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Evaporative cooling systems to improve internal comfort in industrial buildings

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Abstract

Several studies were carried out to determine how hot or cold environments can affect task performance and can influence productivity. Usually, HVAC plants are exactly designed in order to guarantee comfortable internal conditions inside built environments, but not all kind of buildings are equipped with a heating or cooling plant, like for example, some industrial buildings. These buildings are often characterized by high internal thermal loads. For those buildings the ability of different plant configurations to improve indoor thermal conditions was considered taking into account the influence of several parameters, like weather conditions, internal gains, thermal transmittance, ventilation air flow rate, etc. Simulation results are compared in terms of energy savings and thermal comfort.

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Keywords: Evaporative cooling, TRNSYS, industrial building

1. Introduction

Industrial buildings have some features that make them very different than all other types of buildings like dwelling, offices, schools, hospitals, ecc… Firstly, in most of cases, these buildings don't have a heating or cooling system except for office space; secondly, internal heat gains have a huge impact on the energy balance of the building cause of work processes and electrical equipment. Electrical energy consumptions due to production processes, especially in case of high process loads, exceed, by far, heating and cooling energy requirements, that not are therefore the main purpose of energy saving strategies.

Moreover, the occupancy pattern and lighting affect significantly the thermal conditions inside the building.

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The industrial sector is one of the largest consumers of energy. In Europe, the sector used 26% of the total energy consumption in 2011 (Eurostat, 2013), while in the US, this sector consumed 31% in the same year (EIA, 2012) [1].

In last years a few guidelines on the energy saving potential of industrial halls have been published (ASHRAE, 2008; ASHRAE [2], 2011; NREL, 2009a; TargetZero, 2011), but their focus, for halls with higher process loads, was to reducing energy consumption due to production activity because it represents the main component of energy consumptions. Particularly in the "data processing centers" (CED) and in many dairy industries, defined today among the areas with the highest demand for energy for specific use [3,4].

This study deals with industrial halls with high process loads and the aim of the study is the evaluation of the behavior of several air-systems for increase the comfort conditions inside the hall.

In industrial buildings the evaporative cooling can represent a suitable way to make more comfortable the internal environment ether with direct or indirect system. An application of an indirect system for building cooling, based on a passive component, without forced ventilation, is presented and analyzed in [5]. However, evaporative cooling systems, based on mechanical ventilation, can be profitably installed in industrial sheds.

In a previous paper, a study about the ability of an evaporative cooling system to guarantee inside comfort conditions, saving electrical energy, has been carried out for a sample industrial building [6]. In particular, four different configurations have been analyzed: no system, external air ventilation system, external air ventilation system with direct evaporative cooling and external air ventilation system with indirect evaporative cooling. In this paper the behaviour of the same systems by means of TRNSYS simulations has been investigated taking into account the influence of several parameters, like weather conditions, level of internal gains, thermal transmittance of the envelope components. Moreover, in order to evaluate the effect of different systems in reaching internal comfort conditions, the wet bulb globe temperature index has been introduced, according to ISO 7243 [7].

2. Reference industrial building

2.1. Building parameters

An existing typical industrial building is used as the baseline for the present study. The industrial shed is a single floor building with a total gross floor area of 4800 m^2 and 7 m high. The glass area takes up the upper zone of the external wall, but, often, buildings are equipped with skylights, that are a very effective way to provide daylight over a large area of a single-storey building [8]. The characteristic of the building envelope are reported in Table 1.

Table 1. Details and thermal properties of building construction components.

