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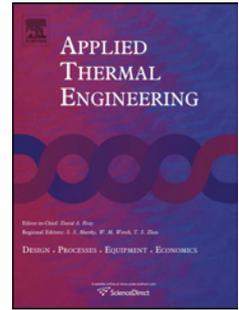
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Experimental investigation to optimise a desiccant HVAC system coupled to a small size cogenerator

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ABSTRACT

In the Mediterranean area, the increasing demand of summer cooling in residential and tertiary sectors is usually satisfied by electrically-driven units; this often determines electric load peaks and black-outs. Thus, a wide interest is spreading in small scale natural gas-fired polygeneration systems: a prime mover drives (mechanically, electrically, thermally) electric generators and/or heat pumps, desiccant wheels, etc., matching thermal (heating and cooling) and electric end-user requirements.

In this paper, laboratory tests have been considered to experimentally evaluate a small scale polygeneration system based on a natural gas-fired Micro-CHP and a desiccant-based HVAC system. Cogenerated thermal power is used for the desiccant wheel regeneration, while electric power for auxiliaries, chiller and external units. The HVAC system can also interact with electric and thermal separate “production” systems. The main results of the experimental tests are shown, stating the increase of the COP of the chiller in desiccant-based HVAC systems. Then the paper identifies the operating conditions (outdoor and supply air thermal-hygrometric conditions, electric grid efficiency and partial load operation of the MCHP) which guarantee significant primary energy savings (up to around 30 %) and CO₂ equivalent emission reductions (up to around 40 %) of the polygeneration system compared to the conventional HVAC system.

Keywords: decentralized polygeneration, Micro-CCHP, desiccant wheel, hybrid HVAC system, experimental analysis.

1. Introduction

During last years, great attention was focused on the transition from centralized to decentralized energy “production” systems (Decentralized or Distributed Generation, DG), to reduce T&D energy losses: a miniaturization process (“size” effect) is in progress.

A comparison between centralized and distributed power systems is carried out in [1]: small, modern generators can be more efficient and less costly to operate than large and old generators. This could lead to conclude that there is no longer an economy of scale in power “generation”, but large modern power units have higher electric efficiency and lower operating cost than small modern DG units based on the same technology. Since the “size” effect does not always lead to energy savings and pollutant emission reductions, there is the need to support the diffusion of on-site small complex energy conversion devices, Decentralized Polygeneration, DP. A widespread use of DP systems could allow energy, economic and environmental benefits: the benefits and drawbacks of DP are analysed in [2].

Furthermore, especially in Mediterranean area, there is an increasing demand of summer cooling energy in domestic sector, usually satisfied by electrically-driven units; this involves electric load peaks and black-outs. Thus, an increasing interest occurs in small scale polygeneration systems fuelled by natural gas.

Micro Combined Cooling, Heating and Power (MCCHP, electric power output ≤ 15 kW) systems represent typical decentralized energy “production” systems, particularly suitable for use in Mediterranean areas.

In the CCHP systems, the prime mover can be based on different technologies (Stirling, Reciprocating Internal Combustion - RIC, Fuel Cell, Gas Turbine, etc.). The most mature technology available on the market, gas-fired RIC engines, allows small installation space, high thermal efficiency, low noise, vibrations and maintenance and long life service. These engines, coupled to electric generators, can drive electric heat pumps, thermal heat pumps, Desiccant Wheels - DW, etc., allowing a wide range of operating conditions to match thermal (heating and cooling) and electric end-user requirements.

In summer, to supply cooling energy, recovered thermal energy can be used for the regeneration of the desiccant wheel, as well as for the activation of absorption/adsorption heat pumps, while, in winter, engine thermal wastes allow to satisfy domestic hot water and heating purposes. Thermal energy input for desiccant wheel regeneration can also be derived from solar source in Solar Heating and Cooling (SHC) systems [3-5]. Solar collectors are typically integrated with fossil fuel-based technologies to match the entire heat demand, SHC-CCHP systems. These systems offer the typical advantages of natural gas-fired polygeneration systems coupled to a renewable energy source. Despite of these advantages, coupling solar collectors and a CCHP unit for heating, cooling and electric purposes presents certain critical issues [6]; furthermore, this energy conversion system (solar thermal collectors, storage systems and CCHP) is very complex, both in design phase, to optimize the size of each subsystem, and during operation, its performance being dependent on many variables (energy tariff structure, load profiles, etc.).

Few investigations have been carried out on solid desiccant systems coupled to small scale combined cooling, heat and power systems, and little attention has been paid on both energy and environmental performances [7 – 10].

Therefore, this paper experimentally analyses a desiccant-based MCCHP system which includes a hybrid Air Handling Unit - AHU - (the term hybrid refers to the contemporary presence of a desiccant wheel and an electric chiller that, in succession, dehumidify and cool the air to be introduced in the conditioned space). This HVAC system is suitable above all for residential and small commercial users. An energy and environmental analysis for different operating modes has been carried out.

The system performance has been evaluated as a function of various working conditions (outdoor and supply air thermal-hygrometric conditions and MCHP partial load operation), in order to establish the effectiveness of such systems compared to a conventional HVAC system based on separate electric and thermal “production”. Finally, the influence of the electric grid efficiency to analyse the polygeneration system in different electricity mix scenarios has been considered, as well as the effect of the regeneration temperature on the energy savings.

