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Publication date
2017

Document Version
Final published version

Published in
Proceedings 12th IEA Heat Pump Conference

Control strategies of CO_2 refrigeration / heat pump system for supermarkets

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Abstract

CO_2 is a promising refrigerant compared to traditional HFCs due to its insignificant global warming potential and nonthreatening to the ozone layer. It has been used as refrigerant in industrial and commercial refrigeration in recent years. With high compactness and the ability to recover heat, CO_2 booster systems have been widely installed in newly constructed supermarkets in the Netherlands. One remarkable advantage of this system is that great amount of heat can be recovered from the gas cooler for heating use due to high temperature driving force from CO_2. Sometimes the COP is sacrificed to fully satisfy the heating demand. Within the present work, a quasi-steady-state computer model has been developed to study the performance of the system based on a typical Dutch supermarket. The model has been validated using experimental data. By altering condensing pressure and gas cooler capacity using different methods, various control strategies to satisfy both cooling and heating demand have been proposed and compared. The results from the simulation illustrate that some control strategies have lower energy consumption and easier operation compared to the others. The study also shows that CO_2 booster refrigeration system has a good potential for heat recovery. It has the potential to save 13% primary energy compared to conventional heating method in supermarkets.

Selection and/or peer-review under responsibility of the organizers of the12th IEA Heat Pump Conference 2017.

Keywords: CO_2 refrigeration / heat pump system; supermarket; heat recovery; control strategies.

Nomenclature

h	Enthalpy	kJkg^{-1}	
$\dot{m}$	Mass flow	kgs^{-1}	
PR	Pressure ratio	-	
$\dot{Q}$	Heat flow	kW	
T	Temperature	ºC	
$\dot{V}$	Volume flow	m^{3}s^{-1}	
Greek symbols			
ΔT	Temperature difference	K	
η	Efficiency	-	
ρ	Density	kgm^{-3}	

Subscripts

amb	Ambient
evap	Evaporator
g	Flash gas
is	Isentropic
l	Low pressure
L	Leaving cabinets
m	Medium pressure
U	Superheating
UL	Superheating in suction line
v	Volumetric

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1. Introduction

Since CO_2 has no ozone depletion potential and insignificant global warming potential compared to HFCs, it has been researched and widely used as refrigerant in commercial and industrial refrigeration in recent years. Environmentally friendly natural refrigerants are considered to be a long-term solution for refrigeration and heat pump applications especially for supermarkets. New possibility for heat recovery can be provided by refrigeration systems which work with natural refrigerants. Several combinations of refrigeration and heat recovery solutions have been analyzed by Cecchinato et al. [1]. These authors also compared these solutions with conventional systems and concluded that some natural based solutions showed higher energy efficiency compared to conventional solutions. Sawalha [4] considered certain system solutions, based on CO_2 as the refrigerant, as efficient solutions for supermarkets. As intensive energy consumers, supermarkets need simultaneously large amounts of cooling and heating. CO_2 booster refrigeration systems have been introduced to satisfy both environmental and technical requirements,. With a high compactness and the ability to recover heat for heating use, CO_2 booster refrigeration systems are widely used in newly constructed supermarkets in the Netherlands. Sawalha [5] has identified the match between the heating and cooling demands of the supermarket as an important parameter for the system for heat recovery from the supermarket refrigeration system. When ambient temperature and relative humidity are low in cold weather, the cooling demand of the supermarket is low. At the same time the heating demand of the supermarket increases. Therefore an efficient refrigeration system with considerable capacity should be designed for simultaneous cooling and heating demands. Colombo et al. [2] studied energy saving when heat recovery is applied in CO_2transcritical refrigeration system in an existing supermarket. They concluded that considerable energy savings can be obtained and the CO_2 system showed lower energy consumption compared to an existing R404A system. Tambovtsev et al. [7] discussed CO_2transcritical refrigeration systems and emphasized the importance of applying the gas cooler by-pass and an optimally tuned control algorithm. Sawalha[5] theoretically investigated the performance of a CO_2transcritical booster refrigeration system when operated to cover the simultaneous cooling and heating demands in an averaged size supermarket in Sweden. Although CO_2 booster refrigeration systems have the ability to deliver heat recovery, sometimes the COP needs to be sacrificed to fully satisfy the heat demand. The present work shows a model which has been developed using computer software to simulate a CO_2 booster system installed in a specific supermarket in the Netherlands. Different control strategies of the system are studied and compared. The paper covers the following items: - Modeling of the CO_2 booster refrigeration system - Validation of the model - Proposal of different control strategies - Comparison of different control strategies and identification of the one delivering the best performance.

