Influence of the hydronic loop configuration on the energy performance of a CO_2 heat pump for domestic hot water production in a multi-family building

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Abstract

In this work, a numerical analysis of the influence of the hydronic loop on the energy performance of a CO_2 heat pump dedicated to DHW production for an apartment block located in Bologna (Italy) is presented. The energy model of the whole heating system, implemented in TRNSYS17, has been validated by means of a monitoring campaign performed during the winter season of 2017-2018. The experimental results highlighted a poor and unexpected energy performance of the heat pump. The comparison between experimental and numerical results showed a significant penalty of the heat pump performance due to an erroneous use of the hot stratified thermal storage system. Outcomes of this paper confirm that CO_2 heat pumps are very sensible to the temperature of the fresh water at the inlet of the gas cooler. This value can be strongly reduced thanks to the presence of the stratified tank in the hydronic loop.

Key Innovations

- Numerical simulation of a CO_2 heat pump dedicated to DHW production for a large multi-flat building.
- Calibration of the numerical model using the measured water temperature at the heat pump inlet.
- Analysis of the influence of storage thermal stratification on the heat pump energy efficiency.

Practical Implications

Energy performance of CO_2 heat pumps dedicated to DHW production is strongly influenced by the hydraulic configuration of the system and the heat pump operating mode. Thermal stratification within the storage tank must be guaranteed during the whole season in order to maintain low water temperature values at the inlet of the heat pump. Contemporaneity factors for DHW in multi-family buildings have to be considered for an optimal design of the thermal storage system.

Introduction

In Europe, the building sector is currently responsible for about 40% of the overall energy demand and nearly 36% for greenhouse gas emissions (Gulotta et al., 2021). For this reason, it is evident how the refurbishment of the existing building stock and the development of sustainable and efficient constructions have a great importance for the reduction of environmental issues. Within this frame, it is worth to focus efforts especially on residential buildings since they account for approximately 25% of the European final energy demand (Karytsas et al., 2019).

In recent years, the main actions of policymakers and stakeholders have been focussed on the reduction of buildings energy demand for space heating and cooling, as well as on the enhancement of the performance of HVAC systems. However, little consideration has been given to the reduction of the energy use linked to domestic hot water (DHW) preparation, even though its contribution reaches about 15-20% of the total energy need of residential buildings (Hervas-Blasco et al., 2020). Furthermore, with increasing building thermal insulation, it is expected that the share of DHW on the total heat demand in new residential constructions could be as high as 50% or more (Braas et al., 2020).

Heat pumps can be considered as an efficient alternative to traditional systems to decrease the primary energy inputs for ambient conditioning and DHW production. In residential buildings, air-source heat pumps (ASHPs) represent the most widespread solution according to the huge availability of the heat source, the relatively low installation cost and the considerable energy efficiency. Within this field, carbon dioxide (CO_2 or R744) heat pumps operating with a trans-critical cycle are considered as one of the most efficient option for DHW production. As clearly demonstrated by many Authors (Bruno et al. (2019) and Dilshad et al. (2020) among others), the gas cooling process allows to obtain a large temperature lift in the fresh water stream without the significant reduction of *COP* obtained with conventional heat pumps based on sub-critical cycles.

In order to achieve the best energy performance from a CO_2 heat pump for DHW production, the inlet water temperature of the gas cooler must be kept as low as possible by using a stratified storage tank coupled to the heat pump (Smitt et al., 2021). The motivation is linked to the peculiarity of the trans-critical cycle: the thermodynamic performance is mostly influenced by the water inlet temperature at the gas-cooler side and not by the maximum temperature of hot water (He et al., 2021).

Many Authors have investigated the influence of the storage tank stratification on the energy performance of CO_2 heat pumps but the works of Nawaz et al. (2018) and Tammaro et al. (2016) deserve a mention for novelty and

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accuracy. A calibrated numerical model was used by Nawaz et al. (2018) to run a parametric analysis for a trans-critical CO_2 heat pump. They outlined that the heat pump performance is strongly influenced by the water flow rate, which directly impacts the storage thermal stratification, and suggested to adopt a variable-speed circulating pump. Tammaro et al. (2016) evaluated the optimal volume of the thermal storage coupled to a CO_2 heat pump as a function of the heat pump size and the system control strategy. They evidenced that the users' discomfort decreases when the tank volume increases, while the heat pump seasonal performance is essentially unaffected by the thermal storage size.