2. Test facility

At Sannio University, in Benevento (Southern Italy), a desiccant Air Handling Unit coupled to a cogenerator based on a natural gas-fired reciprocating internal combustion engine, an electric chiller and a natural gas-fired boiler, has been experimentally analysed. The hybrid HVAC system is based on dehumidification of outdoor air by a desiccant wheel and subsequent cooling by an electric chiller.

In Fig. 1, the layout of the test facility is shown [10].

Nominal characteristics of the equipment are the following:

- cogenerator: $P_{el} = 6.0$ kW (0.200 kW for the cogenerator auxiliary electric loads), $P_{th} = 11.7$ kW, $\eta_{el} = 28.8\%$, $\eta_{th} = 56.2\%$; the MCHP supplies thermal power for the desiccant wheel regeneration by recovering heat from the exhaust gas and from the engine. The maximum thermal output is available for a water flow rate of 33.5 l/min, at a temperature in the range 60 – 65 °C. The actual flow rate (17.0 l/min) during the tests is lower than the nominal one and consequently the hot water is available at 72.5 °C. Due to thermal losses in the distribution pipes and to the effectiveness of the air-to-water heat exchanger, the maximum achievable regeneration air temperature is 65 °C; the MCHP mainly supplies electric energy for AHU electric loads (fans, pumps, desiccant wheel, etc.), chiller and further external electric devices (computers, lights, etc.).
- air-cooled water chiller: $P_{co} = 8.50$ kW, COP = 3.00;
- boiler: $P_{th} = 24.1$ kW, $\eta_b = 90.2\%$. The boiler can be either used to supply thermal energy when the hybrid HVAC is powered by separate “production” systems, or to integrate the thermal power available from the MCHP. In fact, in hot and humid climates, to reach the desired ω_s , the regeneration air temperature should be consequently increased, so thermal power is supplied also by the boiler, allowing to reach regeneration temperature of 70°C.

The AHU treats 800 m³/h of air that achieves the summer supply conditions required by the room ($T_s = 13-19$ °C, $\omega_s = 7-11$ g/kg).

There are three outdoor air streams:

- process air, dehumidified by the desiccant wheel, pre-cooled by the cooling air stream in an air-to-air cross flow heat exchanger, finally cooled to the supply temperature by a cooling coil interacting with the chiller;
- DW regeneration air, heated by the heating coil interacting with the MCHP and/or by the heating coil interacting with the boiler;
- cooling air, cooled by a direct evaporative cooler and then used to pre-cool process air.

The weight of the DW is 50 kg; its dimensions are 700 mm x 700 mm x 440 mm (height x width x thickness), with a diameter of 700 mm. In reality, the frontal area of the rotor effectively exposed to process and regeneration airflows is relative to a diameter of about 600 mm, since a circular crown of the total area is obstructed by the metallic frame in the AHU. The rotor has the following configuration: 60% of the rotor area is crossed by the process air, while the remaining 40% by the regeneration air. This is often used when there is low temperature regeneration thermal energy.

In fact, the DW is filled with silica-gel, a desiccant material that can be effectively regenerated at temperatures as low as 60-70 °C, values obtainable by using the thermal recovery from the MCHP and, only when necessary, the natural gas boiler. The rotor matrix is composed of alternate layers (smooth and wavy) of silica-gel sheets and metallic silicate, chemically bound into an inorganic fiber frame.

The so realized “honeycomb” frame has several advantages, such as the maximization of the superficial contact area, low pressure drops, low weight but high structural durability.

In Fig. 2, the Sankey diagram of the polygeneration system is shown, to highlight the main power flows and losses (outdoor thermal-hygrometric conditions: $T_o = 33.9$ °C, $\omega_o = 10.3$ g/kg; regeneration temperature 65.0 °C). For the global system and for its main subsystems, power inputs and outputs are reported, also considering electric loads of the MCHP and AHU auxiliaries (including circulation electric power absorption).

3. Energy and environmental analysis

The hybrid HVAC system has been experimentally tested in stationary conditions for a length of 30 minutes for each test. The results of the experimental analysis allowed to compare the above described alternative system, AS, to the following usually adopted conventional air conditioning system, CS.

In the latter, outdoor air is cooled and dehumidified by an electric chiller powered by the electric grid; then, it is reheated by a natural gas-fired boiler. Auxiliaries and external electric devices are powered by the electric grid.

According to the European Directive 2004/8/EC, [11, 12], the reference energy efficiencies for both electric grid and the boiler has been evaluated, with respect to Italy,:

- Electric grid: $\eta_{eg} = 45.2 \%$, CO_2 equivalent emission = $0.531 \text{ kgCO}_2/\text{kWh}_{el}$ [13];
- Boiler: efficiency = 90% , CO_2 equivalent emission = $0.20 \text{ kgCO}_2/\text{kWh}_p$; natural gas lower heating value = $9.59 \text{ kWh}/\text{Nm}^3$.

The comparison is carried out assuming that both AS and CS deliver equal energy (electric and cooling energy; thermal energy recovered from the MCHP is fully used to DW regeneration). Moreover, the two systems handle $800 \text{ m}^3/\text{h}$ of process air and operate at the same outdoor and supply thermal-hygrometric conditions. For example, Fig. 3 shows the energy flows of AS and CS.

