Analysis of heat gain decrease achieved by ventilation heat recovery in solar cooling building: case study

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Abstract

The aim of this work is to present the improvement of the operation of the solar building cooling system by using ventilation with heat recovery, and thus reducing required cooling power of air conditioning system. The heat recovery system is divided into zones with counter-flow plate heat exchangers (laboratories, required system tightness, no air recirculation) and counter-flow spiral heat exchangers (office and conference rooms). The analysis was carried out for the real indoor conditions, measured on the site and climatic data of a typical meteorological year.

Keywords: heat recovery, solar cooling, TMY

1. Introduction

Energy has become a key issue in our daily lives. Civilization development and population growth mean that the demand for energy is constantly growing. Its production, processing and consumption can have a devastating impact on the environment. Awareness of this threat has created in developed countries the pursuit of energy conservation and abandoning the use of its forms that lead to the greatest pollution of the environment. The use of renewable energy sources is becoming more and more popular all over the world. In 2017, renewable energy accounted for around 18% of total final energy consumption (REN21, 2019). Progress in renewable energies remains concentrated in the power sector, while much less growth has been recorded in heating and cooling. Despite this, solar building cooling systems are being developed and are getting the attention of both researchers and potential users (Ge et al., 2018; Lazzarin and Noro, 2018).

Modern architecture is characterized by high tightness of external partitions and windows. Providing the necessary amount of fresh air to users of rooms in insulated and almost hermetic buildings is a basic condition for maintaining their well-being and health. The purpose of the work undertaken and described in the article is to determine the possibility of reducing the building's cooling demand by applying heat recovery associated with a ventilation system. This analysis focuses on the integration of solar-assisted cooling system with heat recovery in a Solar Research Energy Center (CIESOL) located in South Spain. A detailed description of the building and solar-assisted air-conditioning system is provided in (Rosiek, 2018; Rosiek and Batlles, 2013, 2012). The solar-assisted air-conditioning system presently installed in CIESOL consists of a flat-plate collector array facing due south and titled at an angle of 30° to the horizontal plane, with a total surface area of 160 m^2, a hot water driven single-effect LiBr-H_2O absorption chiller, a cooling tower, a shallow geothermal heat dissipation system, two hot water storage tanks, each with a 5,000 L capacity, two chilled water storage tanks of 2,000 L and 300 L, respectively; along with an auxiliary gas boiler, a plate heat exchanger and peripheral equipment such as valves and pumps. Due to Almeria’s particular meteorological conditions, the solar-assisted air-conditioning system makes a huge contribution to CIESOL’s cooling for 5 months (from May to September) of the year. In summer mode, the building's water temperature supply is between 7°C and 12°C, to cover CIESOL's cooling load. Figure 1 illustrates the system’s general layout with each component’s minimum and maximum temperature range, operating in the summer. To produce chilled water in the desired temperature range, it is necessary to supply the absorption chiller with hot water between 70°C and 95°C; this is mainly obtained from the solar collector array. When the solar hot water temperature is too low, the warm water from the hot water storage tanks can be used; otherwise the auxiliary heater is used to drive the absorption chiller. A conventional HVAC system was also installed to provide thermal comfort conditions in case of solar-assisted air-conditioning system failure.

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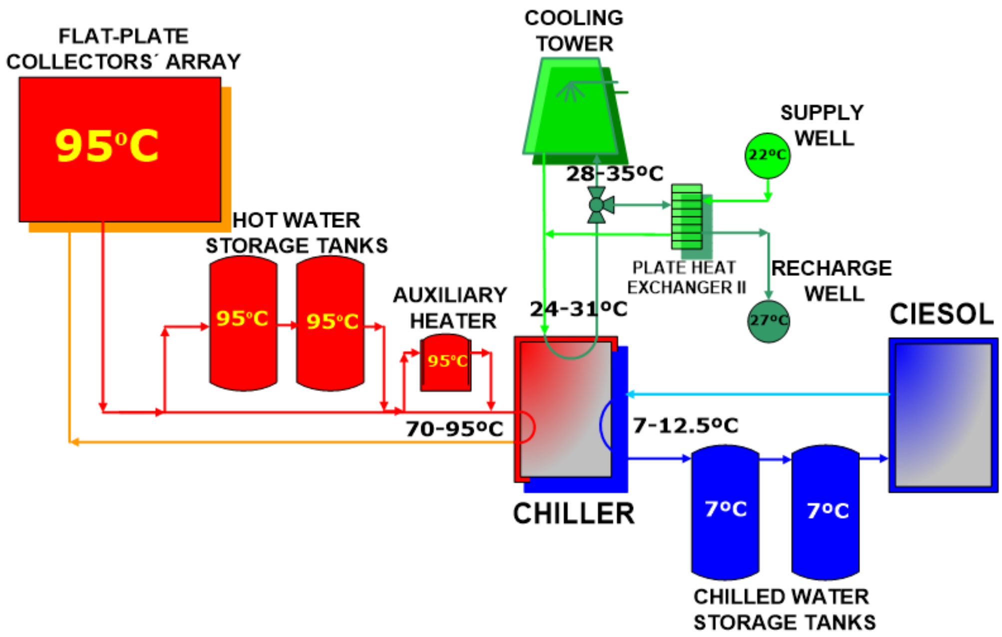

