Energy performance of a ventilation system for a block of apartments with a ground source heat pump as generation system

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Abstract. This work presents a numerical simulation of the annual performance of two different systems: a traditional one composed by a gas boiler-chiller pair and one consisting of a ground source heat pump (GSHP) both coupled to two thermal storage tanks. The systems serve a block of flats located in northern Italy and are assessed over a typical weather year, covering both the heating and cooling seasons. The air handling unit (AHU) coupled with the GSHP exhibits excellent characteristics in terms of temperature control, and has high performance parameters (EER and COP), which make conduction costs about 30% lower than those estimated for the traditional plant.

1. Introduction

Building energy consumption amounts to almost 40% of Europe’s yearly energy needs and contributes to around 36% of CO\textsubscript{2} emissions within the European Union. In order to reduce these figures, EU member States are required to implement energy efficiency measures for buildings in compliance with the Energy Performance of Buildings Directive [1]-[3].

Different strategies can be pursued to achieve the objective of reducing the fossil primary energy consumption, such as increasing the insulation thickness for the building envelope, using more efficient HVAC systems and exploiting renewable energies [4].

With the increase of insulation thickness for the walls and better airtight window glasses and frames, a growing concern is represented by proper ambient ventilation to ensure high standards of air quality. Owing to the decrease in the overall transmittance of the building envelope, thermal losses due to ventilation constitute a growing share of the total heating/cooling demands, between 20 and 50% depending on building insulation and compactness, air change rate, indoor heat sources, temperature and relative humidity set points and outdoor climatic conditions [5]. Heat Recovery (HRV) from exhaust air in buildings represents a widespread energy saving strategy, which allows to transfer energy between exhaust air and fresh supply air by means of air-to-air heat exchangers integrated within the air handling unit (AHU). Savings in primary energy due to HRV can be significant, depending on the type of air supply system, heat exchanger configuration and power needed to operate the heat recovery unit [6].

Regarding renewable energy, in 2009 the European Parliament recognizes aero-thermal, geothermal and hydrothermal energy as renewable energy source (RES) [7]. The employing of ground-coupled heat pump (GCHP) systems for heating and cooling of new and existing buildings satisfies both the directive
on renewable energy and the reduction of primary energy consumption needs. GCHP can reach higher energy efficiency than those of an air source heat pump system because the soil can provide higher temperature for heating and lower temperature for cooling than air does [8].

The issue of energy saving in HVAC is of particular interest for buildings in locations whose weather conditions on a typical year provide for significant heating and cooling loads, making it necessary to operate the air conditioning system almost continuously over the whole year, switching from heating mode to cooling mode depending on the season. This is the case in Italy for the river plain area in the North, especially its eastern part, where winters are cold and summers are very warm and humid, as for Bologna [9].

Traditionally, AHUs for the maintenance of the thermal comfort in buildings consist of heating coils fed with water heated in a gas boiler and cooling coils fed with water chilled in an air-cooled chiller (when intermediate fluid configuration is preferred instead of one with direct expansion); moreover HRV systems are increasingly integrated within the AHU for a better overall efficiency [10].

The aim of this work is to compare the heating and cooling energy performance of a traditional AHU serving a block of flats in Bologna with those of an AHU employed for the same task but coupled with a ground-source heat pump for the production of both hot and cold water. Both systems analyzed make use of sensible heat recovery systems for the preheating or precooling of supply air, depending on the operation mode, and thermal energy storage tanks in order to uncouple hot and cold production systems from sudden load variations within the building.

A numerical investigation method has been adopted, carrying out simulations over a typical year exploiting the huge potential of TRNSYS, a lumped-parameter simulation program for energy system transient analysis [11].

The temperature control inside the building has been analyzed in terms of displacement from the defined set point; besides, the temperature oscillations inside the heat storage tanks, the consequent cycling and the performance of GSHP in terms of COP and EER have been investigated. Finally, the annual energy consumption and operational costs for the two technical solutions are evaluated and compared.

