EVAPORATIVE COOLING FOR AN AUDITORIUM DURING THE DRY SEASON IN A HOT SEMI-HUMID CLIMATE

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Abstract

The performance of an evaporative cooling system in an auditorium to reduce daytime overheating during the hot dry season is analyzed. The auditorium is located in Temixco Morelos, Mexico. This region presents a hot semi-humid climate with pronounced dry and raining seasons. The auditorium was designed with a bioclimatic approach in order to operate without an air-conditioned system. Measurements show that hygrothermal comfort conditions are achieved during operative hours (daytime) for most part of the year. However, overheated conditions are present during the hot dry season (from March to June) in 33% of the daytime hours. An evaluation of the climatic conditions shows that evaporative cooling (EC) can be used in Temixco. Thus, the used of an EC system composed by small injection-type direct evaporative coolers is proposed. The Energy Plus software was used to simulate the auditorium using a fan coil air conditioning system; the consumed energy and the building thermal loads were calculated. The interior air temperature of the auditorium using the proposed EC system was obtained through an energy balance. The energy consumed by the EC system was compared with that consumed by the fan coil air conditioning system. More than 70% of energy can be saved using the EC system.

Keywords: Passive control, Bioclimatic, Evaporative cooling, Climate.

1. Introduction

The applicability of low energy cooling systems, including evaporative cooling, is determined by the climate conditions, as mentioned by Givoni (1994). Correia da Silva's (1995) numerical model shows that a solar chimney and an evaporative cooling tower system are suitable to improve thermal comfort in an auditorium for a hot dry climate.

In the central part of Mexico, a mild climate predominates but from March to July, temperatures as high as 35ºC can be reached, giving place to overheating conditions in many buildings. In this paper, the use of an evaporative cooling (EC) system is proposed for an auditorium in a hot semi-humid climate (Aw). The energy consumed by the evaporative cooling system is compared with that consumed by a fan coil system.

2. Description of the auditorium

The auditorium of The Centro de Investigación en Energía (CIE) was built in 1985 in Temixco Morelos, Mexico, where the clime is hot semi-humid. The auditorium was designed with a bioclimatic approach in order to operate without an air-conditioned system. The building's length is 16.45 m, its width is 11.5 m, and its height is 4.5 m. The walls are made of two layers of brick and a layer of 0.6 m air space between them. The walls present two non-operable windows on the east wall and two non-operable windows on the west wall to provide daylighting. Natural ventilation is given by louvers located also on the east and west walls. The roof is made of 5 layers. From bottom to top, the layers are the following: 0.40 m of hollow blocks and T-beams system, 0.10 m of concrete, 0.10 m of expanded loose fill/powders-perlite, 0.02 m of brick, and 0.01 m of asphalt. The auditorium presents a false ceiling hanging 1.5 m from the ceiling forming a plenum. There are six windmills that connect the area of the auditorium with the exterior through tubes that go through the plenum. The function of these six windmills is to help the interior air to move from bottom to top. A schematic representation is shown in Figure 1. Thermal comfort zone was calculated according to

Szokolay (1984), being the upper limit 27.2ºC and the lower limit is 24.2ºC. Interior and exterior temperatures have been measured during 2007. Since the louvers were fix-open and the windmills operated all the time, the thermal lag of the interior temperature was less than one hour. Hygrothermal comfort conditions are achieved during operation hours (daytime) for most part of the year. However, overheat conditions are present during the hot dry season (from March to June), in 33% of the daytime hours.

