MANAGEMENT STRATEGY FOR IMPROVING THE ENERGY EFFICIENCY OF A GROUND COUPLED HVAC SYSTEM

N. Pardo[1,*], Á. Montero[1], A. Sala[2], J. Martos[3] and J.F. Urchueguía[1]
[1]Department of Applied Physics, Univ. Politécnica de Valencia, Spain
[2]Inst. for Industrial Control Systems and Computing, Univ. Politécnica de Valencia, Spain
[3]Department of Electronic Engineering, Univ. de Valencia, Spain

Tel*: +34 963877007 ext. 85247, e-mail*: nipargar@doctor.upv.es

ABSTRACT

In this work, we present a new managament strategy for a ground coupled HVAC system in a cooling dominated office area. The idea is disminish the electrical consumption while keeping the thermal comfort requirements. This objective is achieved by means of the proper management of some parameters of the system: the air mass flow in the fan, the mass water flow in the internal and external hydraulic circuit and the set point temperature in the water to water heat pump. The electric consumption of the HVAC system is calculated for this new management strategy and compared with the results obtained for a conventional management strategy. In our simulation, the annual electrical energy savings of the ground coupled HVAC system are bigger than the 24% when it is managed by the new one.

INTRODUCTION

Nowadays, the increasing of the energy consumption is producing serious changes in the natural environment as the global warming. Around the 40% of all greenhouse gas emissions in developed countries come from the building equipments, where approximately 60% are produced in cooling and heating systems. Therefore, the development of HVAC systems has to address the issues of improving its energy efficiency and reducing its environmental impact.

Ground coupled heat pumps (GCHP) are an attractive solution for cooling and heating commercial buildings due to their higher efficiency compared with the conventional air to water heat pump (e.g. Lund et al., 2001). The Environmental Protection Agency (EPA) recognizes ground source heat pump systems among the most efficient and comfortable heating and cooling systems available today (see EnergyStart, 2008).

In the design of the air conditioning systems, the references taken to estimate the heating and cooling capacity of the heat pump to install are the coldest and the warmest day throughout the year. Therefore, most of the time the air conditioning system is underused. In this context, an energy management in HVAC systems is thought to improve its energy efficiency while satisfying the thermal comfort demand.

There are different methods to estimate the comfort state based on defining a parameter quantifying it. The most popular choice is the Predicted Mean Vote index (PMV) that predicts the mean value of the votes of a large group of persons in the Thermal Sensation Scale (see ISO7730-1994, 1994). To achieve the desired PMV with the minimal power consumption, management strategies can be implemented to reset the operation parameters. Its experimental evaluation supposes high costs and long duration. Therefore, numerical simulations are a good alternative to test them. This study uses TRNSYS software package for this purpose.

The aim of this work is to evaluate the energy savings that a new management strategy can produce in a HVAC system composed by a ground coupled heat pump and a central fan coil for a standard office space. In this strategy, the air mass flow in the fan, the water mass flow in the internal and external hydraulic system and the set point temperature in the water to water heat pump, usually fixed in conventional strategies, have the possibility of a continuous regulation that allows us to design a more efficient energy way to achieve the desired thermal comfort fixed by the PMV.

SIMULATED SYSTEM

In this section, we present a description of the office parameters as well as PMV index, which is the criteria used to evaluate the thermal comfort. Afterwards, we describe the energy model of the employed devices in the air conditioning system, the total electrical power equation of the air conditioning system and the new management strategy and the conventional one.

Simulated office area

The simulated office area comprises 108 m² (12m x 9m) with two windows in the façades north and south and three in the façades east and west. To perform the simulation, the office area is characterized by its building materials, its dimensions, distribution and orientation.

There are four different kinds of construction elements: external walls, floor, roof and window glasses. External walls are defined as ventilated façades composed by four elements: perforated brick, 5 cm of insulation, air chamber and a Naturex plate cover; its global conductivity is 0.51 W/m^{2}K. The floor and the roof are built with hollow block with 5 cm of insulation; its global conductivity is 0.51 W/m^{2}K. Finally, the window is composed by a window glass, with solar radiation transmissivity equal to 0.837 and conductivity equal to 5.74 W/m^{2}K, and a window frame, with conductivity equal to 0.588 W/m^{2}K. The windows size is 1.5 square meters, dedicating a 15% of this area to the frame surface. The internal and external shadow factor for these windows is estimated in 0.7.

