Performance Assessment and Active System Monitoring for Refrigeration Systems

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PhD thesis, June 2012

DTU Electrical Engineering
Department of Electrical Engineering

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DTU Elektro – PhD Thesis
Kongens Lyngby June 2012

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Summary

The refrigeration system in a supermarket is an important part of the business for the supermarkets, both in terms of the possibility it provides and because of the associated cost of operating the system. It provides the possibility of selling chilled and frozen food but on the other hand the operation of the refrigeration system is associated with a significant cost. Cost efficient operation of the refrigeration system is therefore very important for the supermarkets. To ensure that the systems are operated cost efficient a performance assessment scheme is required. In addition, there exists a need for algorithms that ensures or improves the performance of the system.

A supermarket refrigeration system is usually a complex and distributed control system, and it can therefore be difficult to assess the performance without a formal method. The main interest for a supermarket, with respect to the refrigeration system, is to optimise the total cost of ownership, (TCO). However, directly measuring TCO provides some challenges. It can therefore be beneficial to divide TCO into performance criteria, which can be quantified and measured. For supermarket refrigeration systems the performance criteria can be divided into three categories: quality-, energy- and reliability-related criteria. Hence, it is important to operate the refrigeration system such that it ensures good quality of the stored goods as energy efficient as possible without compromising the reliability of the system.

A performance function that quantifies and measure the criteria has been developed in this project. The quality is measured by the control errors in the system because there is a connection between the quality of the stored goods and the ability of the refrigeration system to provide the required temperature. A deviation from the controller set-point corresponds to a temperature deviation,

which will eventually harm the stored goods. The energy efficiency is measured by the coefficient of performance, COP, which basically is the delivered cooling power divided by the consumed electrical power of the system. The reliability criteria is measure by the switch frequency of the compressors in the refrigeration system. The reason is that excessive compressor switching will wear down the compressors too fast and thereby decrease the reliability of the system due to a higher demand for maintenance. The proposed performance function provides a method for assessing the operational performance at a plan-wide level and is therefore providing a tool for improving the plant-wide performance.

The performance function has been used in different setups to improve the performance of the refrigeration system. Static and the dynamic performance of the refrigeration system has been addressed in the project. The proposed methods for improvement relies on a minimum of detailed knowledge about the refrigeration system. In addition, since a refrigeration system often operates in steady state an active system monitoring setup has been proposed, to enable improvement of the dynamic performance.

Dansk Resumé

Kølesystemet i et supermarked er en vigtig del af forretningen for supermarkedet, både på grund af de muligheden systemet tilføjer, og de omkostninger der er forbundet med driften af systemet. Kølesystemet giver mulighed for at sælge både køle og fryse varer, men på den anden side er driften af anlægget forbundent med en betydelig omkostning. Omkostningseffektiv drift af køleanlægget er derfor meget vigtig for supermarkederne. For at kunne sikre en omkostningseffektiv drift anlægget, er der derfor behov for en evalueringsmetode. Ud over det er der behov for algoritmer og metoder til at sikre og forbedre driften af systemet.

Et køleanlæg i et supermarked er normalt et komplekst og distribueret reguleringssystem, og det kan derfor være vanskeligt at vurdere den samlede driftskvalitet uden en formel metode. Hovedinteressen for et supermarked, med hensyn til kølesystem, er at optimere de samlede ejeromkostninger for systemet. En direkte måling af de samlede ejeromkostninger er for forbundet med en del udfordringer. Det kan derfor være en fordel at opdele de samlede ejeromkostninger i driftskriterier som kan kvantificeres og måles. Disse driftskriterier kan for et supermarked kølesystem opdeles i tre kategorier: Kvalitets-, energi- og pålidelighedsrelaterede kriterier. Med andre ord er det altså vigtig at sikre fødevarekvaliteten så energieffektivt som muligt uden at kompromittere systemets pålidelighed.