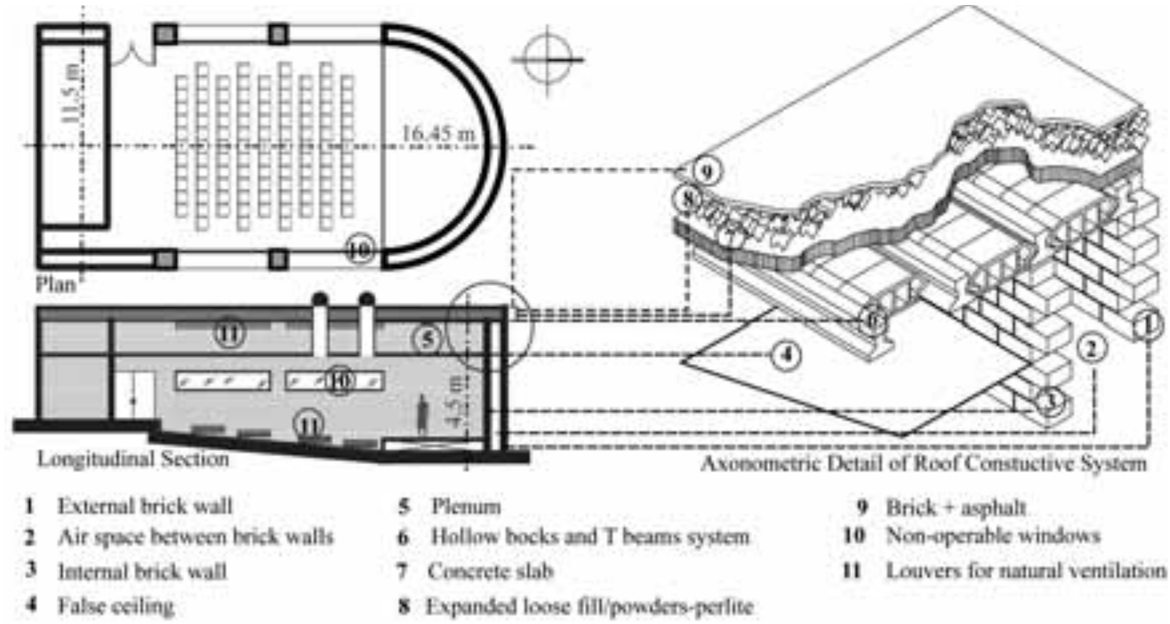

Fig. 1: Schematic description of the Auditorium and of the roof and walls constructive system.

3. Auditorium thermal conditions without cooling system

Interior and exterior measured temperatures for the auditorium without a cooling system, for ten days, are shown in Figure 2. The interior temperatures overpass the upper comfort limit for about twelve hours, some of them corresponding to the maximum occupancy period.

