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The influence of evaporation pressure dynamics on energy consumption

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ABSTRACT

The evaporation pressure in a supermarket refrigeration system constantly fluctuates around its setpoint. Refrigerated display cases play a major role in driving the dynamic behaviour of the evaporation pressure. A significant part of this dynamic is influenced by the controllers and their control actions. This holds true for the individual controllers on the display cabinets, as well as the controller on the compressor rack. The result of that are complex interactions between the display cases and the compressor rack. It is suspected, that these fluctuations negatively influence the energy demand of the refrigeration system.

In this paper, a dynamic model of a supermarket is presented and validated against real operating data. The model is able to represent the dynamics of real display cabinets and a compressor rack including all interactions relevant for the refrigeration system. Based on the model an investigation into the influence of evaporation pressure fluctuations and their impact on the energy demand is performed.

Keywords: Supermarket, Refrigeration, Carbon Dioxide, Display Cases, Control

1. INTRODUCTION

In supermarket refrigeration systems the evaporation pressure is constantly fluctuating. This fluctuation is mainly driven by the control behaviour of the display cabinets in the store. As a result, the capacity controller on the compressor rack need to constantly compensate for the fluctuations. It is suspected that these fluctuations contribute to higher energy consumption on the compressor rack.

Simulation models form a good method to investigate this behaviour and can aid in developing more sophisticated control algorithms. In order to investigate the control algorithms in a supermarket refrigeration system, a high level of detail is required to capture the dynamic behaviour correctly. This is because the refrigerant circuit of a supermarket refrigeration system is characterised by complex interactions between the display cabinets and the compressor rack. Fluctuations in the evaporation pressure influence the controllers of the display cabinet, mainly the superheat control in the display cabinets. On the other hand, the opening and closing of the expansions valves results in rising and falling pressures in the suction line. As a result, the pack controller reacts to the change in the suction pressure by increasing or decreasing the engaged compressor capacity.

To capture these dynamic interactions, a model has been built where 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. To identify the relevant parameter for the simulation model monitoring data for a real supermarket was used. Furthermore, for the individual compressors of the compressor rack physics-based models are used to describe their performance characteristics. The parameter for the compressor models where identified from manufacturer data. The simulation model for the refrigeration cycle is built in *Modelica* (Modelica, 2021) using the *TIL-Library* (TLK-Thermo, 2022). For the pack controller a Simulink model of the Danfoss pack controller is used.

In a first step the model is validated against monitoring data, combined with additional measurements for the refrigeration system, e.g. mass flow meters installed at the supermarket. For that not only the average values, but also the fluctuations are compared. Beyond that, the model is also visually compared against the monitoring data for an exemplary day.

The simulation model is then used to evaluate the influence of fluctuating evaporation pressures on the energy consumption. This is done by comparing the compressor power of the supermarket in its original configuration against a variant that is enabled to maintain a more stable evaporation pressure.

2. MODELING

In Figure 1 the typical layout of a CO_2 supermarket refrigeration system is shown. In this system there are two evaporation pressure levels, one for the medium temperature (MT) display cabinets and one for the low temperature (LT) display cabinets. 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

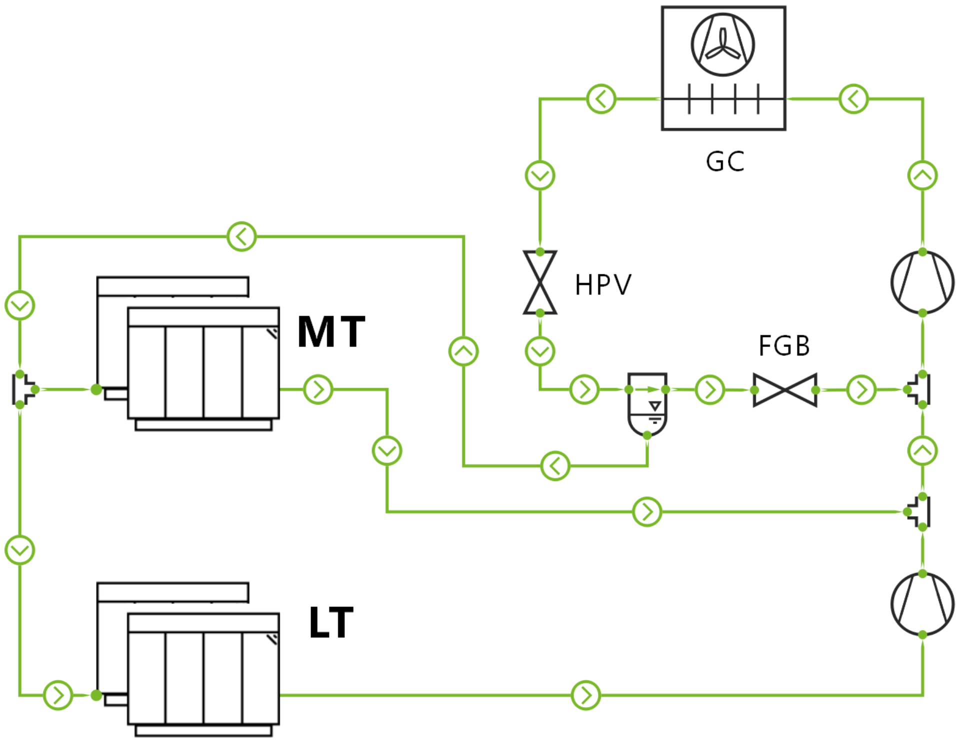

Figure 1: Principle Flow Chart for a so called transcritical CO_2 Booster System used in supermarket refrigeration.

variable speed compressors, providing either a fixed capacity or a variable capacity respectively. The specific arrangement of compressors is determined by the manufacturer of the compressor rack. From the compressors the refrigerant flows to the gas cooler (GC) on the outside of the supermarket. 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 CO_2 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 expansion valve there 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.