2. CO_2 booster systems in supermarkets

2.1. System analysis

Fig. 1 shows a typical CO_2 booster system installed in supermarkets.

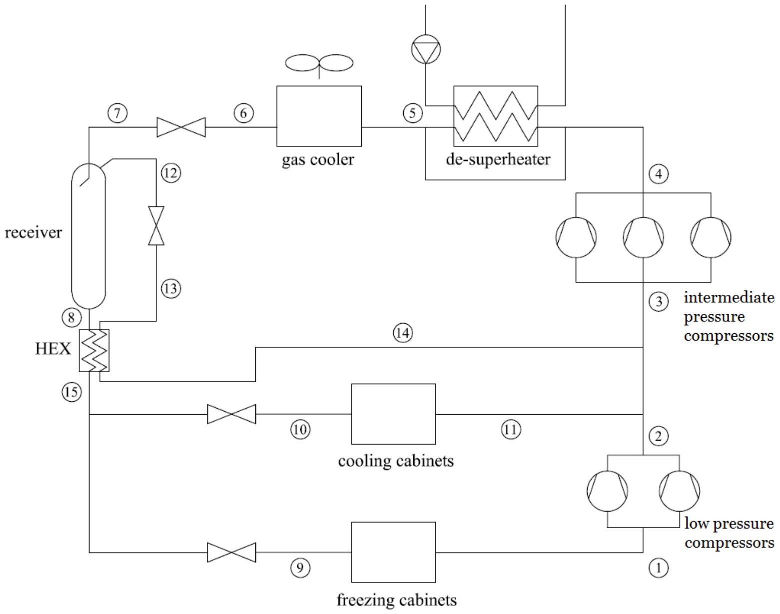

Fig. 1. Typical CO_2 booster refrigeration system for supermarkets.

The system has two different evaporating pressure levels: one to serve the fresh products cabinets and one to serve the frozen products cabinets. Usually by experience, lower evaporating temperature is set to be around -33ºC, while intermediate evaporating temperature remains at around -8ºC. After being compressed to intermediate pressure level in the lower pressure level compressors, the CO_2 stream from the freezing cabinets is mixed with the streams coming from the cooling cabinets and receiver. The total flow will be compressed by the intermediate compressors to reach the condensing pressure. When heating is needed for the supermarket, CO_2 flow will pass through the desuperheater to heat up the heating water on the other side. The heating water will be sent by a circulation pump to fan coils or other equipment to heat up the space air and then return to the desuperheater. If there is no heating requirement, the CO_2 flow will directly go to the gas cooler through a bypass of desuperheater and will reject redundant heat to the environment. After cooling (or condensation) in the gas cooler, CO_2 fluid is throttled to a fixed temperature which is a little bit higher than the intermediate evaporation temperature and is fed into the receiver. The functionality of this receiver is to keep high efficiency of the lower temperature stage in the booster system. Also this receiver can act as a volume storage to adjust the mass flow of the system when operation is changed. The flash gas from the receiver will be further throttled to the same intermediate pressure as that in the cooling cabinets and will get superheated in the internal heat exchanger (HEX in Fig. 1), and will finally mix with the vapour coming out of the cabinets. Saturated liquid coming out of the receiver will be sub-cooled in the internal heat exchanger(HEX in Fig. 1) by the flash gas from the receiver, then further throttled in different valves to different evaporation pressures to feed cooling and freezing cabinets.

2.2. Heat Recovery from the System

In the system, heat recovery takes place in desuperheater. The amount of heat that can be recovered from the system is determined by the mass flow of the refrigerant and the specific enthalpy difference of the refrigerant across the desuperheater. Normally to get higher refrigeration COP, the gas cooler should be operated at the lowest possible pressure, which means the condensing pressure should be as low as possible. But when the refrigeration system is operated to get heat recovered, if the recovered heat at lowest possible condensing pressure cannot satisfy the heating demand, condensing pressure should be raised up to recover more heat. This is explained using Fig. 2.