Fig. 1. The general layout of the solar-assisted air-conditioning system installed at CIESOL working in the summer season

Based on the previously performed thermal balance of the building (Gil et al., 2019), it was shown that the maximum annual demand for cooling capacity is 61.4 kW. Analyzes have also shown that as much as 35% of the heat gains come from the treatment of ventilation air necessary for people staying inside the building. The summary of the balance and the key conclusion was that the dominating problem of the CIESOL building is the lack of mechanical ventilation with heat recovery, which directly translates into a greater demand for cooling power, resulting in energy wastage and inefficiencies. Very few design strategies can be adopted in an overcrowded building, since the solar-assisted air-conditioning system is working to handle the total expected cooling demand. It was demonstrated that over 20 kW of cooling power is lost through ventilation with the maximum thermal load of the building (Figure 2). Therefore, the use of the simplest recuperative heat exchanger would result in a significant decrease in the demand for cooling power during the summer. Consequently, it was decided to analyze the potential benefits resulting from the use of heat recovery combined with mechanical ventilation of the building.

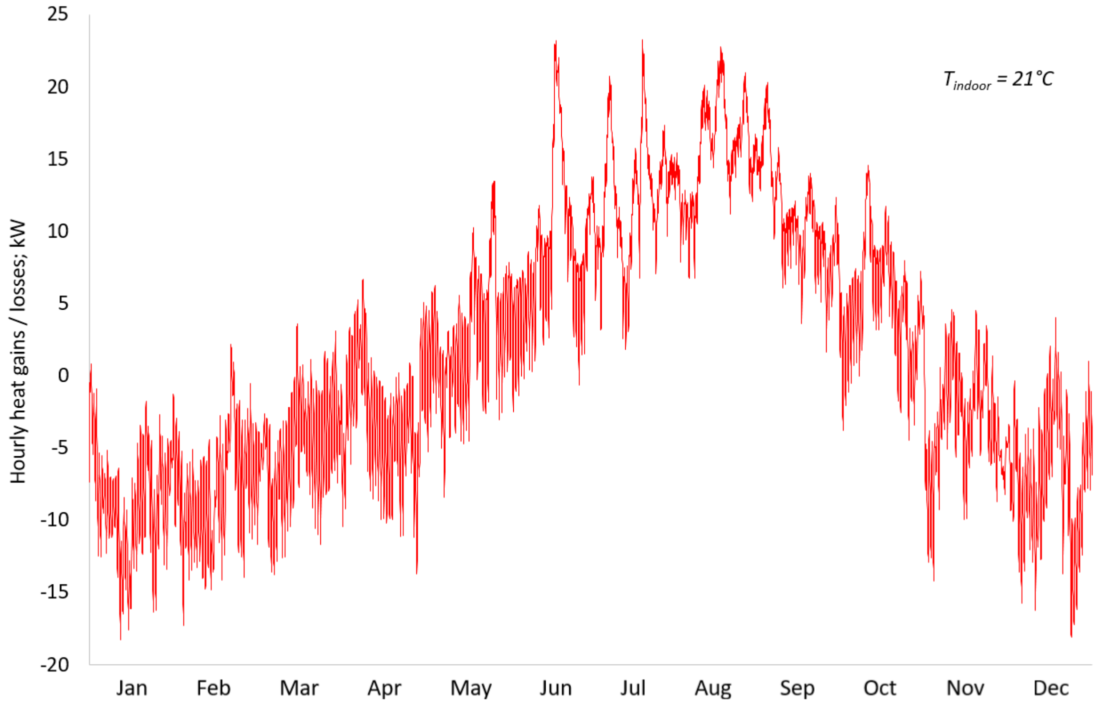

Fig. 2: Heat gains/losses by the ventilation in CIESOL building (calculated using TMY data)

