Energy and economic analysis of an integrated multi-source heat pump system for a school building

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Abstract

In 2009 a new school building was built up in Agordo town, located in northern Italy. The main design features of the building incorporate a well insulated envelope and a space heating and ventilation system driven by an innovative multisource heat pump system. The latter incorporates outdoor air, ground heat, solar heat, and heat recovery, so enhancing the performance of the heat pumps, both in terms of heating capacity and overall efficiency. However the use of different sources requires additional investment costs, so it is worth to investigate the economic advantage of such a solution. This paper presents a study conducted on the base of both energy and economic side: a period of approximately four heating seasons was considered and the behaviour of the system was assessed and the incorrect settings and operation of the plant were identified. The economic balance of the multisource heat pump system is evaluated with respect to a single source heat pump HVAC plant.

Keywords: ground coupled heat pump; heat recovery; multisource system; thermal collector

1. Introduction

Low energy building are based on the reduction of the primary energy demand through a high insulation level, the use of high efficiency heating/cooling systems and the integration of renewable energy sources into the building HVAC plant [1]-[3]. The heat pump is one of the most advantageous systems to be considered. As it is well known, particular care is advisable with regard to the selection of both the heating system (in order to lower heat supply temperature) and the heat source. In particular, this second aspect should be carefully evaluated as the potential advantages of alternative heat sources could be significant. External ambient air is the most diffused but the worst from thermodynamic point of view as the buildings heating loads generally increase as air temperature decreases [4]. This is why there is an increasing interest in dual source systems during the last decades both from the experimental and the theoretical point of view.

The main idea in dual source systems is that the heat pump absorbs heat by two heat sources. Two arrangements widely studied in literature are air source heat pump/solar collectors and ground source heat pump/solar collectors with two typical configurations: “in series” (the two sources are aligned in series so that the former raises the temperature before that heat is taken from the latter) or “dual source” (the heat pump takes heat choosing time by time the most favourable source from the thermodynamic point of view) [5].

The present paper refers to an example of the latter configuration into a real application, the new school building (operative since 2009) in Agordo, a mountain resort in Belluno province – North Italy. In this case a multi-source

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(solar+ground and exhaust ventilation air+ground) absorption heat pumps system was implemented. The paper presents the evaluation and analysis of data obtained through real time monitoring of the HVAC system in operation for the last four heating seasons (from May 2012 till April 2016). The plant was previously monitored in the first period of operation (from end of October 2009 to end of March 2011) and the results of that analysis were already published [6] [7]. It is interesting to analyze the performance of the plant after some years of operation. During this period the behavior of the system has been assessed and incorrect operation of the plant has been identified that penalized the energy performance of the plant. The energy balance indicates that the integration of different sources, when operating, not only increases the thermal performance of the system as a whole, but also optimizes the use of each source, as resulted also from the previous study ([6] [7]).

2. The building and the HVAC plant

The detailed information about the climatic conditions, building composition, building energy load calculations and HVAC plants description are reported in previous works of the Authors ([6]-[8]). Here only the main data are briefly reported. The town of Agordo lies in the geographic area of the Dolomiti mountains (North-East of Italy) in a valley at 611 m a.s.l.; it has a temperate climate (Köppen climate classification Cfb-Cfc) with cold winters (3,376 Heating Degree Days - HDD) and mild or cool summers, frequently associated with abundant rains. Monthly mean air temperatures are -2.9 °C and 18.3 °C in January and July respectively.

The main building features can be shortly indicated in 5680 m² of heated area with an enclosed gross heated volume of 19640 m³. The outdoor conditions used in the following analysis are given by the weather dataset by the Environment Agency of Veneto Region [9], in the form of hourly sampling of outdoor temperature, relative humidity, global solar radiation, wind speed, for the period May 2012-April 2016.

The plant was designed to serve for the sole purpose of heating and ventilation, due to the severe climatic condition and the use of building, closed in summertime; moreover the demand for hot tap water is negligible and is fulfilled by means of electric water heaters. For sake of simplicity a reduced functional diagram of the plant is shown in Fig. 1 reporting only the main hydraulic streams and energy flows within the plant.

The HVAC plant is divided into the space heating section and the ventilation section. The heating section has two ammonia-water absorption heat pumps (HP3 and HP4 in Fig. 1) with geothermal exchangers in parallel (960 m, 6 x 160 in a row, of vertical tube heat exchangers designed with double-U pipes with a outer diameter of 32 mm and a thickness of 2.9 mm), producing thermal energy at 45 °C. Moreover the plant includes the setting up of 50 m² of flat type solar heat collectors (4 arrays in parallel, each of those made of 5 modules in series), and can be operated in four different modes, the control acting on a “threshold radiation” calculated as a function of the solar collectors features and the desired outlet temperature (see next section).