Several studies can be found in the literature dealing with the energy performance of trans-critical CO_2 heat pumps coupled to different kind of buildings, such as hotels (Smitt et al., 2021), supermarkets (D'Agaro et al., 2018), office buildings (Visser, 2019), hospitals and schools (Tammaro et al., 2017). Moreover, R744 is nowadays accepted as a suitable alternative to traditional refrigerants also in the residential sector in both heating and DHW applications. Numerous papers have studied the performance of CO_2 heat pumps for combined production of DHW and hot water for heating in residential buildings both theoretically (Brodal and Jackson, 2019) and experimentally (Bastani et al., 2020).

Nevertheless, at the best of the Authors' knowledge, few studies have investigated the effective energy performance of R744 heat pumps dedicated to pure hot water heating in multi-family residential buildings. Stene and Alonso (2016) published the results of an annual monitoring campaign conducted on a brine-to-water CO_2 unit, coupled to a block of 243 flats located in Norway, for pure DHW production. Refrigerant temperatures and pressures, as well as brine and water temperatures, were collected during the monitoring campaign to evaluate the gas cooler heating capacity. Nonetheless, due to limitations in the measuring and monitoring system, the seasonal performance factor of the heat pump was only estimated and not directly measured.

In this work, the annual energy performance of an air-to-water CO_2 heat pump coupled to a multi-storey residential building and dedicated to DHW production is investigated by means of numerical simulations. The numerical model of the building and of the heat pump-based DHW system has been built using TRNSYS17, one of the most widespread software for energy modelling of both buildings and complex HVAC systems. The model has been calibrated using measured data collected during a monitoring campaign performed for three months. Particular effort has been dedicated to the definition of an overall hot water tap profile, introducing the contemporaneity factor between the requests of each flat, and the control strategy of the system. The influence of the thermal stratification of the storage is also assessed, highlighting how the effective thermocline within the tank must be always correctly reproduced in the numerical model in order to obtain realistic values for the system performance indicators.

Building and heating system description

Characterization of the flat complex

The multi-family building selected for this analysis is an apartment block located in Bologna, in the north of Italy (latitude = 44.50° N; longitude = 11.34° E). The apartment building is characterized by an unheated basement, in which cellars and garages are present, and eight storeys. One office and the stairwell entrances are located at the ground floor, while each storey between the first and the sixth ones consists of four flats and is characterized by the same layout. Moreover, at the seventh floor two attics are present: therefore, a total of 26 apartments are present in the block. The dimensions of the flats are significantly different: the net floor area of each apartment ranges between 110 m^2 and 150 m^2, while the attics are characterized by a larger floor area (about 180 m^2 each one). The overall net floor area of the building is equal to 3826 m^2, corresponding to a gross heated volume of 11477 m^3.

The block was built at the end of '60s and, recently, has been partially renovated with typical retrofitting measures: 10.5 cm of thermal insulation were installed on the horizontal roof and new double-glazing windows were adopted. Furthermore, the centralized heating system of the building was completely retrofitted. The old installation, based on a traditional gas boiler for combined space heating and DHW preparation, was dismantled and different heat generators were installed. Specifically, space heating is provided by 3 air-to-water heat pumps, connected in parallel and coupled to a thermal storage tank, while DHW is prepared by means of a trans-critical CO_2 heat pump, coupled to a thermal storage system described in detail later. A condensing gas boiler is used as back-up heater for both services.

Configuration of the heating system

Figure 1: CO_2 heat pump installed in the flat complex.

DHW production is provided by an air-to-water CO_2 heat pump installed in the shared yard of the flat block. In Figure 1 a picture of the unit is presented.

On the air side of the device, a fin-and-tube heat exchanger with copper tubes and straight fins is used as evaporator, while on the water side, a copper tube coil is used as gas cooler for hot water heating. The heat pump

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is equipped with a dual-stage compressor, in which different typologies of compressor (i.e. rotary and scroll) are used as first and second compression stages.

The hydraulic loop between the heat generator and the multi-family building is represented in Figure 2. Two thermal energy storages (namely TES 1 and TES 2 in that figure) are installed between the heat pump and the building DHW distribution loop. Each thermal storage has a capacity of 500 litres and a cladding of polyurethane foam (7 cm) is applied to decrease thermal losses.

