Multisource heat pump system from design to operation: the case study of a new school building

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Abstract

Heat pump features can take advantage of better sources than air, for instance the ground heat, solar heat and heat recovery. A multisource system aims to enhance the performances of the heat pump, leading to a significant amount of primary energy being saved. The present work shows data monitoring and analysis for a real working application in northern Italy, for 12 months. The energy balance indicates that the integration of different sources not only increases the thermal performance of the system as a whole but also optimizes the use of each source.

Keywords: low-energy building; multisource heat pump; solar-assisted heat pump; primary energy savings

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1 INTRODUCTION

To reduce the energy need of the building and the installed heating and cooling capacity in the temperate climate (according to the Koppen climate classification), good options are:

(i) to design a low-energy building envelope with good thermal insulation;
(ii) to choose energy-effective technologies based on a high-efficiency generation system, such as heat pumps;
(iii) to integrate them as much as possible with heat-recovery devices and renewable energy sources.

Designed in 2006–07, the new High School Building of Agordo (province of Belluno, northern Italy)—enclosing the three features mentioned above—started its operations in autumn 2009. The building is operated by the Belluno Province Administration, appointed for the public education service.

The town of Agordo lies in the geographical area of the Dolomiti mountains in a valley at 611 m asl, where the climate is severe during wintertime (3376 degree-days). The building has a total floor area of 5680 m^2, an outward surface of 13 608 m^2 and an enclosed gross heated volume of 19 644 m^3; the envelope is well insulated, the outer walls and the roof allowing an average thermal transmittance of ~0.16 W/(m^2 K), the floor having a thermal transmittance to the ground [1] of 0.4 W/(m^2 K) and the glazing system having a thermal transmittance of 1.38 W/(m^2 K).

From the architectural and functional point of view, the building is made up of two main wings and a central belt: the south-east wing is on three storeys and houses teaching rooms, used mainly (but not only) in the morning; the west wing is on two storeys and houses the laboratories, used mainly (but not only) in the afternoon; finally the central belt on three storeys houses the administrative offices, manned all the day, and the auditorium, used occasionally.

Lessons are stopped from the middle of June to the end of August and so the climatization system does not provide summer cooling, but only ventilation and space heating.

Through dynamic simulation in the TRNSYS environment, different solutions were evaluated with respect to the heating system [2, 3]. A multisource absorption heat pump system has been designed to fulfil the needs of the building: the sources are ground, sun and recovery on ventilation. The system was designed in such a way that the size of the borehole ground exchangers field and the solar system was optimized to find the most viable mix [4]. The ratio between the thermal input from the source and the thermal output is much lower for an absorption heat pump than for a compression one, which would need a much larger (and much more expensive) borehole field. This consideration applies only to the heating mode of course.

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2 DESCRIPTION OF THE HEATING, VENTILATING AND AIR CONDITIONING PLANT

The heating, ventilating and air conditioning (HVAC) system in teaching rooms, laboratories and offices provides space heating by means of a radiant floor and ventilation by means of three independent AHUs (air handling unit), each of which serves a single-duct system [5]. The auditorium is served by an all-air system. From the design calculations, performed according to ref. [6], the space heating requires a maximum power of 146 kW and the ventilation system requires 122 kW of sensible heat, the design indoor conditions being as follows: 20°C of air temperature and neutral ventilation (air supplied at 20°C).

Under the ground floor of the south-east wing lies the central heating plant. For the sake of simplicity, a reduced functional diagram of the plant is shown in Figure 1; the hydraulic streams and, therefore, the energy flows within the plant can be easily understood, while many hydraulic attachments (hydraulic separators, filters, valves and pumps) were hidden as well as the secondary circuits of the radiant floor.

The HVAC system is split into a ventilating section (left) and a space heating section (right). The system is equipped with gas-fired absorption heat pumps and a backup boiler, and the size of the units was selected to cover the peak load with the use of fossil fuel.