The ventilation section serving the AHU (Air Handling Unit) hot batteries is also made up of two ammonia-water absorption heat pumps (HP1 and HP2 in Fig. 1) with geothermal heat exchangers (750 m, 6 x 125 m in a row, of vertical tube heat exchangers). In order to serve the AHU the heat pumps produce thermal energy at 55-60 °C. The heat recuperators downstream of the AHU are static cross flow type with an efficiency of 50 %.

Component	Rated capacity (kW)	Rated efficiency
HP1 + HP2	74 (B0W60)	1.25 (GUE [10])
HP3 + HP4	76 (B0W40)	1.40 (GUE [10])
Boiler	114.4	1.02 (condensing)

Table 1 - Heating generators of the central HVAC plant.

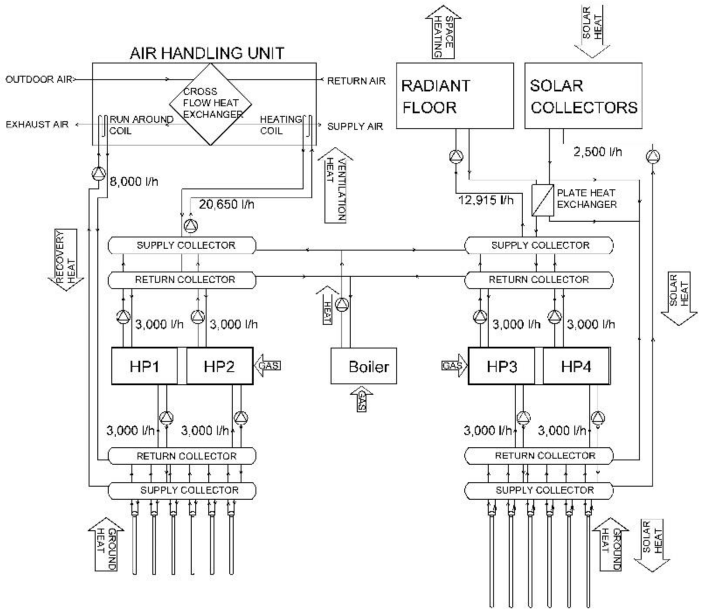

Fig. 1. Reduced functional diagram of the HVAC plant. The main mass and energy flows (gas, ground, solar, recovery) are shown [7].

In order to increase thermal recovery still further when the external temperature exceeded 0 °C, run-around coils were installed at the outlet of two out of four of AHUs (the one of the laboratories and the one of the teaching rooms), for a global volume flow of 20600 m³/h out of 25000 m³/h. The run-around coils can operate on the exhaust flow of a heat exchanger, either sensitive or latent; the heat recovered can be sent to the evaporator of the absorption equipment (see next section for the control logic).

The HVAC system in teaching rooms, laboratories and offices provides space heating by means of radiant floor and ventilation by means of three independent AHU each of those serving a single-duct system. The auditorium is served by an all-air system. From the design calculations, performed according to the UNI EN 12831:2006 Standard, the space heating requires a maximum power of 146 kW and the ventilation system requires 122 kW of sensible heat, being the design indoor conditions set to 20 °C of air temperature and the ventilation being neutral (air supplied at 20 °C).

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Boiler	114.4	1.02 (condensing)

Table 1 reports the main characteristics of the heating generators; the heat pumps were designed to cover the base load, the condensing boiler to supplement the peak load as well as to backup an eventual fault of one or more heat pumps.

3. Operation and monitoring of the HVAC plant

The operation control logic of the HVAC plant as designed is here briefly described ([6]-[8]).

The need for sensible heating of the AHUs depends directly on the outdoor air temperature and on the volume flow, which is constant for every AHU. The teaching room AHU is scheduled to work from 6 am to 2 pm, that of the laboratories from 10 am to 6 pm and that of the offices from 6 am to 6 pm. Heat pumps HP1, HP2 and the boiler are activated once the return temperature falls below the given thresholds. The run-around coil is activated when the temperature of the exhaust flow downstream the cross-flow recuperator rises above 10 °C, which is considered to be the highest potential temperature that the ground can supply.