The Primary Energy Saving, PES, has been considered for the energy comparison, [14]:

$$\text{PES} = \frac{E_p^{CS} - E_p^{AS}}{E_p^{CS}} \quad (1)$$

Furthermore, the avoided CO_2 equivalent emissions of the alternative system with respect to the conventional one have been evaluated too.

4. Experimental analysis

The start-up of the experimental test facility (calibration of the measuring systems, design of data acquisition and monitoring software, etc.) has been initially carried out. A user-friendly graphical interface records the main parameters, such as: air temperature and relative humidity in the main AHU sections, inlet and outlet water temperature for heat exchangers, boiler, MCHP and chiller, volumetric

flow rate (water and natural gas), electric energy and power flows. So, thermal and electrical MCHP efficiencies, COP of the chiller, PER, etc., have been evaluated too.

After a starting phase [10], during summer 2009, many tests have been carried out, representing a wide range of outdoor air thermal-hygrometric conditions and different AHU operating modes.

4.1 Comparison between AS and CS

AHUs based on chemical dehumidification have the advantage of reducing cooling energy demand, for the lack of cooling-based dehumidification, on which the conventional air conditioning systems are instead based. Furthermore, the refrigeration unit can produce chilled water at higher temperatures, compared to a chiller that operates in a conventional HVAC system, and consequently it will operate with higher COP.

For these reasons, attention has been paid to the evaluation of the performance of the electric chiller in the CS: a detailed model, based on well known simulation codes of inverse machines, allows to evaluate the performance of the air-cooled water chiller interacting with external secondary fluids, air and water [15, 16]. Both full and part load operating conditions have been considered, in agreement with literature [17].

Many tests have been realized, considering different operating conditions and electric energy quantities supplied to the final user.

In Fig. 4, the full load COP of the chiller in alternative and conventional HVAC systems is reported as a function of T_o and for different ω_s values. The COP in the alternative system (“DW”) obviously does not depend on ω_s because the dehumidification is carried out by the desiccant wheel, not by the chiller.

Contrariwise, the COP of the chiller interacting with the cooling/dehumidification coil strongly decreases on ω_s reduction. For many operating conditions, the chiller interacting with the hybrid HVAC system performs better than the conventional one that has also a “size” (cooling capacity) more than twice. Only for low outdoor temperature (< 29 °C) and high values of supply air humidity ratio (> 10.8 g/kg) COP of the chiller interacting with the conventional AHU is greater than the alternative one.

To highlight the influence of outdoor air properties on the energy performances of AS and CS, in Fig. 5 outdoor air thermal-hygrometric conditions that get a positive PES are shown. The hybrid HVAC system

interacting with the MCHP requires less primary energy than the conventional system for ω_o lower than about 11.5 g/kg and T_o in the range 25-36 °C. For $\omega_o > 11.5$ g/kg, the lower limit of the previous temperature range increases: for $\omega_o = 13.0$ g/kg, AS is preferable only for $T_o > 28$ °C. Finally, for $\omega_o > 13.0$ g/kg, AS is no more energetically suitable.

In Fig. 6, the PES as a function of ω_o and for three different values of T_o is shown. PES increases when ω_o decreases, reaching a maximum value for $\omega_o = 8.00$ g/kg (24 %, 31 % and 35 % for T_o equal to 25.0 °C, 29.0 °C and 33.5 °C, respectively). PES becomes positive when ω_o is lower than a certain value, depending on outdoor air temperature (11.4 g/kg, 12.6 g/kg and 13.0 g/kg for the three T_o values, respectively). Moreover, PES increases with T_o .

In the range of operating conditions of Fig. 6, the avoided CO₂ equivalent emissions show the same trend as PES, achieving a maximum value of 43%. The avoided emissions go to zero when ω_o is in the range 14.5-16 g/kg (depending on T_o value), while PES goes to zero when ω_o is in the range 11.2-13.1 g/kg. Therefore, the ambient convenience remains in a wider outdoor humidity ratio range compared to the energy convenience.

As explained above, the electric grid efficiency remains a key factor. In Fig. 7, at constant operating conditions (Table I), the influence of η_{eg} on PES is reported. In the base case ($\eta_{eg} = 45.2$ %), PES is about 22 %; then it increases with η_{eg} reduction.

The influence of the partial load operation of the MCHP on the global energy performance has been analysed too. The net electric power for computers, lights, etc., has been gradually increased up to 1.5 kW to allow the full load of the cogenerator (for $T_o = 29.5$ °C and $\omega_o = 10.2$ g/kg, the electric power supplied by the MCHP to the chiller and to the MCHP and AHU auxiliaries is about 4.5 kW). Fig. 8 shows that PES increases with the net electric power supplied to the final user, so it is convenient to operate the MCHP at full load for the maximum number of hours (PES of around 24 %): this, in fact, causes an increase of electric efficiency of the micro-cogenerator. Also in this case, the reduction of CO₂ equivalent emissions show the same trend as PES; it achieves the maximum value (35%) at full load operation of the MCHP.

In a previous paper, the influence of the regeneration temperature on the selected DW performance has been analyzed, [18]. It has been shown that the dehumidification capability of the wheel, $\Delta\omega$, increases when regeneration temperature rises: with $T_o = 30.5$ °C and $\omega_o = 10.7$ g/kg, $\Delta\omega$ varies from 1.0 g/kg to 5.6 g/kg when regeneration temperature increases from 34 to 72 °C.