The orientation of the office area, the layout of windows and the fan coil is shown in figure 1.

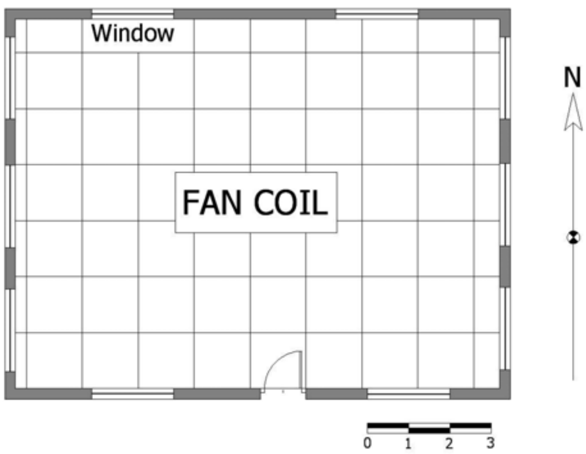

Figure 1 Layout of windows and the fan coil in the office area.

All these parameters are included in the TRNSYS software package through the TRNBuild tool specifically designed to simulate the thermal behaviour in a multi-zone building area. Finally, the weather data base of the Spanish city of Valencia is used to characterize the Mediterranean coast weather.

Comfort criteria

The Predicted Mean Vote (PMV) index is one of the criteria to estimate the comfort state proposed by the ISO7730-1994 standard. We choose this parameter to evaluate the comfort state in our simulation. PMV predicts the mean value of the votes of a large group of persons on the seven point Thermal Sensation Scale.

The PMV index is based on heat balance of the human body. A person is in thermal balance when the internal heat production in the body is equal to the loss of heat to the environment, and it can be determined when the activity and the thermal resistance of clothing are estimated, and the following environmental parameters are measured: air temperature, mean radiant temperature, mean air velocity and partial water vapour pressure.

PMV index in the office area is calculated at each simulation step time. To do it, three parameters have to be estimated: activity, thermal resistance of clothing and mean air velocity. We follow ISO7730-1994 recommendations for this purpose. The value for the activity parameter is 1.2 met for moderated office activity. The thermal resistance of clothing is 1.0 clo, when the considered working clothes are shirts, trousers, jackets, and shoes. Finally, the value for the mean air velocity is estimated in 0.1 m/s. The other three parameters, air temperature, mean radiant temperature and partial water vapour pressure are computed at each time step.

Air conditioning system and energy model of its devices

The air conditioning system linked to the simulated office area is based in a ground coupled heat pump. This system is composed by two parts: the external circuit composed by the external water pump (EWP) and the geothermal heat exchanger (GHE), and the internal circuit, composed by the internal water pump (IWP) and a central fan coil (FC) linked to the office area. The water to water heat pump (WWHP) is a reversible heat pump and it is the connecting element between both circuits. In figure 2 we include a schematic diagram showing the layout of these devices.

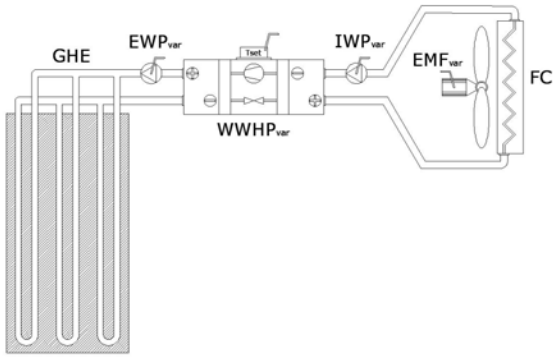

Figure 2 Air conditioning system: geothermal heat exchanger (GHE), external water pump (EWP), internal water pump (IWP), water to water heat pump (WWHP), fan coil (FC), electric motor of the fan (EMF).

The only difference between the hydraulic circuit for the new management strategy and the conventional one is in the active devices (WWHP, EWP, IWP, EMF) employed. For the new management strategy, the active devices have a continuous regulation, whereas for the conventional one these devices have only two positions: switched on or switched off.