The ventilation section is made up of AHUs equipped with heating coils connected to the heat pumps; there are also sensible heat recuperators made up of cross-flow heat exchangers; they were sized with an efficiency limited to 50% in order to avoid frosting problems. The thermal source for the ventilation heat pumps can be either the ground (750 m, 6 × 125 m in a row, of vertical tube heat exchangers) or the exhaust airflow [7]. By means of a run-around coil (shown in Figure 1), downstream the cross-flow recuperator on the exhaust flow, a total recovery is then produced at the absorption heat pump evaporator level.

In the simplified scheme, special attention shall be paid to the AHUs since only two out of four of them (for a global volume flow of 20 600 m^3/h out of 25 000 m^3/h) are equipped with the run-around coil, while the two left (4400 m^3/h) save energy only from the cross-flow recuperator.

The heating section is made up of radiant floors connected to the heat pumps. Solar collectors (four arrays in parallel, each of which are made up of five modules in series) can serve directly the radiant floor, by means of the plate heat exchanger (Figure 1, top right side).

The thermal source for the space-heating heat pumps can be either the ground (960 m, 6 × 160 in a row, of vertical tube heat exchangers) or the solar section. Boreholes heat exchangers were designed with double-U pipes with an outer diameter of 32 mm and a thickness of 2.9 mm. In summer the solar system can re-generate the ground.

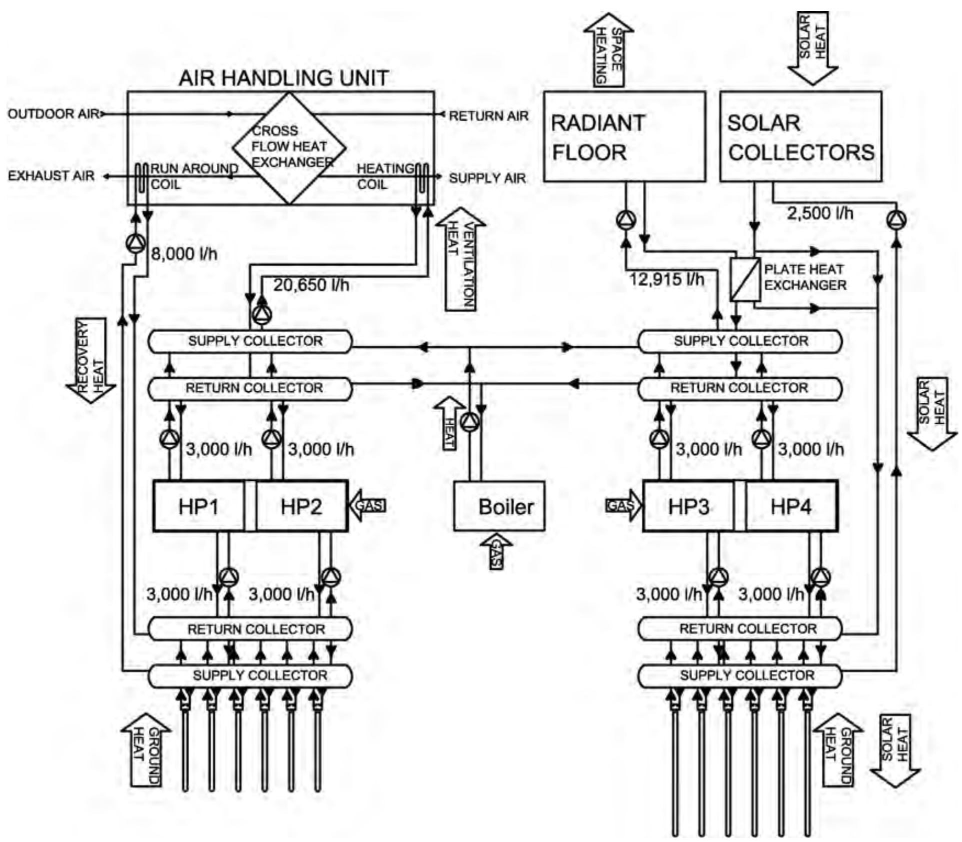

Figure 1. *A simplified functional diagram of the HVAC plant.*

Table 1. *Heating generators of the central HVAC plant.*

Component	Rated electric consumption (kW)	Rated capacity (kW)	Rated efficiency
HP1 + HP2	0.15	74 (B0W60)	1.25 (*GUE* (UNI [11]))
HP3 + HP4	0.15	76 (B0W40)	1.40 (*GUE* (UNI [11]))
Boiler	0.18	114.4	1.02 (condensing)