Building component	A [m^2]	U [Wm^{2}K^{-1}]
External walls	1396	0.25 - 1
External doors	240	0.461
Glass surface	576	5.68
Roof/ground surface	4800	0.666 / 1.75

Industrial buildings are built using simple steel or concrete construction methods; in both cases, often, the suitable choice is steel sandwich panels or concrete panels in order to minimize the construction time and therefore construction costs. In order to evaluate the influence of the envelope heat transfer coefficient on the building thermal behaviour, several simulations have been carried out varying the thermal transmittance of the external walls. Comparisons among the results of the simulations has been made using as parameter the Cooling Degree-Hours, Base26 (CDH_26), defined by the following equation:

$$CDH(26°C) = m_k \sum_{k=1}^{n} \left(\vartheta_{ek} - 26\right) \tag{1}$$

The index has been developed similar to the Cooling Degree-Days index proposed by ASHRAE [2] and calculated by the following equation:

$$
\begin{aligned}
&CDD(\vartheta_{tc}) = m_k \sum_{k=1}^{n} \left(\vartheta_{ek} - \vartheta_{tc}\right) \\
&m_k = 1 \quad if \quad \vartheta_{ek} \geq \vartheta_{tc} \\
&m_k = 0 \quad if \quad \vartheta_{ek} < \vartheta_{tc}
\end{aligned}
\tag{2}
$$

where m_k is an dimensionless parameter, $CDD(\vartheta_{tc})$ is the amount of cooling degree days, ϑ_{ek} is the external air temperature [°C] and ϑ_{tc} denotes the threshold temperature for cooling [°C], also referred to as the balance point temperature [9] and defined as the value of the outdoor air temperature at which the total heat loss is equal to the internal and solar gains Φ_g [kWm^{-2}].

2.2. Sensitivity analysis

A first group of simulations deals with the influence of some parameters on the achievement of the internal comfort conditions. The considered parameters are the following: internal gains Φ_g (10; 20; 30; 40; 50; 60), thermal transmittance of the envelope U (0.25; 0.30; 0.35; 0.40; 0.50; 1.00) [Wm^2K^{-1}], and ventilation rate, ACH_{max} (4; 5; 6) [h^{-1}]. For each value of every single parameter, all the other parameters have been changed, carrying out 108 different simulations. Results in terms of *CDH (26)* has been compared.

In particular, it appears that the influence of the thermal transmittance of the building U is very small and it reassures us about uncertainty in knowledge of thermal transmittance of envelope.

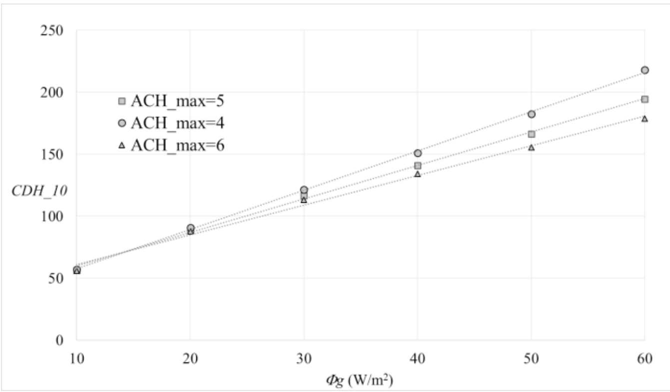

Fig. 1. *CDH (26°C)* VS Internal gains

On the other side, a parameter that strongly affects the internal conditions, is the amount of internal gains Φ_g, as can be seen in Fig. 1, where each line represents a trend line based on all thermal transmittance values for a given air ventilation rate *ACH*. In the same figure also the influence of the maximum rate of ventilation ACH_{max} can be seen; this influence depends on the value of the internal gains, but it is however more limited that the influence of the internal gains level. In order to compare the different plant configurations the following values has been considered: $U = 1$ $Wm^{-2}K^{-1}$; $\Phi_g = 50$ Wm^{-2} and $ACH_{max} = 4$.

3. Simulations

3.1. Simulations parameters

The building energy analysis has been carried out using the transient simulation program TRNSYS, which can be used to assess the performance of thermal and electrical energy systems. Different building locations were considered in order to evaluate the influence of weather conditions on the behaviour of cooling plant.

For the simulations, running with a time step of 6 min, a cooling period from 15th May to 30th September was considered. The following time scheduling for the workers is assumed: 24 hours a day, starting from Monday at 2:00am to Saturday at 2:00am. The infiltration airflow rate was assumed to be a sum of a variable part and a fixed part, set equal to 0.2 air changes per hour *ACH*, which is a suitable value for a not air-conditioned building during the summer period.