The need for space heating is determined by the number of active circuits (each room has its own thermostatic on–off control, according to Italian regulations). The same scheduling of the AHUs applies to the heating system. HP3, HP4 and the boiler are activated once the return temperature falls below the given thresholds. The solar system is activated whenever the measured solar radiation on the field exceeds a minimum threshold. If the solar circuit outlet temperature exceeds 38 °C (i.e. the radiant floor supply temperature increased by 3 °C), the solar outlet is connected to the plate heat exchanger, or otherwise the exchanger is bypassed (Fig. 1). Then the solar outlet feeds the HP3 and HP4 evaporators collector, thus increasing the evaporation temperature. When there is no need for space heating, the solar outlet is directed to the borehole heat exchangers.

The monitoring of the plant was set-up with the cooperation of the building's and the plant's controller designers. The oldest data available on the PC that controls the plant, accessible from remote terminal via VPN after authentication, was from the begin of May 2012. During the period May 2012-April 2016 the following cumulative energy flows (mass flow times the temperature difference between the inlet and the outlet, via simple thermal energy meters) were logged hourly and the results are presented in the next section on a monthly basis:

- Condenser and evaporator of each heat pump (at the collectors);
- Ground circuits, separately for ventilation and space heating;
- Primary circuit of AHU heating coils and run-around coils;
- Solar circuit;
- Primary circuit of the radiant floor.

All of the energy meters are located in the central heating plant, and therefore the energy delivered to each circuit was measured as the gross value including distribution losses. At times, the monitoring software did not function properly, so it was necessary to complete missing data using assumptions drawn from general system performance.

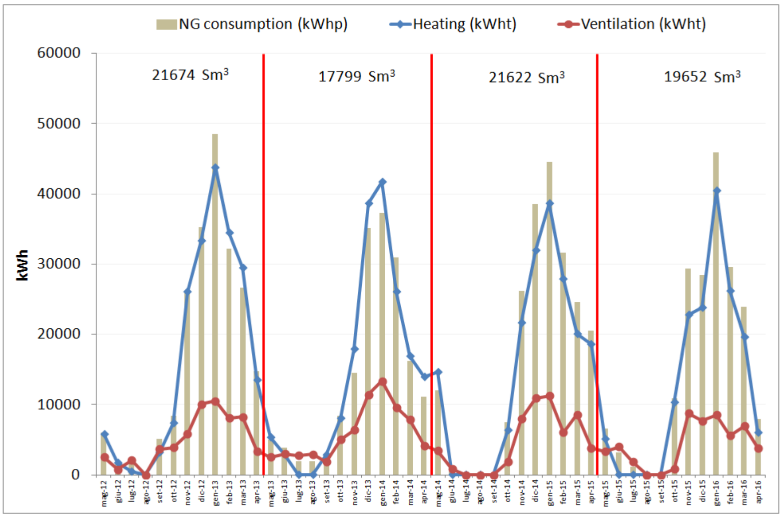

Fig. 2. Monthly Natural Gas consumption (expressed as primary energy), heating and ventilation energy needs (expressed as thermal energy), and seasonal NG consumption (heating season is considered from May till April of the following year).

Values of Natural Gas (NG) consumption were obtained from the natural gas bills, with all consumption being attributable to the heating/ventilation system. Fig. 2 depicts the monthly data during the analyzed period and the total volume consumption during the heating seasons (from May till April of the following year) 2012-2013, 2013-2014, 2014-2015 and 2015-2016. For comparison, a total volume of 20832 Sm³ and 22033 Sm³ was used in 2009-2010 and 2010-2011 seasons respectively. The LHV was assumed to be 9.53 kWh/Sm³, since the gas provider only gave the HHV equal to 38.1 MJ/Sm³. Fig. 2 reports also monthly energy needs for heating and ventilation as measured by thermal energy meters.

4. Energy analysis for the period 2012-2016

4.1. Energy performance indexes

Once the energy flows of the plant were available some assumptions were made in order to complete the analysis. Energy performance was evaluated on the basis of different indexes:

- Primary Energy Ratio (PER) calculated for the whole HVAC plant and also separated for the two sections (heating and ventilation). It is the ratio between the useful thermal energy produced (for heating - subscript h - and for ventilation - subscript v - by heat pumps, boiler, and free renewable energy (solar+static recuperator)), and the primary (fossil) energy consumed (subscript p):

$$PER_{tot} = \frac{E_{h,Boiler} + E_{h,HP3+HP4} + E_{h,Solar} + E_{v,Boiler} + E_{v,HP1+HP2} + E_{v,Static\_rec}}{E_{p,h,Boiler} + E_{p,v,Boiler} + E_{p,HP1+HP2} + E_{p,HP3+HP4}}$$

$$PER_h = \frac{E_{h,Boiler} + E_{h,HP3+HP4} + E_{h,Solar}}{E_{p,h,Boiler} + E_{p,HP3+HP4}}$$

$$PER_v = \frac{E_{v,Boiler} + E_{v,HP1+HP2} + E_{v,Static\_rec}}{E_{p,v,Boiler} + E_{p,HP1+HP2}}$$