The efficiency of heat recovery depends on the size of the exchanger's active surface and the parameters of the supply and exhaust air streams. During the actual operation of the device, these parameters are variable, so it is reasonable to examine the consequences of their changes. In addition, according to EN 308:1997 *Heat exchangers. Test procedures for establishing the performance of air to air and flue gases heat recovery devices*, heat transfer efficiency should be determined under nominal operating conditions, which are as follows:

- Dry bulb temperature of the extract and intake air are 25℃ and 5℃, respectively;
- Wet bulb temperature of the extract air is below 14℃, which corresponds to approximately 30% of relative humidity; when measuring this temperature the air velocity should be in the range of 3.5 ÷ 10 m/s, (the recommended value is 5 m/s).
- Air density in the range: 1.16 ÷ 1.24 kg/m^3.

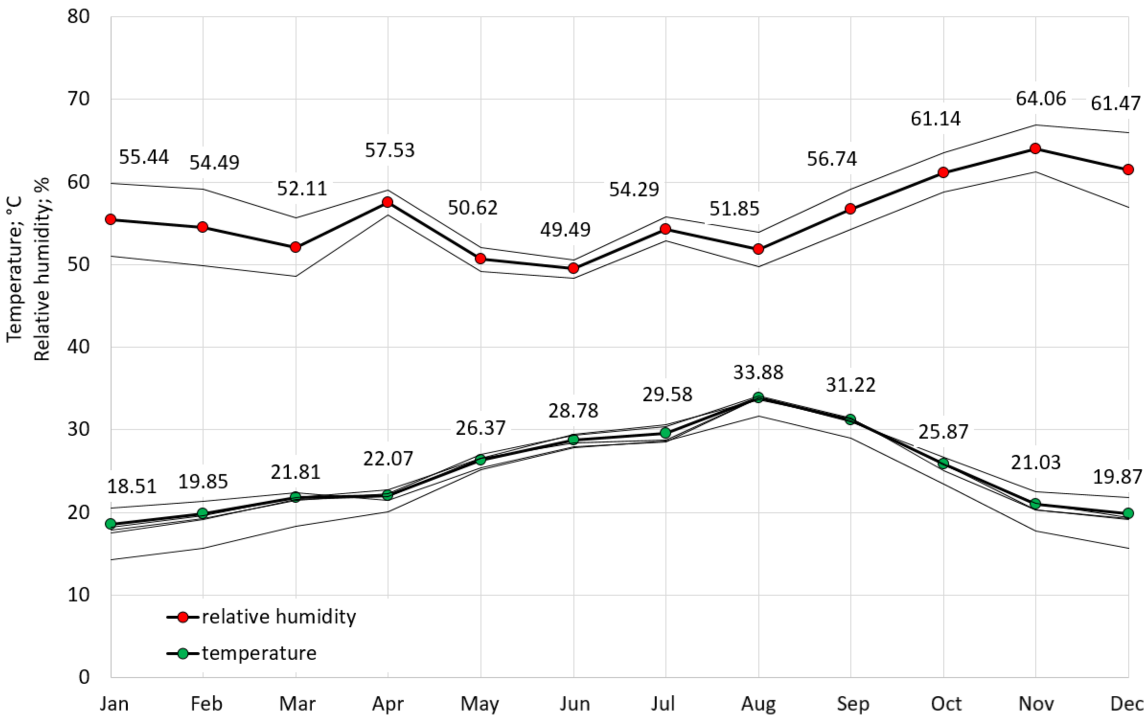

Fig. 3: Real value of temperature (monthly average) and relative humidity of the ambient air in Almeria, Spain

Due to the location of the analyzed building in the Hot-summer Mediterranean climatic zone (Csa) the operating conditions of the system differ significantly from the normative values - the course of annual temperatures and relative humidity are shown in Figure 3. The data presented are actual values (monthly average) collected from the annual observation carried out with sensors mounted on the site. In addition, the efficiency of heat recovery devices is tested in conditions corresponding to winter (heat recovery), while the purpose of their application in this work is cold recovery. Therefore, the heat transfer efficiency specified in the technical documentation of the devices was not adopted for the calculation, but it was determined experimentally for the corresponding climatic conditions.