2.1. Display cabinet

Display cabinets have been described in different detail levels in literature. Typically, the level of detail is a result of the intended purpose of the research. Many Models focus on the air flow inside the cabinet. Among them are different varieties of CFD simulations as for example in (Chaomuang et al., 2017), (Foster et al, 2005) and (Wang et al, 2021). In a similar manner multiple authors model the temperature distribution inside the cabinet for example in (Chaomuang et al., 2019), (Chaomuang et al., 2020) and (Ben-abdallah et al, 2018). For control-oriented investigations of the whole supermarket refrigeration system these models are usually too complex to parametrise and computationally too expensive for system simulations.

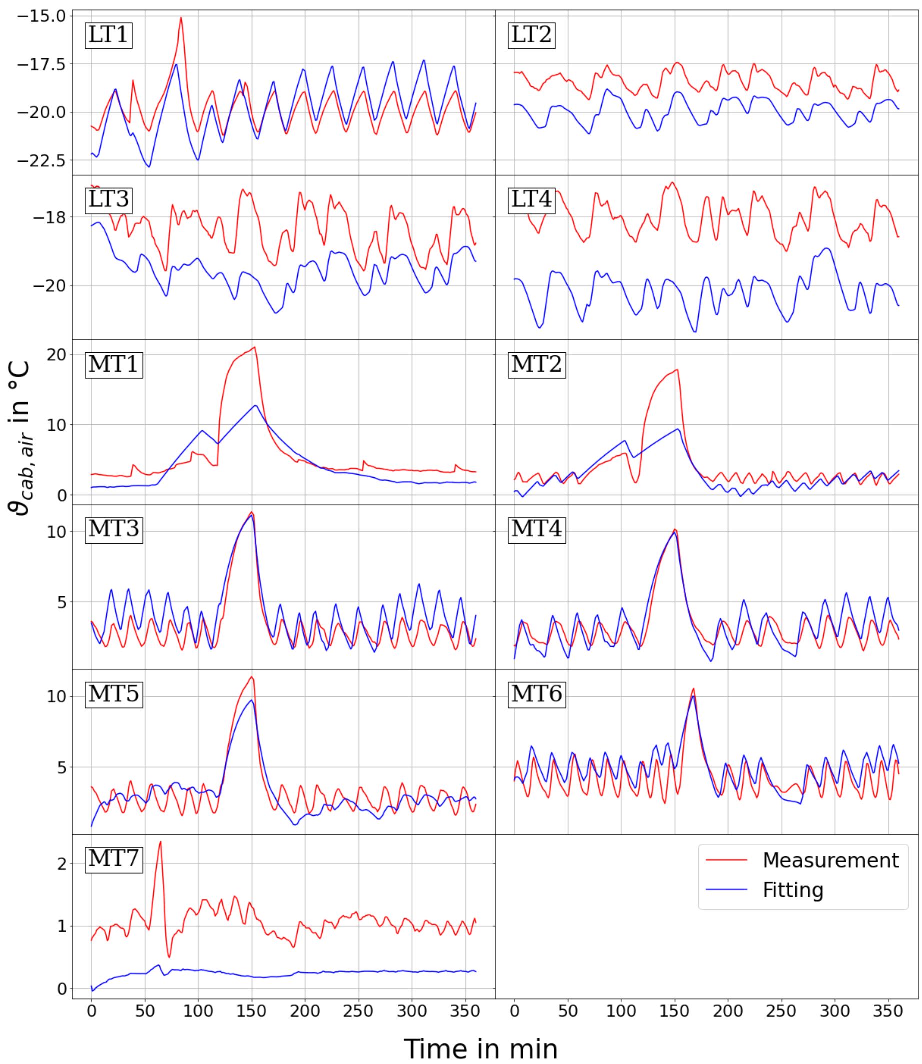

Figure 2 Fitting results for the individual display cabinets in the supermarket

For investigating the refrigerant side of a supermarket refrigeration system, it is important to model the correct behaviour of the refrigerant side. This means the focus of the display cabinet model lays on the refrigerant side and not on the air side. Therefore, the air side of the display cabinet is not modelled in great detail.

Models using a simplified description of the air-side are more appropriate for this purpose. The display cabinet model used in this work uses a thermal network approach for the air side. This approach has been described by the authors in a variety of version in different publications for example (Larsen et al., 2007a), (Larsen et al. 2007b), (Leerbeck et al., 2021), (Leerbeck et at., 2022) and (Schulte et al., 2022), as well as by other authors in literature (Shafiei et al., 2013). The specific version used here is (Schulte et al., 2022). Here the air side is represented by two thermal capacities, one for the air and one for the goods. For the evaporator a finite volume approach, similar to (Schulze, 2012), is used. The refrigeration load on the cabinet and the relevant parameter for the cabinet, e.g. the heat capacity of the goods and the air, have been identified from monitoring data. The data needed for the calibration of the display cabinet models is taken from the monitoring system of a supermarket in Denmark. The data was collected between 2012 and 2016. The valve areas for the individual cabinets are taken from (Fredslund, 2013). Figure 2 shows the results of the parameter identification, as they relate to the air temperature of the display cabinet. It can be seen, that the simulation models match the observed dynamics from the monitoring data reasonably well.

2.2. Compressors

For the CO_2 supermarket refrigeration system two or more compressors are operating in parallel on each stage connected with LT and MT cabinets. The compressors on the LT stage are subcritical and on the MT stage transcritical electric driven reciprocating compressors. In each stage one of the parallel compressors is speed controlled and the others are run at constant speed with on/off control.

For the simulation model the physics-based model approach for semi-hermetic reciprocating compressors is used (Schedel et al, 2013). Figure 3 describes the main model structure of the dynamic physics-based compressor model with isentropic compression between suction and discharge volume (1) taking main loss effects into account due to valve flow (2), leakage flow (3), friction + electric motor losses (4), re-expansion losses (5) heat transfer to the ambient (6) and considering internal heat capacities (7). The compressor models are designed in that way in which all losses are scalable with the displacement. The parameter relative displacement can be used to change the overall compressor size with similar efficiencies.

