Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 136 (2017) 91–96

Energy Procedia

www.elsevier.com/locate/procedia

4th International Conference on Energy and Environment Research, ICEER 2017, 17-20 July 2017, Porto, Portugal

# Energy study in water loop heat pump systems for office buildings in the Iberian Peninsula

F. J. Fernández[a,*], M. B. Folgueras[a], I. Suárez[a]

[a]*University of Oviedo, Polytechnic School of Engineering, C/Wifredo Ricart s/n, Gijón 33204, Spain*

**Abstract**

The energy consumption is analyzed for the air conditioning of an office building in four important cities of the Iberian Peninsula. A water loop heat pump (WLHP) system is compared with a conventional water system. Energy redistribution is an important advantage, but significant savings come from heat pumps high efficiency parameters and minor air flow rates in the cooling tower. Even using natural gas as energy source, 8.1% decrease of $CO_2$ emissions is reached, but additional important reduction can be easily obtained by using a solar thermal energy system as energy source.


Peer-review under responsibility of the scientific committee of the 4th International Conference on Energy and Environment Research.

*Keywords:* Water loop heat pump; heating ventilating and air conditioning; office building energy consumption.

## 1. Introduction

Water loop heat pumps (WLHP) systems are usual in air conditioning of commercial and office buildings. In this scheme one water loop circuit receives energy from the condensation and yields it to the evaporation of reversible heat pumps that attend thermal loads of different zones of the building. The net necessary energy to keep the water loop temperature in a range can be obtained from gas boilers or other energy production systems and dissipated by cooling towers. One important advantage of these systems is the transfer of energy between zones of the building when serving

* Corresponding author. Tel.: +34-985182112; fax: +34985182143.
*E-mail address:* javierfernandez@uniovi.es

1876-6102 
Peer-review under responsibility of the scientific committee of the 4th International Conference on Energy and Environment Research.
10.1016/j.egypro.2017.10.292

loads of opposite sign. Besides, heat pumps using the water loop as a source, work with very good efficiencies, EER in refrigeration or COP in heating mode.

Some references to WLHP systems can be found in specialized literature. Their performance has been analyzed in representative weathers in China [1] and in several European climatic areas [2]. Yuan and Grabon [3] optimized their working parameters by mathematical modelization.

The present study analyzes the behavior of a WLHP system in a common office building under climatic conditions of four important cities in the Iberian Peninsula. Energy consumptions of this system and other more conventional system are compared. It is a water system with four tube connection design to allow simultaneous heating and cooling loads, fan-coils, gas boiler and a conventional chiller. The objective is to obtain important information that can be useful to reduce energy consumption in WLHP systems for HVAC.

## 2. Calculation of energy demands and systems energy consumptions

### *2.1. Energy demand in an office building*

A regular office building was studied to obtain the detailed energy demand profiles corresponding to four representative cities. The building has three occupied plants, as well as not habitable attic and ground floor zones. Total inhabited gross area is 918 m$^2$, and it has external zones with four orientations and an inner zone, as shown in Fig. 1 (a) and (b).

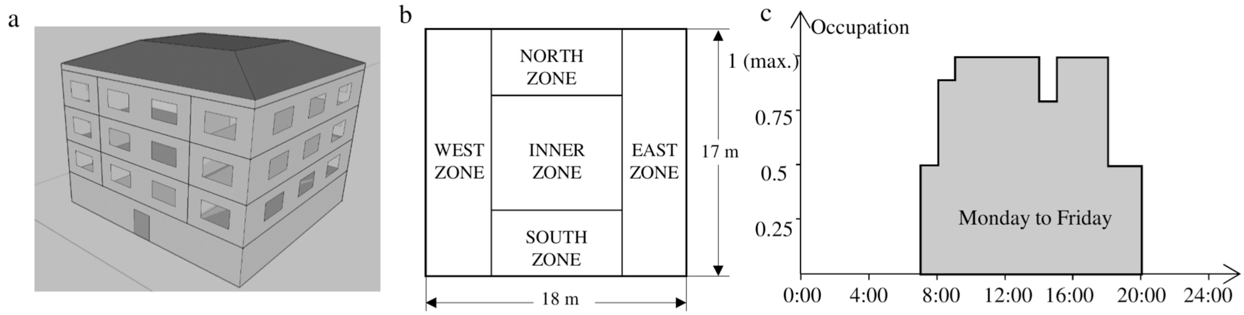


Fig. 1. (a) External view of the building; (b) distribution of thermal zones in the occupied plants; (c) office occupation profile.