2. Research methodology and description

The research was based on two types of heat exchangers (Figure 4). For office and conference rooms, the use of a spiral counter-flow heat exchanger, ensuring low energy consumption with high heat exchange efficiency, was assumed. This type of heat exchanger is based on the patent (Walczak and Mościcki, 2013), and its construction is distinguished by the use of partial ribbing of the space between the membranes separating the processed air streams. The exchanger has the shape of a cylinder with a circular cross-section and consists of two flat plates, rolled around one axis, forming a double helix and contains two slotted channels (with sealing elements) for air flow. These panels are made of aluminum sheet with a thickness of 0.085 mm and tightly separate the exhaust and supply air ducts. The unit is enclosed in a polyvinyl chloride housing and is insulated with a 30 mm thick polyurethane foam and additionally padded from the inside with a 10 mm rubber mat, designed to thermally insulate and dampen the noise that constantly accompanies the device.

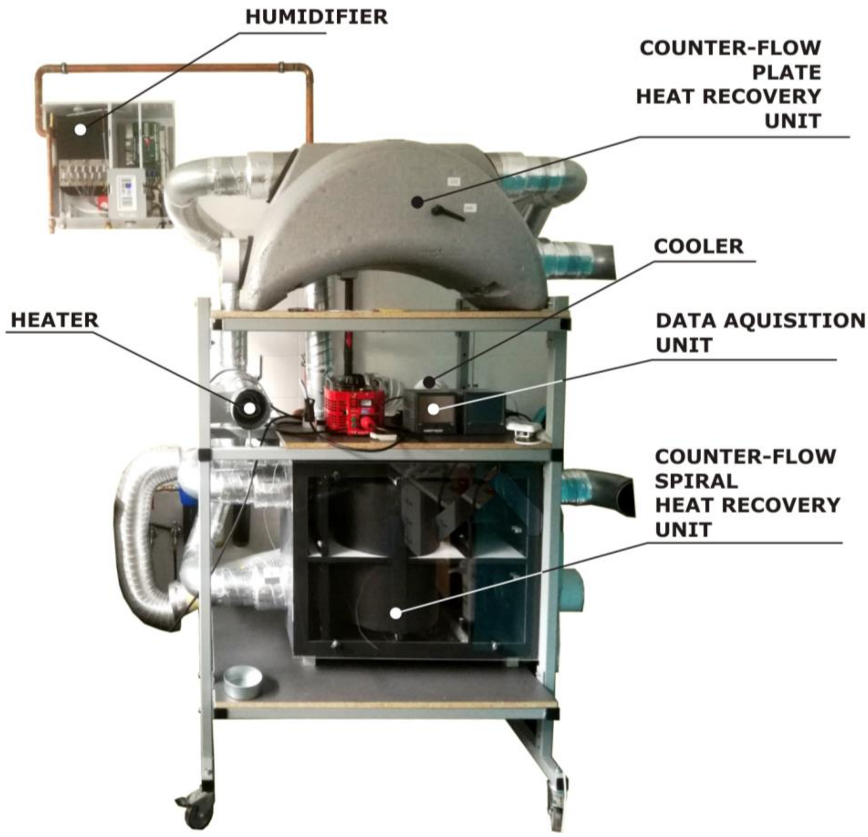

Fig. 4: View of the test bench with heat recovery units

In the laboratories, counter-flow plate heat exchangers, which is based on the opposite principle of the cross-flow heat exchanger, were used to ensure complete separation (tightness) of air streams. However, construction of counter-flow plate heat exchangers features a parallel section, giving it a larger contact surface than the cross-flow plate heat exchanger, which results in higher efficiency of heat recovery. The supply air and extract air flow in completely separate ducts, meaning that any pollution in the extract air cannot return to the supply air. Both recovery units used in the tests have similar overall dimensions and are dedicated to ensuring air exchange in small buildings. The air flow during the tests for both devices was 200 m^3/h. Manufacturers of both devices state that temperature transfer efficiency can reach up to 95%. At first glance, the devices are therefore identical, unfortunately none of the manufacturers provides information about area of heat exchange. The variability of operating conditions, as well as the lack of accurate technical data are the premises for experimental studies of the heat recovery efficiency of both devices.

