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SIMULATIONS AND FIELD TESTS OF A CO_2 REFRIGERATING PLANT FOR COMMERCIAL REFRIGERATION

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ABSTRACT

During the refurbishment of a supermarket in northern Italy, the HFC refrigeration plant has been replaced by a new CO_2 transcritical system. A deep synergy has been favoured with the HVAC system, by allowing mutual heat exchanges at various temperature levels. Simulations have been performed at the design stage, with a TRNSYS based tool and in-house types for the refrigerating unit and the display cabinets. The plant was fully instrumented, and the availability of measured data on the new system allowed assessing its actual performance and validating the model. It was then possible to investigate the effectiveness of the heat recovery solutions applied, and to optimize their control rules.

Keywords: CO_2 booster system, Commercial refrigeration, Heat recovery, HVAC integration

1. INTRODUCTION

The reduction of the environmental impact due to the energy use of shopping malls is the main driver for strategies of heat recovery from the commercial refrigeration systems in favour of the HVAC plant. This is especially true when CO_2 is employed as refrigerant, with the aim of removing any direct greenhouse effect. With this refrigerant the transcritical operation and the high discharge temperature at the compressors allow great opportunities for heat recovery solutions (Polzot 2015, 2017). Also Karampour (2017) and Sawalha (2013) evaluated that heat recovery from a CO_2 system can cover a significant fraction or the entire heating demand of an average size supermarket depending on the climate, with a comparable annual energy use against a conventional R404A system where a separate heat pump is used for the heating needs. However, heat recovery affects the performance of the refrigeration system, as its condensing (or gas cooling) pressure has to be raised to match the heating demand. This requires a thorough evaluation at the design stage of the benefits deriving from this configuration, which can be performed by modelling the whole system (Karampour 2014). In this paper the model of a CO_2 booster refrigeration system with heat recovery is described, and validated against measurements taken on a plant located in Italy, at "humid subtropical climate" conditions (Peel 2007). The model is built in the Trnsys environment (Klein 2010), each component (i.e. display cabinets, the whole refrigerating plant, heat pumps) has been described by in-house Types, based on thermodynamic relations. A time dependent simulation is performed, with a 1 min time step based on actual weather and room temperature conditions, to validate the model. Annual simulations with an hourly time step are then performed for the estimation of power profiles and annual energy utilization, in order to optimize the control rules.

2. REFRIGERATION SYSTEM

The supermarket considered for refurbishment in the framework of the FP7 European Project CommON*Energy* (Cortella 2014, Commonenergy 2017) is a small supermarket of ca. 1200 m^2 selling area, located in Modena (IT). It underwent renovation during the summer 2016 and the retrofitting process involved several aspects such as envelope retrofitting, solutions for both artificial and natural lighting. In particular, the solution set which involved the refrigeration system was also focused on its integration with the HVAC.

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2.1 Refrigeration system description

The previous refrigeration system has been replaced with a new generation type, which is composed by a transcritical CO_2 booster system as commercial refrigeration unit (CRU) and closed refrigerated display cabinets, both for chilled and frozen food.

The new refrigerated display cabinets consist of approximately 14 m for the low temperature level (LT) cabinets and 63 m for medium temperature level (MT), thus the estimated nominal compressor capacity is of 70.5 kW for the MT and 10.8 kW for the LT.

Figure 1 shows a schematic drawing of the booster system installed.