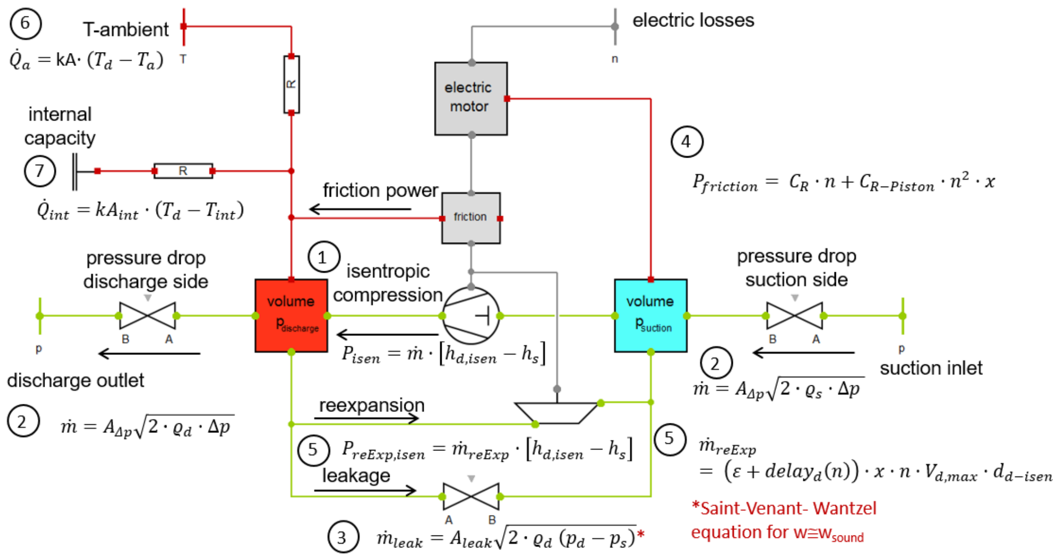

Figure 3 Physical based compressor model with isentropic compression between suction and discharge volume (1) taking main loss effects into account due to valve flow (2), leakage flow (3), friction + electric motor losses (4), re-expansion losses (5) heat transfer to the ambient (6), and considering internal heat capacities (7) (similar to (Schedel et al., 2013)).

The individual compressors of the compressor rack are modelled using physics-based models. Their parameters are fitted to steady state manufacturer data using Levenberg-Marquardt-Algorithm. The dynamic thermal behaviour is considered by internal thermal heat capacities.

2.3. 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 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. For this the pack controller can use the engaged compressor capacity to control the evaporation pressure of the respective pressure level. Furthermore, the gas cooler outlet temperature and the high pressure are controlled by the gas cooler fans and the opening degree of the HPV. The receiver pressure is controlled by the opening degree of the FGB. For the simulation of the pack controller a Simulink model of the real controller was used.

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 to control the superheat at the outlet of the evaporator. The temperature control can operate in two modes, either thermostatic or modulating (Danfoss, 2018). In Thermostatic mode the controller begins opening the expansion valve at a “cut-in” temperature and continues until a “cut-out” temperature is reached, where the expansion valve is closed. In modulating mode, the controller uses the expansion valve to maintain a constant temperature in the cabinet, while simultaneously ensuring a minimum superheat is maintained.

3. INVESTIGATION

Before using the model for investigations, it has to be validated. For this the model is validated with data from the monitoring systems, as well as additional sensors installed in the supermarket. The data is from a supermarket in Otterup, Denmark and was collected in a previous project ESO-2 (Fredslund, 2013). The collected data spans from 2012 till 2016 and contains measurements in a one-minute resolution. During this time additional measurements from mass flow meters for MT and LT suction mass flow, as well es separate power meters for the MT and LT compressors are available. The data was split into day and night time operation, to account for an open and closed store.

3.1. Validation

The first and primary validation is on the summarised refrigeration load of the display cabinets. For this the measurements of a mass flow meter in the suction line, the pressure in the receiver and the pressure and temperature in the suction line were used to calculate the cumulated cooling load of display cabinets. The mass flow meter provides an independent source of truth for the refrigerant mass flow in the system. Furthermore, the simulated power of the compressors was compared to the actual power measured by power meters. Here individual power meters of the MT and LT compressors where installed.

Figure 4 shows the comparison of the refrigeration load and the compressor power. For the MT refrigeration load of the supermarket a good agreement between the model and the data can be observed, with only slight deviations. While the LT refrigeration load of the model also generally agrees with the data, it can be seen, that the use of a constant refrigeration leads to some difference at the high and low ambient temperatures. For the compressor power both the MT and LT compressor power show good agreement with the data. This is both in absolute values as well as for the fluctuations.