The following paragraphs detail the model describing the behaviour of each component. We also derive the way to compute the electrical power consumption of each one, for given working conditions. Notice that the purpose of this study is to improve the energy efficiency of the system keeping the comfort requirements. Therefore, the knowledge of this power consumption behaviour allows us to develop a suitable management strategy to achieve this objective.

*Ground Heat Exchanger (GHE)***.** The objective of the ground heat exchanger is to interchange heat with the ground. A heat carrier fluid is circulated through the ground heat exchanger and either rejects heat to, or absorbs heat from the ground depending on the temperatures of the heat carrier fluid and the ground. In typical U-tube ground heat exchanger applications, a vertical borehole is drilled into the ground. A U-tube heat exchanger is then pushed into the borehole. The top of the ground heat exchanger is typically several centimetres below the surface of the ground. Finally, the borehole is filled with a fill material.

We use ‘Duct Ground Heat Storage Model’ to simulate our GHE (see Hellström, 1989); this model assumes that the boreholes are placed uniformly within a cylindrical storage volume of ground. There is convective heat transfer within the pipes, and conductive heat transfer to the storage volume. The temperature in the ground is calculated from three parts; a global temperature, a local solution, and a steady-flux solution.

Three U-tube vertical boreholes of fifty meters depth placed in parallel compose our GHE. The boreholes are filled with bentonite. Finally, description parameters for the GHE used in the simulation are shown in table 1.

Table 1 Description parameters of the Ground Heat Exchangers.

Parameter	**Value**
Number of boreholes	3
Borehole length	50 m
Borehole radius	0.1016 m
Storage thermal conductivity	2 W/m K
Storage Heat Capacity	2016 kJ/m^3/K
Outer radius of U-tube pipe	0.01664 m
Inner radius of U-tube pipe	0.01372 m
Centre to centre half distance	0.0254 m
Fill thermal conductivity	1.3 W/m K
Pipe thermal conductivity	0.42 W/m K

Water to Water Heat Pump (WWHP). The water to water heat pump is the component that supplies thermal energy to the office area. It is a reversible heat pump; when the system is working in heating mode this device extracts heat from the ground and uses it to heat the offices. When the system works in cooling mode makes the opposite action.

Now, we describe the way in which the model calculates the heat pump power consumption from the operational variables. By definition, heat pump power consumption is equal to the ratio between the heat pump capacity, G, and its nominal coefficient of performance, COP. Nevertheless, the model takes into account that the heat pump is working at partial load. In this case, the actual heat pump power consumption is calculated multiplying this theoretical consumption by the fraction of full load power, f_{flp} , which is a function of the heat pump partial load ratio, PLR (ratio between the load, Q_{load}, and the heat pump capacity, G). Then, the actual consumption for the heat pump is defined as the following expression:

$$P_{HP} = \frac{G}{COP} f_{flp}\left(PLR\right) \tag{1}$$

These three quantities needed to calculate P_{HP} are related on different ways with the operational variables. The heat pump capacity, G, and the coefficient of performance, COP, are calculated using the following expressions:

$$G = G_{rate} G_{ratio}\left(T_{set}, T_{ground,out}\right) \tag{2}$$

$$COP = COP_{rate} COP_{ratio}\left(T_{set}, T_{ground,out}\right) \tag{3}$$

First, G_{rate} and COP_{rate} are the heat pump capacity and the coefficient of performance both in nominal conditions. In table 2, we include the values for G_{rate} and COP_{rate} used in our simulations. And second, G_{ratio} and COP_{ratio} are dimensionless coefficients calculated from the values of the operational variables T_{set}, set point heat pump temperature, and $T_{ground,out}$, temperature of the water returning from the GHE. These quantities are read from two data files and characterize the behaviour of the heat pump for the different values of the operational conditions. Another data file is employed to relate the fraction of full load power, f_{flp}, and the partial load ratio, PLR.