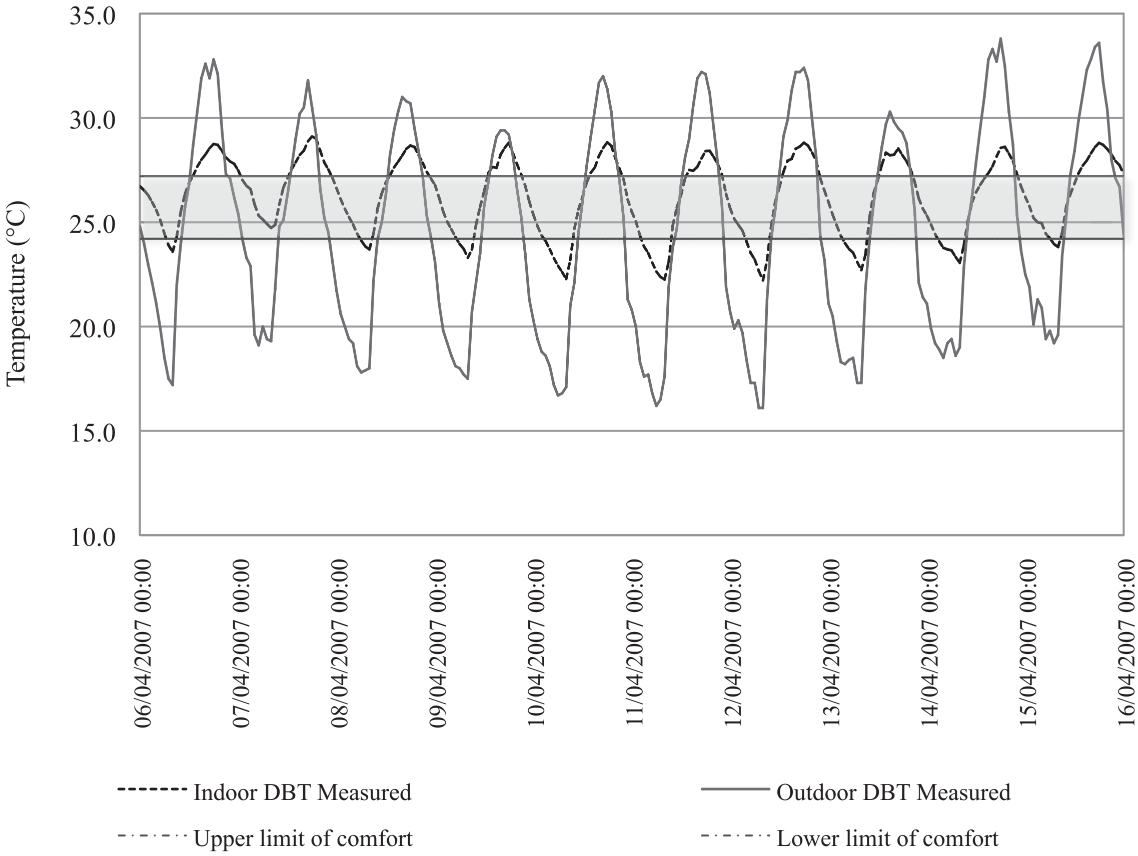

Fig. 2: Indoor thermal pattern of the CIE auditorium.

For the hot dry season (2,928 hours), the auditorium was in thermal comfort conditions only 21% of the time, in hot thermal discomfort 33% and in cold thermal discomfort 46% of the time. The corresponding distribution for each month is presented in Table 1. Since the cold thermal discomfort was mainly during the night when the auditorium was not occupied, the principal objective was to reduce the large number of hot thermal discomfort hours.

Tab. 1: Hours of percentage of comfort, hot discomfort and cold discomfort.

	Comfort	Hot discomfort	Cold discomfort
March (hours)	144	253	347
%	19	34	47
April (hours)	138	312	270
%	19	43	38
May (hours)	159	297	288
%	21	40	39
June (hours)	171	108	441
%	24	15	61
Total (hours)	612	970	1346
%	21	33	46

4. Potential of the evaporative cooling for the location

An evaporative cooling system can be used in climates where the dry bulb temperature (DBT) is not higher than 46°C, the wet bulb temperature (WBT) is not higher than 22°C, and there are not problems of water availability (Givoni, 1994). In Temixco, the third condition is fulfilled, in the following, the first and second conditions are analyzed for the hot dry season (from March to June).

4.1. DBT in Temixco 2007

From March to June, the highest average of maximum DBT (DBT Max Av) was 33.0°C, as shown in Figure 3. This value is below of the upper limit for the use of an evaporative cooling system. In this period the maximum thermal oscillation amplitude was 14.3°C in March, this large oscillation amplitude was due to the reduced water vapor in atmospheric air.