In addition to the heat exchangers mentioned above, the test bench included an air heater and a humidifier mounted on the intake air duct, which served to establish air conditions in accordance with the external air data. Similarly, an air cooler was installed on the extract air duct to simulate air conditions in the room corresponding to spring and autumn periods. Temperature sensors (K-type thermocouples, class 1) and humidity sensors (P18L transducers) have been mounted on each duct. Signals from these sensors were collected by the data acquisition system. Due to the inertia of the humidity transducers, the data was recorded with a step of 10 seconds. On the basis of measurements taken, the temperature efficiency of heat recovery was determined. The layout of the test bench with the measuring points is shown in Figure 5. The test results allowed to determine the real values of heat recovery efficiency, determined by the following formula:

$$\eta_T = \frac{T_{intake} - T_{supply}}{T_{intake} - T_{extract}} \qquad \text{(Eq. 1)}$$

Using the law of propagation of uncertainties as a geometric sum of partial differentials (Eq. 2), standard uncertainty of heat recovery efficiency was also determined; values are given in Table 1.

$$u_c(y) = \sqrt{\sum_{j=1}^{k} \left[\frac{\partial f}{\partial X_j}(\overline{X_1}, \overline{X_2}, \dots, \overline{X_k})\right]^2 u^2(\overline{X_j})} \qquad \text{(Eq. 2)}$$

Heat recovery efficiency values were determined for the period from May to September, which corresponds to the chiller's operating period for the needs of building air conditioning. Determination of the average monthly psychrometric properties of the ambient air and indoor conditions based on real measurement data allowed to state that the dew point of the intake air is always below the temperature of the air removed from the rooms.

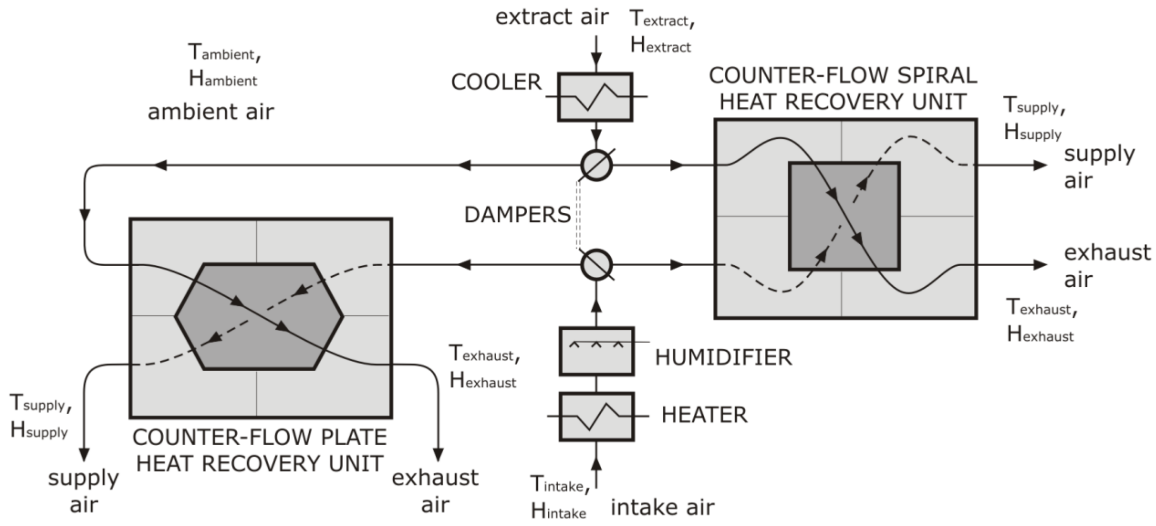

Fig. 5: Scheme of the test bench with heat recovery units

Thus, moisture recovery is not possible and the system can only recover sensible heat contained in the extract air stream. Lack of moisture condensation allows to apply Eq. (3) to determine the amount of recuperated heat and thus to determine the reduction of cooling power demand.

$$Q_r = \dot{m} \cdot c_p \cdot (T_{intake} - T_{supply}) \qquad \text{(Eq. 3)}$$

3. Results

In order to determine the reduction in the demand for cooling power, resulting from the use of heat recovery in a building, it is necessary to determine the effectiveness of the system in particular seasons. For each of the analyzed months, the efficiency of heat recovery was determined by simulating outdoor and indoor air conditions, based on data obtained from sensors mounted on the building (see Figure 3). Due to similar conditions in some months, 3 points were finally simulated, corresponding to the following months: May, June/July, and August/September. An example diagram of the simulation of air conditions on the heat exchanger for August and September is presented in Figure 6. The graph shows a sample (five minutes) of tests for the set temperature and humidity as well as adequate heat recovery efficiency. The average efficiency value from each cycle was taken for further calculations. The comparison of heat recovery efficiency in individual months for both heat exchangers used in the tests is presented in Table 1.