It is evident that the reduction of process air humidity ratio that the DW realizes, $\Delta\omega$, is strictly related to the room latent load it has to balance. Therefore the regeneration temperature can be used as a control parameter, in desiccant-based HVAC systems, to dynamically follow the latent load to handle. If, for example, the latter decreases, the regeneration temperature can be automatically reduced.

If the MCHP continues working at full load, as this is the best operating condition in terms of energy performance (see Fig. 8), the reduction of the regeneration temperature can be achieved, at constant regeneration air flow rate (800 m³/h), only by wasting a part of the available thermal power. Therefore the PES obviously reduces. In fact, in the same outdoor operating conditions as above ($T_o = 30.5$ °C and $\omega_o = 10.7$ g/kg), it decreases from 30% to -10 % when the regeneration temperature reduces from 65 to 34 °C.

The desiccant-based HVAC system can also be powered by separate “production” of electric and thermal energy, respectively provided by the electric grid and a natural gas boiler. Considering the above reported values for the boiler (90%) and the electric grid efficiency (45.2%), the desiccant-based HVAC system powered by the MCHP (AS) guarantees an average PES of 18% with respect to the same system powered by separate “production”. Even if the electric energy “production” is based on the Best Available Technology (BAT, i.e. with natural gas combined cycle power plants, $\eta_{eg} = 0.58$), PES remains positive (8.2% on average).

The PES values obtained in this work are in good agreement with quite similar polygeneration systems and operating conditions [7-8].

5. Conclusions

Experimental tests have been carried out in Benevento (Southern Italy) to analyse a hybrid desiccant HVAC system driven by a MCCHP, in order to evaluate its energy and environmental performances compared to a conventional system based on an electric chiller.

The performances of the two systems are strongly influenced by outdoor thermal-hygrometric conditions: the alternative system can guarantee a Primary Energy Saving when outdoor air humidity ratio is lower than a certain value (11.5 g/kg) and outdoor air temperature is in the range 25 – 36 °C. Moreover, PES increases with T_o and when ω_o decreases, reaching a maximum value for $\omega_o = 8.0$ g/kg, 24 % for $T_o = 25.0$ °C, 31 % for $T_o = 29.0$ °C, 35 % for $T_o = 33.5$ °C. The avoided CO₂ equivalent emissions show the same trend as PES, achieving a maximum value of 43%.

Obviously, PES decreases when electric grid efficiency increases.

Moreover, the best performance (PES of around 24 %, considering the operating conditions of the specific test) is obtained when the MCHP works at full load; so a detailed analysis of the electric loads and a correct sizing of the small-scale cogeneration system have to be carried out to minimize the operating hours at partial load. Also in this case the reduction of CO₂ equivalent emissions show the same trend as PES; it achieves the maximum value (35%) at full load operation of the MCHP.

With the MCHP working at full load, if the regeneration temperature has to be automatically decreased for control purposes (to reduce the latent load balanced by the desiccant wheel), the PES decreases until it becomes negative for very low regeneration temperatures (minus than 42 °C).

Furthermore, the desiccant-based HVAC system powered by the MCHP (AS) guarantees an average PES of 18% with respect to the same HVAC system powered by separate “production” of electric and thermal energy. Even if the electric energy “production” is based on the Best Available Technology (BAT, $\eta_{eg} = 0.58$), PES remains positive (8.2% on average).

Further investigations are required as regards economic savings and the introduction of a solar collecting system in the existing polygeneration plant.

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Nomenclature

CO₂ Carbon dioxide equivalent emissions [kg/h]

E Energy [kJ]

P Power [kW]

PES Primary Energy Saving [-]

r_e Electric energy share provided to the chiller [-]

RH Relative Humidity [%]

T Temperature [°C]

η Efficiency [-]

ω Air humidity ratio [g/kg]

$\Delta\omega$ Air humidity ratio variation [g/kg]

AHU Air Handling Unit

AS Alternative System

BAT Best Available Technology

CCHP Combined Cooling Heating and Power

CHP Combined Heat and Power

COP Coefficient Of Performance

CS Conventional System

DG Decentralized or Distributed Generation

| | |
|-------|---|
| DP | Decentralized or Distributed Polygeneration |
| DW | Desiccant Wheel |
| HVAC | Heating Ventilation and Air Conditioning |
| MCCHP | Micro Combined Cooling Heat and Power |
| MCHP | Micro Combined Heat and Power |
| PM | Prime Mover |
| RIC | Reciprocating Internal Combustion |
| SHC | Solar Heating and Cooling |
| T&D | Transmission and Distribution |

Subscripts

| | |
|-----|-------------------|
| AHU | Air Handling Unit |
| b | boiler |
| co | cooling |
| o | outdoor air |
| eg | electric grid |
| el | electric |
| p | primary |
| s | supply air |
| th | thermal |

Superscripts

| | |
|----|---------------------|
| AS | Alternative System |
| B | Boiler |
| CS | Conventional System |
| CH | Chiller |
| EG | Electric Grid |
| US | End User |

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FIGURE CAPTIONS

Fig. 1 - Layout of the test facility

Fig. 2 - Sankey diagram of the polygeneration system

Fig. 3 - Energy flows of Alternative System and Conventional System

Fig. 4 - COP of the electric chiller in the conventional HVAC system (for different values of supply air humidity ratio) and in the desiccant-based HVAC system, as a function of the outdoor air temperature

Fig. 5 - Psychrometric chart showing the area where PES is <0 and that where PES is >0