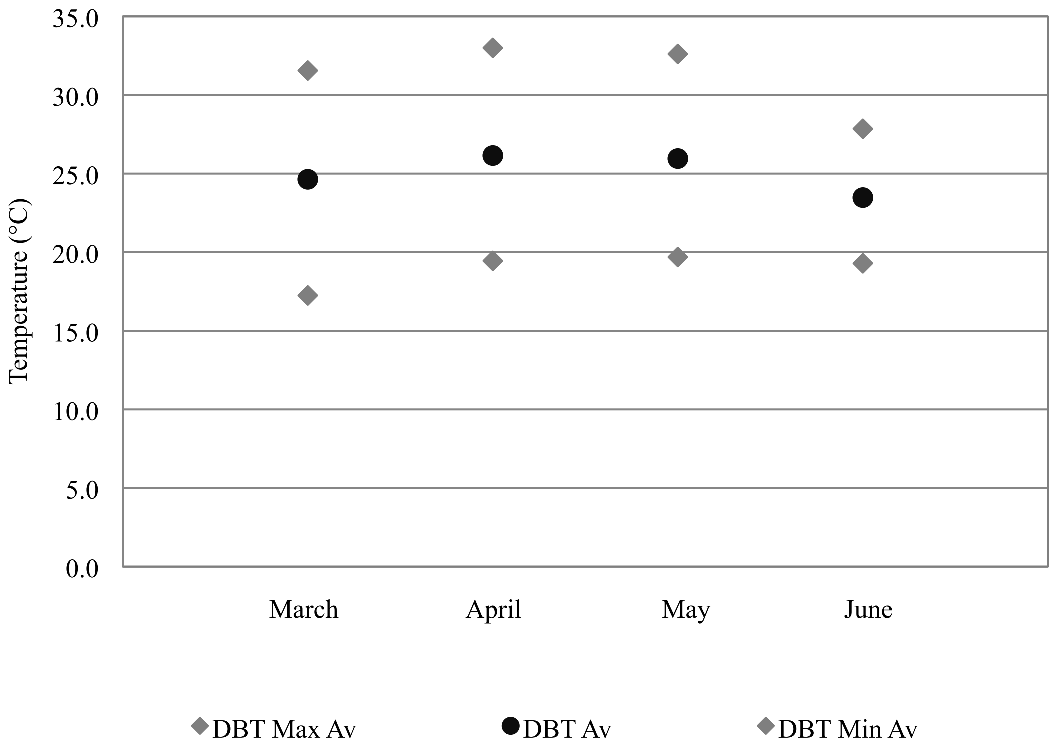

Fig. 3: Dry Bulb Temperatures from the CIE weather station.

4.2. WBT in Temixco 2007

The WBT data was obtained from the measured hourly values of DBT, relative humidity (RH) and the local atmospheric pressure (p) using the polynomial expression of Tejeda-Martínez (1994):

$$WBT = \left\{-\frac{Q}{2}+\left[\frac{Q^2}{4}+\frac{S^3}{27}\right]^{\frac{1}{2}}\right\}^{\frac{1}{3}} + \left\{-\frac{Q}{2}-\left[\frac{Q^2}{4}+\frac{S^3}{27}\right]^{\frac{1}{2}}\right\}^{\frac{1}{3}} + \frac{b}{3a} \quad \text{(eq. 1)}$$

where Q is obtained by:

$Q = 8264.65 - 1480.45\,(RH/100)e_s - 0.966pDBT$ (eq. 2)

The saturated vapor pressure is calculated by:

$e_s = 6.6x10^{-4}DBT^3 + 4.6x10^{-3}\,DBT^2 + 4.58x10^{-1}DBT + 6.63$ (eq. 3)

And the standard atmospheric pressure p in hPa by the expression:

$p = 1013.25\,(1-2.25577x10^{-5}Z)^{5.2559}$ (eq. 4) where Z is the local altitude in meters.

The S variable is obtained by:

$S = 662.23 + 0.97p$ (eq. 5)

The last term in equation 1 was considered constant:

$b/3a = -1.0$ °C (eq.6)

Figure 4 shows the WBT for the period of March to June. The highest average of maximum WBT (WBT Max Av) was 22.0°C. This value is equal to the upper limit for the use of an evaporative cooling system.