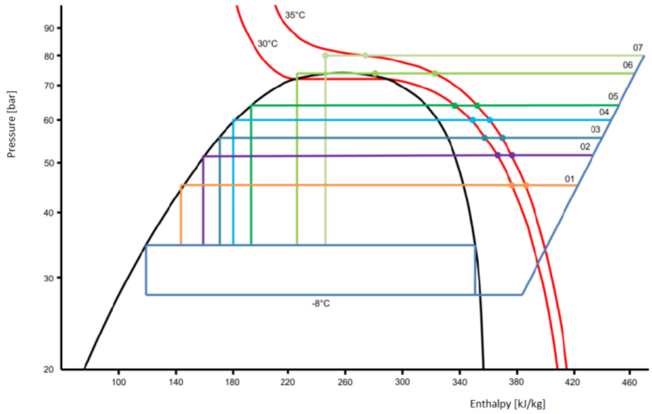

Fig. 2. Effect of condensing pressure in heat recovery.

Assume fixed temperature at the outlet of the desuperheater. From Fig. 2 it can be observed that when the high pressure level is increased from state 1 to state 7, the specific enthalpy difference in desuperheater is increasing. Especially when the isotherm of outlet temperature is in its flatter region around the critical point, the enthalpy difference increases significantly. Also thanks to the receiver, the total mass flow will increase when the discharge pressure increases. These findings theoretically prove that heat recovery can be enhanced by increasing the discharge pressure of the system. While on the other hand, the increase of discharge pressure will definitely cost more compressor work, which will in return negatively influence the refrigeration COP. In the following sections a model will be developed to study this method of operation so that quantitative result can be presented.

3. Model description

3.1. Supermarket information

This study discusses a specific supermarket plant installed in the Netherlands. The floor area of this supermarket is 1,300 m^2. A CO_2 booster refrigeration system is installed which delivers both refrigeration and heating. When heating is needed, heat can be recovered making use of the desuperheater. The plant layout corresponds to the schematics given in Fig. 1. The supermarket is open during 6 days per week. Opening hours is from 8:00 to 20:00. There is no specific air-conditioning system installed in the supermarket. To maintain the temperature inside the supermarket in an appropriate range, the cooling demand of the supermarket caused by high ambient temperature is delivered by the cooling cabinets. The maximum heating demand of the supermarket is 91.0 kW in day time and 71.5 kW at night. Maximum cooling demand for the cooling cabinets is 119.1 kW and freezing demand for the freezing cabinets is 17.1 kW.

3.2. Model description

Different components and parameters of the system are modelled in Matlab using detailed or simplified methods, depending on the specific requirements. RefProp[3] is used to determine the required fluid properties. The state points mentioned in the following paragraphs correspond to those stated in Fig. 1.

Gas cooler

For oil return reasons, the condensation temperature shouldn't be lower than 10°C. For this reason it is assumed that, when the ambient temperature is less than 5°C, the corresponding condensation temperature is 10°C (which means condensation temperature is 5 K higher than ambient temperature when T_{amb}=5°C), and the corresponding gas cooler exit temperature is 8°C (which means sub-cooling in gas cooler is 2 K when T_{amb}=5°C). When the ambient temperature increases from 5°C to 2 K lower than the critical temperature of CO_2 (critical temperature is around 31°C), the temperature difference between condensing and ambient linearly decreases from 5 K to 2 K, in the meantime, the sub-cooling in gas cooler linearly decreases from 2 to 0K. And if ambient temperature keeps increasing, the discharge pressure becomes higher than the critical pressure, what means that the system will operate in transcritical conditions. When the system operates in transcritical conditions, the gas cooler exit temperature is 2 K higher than the ambient temperature.