Fig. 6 - PES as a function of outdoor air humidity ratio for three different values of outdoor air temperature

Fig. 7 - PES as a function of electric grid efficiency

Fig. 8 - PES as a function of net electric power supplied to the final user

Table I - operating conditions for the test of Fig. 7

| Outdoor air temperature [°C] | Outdoor air humidity ratio [g/kg] | Supply air temperature [°C] | Supply air humidity ratio [g/kg] | Regeneration Temperature [°C] |
|---|--|--|---|--|
| 33.9 | 10.3 | 20.1 | 6.50 | 65.0 |

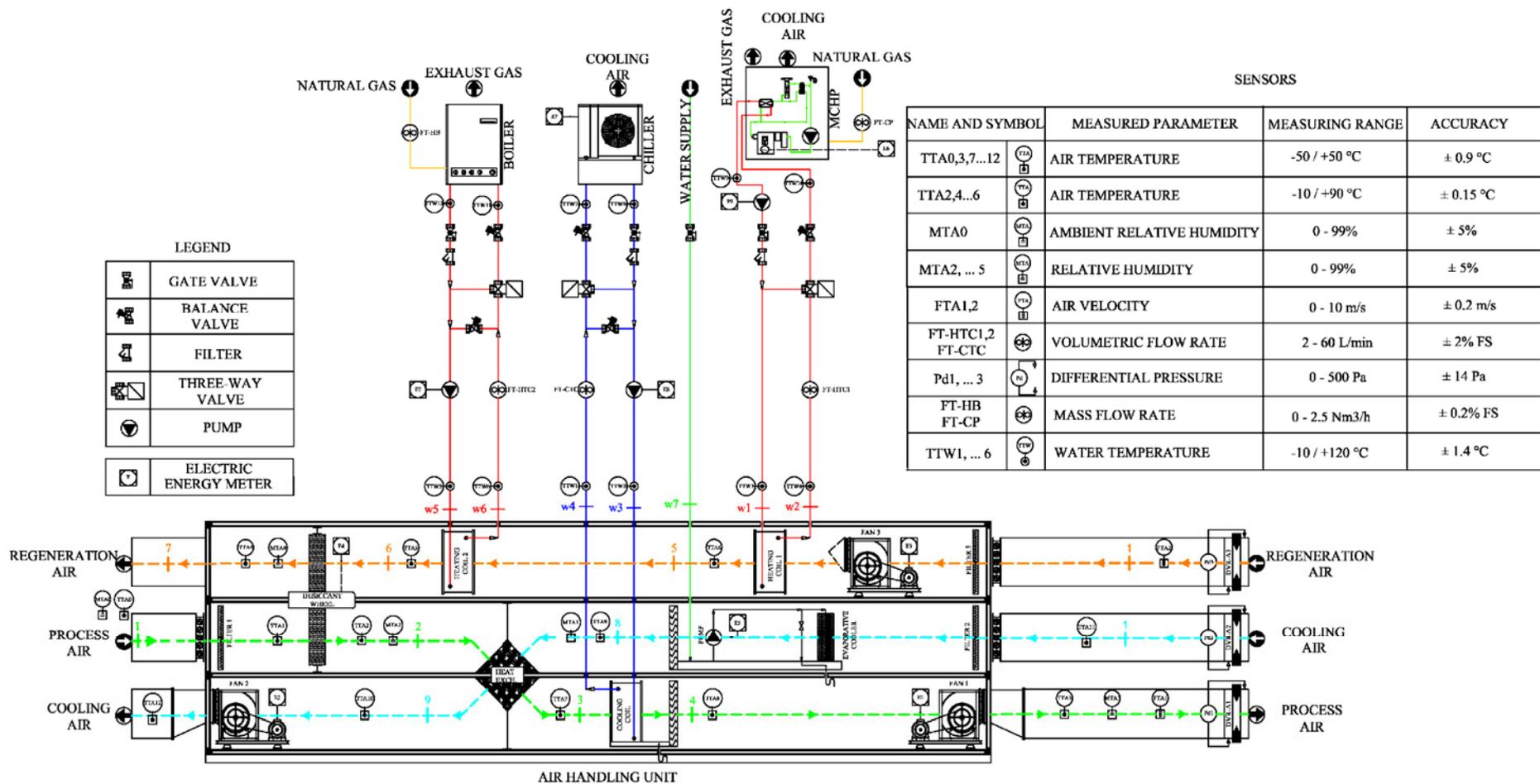


Fig. 1

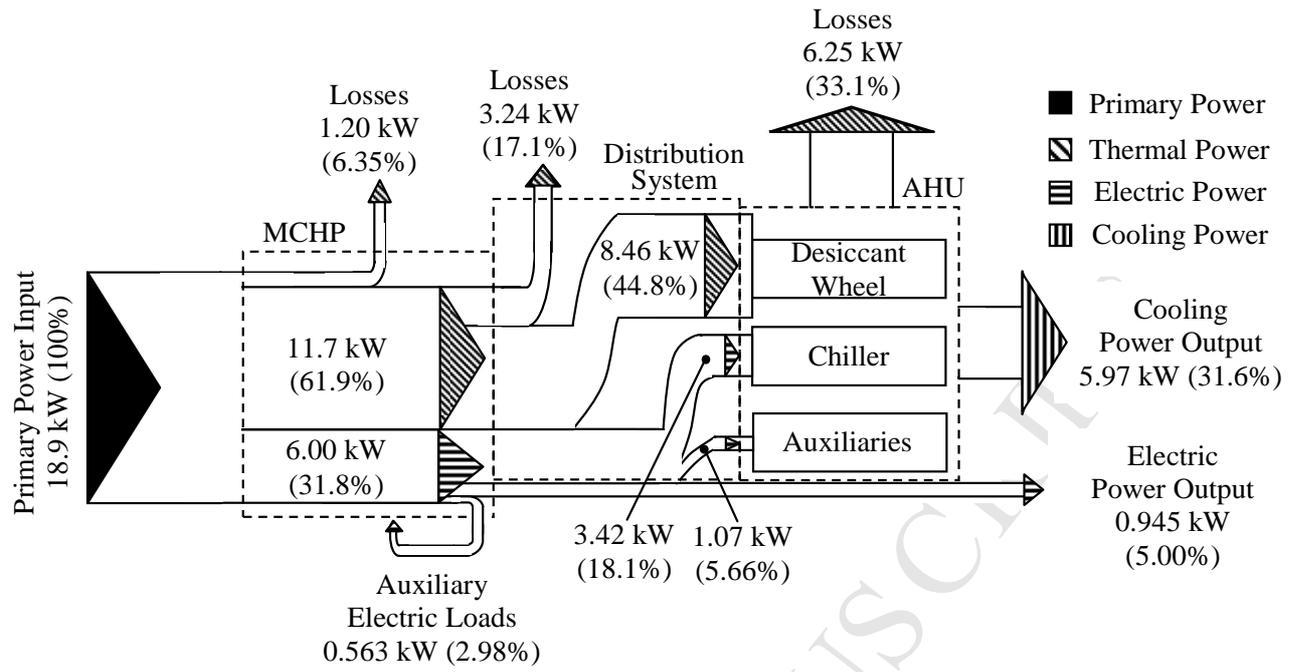


Fig. 2

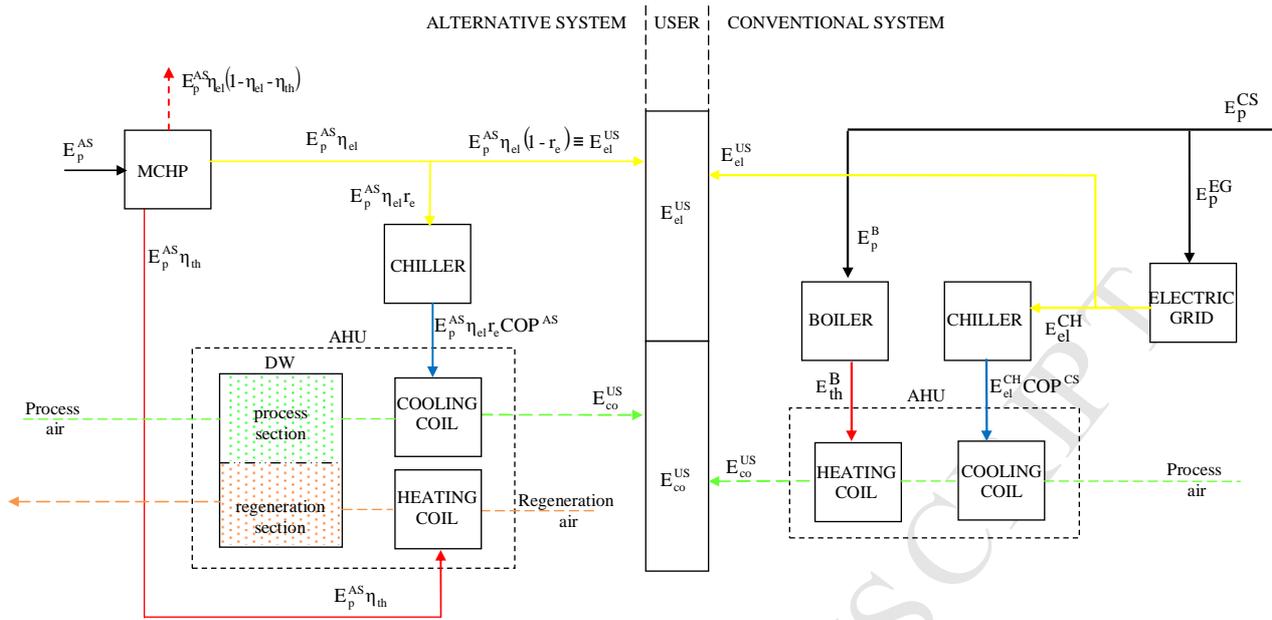


Fig. 3

