the indoor temperature, higher gains will lead to higher supply temperatures and therefore to a less comfortable climate. The amount of moisture buffering material on the other hand rather influences the stability of the climate and does not have a large effect on the performances of IEC. Higher ventilation rates are favourable to help remove moisture and therefore increase the indoor comfort. Due to the reduced thermal mass, the number of comfort hours will be less in case of a light construction type.
NOMENCLATURE

\( \varepsilon \) Effectiveness of indirect evaporative cooling installation [%]

\( \theta \) Temperature [°C]

\( Q_c \) Cooling demand [kWh]

\( h_v \) Latent heat of evaporation [2.5 \( 10^6 \) J/kg]

\( n \) Air change rate [h⁻¹]

\( \xi \) Vapour capacity of air [6.1 \( 10^{-6} \) kg/kg/Pa]

\( V \) Volume [m³]

\( p \) Vapour pressure [Pa]

\( \rho_a \) Density air [kg/m³]

\( RH \) Relative humidity [%]

Subscripts

s supply air

r return air

1 air stream entering heat exchanger

2 air stream leaving heat exchanger

i indoor

e outdoor

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REFERENCES


