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► **To cite this version:**

N. Pardo, Á. Montero, A. Sala, J. Martos, J.F. Urchueguía. Efficiency improvement of a ground coupled heat pump system from energy management. *Applied Thermal Engineering*, 2010, 31 (2-3), pp.391. 10.1016/j.applthermaleng.2010.09.016 . hal-00692337

HAL Id: hal-00692337

<https://hal.science/hal-00692337>

Submitted on 30 Apr 2012

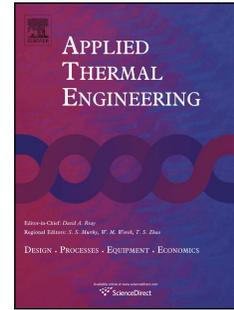
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Accepted Manuscript

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PII: S1359-4311(10)00403-5

DOI: [10.1016/j.applthermaleng.2010.09.016](https://doi.org/10.1016/j.applthermaleng.2010.09.016)

Reference: ATE 3243

To appear in: *Applied Thermal Engineering*

Received Date: 26 February 2010

Revised Date: 14 July 2010

Accepted Date: 20 September 2010

Please cite this article as: N. Pardo, Á. Montero, A. Sala, J. Martos, J.F. Urchueguía. Efficiency improvement of a ground coupled heat pump system from energy management, *Applied Thermal Engineering* (2010), doi: [10.1016/j.applthermaleng.2010.09.016](https://doi.org/10.1016/j.applthermaleng.2010.09.016)

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Efficiency improvement of a ground coupled heat pump system from energy management

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ABSTRACT

The installed capacity of an air conditioning system is usually higher than the average cooling or heating demand along the year. So, most of the time, the system is working under its actual capacity. In this contribution, we study the way to improve the efficiency of a ground coupled heat pump air conditioning system by adapting its produced thermal energy to the actual thermal demand. For this purpose, an air conditioning system composed by a ground coupled heat pump and a central fan-coil linked to an office located in a cooling dominated area was simulated, and a new management strategy aiming to diminish electrical consumption was developed under the basic constraint that comfort requirements are kept. This strategy takes advantage of the possibility of managing the air flow in the fan, the water mass flows in the internal and external hydraulic systems, and the set point temperature in the heat pump to achieve this objective. The electrical consumption of the system is calculated for the new management strategy and compared with the results obtained for a conventional one, resulting in estimated energy savings around 30%.

Keywords: Ground coupled heat pump; Energy efficiency; Numerical simulation.

1. - INTRODUCTION

It is a well known fact that building equipments are responsible of around the 40% of all greenhouse gas emissions produced in developed countries, being approximately 60% produced by cooling and heating systems [1-3]. In this context, the development of air conditioning (HVAC) systems should be oriented to improve its energy efficiency and to reduce its environmental impact.

Ground Coupled Heat Pumps (GCHP) systems represent a technically viable technology for heating, cooling and domestic hot water systems in buildings [4-9]. They offer several interesting characteristics for the potential user, such as a lower electrical demand and maintenance requirements than conventional air to water heat pump systems and, therefore, lower annual cost [10-13]. From a non economical point of view, they offer competitive levels of comfort compared with standard technologies, reduced noise levels and visual impact, savings of greenhouse gas emissions and reasonable environmental safety. Furthermore, the Environmental Protection Agency (EPA) recognizes ground source systems among the most efficient and comfortable heating and cooling systems available today [14]. A proof of the goodness of this technology is the amount of installed units worldwide, estimated in 1.1 million [15].

In the standard design of an air conditioning system, the references taken to estimate the heating and cooling capacity of the system to be installed are usually based on the coldest and the warmest day along the year. A ground coupled heat pump air conditioning system can be designed to meet the whole load or just a part of it, using an auxiliary system to achieve comfort conditions in days in which thermal demand is higher than the capacity of the heat pump system. In both cases, the thermal energy required by the thermal load is under the actual capacity of the system during most part of the time. In this context, the development of strategies for the operation of the air conditioning system and, particularly, for the system based on the ground coupled heat pumps, allowing to adapt the thermal energy generated by the system with the thermal load is a good way to improve the system energy efficiency while satisfying the thermal comfort [16-19]. An approach to achieve this objective based in the combination of the ground coupled system with other production system and in the decoupling of energy production from energy distribution was presented in [20]. This contribution tries to achieve the same objective just taking advantage of the possibility of managing the operational variables of the ground coupled heat pump system.