Receiver

CO_2 fluid from the gas cooler is throttled in a throttling valve and then fed to the receiver. Enthalpy does not change after passing the valve. The receiver operates at constant pressure corresponding to a saturation temperature of -0.9°C.The pressure drop through the receiver is considered negligible. For the receiver, the mass flow ratio between flash gas and saturated liquid should be calculated. The mass flow through the freezing cabinets is taken as $\dot{m}_l$, the mass flow through the cooling cabinets as $\dot{m}_m$, the mass flow of flash gas from the receiver as $\dot{m}_g$. Based on energy conservation, the following relation can be derived:

$$(\dot{m}_l + \dot{m}_m + \dot{m}_g) \times h_7 = (\dot{m}_l + \dot{m}_m) \times h_8 + \dot{m}_g \times h_{12} \tag{1}$$

Then, given the inlet enthalpy, the mass flow ratio can be calculated:

$$\frac{\dot{m}_g}{\dot{m}_l + \dot{m}_m} = \frac{h_7 - h_8}{h_{12} - h_7} \tag{2}$$

Freezing cabinets

The freezing cabinets work as evaporators in the refrigeration cycle. The evaporating temperature in the freezing cabinets, $T_{evap\_freezing}$, is −33°C. The useful superheating in the evaporators, $\Delta T_{U\_freezing}$, is taken as 8 K, while the superheating along the suction pipeline (useless superheating), $\Delta T_{UL\_freezing}$ is taken as 12 K. It is further assumed that the pressure drop in freezing cabinets and suction pipelines is negligible.

Cooling cabinets

The cooling cabinets work also as evaporators in the refrigeration cycle. Evaporating temperature in the cooling cabinets, $T_{evap\_cooling}$, is −8 °C. The useful superheating in the evaporators, $\Delta T_{U\_cooling}$, is taken as 8 K, while the superheating along the suction pipeline (useless superheating), $\Delta T_{UL\_cooling}$ is taken as 4 K. It is further assumed that also the pressure drop in cooling cabinets and suction pipelines is negligible.

Heat exchanger (HEX)

Flash gas from the receiver is throttled to the intermediate pressure of the system and exchanges heat with the saturated liquid from the receiver in the internal heat exchanger. Saturated liquid is sub-cooled in the heat exchanger, what contributes to larger enthalpy difference in both cooling and freezing cabinets and leads to a higher COP of the system. The enthalpy does not change after passing the throttle valve. From experimental experience it is known that the outlet temperature of the heat exchanger (state point 14) is 5 K higher than that of state point 10 (which is the evaporation temperature in cooling cabinets). The enthalpy at the liquid side outlet of the HEX can be calculated from energy conservation:

$$h_{15} = h_8 - \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_m} \times (h_{14} - h_{13}) \tag{3}$$

Where $\dot{m}_g / (\dot{m}_l + \dot{m}_m)$ is the ratio between gas and liquid mass flow which has already been determined for the receiver in eq. (2).

Mass flow in freezing / cooling cabinets

The sub-cooled CO_2 liquid coming out of the heat exchanger is separated into two streams, one is throttled to the low pressure level and fed into the freezing cabinets, the other is throttled to the intermediate pressure level and fed into the cooling cabinets. For the stream fed into the freezing cabinets, enthalpy does not change through the valve. The mass flow needed for the freezing cabinets can then be determined based on energy conservation:

$$\dot{m}_l = \frac{\dot{Q}_{freezing}}{h_{L\_freezing} - h_9} \tag{4}$$

where $\dot{Q}_{freezing}$ is the freezing demand in kW. The volumetric flow needed for the freezing cabinets can be expressed as::

$$\dot{V}_l = \frac{\dot{m}_l}{\rho_1} \tag{5}$$

The mass flow needed for the cooling cabinets can also be determined based on energy conservation:

$$\dot{m}_m = \frac{\dot{Q}_{cooling}}{h_{L\_cooling} - h_{10}} \tag{6}$$

where $\dot{Q}_{cooling}$ is the cooling demand in kW.