Typical compositions were chosen for the opaque surfaces, resulting in 0.517 W/m$^2$-K U-factor values for walls and 0.563 W/m$^2$-K for floors and ceilings. U-factor for windows is 2.9 W/m$^2$-K and its solar heat gain coefficient is 0.72. Internal heat gains were included for people, lighting and office equipment: people activity was estimated in 130 W/p, with an occupation density of 12 m$^2$/p; the heat gain from lighting was fixed in 7 W/m$^2$ and 8 W/m$^2$ internal heat gain for the office electric equipment. The ventilation air volume was fixed in 12 l/s-p. These are maximum values for people, lighting, equipment and ventilation loads. Their profile follow the occupation schedule in Fig. 1 (c). Constant air infiltration values were fixed for not habitable zones: 2 air changes per hour in the attic and 3 changes per hour in the ground floor. The loads were calculated in ideal air loads mode, with cooling and heating thermostat schedules that keep air temperature in a range between 21 and 25ºC in working hours, from Monday to Friday, and humidity controls to keep humidity ratios between 45 and 55% in the same schedule.

Calculations were performed with the EnergyPlus [4] simulation software, using the OpenStudio$^{(R)}$ [5] platform to define the building, loads and weather data. The energy demand was calculated for climatic conditions of four cities in the Iberian Peninsula: Madrid, Barcelona, Zaragoza and Porto. The weather data files were obtained from the EnergyPlus site, choosing weather data from the SWEC (Spanish Weather for Energy Calculations) database for Spanish cities and from the IWEC (International Weather for Energy Calculations) for the Porto weather data.

## 2.2. HVAC systems

The energy consumption necessary to attend these energy demands will be compared for a WLHP system and a conventional 4 tube fan-coil water HVAC system. Fig. 2 shows two simplified configuration schemes for these systems. One storage tank is included in the water loop system.

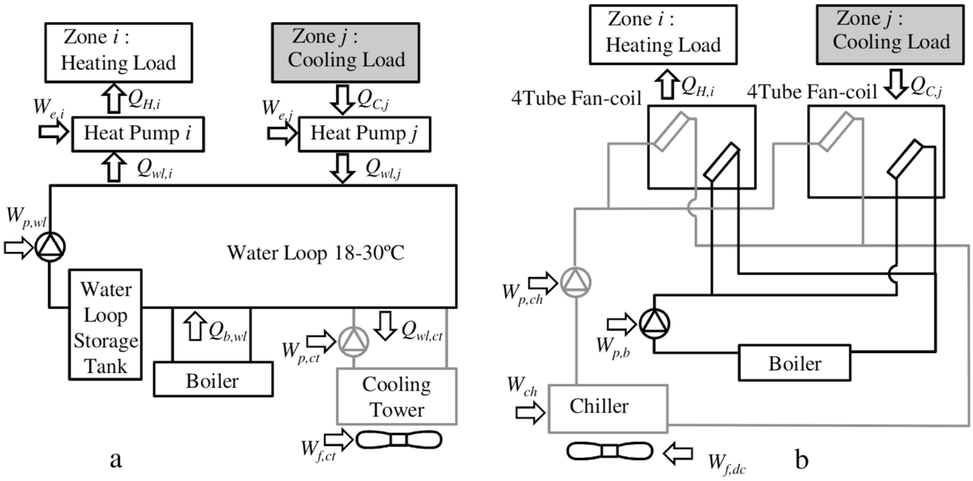


Fig. 2. (a) Water Loop Heat Pump system; (b) 4Tube fan-coil water HVAC system.

The energy consumption in the WLHP system is estimated by equations (1) to (9):

$$Q_{hp,wl} = \sum_{j(Cooling)} \frac{Q_{C,j}(EER_j+1)}{EER_j} - \sum_{i(Heating)} \frac{Q_{H,i}(COP_i-1)}{COP_i} \tag{1}$$

$$COP_i = \xi_C \frac{T_{cond,i}}{T_{cond,i}-T_{evap,i}}; \;\; (T_{cond,i} = 50\ ^{\circ}\mathrm{C};\ T_{evap,i} = T_{wl} - 8\ ^{\circ}\mathrm{C}) \tag{2}$$

$$EER_j = \xi_C \frac{T_{evap,j}}{T_{cond,j}-T_{evap,j}}; \;\; (T_{cond,j} = T_{wl} + 8\ ^{\circ}\mathrm{C};\ T_{evap,j} = 2\ ^{\circ}\mathrm{C}) \tag{3}$$

