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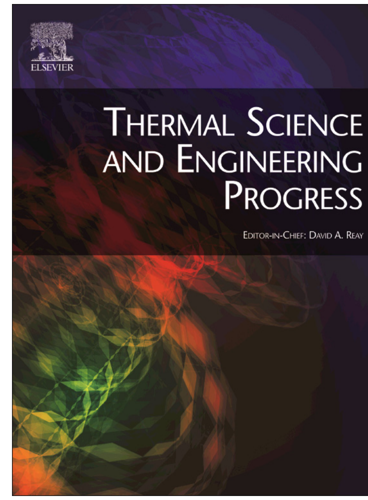

The Importance of Monitoring Renewable Energy Plants: Three Case Histories

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THE IMPORTANCE OF MONITORING RENEWABLE ENERGY PLANTS: THREE CASE HISTORIES

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Abstract. Many renewable energy plants are put into operation without providing a monitoring system to evaluate their performance over time. Then if is often difficult to realise the bad working of the system and the loss of efficiency results in an economic loss. In the Author's experience as designer or supervisor of such plants, he came across various examples that pointed out the advantages of having installed a monitoring system, of course with a careful data analysis. Problems sometimes arose from poorer performance than anticipated in the design, but more often from inefficient plant operations after some months or years from the starting.

Three quite different examples, derived from the Author's direct experience, are reported to illustrate how real performance can be lower than designed due respectively:

1. To bad settings of the parameters;
2. To a hurried commissioning that did not reveal the mistakes in the design of the plant;
3. To a failure of a single component over time.

Key words
Renewable energy, heat pumps, bad control

Highlights

- Many renewable energy plants are not provided with a monitoring system
- It is often difficult to realise the bad working of a plant without monitoring it
- It is not straightforward to decide whether the energy bill is appropriate
- Three different practical examples demonstrate the importance of monitoring

1 Introduction

Renewable energy plants are seldom simple plants except small installations such as DHW solar plants or small photovoltaic plants. They often include the renewable section and conventional auxiliaries, usually boilers or chillers. A hot and/or cold storage is/are usually present and a suitable control system governs the plant. The plant manager is frequently a person other than the designer and he does not always possess a comprehensive understanding of the system control logic.

Many renewable energy plants are put into operation without providing a monitoring system to evaluate their performance over time. It happens that seasonal performance may be well below the values planned in the design or recorded during commissioning. Or wrong setting of parameters or the failure of some components prejudices system performance: the system goes on operating but at a lower efficiency than designed. In the absence of a supervision it is difficult to realise the bad working of the system. In fact the auxiliary provides the service and it is not straightforward to decide whether the energy bill, that often covers a whole month or more different needs than the considered plant, is appropriate. The loss of efficiency results in an economic loss that might go on for years, losing the expected benefits of a more expensive, but potentially also more efficient installation.

Scientific literature sometimes reports analysis of performance gaps between predicted and real data, however regarding small or simple plants. Analyses regarding complex plants or a whole building are sometimes published on technical literature. Connelly and Fedoruk [1] report on a large building in San Francisco (about 2,000 m^2 floor area), but their analysis, regarding two years of operation (2014, 2015), compares predicted vs. measured energy use. Then the discrepancies were attributed to a warmer winter or to an underestimating the performance of the Variable Refrigerant Flow (VRF) system in heating and not to bad working of system or components. Another long term survey regarding the resulting of refurbishment of a large building is offered by Vaughn [2] who considers various aspects in the building management from comfort to Indoor Air Quality (IAQ) and electric loads, evaluating even water consumption and acoustics. However the analysis is mainly qualitative and no information is offered of energy consumption of the Heating Ventilating Air Conditioning (HVAC) plant in the considered period.

Other extensive surveys are available, however regarding the comparison of many similar plants, or between each to another or with predicted performance by simulation. An excellent example is due to Miara et al. [3] that report on an extensive investigation regarding 250 air-to-water and brine-to-water heat pumps systems in single family dwelling for a period over 10 years. The comparison concerns mainly the laboratory COP values with the field monitoring data. Only few hints are dedicated to the possible explanations of the difference, indicating generic responsibility of the manufacturer, the planner, the installer and residents.

A similar careful investigation regarding ground source heat pump (GSHP) systems is proposed in 5 paper by Kavanaugh and Kavanaugh [4-8]. They completed an 18-month data collection and analysis project identifying common characteristics of successful GSHP systems and the prevalence of unacceptable long-term temperature change. The goal of the project was "*to enhance the ability of GSHPs to minimize energy consumption, electrical demand, and maintenance requirements while being cost effective and environmentally responsible*". They evaluated the performance of 35 buildings of different characteristics regarding the geographic location, the use (all commercial buildings), air distribution systems, control methods and so on. Particular attention was devoted to 22 buildings that attained an Energy Star [*] designation. In this long time survey the attention is dedicated to a general comparison of the various buildings performance with the Energy Star standard, in simple terms between predicted and real performance. No significant analysis is proposed on the reason of the difference. No surprise on that as the purpose of the investigation was not the single building but the behavior of a group of different buildings. Certainly the gap between anticipated plant performance from planner simulations and real results, or even between the performance surveyed during commissioning and after some

[*] Energy Star (trademarked *ENERGY STAR*) is an international standard for energy efficient products originated in the United States

months or years of operation is sometimes impressive. Recently Mahdavi and Ghiassi [9] presented a report on solar houses where the gap was up to 100%.