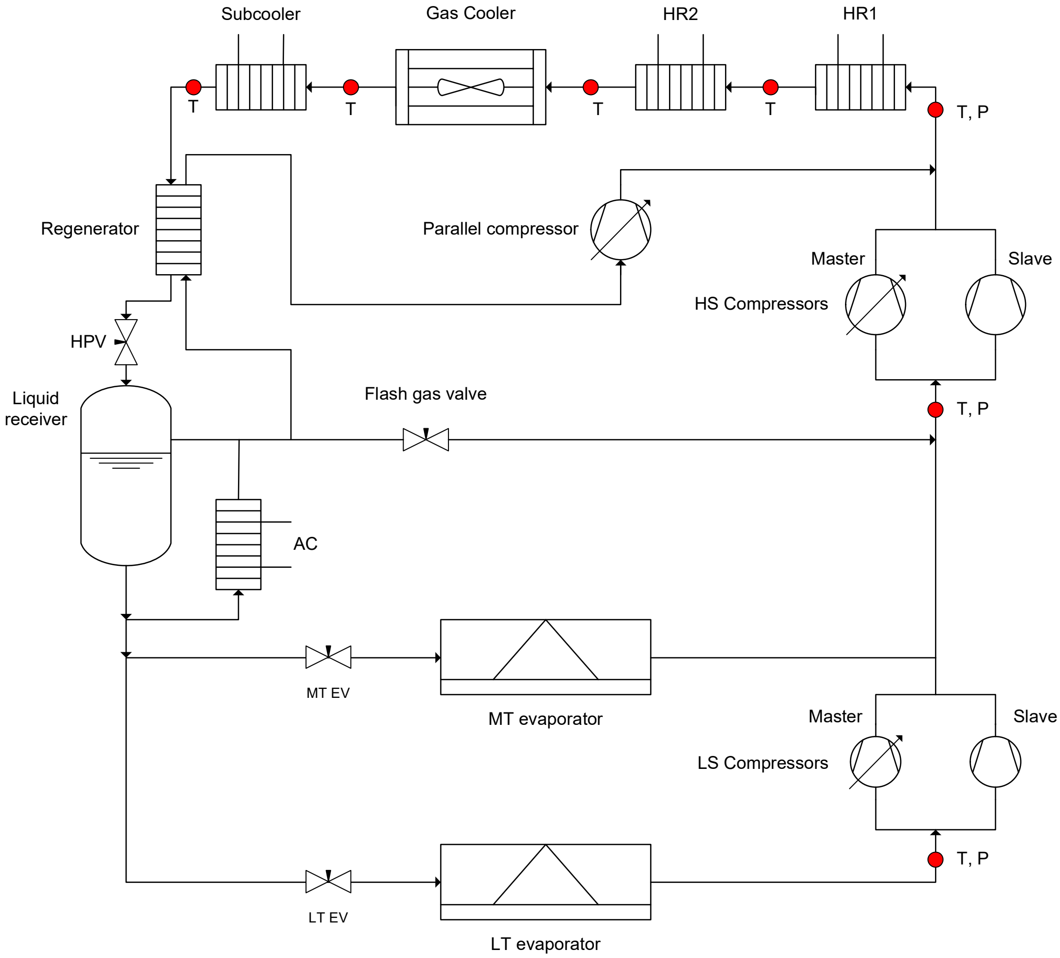

Figure 1: Schematic drawing of the CO_2 refrigeration system with the indicative position of the probes (red flags).

It works at two temperature levels, LT and MT, in this case fixed at respectively -35°C for frozen food equipment and -10°C for chilled food. Each stage has its own compressors, the LS (Low Stage) work between the LT and the MT levels, while the HS (High Stage) work between the MT level and the high stage pressure driven by the outdoor conditions. A third compressor stage, parallel compressor, is used to process the flash gas coming from the liquid receiver placed after the first expansion valve, HPV (High Pressure Valve), from an intermediate pressure p_{INT} of 37.7 bar to the high stage pressure p_{HS}.

Each compressor rack, LS and HS, is composed by two compressors: the master is controlled by an inverter and the slave is an ON/OFF type. The high stage nominal power is of 23.7 kW, 8.1 kW for the master compressor and 15.6 kW for the slave; similarly, the low stage has a nominal power of 2.4 kW, 1.05 kW for the master and 1.35 kW for the slave. Other 9 kW of nominal power must be considered for parallel compression.

Because of the small size of the supermarket with a considerable prevalence of food sales area, recovered waste heat from the refrigeration system can significantly contribute to reduce the supermarket energy use for heat production. Thus, the system allows for heat recovery at two temperature levels in order to recover heat for Domestic Hot Water (DHW) production (HR1 exchanger) and for space heating (HR2 exchanger).