Fan Coil (FC). A central fan coil is used to heat or cool the office area. The simplified fan coil model considers that the final air temperature is the average temperature of the fluid in the coil. Note that the inlet air temperature to the coil is the ambient temperature plus the heating effect due to the heat losses of the fan electric motor. The heat absorbed by the fluid is the same extracted from the air minus the losses due to condensation as can be read from equation (4).

$$Q_{fluid,fc} = \dot{m}_{air,fc}\left(h_{air,in,fc} - h_{air,out,fc}\right) - \dot{m}_{con,fc} h_{cond,fc} \tag{4}$$

The electrical consumption of this device comes from the fan electric motor, and it is computed as follows:

$$P_{fan} = P_{rated,fan}\left(\frac{\dot{m}_{air}}{\dot{m}_{rated,air}}\right)^2 \tag{5}$$

where $P_{rated,fan}$ and $\dot{m}_{rated,air}$ are the rated fan power consumption and the rated air mass flow when the fan is operating at full-speed (see table 2). Finally, $\dot{m}_{air}$ is the air mass flow through the fan in each time step.

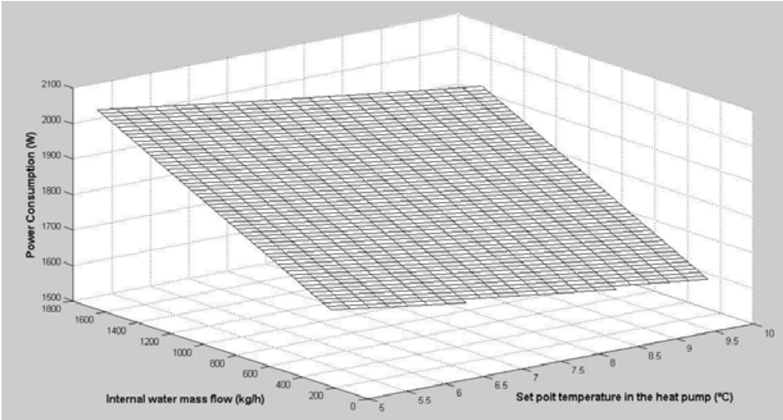

Figure 3 Electrical power consumption for the heat pump and the internal water pump when the heat pump supplies 5 Kw in cooling mode with a heat pump inlet temperature from the source of 24ºC.

Water Pump (IWP, EWP). We use variable speed water pumps for the internal and external hydraulic circuit. A pressure regulator will adjust the pump speed, therefore, the pressure drop keeps constant in the hydraulic circuit. As a consequence, the electric pumping consumption is linear with respect to the water mass flow through the pump. This linear relation is defined in the following expression:

$$P_{pump} = P_{rated,pump} \frac{\dot{m}_{water}}{\dot{m}_{rated,water}} \tag{6}$$

where $P_{rated,pump}$ and $\dot{m}_{rated,water}$ are the rated pump power consumption and the rated water mass flow when the pump is operating at full capacity (see table 2). Finally, $\dot{m}_{water}$ is the water mass flow through the pump in each time step.

Table 2 Value of the parameters for the heat pump, fan coil and water pumps

Device	Parameter	Value
WWHP	Rated G in heating Rated G in cooling Rated COP T_{set} in cooling T_{set} in heating	10 kW 9 kW 3 5 – 10ºC 35 – 45ºC
FC	Rated air mass flow EMF rated power	2700 kg/h 366 W
IWP	Rated water mass flow Rated power	1650 Kg/h 400 W
EWP	Rated water mass flow Rated power	1500 Kg/h 360 W

Electrical of the air conditioning system

The addition of the consumption from the different devices is the total electrical power consumption of our air conditioning system. Equation (7) allows us to calculate this total power consumption at every time step. In this expression, the constant coefficients C_{FC}, C_{IWP}, C_{EWP} and C_{WWHP} can be obtained by looking in the previous electrical consumption expression for each device (see reference (1), (5) and (6)).

$$\begin{aligned} P_{total} &= C_{FC}\dot{m}_{air}^2 + C_{IWP}\dot{m}_{IWP} + \\ &+ C_{EWP}\dot{m}_{EWP} + C_{WWHP}\frac{G_{ratio}(T_{set},T_{ground,out})}{COP_{ratio}(T_{set},T_{ground,out})}\cdot \\ &\cdot f_{flp}(T_{load,in},T_{ground,out},T_{set},\dot{m}_{IWP}) \end{aligned} \tag{7}$$

This equation shows the relation between the electrical consumption of the air conditioning system and six system variables: air mass flow ($\dot{m}_{air}$), internal water mass flow ($\dot{m}_{IWP}$), external water mass flow ($\dot{m}_{EWP}$), heat pump set point temperature (T_{set}), outlet temperature of the ground heat exchanger ($T_{ground,out}$) and heat pump inlet temperature from the load ($T_{load,in}$). Out of these six variables, the first four are management variables ($\dot{m}_{air}$, $\dot{m}_{IWP}$, $\dot{m}_{EWP}$, T_{set}). A proper understanding of the behaviour of this power consumption expression is the key to design a strategy that will improve the energy efficiency of the system.

