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Summer Energy Saving In Indoor Environments Using A New Free-Cooling System

C. Isetti[1], E. Nannei[2] and B. Orlandini[3]

[1] Department of Department of Architectural Design and Construction
DIPARC, University of Genoa
Stradone S. Agostino, 37 – 16123 Genoa (Italy)
Phone number: 0039 010 2095890, e-mail: **isetti@leonardo.arch.unige.it**

[2] Department of Production Engineering, Thermo–energetic and Mathematical Models
DIPTEM/TEC, University of Genoa
Via all'Opera Pia, 15 A - 16145 Genoa (Italy)
Phone number: 0039 010 3532875, e-mail: **nannei@ditec.unige.it**

[3] Department of Production Engineering, Thermo–energetic and Mathematical Models
DIPTEM/TEC, University of Genoa
Via all'Opera Pia, 15 A - 16145 Genoa (Italy)
Phone number: 0039 010 3532578, e-mail: **bruno.orlandini@unige.it**

Abstract.

This paper analyses a new indirect evaporative cooling system characterized by low energy demand for use in summer conditions. The process is carried out by means of a membrane contactor (MC) equipped with a hydrophobic membrane that is permeable to vapour but not to the liquid phase. Chilled water feeds a radiant ceiling panel in order to reduce the mean radiant and indoor air temperatures in the environment served and, consequently, the energy required by a heat pump working to achieve comfortable conditions. The energy needed by this system is compared with the one required by the heat pump alone to get the comfortable conditions. The results yielded by a SIMULINK program in different summer climatic conditions, in the Mediterranean area, show energy savings of 30 ÷ 60 %.

Key words

Energy saving, Thermal comfort, Summer cooling, Radiant Panel, Membrane contactor.

1. Introduction

Indoor cooling systems with low energy requirements might yield significant energy saving in warm and mild summer climates. In this regard, cooling by means of water evaporation is of particular interest [1-3] since energy consumption is limited to the fan load. A first distinction is usually made between systems that use a direct approach and those that use an indirect approach. In the former type of system, cooled air is directly conveyed to the room, while the latter makes use of a heat exchanger to chill the air flow to the room, in order to avoid increasing indoor air humidity [3]. The cooling system proposed in this paper combines a radiant ceiling panel [4-6] with a new indirect evaporative process carried out by means of a MC equipped with a hydrophobic membrane that is permeable to vapour but not to the liquid phase [7-10]. In comparison with a direct-contact air-water evaporative process, the MC allows the frontal velocity of the air to be varied with a flexibility of up to 4-5 m/s, while avoiding the carryover of water droplets, facilitating compactness and achieving high cooling efficiency. This paper presents a theoretical analysis of energy saving when such a system is coupled with a heat pump in different summer conditions in the Mediterranean area.

2. Membrane Contactor Cooling System

The system studied combines two subsystems: a radiant ceiling panel (RCP) fed by chilled water, and the membrane contactor (MC) with the heat exchanger (HE) (Fig. 1). As can be seen, an outside air flow is driven by the fan (F1) through the MC, to allow water evaporation from the liquid side to the air through the membrane. Before entering the MC, the outside air is pre-chilled by the HE to enhance the water cooling effect. The chilled water is first conveyed to a tank by means of a pump (P2) and then to the RCP by another pump (P1). The upper side of the RCP is insulated to reduce parasitic thermal exchanges.

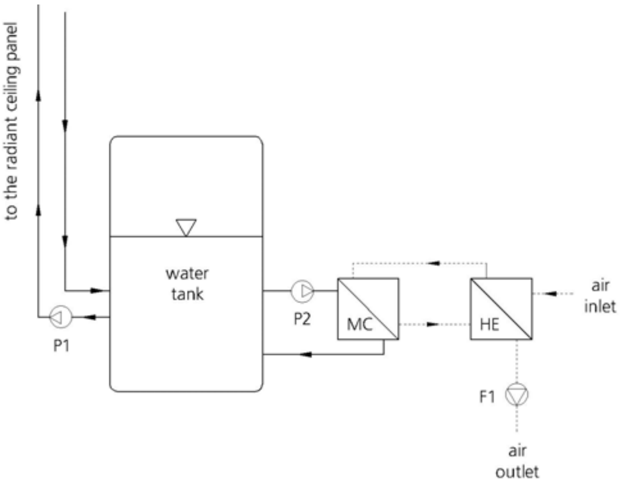

FIG. 1. The indirect evaporative cooling system; MC coupled to HE on the air path

Fig. 2 shows a schematic representation of the standard contactor (1 m^2 membrane area with 20 plate membrane modules), manufactured by an Italian company (dimensions: $L_x = 200$ mm, $L_y = 340$ mm, $L_z = 210$ mm).