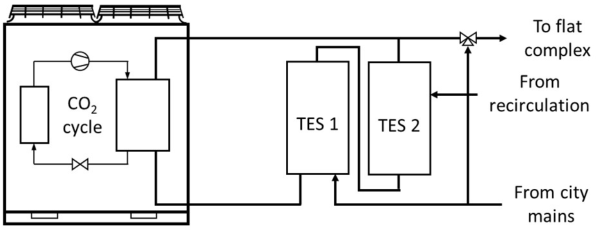

Figure 2: Hydraulic connections of the heating system.

As pointed out by Figure 2, thermal storages are connected in series and a particular hydraulic configuration is used. First, the DHW temperature is controlled by means of a three-way tempering valve: hot water from the heating system and cold water from the city mains are mixed to obtain the comfort DHW temperature. In this case, the set-point temperature at the exit of the three-way valve is maintained constant along the year and equal to 45°C. The town mains are also connected to the bottom part of TES 1; in this way, a cold volume of water can be isolated within this tank and the water temperature at the inlet of heat pump gas cooler can be significantly reduced. The hot water produced by the heat pump can be supplied to the storage tank system or directly to the three-way valve following the rules described in the next section. The adoption of this hydraulic configuration has a twofold motivation: i) to keep the minimum water temperature at the gas cooler inlet by establishing the maximum thermal stratification within TES 1; ii) to store DHW in the system, guaranteeing the best comfort for flat residents, even for a large request of DHW. Moreover, the hot water distribution system presents a recirculation loop, connected to the hotter portion of TES 2.

The heat pump presents a fixed speed circulating pump on-board, provided by the manufacturer. When the circulator is switched on, the mass flow rate of fresh water entering in the gas cooler is equal to about 505 kg/h.

The nominal heating capacity of the heat pump, $P_{th,nom}$, and the nominal *COP*, COP_{nom}, are respectively equal to 30 kW and 4.70 in correspondence of a dry bulb ambient temperature (T_{amb}) equal to 16°C, with a production of DHW from 10°C to 60°C. In Figure 3 heating capacity and *COP* values are reported as functions of the ambient temperature and the inlet water temperature, $T_{w,in}$. These data confirm how the heat pump performance is significantly influenced by the temperature of fresh water at the inlet of the gas cooler, while the ambient temperature has a lower impact on the heat pump behaviour.

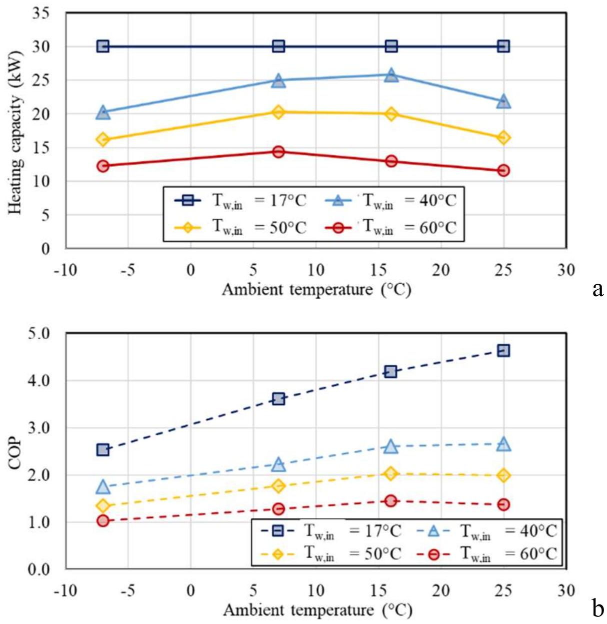

Figure 3: CO_2 heat pump performance data: heating capacity (a) and COP (b).

Control strategy of the heating system

The heat pump operation is supervised by means of an on-off control logic. The temperature of water in the top part of TES 1, $T_{TES1,top}$, is used as monitored variable: specifically, when this temperature is lower than a threshold value, the thermal storage system is considered empty (i.e. no thermal energy is stored) and the heat pump is switched on. On the contrary, when $T_{TES1,top}$ is higher than another threshold value, the heat pump is switched off since both the storages have been charged with hot water. Both the threshold parameters have been fixed by means of the master supervisor of the heat pump on-board controller: the heat pump is activated when $T_{TES1,top}$ drops below 46°C, while the unit is switched off when $T_{TES1,top}$ increases above 50°C.

Depending on the DHW request from the flat block and the quantity of DHW stored in both the thermal energy storages, four working modes can be identified: Figure 4 reports the possible operating conditions of the system.