The consumptions of the fan coil electric motor and of the water pumps only depend on the air mass flow and the water mass flow respectively, and they are independent of the electric consumptions of the other devices. In contrast, the consumption of the heat pump depends on its set point temperature and the internal water mass flow and this last variable is included in the consumption equation of the internal water pump.

Let study this expression for given conditions of $T_{ground,out}$ and $T_{load,in}$. The non-trivial behaviour of equation (7), corresponding to the heat pump plus water pumping power consumption is shown in figure 3. The figure shows the overall electric consumption of these two devices as a function of the internal water mass flow and the set point for the outgoing temperature in the cold side when delivering 5 KW in cooling mode with a heat pump inlet temperature from the ground heat exchanger of 24ºC.

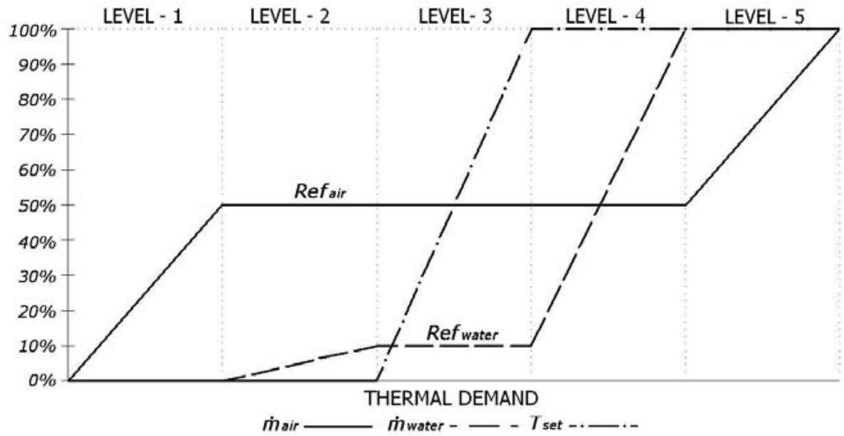

Figure 4. Diagram illustrating the classification in capacity levels of the system capacity given by the steady state values of the management variables in the new management strategy.

From Figure 3, we see that to achieve energy savings it is desirable to work with low water flows and high set point temperatures. Notice that the change in electric power consumption is bigger in the direction of the internal water mass flow than in the set point temperature of the heat pump. Therefore, if it is necessary to supply more energy it is more convenient to modify first the set point temperature keeping low water mass flows. Once the set point temperature achieves its minimum allowed value the water pump starts to increase the water mass flow rate.

In addition to the conclusions derived from figure 3, it would be desirable to have similar flows in both sides of the heat pump. Therefore, the ratio between internal and external water mass flow must be the same with the time. Finally, notice that the electric motor of the fan is the unique device which can be switched on with independence of the heat pump. The other devices, the internal and external water pumps along with the heat pump have to be switched on all together.

Management strategy

This section discusses the two management strategies: the conventional one versus a new one based on the behaviour of equation (7).

For both systems, we have fixed two aspects: the working period and the heating and cooling seasons. The working period in the office is fixed from 9:00 to 18:00, which is the period when is switched on the air conditioning system. The heating season is considered from January to March and from November to December and the cooling season from April to October. Finally, there are not holiday periods; therefore, the air conditioning system works every day.

New management strategy (N.M.S.): The objective of the new management strategy is to improve the energy efficiency of the air conditioning system by adapting its thermal capacity to the actual thermal comfort demand in the office. The new management strategy is detailed in the following paragraphs.