Table 2. *Electric power of the auxiliaries.*

Auxiliary	Electric power (kW)
Primary pumps space heating	0.42
Primary pumps ventilation	0.42
Evaporator pumps space heating	0.42
Evaporator pumps ventilation	0.42
Ground borehole pumps heating	0.44
Ground boreholes pumps ventilation	0.42
Solar circuit pump	0.47
Run-around coil pumps	0.37
Secondary pumps space heating	1.10
Secondary pumps ventilation	0.80
AHU fans	1.31

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

In Table 2, the rated electric consumption of the auxiliaries is shown. It is more organized to group the pumps according to their function than just to list them.

3 OPERATION OF THE HVAC PLANT: MONITORING

Now the control strategy is explained.

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 AHUs are scheduled to work from 6 am to 2 pm, those of the laboratories from 10 am to 6 pm and those of the offices from 6 am to 6 pm. Heat pump (HP)1, HP2 and the boiler are activated once the return temperature falls below the given thresholds. The sequence of activation of the HPs is swapped every 2 weeks. 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 an upper bound for the highest temperature that ground could offer.

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, and the sequence of activation of the HPs is swapped every 2 weeks. The solar system is activated according to the measured solar radiation I_β (W/m^2).

The efficiency of the solar field can be roughly defined with a first-order model (the reason is explained a few lines below) as

$$\eta = \eta_0 - a_1 \frac{T_m - T_{oa}}{I_\beta} \tag{1}$$

where η_0 is the zero-loss solar system efficiency, a_1 the heat-loss coefficient, T_m the mean temperature of the fluid, and T_{oa} the outside air temperature. η_0 and a_1 can be calculated from the respective values with respect to the single collector, i.e. 0.75 and 4.14 W/(m^2 K), multiplied by *CF5*, which is the series correction factor [8], equal to 0.881 (for the five collectors in series).

Let the threshold radiation I_T be defined as follows (the first-order model adopted does not lead to significant errors for flat-plate collectors):

$$I_T = \frac{a_1(T_m - T_{oa})}{\eta_0 - \eta_{min}} \tag{2}$$

thus being the radiation that provides the minimum acceptable efficiency η_{min}, given the solar field characteristics, the minimum average temperature desired T_m and the outdoor temperature T_{oa}. The desired T_m was set at 5°C and η_{min} at 0.01. Whenever solar radiation on the field I_β exceeds I_T, the solar loop pump is activated. 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. Then the solar outlet feeds the evaporator collectors, 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 accomplished with the cooperation of the building's and the plant's controller designers. The interface of the monitoring systems, running on the PC that controls the plant, has been accessible from any remote terminal via a virtual private network (after authentication), from the end of October 2009 to the end of March 2011. During this first phase, the remote access was open to allow the designer to make small adjustments on the control variables of the plant, after which the access to the control PC was discontinued, as well as the data logging, and the control of the plant was allowed from inside the building only.

During the period indicated above, the following cumulative energy flows (mass flow times the temperature difference between the inlet and the outlet, via simple thermal energy meters) were logged monthly:

- condenser and evaporator of each heat pump (at the collectors);
- ground circuits, the one for ventilation and that for space heating;

- primary circuit of AHU heating coils and run-around coils;
- solar circuit;
- primary circuit of the radiant floor.

All the energy meters are located in the central heating plant; therefore, the energy delivered to each circuit is the gross value including distribution losses.

The previous listed ones and other significant parameters, for instance inlet–outlet temperatures for hydraulic circuits and supply–return temperatures in air ducts, room temperatures, mixing valve positions and on–off state of the heat pumps, were logged hourly, but presumably due to malfunctions in the monitoring software, the sequence is often broken and so it was necessary to post-process the data in order to fill the gaps. However, the hourly data were of a great help to reveal some improper use in the control of internal temperatures, which will be detailed in a following paragraph.

