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Energy efficient control strategies in supermarket refrigeration systems

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ABSTRACT

The evaporation pressure in a supermarket refrigeration system exhibits constant fluctuations around its setpoint. A significant part of this behavior is influenced by the controllers and their control actions. This holds true for the individual controllers on the display cabinets, as well as controllers on the compressor rack. As a result, complex interactions between the display cases and the compressor rack arise. These complex interactions can result in higher energy demands for the compressor rack.

A simulation model, that can represent the dynamics of real display cabinets and a compressor rack including all interactions relevant for the refrigeration system, is used to investigate different control strategies for the compressor rack and the display cabinets. The different control strategies are evaluated based on their influence on the energy consumption, as well as the influence on the number of compressor start/stops. Control strategies that reduce the fluctuations in the evaporation pressure also led to lower energy consumption on the compressor rack.

Additionally, field tests were carried out in a supermarket comparable to the simulation model. It could be seen that changing the control strategy reduces the fluctuations of the evaporation pressure. As a result, also a decrease in average compressor power was observed. The first measurements indicate that changing the control strategy reduces the compressor energy demand by more than two percent.

Keywords: Refrigeration, CO2, Supermarket, Control, Efficiency, Simulation

1. INTRODUCTION

In Figure 1 the typical layout of a CO_2 supermarket refrigeration system is shown. In this system there are two temperature levels, one for refrigerated foodstuff (medium temperature/ MT) and one for frozen food (low temperature/ LT). For each pressure level there is a set of compressors. In general, more than one compressor is installed per pressure level. The compressors can either be fixed speed or variable speed compressors, providing either a fixed capacity or a variable capacity respectively. The specific arrangement of compressors is often determined by the manufacturer of the compressor rack and is a cost driven balance between the needed capacity modulation and the complexity of the configuration. From the compressors the refrigerant flows to the gas cooler (GC) placed on the outside of the supermarket building. Here the CO_2 rejects heat to the ambient air. Leaving the gas cooler, the CO_2 gets throttled by the high-pressure valve (HPV) into the receiver, where the liquid and gaseous phase separate. The gaseous CO2 is throttled back into the suction manifold of the compressors, via the flash gas bypass valve (FGB). The liquid refrigerant is fed to the display cabinets in the store room, where, in each individual display cabinet, an expansion valve throttles the CO_2 down to the evaporation pressure. From the expansion valve the CO_2 passes through the evaporators of the display cases, where it evaporates and then flows back into the suction manifold of the compressors.

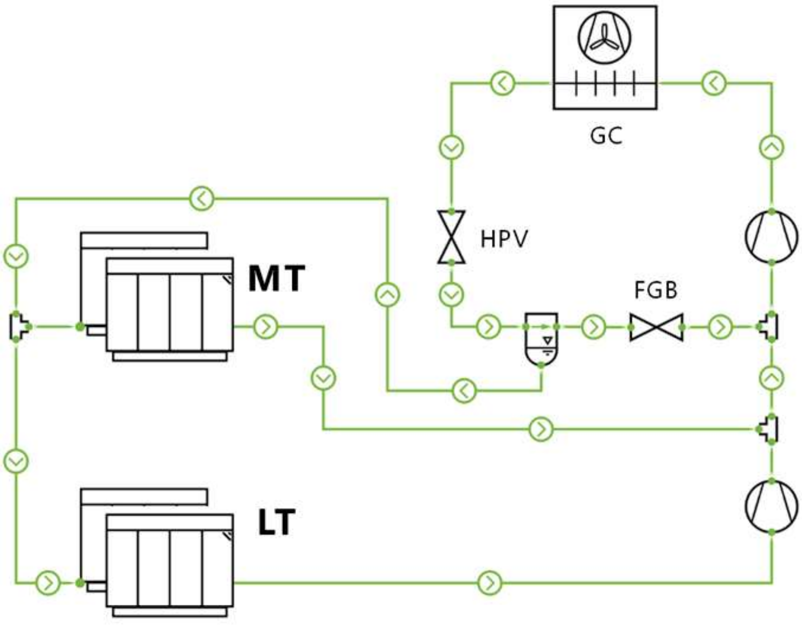

Figure 1 Typical layout of a CO2 refrigeration system

When operating a supermarket system typically the level of the suction pressure and the number of compressor start/stops are monitored. The suction pressure level is important for the power consumption of the compressors. The number of start/stops gives some indication of the stability of the control and the lifetime for the compressors. Typically, the evaporation pressure in a supermarket refrigeration system is constantly fluctuating. It is not fully understood how this kind of fluctuations impacts power consumption.

