Potential of HVAC and solar technologies for hospital retrofit to reduce heating energy consumption

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Abstract. The study presents a combination of several energy efficient technologies together with their potential to reduce the energy consumption and to increase the comfort through the retrofit of a hospital building. The existing situation is characterized by an old and inefficient heating system, by the complete missing of any ventilation and by no cooling. The retrofit proposal includes thermal insulation and a distributed HVAC system consisting of several units that includes air to air heat exchangers and air to air heat pumps. A condensing boiler was also considered for heating. A solar thermal system for preparing domestic hot water and a solar photovoltaic system to assist the HVAC units are also proposed. Heat transfer principles are used for modelling the thermal response of the building to the environmental parameters and thermodynamic principles are used for modelling the behaviour of HVAC, solar thermal system and photovoltaic system. All the components of the heating loads were determined for one year period. The study reveals the capacity of the proposed systems to provide ventilation and thermal comfort with a global reduction of energy consumption of 71.6 %.

1 Introduction

The study is enrolled in the global effort dedicated to the reduction of energy consumption in buildings, which according to [1] accounts 40% of final energy, in Europe.

Due to their low thermal efficiency the existing buildings are responsible of most of this energy consumption [2-4] and 95% of the existing buildings do not achieve energy efficiency standards and their retrofit provides a lot of potential for reducing the energy consumption [4]. In the particular case of Romania 80% of buildings need to be renovated from thermal considerations [5].

Important energy savings can be realised through the simple retrofit of the envelope and by mounting heating, ventilation and air conditioning system (**HVAC**) systems [6].

In the actual European context regulated by the directive on the energy performance of buildings [1], the near zero energy buildings (**nZEB**) concept should be considered for buildings retrofit. There most important strategies for the retrofit of existing buildings to **nZEB** are to reduce the energy loads and to implement technologies based on renewable energy sources [6-8].

In Romania the energy requirements set for new buildings of **nZEB** level are of (93–217) kWh/m²/year for residential buildings and of (50–192) kWh/m²/year for non-residential buildings [8].

Hospitals represent 7% of the non-residential buildings in Europe [8]. Different strategies for the thermal retrofit of a hospital building envelope are analysed in [9]. Different technologies were considered in hospitals retrofit: substation climatic 3-way valve, radiators thermostatic valves and air handling unit (**AHU**) with programmable regulation [10], or demand control ventilation (**DCV**) of the existing heating, ventilation and air conditioning (**HVAC**) units [11] or control systems, cogeneration, heating regulation, new lighting systems and solar panels [12]. Renewable energies considered in the hospitals retrofit are solar panels [12] or a combination of fuel cells, solar thermal system and solar photovoltaic system [13]. A trigeneration system is considered in [14].

The goal of the study is to investigate the potential of different energy efficient technology to be integrated in a retrofit plan of an existing hospital building in Romania to reduce energy consumption. The considered retrofit technologies are a new condensing heating boiler, new radiators, a distributed **HVAC** system based on several units that integrates air to air heat recovery (**AAHR**) and air to air heat pumps (**AAHP**), a solar thermal system and a photovoltaic system. This study continues the concerns regarding to the energy efficiency in buildings at the Technical University of Cluj-Napoca [15-20]).

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2 Material and method

The study refers to a building of the Municipal Hospital of Sighetu Marmaţiei in the northern part of Romania at the coordinates: 47.927957 N, 23.876285 E. The position of the city on the map of Europe is presented in figure 1 and the position of the Municipal Hospital on the map of the city is presented in figure 2.

Fig. 1. Location of the city on the map of Europe (Google Maps).

Fig. 2. Location of the city on the map of Europe (Google Maps).

The building considered for refurbishing is the Maternity located as indicated in figure 3.

Fig. 3. Location of the hospital building considered in the study (Google Maps).

The frontal and the rear view of the building are presented in figure 4 and figure 5.

Fig. 4. Frontal view of the building.

Fig. 5. Rear view of the building.

The built area of the building is of 1236 m^2, the unfolded built area of the building is of 4576 m^2 and the heated area of the building is of 4079 m^2.

The walls are realized of autoclaved cellular concrete bricks with the width of 30 cm.