The gas consumption (standard m^3, Sm^3) was given from the natural gas bills (the heating/ventilation plant being the only gas consumer in the building) of the seasons 2009–10 and 2010–11, respectively, 20 832 and 22 033 Sm^3. The LHV is assumed to be 9.55 kWh/Sm^3 (34.38 MJ/Sm^3), since the gas provider only gives the HHV equal to 38.32 MJ/Sm^3.

4 ENERGY PERFORMANCES 2009–2011

Once the energy flows of the plant described above are available, few assumptions need to be made in order to complete the analysis.

With respect to the heat pump, the gas utilization efficiency (*GUE*) is defined as follows [8]:

$$\mathrm{GUE} = \frac{E_{\mathrm{AC}}}{E_{\mathrm{in}}} \tag{3}$$

where E_{AC} is the energy delivered to the primary circuit from the heat pump condenser–absorber and E_{in} is the input energy (gas) to the heat pump. E_{in} is not measured directly, but the first law for an absorption heat pumps offers

$$E_{\mathrm{G}} + E_{\mathrm{E}} = E_{\mathrm{AC}} \tag{4}$$

where E_{E} is the energy supplied to the evaporator from the heat source; the generator is fired by natural gas; therefore, $E_{\mathrm{G}} = E_{\mathrm{in}} \times \eta_{\mathrm{G}}$, then *GUE* becomes

$$\mathrm{GUE} = \frac{E_{\mathrm{AC}}}{E_{\mathrm{AC}} - E_{\mathrm{E}}} \cdot \eta_G \tag{5}$$

where E_{AC} and E_{E} are known.

The same rationale applies to the backup condensing boiler. Italian national regulations set the minimum required efficiency for a boiler as 90% + 2 × log10 (P_n), where P_n [kW] is the rated capacity of the burner. For a conservative approach, it has been decided to set the annual average efficiency of the burners (those of the heat pumps, having a rated efficiency of 97.5% and that of the condensing boiler, having a rated efficiency exceeding 100%) at 85%, in order to account for part load operation and the several on/off cycles.

The electricity consumption of the auxiliaries (pumps, fans) was not measured; however, it has been estimated considering the rated electric consumption of each and the scheduling of the building (omitted for sake of brevity) for primary and secondary circuit pumps, and from the on/off time of each with respect to ground, solar and recovery circuit logged data.

Figure 2 illustrates the space heating energy distribution between heat pump, direct solar and boiler. When the balance is made on the whole year 2010, the solar direct contribution is 8.2%, exceeding the backup boiler that contributes 6.1%.

Figure 3 reports the heating energy share for ventilation, while Figure 4 shows the total HVAC system heating share. The solar source influence is really appreciable. In the whole 2010, the solar energy delivered to the evaporator is equal to the 13% of the total energy required at evaporator level. Indeed after that of the heat pumps, the most relevant contribution is given from the static recuperator, with monthly efficiency ranging from 35 to 50%.