As just illustrated studies regarding the long term survey of a single plant are, to the Author's knowledge, practically absent. The reason is, probably, that the analysis requires time and expertise with related costs. In fact most of the previously presented studies were financed by companies or institutions. The building owner does not consider that those costs are investment in keeping a good efficiency of the plant or in correcting mistakes in the design or in the installation or in the management of the plant. Among other things the required instrumentation does not need the accuracy of scientific instruments: common heat meters and thermocouples or even thermistors can supply sufficient information for an analysis. Moreover the costs of the instrumentation, recording and transmitting devices dropped dramatically in these years.

The Author in his experience as designer or supervisor of renewable energy plants always required the installation of a monitoring system. Thus he had the possibility of long term analysis of plant operations, producing often scientific and technical publications on this unfrequented topic with his research team.

Three quite different examples from the Author's direct experience are here reported to illustrate how real performance can be lower than designed due respectively:

1. To bad settings of the parameters;
2. To a hurried commissioning that did not reveal the mistakes in the design of the plant;
3. To a failure of a single component over time.

2 The first example: a motor driven heat pump HVAC plant

The Department of Management and Engineering of the University of Padova (DTG) is located in the historic heart of the town of Vicenza since 1999. The building has a volume of 14,300 m^3 and a net floor area of about 4,200 m^2.

The building HVAC plant is based on:

- Two pipes fan-coils to distribute heating and cooling.
- An Air Handling Unit (AHU) that treats 21,700 m^3/h supplying ventilation and providing humidity control. The AHU equipped with a cross flow heat exchanger and, after a pre-heating section, the coils for cooling/dehumidification, followed by the post heating.

A gas reciprocating engine driven heat pump provides heating and cooling production with two condensing boilers as auxiliary.

An air to water heat pump equipped with a four-way inversion valve can operate both in heating and cooling mode, driven by a gas reciprocating engine. The heating/cooling capacity can be modulated in a continuous way by varying the engine speed from 1800 to 2600 rpm, through a throttle butterfly valve. The inlet valve on the compressor cylinders can be unloaded, obtaining a further modulation down to 40% of the rated heating/cooling power. The rated heating capacity is 380 kW with a gas consumption of 19 Nm^3/h (for 10 °C outdoor air and condenser inlet/outlet 40/45 °C). The rated cooling capacity is 276 kW with a gas consumption of 22 Nm^3/h (for 35 °C outdoor air and evaporator inlet/outlet 12/7 °C).

Exhaust and cooling jacket water plus lubricant oil cooling provide the recovered heat, as to achieve a rated recovery of 109 kW when in heating mode and of 126 kW when working in cooling mode, available at about 70 °C. The EER in refrigeration mode is 4.5, with a PER (Primary Energy Ratio, i.e. the ratio of the cooling/heating capacity over the fuel consumption in terms of primary energy) of 1.4 in cooling mode; in heating mode the COP is higher than 5 and the PER is of about 1.5.

A double hydraulic circuit takes the engine recovered heat providing hot water (70/65 °C) and chilled water in summer (7/12 °C) and warm water in winter (45/40 °C) supplied by the heat pump. The control is allowed by a digital display controller: it operates on the engine speed and the unloader of the compressor cylinders. The system has two separate storage tanks, 3000 l on the heat pump circuit at a temperature of about 45 °C and 1500 l on the heat recovery circuit at a temperature of about 70 °C. An auxiliary heating capacity of 285 kW is offered by two condensing boilers.

In winter the fan-coils receive warm water by the heat pump circuit (45 °C). AHU pre and post heating coils receive hot water from the high temperature circuit (70 °C). The two condensing boilers C1 and C2 are activated only as a backup for heat pump failure or as an integration of the heat pump in the case of an unexpectedly high heating demand (figure 1).