A further integration with the HVAC system involves the possibility for the CRU to provide cooling capacity to the air handling unit by means of a heat exchanger (AC) that works with refrigerant at the intermediate pressure level, where the temperature is suitable for chilled water production. On the other way round, a subcooler can be used at the expense of the HVAC system to further lower down the gas cooler outlet temperature.

In the present paper, we focus on heat recovery in favour of DHW (HR1) and space heating (HR2), as no monitored data is available yet to validate the integration in terms of AC heat exchanger and subcooler.

2.2 Model of the refrigeration system and control rules

The mathematical models of the components of the refrigeration system, such as display cabinets, cold rooms and commercial refrigeration unit, have been developed in the TRNSYS environment, as it allows carrying out dynamic simulations of a complex system, which includes mutual interactions with the building and with the HVAC system.

The models of the display cabinets and of the cold rooms, which are described in detail in Polzot *et al*. (2016), estimate the total cooling load on the evaporators of the commercial refrigeration unit (CRU) by tuning the performance at rated conditions in accordance with the realistic and time-dependent working conditions in a supermarket. At the same time, the sensible and latent credits from the refrigerated equipment to the HVAC system are calculated via the evaluation of the heat and mass transfer between the refrigerated volume and the surroundings.

The CO_2 transcritical booster system with parallel compression is modelled through the definition of the thermodynamic cycle in which the operating conditions are dynamically modified as a function of the instantaneous cooling load at the evaporators and of the outdoor temperature. A detailed description of a general booster system can be found in Polzot *et al.* (2016). Here we underline that the properties of the refrigerants at different points of the thermodynamic cycle are given by linking, in Trnsys environment, our in-house routines to the CoolProp libraries (Bell *et al.*, 2014), and that correlations from BITZER Software (Bitzer, 2017) have been used to define the performance of compressors.

The control rules of the plant, inferred from the monitoring data, have been implemented in the model as follows:
- when the heating demand from the HVAC system in the supermarket is zero ($q_{heating}$=0), the high stage pressure p_{HS} is driven by the outdoor temperature. In particular, in transcritical operation the high stage pressure p_{HS} is set according to a correlation that maximizes the COP (Polzot *et al.*, 2016) and the auxiliary compressor is switched on.
- when the HVAC system in the supermarket asks for heating ($q_{heating}$ >0), the refrigeration central unit is set to transcritical mode in order to insure that the temperature at the exit of HS compressors is high enough for providing space heating. The discharge pressure p_{HS} is increased to 78 bar and heat recovery in HR2 is activated in series to HR1. In winter season, the auxiliary compressor is switched off because the temperature at the exit of the gas cooler keeps a low production of flash gas. In both cases the heat recovery for DHW production is active whenever the HS outlet temperature is suitable to heat the water up. The amount of heat transferred is the maximum available.

The LS and HS compressors are set to operate in order to cope with the cooling demand at the two evaporation temperature levels at each instant, i.e. to process the required flow rate.

As already mentioned, each stage is composed by two compressors, one controlled by an inverter and the other one not. The operation range of the variable speed compressor is from 40% to 100%, respectively from 30 Hz to 60 Hz. Due to this constraint, at the switch between the two compressors the refrigerant flow rate may not coincide with the one required by instantaneous cooling capacity.

The heat exchangers HR1 and HR2 are modelled simply by assuming an appropriate approach temperature value. HR1 provides hot water at 60 °C to avoid the growth of legionella. The supply temperature and the minimum return temperature of the heating system is fixed at 45 °C and at 35 °C, respectively.

The values of the main design parameters are reported in Table 1.