To maintain neutral comfort conditions in the office the PMV value should be zero in the Thermal Comfort Scale. The deviation of the PMV from zero indicates a variation of the thermal demand in the office area. Notice that to satisfy this thermal demand there are several configurations of the management variables that the system can achieve in steady state conditions which will be suitable to compensate this PMV deviation. Our management strategy is based on the choice of a particular configuration of the management variables. This choice classifies the air conditioning system capacity in five capacity levels given by the steady state value of the management variables. In figure 4, we show a diagram illustrating this classification. We now describe each capacity level.

- First capacity level; in steady state conditions the fan capacity is between the 0% and the 50% of its maximum capacity and the other devices are switched off. In this level the blown air by the fan modify the convective factor and homogenize the temperature in the office to achieve the desired thermal comfort, PMV equal to zero.
- Second capacity level; the fan, the hydraulic pumps and the heat pump are switched on. In steady state conditions, the air blown by the fan is fixed to the 50% of its maximum capacity. The water mass flows of the internal and external hydraulic pumps are between the 0% and the 10% of its maximum allowed value. And the set point temperature of the water to water heat pump is 0% of its capacity level, which indicates the lowest or highest set point temperature for heating or cooling mode respectively.
- Third capacity level; in steady state conditions the air blown by the fan is fixed to the 50% of its maximum capacity. The water mass flows of the internal and external hydraulic pumps are fixed to the 10% of its maximum allowed value. And the set point temperature of the water to water heat pump is between 0% and 100% of its

capacity level, meaning that the set point temperature can be any value of its range in heating or cooling mode.

- Fourth capacity level; in steady state conditions the air blown by the fan is fixed to the 50% of its maximum capacity. The water mass flows of the internal and external hydraulic pumps are between the 10% and the 100% of its maximum allowed value. And the set point temperature of the water to water heat pump is 100% of its capacity level, which indicates the highest or lowest set point temperature for heating or cooling mode respectively.
- Fifth capacity level; in steady state conditions the fan capacity is between the 50% and the 100% of its maximum capacity. The water mass flows of the internal and external hydraulic pumps are fixed to the 100% of its maximum allowed value. And the set point temperature of the water to water heat pump is 100% of its capacity level.

If the energy supplied by the air conditioning system when all the management devices are given the 100% of its capacities is not enough to maintain the thermal comfort conditions, the PMV diverts from zero and the thermal demand is not satisfied.

The choice of these five capacity levels is based on the behaviour of equation (7). This equation, explained in previous section, indicates that to achieve energy savings is desirable to work with low water mass flows in the air conditioning system. For this reason, the new management strategy tries to achieve a steady state in which the water mass flow is maintained as low as possible, using first all other possibilities to supply energy.

We also want to point out that in actual conditions a high air flow blown by the fan can produce an excessive noise due to the vibration in the fan coil, to avoid this effect, the new management strategy tries to maintain in steady state conditions the air mass flow blown by the fan around half of its maximum capacity.

Conventional management strategy (C.M.S.): In the following paragraphs, we are going to describe the conventional management strategy for the air conditioning system.

This strategy uses on/off regulators to manage the air conditioning system. This kind of regulators has only two possible operation points. They give its maximum capacity when switched on and nothing when switched off. The on/off regulators are installed in active elements: the water to water heat pump, the electric motor in the fan coil and the internal and external water pumps.

In the conventional management strategy, three aspects are important. First, the difference between the set point temperature and the inlet temperature of fluid in the water to water heat pump load from the office area indicates the connection or disconnection of the generator energy system, composed by the heat pump, the external water pump and the ground heat exchanger. Therefore, the internal water pump is always switched on to have a measure of this difference during the working period. Second, to maintain the comfort conditions the PMV is located in a comfort band between 0.5 and -0.5, in order to try to avoid an excessive number of connections and disconnections of the system, which in a real situation, could damage the actuators of the fan coil. Finally, the set point temperature in the heat pump is constant, for the heating season is fixed to 45ºC and for the cooling season to 7ºC.

In heating mode, the conventional management strategy works as follows. When the value of the PMV is below than the lower limit of the comfort band, PMV equal to -0.5, the electric motor of the fan is switched on. Then, the air goes through the coil, where a constant water mass flow is pumped from the internal circulation pump. This heat exchange produces a variation of the temperature of the water in the internal circuit, which is detected by the water to water heat pump. Immediately, this device and the external water pump are switched on to provide the necessary energy to fix the value of the PMV to the upper limit of the comfort band, PMV equal to 0.5. After achieving this value, the heat pump, the external water pump and the electric motor of the fan are switched off.