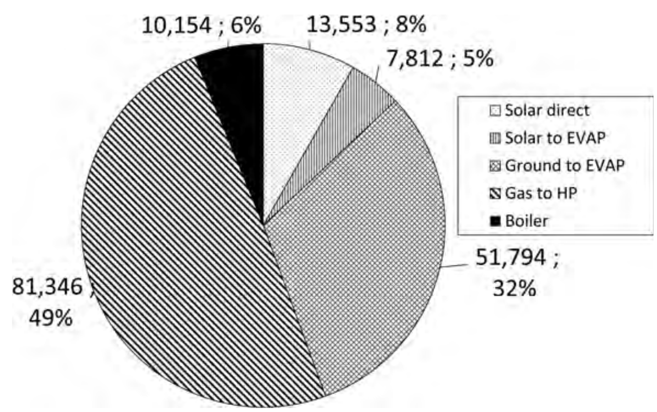

Figure 2. *The space heating share of energy (kWh; %), 2010.*

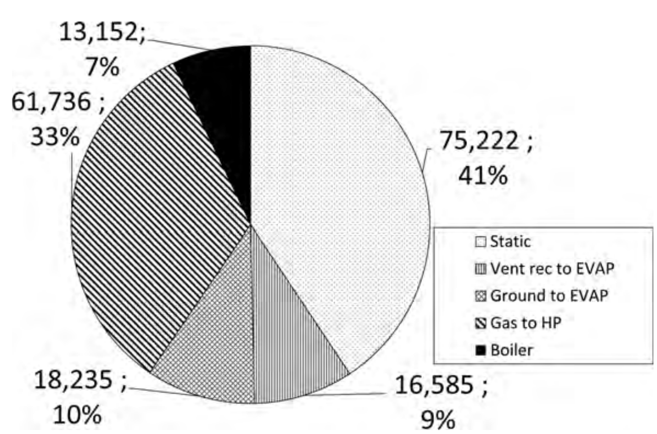

Figure 3. *The ventilation share of energy (kWh; %), 2010.*

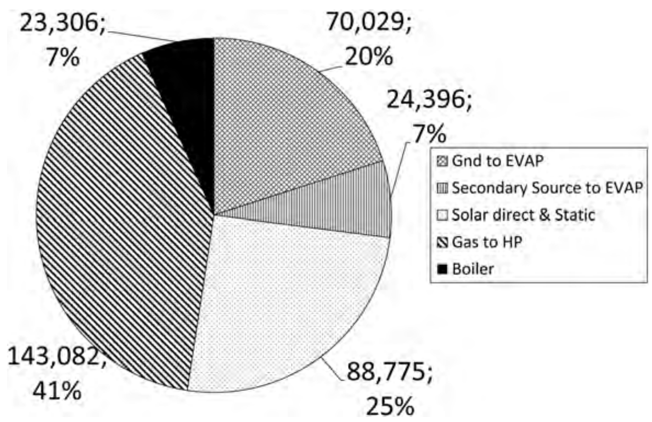

Figure 4. *The HVAC global share of energy (kWh; %), 2010.*

Table 3. *GUE of the heat pumps when working on a specific heat source.*

	HP heating	HP ventilation
GUE	1.473	1.322
GUE_{gnd}	1.465	1.290
GUE_{ss}	1.530	1.375

Finally, Table 3 shows the *GUE* of the heat pumps for both sections, yearly averaged or considered for a specific source, ground (gnd) or secondary source (ss, solar and heat recovery, respectively, for heating and ventilation). The secondary source influence is remarkable: it raises the *GUE* from around 1.47 to 1.53 for heating and from 1.29 to 1.38 for ventilation.

As it can be seen in Table 3, ventilation heat pumps' *GUE* is lower than that of space heating. The ventilation-dedicated heat pumps produce water at a higher temperature (55–60°C) than the space heating ones, therefore it is reasonable to expect that the GUE of the ventilation dedicated units might be higher than that of the space heating ones.

For the year 2010, it is then possible to look into Table 4 for a detailed energy balance of the heating plant. The quantity *GUE** is the gas utilization efficiency as it should be calculated according to thermodynamics; as per ref. [8] the *GUE* only accounts for gas consumption, while *GUE** includes the electric consumption of the heat pump (converted to primary).

The primary energy ratio (*PER*) is calculated as the ratio between the thermal energy produced and the primary energy input, including—for space heating—electric auxiliaries of primary, secondary, source circuits (ground solar circulation pump) and the electric consumption of heat pumps; the conversion factor from primary to electricity is 0.46, according to Italian regulation [9]. The same rationale applies to ventilation, thus including fan consumption.

The *PER* CP is the primary energy ratio calculated at the central heating plant excluding the secondary distribution pumps and/or fans, and the *PER* w/o electricity is the ratio of the delivered thermal energy to the thermal energy consumption. If compared with the *PER*, which is the most important parameter with respect to primary energy conservation, these figures offer some interesting information about the weight of the auxiliaries (distribution system) on the global energy consumption for climatization.