In order to allow a correct numerical simulation for TRNSYS the following logical conditions, governing the opening and closing of windows, have been defined: $\vartheta_i > \vartheta_e$ and $\vartheta_e > 14°C$ where ϑ_i is the internal air temperature and ϑ_e is the external air temperature. These conditions are satisfied when the internal environment is particularly warm ($\vartheta_i > \vartheta_e$) and only during the summer period ($\vartheta_e > 14°C$), since during winter period heat losses contribute to mitigate the indoor temperature. The activity level of the occupants is set as "hard work" the associated heat gain being 90W and 95W for sensible and latent loads, per person, respectively.

4. Cooling systems analised

Four different ways to cool the internal environment of an industrial building have been considered and are listed in Table 2. The cooling system by means of natural ventilation is at the same time the less expensive and the less effective way to reduce the internal air temperature.

Table 2. Data for sensitivity analysis.

Case	Infiltration	Mechanical ventilation	Evaporative equipment
Natural infiltration NV	✓	-	-
Mechanical ventilation MV	✓	✓	-
Direct evaporative cooling DEC	✓	✓	Direct
Indirect evaporative cooling IEC	✓	✓	Indirect

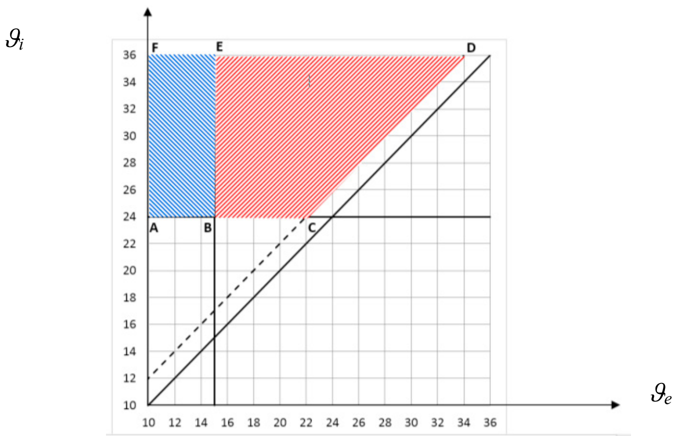

Fig. 2. Needed conditions to activate mechanical ventilation with external air.

In order to evaluate air infiltration rate a new method proposed by Brinks et al. [10] has been used. The method supplies a correlation to evaluate averaged monthly infiltrations e_{month} valid for buildings with a height of 4-20 m and a floor area of 1000-10000 m^2. The monthly coefficient e_{month} can be evaluated by the following equation:

$$e_{month} = b + x_1 \cdot h + x_2 \cdot A + x_3 \cdot \vartheta \quad (3)$$

where b, x_1, x_2, x_3 are reported for every month, h is the mean building height H [m], A [m^2] is the floor area and ϑ [°C] is the inside temperature during operational time.

A mechanical ventilation system based only on external air has been considered and fans equipped with inverter has been simulated. The starting of the fans depends on the following conditions:

- time slot: worktime or stoptime
- thermostat signal: on if $\vartheta_i > 24°C+2°C$; off if $\vartheta_i < 24°C+2°C$
- air external temperature value: $\vartheta_e < \vartheta_i - 2°C$
- state of the other cooling equipments: if direct/indirect evaporative cooling equipment works, ventilation with external air turns off.

The third condition is due to the hypothesis according to which external air increases its temperature when pass through the fan; some empirical studies report a temperature increment equal to 2°C.

Therefore, if the outdoor temperature is two degrees lower than the upper limit of the comfort temperature, then introducing external air is possible. With reference to Fig. 2, all points that are contained in trapezoid ACDF satisfy that condition, but only points that are contained in trapezoid BCDE represent valid conditions for the ventilation with the external air. In fact, there is a threshold value of the external temperature ϑ_t (equal to 15°C) under which external air can't introduce inside the building; this temperature identify a rectangular zone ABEF, that corresponds to too much low values of entering air temperature.