The fluctuations can have either an exogenous or endogenous source. The exogenous disturbances are coming from ambient temperature/humidity changes, customers visiting the store, goods being loaded/un-loaded in the cold rooms etc. Investigating data and simulation models it can be found that most of the exogenous disturbances results in a slow dynamic impact on the system load. When customers do their shopping and, for example, open glass lids in the cabinets this leads to an immediate increase in infiltration of air and humidity into the refrigerated cabinet. This infiltration, however, eventually only leads to a very lowpass filtered change in the cabinet load as the thermal capacity of goods and the metal in the cabinet absorbs immediate changes. In contrast, loading of cold rooms (leaving the door open) does lead to an immediate change in the load that can be observed in the compressor capacity, however these events are isolated to distinct times a day and hence typically do not affect the overall stability of the system. All in all, this leads to the conclusion that the in-stable operation of a supermarket system mainly is driven by endogenous disturbances. Endogenous disturbances are self-inflicted and are mainly caused by various un-coordinated and frequent “events” in the control system such cut-in/cut-out of refrigerated cabinets and compressors. These events can lead to chaotic behaviors such as e.g., synchronization of cabinets as described in Larsen et al., 2007b.

This paper investigates the effect of combination of various typical used control strategies in supermarkets refrigeration systems.

2. MODELING

Simulation models form a good method to investigate the endogenous disturbances in the refrigeration system and can aid in evaluating different control settings and system setups. To describe the dynamic behavior correctly a simulation model with a high level of detail is required. To this end an existing simulation model of a supermarket (Schulte et al., 2023) is used.

In this simulation model the relevant parameters for the display cabinets were identified from monitoring data for a real supermarket. The parameters for the compressor models were identified from manufacturer data. The simulation model for the refrigeration cycle is built using the programming language Modelica (Modelica, 2021) using the TIL-Library (TLK-Thermo, 2022). For the pack controller a Simulink model of the Danfoss pack controller is used. The setup of the simulation model and its validation was described in Schulte et al., 2023.

To describe these dynamic interactions, the display cabinets are modelled individually, as well as the compressor rack with its individual compressors. This model is designed to represent the dynamics on the refrigerant side of the cabinets in sufficient detail, while making some simplifications on the air side of the display cabinet. Thus, the air side is represented by two thermal capacities, one for the air and one for the goods, while the evaporator is modelled by a finite volume approach, similar to Schulze, 2012. The specific model structure, as well as the parameters were described in Schulte et al., 2022. Comparable versions of this modelling approach for display cabinets are also shown in (Larsen et al., 2007a), (Larsen et al., 2007b), (Leerbeck et al., 2021), (Leerbeck et at., 2022) and (Shafiei et al., 2013).

The individual compressors of the compressor rack are modelled using physics-based models from the TIL-Library (TLK-Thermo, 2022). Their parameters are fitted to steady state manufacturer data using a Levenberg-Marquardt-Algorithm. The dynamic thermal behavior is considered by internal thermal heat capacities. In this model of a CO_2 supermarket refrigeration system two compressors are operating in parallel on each stage. The compressors on the LT stage are subcritical and on the MT stage transcritical electric driven reciprocating compressors. The designation transcritical indicates that the MT compressors operate in this model only when required by the ambient temperature, thus in a typical Danish climate most operating hours will be in subcritical model. In the MT stage one of the compressors is speed controlled and the others are fixed speed as are the LT stage all compressors. Fixed speed compressors are controlled by an on/off control.

The model was validated in Schulte et al., 2023, using data from the ESO-2 Project (Fredslund, 2013). Overall good agreement between measurement data and the simulation model is achieved. Especially the main interactions between the refrigerated cabinets and compressor rack are well represented. This also extends to the chosen control actions of the respective controllers.