In cooling mode the conventional management strategy is the same except that the references of the comfort band are inverted. The electric motor of the fan is connected when the PMV is higher than the upper limit of its comfort band, PMV equal to 0.5, and the air conditioning system stops supplying energy when the PMV arrives to the lower limit of its comfort band, PMV equal to -0.5.

RESULTS AND DISCUSSION

In this section, we present the energy consumption of the air conditioning system for both management strategies, and we evaluate the energy savings achieved by the new one. In the simulation conditions, the weather database employed models the Mediterranean coast weather, which is characterized to have hot summers and warm winters and the working period coincides when the external ambient temperature and the solar radiation is the highest throughout the day. The combination of these two factors produces that the energy demand in cooling mode is much higher than in heating mode.

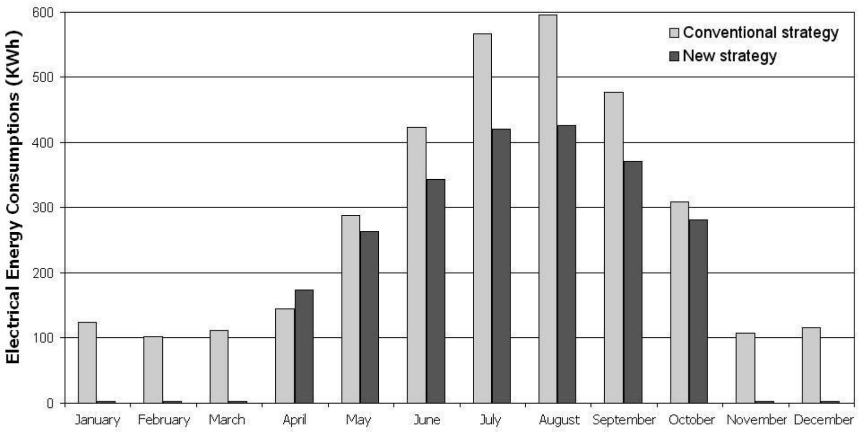

Figure 5 Electrical energy consumptions for both management strategies.

These results are presented from simulations with time step equal to 1.8 seconds. We want to point out that we have performed a convergence study of the results for the following time steps: 14.40 seconds, 7.20 seconds, 3.6 seconds, 1.8 seconds and 1.44 seconds. And we have obtained a clear tendency towards a value independent of the time step employed.

In figure 5, we present the monthly electrical energy consumptions of the air conditioning system for the two management strategies. From this figure we can see that in all months, except April, the system consumes less electrical energy when it is managed by the new management strategy.

In winter season, our simulation results show that the influence of the external temperature and the solar radiation in the office area during the working period is enough to maintain the thermal comfort in it, therefore the energy provided by the air conditioning system is very low. A remarkable difference between both management strategies exist because the conventional management strategy employs the temperature change of the water in the internal hydraulic circuit to communicate the energy demand in the office room to the generator system. Therefore, the internal water pump is always switched on during the working time and, therefore, consuming electrical energy independently of the air conditioning necessities. Whereas, in the new management strategy, the air conditioning system only consumes electrical energy according with the thermal demand in the office area. In this way, we avoided the unnecessary electrical consumptions, which can suppose a great energy lost throughout the year.

In summer season, the influence of the external temperature and the solar radiation, which were an advantage during the winter season, are now a disadvantage. Therefore, both management strategies are more active to provide the necessary high cooling power to achieve the thermal comfort. In these conditions is when the new one has more chances to manage the air conditioning system to reduce its electrical consumption. In figure 5, the electrical consumption for both management strategies grows up in function of the cooling demand, being the maximum electrical consumptions during the warmest months, July and August. In all months during the cooling season, except April, the electrical consumptions of the air conditioning system when is employed the new management strategy are significantly smaller than the ones achieved when using the conventional management strategy. Furthermore, the new management strategy improves the efficiency of the air conditioning system as the cooling demand increases.