Table 1. Main design parameters for the commercial refrigeration unit

Parameter	Value	Unit
LT Evaporating temperature	-35	°C
MT Evaporating temperature	-10	°C
Minimum condensing temperature	8	°C
Liquid receiver pressure p_{INT}	37.7	bar
p_{HT} with heat recovery in HR2	78	bar
Subcritical subcooling	3	K
Gas Cooler/Condenser approach temperature	4	K
HR1 and HR2 approach temperature	5	K

3. MODEL VALIDATION

The collected data have been thoroughly analyzed in order to prove the values of the setting parameters (Table 1) and infer the values which are intrinsic characteristics of the plant itself and thus necessary to calibrate the model. As an example, the superheating at the two temperature levels are almost constant in time, thus suitable averaged values have been introduced in the model: 30 K for LT superheating and 20 K for MT one. Eventually the model has been validated with the monitored data, which were available from April to November 2017. In the following, for the sake of brevity, the comparison between the results from simulation and the monitored data is reported for three selected weeks, each representative of a different operating condition: subcritical (week 1, from 28th of April to 5th of May), transition (week 2, from 14th to 21st of May), transcritical (week 3, from 30th of June to 6th of July).

3.1 Compressor section

As a starting point, the compressor section has been validated and thus, the instantaneous operating condition of each compressor from the monitored data has been given as input to the model.

Figure 2 and figure 3 show the comparison in terms of pressure and temperature, respectively, at the outlet of the MT compressors. The computed pressure profiles align quite well with the measured data confirming that the compressor correlations and the control rules of the high stage pressure implemented in the model describe accurately the real system.

As regards the High Stage outlet temperature, it is important to remark that in all the weeks considered it appears that the difference between the value from simulations and the measured one is approximately constant and is, on average, around 13 K. Such temperature difference indicates that there is an appreciable heat loss in the line between the compressor outlet and the position of the corresponding temperature probe. As this temperature is fundamental in the evaluation of the heat recovery potential, the temperature drop has been introduced in the model as part of the calibration setting.

One of the most important outputs of the model is the power and energy use of the system. Figure 4 shows the comparison between the electric power values predicted and measured. In both, two patterns are present; the power values switch from the lower to the upper one when the slave HS compressor is activated. The energy meter registers all the electrical energy utilization of the CRU, including auxiliaries which account for 5 kW on average. This is the reason for the slight difference between measured and computed values in Fig. 4.

3.2 Cooling capacity

Once the compressor section of the CRU model was calibrated, the prediction of the cooling capacity has been introduced. Some monitored data, mainly related to direct energy utilization of the refrigeration equipment, have been used to refine the models of the display cabinets and cold rooms. As an example, the signal of defrost operation has been used to correct the defrost frequency and duration thus influencing the daily profile of the cooling capacity.

Unfortunately, no data are available to infer the cooling capacity of a single display cabinet nor a group of them, thus the mass flow measured at the exit of the receiver is the only data available for the validation of the total cooling capacity. The refrigerant mass flows computed from the simulated cooling capacity at the LT and MT levels define, as stated above, the status of the compressors, which in turn gives the instantaneous mass flow processed by them.

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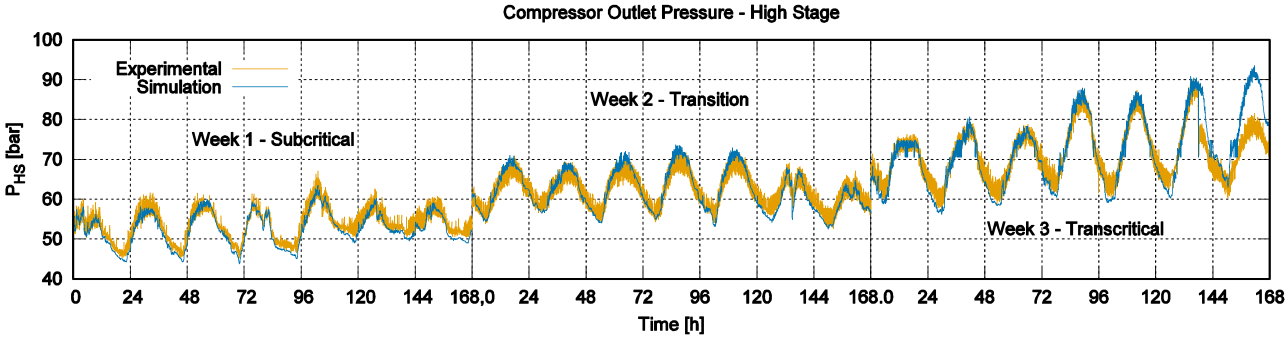