**Figure 1.** Floor drawing of the block of flats.
2. Simulation

The building chosen for the simulation consists of a block of flats located in Bologna. Its structure is typical of the last decade of the 20th century for this kind of constructions in the area. It is a three-storied building with six flats per floor of three types which differ in the net surface area ($A_1=64.46 \text{ m}^2$, $A_2=42.04 \text{ m}^2$ and $A_3=63.37 \text{ m}^2$ respectively) and with the East-West direction acting as a mirror line. The total net surface of the flats is $A_{nf}=1025 \text{ m}^2$, whereas the total area, including walls, stairs and lifts is $A_{tf}=1369 \text{ m}^2$. The net volume of the building to be conditioned is $V_n=2768 \text{ m}^3$, and the envelope through which heat is exchanged with the ambient is $A_n=1933 \text{ m}^2$. Fig.1 reports a floor drawing of the block of flats.

![Figure 1. Floor drawing of the block of flats.](image)

Figure 2. Sketch of the HVAC systems.

The two HVAC systems investigated serve the whole building and share some common components, namely the storage tanks for hot (HWT) and chilled (CWT) water, as it emerges in Fig.2, where a sketch of the HVAC system in the two configurations has been reported. Each flat is served by a ventilation system supplying treated air to the load by means of an independent air handling unit (AHU), which draws the thermal power needed from the HWT or from the CWT depending on the time of the year. The HRV within the AHU exchanges sensible heat only (i.e. no latent heat due to condensation/evaporation of moisture in the air is recovered) and its efficiency can be calculated as:

$$
\varepsilon_S = \frac{T_{PR}-T_o}{T_e-T_o} 
$$

$\varepsilon_S$ is the sensible heat efficiency, $T_{PR}$ is the supply air temperature after the heat recovery process, while $T_o$ and $T_e$ are respectively the outdoor and the exhaust air temperatures. In this study, a fixed-plate cross-flow heat recovery system has been considered and a value of 0.7 has been adopted for the sensible efficiency, according to indications given in [9] and assuming a medium-to-high efficiency heat exchanger.

The water tanks act as buffer systems, to dampen temperature oscillations due to sudden changes in outdoor conditions and to cover peak loads that might occur. The tanks are modeled as stratified, with a volume capacity $V=4 \text{ m}^3$ each. Depending on the choice of the HVAC system, the HWT and CWT are fed separately by a gas boiler and an air-cooled chiller, or together by a water-to-water heat pump (HP),
which exchanges thermal energy with the ground by means of a ground source heat exchanger (GSHX), where a mixture of water and glycol circulates. On the basis of the design indications given in [12], the geothermal field consists of 10 vertical probes, whose buried length is 100 m; the total surface area occupied by the geothermal field is about 490 m². The nominal thermal power the gas boiler can supply is $P_b = 56$ kW with an overall efficiency $\eta_b = 0.85$. Referring to an existing unit, the chiller can supply a cooling power $P_c = 42.6$ kW with a rated energy efficiency ratio $EER_c = 3.15$, considering water inlet and outlet temperatures of 12°C and 7°C at the evaporator and air temperature of 35°C at the condenser; in the simulation the data of the existing unit were employed in order to establish EER values as a function of the outdoor air temperature, as reported in Fig.3, where EER curves are plotted for different values of the evaporator outlet temperature.

The ground source heat pump was also modeled to replicate an actual machine. The nominal cooling power is $P_{hp_c} = 45$ kW with nominal $EER = 4.10$, when the temperatures of the fluid at the evaporator are 12/7°C and the temperatures of the fluid at the condenser are 30/35°C; the nominal heating power is $P_{hp_h} = 53$ kW with a coefficient of performance $COP = 4.10$, when the temperatures of the fluid at the evaporator are 10/7°C and the temperatures of the fluid at the condenser are 40/45°C. Actual EER and COP curves were deduced from builder’s data to better replicate the machine’s actual behavior.

The temperature set points of water feeding the heat exchangers within the AHU are 40°C and 11°C for heating coil and cooling coil respectively, assuming a dead band of ±2°C for temperature control purpose.

![Figure 3. EER curves for the chiller.](image)

Fig.4 shows EER curves versus condenser inlet temperature for different values of the evaporator inlet temperature, while COP values versus evaporator inlet temperature for fixed condenser inlet temperatures are plotted in Fig. 5 for the working ranges of interest.

![Figure 4. EER curves for the heat pump.](image)  
![Figure 5. COP curves for the heat pump.](image)

The layout of the building-HVAC system for the case of the ground-source heat pump is shown in Fig.6.
The AHU is not shown in Figure 6 as a single block: its components are represented instead, namely the heating and cooling coils, the heat recovery unit, the fan coils and the exhaust air mixer. The ground source heat pump is also shown separately from the borehole heat exchanger loop. The control system to keep temperature in all rooms at their set values is not shown in the picture. A similar setup was employed for the traditional system with gas boiler and chiller, not shown here.