Tab. 1: Heat recovery efficiency

Month	Heat recovery efficiency - counter-flow spiral heat exchanger	Heat recovery efficiency - counter-flow plate heat exchanger
May	0.824 ± 0.094	0.424 ± 0.117
June	0.853 ± 0.062	0.490 ± 0.134
July	0.853 ± 0.062	0.490 ± 0.134
August	0.604 ± 0.078	0.320 ± 0.108
September	0.604 ± 0.078	0.320 ± 0.108

Experimental studies have shown that counter-flow spiral heat exchanger achieves almost twice as high heat recovery efficiency as counter-flow plate heat exchanger. This is due to the longer contact of the air stream with the heat exchange surface, obtained by additional winding the exchanger into a spiral. For both exchangers, the highest efficiency were obtained for June/July, 0.853 and 0.490. Moreover, the efficiency of the counter-flow plate heat exchanger in the whole range of tests did not exceed 0.5. It should be noted that the uncertainties in determining the heat recovery efficiency for the counter-flow plate heat exchanger are definitely higher due to the worse heat exchange and larger temperature differences. The obtained results meant that heat recovery from ventilation air using plate heat exchangers is considered only in chemical laboratories located on the ground floor of the building, in order to avoid potential pollution of the supply air. Higher efficiency spiral heat exchangers are assumed for the remaining rooms.

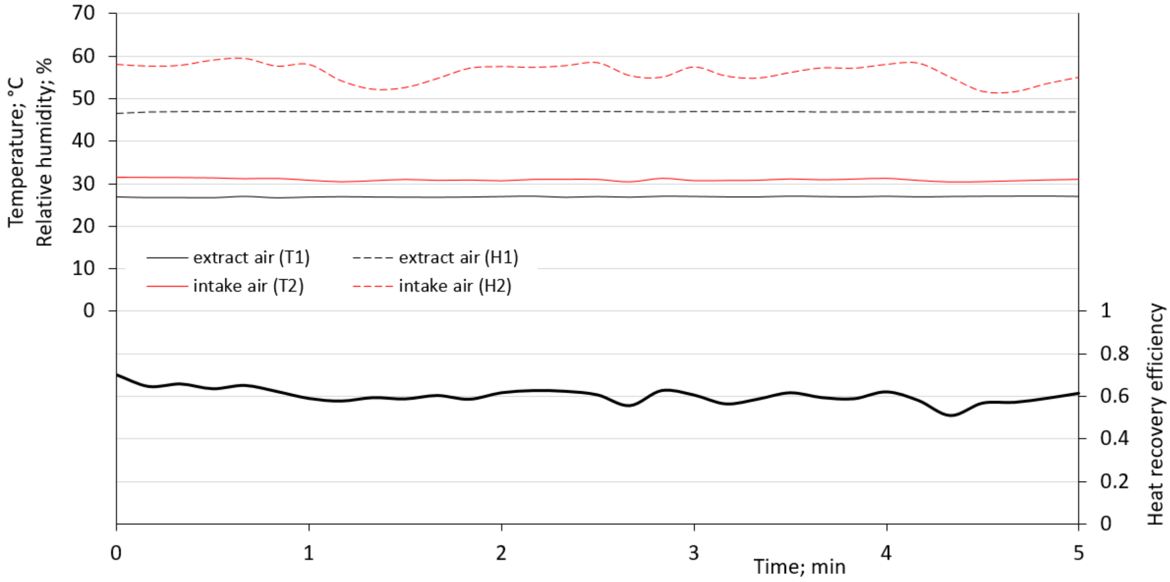

Fig. 6: Simulation of intake and extract air parameters and achieved heat recovery efficiency

In order to compare energy consumption with and without a heat recovery system, the previously made thermal balance of the building (Gil et al., 2019) was used. To calculate the necessary amount of fresh air and ventilation rate in each room, the parameters specified in (Rosiek and Batlles, 2013) were adopted.