When the sanitary hot water request of the building is absent or negligible and the temperature in the thermal energy storage system decreases due to ambient losses, the heating system operating mode follows the scheme reported in Figure 4a. In this working mode, the heat pump is switched on to charge the two storage tanks. On the contrary, when the thermal storage system is charged and the building DHW request is low, the system is operated following the scheme of Figure 4b: the CO_2 unit is switched off while DHW is prepared by mixing cold water from the city mains and hot water from the storages at the 3-way valve.

When the overall DHW consumption increases, the temperature within the tanks decreases and, consequently, the heat pump is activated to supply the load. Two operating conditions can be established in this case. When the heat pump heating capacity is higher than the thermal

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load required for DHW preparation, the unit simultaneously charges the storages and meets the building draws (see Figure 4c). Finally, as pointed out by Figure 4d, DHW consumption is satisfied by both the heat pump and the thermal storage system when the hot water request reaches the maximum values.

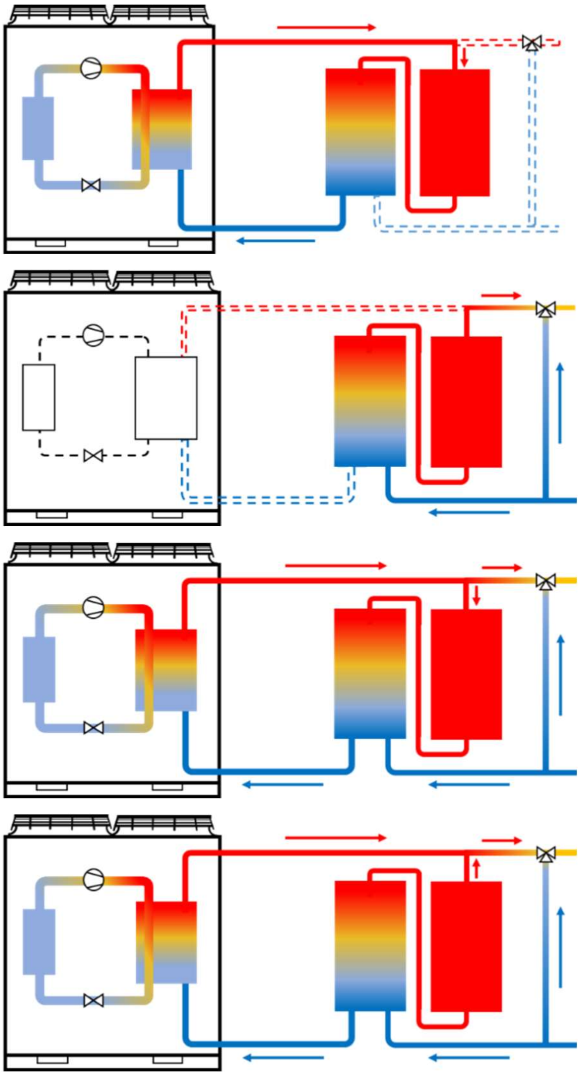

Figure 4: Working modes of the heating system dedicated to DHW production: TESs charging with no DHW request (a); DHW supply from TESs only (b); DHW supply and TESs charging (c); DHW supply from heat pump and TESs (d).

In order to avoid legionella proliferation within the storage tanks, a daily anti-bacteria treatment is provided by the heat pump. Numerical simulations showed that the heat pump seasonal performance is slightly influenced by this treatment.

DHW use and hourly average consumption

Accurate information on DHW consumption and load profile are essential to clearly evaluate the energy performance of heating systems dedicated to hot water preparation. Several models for the estimation of hot water consumption are reported in the literature. Most recent models can be found in the comprehensive review published by Fuentes et al. (2018), who reported the most used DHW draw-off patterns and identified the main variables which influence DHW use, such as seasonality and building type. Moreover, the Authors showed how the typical DHW use in residential buildings significantly varies among different countries, demonstrating that the daily DHW consumption may range between 40 and 94 litres per occupant.

In this paper, the model proposed by Evarts and Swan (2013) has been considered for the estimation of the daily DHW use of each person. Their model, in fact, takes into account the influence of the number of occupants in each dwelling to compute the hot water demand, instead relying on standard consumption. According to data recorded by the block administrator, the total number of residents is equal to 81, corresponding to about 3.1 occupants per household. For this reason, according to Evarts and Swan (2013) the average daily DHW use of each person has been fixed equal to 67 litres with a comfort delivery temperature of 45°C.