All building details were modelled for the purpose of thermal analysis by determining their U-values, a summary of which is reported in Table 1 together with the corresponding thickness.

| Structural Component | Thickness (m) | U-value (W m⁻² K⁻¹) |
|----------------------|--------------|---------------------|
| Outer Walls          | 0.36         | 0.45                |
| Inner Walls          | 0.24         | 0.54                |
| Ground Floor         | 0.50         | 0.24                |
| Inner Ceilings       | 0.45         | 0.75                |
| Windows              | -            | 2.83                |

Figure 6. Layout of the heat pump system and load.

In the Bologna region, regulations concerning the heating season define it as the period from October 15ᵗʰ to April 15ᵗʰ, while the cooling season is the remaining part of the year. During the heating season, room temperature is set at $T_{\text{set,h}}=20°C$, and heating operates from 6:00 am to 12:00 pm. In the remaining hours, heating is switched on only when temperature in any one room drops below $T_l=18°C$. During the cooling season, the set point temperature is $T_{\text{set,c}}=26°C$, and the system operates in the same hours as in winter, but it is never switched on between 0:00 am and 6:00 am. Over the whole year, whenever outdoor temperature is between 20°C and 26°C, the system operates in free cooling mode. Weather data for the city of Bologna were obtained from the Meteonorm database.
For both system configurations, simulations were run at time steps of 5 minutes over the whole year, spanning both the heating and cooling seasons. Several data were collected as output and the most significant shall be discussed in the following section.

3. Analysis of the results

Temperature control for the rooms was more than satisfactory, as exemplified in Fig.7. The outdoor and bedroom temperatures over a 24-hours period are plotted for two days in winter and two in summer. Two of the chosen days are representative of the average temperature of the period (November 2nd and June 30th for winter and summer respectively) while the remaining two are those when outdoor temperatures reached their extreme values (January 13th and August 18th respectively): for all cases temperature conditions in the bedroom were quite stable, also owing to the capacitive effects of the storage tank. Even when outdoor temperature dropped below the design value of -5°C (night hours of Jan 13th, where it was -6°C on the average) conditions in the room never fell below 18°C. The other rooms had a similar behavior. In Fig.7 it is also possible to notice how bedroom temperatures displayed a smoother trend during days of average conditions while they had a slightly oscillating behavior during the days of extreme conditions.

![Figure 7. Outdoor and room temperature for different days of the year.](image1)

![Figure 8. Temperature of water from storage tank to load.](image2)

To assess the behavior of the storage tank, the temperature of the water fed to the load (building) was analyzed during the same days as for temperature control of the thermal zones, as shown in Fig.8. In this case, temperature related to winter days are higher (as thermal energy must be supplied to the building) than those occurring in summer days (when the thermal load must be removed from the building). During the days representative of the period, the water temperature oscillates regularly between 40±2°C in winter and 11±2°C in summer, which indicates how the tank is working properly as thermal energy buffer. During days with harsher temperature conditions, the amplitude of temperature oscillations is still around ± 2 °C, but frequency decreases, especially in winter, due to the larger energy demands of the load. The trend of temperature in the early hours of Jan 13th, as the set-point increases to 20°C (at 6:00) differs from that of the representative winter day (Nov. 2nd), because of the increased energy demands of the building. The same happens in the summer on Aug 18th, but in the afternoon hours, when outer temperature reaches its maximum.

The performance of the heat pump is analyzed for the representative days in winter and summer. Figure 9 shows the power to load, EER and outdoor temperature for June 30th. It is evident how the thermal energy removed from the load follows the trend of the outdoor temperature: as the latter increase so does the former. Moreover, when the heat pump operates (which corresponds to a decrease of the water temperature to load, with reference to Fig.8) there is an increase in the amount of thermal energy removed. The EER of the heat pump during operating times are rather high, averaging around 5.0. A similar behavior occurs during winter operation, as shown in Fig.10.
this time the power to load and the outdoor temperature have opposite trends: when the temperature outside decreases the thermal energy which needs to be supplied to the building increases. In the same way, while the heat pump operates, the power supplied to load increases due to the increase in the temperature of the water from the storage tank to the load, as shown in Fig. 7.