Following this idea, 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 linked to a standard office space. In this new management 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 heat pump, usually fixed in conventional strategies, have the possibility of a continuous regulation that allows us to design a more efficient way to achieve the desired thermal comfort.

This new management strategy is based on five capacity levels developed from the total electrical power equation of the HVAC system. In our particular case, this equation indicates that to achieve energy savings is desirable to work with low water flows. For this reason, the new management strategy tries to achieve a steady state in which the water mass flow is maintained at the lower level that guarantees the production of enough thermal energy to satisfy the thermal demand.

An office space in a cooling dominated area is modelled to evaluate in these conditions the energy performance of the ground coupled heat pump HVAC system when it is managed by the new management strategy and by a conventional one. The thermal comfort criterion which has to be satisfied is the Predicted Mean Vote index (PMV). This comfort index predicts the mean value of votes of a large group of people in the Thermal Sensation Scale and it is defined by the ISO7730-1994 standard [21].

We compare the annual electrical energy consumption of the HVAC system when it is managed by our new management strategy and by a conventional one and an evaluation of its energy efficiency is presented. We also analyze the factors which allow improving the energy efficiency of the air conditioning system by the new management strategy.

Nomenclature

P_{pump} : Circulation pump electrical power consumption

$P_{rated,pump}$: Rated circulation pump electrical power 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} : Set point temperature – Desired outlet temperature of fluid in the heat pump fluid stream

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

$T_{GHE,in}$: Inlet temperature of fluid to the ground heat exchanger

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

T_a : Temperature of the surrounding ground

β : dumping factor

α_v : heat transfer coefficient between the fluid and the surrounding ground per unit of volume

α_p : heat transfer coefficient between the fluid and the surrounding ground per unit length of pipe

V : Volume of storage

L_p : total pipe length in the storage volume

C_f : Volumetric heat capacity of the fluid

Q_f : Total fluid flow rate in the ground 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 draining 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

2. - SIMULATED SYSTEM

In this section, we present a description of the simulated office area as well as the PMV index which is the criterion used to evaluate the thermal comfort. Afterwards, we describe the ground coupled heat pump system linked to the office area, the energy model of the employed devices, the new management strategy and the conventional one. Finally, these systems are simulated using TRNSYS software package [22].

2.1. - 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/m²K. The floor and the roof are built with hollow blocks with 5cm of insulation; its global conductivity is 0.51 W/m²K. Finally, the window is composed by a glass, with solar radiation transmissivity equal to 0.837 and conductivity equal to 5.74 W/m²K, and a window frame with conductivity equal to 0.588 W/m²K. The windows size is 1.5 m², dedicating a 15% of this area to the frame surface. The internal and external shadow factor for these windows is estimated in 0.7.

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.

2.2. - 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. The PMV index for the air conditioned space is calculated at each simulation step time. To do it, three parameters, activity, thermal resistance of clothing and mean air velocity has to be estimated. 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.

2.3. - Air Conditioning System and Electric Power Consumptions

The air conditioning system linked to the simulated office area is based on 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 1 a schematic diagram showing the layout of these devices is included.

FIGURE 1

The only difference between the hydraulic circuit for the new management strategy and the conventional management strategy is in the active devices (WWHP, EWP, IWP, EMF) employed. For the new one the active devices have a continuous regulation, whereas for the conventional one these devices can be only 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 its electrical power consumption behaviour (see below eq. (9)) allows us to develop a suitable management strategy to achieve this objective.

Ground Heat Exchanger (GHE). ‘Duct Ground Heat Storage Model’ [23] is used to simulate our GHE, composed by three U-tube vertical boreholes of fifty meters depth placed in parallel with a distance between them of 5.3 meters. This model is implemented in TRNSYS library as type557. To estimate the size of the ground heat exchanger the method proposed in reference [24] was employed. U-tube pipes are characterized by an outer radius equal to 0.01664 m., an inner radius equal to 0.01372 m., a center to center half distance equal to 0.0254 m., and a pipe thermal conductivity equal to 0.42 W/m K. Boreholes of radius equal to 0.1016 m. are filled with bentonite (with thermal conductivity equal to 1.3 W/m K). Finally, ground thermal

properties are characterized by a storage thermal conductivity equal to 2 W/m K, and a storage heat capacity equal to 2016 kJ/m³/K.