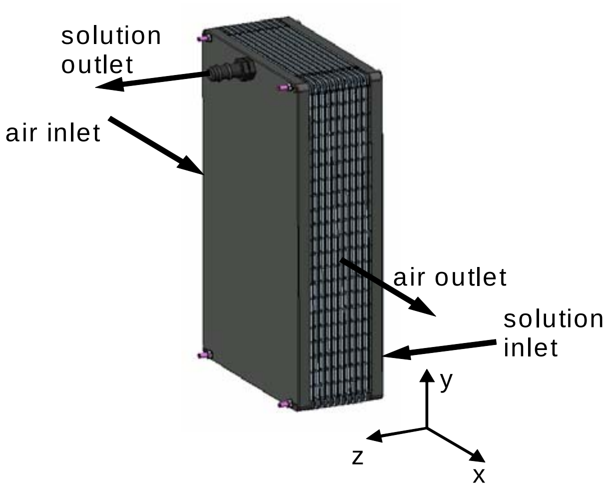

Fig. 2. Schematic representation of the MC

3. Mathematical model

The system was studied by means of a SIMULINK_ MATLAB code particularly suited to simulate the dynamic behaviour of complex systems by reducing their components to graphical interfaces i.e. to blocks in which the inlet and outlet variables are linked together by means of the equations that describe the behaviour of the related components. The simulation model combines two parts: the first dealing with the room and RCP, the second with the water-chilling system represented in Fig.1.

With reference to first part (i.e. the heat balance in the room) the following assumptions have been made:

- the air is assumed to be ideally mixed;
- the emissivity of all surfaces in the room is $\varepsilon = 0.9$;
- the outdoor air renewal rate is n = 0.5 1/h;
- the internal source is the sensible heat produced by two persons engaged in light activity (1.2 Met) [11];
- the heat balance of the internal air takes into account the convective and radiant heat exchanges with the internal structures (floor, ceiling/radiant panel and walls) separately;
- heat conduction through the external wall and the internal structures is uni-dimensional;
- no heat exchange takes place between the room under study and adjacent rooms;
- solar radiation through glass, computed according [1,2], is considered to be diffused in the room;
- the two persons are wearing summer clothing (0.5 clo) in accordance with thermal comfort theory [11];
- the heat pump used to cool the indoor air is a direct expansion system with an average EER = 2.5.

The finite-volume method was used to evaluate the temperatures of all the surfaces facing the enclosure. At each simulation time, both the indoor air temperature and the mean radiant temperature were evaluated with reference to the centre of the room.

In the second part of the model (i.e. the water-chilling system, tank, water flow to the RCP, MC, HE) the following assumptions have been made:

- heat and mass transfer in the MC refers to the solution of the mass and energy equations applied to the repetitive module of the contactor. The finite-volume method is adopted to discretize the conservation equations on the air side and the water side. A description of the numerical model is given in [7-9];
- the MC is equipped with a 2 m^2 composite membrane (PTFE layer, 70 μm thick, laminated on a polyester layer up to a total thickness of ≈ 170 μm). The PTFE layer is faced to the water. The measured vapour transmission property of the membrane, according to [12], is Ret = 1.0 (m^2Pa/W);
- the heat exchanged in the HE is computed on assuming effectiveness $\eta_{HE} = 0.6$ [13];
- the area of the RCP is equal to the ceiling one; the panel fin effectiveness (fig. 3) resulted $\eta_{RCP} = 0.9$ [13].