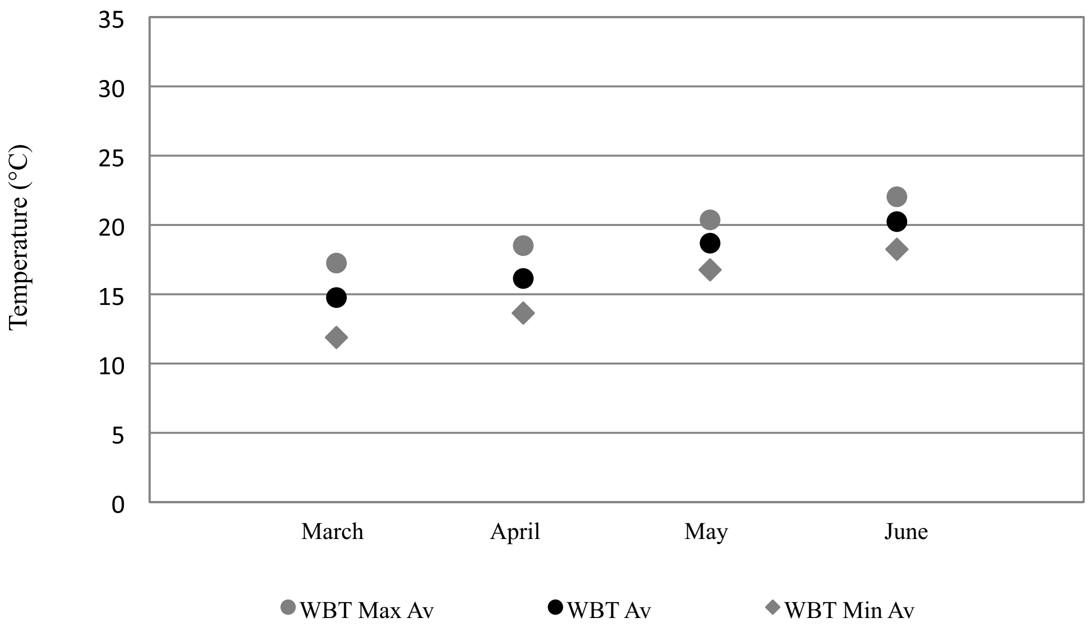

Fig. 4: Mean monthly WBT calculated from eq. 1.

5. Calculations

5.1. Non air condition calculations

The thermal performance of the auditorium without air condition was evaluated with the Energy Plus software and the Design Builder interface. The double walls and the air between them were simulates as a three layered wall. A one dimensional heat conduction equation was used to evaluate the heat transfer from outside to inside and vice versa. The air plenum in the roof was simulated as a separated space, connected to the auditorium by the false ceiling. The thermal conductivity, heat capacity and density of the construction materials were obtained from the design builder software library. The louvers were opened and the windmills operated during the whole day.

5.2. Fan coil air conditioning calculations

A compact fan coil air conditioning unit (AC), to keep the auditorium at 25ºC from 08:00 to 20:00 h in working days, was evaluated as a reference. The AC outlet air temperature was fixed to 12ºC and the cooling distribution losses were considered to be 5%. The hourly heat loads were obtained and used for a thermal balance in the evaporative cooling analysis.

5.3. Evaporative cooling calculations

The use of a direct evaporative cooling system composed by injection-type direct evaporative coolers is analyzed. Each cooler has a centrifugal ventilator having a diameter of 0.25m, which rotates at 1500 rpm. The air crosses a battery of 38 porous plates, each having dimensions 1.20mm x 0.60mm x 3.50mm. The wet (effective) surface of the battery is $50m^2$. The water distribution circuit works on a close loop and has a volume flow rate of 100 kg/h. The air mass flow rate and the exit temperature for this cooler were obtained, from experiments, by Santamouris (1996).

The air mass flow rate $\dot{m}_a$ in kg/h is given by:

$$\dot{m}_a = -39.7 + 1.46x10^{-5}\ \dot{m}_w v_f + 0.2\ v_f \qquad \text{(eq. 7)}$$

$\dot{m}_w$ are the air and water flow mass rate in kg/h and v_f is the fan speed in rpm.

The air volumetric flow rate $\dot{V}_a$ in m^3/h is,

$$\dot{V}_a = \dot{m}_a/\rho \qquad \text{(eq. 8)}$$

The exit temperature of the evaporative cooler, T_{ec} in °C is given by

$$T_{ec} = DBT–0.23\ \dot{m}_w^{0.09}\ (DBT–WBT) – (1.18x10^{-4}\ \dot{m}_w^{0.09}\ (DBT–WBT)^{2.16}) / v_f^{-0.61} \qquad \text{(eq.9)}$$

The air volumetric flow rate of a cooler unit results in 218 m^3/h.