To calculate the electrical energy consumption of the water to water heat pump the outlet temperature of the ground heat exchanger ($T_{GHE,out}$) is needed. This temperature is computed by the 'Duct Ground Heat Storage Model' as:

$$T_{GHE,out} = \beta \cdot T_{GHE,in} + (1 - \beta) \cdot T_a \quad (1)$$

Where β is a dumping factor defined as follows:

$$\beta = \exp\left(-\frac{\alpha_v \cdot V}{C_f Q_f}\right) = \exp\left(-\frac{\alpha_p \cdot L_p}{C_f Q_f}\right) \quad (2)$$

Water to Water Heat Pump (WWHP). The model employed for the water to water heat pump (type 666 from TRNSYS library) calculates its power consumption from the operational variables. The heat pump power consumption (P_{HP}) is calculated as the quotient between its capacity at current conditions (G) and its coefficient of performance at current conditions (COP), all multiplied by the fraction of full load power, f_{flp} :

$$P_{HP} = \frac{G}{COP} f_{flp} \quad (3)$$

The fraction of full load power is a function of the heat pump Partial Load Ratio (PLR), the ratio between the thermal load and heat pump capacity at current conditions. 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:

$$G = G_{rate} G_{ratio}[T_{set}, T_{GHE,out}] \quad (4)$$

$$COP = COP_{rate} COP_{ratio}[T_{set}, T_{GHE,out}] \quad (5)$$

G_{rate} and COP_{rate} are the heat pump capacity and the coefficient of performance both in nominal conditions. In table 1, the values for G_{rate} and COP_{rate} used in our simulations are included. G_{ratio} and COP_{ratio} are dimensionless coefficients calculated from the values of the operational variables T_{set} , set point heat pump temperature, and $T_{GHE,out}$, temperature of the water returning from the GHE. These quantities are read from two data files (standard files from TRNSYS library) and characterize the behaviour of the heat pump for the different values of the operational conditions. Another data file (also available together with type 666 from TRNSYS library) is employed to relate the fraction of full load power, f_{flp} , and the partial load ratio, PLR .

Fan Coil (FC). The fan coil model (type 774 from TRNSYS library) 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 (6).

$$Q_{fluid,fc} = \dot{m}_{air,fc} (h_{air,in,fc} - h_{air,out,fc}) - \dot{m}_{con,fc} h_{cond,fc} \quad (6)$$

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

$$P_{fan} = P_{rated,fan} \left(\frac{\dot{m}_{air}}{\dot{m}_{rated,air}} \right)^2 \quad (7)$$

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. In table 1, the values for $P_{rated,fan}$ and $\dot{m}_{rated,air}$ used in our simulations are included. Finally, \dot{m}_{air} is the air mass flow through the fan in each time step.

Fan power consumption is calculated in common software tools as a function of the percent design airflow using a second order equation [25]. The choice of equation (7) is done to be representative of a generic case. Other choices in which the second order equation slightly differs from equation (7) produces a very low impact in energy performance results, confirming that this choice can be considered as representative of a generic case.

Water Pump (IWP, EWP). We use variable speed water pumps for the internal and external hydraulic circuit. The pump speed will be adjusted by a pressure regulator so that 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:

$$P_{pump} = P_{rated,pump} \frac{\dot{m}_{water}}{\dot{m}_{rated,water}} \quad (8)$$

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. In table 1, the values for $P_{rated,pump}$ and $\dot{m}_{rated,water}$ used in our simulations are included. Finally, \dot{m}_{water} is the water mass flow through the pump in each time step.

TABLE 1

Electrical consumption of the air conditioning system. The addition of the consumption from the different devices is the total electric power consumption of our air conditioning system. Equation (9) 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 expressions for each device, equations (3), (7) and (8).

$$P_{total} = C_{FC}\dot{m}_{air}^2 + C_{IWP}\dot{m}_{IWP} + C_{EWP}\dot{m}_{EWP} + C_{WWHP} \frac{G_{ratio}[T_{set}, T_{GHE,out}]}{COP_{ratio}[T_{set}, T_{GHE,out}]} f_{fp}[T_{load,in}, T_{GHE,out}, T_{set}, \dot{m}_{IWP}] \quad (9)$$

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}), heat pump inlet temperature from the ground heat exchanger ($T_{GHE,out}$) 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 management strategy that will improve the energy efficiency of the system.