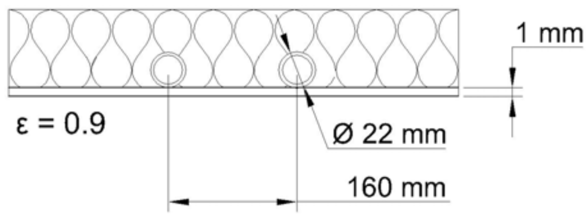

Fig. 3 Sketch of the radiant ceiling panel (RCP)

- the water flow rate tφ the RCP is $q_{RCP} = 300$ kg/h;
- the outdoor air flux through the MC is $G_a = 460$ kg/h;
- the water mass in the tank (10 kg) is maintained constant by replacing the evaporated water by tap water.

All the above-mentioned components were reduced to a Simulink blocks, which were linked the other components in order to obtain the whole system to be simulated.

The performance of the present cooling system is based on the latent and sensible heat exchanges taking place in the MC. The outputs of the MC Simulink block have been previously validated by experiments [8-10]. Fig. 4 shows the measured vapour flux data for the standard MC (1 m^2 membrane area) as a function of the air flow rate through the MC during an evaporative process, in comparison with the computed values (solid line) [8-10]. The predicted results were in good agreement with those yielded by the experiments.

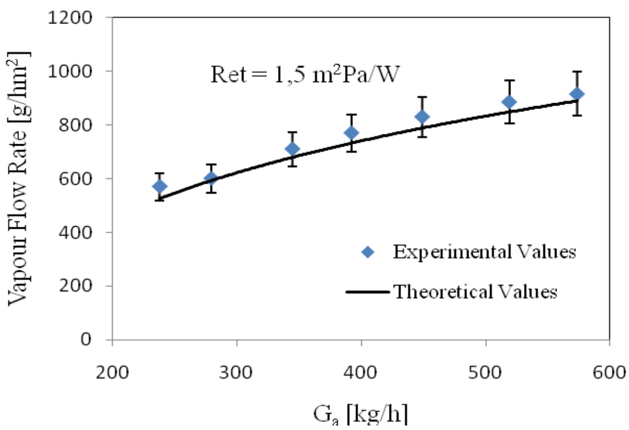

Fig. 4. Computed and experimental evaporative fluxes through MC as a function of air flow rate (air side t_{in} = 20 °C; φ_{in} = 30 %; water side t_{in} = 12.5 °C and q_m = 640 kg/h).

The present theoretical analysis was carried out by comparing the heat pump energy consumptions required to maintain the PMV index at a level not exceeding + 0.5 for a room with or without the cooling system. In order to evaluate the electrical consumptions of the auxiliary components of the cooling system, the pressure losses were derived from the usual technical data; for the MC, reference was made to the experimental data shown in fig. 5 [8]. The total auxiliary power estimated for P1, P2 and F1 was P_{aux} = 75 W.

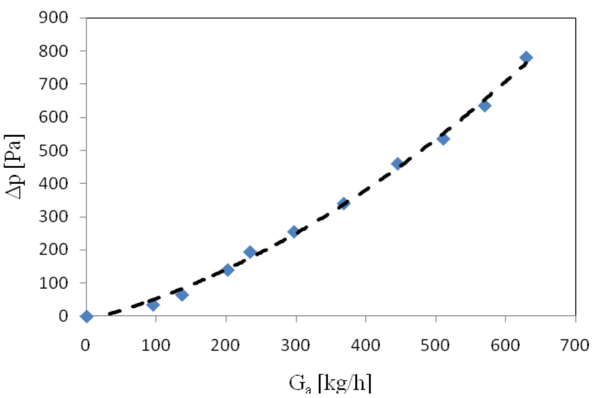

Fig. 5. Experimental MC pressure losses versus air flow rate

4. Results

A preliminary simulation run was focused on selecting the most suitable working conditions of the water-chilling system alone. For the sake of simplicity, we computed only the heat power removed by the panel in reference steady-state conditions of the room; i.e. the outdoor and indoor air temperatures and the mean radiant temperature were set equal (t_e= t_a= t_{mr} = 26 °C).