Low and mid-pressure compressors

Two identical piston compressors have been installed in the low pressure stage. One of the compressors is equipped with a frequency inverter which can adjust the frequency from 0 to 70 Hz. Three identical piston compressors have been installed in the intermediate pressure stage. Also in this stage one of the compressors is equipped with a frequency inverter which can adjust the frequency from 0 to 70 Hz. Important parameters for compressors are isentropic efficiency and volumetric efficiency. The isentropic efficiency has been obtained as a function of the pressure ratio between discharge and suction by fitting the performance data of the respective compressors. For instance, for the low pressure stage compressors the isentropic efficiency has been calculated using eq. (7):

$$\eta_{is} = -0.0154 \times PR^4 + 0.1941 \times PR^3 - 0.9139 \times PR^2 + 1.8744 \times PR - 0.7772 \tag{7}$$

with PR the pressure ratio. Also the volumetric efficiency of the compressors has been obtained from the performance data of the specific compressors. For the same low pressure stage compressors the volumetric efficiency has been obtained from:

$$\eta_v = +0.0005 \times PR^3 + 0.009 \times PR^2 - 0.1783 \times PR + 1.1797 \tag{8}$$

The enthalpy at the inlet of the intermediate pressure compressors has been obtained from an energy conservation balance:

$$h_3 = \frac{\dot{m}_l \times h_2 + \dot{m}_m \times h_{11} + \dot{m}_g \times h_{14}}{\dot{m}_l + \dot{m}_m + \dot{m}_g}. \tag{9}$$

Desuperheater

The method to build this model is: first give initial guess of the total heat transfer rate; apply cell method, divide the total heat transfer volume into 100 identical parts; start from the first cell, calculate heat transfer and pressure drop in cells till the last cell; get the total heat transfer rate; do iteration until these two heat transfer rates are identical. In this way the occurrence of a temperature cross is prevented. Details of the method can be found in Shi [6].

4. Model validation

The model has been validated making use of experimental data of the supermarket mentioned in section 3.1which has been modelled according to the schematic given in Fig. 1. The data used for the validation concerns the period 0:00 to 16:45 of 3-11-2015. Since most of the time the gas cooler was operating under part load conditions and the proposed model can only handle full load conditions, the experimental temperature at the outlet of the gas cooler has been taken as input to the model.

Fig. 3 shows a comparison between the experimental and model results for the cooling (left) and freezing (right) capacities as a function of time. Most of the operating conditions could be closely approached by the model.

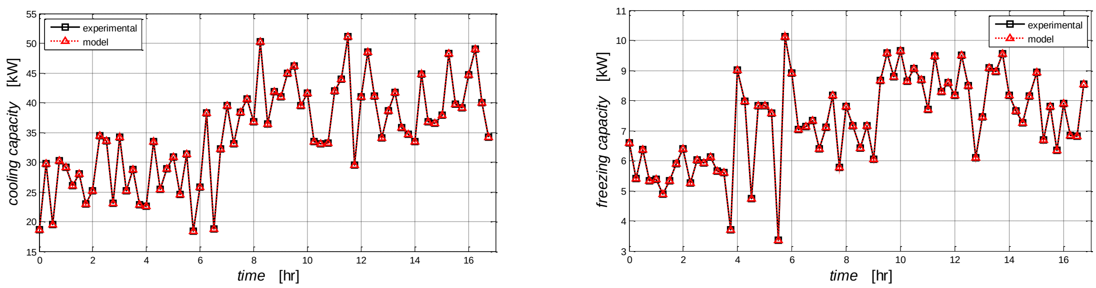

Fig. 3. Comparison between experimental and model cooling (left) and freezing (right) capacities as a function of time.

Fig. 4 shows left the recovered heat and right the temperature at the outlet of the desuperheater both as a function of time. During the night, before 6:00, the predicted temperature at the outlet of the desuperheater is significantly higher than the experimental value. During these occurrences, as can be seen on the left side of the figure, there is no heat recovered so that the desuperheater is by-passed and, in the model, the temperature becomes equal to the compressor discharge temperature. This discrepancy is caused by the quasi-steady-state approach of the model which takes time steps of 15 minutes. While in reality there is a time delay when the by-pass is switched on and off, the model directly switches from one operating condition to the operating condition 15 minutes later.