The auditorium has an air volume of about 600 m^3 and a minimum of 10 air changes (ACH) are required to fulfill air quality construction regulations (SOSDF, 2011).

The indoor air temperature using the evaporative cooling system was obtained from an energy balance, considering a well-mixed air, building thermal loads obtained from the Energy Plus results when using a compact fan coil air conditioning unit, 80 people sited in the auditorium, and air flow from outside,

$$T_{int\,(i)} = [(q_{bu} + q_{pe})/ V\rho Cp + ACH_1(T_{ec} – T_{int\,(i-1)}) + ACH_2(T_{out} - T_{int\,(i-1)})]\ \Delta t + T_{int\,(i-1)} \qquad \text{(eq. 10)}$$

where $T_{int\,(i)}$ and $T_{int\,(i-1)}$ are the internal temperature at actual and previous times, q_{bu} and q_{pe} are the heat flows due to the building thermal loads and the people sited in the auditorium, respectively. V is the volume of the auditorium, ρ the air density and Cp the air specific heat. ACH_1 and ACH_2 are the air changes from the evaporative cooling and must fulfill the condition that $ACH_1+ACH_2 \geq 10$, and T_{out} is the temperature of the ambient.

6. Results

6.1. Non air condition results

The auditorium without an air conditioning system and considering 1.8 ACH was first simulated with the Energy Plus software and the results were validated with measured data within the auditorium as shown in Figure 5. The oscillation amplitude of the indoor DBT simulated is similar than that of the indoor DBT.

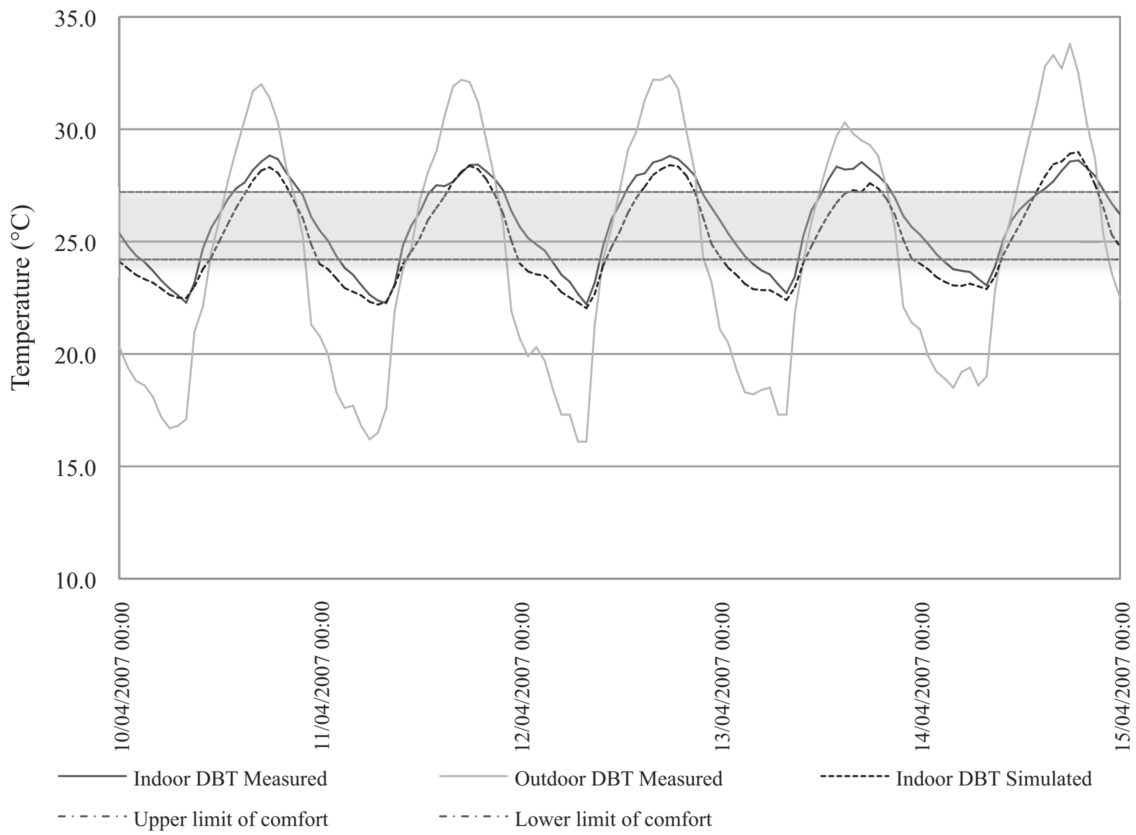

Fig. 5: Experimental and simulated indoor DBT temperatures.