Fig. 6 reports the results (solid lines) obtained for three different air humidity values (φ_e = 40% ; φ_e = 55% and φ_e = 75 %) as a function of the ratio q_m/G_a. The dashed lines show the corresponding results obtained without the heat exchanger (HE) on the air path.

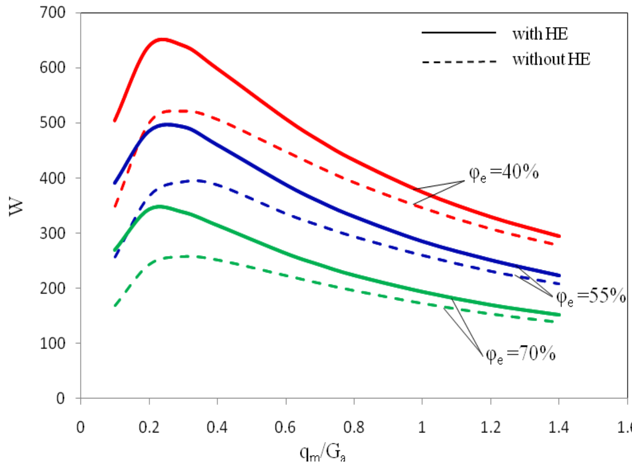

Fig. 6. Heat removed by the panel in reference conditions as a function of the ratio q_m/G_a for different outdoor air humidity values.

As can be observed, the heat power removed by the panel becomes more significant as φ_e decreases, with a maximum value at $q_m/G_a \approx 0.3$. As shown in Fig. 6, without the HE, the heat power removed by the panel decreases by 20% to 30%; it should be noted, however, that the energy requirement of the fan F1 decreases owing to the reduced pressure losses. It is noteworthy that the panel is free from vapour condensation since the cooling effect decreases as φ_e increases.

On fixing q_m/G_a = 0.3, further simulations were carried out to analyse the indoor conditions and the heat pump energy required in order not to exceed PMV = + 0.5.

Fig. 7 and Tab. 1 show the room under consideration and its geometrical and thermophysical properties. The wall west-facing is equipped with a window. Tab. 1 summarizes the main characteristics of the room.

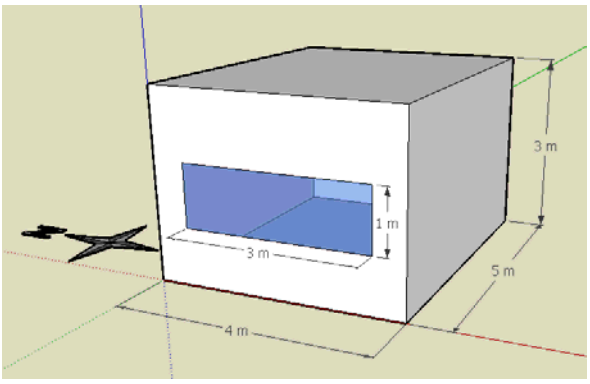

Fig. 7. Geometrical characteristics of the considered room

The outdoor climatic conditions shown in Fig. 8 refer to a sunny day in Genoa, in the north of Italy (44° 24' N), and in Foggia (41° 28' N), in the south, on 21st July [14]. The results reported refer to steady daily variations in all the variables which characterize the behaviour of the room.

Tab. 1: Geometrical and Thermophysical properties of the considered room

External/internal structures	**Typology**	**Thickness (m)**	**Thermal conductivity (W/mK)**	**Density (kg/m^3)**
External wall W facing	1 Brick (outside)	0.15	0.70	1700
	2 Insulation	0.04	0.04	30
	3 Brick (inside)	0.08	0.50	900
Internal partitions	Brick	0.15	0.70	1700
Floor/ceiling	Mortar	0.35	1.6	2000
Window W facing	**Typology**	**Area (m^2)**	**Trasmittance (W/m^{2}K)**	**Shading factor (-)**
	double glazed	3	2.2	0.67