The existing energetically situation is characterized by the following aspects: the building is not thermally insulated; the windows frames are of polyvinyl chloride (**PVC**) type and the glazing are double; windows frames presents a lot of leakage problems; the thermal agent for heating and for domestic hot water (**DHW**) is prepared in a gas boiler in advanced state of deterioration; the building is connected to the thermal plant through an old thermal network; there is no mechanical ventilation and no air conditioning in the building.

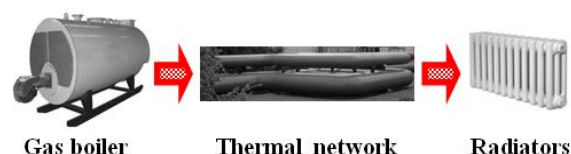The functional scheme of the existent heating system is presented in figure 6.

Fig. 6. Functional scheme of the existent heating system

The efficiency of the existing old boiler is estimated at only 80% due to several technical problems and the efficiency of the existing thermal network from the boiler to the building is estimated at the same value of 80% due to the existing leaks and to the degraded insulation. According to these efficiencies, the global efficiency of the existing heating system is of only 64%.

The proposed retrofit consists in the following updates: insulation of the building with an outer layer of rock wool of 15 cm width (0.036 W/m^{2}K); windows replacement with thicker PVC frames and triple glazing (0.8 W/m^{2}K); replacement of existing old radiators with sanitized radiators in agreement with hospital regulations [21]; dedicated to the building thermal plant of container type, mounted in the immediate vicinity of the building and based on efficient condensing type gas boiler; distributed **HVAC** system each unit including an **AAHR**

(efficiency of 50%) and an **AAHP** (nominal COP=6.3 at interior temperature of 24 °C and exterior temperature of 0 °C); a solar thermal system based on flat thermal collectors (**ST**); and a photovoltaic system (**PV**).

ST is used to assist the **DHW** preparation system and **PV** is used to assist the **HVAC** system. Each of **ST** and **PV** are placed on 50 % of the available flat attic of the building, resulting 119 thermal collectors and 157 PV collectors.

The **HVAC** units are equipped with electric preheaters to provide adequate air temperature if the outside temperature decreases below -10 °C representing the lower operating limit of the equipment and with additional heating coils if the outlet air temperature is not sufficient.

The functional scheme of the proposed ventilation, heating and cooling system is presented in figure 7.
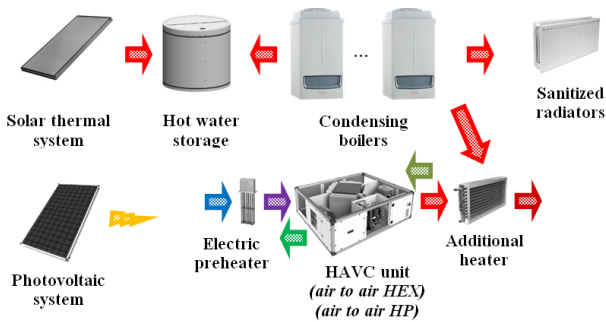

Fig. 7. Functional scheme of the proposed ventilation, heating and cooling system

The global purpose of the proposed improvements is to provide both high comfort and also reduced energy consumption.

To evaluate the thermal behaviour of the building, it was considered simplified as single volume with constant interior temperature of 24 °C.

The thermal loss through walls was determined based on heat transfer through inertial elements equations, according with Romanian regulations.

The thermal load through inertial elements such as walls ($\dot{Q}_{in}$ [W]) was determined as:

$$\dot{Q}_{in} = S \cdot [k \cdot (t_{ac} - t_i) + \eta \cdot \alpha_i \cdot (t_c - t_{ac})] \qquad (1)$$

where:

S [m²] is the lateral area of the inertial element of the building envelope;

k [W/m²K] is the global heat transfer element through the inertial element of the building envelope;

t_{ac} [°C] is the average conventional exterior temperature, influenced by the solar radiation;

t_i [°C] is the interior temperature;

η [-] is the coefficient of temperature oscillation amortization effect;

α_i [W/m²K] is the interior coefficient of convection;

t_c [°C] is the conventional exterior temperature, influenced by the solar radiation.