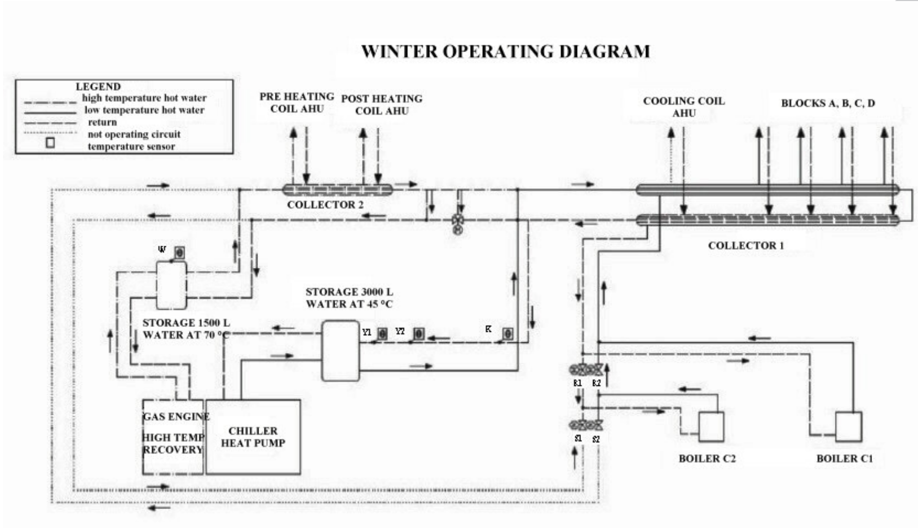

Figure 1 – *The HVAC gas engine driven heat pump circuit*

An analysis of the first year of operation was carried out [10]. The efficiency revealed lower than planned. The malfunction and low efficiency resulted due to a badly managed commissioning phase and to poor and incorrect maintenance and operation of the system. First of all heating was mainly provided by the boilers instead of by the heat pump. Moreover their set point was fixed to produce water at 75 °C. The return water was then at such a high temperature that the boiler did not work even in condensing mode. The heat pump was not continuously modulated; instead it operated with on-off cycles, with a low COP, moreover producing thermo-mechanical stresses on the engine and an over modest heat recovery utilisation. The above described control logic errors made a continuous operation of boiler C2 with only some minutes per hour use of the heat pump. This was the main reason of the poor plant performance in its first year.

The different plant performance with correct control compared to the previous bad operation was evaluated by some test days in winter heating, monitoring the natural gas consumption and the outdoor temperature, to obtain the gas consumption in terms of Nm^3 per "degree-hour" (dh). The plant was operated for three test days in the design mode (heat pump priority) and for other three test days running only the condensing boilers. The results are summarised in Table 1. Condensing boiler mode led to a consumption well higher than 50% with respect to the design operating mode thus demonstrating the great efficiency of the solution of a gas driven heat pump. An evaluation of the annual consumption through the previous estimate of the consumption expressed as Nm^3 per dh was conducted for the town of Vicenza (around 28,000 dh, from 15th October to 15th April). The annual heating average cost passed from 15,059 € in design operating mode to 21,830 € with condensing boilers.

Table 1. *Natural gas consumption per dh recorded respectively in design operating mode and in condensing boiler mode (the consumption per degree-hours is averaged on the three days of each test run)*

Mode	Gas consumption in the test run [Nm^3]	Degree hours [dh]	Consumption per degree-hour [Nm^3/dh]
Design operation (HP + condensing boiler)	70.10	91.1	0.77
Condensing boilers	87.40	77.0	1.14

3 The second example: ground water heat pump for a historical building

The Basilica Palladiana is a historic building, designed by Andrea Palladio in the 16th century, sited in one of the most famous square in the centre of Vicenza. In 2007 the municipality decided to make it a cultural centre for the city. Consequently a HVAC plant was installed for the requirement of air conditioning for many people expected to attend the exhibitions. The HVAC plant is an open loop water source heat pump system using underground water as heat source or as heat sink. Two wells were built (distance about 50 m) to produce and inject the water from the layer (40 m deep).

In the territory of Vicenza a strict regulation limits the use of underground water in heat pump plants for environmental reasons (the fear of damaging the potable aquifers). The local Authority provided a special authorisation to the plant to have a benchmark for the use of underground water for air conditioning uses. The authorisation prescribed the installation of a data logging system in order to evaluate the energy and environmental performance of the HVAC plant. The underground water temperature at 14 °C is used as heat pump source (during heating season to supply hot water at 45-40 °C) or as chiller heat sink (during cooling season to supply cold water at 7-12 °C).

Three main circuits must be considered (figure 2) :

- underground water layer circuit: the well water is circulated in an open loop by two pumps, one backup, 18 kW nominal power at 70 m water column head each, controlled by inverter. Sand filters are provided. The heat exchange with the primary circuit is allowed by two stainless steel heat exchangers (one backup), 700 kW nominal power each. The production well pump is controlled (by an inverter) to provide a flow rate as a function the condensation/evaporation water circuit requirement;
- primary circuit: it avoids the direct use of the underground water. The circuit is equipped with two pumps (cooling pump in figure 2, one backup, constant flow rate – 90 $m^3 h^{-1}$ –, 5 kW power, 12 m head). The heat pump/chiller is an electrical water/water one, six scroll compressors set up in two parallel circuits, R410A as refrigerant; the compressors operate by on/off and step by step logic.