Figure 2: Pressure at the outlet of HS compressors: comparison between monitored data (experimental) and simulated results (simulation) for the selected weeks

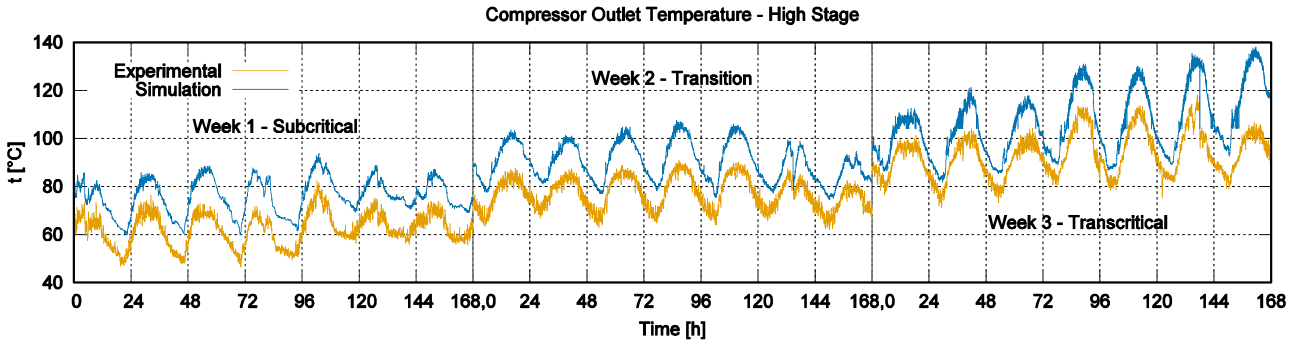

Figure 3: Temperature at the outlet of HS compressors: comparison between monitored data (experimental) and simulated results (simulation) for the selected weeks

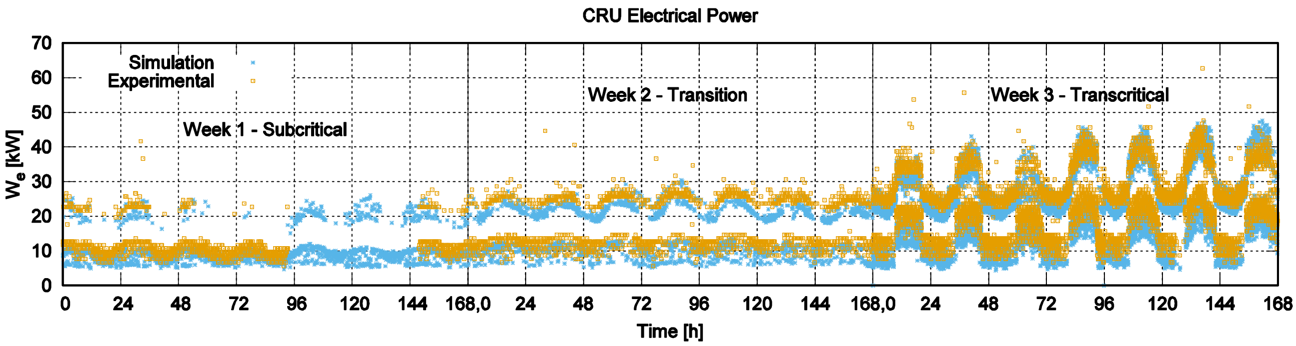

Figure 4: CRU electrical power values: comparison between monitored data (experimental) and simulated results (simulation) for the selected weeks

In Figure 5 the computed flow rate, net of the flash gas flow rate, is compared to the measured mass flow for a week at subcritical conditions. The monitored data are much more scattered around the mean value with respect to the simulated ones, which gather around two values: the lower one corresponding to the status when the slave HS compressor is switched off and the upper one when it is activated. This seems to suggest, on one hand, that the variation in the compressor status is much more rapid in reality than the one simulated and, on the other, that the oscillations in the mass flow at the exit of the MT compressor (computed value) are damped at the exit of the separator (measured value) thanks to the refrigerant volume in the plant.
Nevertheless, the simulated weekly averaged value differs from the measured one by 5% to 16% for the two months considered.