The air ventilation flow rate ranges from a maximum value (4, 5 or 6 *ACH*) to a minimum value corresponding to 30% of maximum. Intermediate values of the flow rate are evaluated using a linear interpolation based on internal air temperature, expressed by the following equation:

$$\dot{m}_x = \dot{m}_{min} + \left(\dot{m}_{max} - \dot{m}_{min}\right) \cdot \frac{\left(\vartheta_x - \vartheta_{min}\right)}{\left(\vartheta_{max} - \vartheta_{min}\right)} \qquad (4)$$

where ϑ_{min} is the temperature value at which the thermostat switch off (22°C), ϑ_{max} is the temperature value over which the air flow rate has to be maximum (26°C), $\dot{m}_x$ is the air flow rate at internal temperature ϑ_x.

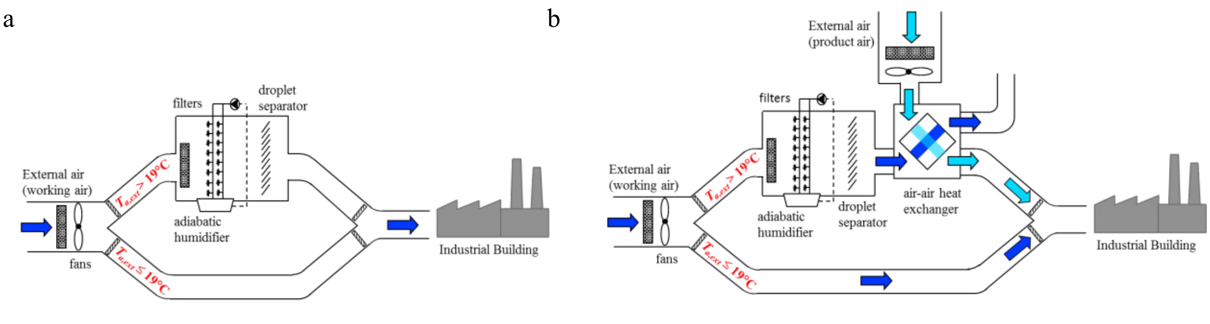

Fig. 3. Sketch of direct (a) and indirect (b) evaporative cooling equipment.

In order to avoid that during night hours the internal temperature increases if the fans are turn off a recirculated rate of airflow, expressed by eq. 5, has been considered; the recirculated air is mixed with external air in order to obtain a suitable inlet temperature of the airflow.

$$\dot{m}_{rec} = \dot{m}_{\min} \frac{\left(\vartheta_t - \vartheta_e\right)}{\left(\vartheta_i - \vartheta_t\right)} \qquad (5)$$

As summarized in Table 3, the support of mechanical ventilation is present in three of the four cases analised. The Direct Evaporative Cooling (DEC) consists of an adiabatic wetting equipment used to lower the temperature of the external ventilation air (Figure 3a). The adiabatic system is supposed to have a wetting efficiency of 90%. The ventilation air flow passes through the DEC apparatus only if the internal air temperature is higher than 25°C and the external air temperature is higher than 19°C.

The starting of the direct cooling equipment depends on the following conditions:

- time slot: worktime or stoptime
- thermostat signal: on if $\vartheta_i > 24°C+2°C$; off if $\vartheta_i < 24°C+2°C$
- air external temperature value: $\vartheta_e > 19°C$
- $\vartheta_{in} < \vartheta_i - 2°C$
- state of the other cooling equipments: if direct/indirect evaporative cooling equipment works, ventilation with external air turns off

In this case the building is equipped with an indirect evaporative cooling system (IEC) instead of a DEC. An external air flow, treated by the evaporative cooler, is used to cool the ventilation air flow by means of an air-air heat exchanger. After passing through filters, external air is saturated within the direct evaporative cooler; then, the working airflows into an air-to-air heat exchanger, which cools the external air without modifying its moisture content. In this way the latter flow is subjected to a sensible cooling. A sketch of this system is represented in Figure 3b and the starting conditions of the indirect cooling equipment are the same one of DEC equipment.