The thermal load through non-inertial elements such as windows ($\dot{Q}_n$ [W]) was determined as:

$$\dot{Q}_n = \dot{Q}_{td} + \dot{Q}_{sr} \qquad (2)$$

where:

$\dot{Q}_{td}$ [W] is the thermal load due to the temperature difference between outside and inside;

$\dot{Q}_{sr}$ [W] is the thermal load due to the solar radiation.

Heat transfer through the triple windows is considered to calculate the corresponding thermal load.

The thermal load corresponding to the fresh air ($\dot{Q}_{fa}$ [W]) was determined as:

$$\dot{Q}_{fa} = \dot{m} \cdot c \cdot \Delta t \qquad (3)$$

where:

$\dot{m}$ [kg/s] is the fresh air mass flow rate;

c [J/kgK] is the specific heat of the air;

Δt [°C] is the temperature difference between the outside and the inside air, considering the air blowing temperature of 24 °C during winter and 23 °C during summer.

In the heating mode with constant interior temperature of 24 °C, the following variations of the **HVAC** units parameters were determined by modelling the thermal behaviour of the equipment, as function of the exterior temperature (t_e [°C]) in the normal operating range of (-10…15) °C:

Thermal power of the **AAHR** ($\dot{Q}_{AA}$ [kW]):

$$\dot{Q}_{AA} = 19.05 - 0.7939 \cdot t_e \qquad (4)$$

Thermal power of the **AAHP** ($\dot{Q}_{HP}$ [kW]):

$$\dot{Q}_{HP} = 19.477 + 0.252 \cdot t_e \qquad (5)$$

Electrical power absorbed by the **AAHP** compressor (P_{HP} [kW]):

$$P_{HP} = 3.0932 + 0.0384 \cdot t_e \qquad (6)$$

Coefficient of performance of the **AAHP** (COP_{HP} [kW]):

$$COP_{HP} = 6.2979 + 0.0035 \cdot t_e - 7e^{-5} \cdot t_e^2 \qquad (7)$$

Global coefficient of performance of the **HVAC** unit (COP_g [kW]):

$$COP_g = 12.487 - 0.3324 \cdot t_e + 0.0035 \cdot t_e^2 \qquad (8)$$

In the cooling mode with constant interior temperature of 25 °C, the following variations of the HVAC units parameters were determined by modelling the thermal behaviour of the equipment, as function of the exterior temperature (t_e [°C]) in the normal operating range of 28…38) °C:

Thermal power of the **AAHR** ($\dot{Q}_{AA}$ [kW]):

$$\dot{Q}_{AA} = 0.7938 \cdot t_e - 19.847 \qquad (9)$$

Thermal power of the **AAHP** ($\dot{Q}_{HP}$ [kW]):

$$\dot{Q}_{HP} = 18.974 + 0.2969 \cdot t_e \qquad (10)$$

Electrical power absorbed by the **AAHP** compressor (P_{HP} [kW]):

$$P_{HP} = 2.9385 + 0.0488 \cdot t_e \qquad (11)$$

Coefficient of performance of the **AAHP** (COP_{HP} [kW]):

$$COP_{HP} = 5.2898 + 0.0047 \cdot t_e - 0.0001 \cdot t_e^2 \quad (12)$$

Global coefficient of performance of the **HVAC** unit (COP_g [kW]):

$$COP_g = 1.50720 + 0.1574 \cdot t_e \quad (13)$$

The calculation of solar geometry, relative position between the sun and the plane of the collectors (**ST** or **PV**), solar thermal energy production and solar electric production, were realised based on previously successful used algorithms [22-24], [17], [25].

The meteorological parameters representing input data for the calculations were taken from the available typical meteorological year (**TMY**) for the climatic zone of Sighetu Marmaţiei. The solar radiation and the ambient temperature are presented in figure 8.

Fig. 8. Global radiation on horizontal plane and ambient temperature for the climatic zone of Sighetu Marmaţiei

The components of the thermal load are the following: for **DHW** (H), through walls (W); through glazing (G) and due to the fresh air (F).