I projektet er der udviklet en driftskvalitetsfunktion, som kan kvantificere og måle kriterierne. Reguleringsfejlene i systemet er brugt til at måle fødevarekvaliteten, fordi der en sammenhæng mellem reguleringsfejlene og den fødevarekvalitet som kølesystemet kan levere. Hvis regulatorerne afviger fra referencen svarer det til at temperaturen afviger fra det ønskede, og det vil i sidste ende

forringe fødevarekvaliteten betydeligt. Energieffektiviteten er målt ved at bruge coefficient of performance, COP, som basalt set er defineret som den leverede køleeffekt divideret med den elektriske effekt som systemet har optaget. Systemets pålidelighed er målt ved skifte-frekvensen for kompressorerne i anlægget. Begrundelse er at overdrevet skift af kompressorerne tilstand mellem tændt og slukke, medfører forhøjet slid og vil derfor nedsætte systemets pålidelighed på grund af forhøjet behov for vedligehold. Driftskvalitetsfunktion kan bruges som en metode til at evaluere driftskvaliteten på system niveau, og kan derfor bruges som værktøj til at forbedre driftskvaliteten og dermed minimere de samlede ejeromkostninger.

Driftskvalitetsfunktionen har været bruget i forskellige sammenhæng til at forbedre driftskvaliteten af køleanlægget. Både statisk og dynamisk driftskvalitet er blevet behandlet i projektet. Metoderne der er blevet foreslået til forbedring af driftskvaliteten afhænger kun af et minimum af detaljeret viden om kølesystemet. For at kunne forbedre den dynamiske driftskvalitet er der blevet foreslået en aktiv systemovervågningstilgang, da et kølesystem for det meste af tiden befinder sig i ligevægtstilstand.

Preface

This thesis was prepared at DTU Electrical Engineering, Automation and Control Group, at the Technical University of Denmark as a partial fulfillment of the requirements for acquiring the Ph.D. degree in engineering. The project was funded by Danfoss A/S, Refrigeration & Air Conditioning, The Agency for Science Technology and Innovation and The Technical University of Denmark.

The thesis deals with performance assessment and operational optimisation of supermarket refrigeration systems. The main focus is to provide a plant-wide performance assessment technique for a refrigeration system and using it for optimisation of the operation of the plant.

The project was supervised by Associated Professor Henrik Niemann, DTU Electrical Engineering and Roozbeh Izadi-Zamanabadi, Danfoss A/S, Refrigeration & Air Conditioning. In addition, the project has been co-supervised by Professor Mogens Blanke, DTU Electrical Engineering and Morten Juel Skovrup, IPU, has been assigned to the project as third-party supervisor. Part of the research was conducted at Laboratoire d'Automatique, Université Libre de Bruxelles with Professor Michel Kinnaert acting as supervisor.

The thesis consist of a summary report an a collection of six papers written in the period 2008-2012

Kongens Lyngby, June 2012
Torben Green

Papers included in the thesis

[A] Torben Green, Roozbeh Izadi-Zamanabadi and Henrik Niemann. On the choice of performance assessment criteria and their impact on the overall system performance – The refrigeration system case study. Published in *Proceedings of the Conference on Control and Fault-Tolerant Systems (SysTol'10)*, pp. 624 – 629, Nice 2010. IEEE. Digital Object Identifier (doi): 10.1109/SYSTOL.2010.5676067

[B] Torben Green, Henrik Niemann and Roozbeh Izadi-Zamanabadi. Performance improvement clarification for refrigeration system using active system monitoring. Published in *Proceedings of the 18th IFAC World Congress* pp. 2845 – 2850, Milan 2011. Available at ifac-paperonline.net

[C] Torben Green, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Design of Excitation Signal for Active System Monitoring in a Performance Assessment Setup. Published in *Proceedings of the 9th European Workshop on Advanced Control and Diagnosis*, Budapest 2011.

[D] Torben Green, Michel Kinnart, Roozbeh Razavi-Far, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Optimising performance in steady state for a supermarket refrigeration system. Accepted for presentation at the *20th Mediterranean Conference on Control and Automation*, Barcelona 2012

[E] Torben Green, Roozbeh Razavi-Far, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Plant-wide performance optimisation – The refrigeration system case. Accepted for presentation at the *2012 IEEE Multi-Conference on Systems and Control*, Dubrovnik 2012.

[F] Torben Green, Roozbeh Izadi-Zamanabadi, Roozbeh Razavi-Far and Henrik Niemann. Plant-wide Dynamic and Static Optimisation of Supermarket Refrigeration Systems, Submitted to the *International Journal of Refrigeration*

Acknowledgements

In the beginning of 2008 I was working as an engineer for Danfoss in a research and development department when I was presented with the great opportunity of pursuing an industrial Ph.D. degree (ErhvervsPhD). I accepted the challenge that I was faced with and began the pursuit of the Ph.D. degree. This pursuit has been made possible by a number of people to whom I owe thanks.