5. Results of the simulations

Different locations has been considered in order to investigate the influence of different weather conditions on the behavior of the considered cooling system. In particular, the following locations has been chosen: Hamburg, Prague, Wien, Udine, Lion and Palermo.

5.1. Thermal comfort conditions

Two indexes proposed by Fanger were considered to evaluate the performance of the different systems in terms of thermal comfort [9]: Predicted Mean Vote (*PMV*) and Predicted Percentage of Dissatisfied (*PPD*). Studies on the comfort of "open.space" environments characterized by large glass surfaces with considerable radiant loads were performed in [11, 12].

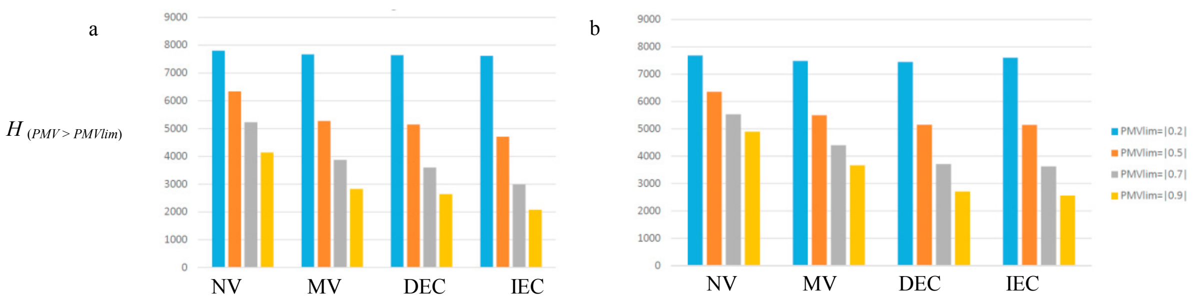

Fig. 4. Hours during which PMV value is greater than PMV limit for different cooling systems considered in Hamburg (a) and in Palermo (b).

The method A (ISO 7730 Annex H [13]) calculates the number or percentage of hours, during which the building is occupied and the *PMV* is out-of a specified range; it is a very simple method, but it does not allow to know how much far from limit is the *PMV* of a room. Another useful method is that called method C. In the method time during which the *PMV* exceeds the comfort boundaries is weighted with a factor w_f which is a function of the *PPD*.

We used the three categories A, B and C, suggested in [13], and a further additional category corresponding to -0,9 < *PMV* < +0,9. The effect of the different cooling systems is summarized in Fig. 4, where e.g., the results for Hamburg and Palermo are reported.

In order to enhance the analysis another comfort index has been evaluated: Wet Bulb Globe Temperature *WBGT*. The index is proposed by UNI EN ISO 7243 to evaluate the hot stress in workplaces. Values suggested by EN ISO 7243 to guarantee comfort conditions for workers depending on metabolic rate M [Wm^2] are reported in Table 3, where different values of *WBGT* are divided into two classes: acclimate and not acclimated persons.

Table 3. Limit values of WBGT varying on metabolic rate.

Metabolic rate class	M [Wm^2]	Reference value of *WBGT* [°C]	
		Acclimated person	Not acclimated person
0	$M \leq 65$	33	32
1	$65 < M \leq 130$	30	29
2	$130 < M \leq 200$	28	26
3	$200 < M \leq 260$	26	23
4	$M \geq 260$	25	20

In Figures 5a and 5b the diagram the values of WBGT obtained for Hamburg and Palermo in cases NV, MV, DEC and IEC during the second week of July are reported. It is clearly shown as WBGT values relative to the IEC system are always the lowest.