2.1. Controllers

Multiple variables in the refrigeration system are controlled by electronic controllers. Two main types of controllers are responsible for the operation of the refrigeration system, the pack controller and the cabinet controller.

The pack controller is responsible for controlling the evaporation pressures, as well as the receiver pressure, the gas cooler outlet temperature and the high pressure between the MT compressors and the HPV. To control the evaporation pressure the pack controller can use the compressor capacity. The ability to match the required compressor capacity with the engaged compressor capacity depends on the system configuration. A typical configuration is to have a variable speed drive on the first (the lead) compressor (VarSpd) and fixed speed compressors on the following steps. The range of the first compressor is usually designed such that it can cover the gap between the capacity steps from the fixed speed compressors, such that a smooth capacity can be reached from low to high capacity. This way the capacity can exactly be matched with the load. Apart from this ideal situation there are also configurations possible where the variable speed compressor cannot cover the full gap of the capacity steps from the fixed speed compressors. Even more so, there are also configurations running entirely with fixed speed compressors giving rise to clear steps/gaps in the available capacity (Stp). This scenario leads to an imbalance between the load and available capacity, resulting in a tradeoff between the precision of the control and the number of compressor start/stops.

The cabinet controllers are responsible for maintaining the temperature in the display cabinets, as well as ensuring sufficient superheat at the outlet of the evaporator. For that the controller can use the opening degree of the expansion valve. The temperature control of the cabinets can principally be done in two different ways; either thermostatic control (Ther) or modulating thermostat control (MTR). In thermostatic control mode the temperature is simply controlled in an ON/OFF manner using a hysteresis control starting and stopping the injection when the temperature reaches respectively the cut-in and cut-out temperature value. This leads to periodic loads from the cabinets depending on if the cabinet is in a state of cut-in or cut-out. In MTR control mode the injection valve is controlled in a continuous manner to maintain the temperature at a reference temperature in the middle between the cut-in and cut-out. At any time, the controller ensures that a minimum superheat is maintained at the evaporator outlet.

3. RESULTS

To evaluate the yearly impact on power consumption (of the compressors) and the number of compressor start/stops, 24h simulations were conducted at constant ambient temperatures. The range of temperature intervals covers ambient temperatures ranging from -10 to +35 °C. The temperature intervals have been weighted with a temperature distribution corresponding to a typical Danish climate.

Table 1 shows the different investigated control modes for the compressors, as well as the cabinet controllers. The ideal variable speed compressor control was investigated to understand the impact of having a completely continuous control. This was done with an idealized variable speed compressor, being capable of covering the whole range from zero to maximum capacity without incurring the typical efficiency losses of a compressor operating at part load. The VarStp version is a very typical configuration for compressor racks, with the first compressor being speed controlled and the following compressors being fixed speed. The Stp configurations only consists of fixed speed compressors.

Table 1 Control modes of the compressor and cabinets

	Compressor Control Mode		Cabinet Control Mode
IdVar	Ideal variable speed compressor capacity control	**Ther**	Thermostatic mode
VarStp	One Variable speed and step compressor capacity control	**MTR**	Modulating temperature control
Stp	Only step/fixed speed compressor capacity control		

The original system consisted of two suction pressure level groups MT and LT. The MT group consists of one variable speed and one fixed speed compressor and seven evaporators. The Variable speed compressor is designed such that at maximum speed it corresponds to approximately 80% of the second step such that there is only a small capacity gap between the first and the second compressor capacity rendering an almost smooth variable control capacity from min speed of the 1st compressor to max capacity. The LT suction group consists of two fixed speed compressors and four evaporators. The LT group can hence be considering step capacity control whereas the MT is a variable capacity controlled, though not ideal, hence included efficiency losses during part load operation and start/stop of the first compressor. The system is operated with a fixed suction pressure reference day and night throughout the year for both MT (-8°C) and LT(-30°C).