The consumptions of the electric motor of the fan coil and 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 $T_{GHE,out}$ and $T_{load,in}$. The non trivial behaviour of equation (9), corresponding to the heat pump plus water pumping power consumption is shown in figure 2. This 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 24°C.

FIGURE 2.

From figure 2 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 higher 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 should increase its water mass flow.

In addition to the conclusions derived from figure 2, 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 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. And the internal and external water pumps along with the heat pump have to be switched on all together.

2.4. - Management Strategies

In this section we present both management strategies: the new one based on the behaviour of equation (9) and the conventional one.

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 November to December and from January to March and the cooling season from April to October. Finally, there are not holiday periods, as a consequence the air conditioning system works every day.

2.4.1. – New management strategy

The objective of the new management strategy is to improve the energy efficiency of the air conditioning system by adapting properly its generated energy to the actual thermal 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 which are suitable to compensate this PMV deviation. Our management strategy is based on the choice of a particular configuration of these 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 3, we show a diagram illustrating this classification. We now describe each capacity level.

FIGURE 3.

- 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 thermal

comfort.

- 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 is 0%, 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 is between 0% and 100%, 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 is 100% 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 is 100%.

If the energy supplied by the air conditioning system when all the active 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 (9). This equation, explained in previous section, indicates that to achieve energy savings is desirable to work with low water 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 at the lower level that guarantees the production of enough thermal energy to satisfy the thermal demand.

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.

A suitable way to implement this new management strategy is through a cascade control structure because allows to quickly achieve the thermal comfort state and, after this, drive the management variables to the steady state of the capacity levels described before without comfort penalties. In the simulations, the action of the controllers has place every 28.8 seconds, which is fast enough in comparison with the building dynamic behaviour.

2.4.2. - Conventional management strategy

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 heat pump, the electric motor of the fan coil and the internal and external water pumps.

In this strategy, three aspects are important. First, the difference between the set point temperature and the inlet temperature of fluid in the heat pump 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 an actual 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 45°C and for the cooling season to 7°C.

In heating mode the conventional management strategy works as follows. When the value of the PMV variable is below 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 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 as the previous one except that the references of the comfort band are inverted. The electric motor of the fan is connected when the PMV is above the upper limit, PMV equal to 0.5, and the air conditioning system stops supplying energy when the PMV arrives to -0.5.

3. - SIMULATION RESULTS AND DISCUSSION

In the following paragraphs, we present and discuss the results obtained from the simulations. First, we evaluate the accuracy of the obtained results. And second, we present the energy consumption of the air conditioning system for both management strategies, evaluating the energy savings achieved by the new one.

To evaluate the accuracy of the simulation results a set of simulations are carried out for several time steps.

The obtained results have to show a tendency towards a value independent of the time step employed to solve the set of differential equations programmed in our software package. Simulations are performed at the following solver time steps: 14.40 seconds, 7.20 seconds, 3.6 seconds, 1.8 seconds and 1.44 seconds.

The results obtained for the conventional management strategy are almost independent of the solver time step chosen, with variations among them always smaller than 1%. Nevertheless, the results from the simulation of the new one show a higher dispersion. This behaviour is due to the fast dynamic introduced by the controllers which are not possible to simulate correctly when the solver time step is not small enough. Indeed, it is a well-known fact that simulating multiple levels of control with widely different time constants leads to a system of stiff ordinary differential equations, for which the numerical integration routines in TRNSYS may not be specially adapted: this is the reason of needing a time step around 1 second in order to reliably simulate a whole year. In spite of this, the tendency of the electrical energy consumptions in all the simulations is the same through the year. In the cases where is used the smallest time steps, 1.80 seconds and 1.44 seconds, the obtained results do not show significant variation between them (always smaller than 3%) and they point out to the same continuum values.

Now, 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.

In figure 4 we present the monthly electrical energy consumptions of the air conditioning system for the two management strategies. These data correspond to the simulations employing a solver time step equal to 1.44 seconds. 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.