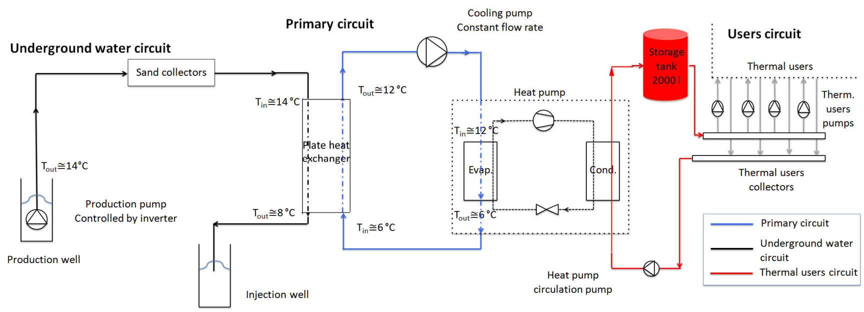

Figure 2 - *Scheme of the underground water open loop heat pump plant with the three main circuits*

- users circuit: hot (45 °C) or cold (7 °C) water exiting the heat pump/chiller feeds a storage tank (2000 l) that is connected by two pumps (one backup, constant flow rate – 85 $m^3\ h^{-1}$ –, 3 kW, 8 m head). Finally, pumps, controlled with inverters, bring hot/cold water to each user circuit.

The large showroom is set up by radiant floor as water terminal unit and displacement ventilation with very low velocity diffusers fed by two air handling units (15000 $m^3\ h^{-1}$ each). A smaller size air handling unit (1500 $m^3\ h^{-1}$) supplies the ventilation of the ticket office. A 600 kW (nominal thermal power) plate heat exchanger connects the plant to the local district heating network (heating backup). During cooling season the backup service provided by a water/water electrical chiller (rated cooling power 330 kW, rated electrical power 66.9 kW, EER=4.95).

A supposedly secondary element of the plant consists of two direct expansion air conditioners (CDZ) which serves two small technical rooms (transformers and general switchboard and Uninterruptible Power Supply (UPS) rooms): they are cooled by the ground water circuit as well.

The first period of the monitoring activity was April-September 2014. Data analysis led to the following observations [11]:

- electrical energy consumption of the primary circuit pumps was extremely high (3000 kWh_{el} per month, 44 % of the total electricity consumption). The underground circuit pumps operated 24 h a day even in periods with very low or null cooling load. The problem arose from the need of cooling the CDZ by underground water in every season. Then the CDZ resulted responsible for 22 % of the total energy use of the pumps;
- the backup chiller consumed 300 kWh_{el} per month constantly only for stand-by operation;
- electrical consumption of the CDZ was quite constant in the range 1400-1700 kWh_{el} per month. When the only cooling load is the CDZ, system operation is extremely inefficient.
- considering that the HVAC plant did not operate during the hottest period (June and July), the electrical consumption in the whole period April-September 2014 was quite high (35500 kWh_{el} excluding the backup chiller), especially due to the continuous operation of the primary circuit pumps even when the HVAC plant was off;
- energy performance indexes revealed fully the poor performance of the plant. In fact the EER_{eq} (energy efficiency ratio of heat pump/chiller equipment) resulted low with respect the design value (3.23 instead of a designed value of 5), but the EER_{tot} (energy efficiency ratio of the whole plant, that is considering the auxiliary needs) resulted really at the awful value 1.21;
- underground water consumption index Q_{wat} (defined as the ratio between the underground water produced and the useful cooling energy) was higher than designed: 223 l kWh_{cool}^{-1} instead of 172 l kWh_{cool}^{-1}. During the off operation periods a consumption of 1000 m^3 $month^{-1}$ was surveyed: clearly the system did not operate correctly.

The oversize of the ground circuit pumps (100 kPa was the real head at the maximum flow rate (80 m^3 h^{-1}) against 500 kPa nominal head of the two installed pumps) was the main reason of the large ground water flow rate with limited control capacity. This gave rise to a very discontinuous operation of the production well pumps with poor modulation capacity and very high on-off frequency (20-30 on-off per day during very low cooling needs periods).

In October 2014 the local Authority approved a series of technical interventions (realized between April and August 2015), after having considered the very poor energy performance and technical problems detected during the first period of operation:

- a modification of the injection well lengthening the tube as to avoid excessive oxygenation and bubbling of water that could cause metals precipitation and well obstruction (frequently occurred during 2014 and 2015);
- improvement of the field operation of the pumps by insertion of two sensors and a balancing valve in the production well to verify the temperature and piezometric gradients;
- equipping the second pump with an inverter, modifying the operation logic of the production well pumps by increasing the operation range;
- modifying the condensation heat recovery system of the ground water chiller and the backup chiller connecting directly the primary circuit to the post-heating coils;
- installing a separate dry-cooler for the CDZ.

The plant behavior after interventions is summarized in Fig. 3. The EER improvement was higher than 40% in summer operation. Above all the system EER improvement was really impressive, passing from 1.21 to 3.80. The main contribution was the strong reduction in electrical energy consumption of the primary circuit pumps, also separating the CDZ from the ground water circuit.