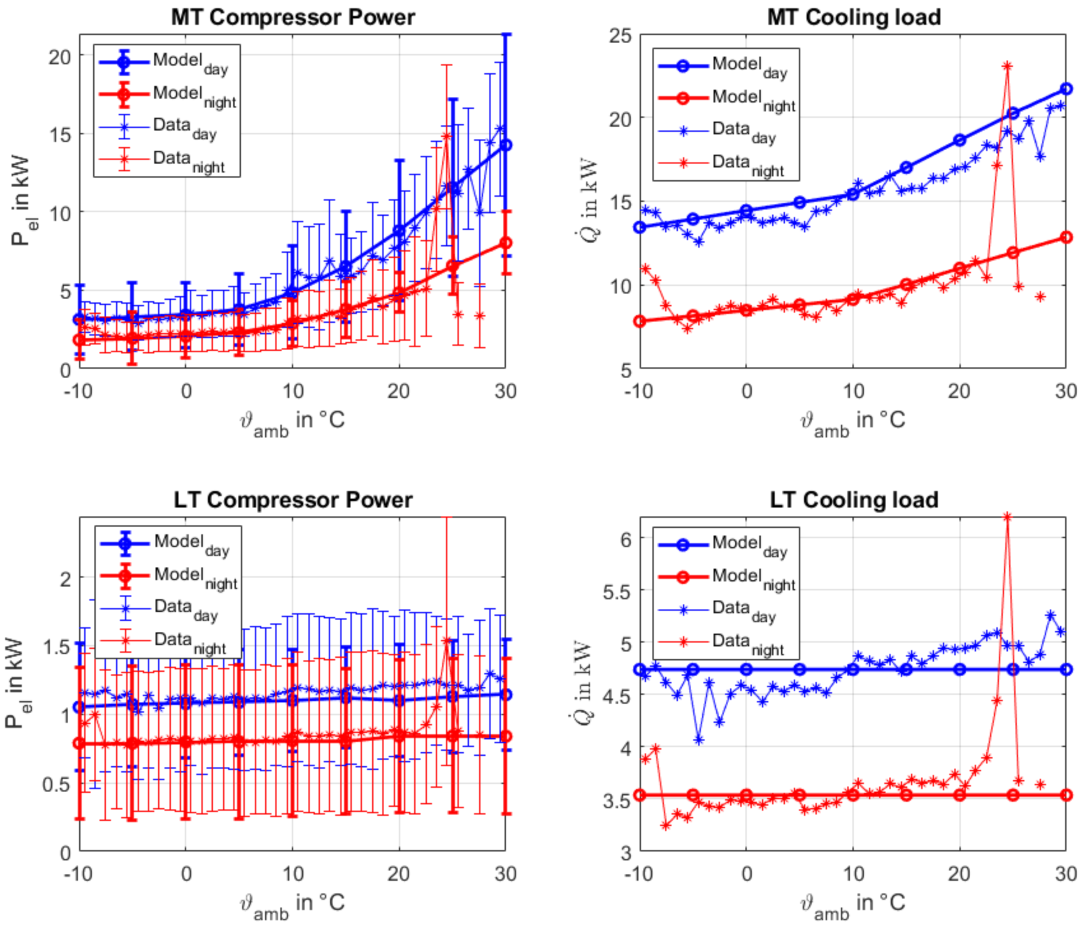

Figure 4 Comparison of measured and simulated compressor power and cooling load, for both MT and LT at day and night. Shown is both the average value and the fluctuations.

In Figure 5 the comparison of the compressor capacity and the suction pressure is shown. For the compressor capacity good agreement is seen both on the MT and LT compressors. For the suction pressure the pressures agree very well with the data, as the pressure in the model is controlled and the setpoint is selected according to the data. Only for the MT pressure at night time the pressure in the model is slightly higher at higher ambient temperatures. The variation in the pressure is matched good as well, in that the fluctuations of the pressure agree reasonably well the fluctuation that where observed in the data.

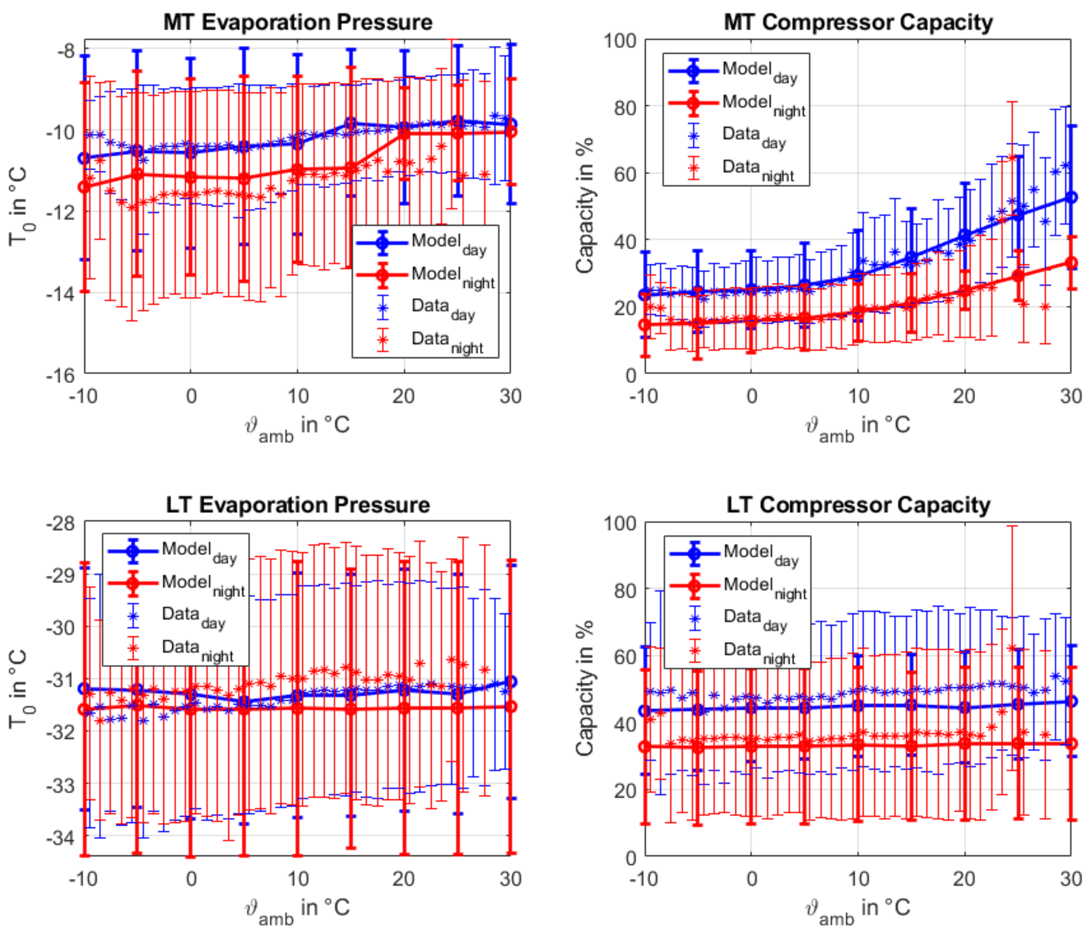

Figure 5 Comparison of measured and simulated evaporation pressure and compressor capacity, for both MT and LT at day and night. Shown is both the average value and the fluctuations.