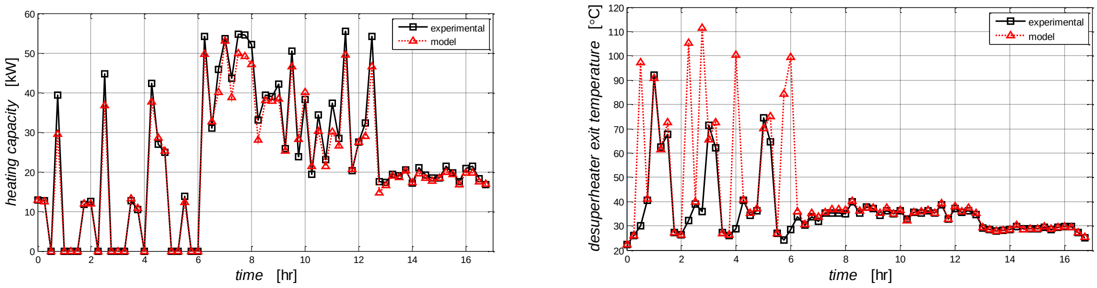

Fig. 4. Comparison between experimental and model heating capacity (left) and desuperheater outlet temperature (right) as a function of time.

Fig. 5 shows a comparison between suction (left) and discharge (right) temperature of the intermediate pressure compressors. The model generally predicts always slightly lower suction temperature while the discharge temperature can slightly be under- or over-predicted. The agreement is very much acceptable.

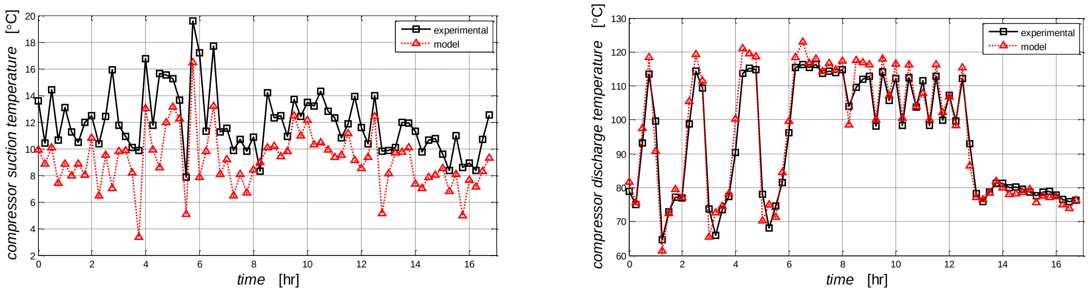

Fig. 5. Comparison between experimental and model suction (left) and discharge temperature (right) of the intermediate pressure compressors as a function of time.

5. Comparison of alternative control strategies

5.1. Operating hours and power requirements

Based on historical weather data, most of the time the Dutch ambient temperature ranges between -10 and 35ºC.The adopted relation between ambient temperature and opening hours of the supermarket is given in the left side of Fig. 6. The heating demand of the supermarket is assumed to be only a function of the ambient temperature and of the supermarket being open or closed. It has been experimentally verified that heating is not required when the ambient temperature is 13 ºC or higher. The yearly heating demand of the supermarket under consideration is 126,970 kWh. The right hand side of Fig. 6 shows the heating power demand as a function of the ambient temperature.

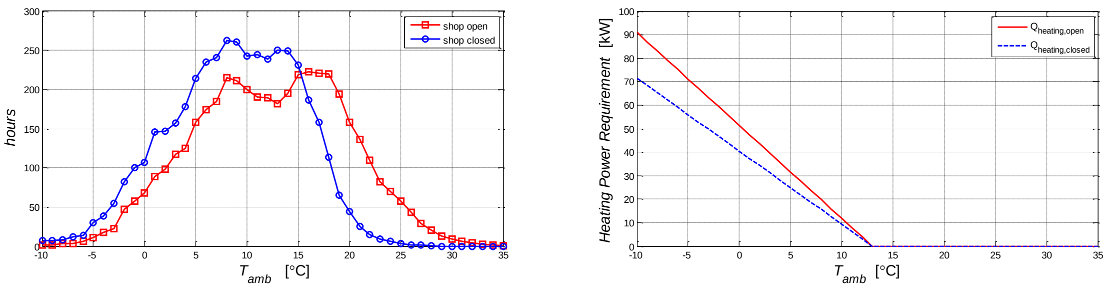

Fig. 6. Relation between opening hours and ambient temperature (left) and heating power requirement as a function of ambient temperature (right).