The electricity consumption of the auxiliaries (e.g. pumps, fans) was not measured and so it was not considered. In the previous analysis ([6] [7]) an estimation was made for the primary and secondary circuit pumps, based upon the known rated electrical consumption of each element and the operational schedule of the building. For ground and solar related data, and the recovery circuit power requirements, the on/off time was used to estimate energy consumption. In this case, no complete access to the remote data logging system was possible so this kind of monitored data was not available. For this reason the PER calculated without considering electrical energy consumption is overestimated respect to the real one.

- Gas Utilization Efficiency (GUE) for the heat pumps [10]:

$$GUE = \frac{E_{AC}}{E_p}$$

where E_{AC} is the energy delivered to the primary circuit from the heat pump condenser-absorber, and E_p is the input energy (NG) to the heat pump. E_p is not measured directly, but can be calculated through the first law for an absorption heat pump:

$$E_G + E_E = E_{AC}$$

where E_E is the energy supplied to the evaporator by the heat source. As the generator is fired by natural gas combustion whose efficiency is η_G, E_G is equal to $E_p \cdot \eta_G$. The GUE can therefore be found as:

$$GUE = \frac{E_{AC}}{E_{AC} - E_E} \cdot \eta_G$$

where E_{AC} and E_E were measured.

An estimation of the hourly and then monthly average efficiency (on the LHV) of combustion burner η_G of both the heat pumps (rated efficiency of 97.5 %) and the condensing boiler (rated efficiency exceeds 100 %) was made based on the first law energy balance of the plant. This accounted for the various operational on-off cycles, and also included some adjustment for when the system is functioning at part load.

- Solar fraction, defined as the ratio between the useful solar energy produced by the solar collectors (direct contribution to space heating, energy flows to the HP3 and HP4 evaporators during heating period and regeneration of the ground during summer months) and the total incident solar radiation on the collectors. The latter was calculated on the basis of the weather dataset for Agordo by the Environment Agency of Veneto Region [9] (total incident solar radiation on the horizontal) by means of the procedure described in previous works ([11] [12]).

4.2. Energy performance of the heating section

Fig. 3 shows the amounts of monthly energy leaving the central heating plant to supply the space heating circuits. The bars show the contribution from different system components: boiler, solar (direct), and heat pumps. When considering the thermal energy data for the four entire heating seasons (Fig. 4) it can be seen that the solar direct contribution to space heating was meaningful during only 2013-2014 season (6.7 %, similar to values calculated in previous analyses ([6] [7])), while in other periods it was lower or null. This was due to a stop of operation of the solar circuit from September 2012 till December 2013 due to damage of one collector, and from August 2014 till April 2016 for unknown reasons.

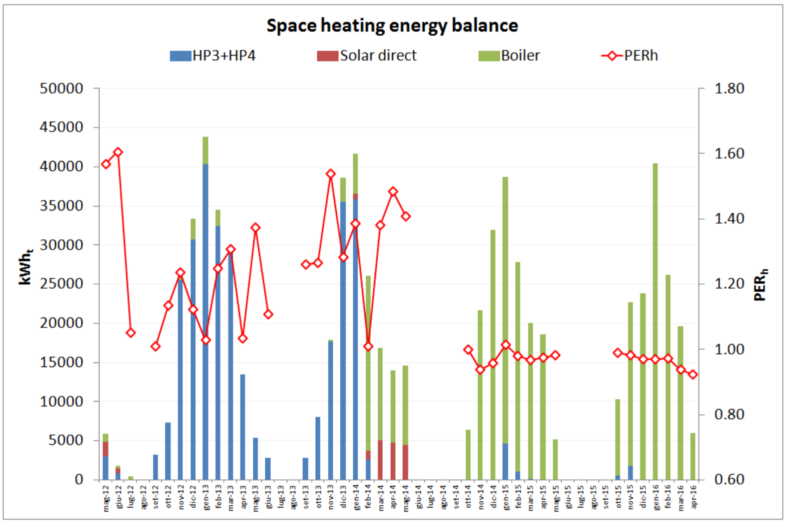

Fig. 3. Monthly energy contribution from the different sources (heat pumps, solar direct and boiler, expressed as thermal energy) to the heating energy needs, and PER of the heating section of the plant.