FIGURE 4

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 during most days of the winter season, therefore the energy provided by the air conditioning system is very low. A remarkable difference between both management strategies exists because the conventional 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. As a consequence, 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. Then, the unnecessary electrical consumptions, which can suppose a great energy loss throughout the year, are avoided.

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 management strategy has more chances to manage the air conditioning system to reduce its electrical consumption. In figure 4, the electrical consumption for both 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 one are significantly smaller than the ones achieved when using the conventional one. Furthermore, the new 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 strategy keeps switched off the fan, the heat pump and the external water pump. Meanwhile, the new strategy activates these three devices to supply energy to the office to compensate the small deviation of the PMV from the neutral thermal comfort state. 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.

Finally, the annual electrical consumption of the air conditioning system when it is managed by the conventional management strategy is estimated in 3359 kWh, and in 2289 kWh when managed by the new one (data for solver time step equal to 1.44 s.). Then, the annual energy savings achieved are estimated in a value around the 30%.

4. - CONCLUSIONS

In this paper, a new management strategy for a HVAC system composed by a ground coupled heat pump and a central fan coil linked to an office area is presented. The air mass flow in the fan, the water mass flow in the internal and external hydraulic systems 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. This new management strategy is based on five capacity levels developed from the total electrical power consumption equation of the HVAC system. This equation indicates that to achieve energy savings is desirable to work with low water 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 at the lower level that guarantees the production of enough thermal energy to satisfy the thermal demand.

The total power consumption of the HVAC system when it is managed by the new strategy and by a conventional one are compared, obtaining that the annual electrical consumption is reduced in a value around the 30% when the air conditioning system is managed by the new management strategy.

To conclude, the energy savings obtained in this study are particular of the chosen conditions of our simulations. Nevertheless, we think that this improvement in the energy efficiency of the HVAC systems

driven by ground coupled heat pumps is an estimation of the effect when these kinds of systems are managed by the new management strategy. The presented new management strategy can be easily adapted to other areas with similar characteristics and conditions and the energy savings achieved should be of the order of the savings obtained in this work.

5. - ACKNOWLEDGEMENTS

This work has been supported by the Spanish Government under projects “Modelado y simulación de sistemas energéticos complejos” (2005 Ramón y Cajal program), “Modelado, simulación y validación experimental de la transferencia de calor en el entorno de la edificación” (ENE2008-0059/CON). A. Sala is grateful to the financial support of grants DPI2008-06731-c02-01 (Spanish Government), and Generalitat Valenciana prometeo/2008/088.

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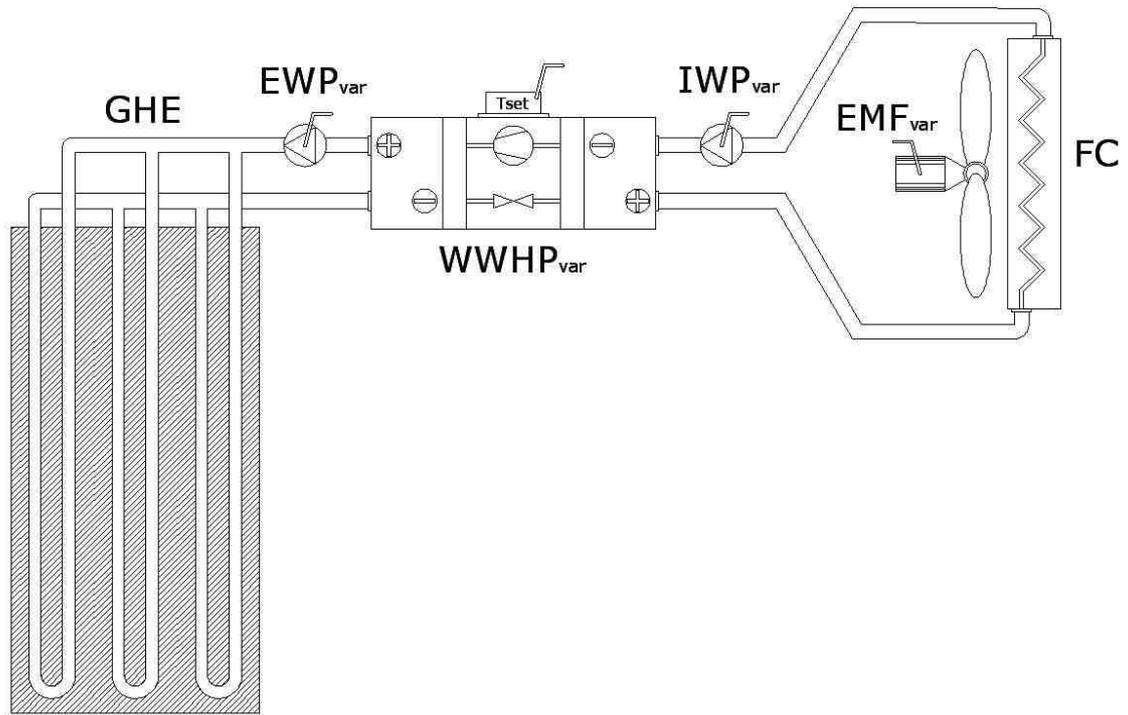