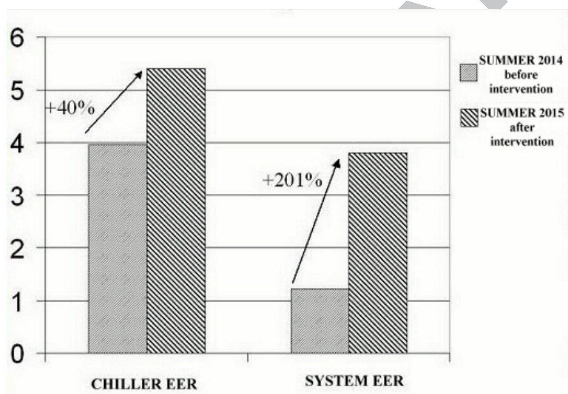

Figure 3 - *Comparison of the Energy Efficiency Ratio for the two periods before and after the interventions*

3 The third example: dual source heat pump heating of a school building

A High School Building was built up starting its operations in autumn 2009 by the town of Agordo. The town is located in the geographical area of the Dolomiti mountains in a valley at 611 m asl, where the climate is severe during wintertime (3376 degree-day). The main data describing the building are the following:

- total floor area of 5,680 m^2;
- an outward surface of 13,608 m^2;
- a gross heated volume of 19,644 m^3.

- the average thermal transmittance of the outer walls and the roof is 0.16 $Wm^{-2}K^{-1}$;
- the thermal transmittance of the floor to the ground is 0.4 $Wm^{-2}K^{-1}$;
- the thermal transmittance of the glazing is 1.38 $Wm^{-2}K^{-1}$.

The building can then be considered well insulated and a low energy building.
The plant was designed to serve only heating and ventilation, as the building is closed in summertime and the demand for hot tap water is negligible. A simplified functional diagram of the plant is shown in Fig. 4 reporting only the main hydraulic streams and energy flows within the plant.

The HVAC plant is divided into two sections: the space heating and the ventilation. The heating section is equipped with two ammonia-water absorption heat pumps (HP3 and HP4 in figure 4) 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. A 50 m² area of flat type solar heat collectors is provided (4 arrays in parallel, each of those made of 5 modules in series).

The solar section control is particular as it is based on a “threshold radiation”, operating in three different modes. When solar radiation is high enough to allow a suitable temperature (considered the efficiency curve of the collector and the outside temperature), solar heat is used for direct heating, otherwise the possible operation temperature is compared with the ground and the heat pump takes heat either from solar collectors or the ground accordingly. Finally in summer solar collectors recharge the ground. The ventilation section serving the AHU hot batteries is also equipped with two ammonia-water absorption heat pumps (HP1 and HP2 in figure 4) with geothermal heat exchangers (750 m, 6 x 125 m in a row, of vertical tube heat exchangers). The heat pumps produce thermal energy at 55-60 °C to feed the AHU. A heat recuperator is installed downstream of the AHU: it is a static cross flow type plate heat exchanger with an efficiency of 50 %.

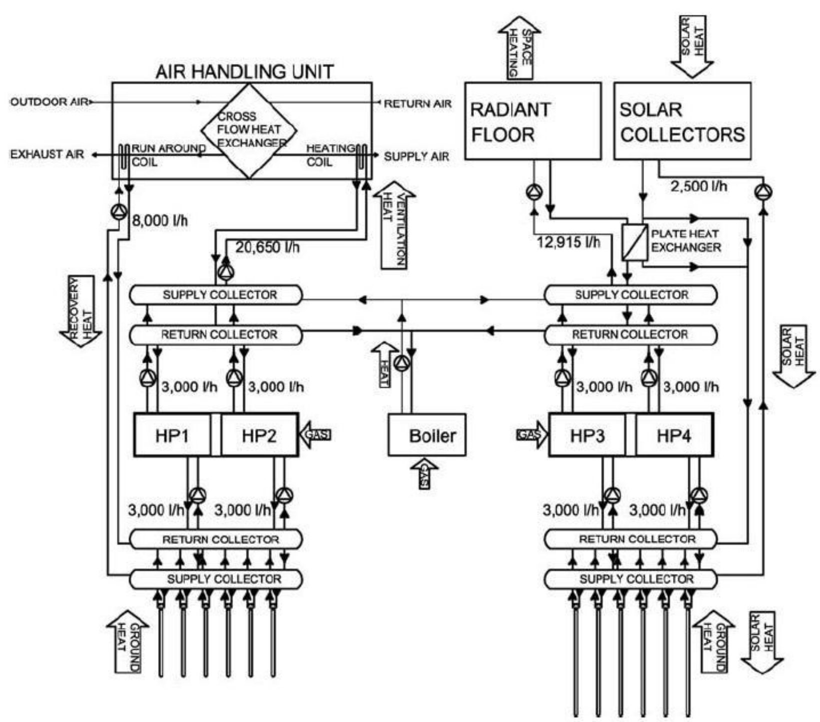

Figure 4 – *Schematic of the dual source heat pump for heating a school building*

After the cross flow heat exchanger run-around coils are present. They recover heat to be sent to the evaporator of the absorption equipment, when the external temperature exceeds 0 °C.