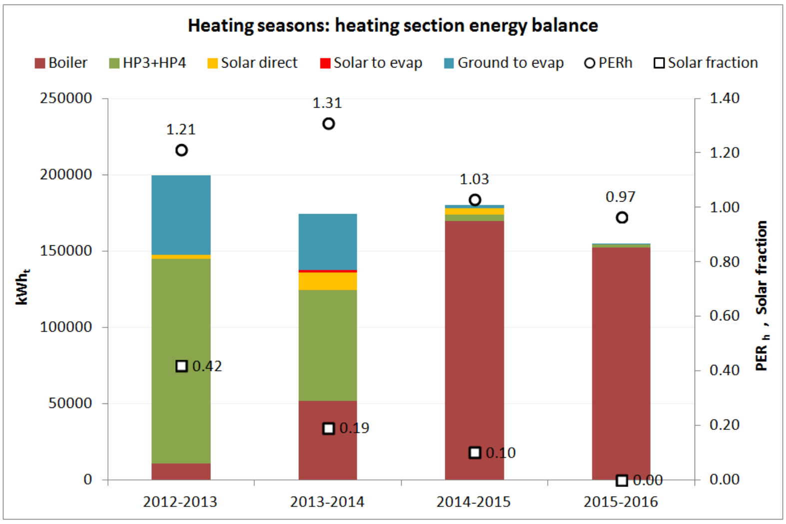

Fig. 4. Seasonal energy contribution from the different sources (energy from heat pumps generators, energy from heat pumps evaporators both from ground and solar, solar direct and energy from boiler, expressed as thermal energy) to the heating energy needs, PER and solar fraction of the heating section of the plant.

Another malfunction of the heating section is depicted in Fig. 3 and Fig. 4: during the period March-December 2014 the HP3 and HP4 heat pumps did not operate at all. They operated in the first three months of 2015 but producing very low energy and then they stopped again. This increased the backup boiler contribution that raised from 5.4 % till more than 98 % in the four heating seasons (Fig. 4) (in 2010 it was 6.1 %).The main effect is the very low value of PER in the last two heating seasons that tended to be equal to the condensing boiler efficiency (around 1).

The solar energy delivered to the evaporator is quite negligible, so all the energy delivered to the HP3 and HP4 evaporators was by the ground. The GUE of the heat pumps resulted to be between 1.2 and 1.5 while in the previous analysis, when solar collector field operated correctly, it was always between 1.45 and 1.5 ([6] [7]). In the months when solar circuit operated, the solar energy system utilisation exceeded 50 % most of time, even during the coldest season. It is very important to restore the correct operation of the solar field in order to increase energy performance of the heating section and to avoid aging and possible damages of the collectors.

4.3. Energy performance of the ventilation section

Fig. 5 illustrates the distribution of energy contributions relating to the ventilation section. The difference between sensible energy demand for ventilation and energy supplied to the AHU from the central heating plant is equal to the amount of energy recovered via the static recuperator. This was calculated ranging from 30 to 40 % of the ventilation needs during the coldest months, such percentage raised to 40-50 % during mild months. The same rationale for PER as explained for space heating was applied to calculations relating to the ventilation system. Values of PER were nearly always greater than 1.5 and reached 2.5 in mild months thanks to correct operation of the ventilation section that followed the previously described control logic. The effect of the heat recovery by means of the run-around coils was significant, as in mid seasons (from April to October) the evaporator was fed mostly by the run-around coils circuit (Fig. 6).

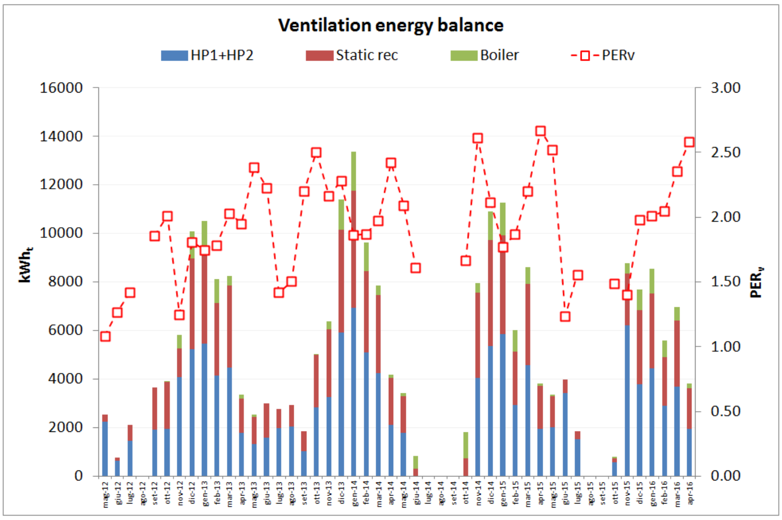

Fig. 5. Monthly energy contribution from the different sources (heat pumps, heat recovery from run-around coils in AHUs and boiler, expressed as thermal energy) to the ventilation energy needs, and PER of the ventilation section of the plant.