I consider myself proud and lucky to have been presented the challenge by my boss Peter Eriksen and my former colleague at Danfoss Dr. Claus Thybo. The process of formulating the project idea was made significantly easier by the insightful contributions from Dr. Lars Finn Sloth Larsen.

The project has been supervised by four competent profiles that throughout the project has pushed me further than I ever expected. Working together with my academic mentor Associated Professor Henrik Niemann has been very inspiring and productive. Precise and valuable criticism has been provided by my co-supervisor Professor Mogens Blanke. Dr. Morten Juel Skovrup has with his insightful contribution helped to widened my refrigeration knowledge. My company supervisor Dr. Roozbeh Izadi-Zamanabadi has skillfully taught me how to work in the field between academia and industry and provided me with guidance and ideas. To all four: Thanks for the support.

I would also sincerely like to thank Professor Michel Kinnaert for my short, but nevertheless very productive and effective research visit at Laboratoire d'Automatique, Université Libre de Bruxelles.

My Family has throughout these years provided me with continues support, and showed me that they believed in me. For that I would like to gratefully thank them all.

Last but definitely not least important, I would like to express a heartfelt thanks to my wife, Lotte Ellemann Green, for her immense and persistent support, which has provided me with the strength to pursue the goal.

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Contents

CHAPTER 1

Introduction

This thesis is written as a summary report and a collection of papers. Each of the papers are placed in the back of the thesis. They have all been reformatted and represents individual pieces of work. The summary part of the thesis is divided into 4 chapters. This chapter gives a basic introduction to the project and chapter 2 provides a description of the main application used in the project, i.e. the supermarket refrigeration system. In chapter 3 a description of the research contributions and a concentrate of the results of the project is presented. In addition the chapter gives an presentation of the state of the art within the different research areas.

This chapter is organised as follows. The problem formulation is presented in 1.1 and is used an guideline throughout the chapter. In section 1.2 the background for the project is presented and section 1.3 provides a description of the motivation for the project.

1.1 Problem Formulation

The overall goal can be formulates as trying to find the answer to the following question: How can sufficient refrigeration be provided with as low cost as possible? Pursuing an answer for that question provides many different solution paths. The general solution path that has been chosen in this research project is to provide more cost optimal performance of the refrigeration plant by providing better control. However, to pursue that solution path the problem has been reduced to answering the following thee governing questions:

1. *Is is possible to establish performance measures for control systems of a supermarket refrigeration system?*
2. *Furthermore, is it possible to develop methods or algorithms to monitor the systems and analyse its performance by means of the established measures?*
3. *In addition is it possible to develop methods that can accommodate for a detected performance degradation of the system?*

Question 3 is interpreted in the following two ways. The first interpretation is that the methods should detect a performance degradation and then accommodate for that degradation. The second interpretation is that the system is assumed not to be operating optimal and the method should therefore try to improve the performance. The second interpretation has been the predominant one use in the project.

Fig. 1.1 illustrates the general solution path that has been pursued in an effort to provide an answer to the governing questions. The operational perfor-

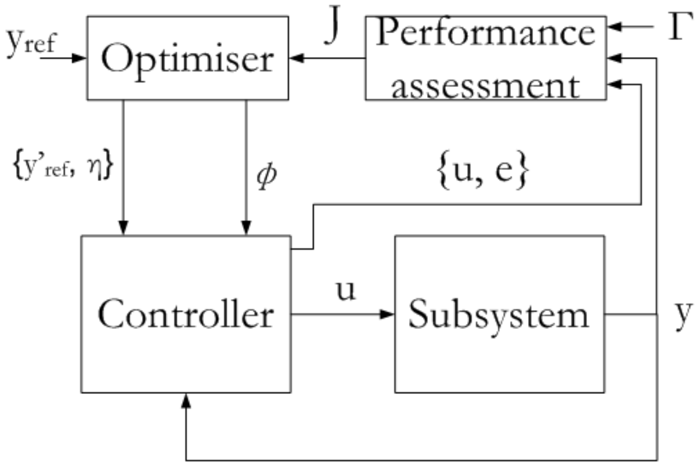

Figure 1.1: Block diagram of system-wide performance assessment and optimisation

mance, J, of the entire plant is provided by the performance assesment block, based on the control error, e, the control output, u, the output from the subsystem, y, and the contributions from the other subsystem in the entire system, Γ. The task of the optimiser is then to improve the performance, which can be done by manipulating the reference signal, y'_{ref} or by applying an excitation signal, η, and adjusting the controller parameters, ϕ.