The monitoring of the plant was set-up with the cooperation of the building's and the plant's controller designers. The following cumulative energy flows measured by simple thermal energy meters were hourly recorded:

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

The monthly natural gas bills supply the values of Natural Gas (NG) consumption, with all consumption being attributable to the heating/ventilation system.

As the surveys of the first two years of operation were satisfactory from the point of view of performance [12,13], no other analysis of the recorded data was operated till recently. Moreover the gas consumption changed not so much from one season to the other, supposedly according the degree-days of the various years.

In June 2017 a careful analysis was carried out for the data recorded during the period May 2012-April 2017 [14].

The analysis of the last 5 years revealed quite a different situation. Whereas the ventilation section behaved not so differently from the first two years, this was not the case of the space heating section (figure 5). Just in the middle of 2013-14 winter heat pumps no longer operated with a sudden fall in the Primary Energy Ratio (PER).

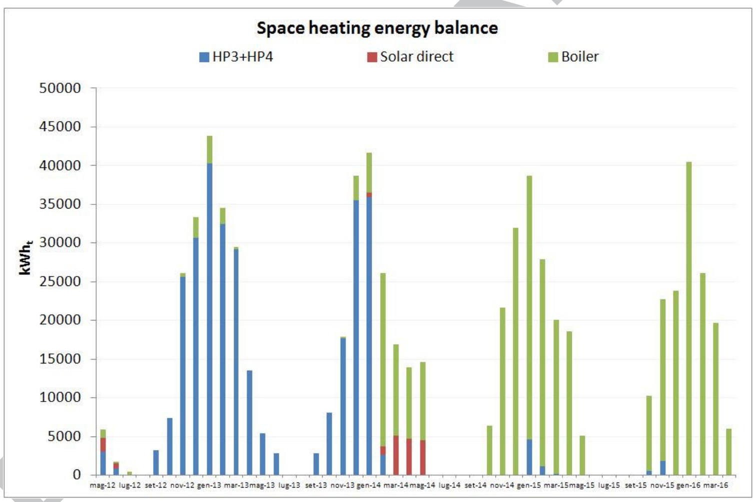

Figure 5 - *Space heating energy balance of heat pumps (HP), solar direct and boilers during 4 heating seasons*

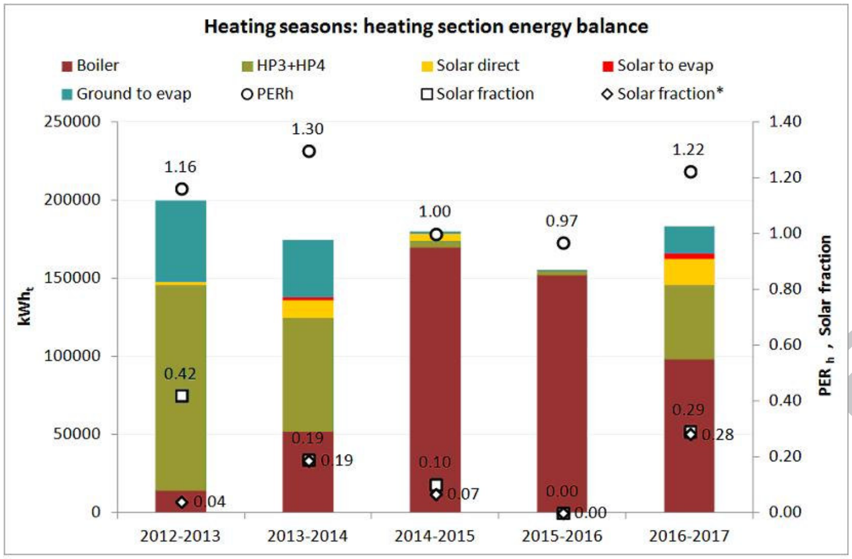

Figure 6 - *Seasonal energy contribution from the different sources (energy from heat pumps generators (HP3+HP4), energy from heat pumps evaporators both from ground and solar, solar direct, and energy from boiler, expressed as thermal energy) to the heating energy needs. The primary energy ratio, the solar fraction (as defined above) and the solar fraction* (without considering the use of solar energy for summer ground regeneration) are reported as well)*

The effect can be easily seen in figure 6, where for every monitored season the contribution of the different plant componen[illegible]epresented. The PER falls from 1.30 in 2013-14 to 1.00 the following season and to 0.97 in 2015-16. It returns to 1.22 during last heating season thanks to the interventions described below. The figure reports also the 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 + regeneration of the ground during summer months) and the total incident solar radiation on the collectors. Most of the energy was suddenly supplied by the boiler unlike during the previous periods in 2014-15. During the left part of the season the direct solar energy contribution raised system performance. However in the following winter, the solar section was completely excluded and the heat pumps occasional contribution appears as an auxiliary integration to a system where the priority is assigned to the boilers.