Also the cooling and freezing capacities are a function of the ambient temperature and of the supermarket being open or closed. The assumed demands are given in Fig. 7.

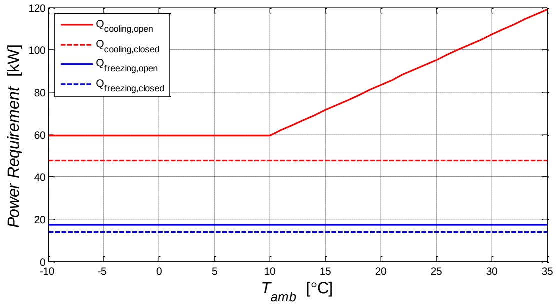

Fig. 7. Relation between opening hours, ambient temperature and cooling and freezing power requirement.

At an ambient temperature of 35 °C the refrigeration system delivers its full cooling capacity (120 kW) while at 10 °C and lower temperatures it delivers half of its capacity (60 kW). When the supermarket is closed the cooling demand is 40% of full load (48 kW).

5.2. Control with floating condensation temperature

The floating condensation temperature control meets only the cooling and freezing demand of the supermarket. The condensing pressure in this control will automatically operate at the lowest possible value, and the gas cooler operates at its full capacity to get more sub-cooling for higher COP. The amount of heat that can be recovered from this operation mode is the free recoverable heat. Fig. 8 shows left the recoverable heat as a function of the ambient temperature when the supermarket is open. The right hand side of the figure shows the corresponding COP of the supermarket plant considering only the intermediate pressure compressors (top) or the whole system (lowest values).

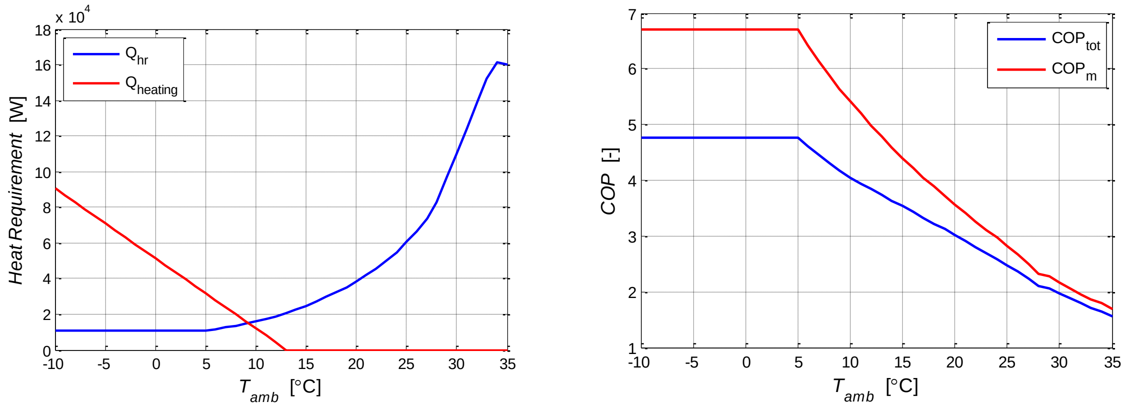

Fig. 8. Recoverable heat when the floating condensation control is used and heat requirement (left) and COP of the intermediate pressure part and of the total system (right) for the hours the supermarket is open.

Fig. 8 (left) makes clear that the recoverable heat only can cover the heating requirement between 13 and 9 °C. Lower ambient temperatures will require an additional heating system. In this temperature range the COP of the system is quite high: 4.8, as can be seen in the right side. The curves of the recoverable heat and COP show a change of slope when the system starts operating at supercritical conditions, at 29 °C ambient temperature. Since the system has been designed for a maximum temperature of 32 °C, above this temperature the intermediate pressure compressors cannot handle the cooling capacity requirement and also the recoverable heat slightly decreases. The annual consumption of the cooling and freezing plant making use of this capacity control method is 175,060 kWh.