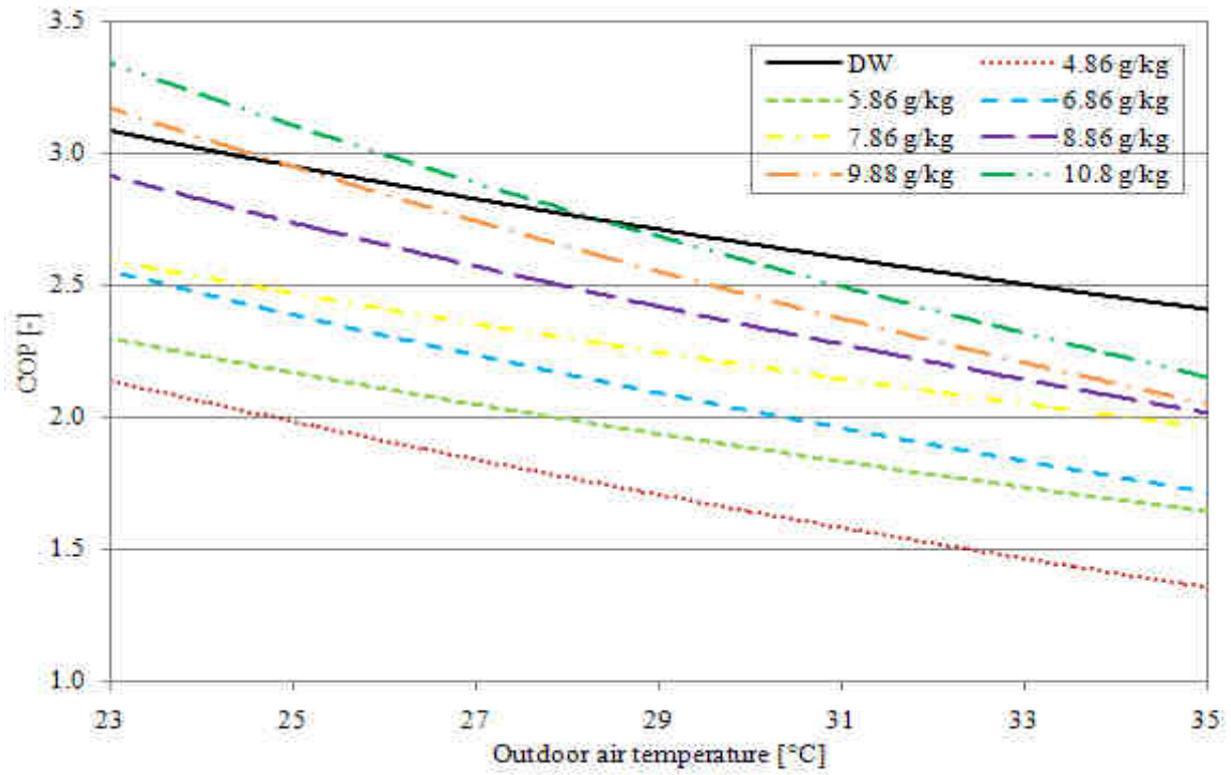


Fig. 4

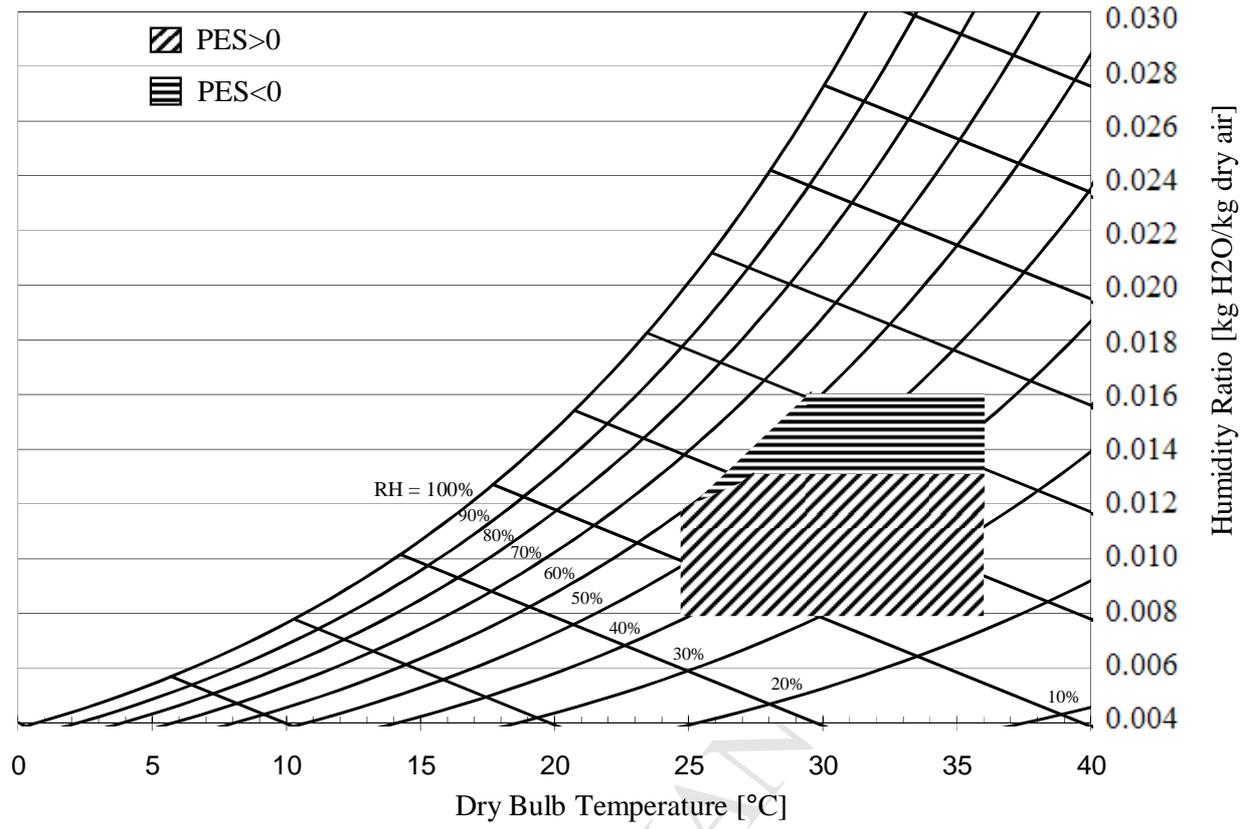


Fig. 5

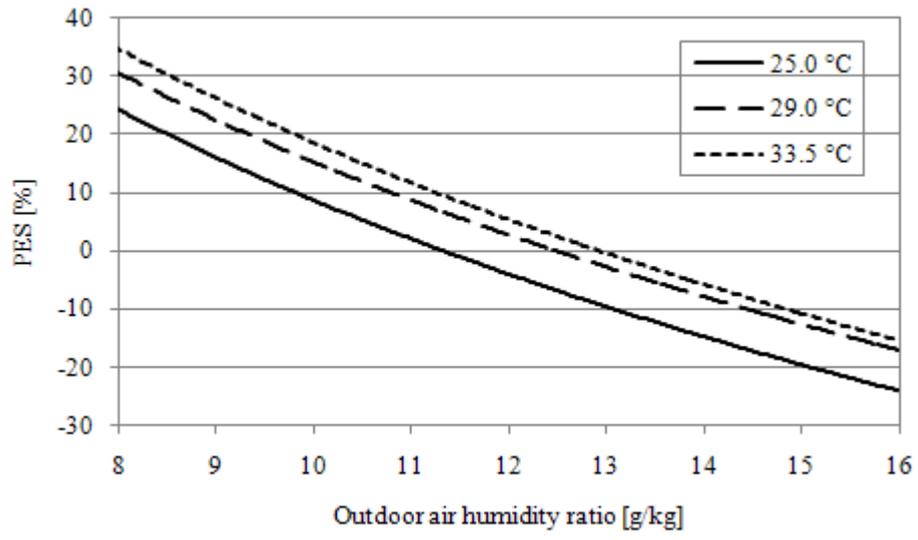


Fig. 6

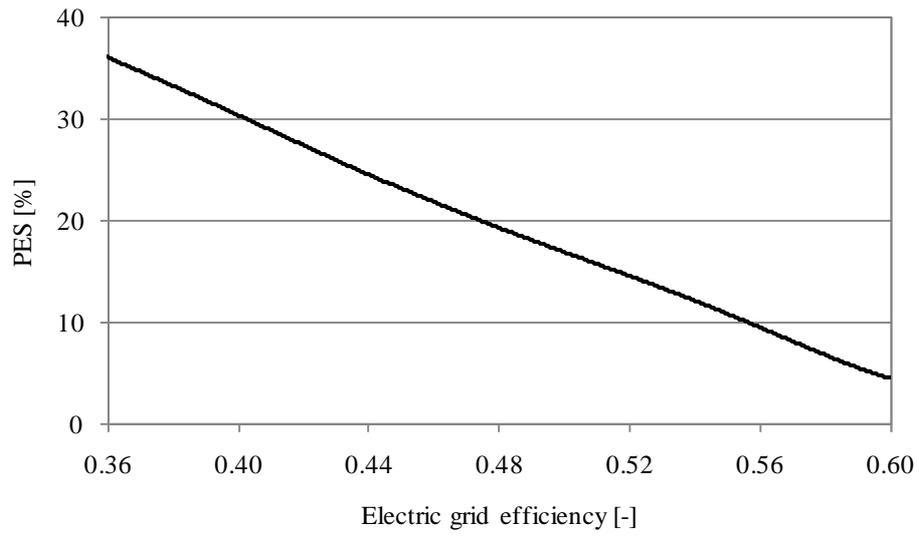


Fig. 7

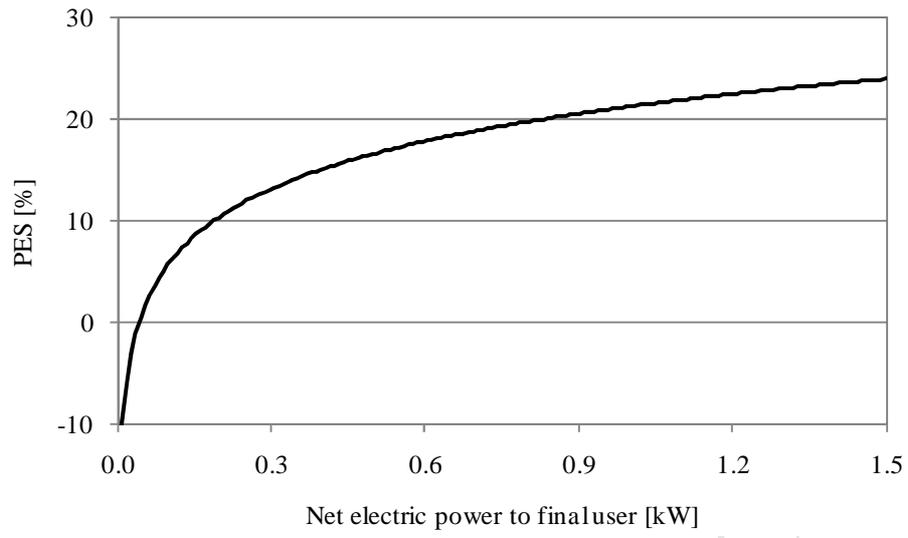


Fig. 8