The values of the coefficient of performance are lower than those obtained for the EER, but they are still satisfactory. This difference is largely due to the smaller temperature difference at which the ground-source heat pump operates during the summer months as opposed to the operating conditions in winter, as can be seen in Fig. 11 and Fig. 12, where the EER and COP trends are shown for different fluid inlet temperatures at the two ends of the heat pump for two representative days, focusing on two operating periods. At the ground side, the fluid inlet temperature is about 20°C in summer and 10°C in winter, with a soil temperature (not shown here) which is about 15°C throughout the year.

The above results show that the geothermal plant and storage tanks are correctly designed to cover the thermal demands of the block of flats over the entire year. The ground source heat pump operates with a thermal reservoir, the soil, which is on the average at a far more stable and higher temperature than ambient air (which is employed in air-to-air and air-to-water heat pumps).

A qualitative analysis on the possibility of soil thermal depletion has also been carried out: during the heating season, the thermal energy subtracted from the ground was estimated in $E_{g_{\text{win}}}=64.416 \text{ MWh}$, while during the cooling season the thermal energy supplied amounts to $E_{g_{\text{sum}}}=43.768 \text{ MWh}$. 

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**Figure 9.** Summer performance of heat pump.

**Figure 10.** Winter performance of heat pump.

**Figure 11.** EER in a representative summer day as function of working fluids inlet temperatures. Particular of an operating period.

**Figure 12.** COP in a representative winter day as function of working fluids inlet temperatures. Particular of an operating period.
Neglecting solar radiation at ground surface and conduction from surrounding soil, a ground temperature decrease is to be expected. The seasonal electricity consumption of the GSHP is \( L_{el\_win} = 22.334 \text{ MWh} \) in winter and \( L_{el\_sum} = 7.324 \text{ MWh} \) in the summer, which yield a seasonal coefficient of performance \( \text{SCOP} = 3.88 \) and a seasonal energy efficiency ratio \( \text{SEER} = 4.97 \). The high values achieved are due to the comparatively high and constant temperature of the outside heat source/sink (the ground) with respect to ambient air.

### Table 2. Electricity and natural gas yearly consumption and costs.

|                          | Traditional | GSHP     | \( \Delta \) | \( \Delta \% \) |
|--------------------------|-------------|----------|--------------|-----------------|
| Annual electricity consumption (kWh) | 37410       | 58809    | +21399       | +57             |
| Annual natural gas consumption (Sm\(^3\)) | 11417       | 0        | -11417       | -100            |
| Total annual cost (€)    | 16615       | 11762    | -4853        | -29             |
| Average annual cost per flat (€) | 923         | 653      | -270         | -29             |

The operating costs for the ground-source heat pump have been compared to those of the traditional heating and cooling solution which employs a gas boiler and an air-cooled chiller for winter and summer conditions respectively. The results are shown in Table 2. The total energy demand is obviously the same for both systems, but the GSHP uses electrical energy only, whereas the boiler-chiller pair employs natural gas to operate in the winter months. The amount of electrical energy in kWh for the chiller and GSHP and the amount of natural gas for the boiler over a year are shown in the second and third row from the top, with the difference reported as absolute and percent values in the last two columns right. The last two rows show the total yearly costs and the yearly costs per flat. The price of electricity and natural gas for Italy have been estimated to be 0.20 € (kWh\(^{-1}\)) and 0.80 € (Sm\(^3\))\(^{-1}\) respectively, taxes and duties included. From the results above it turns out that the use of an AHU coupled with a GSHP instead of a traditional plant brings yearly savings in operating costs of about 30%; however, the cost analysis carried out is not sufficient to justify the installation of such a system, because it must be accompanied with a detailed economic analysis of the initial investment.

### 4. Conclusions

In the present work a ground source heat pump system for heating and cooling a block of flats located in northern Italy (city of Bologna) has been numerically simulated. The energy consumption results have been compared with the ones obtained by employing a traditional system composed by a gas boiler for winter heating coupled with an air-cooled chiller for summer cooling. The simulations show that the yearly primary energy and the operating costs for the ground source heat pump system are lower than the ones for the traditional boiler-chiller combination system. Moreover, the GSHP systems satisfy the requirement of the European directive on the employing of renewable energy technology for space heating and cooling.

### References

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