6.2. Fan coil air condition results

With the Energy Plus software, the electrical energy consumed by a compact fan coil air conditioning unit during the hot season was evaluated. The energy consumed per month from March to June was estimated to range from 1400 kWh and 1500 kWh (See Figure 7).

6.3. Evaporative cooling results

In Figure 6, the indoor temperature using the evaporative cooling system (T_{ec}), for the case where 21 cooler units and 9 ventilators (with the same air volumetric flow rate of a cooler unit) were used, is presented. For comparison, the indoor measured temperature without air conditioning (Indoor DBT measured) is included. The mean interior temperature when using this EC system is 23.2°C, 1.8°C lower than the corresponding to the auditorium without cooling, which is 25.0°C. The time in hot thermal discomfort for the week of 10 to 14 of April was of 14 hours with this EC system and 42 without air conditioning.

Since the cooling demand depends on the thermal loads, the number of evaporative cooling units used should be varied along the time to get temperatures within the comfort zone, the maximum value will be the number of cooler units of the system. These calculations will be done in a future work, but present results allow predicting that no more than 30 cooler units will be needed.

The energy used by an EC system composed by 30 cooler units was estimated from the fans and water pumps energy consumption. Typically, for the proposed small cooler units, the fan and the water pump consume approximately 50 Wh per hour; with an operation schedule from 08:00 h to 20:00 h during working days, the monthly electrical energy consumption is about 400 kWh/month.