In the existing situation, the energy is distributed in the building only as hot water (HW) and through the existing radiators (ER) while in the proposed situation it has the following components: hot water (HW) through proposed sanitary radiators (SR); through the electric preheaters of the **HVAC** units (EP); through the air to air heat exchangers (AA); through the air to air heat pumps (HP) and through the supplementary heating coils of the **HVAC** units (SH).

In the existing situation the primary energy is represented only by the consumption of natural gas in the existing boiler (NGB) while in the proposed situation it has the following components: consumption of natural gas for hot water (NGH); consumption of natural gas in condensing boiler (NGB); consumption of natural gas in the supplementary coils of the **HVAC** units (NGS); consumption of electricity in the preheaters (EEP); consumption of electricity in the compressors of the heat pumps (EHP) and consumption of electricity in the fans (CEF).

2 Results and discussions

The components of the thermal load are determined hourly for both existent and proposed situation as presented for the proposed situation in figure 9.
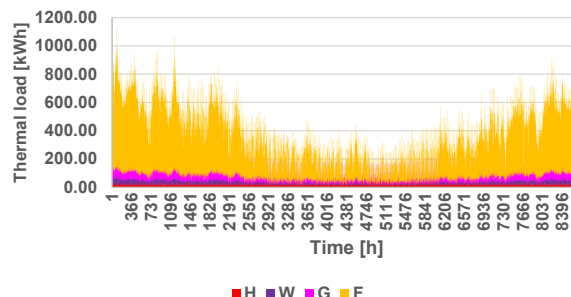

Fig. 9. The hourly components of the thermal load for the proposed situation

It can be observed that the thermal load due to the fresh air is the most important one in the context of reduced loads due to the proposed retrofit measures.

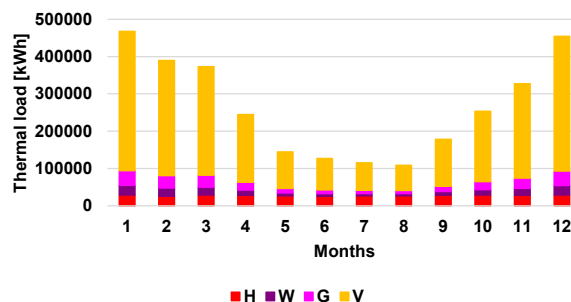The monthly components of the thermal load are presented in figure 10.

Fig. 10. The monthly components of the thermal load for the proposed situation

It can be observed the difference between the thermal loads in summer and in winter. The total thermal load varies between 109000 kWh in August and 467500 kWh in January.

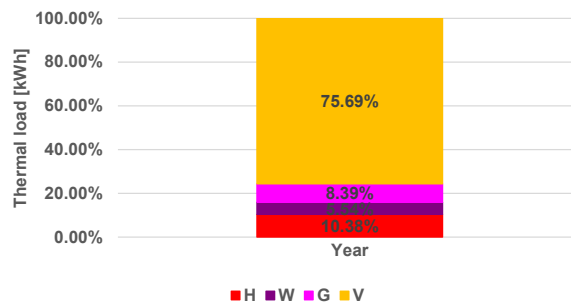The shares of the yearly thermal loads are presented in figure 11.

Fig. 11. The yearly shares of the thermal load for the proposed situation

The **DHW** share of 10.38 % is higher than the ones corresponding to the walls and to the windows, but the higher share of 75.69 % corresponds to the fresh air.

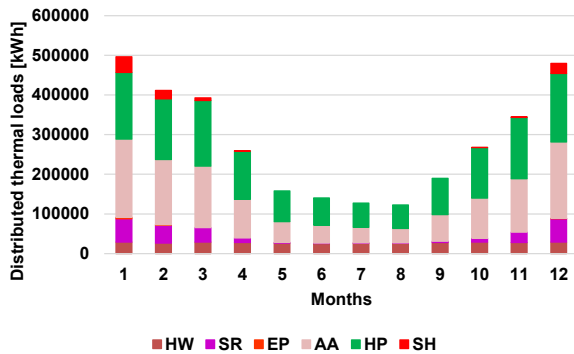The monthly components of the distributed thermal loads are presented in figure 12.