Figure 1. Air conditioning system: the geothermal heat exchanger (GHE), the external water pump (EWP), the internal water pump (IWP), the water to water heat pump (WWHP) and the fan coil (FC) with the electric motor of the fan (EMF).

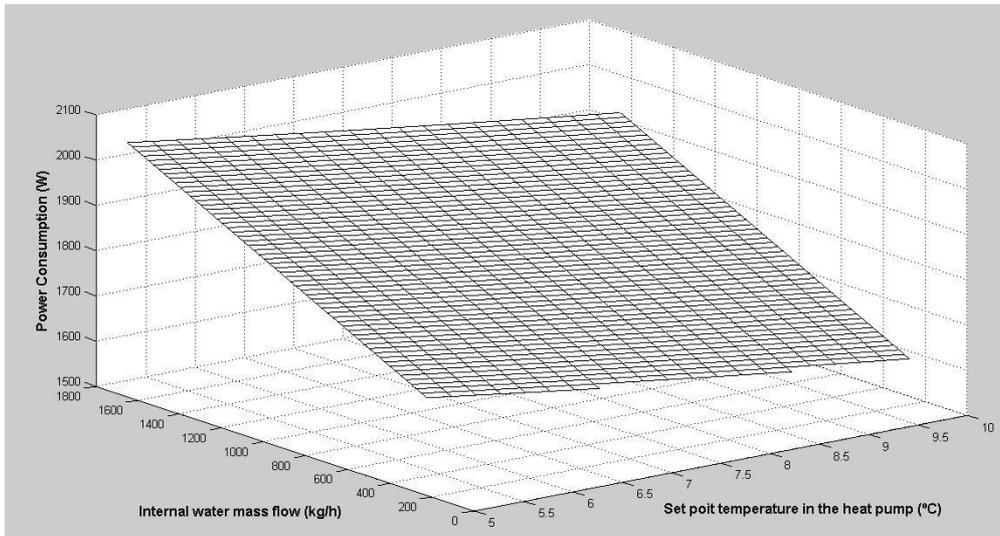


Figure 2. Electrical power consumption for the heat pump and the internal water pump when the heat pump supplies 5 kW in cooling mode with a heat pump inlet temperature from the ground heat exchanger of 24°C.