1.2 Background

Demand for energy efficiency and lower cost level has been some of the reasons for the rising attention that optimal operation of the refrigeration systems in supermarkets has been given over the past years. The refrigeration system has a lot of costs associated. They are the initial invenestment and the costs of operation. The operational costs for refrigeration plants are, maintenance, energy and additional costs. Additional costs are for example the cost of damaged food and reduces sales due to maintenance. The energy cost of a typical refrigeration plant in a supermarket accounts for 40% to 60% of the total electrical energy consumption in a supermarket, see [10]. Thus, the incentive to have a refrigeration plant that is energy efficient, reliable and does not require maintenance often, exists. This is also the reason why it is important to provide answer to the governing questions 1 and 2.

Questions 1 and 2 of the governing questions for the project, might seem trivial to answer, however they pose some problems in real life applications which will be explained hereafter. Plant-wide performance assessment for any industrial complex system requires that an appropriate set of performance criteria has been defined. For a refrigeration system in a supermarket these criteria include the food quality of the stored goods, and the ability to suppress external disturbances, the energy efficiency and the reliability of the plant. The main objective of a supermarket refrigeration plant including the control system, is of course to ensure that the goods are stored at the correct temperature, and at the same time ensuring cost optimal operation of the plant. This can only be achieved if the refrigeration system including the control system is designed properly. The design of a refrigeration system can be split into two main tasks, which are: designing the layout of the refrigeration system and the second task is to design the controller structure and the control system. As it can be seen on Fig. 1.2 a refrigeration system for a supermarket is equipped with many components that require control. In addition all of these components are spatially distributed across the supermarket which calls for a decentralised controller setup with many different control loops. Thus, the evaluation of the plant wide performance is not a trivial task. The distributed control setup provides an

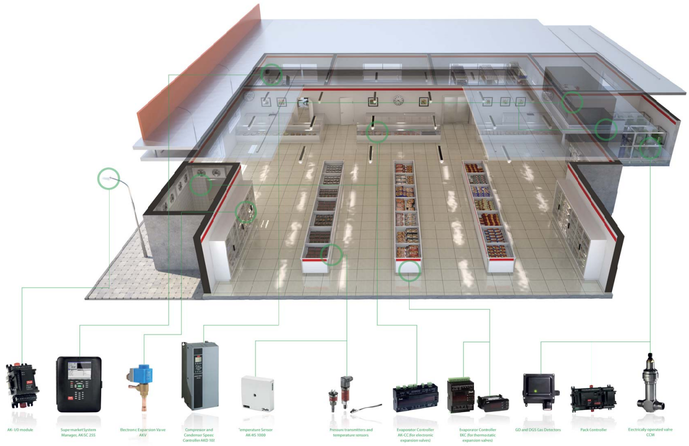

Figure 1.2: Picture of a supermarket refrigeration system including the controller hardware

immediate challenge with respect to developing the methods and algorithms to measure and evaluate the performance of the entire refrigeration plant.

The performance assessment problem is of high interest because it is necessary to be able to define and measure the performance when the overall objective of the control system is to provide optimal performance of the operation of the plant. Hence, minimising total cost of ownership requires an ability to assess the dynamic and static performance of the plant on different time scales. Therefore, answers to question 1 and 2 is required to be able to provide solutions that enable a positive response to question 3 from the hypothesis.

Providing a positive response to question 3 from the hypothesis is another important reason for initiating the research project. Moreover, optimising the performance of the operation is not a trivial task. For example, optimising the performance by manually adjusting the parameters in all the controllers to obtain the optimal performance is almost an impossible task. Hence, tuning the control system to provide optimal performance is almost never achieved, simply because the task is too overwhelming to complete manually. Thus, enabling the supermarkets to achieve optimal operation of their refrigeration plants has also been an important reason for initiating the research project.

In the design process for the controller products, Danfoss has given extra attention to ease of use of the product. Thus, the idea of developing algorithms that can assist the supermarkets to achieve optimal operation of the refrigeration system has also been a driving factor for the research project.

1.3 Motivation

The interest in the performance assessment problem, originates from the strategic aim of Danfoss to provide, the best energy efficient and most sustainable solutions, to all the different markets that Danfoss operates within. This can basically be interpreted as a driving factor for providing positive responses to questions 1 through 3 in the hypothesis.