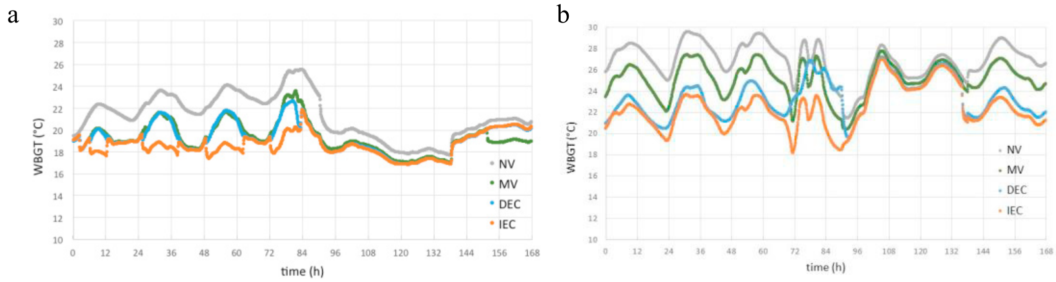

Fig. 5. WBGT index during the second week of July for different cooling systems considered, in Hamburg (a) and in Palermo (b).

5.2. Energy saving

Concerning electrical energy saving that different cooling systems allow, a comparison was carried out, in terms of electrical consumption, between a plant with fan-coils coupled with a chiller and a plant with evaporative cooler.

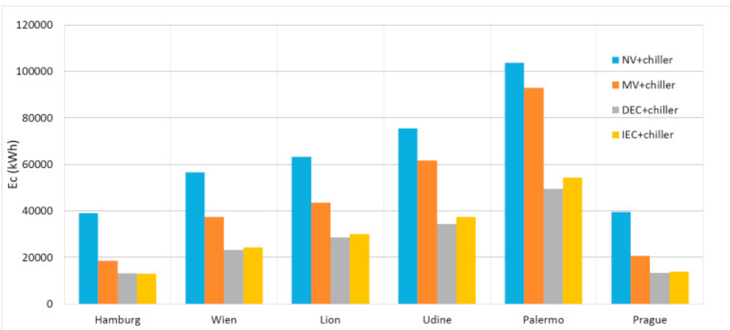

Fig. 6. Annual energy consumptions with a COP of the chiller equal to 2.5

The chiller supplies to the cooling load and it requires an electrical energy evaluated considering a chiller coefficient of performance (*COP*) constant and equal to 2.5. When the chiller operates, the cooling and the electrical load are calculated for every simulation time step. In Fig 6 annual energy consumptions are shown.

6. Conclusions

In present paper different cooling systems for an industrial building has been analyzed: a natural ventilation system, a mechanical ventilation system, a direct and an indirect evaporative cooling systems.

The results of the sensitivity analysis showed that the thermal transmittance of the building envelope and the maximum air flow rate don't affect significantly the thermal comfort conditions. Internal gains are the parameter that more strongly affects the internal environment conditions, in particular the value of internal temperature: an increment from 10 W/m^2 to 60 W/m^2 leads to a 400% increment of the total amount of time during which the temperature is greater than 26°C. Moreover, the simulations with the maximum value of internal gains, without any cooling plant, showed that the internal temperature value is always greater than 26°C for all considered locations.

With reference to *PMV* index values the mechanical ventilation case leads to a slight reduction of the total amount of discomfort time respect to the natural ventilation case. The direct evaporative cooling system presents a greater reduction, in particular for the hot climate, but the indirect evaporative cooling system is the best solution for the thermal comfort, because it guarantees a lower internal humidity ratio. Also the analysis carried out by means of *WBGT* index confirmed the trend of results obtained with the *PMV* analysis, but it shows more clearly the advantage of the indirect evaporative cooling system in humid climates.

With the aim of comparison, concerning electrical energy consumptions, the reference solution is a typical vapour compression refrigeration system. The mechanical ventilation system solution involves highly variable energy saving depending on weather conditions: in cold cities (Prague, Hamburg) the energy saving is about equal to 50%, while in hot cities it is lower than 10%. The direct and indirect evaporative cooling systems allow an energy saving around 50% and 40% for all considered locations respectively.

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