$$\begin{cases} Q_{wl,ct} = Q_{hp,wl}\ ;\ Q_{b,wl} = 0 & \text{for: } Q_{hp,wl} \geq 0 \\ Q_{wl,ct} = 0\ ;\ Q_{b,wl} = \Delta t \cdot P_b & \text{for: } Q_{hp,wl} < 0 \end{cases} \tag{4}$$

$$\frac{\Delta T_{wl}}{\Delta t} = \frac{Q_{hp,wl}+Q_{b,wl}-Q_{wl,ct}}{M_{wl} \cdot c_{p,l}} \tag{5}$$

$$W_{e,T} = \sum_{j(Cooling)} \frac{Q_{C,j}}{EER_j} + \sum_{i(Heating)} \frac{Q_{H,i}}{COP_i} + W_{p,wl} + W_{p,ct} + W_{f,ct} \tag{6}$$

$$Q_T = \frac{Q_{b,wl}}{\eta_b} \tag{7}$$

$$W_{p,wl} = \frac{g \cdot \Delta H_{wl}}{\eta_p} \sum_i \frac{Q_{wl,i}}{\Delta T_i \cdot c_{p,l}}\ ;\ W_{p,ct} = \frac{g \cdot \Delta H_{ct}}{\eta_p} \frac{Q_{wl,ct}}{\Delta T_{ct} \cdot c_{p,l}} \tag{8}$$

$$W_{f,ct} = \frac{\Delta P_{air,ct} \cdot Q_{wl,ct}}{\eta_f \cdot \rho_{air} \cdot \Delta h_{air,ct}} \tag{9}$$

The calculations are performed in terms of energy analysis in time step periods $\Delta t$ of one hour. Equations (4) and (5) are applied regarding that the boiler and cooling tower avoid the water loop temperature $T_{wl}$ to get out of the 18-30 ºC range, allowing free oscillation inside this range. Thermal power of the boiler $P_b$ is slightly greater than the maximum needed. Coefficient of performance (COP) and the energy efficiency ratio (EER) are estimated with Equations (2) and (3) through the Carnot limit values with an approximation factor $\xi_C$=0.5. The mass of the water loop storage tank $M_{wl}$ is fixed 10 kg/m$^2$, per habitable surface area. Water temperature change in the heat pumps, $\Delta T_i$ is 5 ºC. The water range in the cooling tower $\Delta T_{ct}$ is 8 ºC. The boiler thermal efficiency is $\eta_b$=0.95. The energy consumption for pumping is estimated by Equation (8), with the demanded mass of water for each time step period, with pressure losses of 6 meter water height for the water loop ($\Delta H_{wl}$) and 12 meter for the cooling tower ($\Delta H_{ct}$) circuits. The total pumping efficiency is $\eta_p$=0.45 in both cases. The estimation of the cooling tower fan consumption is based in the evacuated energy, with the air enthalpy change ($\Delta h_{air,ct}$) in the cooling tower from ambient conditions to saturated air conditions at 35 ºC, with a fan total efficiency factor $\eta_f$=0.35, and 250 Pa of air pressure drop through the tower. The equivalent procedure was defined to evaluate the energy consumptions of the conventional 4 tube fan-coil water HVAC system:

$$W_{e,T} = \frac{1}{EER_{ch}} \sum_{j(Cooling)} Q_{C,j} + W_{p,ch} + W_{p,b} + W_{f,ct} \tag{10}$$

$$EER_{ch} = \xi_C \frac{T_{evap,ch}}{T_{cond,ch} - T_{evap,ch}}; \;\; (T_{cond,ch} = T_{amb} + 15\text{ °C};\; T_{evap,ch} = -1\text{ °C}) \tag{11}$$

$$Q_T = \frac{1}{\eta_b} \cdot \sum_{i(Heating)} Q_{H,i} \tag{12}$$

$$W_{p,ch} = \frac{g \cdot \Delta H_{ch}}{\eta_p} \sum_{j(Cooling)} \frac{Q_{C,j}}{\Delta T_j \cdot c_{p,l}} \; ; \; W_{p,b} = \frac{g \cdot \Delta H_b}{\eta_p} \sum_{i(Heating)} \frac{Q_{H,i}}{\Delta T_i \cdot c_{p,l}} \; ; \tag{13}$$

$$W_{f,dc} = \frac{\Delta P_{air,dc} \cdot (EER_{ch} + 1) \cdot \sum_{j(Cooling)} Q_{C,j}}{\eta_f \cdot \rho_{air} \cdot \Delta h_{air,dc} \cdot EER_{ch}} \; ; \tag{14}$$