Fig. 12. The monthly components of the distributed thermal loads for the proposed situation

The electric preheaters are needed only in the months of December, January and February, while the supplementary heating coils are used in October – December and January – April.

The shares of the yearly distributed thermal loads are presented in figure 13.

Fig. 13. The yearly shares of the distributed thermal loads for the proposed situation

The most important shares of the distributed thermal loads correspond to the air to air heat recovery (AA: 37.86 %) and to the air to air heat pumps (HP: 41.96 %).

The annual operating time for each type of heat distribution is presenting in figure 14.
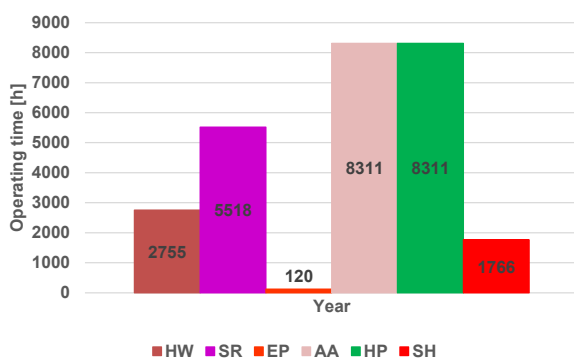

Fig. 14. The annual operating time for each type of heat distribution for the proposed situation

The air to air heat recovery (AA) operates the same number of hours with the air to air heat pumps (HP), because AA can always provide only 50 % of the needed thermal power and HP is needed for the rest of 50 %. Supplementary over the 8311 hours of yearly operation in heating mode, the **HVAC** systems that includes both AA and HP equipment, operates also 319 hours in cooling mode and 130 hours in "only ventilation" mode (no heating or cooling is needed because the outside temperature differs with less than ±3 °C than the inside temperature). The shortest operation period corresponds to the electric preheater (EP 120 hours).

The consumed energy needed to provide the thermal load through the heat distribution equipment is of both thermal type (represented by consumption of natural gas) and of electric type (represented by electricity to power the fans, the heat pump compressors and the electric preheaters).

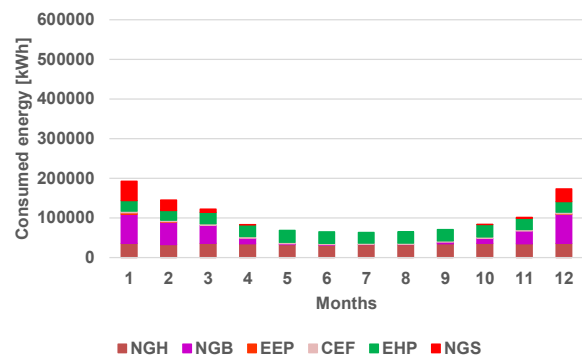The monthly components of the consumed energy are presented in figure 15.

Fig. 15. The monthly components of the consumed energy for the proposed situation

The absolute values of the consumed energy are significant lower than the values of the thermal load and of the distributed thermal load because an important share of thermal load is provided by heat recovery in the air to air heat exchangers and the distributed thermal load of the heat pump is provided with low energy consumption by the air to air heat pump that operates with very high coefficient of performance in the range of (6.25 – 6.34) for all the operating domain of ambient temperatures (-10 – +15) °C.

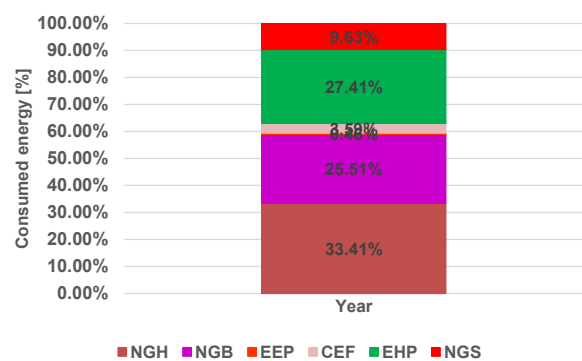The shares of the yearly energy consumptions are presented in figure 16.