In addition, the pattern of hot water draw-offs from the building has to be determined, considering a proper contemporaneity factor for the multi-flat building overall water request. Results reported in Ahmed et al. (2016) have been employed in this work. The Authors developed a series of drawing patterns, classified as a function of the number of residents for different groups of buildings, allowing to simulate the distribution of draw-offs during the day and the contemporaneity of hot water requests. The hourly consumption factor reported for large buildings, characterized by 50 or more occupants, has been selected for the present work. Both the weekly and the seasonal variation of DHW consumption has been considered, by introducing different profiles for workdays and weekends and the monthly corrective factors suggested by Ahmed et al. (2015).

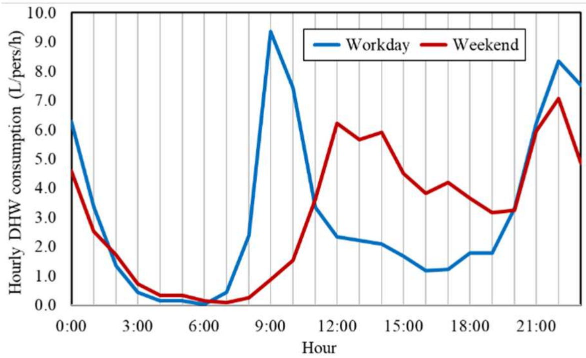

Figure 5: Individual hourly average hot water consumption profile for the month of November.

In Figure 5 the hourly average per-capita DHW use for both workdays and weekends is reported for the month of November as reference. Two sharp peaks can be found in each day, during the morning hours and the evening. It is evident how during workdays the DHW request peak is 3-5 times higher than the average consumption of non-peak hours. During weekends, instead, the daily DHW consumption is smoother; furthermore, the morning peak is shifted 2-3 hours later with respect to workdays, while the evening peaks have the same temporal position.

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The temperature of buried water mains, used as cold source for domestic hot water preparation, is assumed as the soil temperature at a depth of 1.5 m. To model the behaviour of the undisturbed soil, Type77 of TRNSYS has been inserted in the model: this component calculates the temperature of the soil at specific depths. In this way, the influence of seasonality on the energy demand for DHW preparation is taken into account: numerical results point out that fresh water temperature ranges between 12°C and 16°C along the year in Bologna.

Numerical modelling of the system

A complete integrated model of the heating system, including the CO_2 heat pump, the building and all the other system components, has been developed within TRNSYS17. In Figure 6 an extract of the simulation model of the heating system is shown.

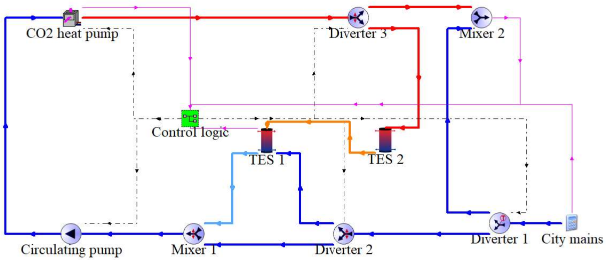

Figure 6: TRNSYS layout of the heating system for the operating mode shown in Figure 4(c).

Yearly simulations have been performed with a timestep of 1 min. Particular attention has been paid for the simulation of the thermal storage system. Type534 has been used to model each tank: the storage volume is subdivided into 10 isothermal constant-volume nodes to simulate the water thermocline within the storage.

Calibration of the numerical energy model with monitoring data

In order to calibrate the numerical model, a monitoring campaign has been conducted during the winter season 2017-2018. Heat pump performance data (i.e. the thermal energy supplied to the heating system and the electric energy need) have been collected between 24th November 2017 and 17th February 2018 at different intervals, ranging from 2 to 7 days.

In Table 1 data obtained from the monitoring campaign and the average *COP* along each interval are reported. It is important to highlight that, reading 11 in Table 1 corresponds to a period of 27 days. For this reason, both the heat pump supplied thermal energy and the electric energy use are much higher with respect to other readings.

It is important to stress how the heat pump performance during the entire monitoring period were very low: the overall *COP* during the period was equal to 1.66, much smaller than the performance declared by the heat pump manufacturer under nominal conditions. In addition, the temperature in the bottom part of TES 1 has been monitored during the same period, in order to verify the effective thermal stratification within the storage tank.