The calculated specific primary energy demand [PE, kWh/(m^2 year)] is then equal to 39.1 for 2010. The target value for the building (according to Italian standards and so allowing standard coefficients for distribution, emission and control efficiency) in the design phase was 30 kWh/(m^2 year). The reasons were then investigated for such a difference. The 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 most of the time was set between 23 and 24°C.

Trial calculations have been performed to estimate the PE in standard conditions, so with 20°C internal temperature. The behaviour of the solar system has been considered equal to that of the real operating conditions, as well as the efficiency of the sensible recuperator on the AHUs, the need of space heating and that of ventilation were considered proportional to the heating degree hours HDH, according to the procedure described in ref. [10]. Therefore, it was assumed

$$\left(\frac{E_{sh}}{\mathrm{HDH}}\right)_{23.5^\circ C} = \left(\frac{E_{sh}}{\mathrm{HDH}}\right)_{20^\circ C} \tag{6}$$

the same applying for ventilation as well. The HDH were calculated (with respect to the 'on' period of the plant) by means of the hourly weather data provided by the Regional Agency for Environment ARPAV, for the weather station in Agordo (BL). The *GUE* of the heat pumps and the boiler efficiency were considered equal to those in real operating conditions. By means of this rough but effective procedure, the PE in standard conditions is estimated to equal 31.5 kWh/(m^2 year) and so only 5% far from the design value. This method keeps this evaluation on the safe side, since it does not account for solar gains, that becomes more and more relevant while decreasing the temperature difference between inside and outside.

Strong recommendations were then reported to the Belluno Province Administration in order to survey the air temperature set-point for the times to come.

5 SCENARIOS WITHOUT MULTISOURCE

It is now interesting to investigate how the concept of multisource system increases the performances of a single-source system. Being the HVAC plant based on water-to-water heat pumps, 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. To determine what happens if one of the supplementary sources is missing, calculations were made to assess different scenarios.

Table 4. *Summary of the energy performance of the plant.*

Space heating			Ventilation			Global		
Quantity	Energy/ performance	Unit	Quantity	Energy/ performance	Unit	Quantity	Energy/ performance	Unit
Space Heating	164 657	kWh_t	Ventilation	184 931	kWh_t	Total energy delivered	349 588	kWh_t
Solar direct	13 553	kWh_t	Static recuperator	75 222	kWh_t	Total gas consumption	196 141	kWh_t
HP3 + 4	140 951	kWh_t	HP1 + 2	96 557	kWh_t			
Gas to HP	95 702	kWh_t	Gas to HP	73 020	kWh_t			
Boiler	10 154	kWh_t	Boiler	13 152	kWh_t			
Gas to boiler	11 946	kWh_t	Gas to boiler	15 473	kWh_t			
El cons HP	495	kWh_{el}	El cons HP	408	kWh_{el}	HP el Consumption	903	kWh_{el}
			AHU fan consumption	1447	kWh_{el}	AHU fan consumption	1447	kWh_{el}
Ground pumps	307	kWh_{el}	Recovery pumps	576	kWh_{el}	Source pump consumption	3869	kWh_{el}
Solar pumps	2922	kWh_{el}	Ground pumps	65	kWh_{el}			
Evaporator pumps	511	kWh_{el}	Evaporator pumps	1142	kWh_{el}	Evaporator pumps	1653	kWh_{el}
Primary pumps	685	kWh_{el}	Primary pumps	1142	kWh_{el}	Primary pumps	1827	kWh_{el}
Secondary pumps	1060	kWh_{el}	Secondary pumps	1305	kWh_{el}	Secondary pumps	2365	kWh_{el}
GUE HP3 + 4	1.473		*GUE* HP1 + 2	1.322				
*GUE** HP3 + 4	1.456		*GUE** HP1 + 2	1.306				
PER space heating	1.365		*PER* ventilation	1.834		*PER*	1.57	
PER CP	1.404		*PER* CP	1.976		*PER* CP	1.63	
PER w/o electricity	1.530		*PER* w/o electricity	2.090		*PER* w/o electricity	1.78	