Fig. 16. The yearly shares of the components of energy consumptions for the proposed situation

The natural gas consumption represents 68.55 % (NGH + NGB + NGS) and the consumption of electricity represents 31.45 % (EEP + CEF + EHP).

The global solar radiation in the plane of the tilted solar collectors, both thermal and photovoltaic, is presented in figure 17.
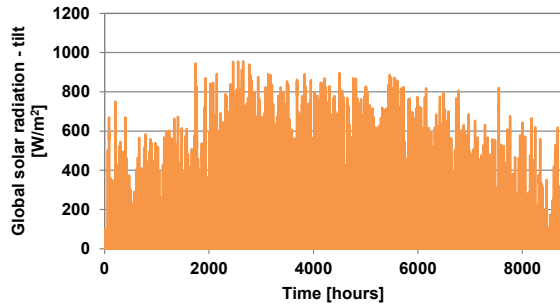

Fig. 17. The global solar radiation in the tilted plane

Due to the tilted plan of the collectors the available solar radiation in this plan is higher than the global radiation in the horizontal plan, available in the **TMY**.

The thermal efficiency of the thermal collectors is presented in figure 18.

The higher value of the considered thermal collectors is of almost 60%.

The solar thermal production of the ST system is presentd in figure 19.

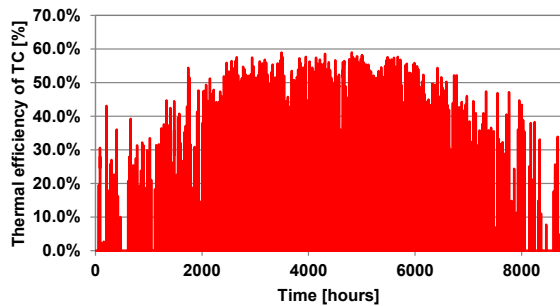The solar thermal production is contributing to the **DHW** preparation as it is presented in figure 20.

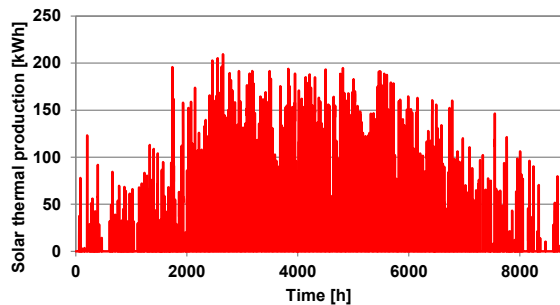**Fig. 18.** The thermal efficiency of the thermal collectors

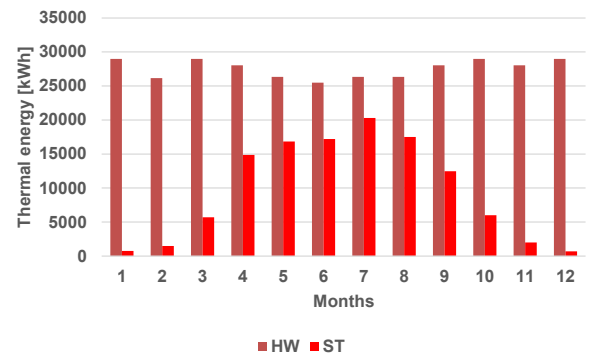**Fig. 19.** The solar thermal production of the thermal collectors

Fig. 20. DHW thermal load and solar thermal production

The total yearly amount of solar thermal production represents about 35 % of the total **DHW** thermal load.

The global efficiency of PV to provide electricity of alternative current (**AC**) type is presented in figure 21.
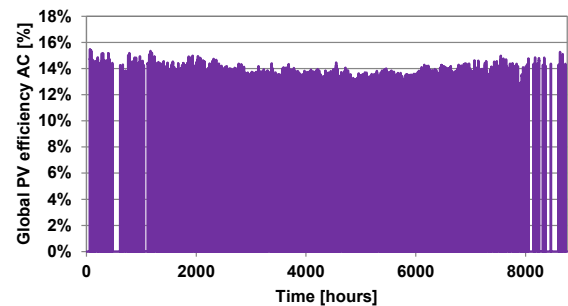