Seemingly at first the solar section was excluded from the heat pump circuit due to a failure of only one solar collector. Even after fixing the solar collector problem, no circulation was reactivated between solar section and heat pump and some solar direct contribution was provided. The final choice of the management was to rely completely on boilers. The result was, as highlighted above, a reduction of the seasonal PER to values lower than 1.

Some revealing considerations are offered by a careful examination of figure 7, where a comparison is carried out from one season to another regarding the annual thermal energy supply, the natural gas (NG) consumption and the heating degree days (HDD) together with the percentage variation of supplied energy and NG consumption with respect to the first heating season. The figure well illustrates that the comparison based on NG consumption only can be misleading. For example in the season 2015/2016 the NG consumption was 10

% lower than in the first (reference) season (2012/2013) with a small difference in HDD (3795 against 3608): however it is the season with the lowest PER_{tot} (0.97). As a matter of fact, the supplied thermal energy in 2015/16 season was more than 20 % lower with respect the reference season (205650 vs. 258478 kWh_t) while NG consumption reduced by less than 10 %.

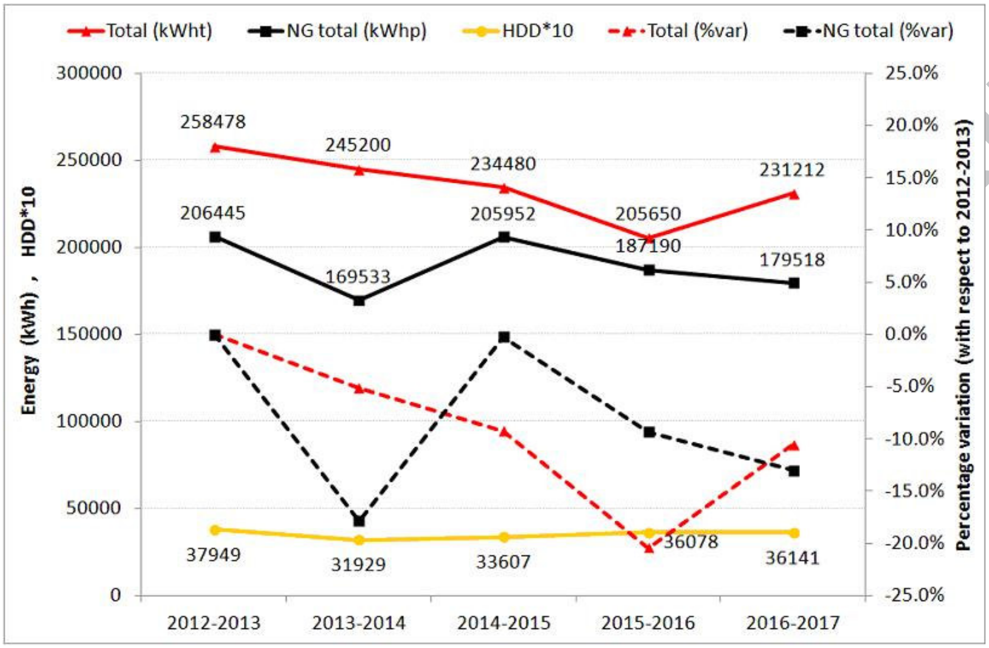

Figure 7 – *Seasonal total thermal energy needs (in kWh_t), NG consumption (in kWh_p), and heating degree days (HDD). The percentage variations with respect to the first heating season are reported as well*

The difference was mainly due to the different values of PER_{tot}. Another meaningful comparison is between 2015/2016 season vs 2013/2014 (the one with the highest efficiency): during the latter period NG consumption was 9 % lower while supplied thermal energy was 16 % higher. The survey on the season 2013/14 of a thermal demand 16% higher even with a 11% lower HDD suggests that other than HDD variables influence the building energy demand. Two possible examples are different settings in the room thermostats, not only with values lower than 23-24 °C that were recorded during the first years of operation, but with some temperature attenuation during Christmas holidays.

Another possible reason is a different seasonal solar irradiation: in effect horizontal total solar radiation in November+December was 63 kWh m^{-2} in 2014 and 102 kWh m^{-2} in 2015.

The above analysis brought back the plant to the initial good performance with correct control setting and repair of failures. The effects are well illustrated by figure 8 where the contribution of the various plant component is given on a monthly basis for the last season.

The corrective action was completed on January 20th 2017.