To calibrate the numerical model, real weather data are also needed. Hourly values of ambient air temperature, relative humidity and total solar irradiation on the horizontal have been obtained from a weather station of the regional Dext3r network (Dext3r, 2020).

Table 1: Monitoring data collected between 24th November 2017 and 17th February 2018 (WD=workday, WE=weekend).

Reading	N° of days-date of reading	Supplied thermal energy (kWh)	Electric energy demand (kWh)	Average *COP*
1	2-WD	354	177	2.00
2	3-WD	493	260	1.89
3	2-WD	269	146	1.84
4	2-WD	308	172	1.79
5	3-WD	278	159	1.75
6	3-WD	577	327	1.77
7	7-WD	1321	788	1.68
8	2-WE	217	125	1.74
9	3-WD	411	237	1.73
10	5-WE	1025	555	1.85
11	27-WE	6455	3735	1.73
12	7-WE	1709	1122	1.52
13	7-WE	1857	1020	1.82
14	6-WD	1352	1122	1.21
15	6-WD	1401	917	1.53
Total	**85**	**18027**	**10862**	**1.66**

A series of calibration runs has been performed to obtain a numerical model able to reproduce the measured data. The Normalized Mean Bias Error (*NMBE*) and the Coefficient of Variation of the Root Mean Square Error *CV*(*RMSE*) have been calculated during the calibration process and used as uncertainty indexes (Guyot et al., 2020). Their values have been verified to be consistent with the limits reported in ASHRAE Guideline 14 (ASHRAE, 2014): following this standard, *NMBE* and *CV*(*RMSE*) must be included within ±5% and below 15% on monthly basis, respectively, and ±10% and 30% on hourly basis, respectively. Collected data have been summed to obtain monthly data and, then, the uncertainty indexes have been evaluated on a monthly basis.

A set of the most influent parameters on the heating system energy demand has been selected and, for each parameter, a variation range has been defined. According to the technical standard EN 15603 (European Committee, 2008), the variation constraints of the selected parameters have been set. For example, the number of occupants can be varied between lower and upper bounds set to 10%, while for the system efficiency the range was limited to 5%. The energy model has been calibrated by using an iterative "trial and error" approach (Guyot et al., 2020), based on the Authors' experience.

The effects of the calibration runs are represented in Figure 7. Here, the heat pump performance data are summarized for the most significant simulations.

Simulation 1 (Sim 1 in Figure 7) is the baseline for the calibration runs. In this case, thermal stratification within the two thermal storages was considered and no variations of input data were introduced. Results point out that this model accurately evaluates the thermal energy supplied

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by the heat pump, while, on the contrary, dramatic differences can be found in terms of heat pump electric energy use and, as a consequence, in terms of *COP*. The average *COP* calculated with the numerical model is close to 4.70 during the whole monitoring period, about 3 times higher than the effective *COP*.

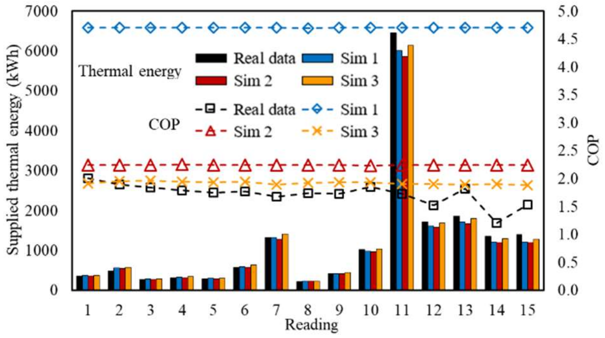

Figure 7: Heat pump performance data obtained during the calibration process compared to effective data.

The comparison with monitored data highlights how this strong difference is related to the temperature gradient within the thermal storage system. In Figure 8 the values of the heat pump inlet water temperature $T_{w,in}$ are reported for each reading, by comparing data from the monitoring campaign with those from simulation 1, in which the thermal stratification within both the TESs is taken into account. It is evident that the temperature gradient within TES 1 is not guaranteed in the real heating system. In fact, $T_{w,in}$ has very high values, up to 50°C, and only for a limited period is below 20°C. On the contrary, when the stratification is considered, the average inlet water temperature is equal to 19°C along the whole period and, for this reason, a better energy performance of the heat pump is obtained. Possible reasons for this lack of thermal stratification are: i). the non-optimal volume of storage tanks connected in series; ii). a too large water flow rate within TES 1.