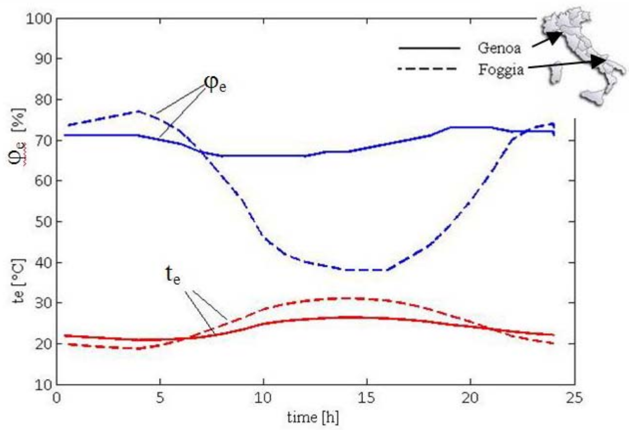

Fig. 8 Daily variations in t_e and φ_e in Genoa and Foggia.

Fig. 8 clearly shows that the climate is warmer and drier in Foggia than in Genoa; these ranges are typical of many Mediterranean locations.

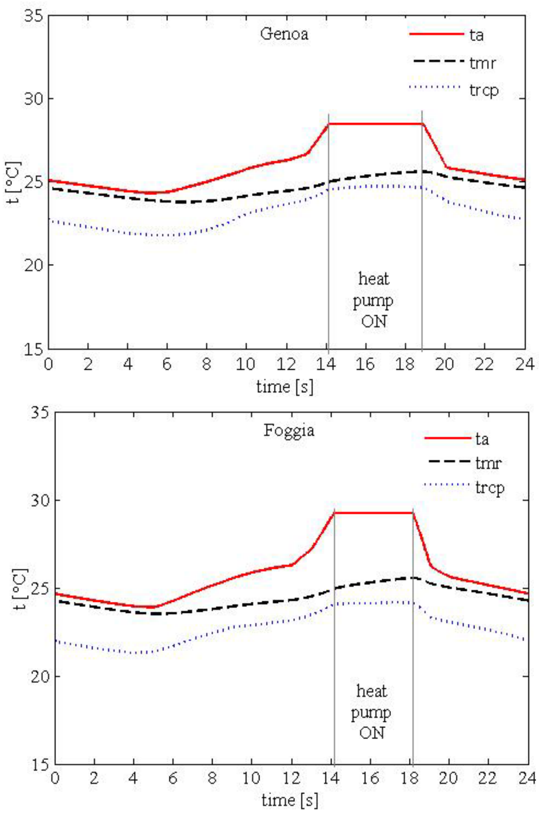

Fig. 9 Daily variations (21st July) in t_a, t_{mr} and t_{RCP} in Genoa and Foggia.

Fig. 9 reports the computed daily variations (21st July) in indoor air temperature t_a, mean radiant temperature t_{mr} and mean RCP temperature t_{RCP} in both cities. Such a figure reveals that t_{RCP} is always lower than t_{mr}; the panel therefore works effectively to lower both the mean radiant and air temperatures in the room, thereby reducing heat pump energy requirements.

Fig. 10 reports the PMV daily profiles with/without the controlling action of the heat pump, which is required for no more than 6-7 hours in the afternoon. Without the heat pump, but with the cooling system on, the PMV of the room reaches a maximum value of about 1.5 in Foggia and 1.1 in Genoa. Without any control system, totally unacceptable values of t_a and t_{mr} are reached (> 40°C with PMV > +3).