Figure 2 depicts the annual weighted coefficient of performance (COP) for the LT suction group, versus the average start/stop of the first compressor of the LT suction group. As can be seen the best efficiency is attained using an ideal variable speed compressor. However, it can also be seen that the yearly COP falls from 1.97 to 1.92 when changing the control strategy for the cabinets from MTR to thermostatic control. The main reason for this drop is the cabinets cutting in and out, causing variations of the suction pressure around the reference. Therefore, there is, on average, a higher compressor capacity at pressures lower than the reference. This leads to a yearly ~2.5% lower efficiency.

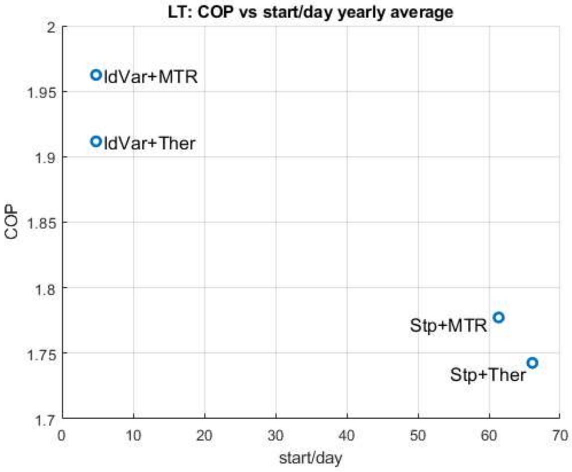

Figure 2 Low temperature compressors COP vs starts per day

The combination of fixed speed compressors (Stp) and Thermostatic control (Ther) shows the most significant drop in efficiency compared to the ideal scenarios. The main reason for this is due to the many operating hours at part load especially during nighttime where the first compressor step (the smallest) is too big compared to the load. The so called "pump-down" limit prevents the last compressor step from stopping before a certain lower pressure level (the pump-down limit) has been reached. This is to avoid frequent compressor cycling. The cost of this is a prolonged operation at a suction pressure below the setpoint, leading to an overall increase in energy consumption. Switching the control strategy to MTR does not fix the main issue, that the compressor capacity of the first compressor is too big at nighttime. However, it slightly improves the operation during daytime, leading to a small reduction in the compressor start/stops.

Figure 3 depict the results for the MT suction group, that is running with one variable speed compressor and one fixed speed compressor. Running the cabinets in MTR mode versus thermostatic mode significantly improves both COP and number of start/stops. In MTR mode the model results show a much more stable load on the compressors hence allowing the controller to better maintain the suction pressure, without frequently starting and stopping compressors.

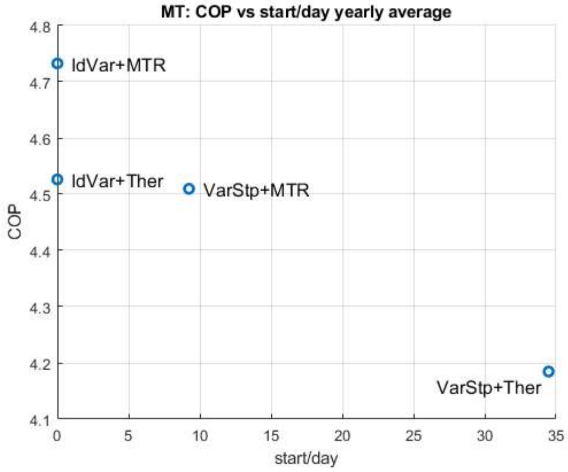

Figure 3 Medium temperature compressors COP vs starts per day

As the average load in MTR mode, at the most the relevant ambient conditions (8°C in Denmark), is enough to keep the variable speed compressor running close to or slightly above rated speed, then operation in part load conditions becomes minimal. On the other hand, running the cabinets in thermostatic mode causes fluctuating load on the compressors which means that even at minimum speed the compressor gets close to or hits the pump down level resulting in prolonged operation in part load at a lower suction pressure. This results in significantly lower efficiency compared to the thermostatic mode for the MT compressors.