Table 5. *Performance indicators in the different scenarios.*

	Ventilation				Space heating			
	Reference	No run-around coils	HP replaces static recuperator	Boiler replaces static recuperator		Reference	HP replaces solar	Boiler replaces solar
Quantity	value				value			
GUE HP 1 + 2	1.322	1.285	1.287	1.285	*GUE* HP 3 + 4	1.473	1.460	1.461
PER	1.834	1.832	1.126	0.962	*PER*	1.365	1.320	1.256

With respect to ventilation, the scenarios are no run-around coils recovery and no static recuperator. In this last case two subcases were defined: in the first the heat pump replaces the heat recovery as well and in the second the boiler replaces the recovery (the first case cannot be a real option since the capacity of the heat pumps is insufficient, and it would require further heating capacity). The fan power has been adjusted to different AHU configurations.

With respect to space heating, two scenarios were evaluated: once the solar is missing, in the first scenario, the HP replaces the solar system, in the second the boiler replaces the solar system. In both cases the solar energy does not contribute to heating purposes or at the evaporator level, neither ground regeneration is provided in summer.

The *GUE* of the heat pump was extrapolated by means of a 10-parameter polynomial function (ARI standard) of the ground loop average temperature (peak load) and supply temperature in the real operating conditions. Then for each scenario, the monthly average ground loop temperatures were estimated by means of the EED software (Earth Energy Designer 2.0, it was also used during the design phase), and finally the balance illustrated in Table 5 was produced.

For the sake of brevity, the complete balance of each scenario is omitted, though the synthetic indicators are reported in Table 5.

The two indicators are important to different considerations. *GUE* always decreases in case of single-source systems, since the ground is heavily stressed, both in the ventilation and in the space heating section. The solar system contribution to the evaporator is less important than that of run-around coil recovery; however, the solar system allows summer ground regeneration. From the calculations performed in EED, however, it seems that summer regeneration produces less benefits than those of a lower extraction during wintertime.

The overall *PER* indicates that the supplementary source has a greater effect on the space heating than in the ventilation system. This is of course due to the fact that the solar system also provides for 'almost free' heating effect, while run-around

Table 6. *Comparison between the old and the new buildings.*

	Gross volume (m^3)	Fuel consumption	LHV	Specific primary energy consumption (kWh/m^3)
Old	17 715	64 160 l	10.48 kWh/l	37.98
New	19 644	21 432 Sm^3	9.55 kWh/Sm^3	10.42

coils recovery only increases the *GUE* of the heat pumps with a reasonable cost in terms of higher fan and pumping consumptions. Moreover, the space heating heat pumps work at a lower condensation temperature than the ventilation ones. According to the following equation [12]:

$$\frac{(\partial \mathrm{GUE}_{\mathrm{CA,h}}/\partial \Delta T)_{T_c}}{\mathrm{GUE}_{\mathrm{CA,h}}} = -\frac{1}{\Delta T} \tag{7}$$

the sensitivity of the *GUE* to the evaporation temperature increases while temperature lift decreases (ΔT is the difference between condensation and evaporation temperature). Therefore, since the heating system works at a lower condensation temperature than ventilation, the heating section benefits from the multisource concept more than the ventilation one.

6 FURTHER REMARKS

From the calculations reported in Table 3, the natural gas consumption in 2010 is equal to 20 532 Sm^3. The average of the two heating seasons 2009–10 and 2010–11 is from the natural gas bill equal to 21 432 Sm^3/year. The difference (4.2%) from the calculated consumption could be ascribed mainly to the following:

- inaccuracy of the energy meters that equipped the hydraulic circuits;
- the incorrect evaluation of the yearly average burners' efficiency.

However, the estimation is quite good, considering that no gas consumption measurement was allowed on the single units.

It is interesting to underline that the previous building that housed the same high school in Agordo enclosed a gross volume of 17 715 m^3 and burned diesel oil for an amount of 64 160 l in 2006–07.

Table 6 finally shows the comparison between the new and the old buildings serving the same purpose.

As it can be seen, the new building specific primary energy consumption is almost a fourth of the old one.

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