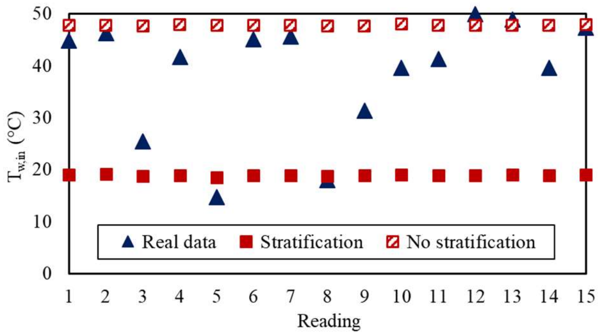

Figure 8: Heat pump inlet water temperature from experimental and numerical data.

According to effective data, thermal stratification within the thermal storages has been neglected during the following calibration runs, summarized as Sim 2 in Figure 7. Specifically, in these simulations the number of tank nodes used in Types534 for the tank modelling is reduced from 10 to 1 to eliminate the thermocline. Obtained results outline how the heat pump performance is strongly reduced, since the average inlet water temperature increases up to 48°C and the average *COP* along the monitoring period drops to 2.24.

Calibration was continued varying: the number of occupants, the thermal insulation of the storages, the distribution temperature and the heat pump performance (i.e. heating capacity and *COP*), keeping no stratification within both the tanks. Main results are summarized as "Sim 3" in Figure 7. The average heat pump *COP* is further decreased to 2.13, while the overall thermal energy supplied by the unit increases up to 17823 kWh, a value really close to the amount of heat given by the heat pump during the monitored period reported in Table 1.

Table 2: Values of NMBE and CV(RMSE) for performed calibration runs.

Simulation run	Uncertainty index	Supplied thermal energy	Electric energy demand
1	*NMBE*	5.1 %	66.5 %
	CV(*RMSE*)	12.1 %	109.3 %
2	*NMBE*	7.2 %	31.3 %
	CV(*RMSE*)	15.4 %	53.1 %
3	*NMBE*	1.1 %	13.9 %
	CV(*RMSE*)	8.3 %	27.7 %

During each step of the calibration process the uncertainty indexes have been calculated on a monthly basis to assess the consistency with limits defined by ASHRAE Guideline 14. In Table 2 the values of both *NMBE* and *CV*(*RMSE*) for the main simulation runs are shown. It is evident how the calibration process allows to reduce the uncertainty of the developed numerical model. More in detail, following the last group of simulation runs (i.e. Simulation 3), the values of *NMBE* and *CV*(*RMSE*) for the heat pump supplied thermal energy are perfectly in agreement with the standard limits. On the other hand, the calibration of heat pump electric energy use is a harder task. In fact, the obtained values are significantly out from the monthly data limits and it can be concluded that the heat pump electric energy input does not meet the calibration criteria. A further reduction of the heat pump declared *COP* below the limit defined by the standard EN 15603 seems justified. In fact, the heat pump performance is affected by other transient phenomena, such as on-off cycling losses and defrost cycles, which may reduce the heat pump efficiency up to 5-10 % on a seasonal basis (Dongellini and Morini, 2019). Both cycling and defrost energy losses are not considered in this model for a lack of data from the heat pump manufacturer. Furthermore, additional data are needed to increase the accuracy of the validation, such as the axial water temperature at different heights of the thermal storage tanks.

In conclusion, the numerical model developed with TRNSYS software package can be considered as partially calibrated (with the limitations reported in the previous paragraph for absorbed electric energy) and the annual performance of the heat pump can be now calculated.

Annual performance of the heating system

The annual performance of the CO_2 heat pump has been calculated with the calibrated numerical model. In Figure

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9, monthly values of supplied thermal energy, electric energy use and *COP* are represented.

Figure 9a reports the global performance of the heating system calculated with the calibrated model (i.e. with no thermal stratification within the storages and including all the variations assessed during the calibration process). Results point out that the heat generator is characterized by a yearly performance factor of 1.84, a very low value, due to the lack of stratification in the storage tank and the high water temperature at the inlet of the heat pump. Furthermore, the seasonality has a significant effect on the monthly thermal energy supplied by the heat pump and a negligible influence on its energy performance. During hot months, in fact, the higher temperature of the cold water from the city mains and the lower request of DHW from the multi-flat building are linked to a lower request of heat from the heat pump.