In order to evaluate the correct modelling of the pressure variations in the refrigeration system an exemplary time dependent data set for a complete day is used to visually compare the dynamics of the evaporation pressure, as well as the running compressor capacity, against the simulation results at the same ambient temperature. Figure 6 shows the comparison of simulation and monitoring data for the day time operation of the supermarket, while Figure 7 shows the night time operation of the supermarket. It can be seen that the general behaviour of the evaporation pressure and the compressor capacity is comparable between the simulation model and the data. While for the MT level the evaporation pressure dynamics are slightly faster than in the monitoring data, the dynamics of the compressor capacity still match the monitoring data quite well. It has to be kept in mind, that the simulation model is deterministic, while the data also contains stochastic influence, as for example costumer behaviour.

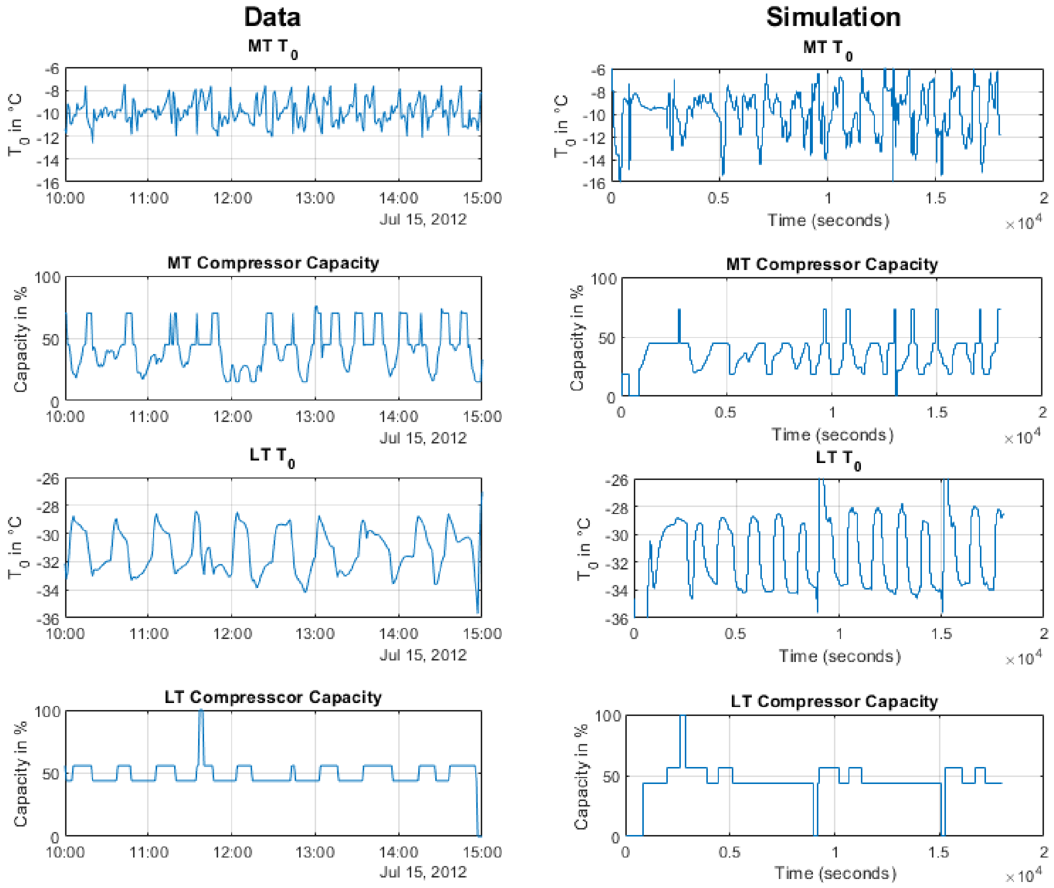

Figure 6 Comparison of the evaporation pressure and compressor capacity for an exemplary day, against simulation results, at day time operation