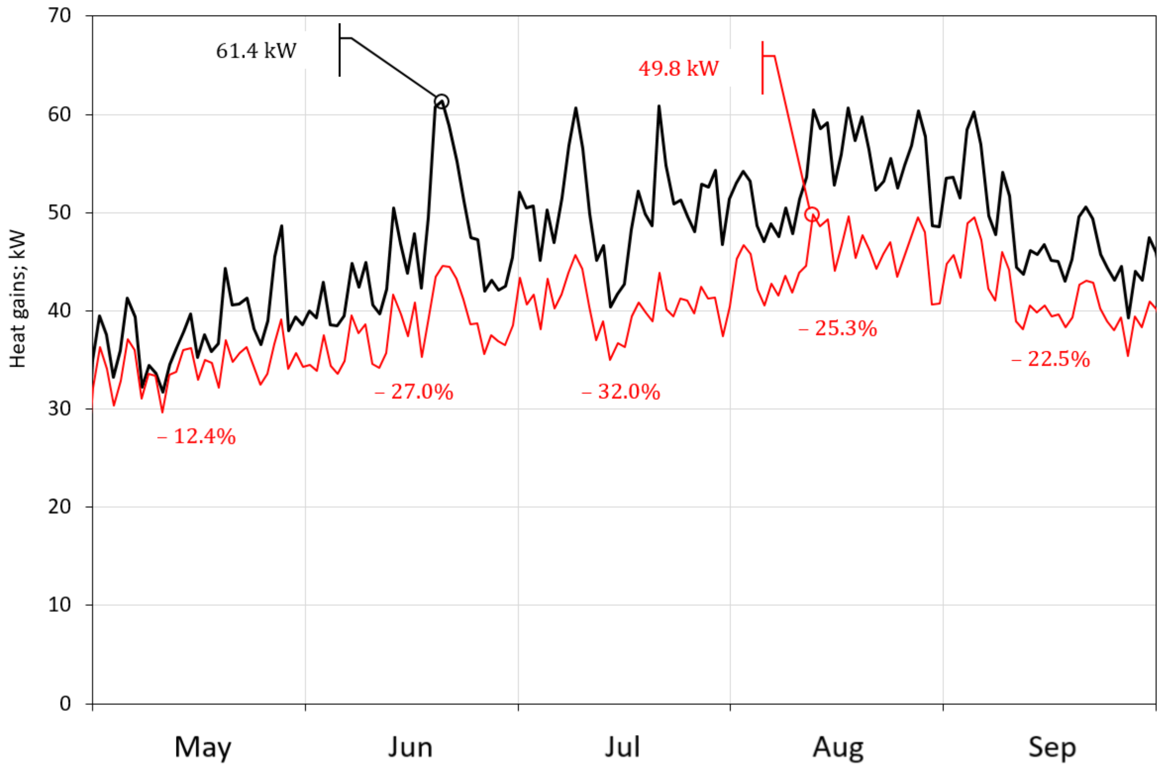

Fig. 7: Building heat gains (cooling demand); variant without heat recovery (black line) and with heat recovery (red line)

The ventilation air flow was determined both by the number of air changes and the amount of fresh air per one person. According to the Standard (*PN-B-03430:1983/Az3:2000*, 2000), the value of 30 m^3 per hour per person was assumed. Energy reduction was calculated by applying heat recovery efficiency values for individual months to the building's thermal balance model. The analysis showed that heat recovery system can significantly reduce the need for cooling and heating. This translates into an increase in the efficiency of the solar air conditioning system by extending the system's operating time. The use of heat recovery from the building's ventilation system significantly reduces the maximum annual cooling demand (Figure 7). It can be seen that the maximum cooling consumption shifts to August, which is an additional advantage, due to the specificity of the building's work, holiday period, partial shutdown of the facility and incomplete demand for air conditioning. The maximum daily reduction of cooling capacity is 27.4% (June 20, reduction from 61.4 kW to 44.6 kW). The greatest benefits from the use of heat recovery system are in July, where it allows reduce the cooling demand by almost 1/3. Total consumption of cold in the reporting period decreased by about 24,000 kWh, 30% of which is in July. The demonstrated reduction in the use of cooling capacity, when provided by a conventional HVAC system, translates into a reduction in electricity consumption of around 9,000 kWh_e.

4. Impact of heat recovery on the operation of the short-term energy storage system

Both the solar-assisted air-conditioning system and the conventional HVAC system installed in the CIESOL building have the option of sending chilled water directly to receivers or feeding a short-term energy storage system (STES). This system consists of two tanks of 2,000 liters each, filled with phase change materials (PCM) S10 and S46 in 88/12 proportions. In addition, there is approximately 1,250 liters of water in each tank. The STES can accumulate an excess of cold during the operation of the solar-assisted AC system and to use it to cover the heat gain during the absence of sufficient solar radiation to drive absorption chiller or at night.