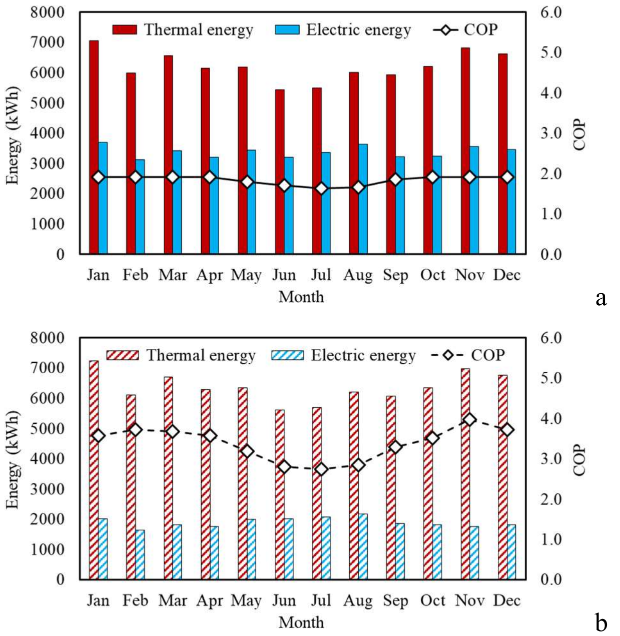

Figure 9: Annual performance of the heat pump without (a) and with (b) thermal stratification within TESs.

As comparison, in Figure 9b the heat pump performance data obtained by considering a perfect stratification within the thermal storage system is reported in order to highlight the achievable heat pump energy performance. On the other hand, the variations of the other parameters investigated during the calibration process have been still considered. By comparing Figure 9a and Figure 9b, it is evident that the annual *COP* of the heat pump is almost doubled, increasing up to 3.36 (+83% if compared to the previous case) due to the lower water temperature at the inlet of the heat pump gas cooler. It is important to stress that with this configuration a larger penalization of the heat pump *COP* during the hot season can be observed. In fact, with an ideal thermal stratification within the tank, higher water temperatures from the city mains have a more remarkable effect on the tank temperature gradient and, thus, on the inlet water temperature. Moreover, also the lower DHW consumption during the summer has a not negligible impact on the thermal distribution within the storages. In fact, the heat pump capacity is more frequently higher than the thermal load required for DHW production and the heat pump meets the hot water request while contemporary charging the storages. In this way, the undesired recirculation of hot water from the heat pump supply to the gas cooler is more frequent and the heat pump *COP* is reduced.

Conclusion

In this paper the annual performance of a CO_2 heat pump dedicated to DHW production for a flat block located in Bologna (Italy) has been assessed by means of numerical simulations. A numerical model including the multi-family building and the heat pump-based heating system has been implemented in TRNSYS. The heat pump performance has been calibrated with monitoring data collected during the winter season 2017-2018. Results of calibration runs point out that the heat pump supplied thermal energy is perfectly in agreement with calibration criteria, while electric energy use does not meet the same criteria. This is due to the lack of a complete set of measured quantities needed for a more accurate calibration of the model.

The comparison between measured data and numerical results shows that the effective heat pump performance is dramatically lower than expected. The main issue is related to the lack of an ideal thermocline within the thermal storage coupled to the heat pump: experimental data highlight how the water temperature at the inlet of the gas cooler is very high, up to 50°C. Similar values have been obtained through numerical simulations neglecting the temperature gradient within the tank. Annual simulations point out that the heat pump efficiency can be almost doubled if a proper temperature gradient is maintained within the storage during DHW production. Moreover, numerical results show that the heat pump performance is lower during the summer period, due to higher water temperatures from the city mains and lower hot water request from the building.

In conclusion, results presented in this work confirm how the energy performance of CO_2 heat pumps is strongly influenced by the configuration of the thermal storage system and, for this reason, the effective temperature distribution within the tank should be always considered in numerical simulations. Moreover, the size of the storages and the water circulation among the tanks and the heat pump should be optimized to maintain the desired thermocline within the storage tanks. The calculation of the total thermal storage volume coupled to the heat pump must be done with the aim to maintain the ideal thermocline within TES 1. The adoption of a variable-speed circulating pump can facilitate this goal by reducing the water flow rate through the tank when needed.

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