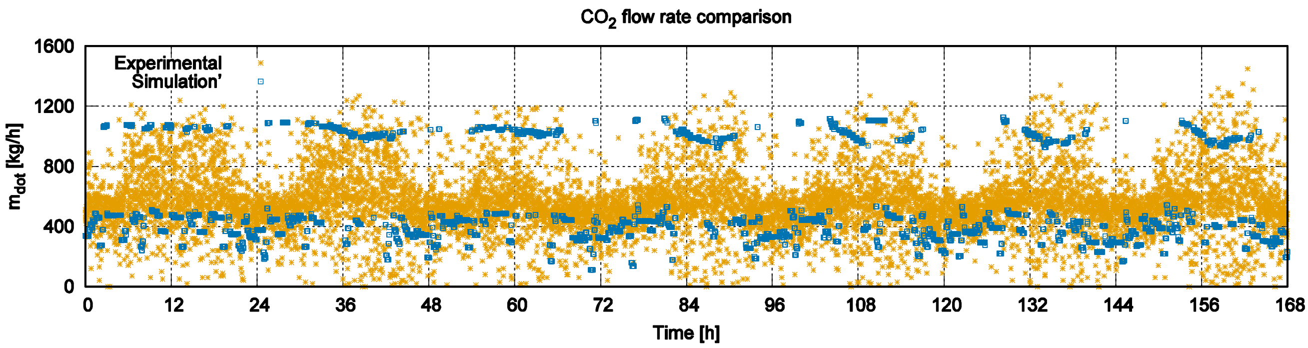

Figure 5: Mass flow rate comparison between monitored data (experimental) and simulated results (simulation) for a subcritical week

3.3 Electrical energy utilization

The comparison in terms of weekly electrical energy demand between monitored and simulated data is reported in Table 2.

Table 2. Comparison between monitored and simulated electrical energy demand for the selected weeks.

Week		Operating conditions	Monitored Electrical Energy Demand [kWh]	Simulated Electrical Energy Demand [kWh]	Error [%]
1	28/04 – 05/05	Subcritical	1340	1270	5.6
2	14/05 – 21/05	Transition	3020	2220	26.5
3	30/06 – 06/07	Transcritical	3970	3750	5.7

It can be noticed that the model predicts very well (within an error of 6%) the CRU energy utilization when the system operates in subcritical or transcritical conditions, whereas the error is much higher (up to 27%) when the system operates in the transition region. This behaviour cannot be due to an incorrect implementation of high pressure control rules in this region given the optimal correspondence between predicted and measured p_{HS} in Fig. 2 for transition week 2. It is rather due to the control rule on HS compressor activation: in transition the power required is normally around the maximum value of the master compressor thus a small variation is enough to activate the bigger slave compressor (the on/off one) which produces a remarkable step increment on the total power. Therefore, a small underestimation of the simulated cooling capacity with respect to the real one implies that the slave compressor is not activated in the model as much as it is in reality and, thus, the estimation of the energy use is much lower.

4. HEAT RECOVERY STRATEGIES

The model that describes the CRU and the refrigerated cabinets of the supermarket considered has been validated under different operating conditions and can be used to perform rather reliable energy use predictions for different scenarios. The main focus is on how to profitably exploit the heat recovery exchangers all year round through optimising the control rules (Ge, 2014).