In April, the electrical consumption of the air conditioning system is smaller when the conventional management strategy is used. This behaviour is because in the first day of this month the air conditioning system changes from heating mode to cooling mode. When the conventional management strategy manages the air conditioning system the value of the PMV index is between 0.0 and 0.5 most of the time belonging to the working periods. In these conditions, this management strategy keeps switched off the fan, the water to water heat pump and the external water pump. Meanwhile, the new one activates these three devices to supply energy to the office to compensate the small deviation of the PMV from the thermal comfort, PMV equal to zero. This shows us the difficulty to choose the suitable dates to change from heating mode to cooling mode and vice versa which are particulars for each system. The previous situation can be avoided by delaying the date to activate the cooling mode of the heat pump.

Table 3 Annual electrical consumptions, in KWh, of the air conditioning system for both management strategies and energy savings (E.S) achieved.

T. step	C.M.S	N.M.S	E. S. (%)
14.4	3365.86	2106.27	37.42
7.2	3365.96	2525.08	24.98
3.6	3365.67	2163.34	35.72
1.8	3365.24	2277.83	32.31
1.44	3359.16	2289.22	31.85

We include in table 3 the annual electrical consumption of the air conditioning system when it is managed by the conventional management strategy and by the new one. We also include in this table the percentage of energy savings achieved by the new one in comparison with the conventional. These results are presented for all solver time steps used in the simulations.

In summary, looking at the results obtained for the different solver time steps used in the simulations, the annual energy savings obtained through the new management strategy are always above 24%. Furthermore, looking at the behaviour of these numerical results it is observed a tendency towards a value around the 30%.

CONCLUSION

In this paper, we present a new management strategy based on total electrical power equation of the air conditioning system driven by a ground coupled heat pump. The air mass flow in the fan, the water mass flow in the internal and external hydraulic system and the set point temperature in the heat pump are managed to obtain the desired comfort in the office while reducing the electrical consumption of the air conditioning system.

The annual electrical energy savings achieved by the air conditioning system managed by the new management strategy are above the 24% of the electrical consumption of the system managed by a conventional one. This result is the lower bound obtained from the simulations. Nevertheless, we have observed a continuum tendency of the savings towards a value around a 30%.

The presented new management strategy can be easily adapted to other buildings with similar characteristics and conditions and the energy savings achieved should be of the order of the savings obtained in this work.

ACKNOWLEDGEMENT

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NOMENCLATURE

P_{pump}: Water pump electrical power consumption

$P_{rated,pump}$: Rated water pump electrical consumption

P_{fan}: Fan electrical power consumption

$P_{rated,fan}$: Rated fan electrical power consumption

Q_{load} : Load met by the heat pump

P_{HP}: Heat pump electrical power consumption at current conditions

G: Heat pump capacity at current conditions

G_{rate}: Heat pump rated capacity

G_{ratio}: Heat pump capacity at current conditions divided by the rated capacity

COP: Heat pump coefficient of performance at current conditions

COP_{rate}: Heat pump rated coefficient of performance at current conditions

COP_{ratio}: Heat pump COP at current conditions divided by the rated COP

f_{flp}: Heat pump fraction of full load power

PLR: Heat pump partial load ratio

T_{set}: Desired outlet temperature of fluid in the heat pump fluid stream (Set point temperature)

$T_{load,in}$: Inlet fluid temperature in the heat pump

$T_{ground,out}$: Outlet temperature of fluid from the ground coupled heat exchanger

$\dot{m}_{water}$: Mass flow rate of fluid passing through the circulation pump

$\dot{m}_{rated,water}$: Maximum mass flow rate of fluid that can pass through the circulation pump

$\dot{m}_{water,IWP}$: Mass flow rate of fluid passing through the internal circulation pump

$\dot{m}_{water,EWP}$: Mass flow rate of fluid passing through the external circulation pump

Cp_{water}: Specific heat of fluid

$\dot{m}_{air}$: Mass flow rate of air passing through the fan

$\dot{m}_{rated,air}$: Maximum mass flow rate of air that can pass through the fan

$\dot{m}_{con,fc}$: Flow rate of condensate from the coil

$h_{air,in,fc}$: Enthalpy of air entering the coil

$h_{air,out,fc}$: Enthalpy of air exiting the coil

$h_{con,fc}$: Enthalpy of condensate draining from the coil