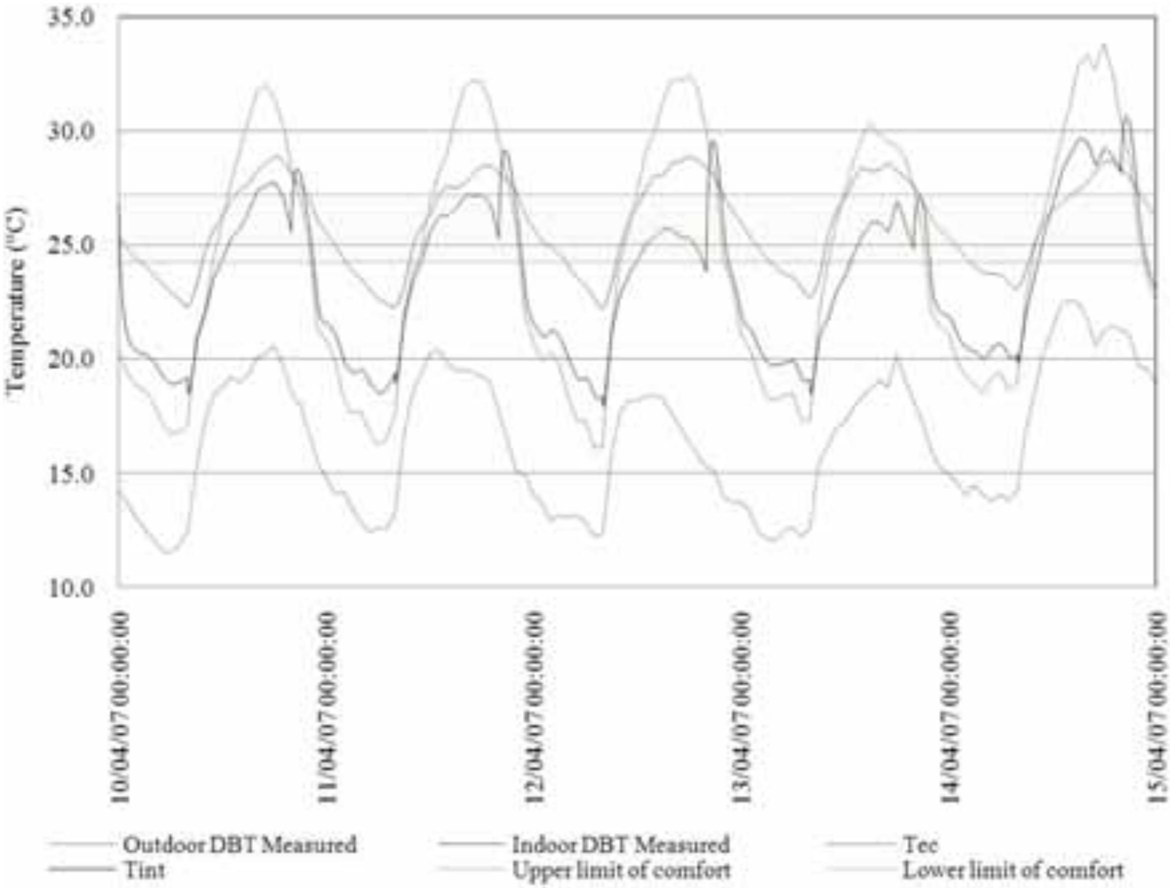

Figure 6. Interior air temperature with evaporative cooling.

In Figure 7, the EC energy consumption per month is compared with that corresponding to the fan coil air conditioning system. Both are calculated considering an operation schedule from 08:00 h to 20:00 during working days. These results show that the monthly energy consumption of the propose EC system ranges from 26% and 29% of that of a fan coil air conditioning system.

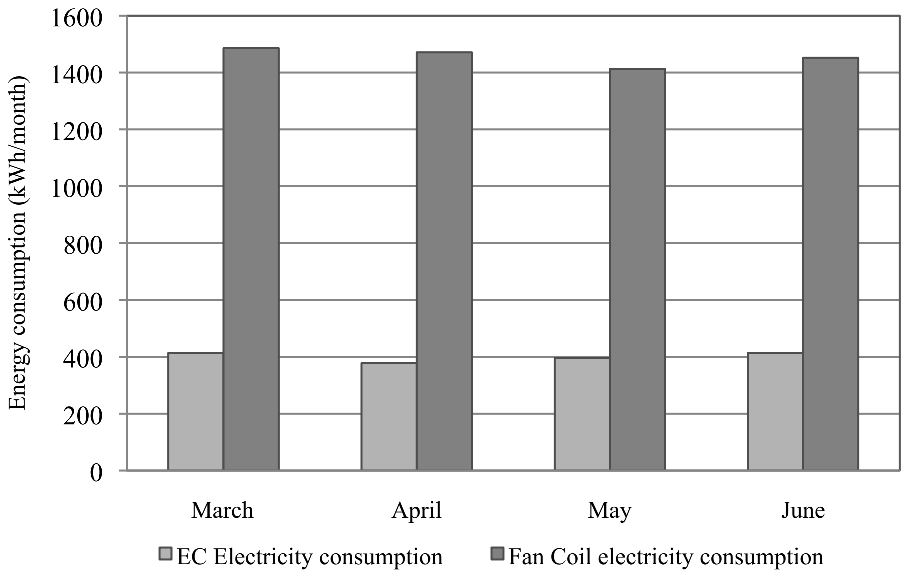

Figure 7: Monthly electrical energy consumption by EC and by AC systems.

7. Conclusions

The results show that the evaporative cooling is a good low energy cooling alternative to reduce the time with overheated conditions in the auditorium of the CIE located in Temixco Morelos, Mexico.

More than 70% of energy can be saved with an evaporative cooling system compared with a fan coil air conditioning system.

8. Acknowledgement

Partial economic support from CONACyT-Morelos 93693 and PROMEP-UAT 103.5/10/0007 projects are acknowledged. Helpful discussions with Raul Rechtman are also acknowledged.

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