The cases that have been analysed in this work are essentially the following:

- Case 0 (reference case): the CRU operates only to cover the refrigerating requirements. DHW and Heating demands are completely satisfied using heat pumps.
- Case A: all the available heat at the two recovery temperature levels is used, while the CRU normally operates to yield the refrigeration cooling capacity required.
- Case B: when heating is required the CRU switches to transcritical mode (with p_{HS} = 78 bar) in order to achieve a suitable temperature level at the HS compressors outlet that guarantees to always yield heat at the HR2 exchanger.
- Case C: when heating is required the CRU switches to transcritical mode, but this time the gas cooler pressure varies between 76 bar and a limit of 85 bar following the heating demand. When the maximum value of the discharge pressure is reached, the gas cooler fan speed is controlled so that the gas cooler outlet temperature

rises and, as a consequence, the refrigerant flow rate rises in turn, resulting with greater heat available at the exchanger (Sawalha, 2013).
In cases A, B and C, the remaining heat needed to fulfil DHW and heating demands is integrated with heat pumps. Two heat pumps are considered, one for DHW production and the other for the heating demands of the HVAC system. Both have been set with a COP value depending on the outdoor temperature (Polzot, 2017). There is no heat storage to allow heat recovery in favour of DHW above 100% of the demand when the energy available permits so.
The DHW and space heating demands yearly profiles used in this analysis result from the building simulation of the supermarket and are shown in figure 6.

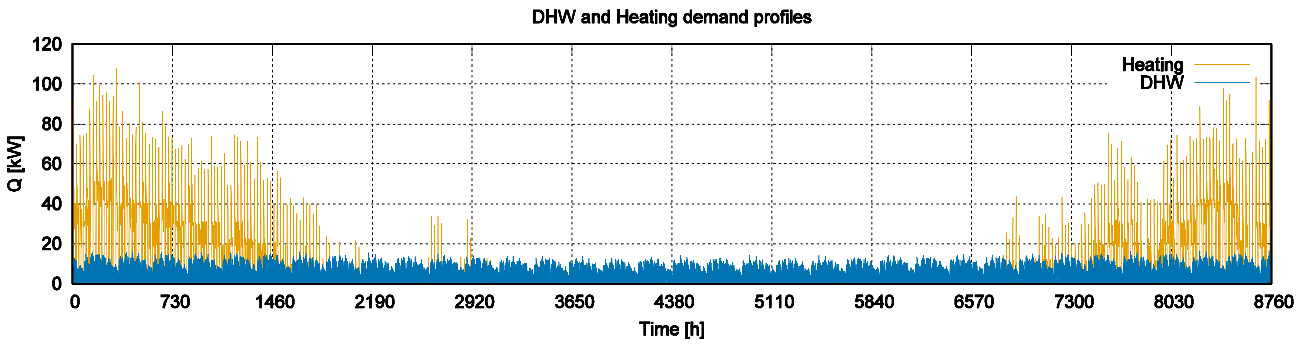

Figure 6: DHW and Heating demand profiles for a whole year.

The electrical energy demand of the CRU and of the Heat Pumps (HPs) for a whole year in the analysed cases are reported in Table 3.

Table 3. Comparison between the annual electrical energy demand of the cases considered.

	Annual Electrical Energy Demand [kWh]			
	CRU	**HPs**	**Total**	**Difference [%]**
Case 0	129643	28412	158055	0,00%
Case A	129643	22190	151833	-3,94%
Case B	141158	15351	156509	-0,98%
Case C	149551	4931	154482	-2,26%

It results that in all cases heat recovery leads to savings in the global energy utilization. Case A is the most favourable mode. This happens because when the CRU switches to transcritical, its electrical energy demand sensibly increases showing that it is better to use the heat pumps instead of forcing the CRU, whose evaporating temperature is on the average lower than that of the heat pump. Once the option of forcing the CO_2 booster to transcritical operation is taken, it is more profitable to implement a control rule which rises the high stage pressure p_{HS} and thus the gas cooler outlet temperature (case C) to a level which allows the direct heating supply. In case C only 5 % of the heating demand remains unsatisfied by heat recovery and in charge of the heat pump, whose installation could be reconsidered.