The meaning of parameters in equations (10) to (14) is similar to the previous ones. The pressure losses for the heating and cooling circuits ($\Delta H_b$, $\Delta H_{ch}$) are 6 meter water height. Temperature changes in the fan-coils ($\Delta T_i$) are 5 ºC for cooling and 15 ºC for heating. The air pressure drop $\Delta P_{air,dc}$, is 150 Pa in chiller dry condenser, and its enthalpy change, $\Delta h_{air,dc}$, is a sensible heating from ambient conditions to 45 ºC. The fan, pump and boiler efficiencies ($\eta_f$, $\eta_p$, $\eta_b$) and the Carnot approximation factor $\xi_C$ have the same values than in the WLHP calculations.

## 3. Results

The monthly and annual heating and cooling energetic demands are shown in Fig. 3 for the four selected cities. The results of the energy consumptions of both systems, WLHP and 4 tube fan-coil water system, are summarized in Table 1.

Two important parameters were selected to analyze the reduction of the total environmental impact of the WLHP system compared to the 4 tube fan-coil water system: the total consumed nonrenewable primary energy, *NRPE* and the total $CO_2$ emissions. The conversion factors were taken from the official Spanish *Institute for the Diversification and Energy Saving* (IDAE) [6], as expressed in Equations (15) and (16). For the scope of comparison, the thermal energy was considered to be generated through natural gas combustion in both cases.

$$NRPE = 1.190 \cdot Q_T + 1.954 \cdot W_{e,T} \tag{15}$$

$$kg\ CO_2 = 0.252 \cdot Q_T + 0.331 \cdot W_{e,T} \tag{16}$$

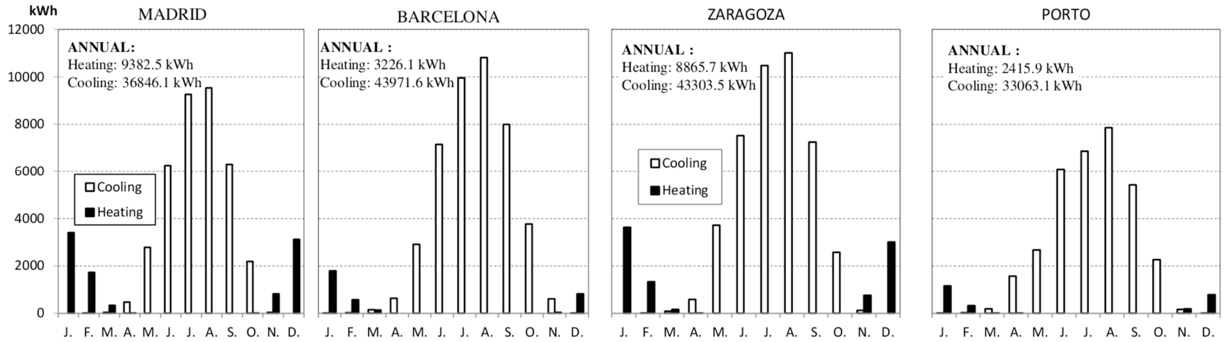


Fig. 3. Energy demand for air conditioning in the office building in: Madrid, Barcelona, Zaragoza, Porto.

Table 1. Annual results for energy consumption and mean efficiency parameters.