4. EXPERIMENTAL INVESTIGATION

Field trials have been carried out after remodeling has taken place of the originally simulated supermarket. Due to the remodeling the supermarket now has a slightly different system setup. Especially the first LT compressor now has a variable speed drive. Furthermore, the store now runs evaporation pressure optimization (p0-Optimization), instead of fixed setpoints for the evaporation pressure. Nevertheless, the store still has a similar size and is comparable in its characteristics to the previous version of the store investigated in the simulation model. For the purpose of this paper the evaporation pressure and the corresponding saturated temperature will be used synonymously.

Figure 4 shows the comparison of the evaporation pressures with thermostatic mode and MTR mode. It can clearly be seen that the fluctuations (indicated by the dashed lines) in the evaporation pressure are reduced by about half. This is true, both during day and night. The evaporation pressure on the MT suction group stays relatively constant over the ambient temperature. Between thermostatic model and MTR mode no significant difference in evaporation pressure could be observed.

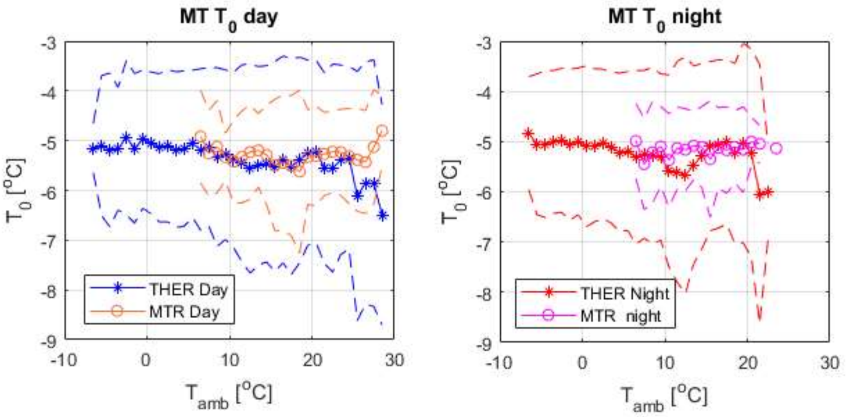

Figure 4 MT evaporation pressure fluctuation for thermostatic and MTR mode. Solid lines show the mean values and dashed line show the standard deviation.

In Figure 5 the comparison of the LT evaporation pressure is shown. As with the MT evaporation pressure, using MTR mode on the LT display cabinet lead to a reduction in evaporation pressure fluctuations. Compared to the MT evaporation pressure, the LT evaporation pressure decreases with ambient temperature. This is to compensate for increased load at higher ambient temperatures.

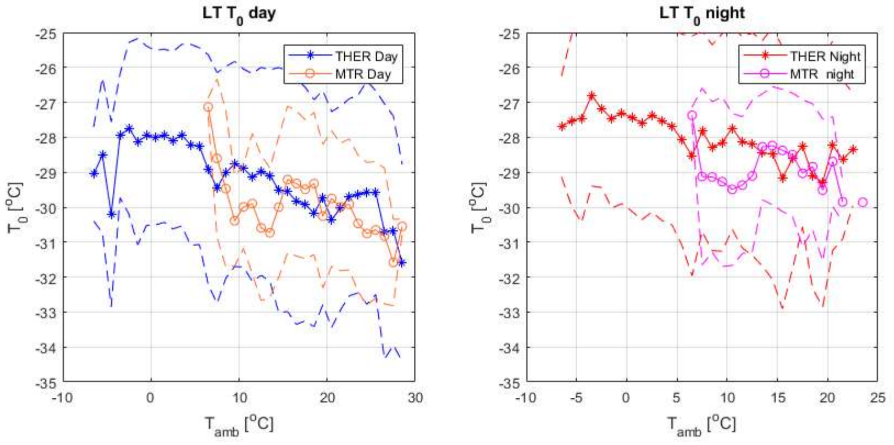

Figure 5 LT evaporation pressure mean and fluctuations for thermostatic and MTR mode. Solid lines show the mean values and dashed line show the standard deviation.

The remaining fluctuations for both the MT and LT evaporation pressure may very well be the result of the neutral zone on the compressor controllers. The neutral zone reduces the controller actions around the setpoint, to prevent excessive controller actions and oscillations. Overall the results qualitatively agree well with the expectations from the simulation model in Schulte et al., 2023.