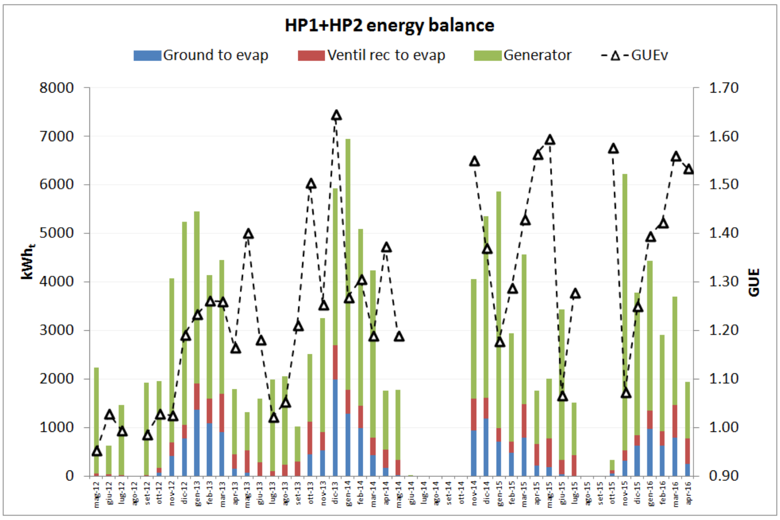

Fig. 6. Energy contribution and GUE for the heat pumps of the ventilation section.

As can be seen in Fig. 6, starting from October 2013 the GUE of HP1 and HP2 was meanly higher than the previous period ([6] [7]). This was due to a lower set point (from 55 °C to 45 °C) of the temperature of warm water produced by the heat pumps. This contributed to the high value of PER. As described in previous works ([6] [7]), analysis of the monitored set-point in the different building wings and the supply-return temperatures of the AHUs revealed that, in the whole building, the air temperature was generally set between 23 and 24 °C. Following the detailed data evaluation, it was possible to provide advice to the Belluno Province Administration regarding system operation. Recommended optimum air temperature set points to obtain maximum efficiency from the heating and ventilation systems were made, which in turn increased the GUE of the ventilation dedicated heat pumps.

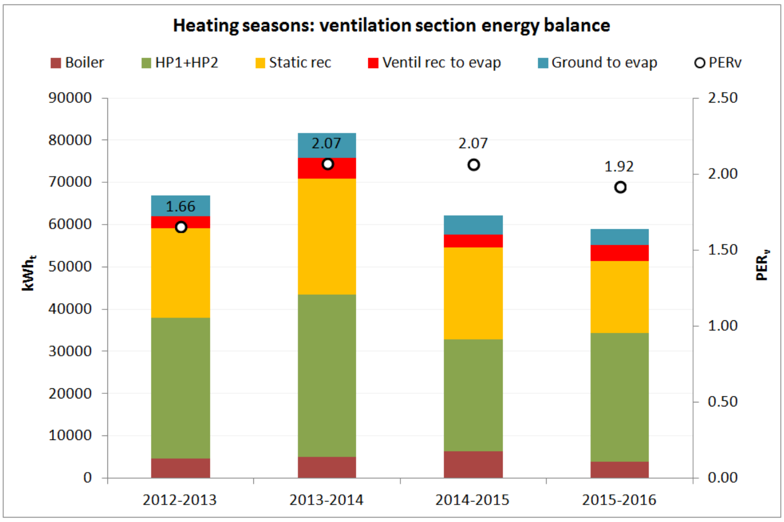

Fig. 7. Seasonal energy contribution from the different sources (energy from heat pumps generators, energy from heat pumps evaporators both from ground and run-around coils in AHUs, heat recovered by static recuperator and energy from boiler, expressed as thermal energy) to the ventilation energy needs, and PER of the ventilation section of the plant.