| Location | Madrid | Barcelona | Zaragoza | Porto |
|---|---|---|---|---|
| **WLHP System** | | | | |
| Total thermal consumption ($Q_T$), kWh | 7279.7 | 2443.3 | 6786.6 | 1751.6 |
| Total electrical consumption ($W_{e,T}$), kWh | 12974.4 | 13550.1 | 14728.9 | 10139.7 |
| Nonrenewable primary energy (*NRPE*), kWh | 34014.8 | 29384.5 | 36856.4 | 21897.3 |
| $CO_2$ emissions, kg | 6129.0 | 5100.8 | 6585.5 | 3797.6 |
| Heat pumps seasonal COP | 4.05 | 4.07 | 4.05 | 4.08 |
| Heat pumps seasonal EER | 3.82 | 3.82 | 3.82 | 3.82 |
| Energy evacuated by the cooling tower ($Q_{wl,ct}$), kWh | 46337.3 | 55287.4 | 54399.6 | 41552.8 |
| Water loop pump consumption ($W_{p,wl}$), kWh | 335.2 | 362.9 | 383.7 | 272.5 |
| Cooling tower pump consumption ($W_{p,ct}$), kWh | 362.5 | 432.5 | 425.6 | 325.1 |
| Cooling tower fan consumption ($W_{f,ct}$), kWh | 318.8 | 434.0 | 405.2 | 300.6 |
| Max. boiler power ($P_{b,max}$), kW | 23.32 | 16.28 | 25.79 | 12.47 |
| Max. cooling tower dissipation rate ($P_{ct,max}$), kW | 80.55 | 98.74 | 104.94 | 79.52 |
| **4Tube Fan-coil Water System** | | | | |
| Total thermal consumption ($Q_T$), kWh | 9876.3 | 3501.1 | 9332.3 | 2543.0 |
| Total electrical consumption ($W_{eT}$), kWh | 12629.3 | 14306.8 | 15001.7 | 9856.3 |
| Nonrenewable primary energy (*NRPE*), kWh | 36430.5 | 32121.9 | 40418.7 | 22285.4 |
| $CO_2$ emissions, kg | 6669.1 | 5617.8 | 7317.3 | 3903.3 |
| Chiller seasonal EER | 3.23 | 3.38 | 3.20 | 3.67 |
| Heating circuit pump consumption ($W_{p,b}$), kWh | 19.6 | 6.9 | 18.5 | 5.0 |
| Cooling circuit pump consumption ($W_{p,ch}$), kWh | 230.6 | 275.2 | 271.0 | 206.9 |
| Dry condenser fan consumption ($W_{f,dc}$), kWh | 969.1 | 1004.0 | 1163.8 | 641.1 |
| Max. boiler power ($P_{b,max}$), kW | 32.62 | 22.77 | 36.08 | 17.44 |
| Max. chiller cooling rate ($P_{ch,max}$), kW | 22.78 | 24.82 | 29.26 | 21.83 |

The main savings of the WLHP system are in thermal consumptions, while the electrical consumptions are more similar for both systems. The seasonal EER of the heat pumps in WLHP are higher than the chiller seasonal EER in the water system. The seasonal COP of heat pumps have good values, above 4.0. Higher consumptions were found for pumping in WLHP systems, but they are compensated by the lower fan consumption of the cooling tower compared to the chiller dry condenser. The resulting savings in terms of nonrenewable primary energy and $CO_2$

emissions have been important, especially in the locations with more cooling demand, where they reach values around 8%. The total mean savings are 6.9% in nonrenewable primary energy and 8.1% in $CO_2$ emissions.

## 4. Conclusion

Calculation of the energy demands of a conventional office building has been performed by means of the building energy simulation software EnergyPlus. The result has allowed the evaluation of the energy consumption of two different systems in four important cities of the Iberian Peninsula. The developed model will help in the adjustment of the design parameters for WLHP systems, as the temperature range of the water loop or the size of its thermal storage. Addition of renewable energy sources will also be considered.

The office buildings have important internal loads, so the energetic analysis resulted in cooling demands much higher than the heating ones, even in locations with severe winter conditions as Madrid or Zaragoza. Not as much coexistence of heating and cooling demands was found, so the advantage of energetic redistribution was not so present. Nonetheless, other advantages allowed important energetic savings of the WLHP compared with the conventional 4 tube fan-coil water system. The use of evaporative cooling towers and the water loop stable temperature as the energy source for heat pumps resulting in high COP and EER values are also important advantages of this systems. The same energy sources were selected for comparison of both systems, but energy inputs in the WLHP system can be easily obtained from solar thermal energy or from geothermal energy systems. Our study of WLHP systems continues with the incorporation of these renewable energetic sources together with the parametric adjustment of their design parameters, with the aim of optimize their advantages, minimizing the environmental impact of building air conditioning.

## References

[1] Lian, Zhiwei, Seong R. Park, and Henian Qi. (2005) "Analysis on energy consumption of water-loop heat pump system in China." *Applied Thermal Engineering* 25 (2005): 73-85.

[2] Buonomano, Annamaria, Francesco Calise, and Adolfo Palombo. (2012) "Buildings dynamic simulation: Water loop heat pump systems analysis for European climates." *Applied Energy* 91 (2012): 222-234.

[3] Yuan, Shui, and Michel Grabon. (2011) "Optimizing energy consumption of a water-loop variable-speed heat pump system." *Applied Thermal Engineering* 31 (2011): 894-901.

[4] www.energyplus.net

[5] www.openstudio.net

[6] www.idae.es