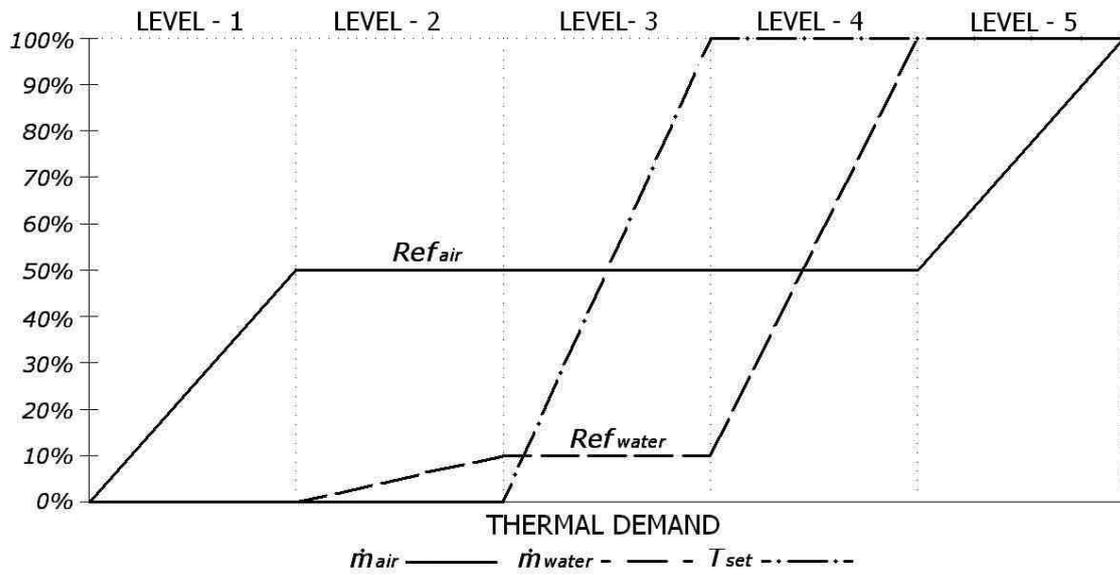


Figure 3. Diagram illustrating the classification in capacity levels of the system capacity given by the steady state values of the management variables. For a given thermal demand there is a unique choice for the operational point of the air conditioning system.

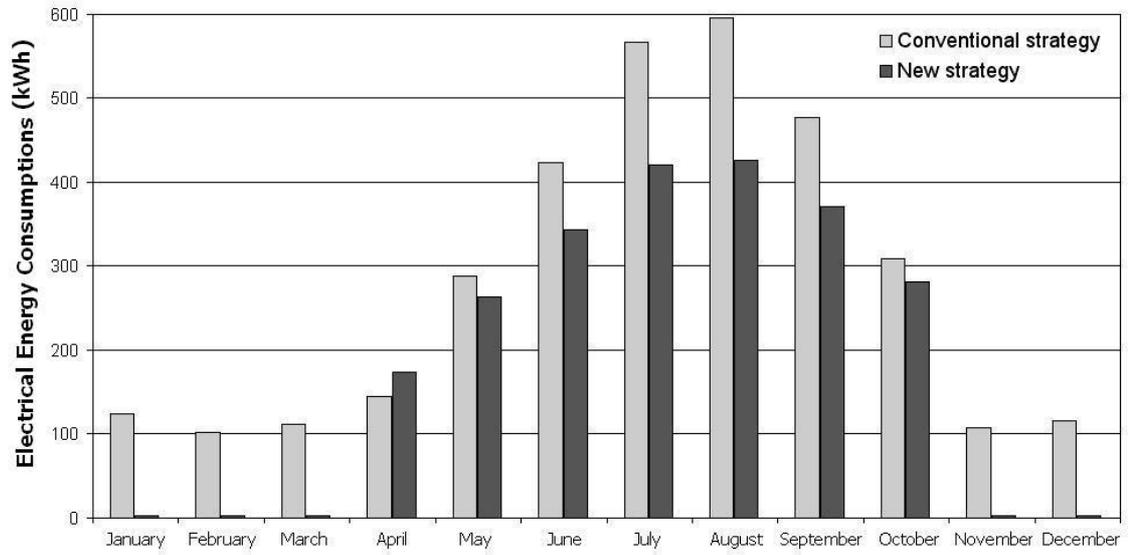


Figure 4. Electrical energy consumptions for both management strategies.

Device	Parameter	Value
Heat pump	Rated capacity in heating of the heat pump ($G_{rate,heating}$)	10 kW
	Rated capacity in cooling of the heat pump ($G_{rate,cooling}$)	9 kW
	Rated coefficient performance of the heat pump (COP_{rate})	3
	Set point temperature in cooling ($T_{set,cooling}$)	5 – 10°C
	Set point temperature in heating ($T_{set,heating}$)	35 – 45°C
Fan Coil	Rated air mass flow for the fan coil ($\dot{m}_{rated,air}$)	2700 kg/h
	Rated power for the electric motor of the fan ($P_{rated,fan}$)	366 W
Water Pump	Rated water mass flow for the internal water pump ($\dot{m}_{rated,water,int}$)	1650 Kg/h
	Rated water mass flow for the external water pump ($\dot{m}_{rated,water,ext}$)	1500 Kg/h
	Rated power for the internal water pump ($P_{rated,pump,int}$)	400 W
	Rated power for external water pump ($P_{rated,pump,ext}$)	360 W

Table 1. Value of the parameters for the heat pump, the fan coil and the water pumps.