Comparing Fig. 4 to Fig. 7 it is interesting to see that, considering the whole period from May 2012 till April 2016, while for space heating the heat pumps delivered only 13.9 % of thermal energy whilst working in conjunction with the solar source, for ventilation the heat pumps delivered 43 % of their thermal energy while working on ventilation recovery. This suggests that it may be possible to make significant cost savings associated with borehole formation if the recovery system is correctly designed and adequately sized.

4.4. Energy performance of the plant

Fig. 8 shows the total energy balance of the whole plant for the four heating seasons, and the percentage contribution of the different devices to space heating and ventilation demands. The figure reports also the total PER and the specific Primary Energy (PE) demand.

The PE was calculated to be between 29.8 and 36.3 kWh/(m^2 year), remarking that it refers to NG consumption only. Considering electricity consumption of pumps and fans as well would increase the PE. According to Italian standards, allowing standard coefficients for distribution, emission and control efficiency, the target value for the building at the design phase was 30 kWh/(m^2 year) (NG + electricity consumption). It can be seen that, except for the 2013-2014 season (when HP3-HP4 heat pumps and solar field operated correctly) there is a large discrepancy between the predicted and measured values, due to the incorrect operation of the heating section of the plant.

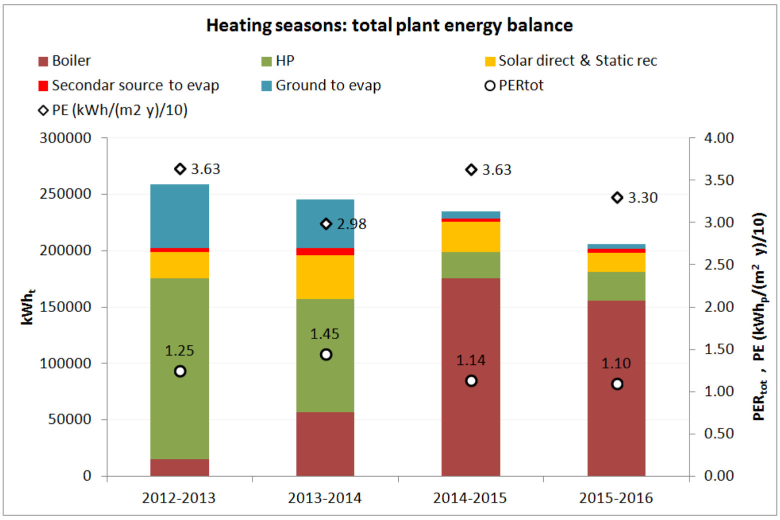

Fig. 8. Seasonal energy contribution from the different sources (energy from heat pumps generators, energy from heat pumps evaporators both from ground and secondary sources, heat recovered by static recuperator, solar direct and energy from boiler, expressed as thermal energy) to the global (heating+ventilation) energy needs, total PER, and specific primary energy consumption.

5. Economic analysis for the period 2012-2016

It is worth to investigate how the concept of multisource system increases the performances of a single source system. The main heat source on which the system relies is the ground, then the ventilation system is supplemented also by the heat recovery source, and the space heating by the solar source. Calculations were made to assess different scenarios to determine what would happen if one of the supplementary sources is missing.

With respect to ventilation, the single source scenario here considered is “no run-around coils recovery and no static recuperator” with the boiler replacing the recovery (the fan power of AHUs was adjusted). With respect to space heating, in the alternative scenario here considered the boiler replaces the solar system (so nor contribution to heating purposes or at the evaporator level, neither ground regeneration in summer are provided). Referring to the previous study [6], PER for the single source scenarios (only ground as heat pump heat source) were calculated to be 0.962 and 1.256 for ventilation and heating section respectively.

Based on a previous study ([13]) an estimation of the extra-investment cost with the dual source scenario with respect to the only-ground source was calculated to be 20000 €. Considering a specific cost of natural gas of 0.7 €/Sm^3, an interest rate of 2.5 % and a period for the economic analysis of 25 years ([13]), the cumulative differential (between the dual and single source scenarios) discounted cash flows are depicted in Fig. 9 basing on energy performance data of the four heating seasons. It is worth to stress that the dual source alternative is more advantageous than the single source one only if the 2013-2014 season is considered, that is the one with solar field operated correctly (at least in the second part of the heating season) and HP3-HP4 heat pumps operated for great part of the season. In this case Net Present Worth (NPW) is about 38000 € and Discounted Payback Period (DPP) is 7 years. Considering the 2012-2013 season, when no useful solar energy was produced and the HP3-HP4 heat pumps operated in only-ground source mode, a substantially economic indifference with the single source scenario is obtained (the curve intersects the axis at 25 years).