Fig. 21. The global efficiency of the **PV**

The solar electric production is presented in figure 22.
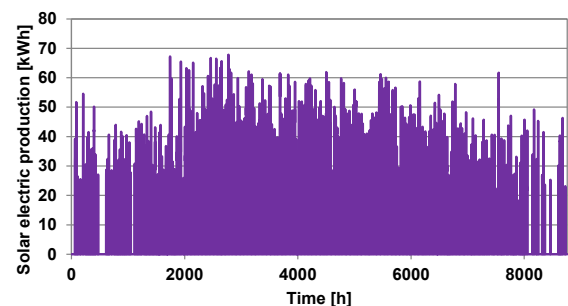

Fig. 22. The solar electric production

The maximum hourly provided electric energy is of 67.8 kWh and is distributed for the building.

The comparative electric consumption of the fans of the distributed HVAC system and the provided electricity is presented in figure 23.
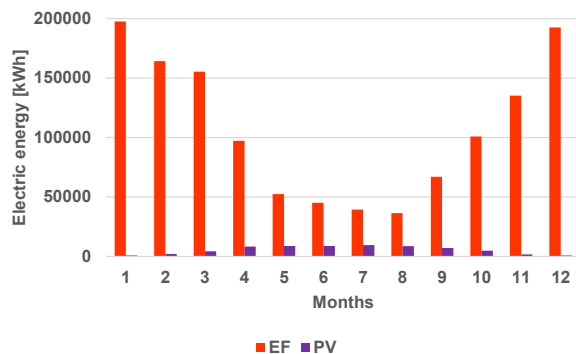

Fig. 23. Fans consumption and solar electric production

The total yearly amount of solar electrical production represents about 5.2 % of the total electrical consumption of the fans.

The thermal and electrical solar energy represents 14.9 % of the global energy consumption of the building.

The comparative global energy consumption in the existent and proposed situations is presented in figure 24.
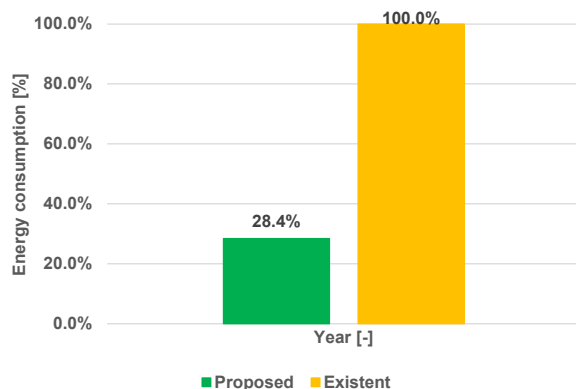

Fig. 24. Existent and proposed global energy consumptions

The global reduction of energy consumption through the proposed retrofit improvements can be obtained in the context of the increased energy demand with 31.1%, determined by the distributed HVAC system capable of ventilation with sufficient air flow rate and also capable of cooling the fresh air.

3 Conclusions

The study proposed a set of retrofit measures to reduce the energy consumption of an existent hospital building.

The proposed retrofit includes classic improvements of building envelope like thermal insulation and windows replacement, but also very efficient improvements from the energy consumption point of view like a distributed HVAC system and two solar systems, one thermal and one photovoltaic.

Despite the increasing of the global energy demand of 31.1% comparing to the existing situation, determined by the need to heat and cool the ventilated fresh air, the global energy consumption of the building can be reduced by 71.6 %.

The specific energy consumption of the building, reported to the heated area, was reduced from 901.3 $kWh/m^2/year$ in the existing situation, at 301.0 $kWh/m^2/year$ in the proposed situation not taking into account the solar technologies and at 256.3 $kWh/m^2/year$ in the proposed situation including the solar technologies. The last value is close to the upper limit of nZEB non-residential building in Romania (192 $kWh/m^2/year$). Other solutions to reach the status of nZEB can be the use of a ground air heat exchanger of to use the facades of the building to produce solar heat or solar electricity.

The solar technologies alone, provides 14.9 % of the global energy consumption of the building.

Under these circumstances the study proved the potential of distributed **HVAC** system including air to air heat recovery and air to air heat pump, together with **ST** and **PV**, to increase the energy efficiency of an existent hospital building.

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