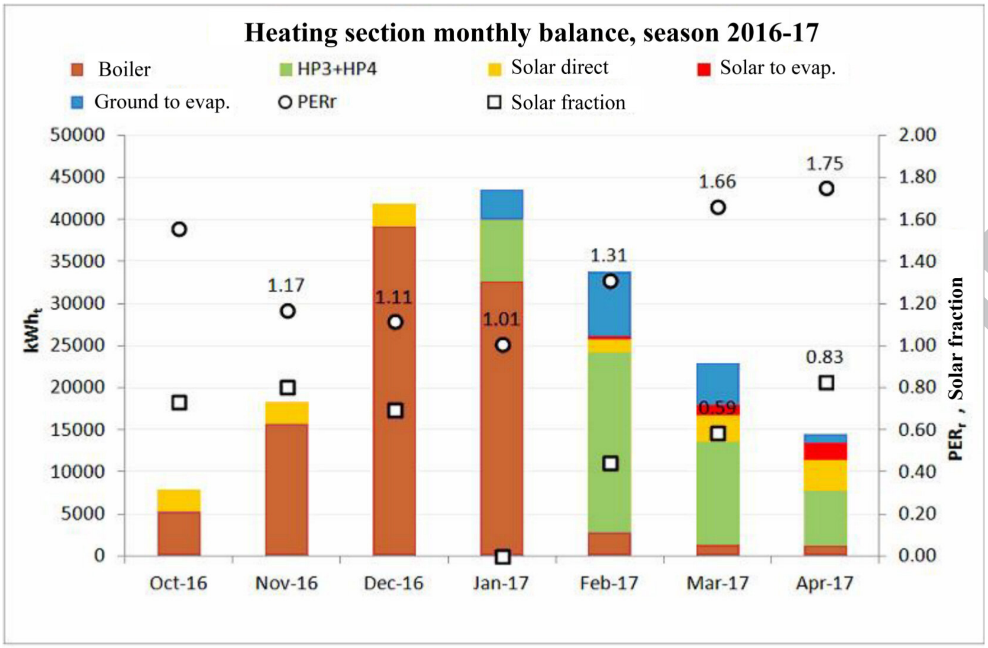

Figure 8 - *Monthly energy contribution from the different sources (energy from heat pumps generators (HP), energy from heat pumps evaporators both from ground and secondary sources, heat recovered by static recuperator, solar direct and energy from boiler, expressed as thermal energy) to the global (heating+ventilation) energy needs, total PER and solar fraction*

To summarize the above analysis in economic terms, if the HVAC plant had been managed in 2015/16 season as in 2013/14 probably NG consumption would have been 25% lower.
The improved performance of plant during the last heating season due to a more thorough operation is well confirmed: with respect to the previous 2015/2016 season, the supplied thermal energy increased by 10 % while the NG consumption decreased by nearly 4 %, even if the intervention was just in the middle of the heating season.
Due to the highlighted errors by the above analysis not only the plant was restored to the design working conditions, but the management company agreed to refund the equivalent of 4816 Sm^3 of gas to the school owner, the Province Administration. At first sight the amount might look of modest entity, but a similar wrong management in a normally insulated building might produce losses even 5-6 times higher.

3 An anecdote

The Author would end with an anecdote. Some years ago he happened to design a heat recovery plant for a thermal bathing establishment in a mountain resort [15]. Here the baths, useful for the treatment of skin diseases, were offered in hourly rounds, giving rise to a sort of batch operation where water was discharged all at the same time at about 40 °C whereas the thermal source was at 27 °C. The instantaneous flow rate of the thermal source was only of 3.8 ls^{-1}, so that a thermal water storage tank was provided. A similar tank was

then built up for the discharged water with a plate heat exchanger between clean thermal water and discharged used water. Finally the discharged water fed the evaporator of a heat pump before the elimination into the sewer. The plant was duly monitored with the main temperatures in the different parts. An analysis revealed a strange temperature reduction during the night in the discharged water tank where the last bathing round discharged water was collected to be used next morning via the heat exchanger and heat pump. This reduction could be not explained with thermal losses from the tank as they well exceeded possible estimated values.
A check on the field explained the strange effect. Recently a new group of urinal built up by the thermal establishment with continuous flow of the cold tap water of the mountains. The plumber had found the most convenient way to reach the sewer via the storage tank !

4 Conclusions

The analysis of recorded data from monitoring of a plant is essential for its correct management, particularly when the plant integrates many different technologies, as it often happens in renewable energy plants. The duty of the plant may result satisfied, even if some components badly work or operate with wrong priority or even fail.

In the considered plants the heating has never been interrupted, and the gas demand sometimes was even in slight decrease, so that the management did not worry. Only the availability of data records and a careful analysis allowed to identify the bad working of the plant and the failures to achieve potential energy savings.

The examples demonstrate from different points of view the paramount importance of adequate monitoring and data analysis of a plant particularly for renewable energy, generally less known by technicians.

Reports on successes or failures can be useful “lessons learned” for future design.

To end with a citation of George Washington: “*We should not look back unless it is to derive useful lessons from past errors, and for the purpose of profiting by dearly bought experience.*”

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- Many renewable energy plants are not provided with a monitoring system
- It is often difficult to realise the bad working of a plant without monitoring it
- It is not straightforward to decide whether the energy bill is appropriate
- Three different practical examples demonstrate the importance of monitoring