Because of some uncertainty on the extra-investment cost, a sensitivity analysis was done. It shows that both DPP and NPW improve (respectively decreases and increases) more rapidly considering the 2012-2013 season besides the 2013-2014 one (Fig. 10). This is because in the 2012-2013 heat pumps operated for all the heating season while in the 2013-2014 they operated in the first part only (Fig. 3).

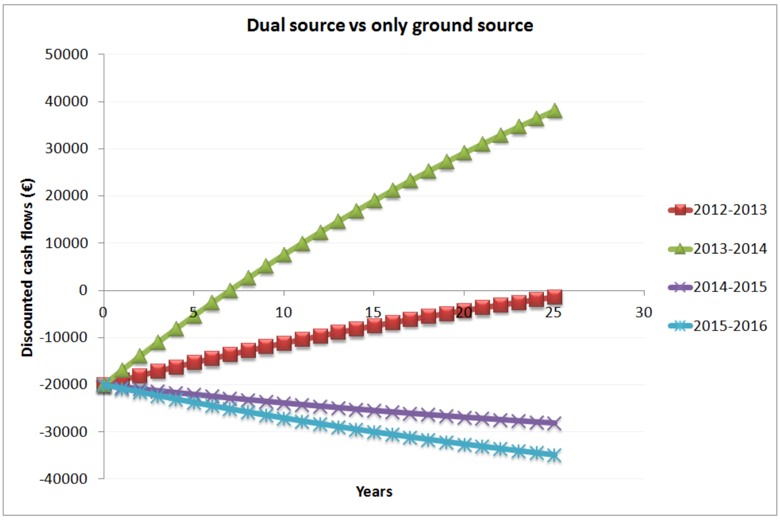

Fig. 9. Cumulative differential discounted cash flows between the two scenarios (dual source and only-ground source).

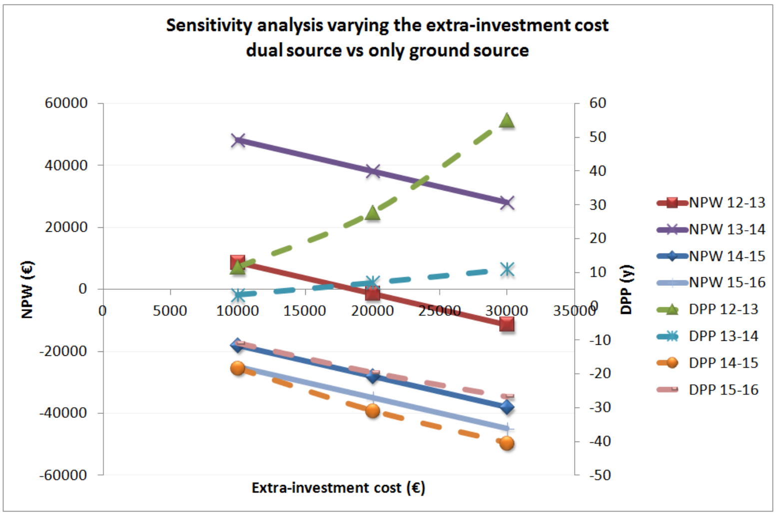

Fig. 10. Sensitivity analysis of the NPW and the DPP varying the extra-investment cost of the dual source scenario with respect to the only-ground one.

6. Conclusions

In this paper the operation of a multisource heat pumps system was described. Based on the monitored data available for the last four years and by means of the First Law balances, an analysis of the energy and economic performance of the system was done. The study revealed that a strong penalization in primary energy ratio was detected due to the malfunction of the heating section. In particular, the stop of operation of solar collectors field and heat pumps for great part of the heating seasons largely decreased the energy efficiency of the whole plant with respect to previous periods. Anyway, in the limited periods when the heating section operated correctly, the solar heat contributed with a meaningful fraction of the space heating need. As already proved by the Authors in a previous study [13], the most efficient solution both from energy and economic point of view for heat pump based systems is to adopt a multi-source system, sizing the ground exchangers to allow a minimum ground inlet temperature of -2 °C, and directing the saving in investment cost to buy solar collectors.

Anyway, concerning the ventilation section of the plant, the integration of different heat sources remarkably increased the efficiency of the system in terms of primary energy consumption. The ventilation recovery downstream the cross flow heat exchangers supplied a large share of the evaporators energy need for ventilation dedicated heat pumps.

A careful investigation of the causes of the malfunction of the heating section (damages to the solar collectors or heat pumps and incorrect control logic of the system) is necessary in a further extension of this work in order to make the Belluno Province Administration aware of the energy and economic penalization it is occurring.

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