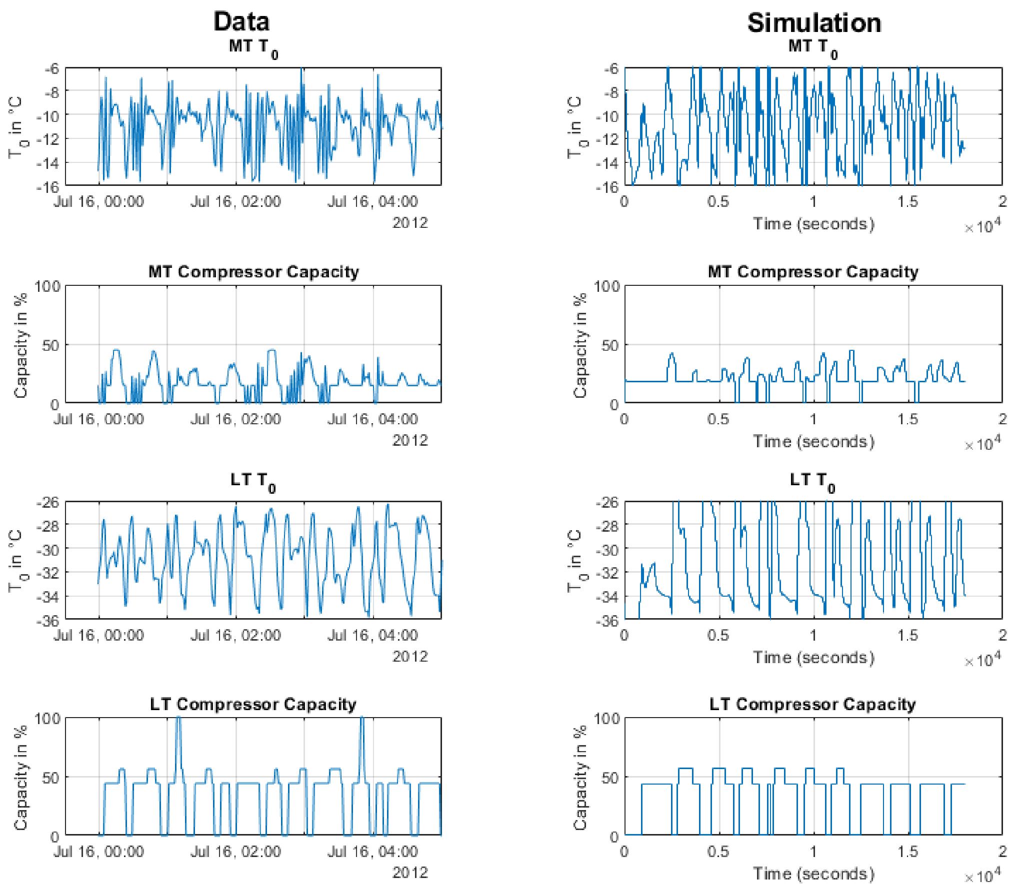

Figure 7 Comparison of the evaporation pressure and compressor capacity for an exemplary day, against simulation results, at night time operation

In Figure 8 the receiver pressure, as well as the gas cooler pressure and the gas cooler outlet temperature are compared. Both gas cooler pressure and outlet temperature are in very good agreement with the measured data. The same holds for the receiver pressure, where also the fluctuations of the receiver pressure match reasonably well with the data.

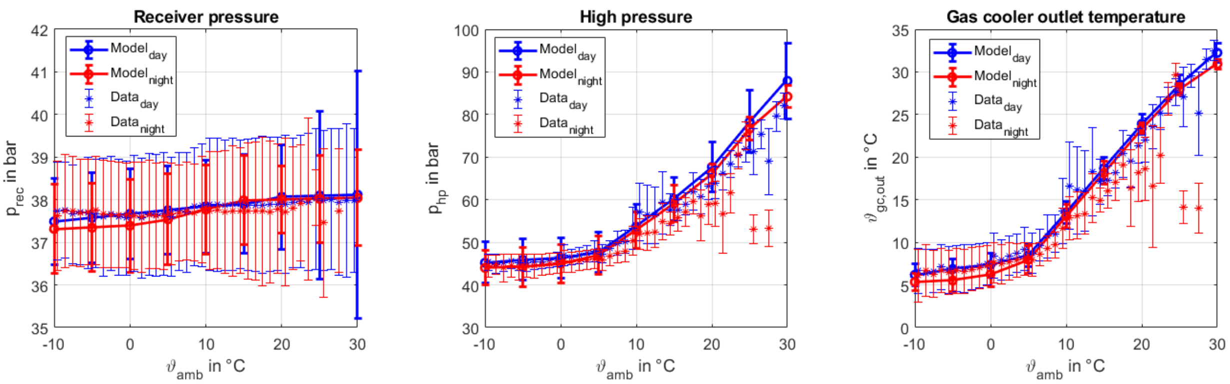

Figure 8 Comparison of measurement data against simulation results for receiver pressure, high pressure and gas cooler outlet temperature. Shown is both the average value and the fluctuations.

Finally, in Figure 9 the dynamic behaviour of the display cabinets is compared. As can be seen, the MT cabinets in the simulations generally match the behaviour of the cabinets from the data. The main difference is, that in the monitoring data, multiple cabinets perform a defrost at 10 o'clock. This defrost behaviour is currently not implemented in the simulation model, therefore no defrost can be observed. For the LT cabinets the behaviour of the simulation models also agrees well with data. Especially three of the cabinets are “floating” and don’t reach their cut-out temperature. Furthermore, for the simulation models it can be observed, that the cabinets seem to enter a limit cycle, resulting in complex periodic behaviour. Due to the deterministic behaviour of the simulation model this phenomenon can be observed, while in the monitoring data stochastic influences make it hard to identify these limit cycles.

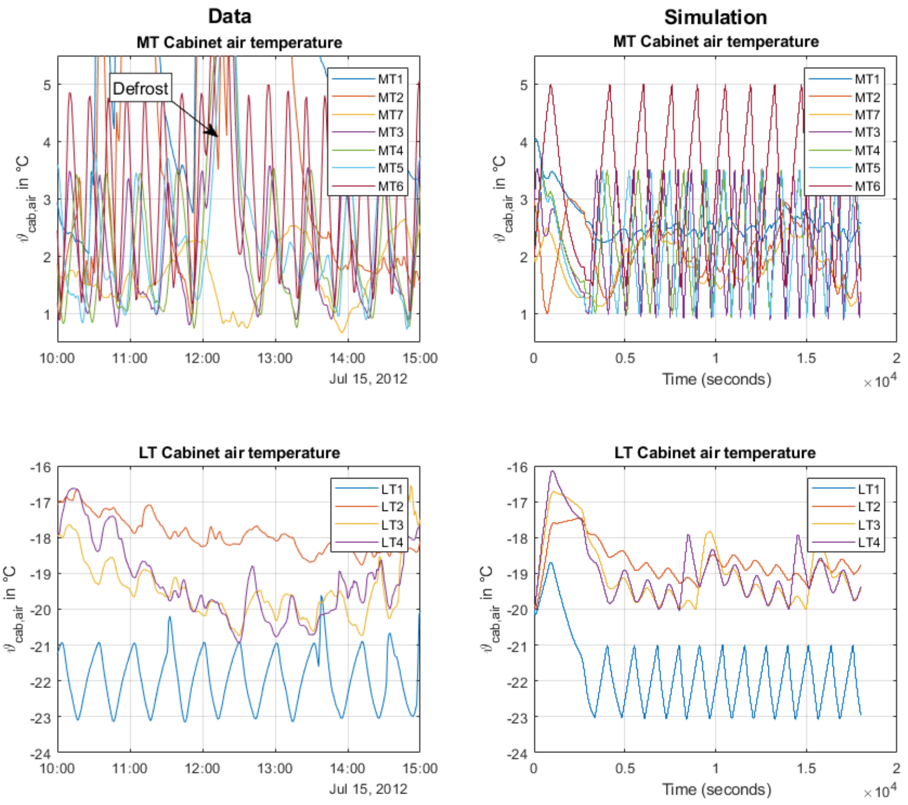

Figure 9 Comparison of cabinet air temperatures for an exemplary day and from simulation data. Relevant is the same basic behaviour, not an exact match

3.2. Influence of the evaporation pressure fluctuations

After validating the simulation model, the model now is considered the baseline for a further investigation into the influence of evaporation pressure fluctuations on the energy demand of the refrigeration system.