Figure 8 shows how the discharge time of both tanks changes with the thermal load of the building. In the absence of heat recovery and a maximum heat load of 61.4 kW assumed, the tanks are able to provide thermal comfort for 18 minutes, providing chilled water for fan coil units at a temperature between 13 and 14.5°C. The use of cold recovery from the building's ventilation air stream extends this time to 26 minutes and 20 seconds, which is an increase of 46%. The benefit of using heat recovery system is even more pronounced during spring and autumn, as well as at night, when the accumulated energy is able to cover the heat demand of the building for several hours.

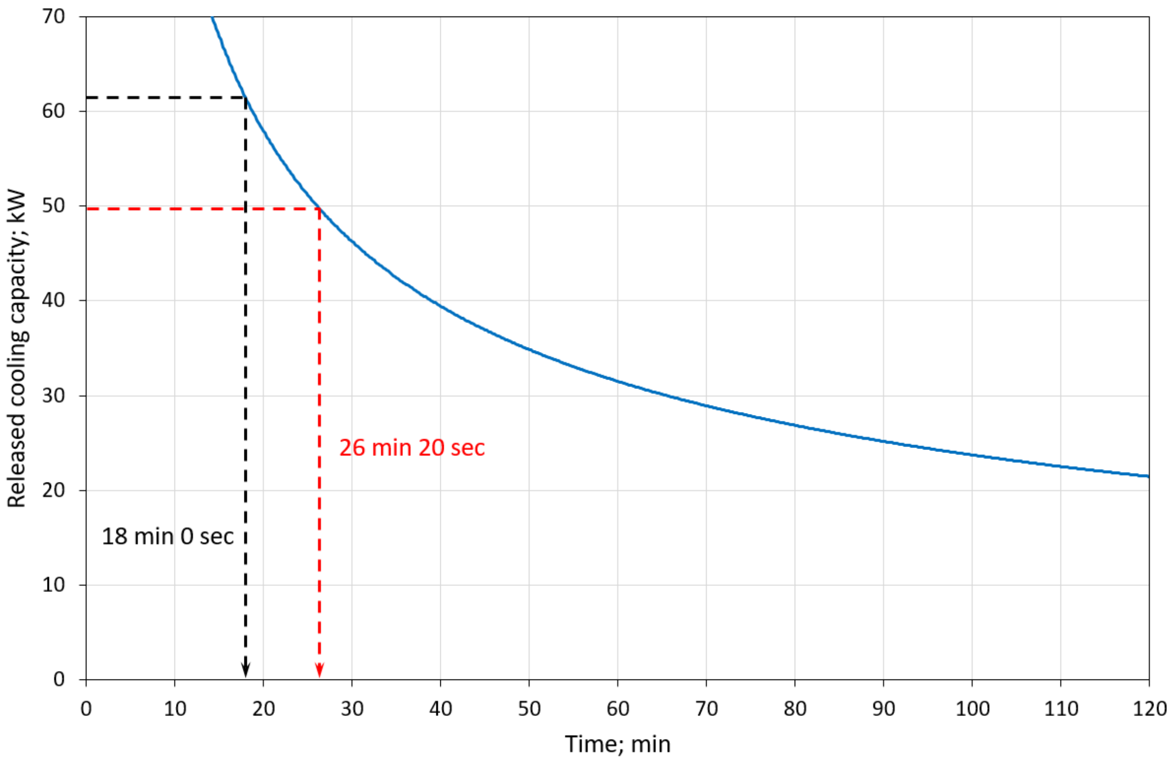

Fig. 8: Cooling capacity released from STES system filled with PCM

5. Conclusions

The paper presents the impact of ventilation heat recovery system on the reduction of CIESOL building heat gains and the operation of the STES system. The tests were carried out using two types of heat exchangers for which climatic conditions in Almeria were simulated. Using the experimentally determined heat recovery efficiency, the potential reduction in building cooling demand from May to September was calculated. The maximum demand for cooling capacity was reduced by 27.4%, from 61.4 kW to 49.8 kW. The results of the simulation showed that during the period of the strongest thermal load of the building, the use of heat recovery from the ventilation system can result in a significant reduction in cooling demand by 27.0%, 32.0% and 25.3%, respectively for June, July and August. This allows to significantly extend the discharge time of storage tanks filled with phase-change materials and water.

6. Acknowledgments

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