5.3. Control for continuous heat recovery

This control method considers four situations:

1. When there is no heating demand or the heating demand is lower than the free recoverable heat, the system operates in the floating condensing temperature control method;
2. When the free recoverable heat is not sufficient, the condensing pressure is increased to obtain more recoverable heat and so to satisfy the heating demand. The gas cooler must operate at its full capacity to obtain the maximum COP;
3. When the discharge pressure is already increased to 90 bar and still the recoverable heat is not sufficient to satisfy the heating demand, the discharge pressure should be maintained at 90 bar and the gas cooler capacity should be decreased to increase the recoverable heat;
4. If the discharge pressure is maintained at 90 bar, the gas cooler is turned off and the heating demand still cannot be satisfied, then auxiliary heating devices should be applied.

The limit of 90 bar has been imposed for safety reasons and to prevent malfunction of the system. Fig. 9 shows the recoverable heat (left) and COP (right) for this control approach when the supermarket is open. Situation 1 applies down to an ambient temperature of 9 ºC; situation 2 applies between 9 and -5 ºC; situation 3 applies below -5 ºC.

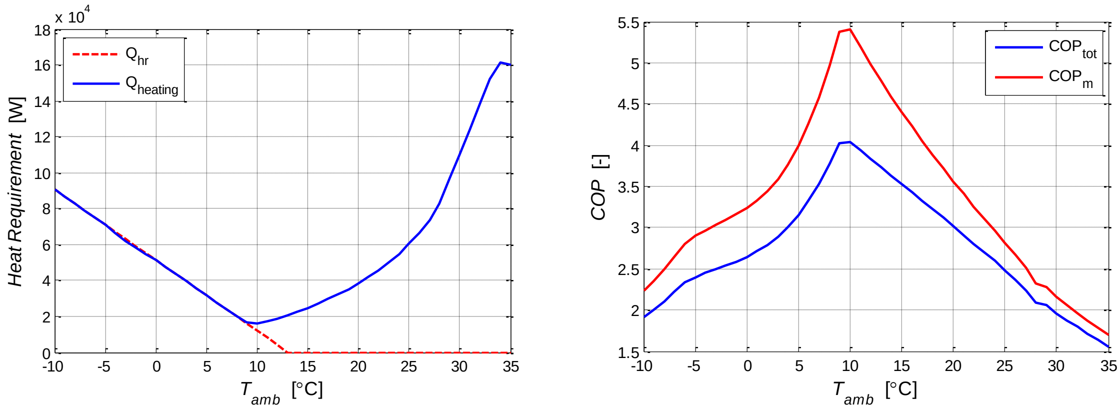

Fig. 9. Recoverable heat when the continuous heat recovery control is used and heat requirement (left) and COP of the intermediate pressure part and of the total system (right) for the hours the supermarket is open.

Fig. 9 shows that with this control approach it is always possible to deliver the heating requirement so that situation 4 does not occur. From -6 ºC the system operates at the highest pressure (90 bar). This is recognizable by the change of curvature of the COP lines in the right hand side of the figure. The COP of the system is significantly lower for this control method. The annual yearly energy consumption of the system (now including heating) is 201,342 kWh.

5.4. Comparison of the control methods

The annual heating demand of the supermarket is 126,970 kWh. It is assumed that the conventional heating method with a high efficiency boiler has a primary energy efficiency of 95% while the electrical energy conversion has a primary energy efficiency of 42%. Table 1 gives an overview of the primary energy consumption of the two alternative control systems.

Table 1. Yearly energy consumption of alternative cooling, freezing and heating methods

Parameter	Gas [kWh]	Electricity [kWh]	Primary energy [kWh]	Savings [%]
Floating condensation temperature + boiler	126,970	175,060	550,462	-
Continuous heat recovery control	-	201,342	479,386	12.9

Integration of the heating and cooling requirements in the same plant allows for about 13% primary energy savings in comparison with using separate systems for heating and cooling requirements.

6. Conclusions

This study has shown that is possible to fulfil the complete heating requirement of a supermarket in the Netherlands making use of the refrigeration system used to remove the cooling and freezing load of the supermarket.

Integration of the refrigeration and heating requirements of the supermarket may lead to primary energy savings up to 13%.

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