To investigate the influence, that the pressure fluctuations have on the energy demand a variant of the refrigeration system has to be modelled, that allows for a more stable evaporation pressure. For this the simulation model was changed in two major ways. First all display cabinets where changed from operating in thermostat mode, where they “cut-in” at an upper temperature limit and “cut-out” at a lower temperature limit, to “modulating temperature regulation” (MTR). In MTR mode the case controller primarily controls the air temperature in the display cabinet, while secondarily ensuring that a minimal superheat is maintained (Danfoss, 2018). Because in MTR mode refrigerant is constantly evaporating in the display cabinet the mass flow of gaseous refrigerant becomes more constant than in thermostatic mode. Therefore, reducing pressure fluctuations that result from cut-ins and cut-outs. As a second major modification the compressors in the refrigeration system where modified in a way that they can provide a continuous capacity without gaps. This allows the controller to always match the needed capacity to maintain the suction pressure. Compared to the baseline system this especially holds true for the LT compressors, where in the baseline system only two fixed speed compressors where installed, resulting in frequent compressor switching, as can be observed in Figure 6 and Figure 7. At low load situations missing capacity at the lower end can also result in frequent pump down cycles and the shut-off of the last running compressor, thus increasing fluctuation in the evaporation pressure.

Figure 10 compares the simulation results for the baseline model with the variant that allows for a more stable evaporation pressure. It can be seen that the fluctuation in the evaporation pressure are significantly decreased. This is especially true for the LT evaporation pressure. One reason for this is, that the LT evaporation pressure is only influenced by the LT display cabinets and the LT compressors, while the MT evaporation pressure is also subject to the influence of the LT compressor discharge and the refrigerant mass flow coming from the FGB. Despite these other influences the MT evaporation pressure also shows a significant reduction in the fluctuations. Further more it can be observed from Figure 10 that the variant model is able to maintain the setpoint for the evaporation pressure, while the baseline model has a lower average pressure. In the baseline model pump-downs and a minimum running capacity of the last compressor result in a lower average evaporation pressure at low ambient temperatures/ low loads.

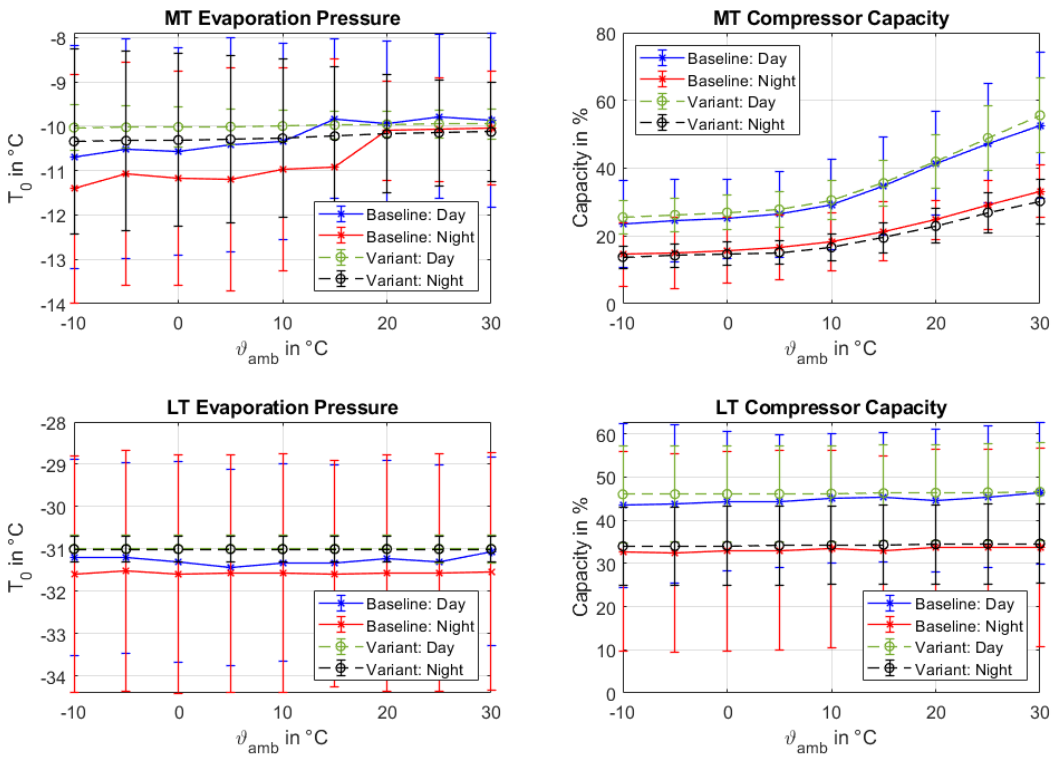

Figure 10 Comparison in both the average value and high of fluctuations for the evaporation pressure and the compressor capacity for both the baseline model and the variant model.

When examining the compressor capacity, it can be seen that the capacity between the baseline model and the variant model remain nearly identical. But looking at the fluctuations of the capacity, a reduction in the fluctuation can be observed.

In Figure 11 the compressor power and the cooling load is compared. Looking at the compressor power it can be seen, that overall the compressor power for the variant model is lower than the baseline model, as well as the fluctuation in the compressor power are reduced. For the cooling load only, the MT load at night seems to be reduced in the variant model. Further investigations into the exact reasons for this behaviour are still need. Despite this, an overall reduction in the compressor power in all scenarios could be observed.