5. CONCLUSIONS

The model has been successfully validated against experimental data. The simulation of display cabinets and cold rooms led to a very good estimation of the cooling capacity profile, and also the energy use has been well estimated especially at full subcritical and full transcritical conditions. It appears that attention has to be paid at defining well the control rules of compressors, especially the switch between variable speed and fixed speed, and heat loss at the discharge of HS compressors, especially in the view of heat recovery.
With this model it is possible to estimate the behaviour of CO_2 commercial refrigeration plants at different operating conditions, thus giving advices for the definition of control rules. As an example, an investigation was performed on the control rules to promote heat recovery by forcing transcritical operation. The result was in favour of increasing fluid pressure and temperature at the gas cooler to reduce the global energy utilization.

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NOMENCLATURE

CRU	Commercial Refrigeration Unit	LT	Low Temperature
DHW	Domestic Hot Water	MT	Medium Temperature
HR1	heat exchanger for DHW heating	p_{HS}	High Stage pressure [bar]
HR2	heat exchanger for space heating	p_{INT}	Intermediate pressure [bar]
HS	High Stage	$q_{heating}$	Heating demand [W]
LS	Low Stage	t	Temperature [°C]

REFERENCES

Bell, I.H, Wronski, J., Quoilin, S., Lemort, V., 2014. Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. Industrial & Engineering Chemistry Research 53(6), 2498-2508.

BITZER, 2017. BITZER v6.7.0 rev1852 – Available at: https://www.bitzer.de/websoftware/ [accessed 12.12.2017].

CommON*Energy,* Converting EU Shopping Centres Into Beacons Of Energy Efficiency*,* http://www.commonenergyproject.eu [accessed 12.12.2017].

Cortella G., Lollini R., Noris F., D'Agaro P., Saro O., 2014. CommONEnergy: re-conceptualizing shopping malls from consumerism to energy conservation, Proc. 3rd IIR International Conference on Sustainability and the Cold Chain, Twickenham, London (UK), 23-25 June 2014, Paper 213, 582-589.

Ge, Y.T., Tassou, S.A., 2014. Control optimizations for heat recovery from CO_2 refrigeration systems in supermarket, Energy Conversion and Management, 78, 245-252.

Karampour M., Sawalha S., 2014. Supermarket refrigeration and heat recovery using CO_2 as refrigerant. A comprehensive evaluation based on field measurements and modelling. https://www.kth.se/polopoly_fs/1.498819!/EP06-supermarket%20refrigeration%20and%20heat%20recovery%20using%20CO2%20as%20refrigerant.pdf, [accessed 12.12.2017].

Karampour M., Sawalha S., 2017. Energy efficiency evaluation of integrated CO_2 trans-critical system in supermarkets: field measurements and modelling analysis. Int. J. Refrigeration, 82, 470-486.

Klein, S.A., Beckman, W.A., Duffie, J.A., 2010. TRNSYS 17, A Transient System Simulation Program, Solar Energy Laboratory, University of Wisconsin, Madison, USA - http://sel.me.wisc.edu/trnsys [accessed 12.12.2017].

Peel M.C., Finlayson B.L., McMahon T.A., 2007. Updated world map of the Köppen-Geiger climate classification, Hydrol. Earth Syst. Sci., 11, 1633–1644.

Polzot, A., D'Agaro, P., Gullo, P., Cortella, G., 2015. Water storage to improve the efficiency of CO2 commercial refrigeration systems. In: Proceedings of the 24th IIR International Congress of Refrigeration; Yokohama, Japan, paper 339.

Polzot A., D'Agaro P., Gullo P., Cortella G., 2016. Modelling commercial refrigeration systems coupled with water storage to improve energy efficiency and perform heat recovery. International Journal of Refrigeration, 69, 313-323.

Polzot A., 2017. Energy benefit assessment of various refrigeration systems integrated with HVAC units in shopping malls. PhD Thesis, Università degli Studi di Udine, Italy.

Sawalha S., 2013. Investigation of heat recovery in CO_2 trans-critical solution in supermarket refrigeration, Int. J. Refrigeration, 36, 145-156.