To evaluate the influence of the reduced evaporation pressure fluctuations on the compressor power demand, Figure 6 shows the electrical power of the compressor rack over the ambient temperature. It can be seen that the electrical power on average is lower, when the display cabinets run in MTR mode. On average a reduction of approximately two percent can be observed in the compressor rack power. The reduction is relatively constant for all observed temperatures. Therefore, it can be extrapolated, that the annual energy demand for the refrigeration system decreases by approximately two percent.

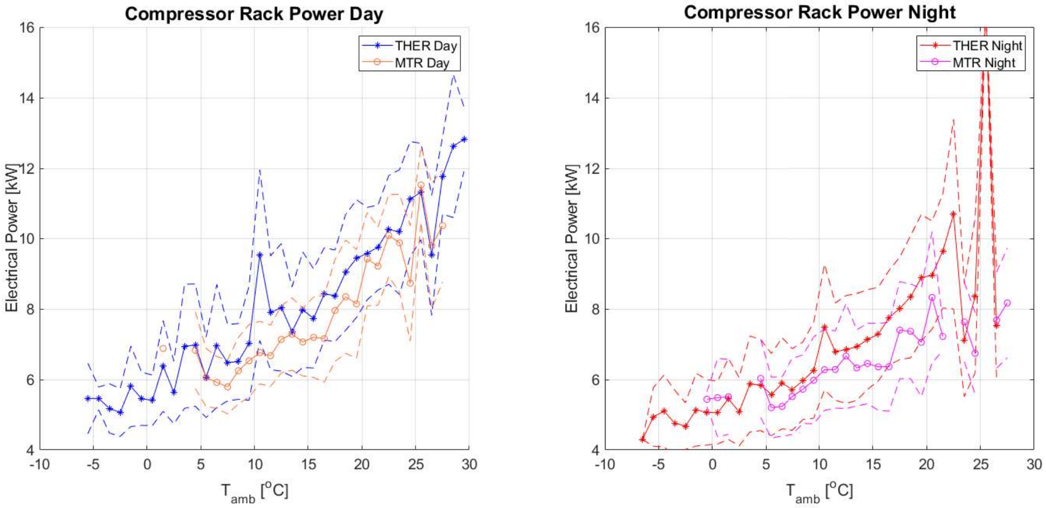

Figure 6 Comparison of compressor rack power for thermostatic and MTR mode. Solid lines show the mean values and dashed line show the standard deviation.

5. CONCLUSIONS

The simulation indicates that when a system is equipped with variable speed compressors there is a benefit in running the cabinets in MTR mode to ensure a more stable load. An important prerequisite is however a well-balanced compressor selection that ensures the variable speed compressor has most operating hours around the rated speed. If only a limited number of fixed speed compressors are used the control strategy on the cabinets does not significantly impact the system performance both in terms of COP and compressor start/stop.

Experimental results from a field test qualitatively confirm the results from the simulation model. A clear decrease in fluctuation of the evaporation pressure can be observed. This also reinforces the hypothesis, that the fluctuations mainly stem from controller behavior. The decrease in evaporation pressure fluctuations goes along with an observable decrease in compressor power, qualitatively verifying the model results.

Further investigations are still ongoing, to investigate the possibility to increase the upper limit of the evaporation pressure optimization. As it is suspected that a more stable evaporation pressure allows for a further increase in evaporation temperature. Also, investigations into the effect of fluctuations on heat recovery of CO_2 compressor racks is ongoing. It is presumed that the fluctuation in compressor capacity also negatively affects the performance of the heat recovery, as disturbances from the capacity controller get transferred into the heat recovery controllers.

ACKNOWLEDGEMENTS

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NOMENCLATURE

T	temperature (K)	*VarStp*	Variable Speed and Fixed Speed Compressor
IdVar	Ideal Variable Speed Compressor	*Ther*	Thermostatic Control Mode
Stp	Fixed Speed Compressor	*COP*	Coefficient of Performance
MTR	Modulating temperature regulation	*FGB*	Flash Gas Bypass Valve
GC	Gas Cooler	*MT*	Medium Temperature
HPV	High pressure valve	*amb*	Ambient

LT Low Temperature

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