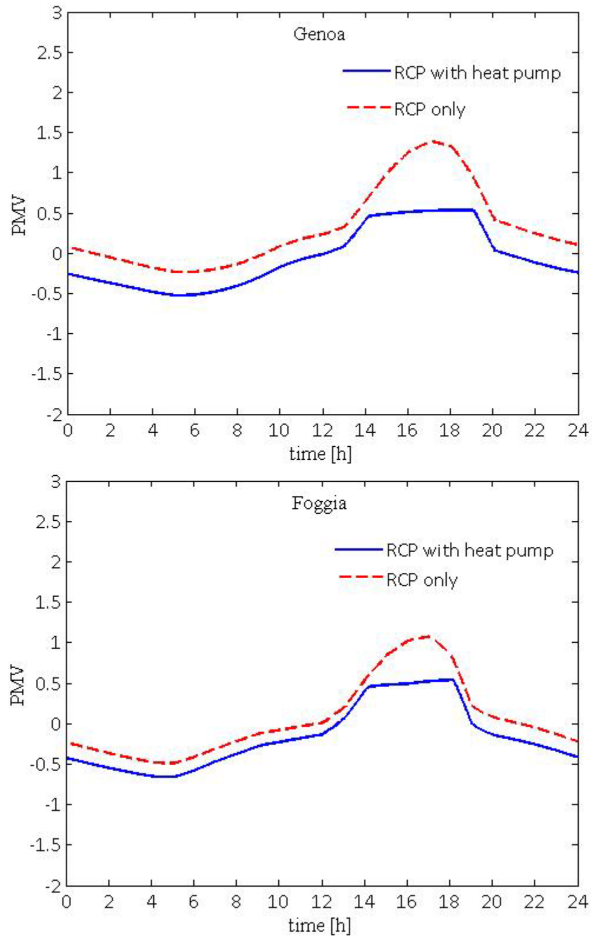

Fig. 10 PMV daily variation for Genoa and Foggia, with and without heat pump.

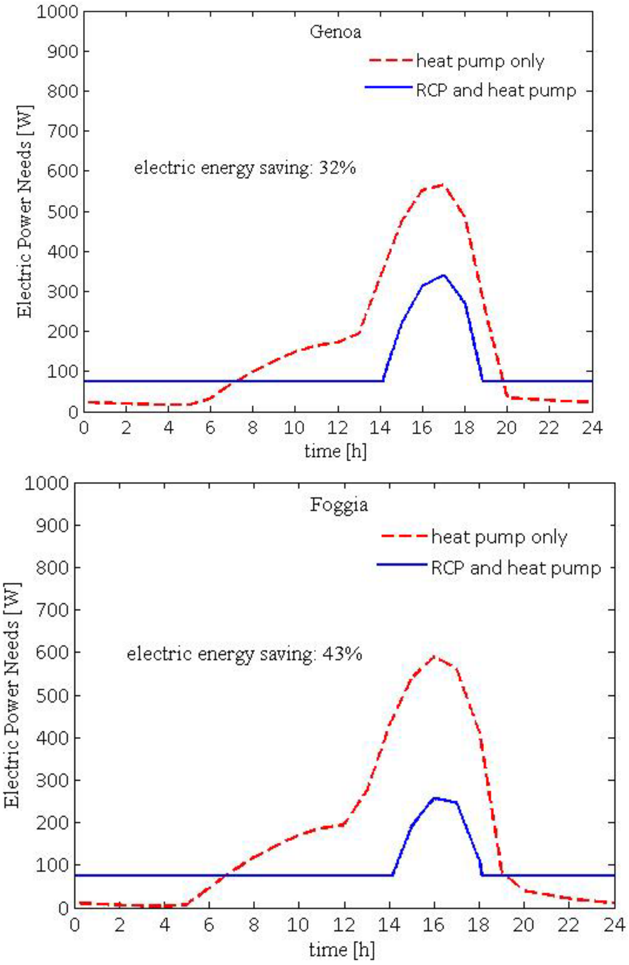

Fig. 11 Electrical power needs (heat pump + auxiliaries) for Genoa and Foggia.