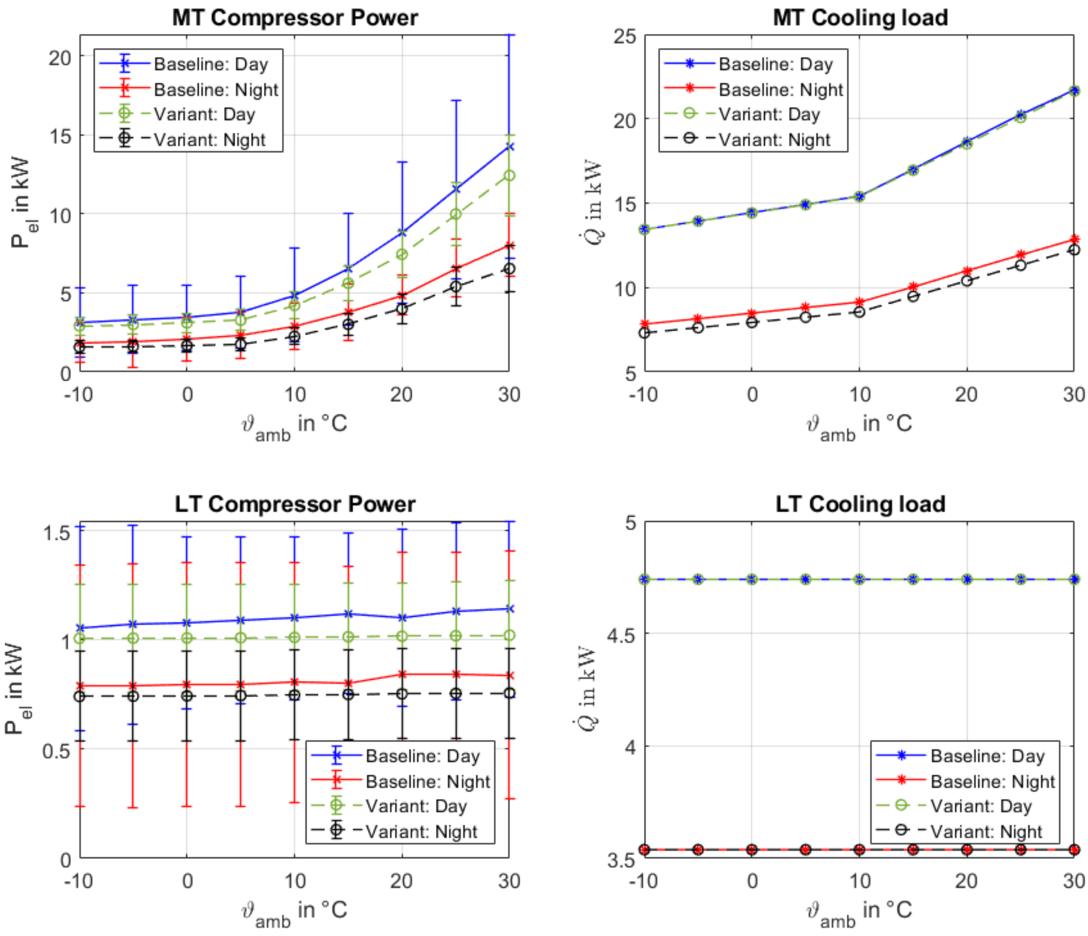

Figure 11 Comparison in both the average value and high of fluctuations for the compressor power and cooling load for both the baseline model and the variant model

It should be kept in mind that the simulation models used here are deterministic and do not consider any form of stochastic disturbances on the refrigeration system, such as humans interacting with the display cabinets in the supermarket. In a real supermarket refrigeration system these stochastic disturbances are unavoidable.

As was stated already in (Larsen et al, 2007b) the biggest leaver for reducing the compressor power is the increase of the evaporation pressure. As could be seen here, a more stable evaporation pressure leads to a better compliance of the actual evaporation pressure, with its setpoint. Furthermore, a more stable evaporation pressure opens up the possibility to increase the setpoint for the evaporation pressure, leading to further energy savings.

4. CONCLUSIONS

A simulation model of a supermarket refrigeration system was shown that is based on a real supermarket in Denmark. The model includes display cabinet models that are parametrised based on monitoring data. The compressor models are parametrised using manufacturer data and the controller models are equivalent to the real controllers used in the store.

The simulation model is extensively validated against existing monitoring data. The stationary behaviour of the simulation model is in good agreement with the observed data for the monitoring system. For the dynamic behaviour of the supermarket refrigeration system two ways to evaluate the agreement were used. First the fluctuations of the simulation model are compared against the fluctuations observed in the simulation. Here it was shown that a good agreement between monitoring data and simulation data was reached. Second a visual comparison between the simulated data and an exemplary day is performed. What could be observed is that the simulation result shows the same general behaviour, that can be found in the monitoring data.

In general, it can be concluded, that the simulation model is able to capture the relevant dynamics that occur in a supermarket refrigeration system. In particular the interaction between the compressor rack and the distributed evaporators is captured with a high degree of accuracy.

Comparing the baseline system against a variant, that was enabled to provide a more stable evaporation pressure, it was shown that overall the compressor power was reduced. These results give a strong indication, that a more stable evaporation pressure can reduce the energy demand for supermarket refrigeration systems. Further work will investigate if similar energy savings can be achieved with advanced control algorithms. For example, control algorithms that can actively match the cooling load to the available compressor capacity by managing the load distribution between the evaporators. Thus, easing the requirement for a continuous and gap-free compressor capacity.

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NOMENCLATURE

p	Pressure (Pa)	T	Temperature (K)
P	Power (W)	T_0	Saturated evaporation temperature
$\dot{Q}$	Heat flow (W)	ϑ	Celsius Temperature (°C)

Subscripts

amb	Ambient	hp	High pressure
cab,air	Display cabinet air temperature	night	Night time operation (Store closed)
day	Day time operation (Store open)	rec	Receiver
el	Electrical	0	Evaporation
gc,out	Gas cooler outlet		

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