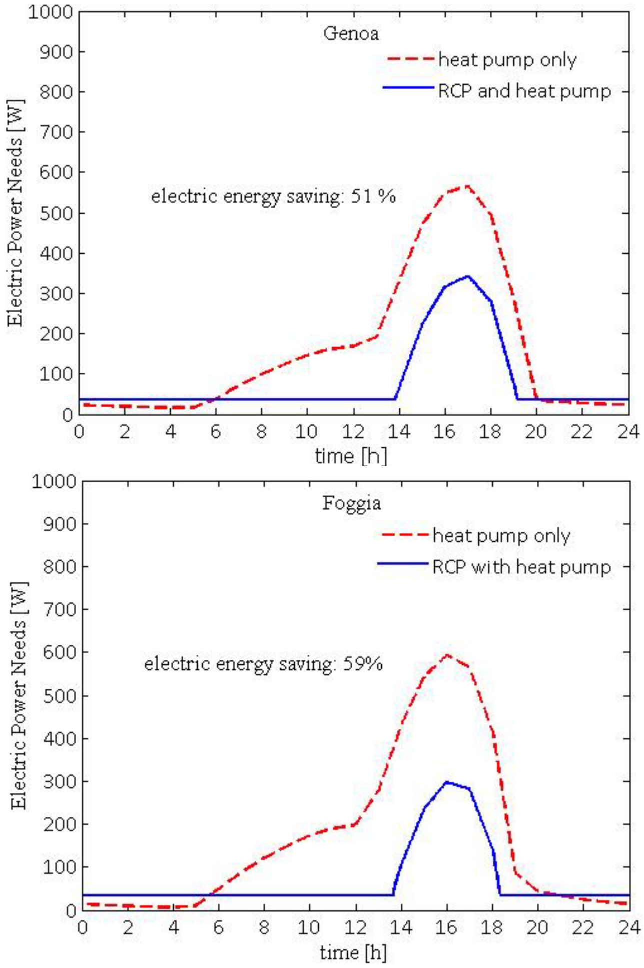

Fig. 12 Electrical needs for Genoa and Foggia without HE (G_a = 350 kg/h; q_m/G_a = 0.3).

Fig. 11 shows the computed electrical power needs and total daily energy consumption (heat pump + auxiliaries) for Genoa and Foggia.
As can be observed, the cooling system significantly reduces the thermal load handled by the pump, particularly in the warmer city (Foggia).
In terms of total daily electrical energy consumption, the saving ranges from 32% (Genoa) to 43% (Foggia).
The daily water consumption is 16.8 kg in Genoa and 23.5 kg in Foggia.
Finally, with a view to reducing the power required by the auxiliaries (mainly fan F1), the system was analysed without the HE and with a reduced air flow rate to the MC (G_a = 350 kg/h), so limiting the power need of the auxiliaries: P_{aux}= 35 W.
Fig. 12 compares electrical needs and total daily energy consumption for Genoa and Foggia for this condition.
As can be observed, despite the lower efficiency of the cooling system, this last operating condition appears more profitable in terms of total daily saving, which increased from 32% to 51% in Genoa and from 43% to 59% in Foggia.

4. Conclusions

The following conclusions can be drawn:

- the optimum ratio between the flow rates of water and air conveyed to MC is close to 0.3;
- the heat loads removed by the cooling system increases significantly as the relative humidity the outdoor air decreases;
- in summer conditions that are typical of many Mediterranean locations, the cooling system enables both the mean radiant temperature and the indoor air temperature of the room to be reduced, thereby lowering the energy requirements of the heat pump;
- without the heat pump, PMV reaches values of 1.2 -1.5 for a limited time (5-6 hours), despite the cooling system; without any control, PMV values becomes totally unacceptable (PMV > +3);
- in the climatic conditions considered, the electrical energy saved by the heat pump is in the range 32%÷ 43%, while maintaining indoor comfort;
- if the heat exchanger is excluded and a lower air flow rate is set, the pressure losses along the air path reduce, the cooling system operates at a lower efficiency, but as a result, the electrical energy saving rises from 43% to 59%.

Nomenclature

EER: energy efficiency ratio (-)
F1: fan
G_a: air mass flow rate to MC (kg/h)
HE: heat exchanger
L: length (m)
MC: membrane contactor
n: air renewal rate (h^{-1})
P1, P2: circulation pumps
P: power (W)
PMV: predicted mean vote
PTFE: polytetrafluoethylene
q_m: water mass flow rate to MC (kg/h)
q_{RCP}: water mass flow rate to the panel (kg/h)
RCP : radiant ceiling panel
Ret: membrane vapour resistance (m^2/PaW)
t: temperature (°C)
x: co-ordinate (m)
y: co-ordinate (m)
z: co-ordinate (m)

Greek letters

ε: emissivity (-)
η: effectiveness
φ: air relative humidity (%)
Φ: outside diameter

Subscripts
a: related to indoor ambient
aux: related to auxiliaries
e: related to external air
HE: related to heat exchanger
in: related to inlet
mr: related to mean radiant
RCP: related to radiant ceiling panel

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