Refrigeration is one of the business areas in which Danfoss aims to fulfill its strategic aim, especially within the food retail segment. The main products that Danfoss produces for the refrigeration industry within the food retail segment are controllers. Thus, the aim is to provide the market with controllers that will ensure optimal operation for a given refrigeration plant. In the past local optimization of individual subsystems was in focus. However, since the control system consists of many different subsystems, plant-wide optimisation is preferable. Optimisation of the plant-wide performance is of course only possible, with a predefined notion of optimal operation at a plant-wide level.

Even tough total cost of ownership is the main optimisation target, measuring total cost of ownership in a suitable fashion can be infeasible. To provide a feasible measurement, that is suitable for optimising the control system, phenomenon on different time scales has to be taken into account. Food quality, component life time and maintenance all impact the total cost of ownership, nevertheless they are difficult to evaluate. Thus, besides a definition of the plant-wide performance a sufficient measurement of such a performance is required. Moreover, even with a definition and a measurement of optimal operation, setting up the control system to ensure plant-wide optimal operation is not trivial. Thus, the project has been motivated by the fact that such challenge exists. The challenge is to assess the performance of the refrigeration plant and secondly how to ensure that the operation is optimal under all conditions.

Providing a positive response to the third governing question poses some problems. Designing control solutions that provides optimal and robust operation requires knowledge about the underlying system. Moreover, having deeper knowledge about individual subsystems enables design of dedicated controllers, which will provide the best performance with respect to both operational optimality and robustness. However, dedicated controller design will lead to lack of generality of the control solution. Thus, if general usability of the control solution is preferable the design solution cannot depend on system specific knowledge. This is exactly the situation when designing control solutions for the refrigeration systems in the food retail segment. The problem of designing control solutions that is optimal with respect to operational performance is difficult due to the information that the design is allowed to depend on.

In the food retail segment Danfoss only provides the components for the control system. The design of the control system and of the physical layout of the refrigeration system is done by contractors, which are hired by the supermarket. The setup of the control system is not done by control engineers or by Danfoss personnel it is done by the contractor who is commissioning the plant. Therefore, any design of performance assessment and optimisation cannot depend on detailed knowledge about the refrigeration plant. Furthermore, it is the aim of Danfoss to provide controllers that are easy to use and thereby decreases the commissioning cost of the control system. This should, of course be taken into account when designing the algorithms for performance assessment.

In an effort to provide the reader with an example of the performance issues appearing in a real supermarket system a short description of the capacity gap problem is presented here. For details see the paper A. The capacity gap problem arises because the compressors are discretely controlled and therefore only have the possibility of being switched on or off. Thus, the switching behaviour of the compressor rack is dependent on the operation point since some operation points will required less switching than others. An example of the system swiching operation point can be seen on Fig. 1.3. To improve the performance of the

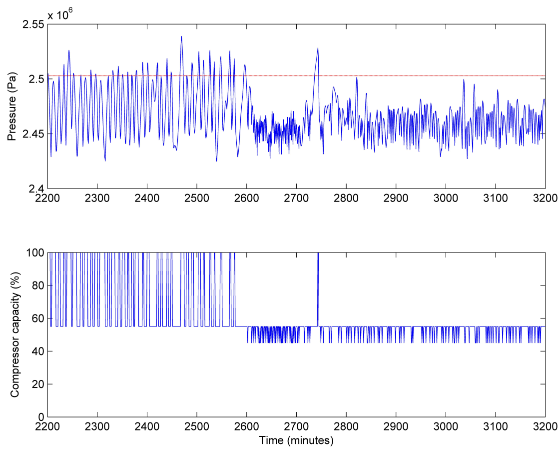

Figure 1.3: Pressure and running compressor capacity, including the transition in operation point due to the closing of the supermarket

refrigeration plant phenomenon like switching should be taken into account when evaluating the performance.

CHAPTER 2

The Supermarket Refrigeration System

The supermarket refrigeration system is chosen as case study for a number of reasons. The main reason is that controls solutions for the food retail segment and especially supermarkets are an important business area for Danfoss. Secondly, the application poses interesting problems with respect to performance assessment and optimisation of the operational performance. The chapter is divided into two sections where 2.1 describes the supermarket refrigeration systems in an effort to provide the reader with insight about the application targeted in this thesis. Section 2.2 provides a description of the simulation model that has been used to test the solution ideas on.

2.1 System description

The supermarket refrigeration system is based on the vapour compression cycle which utilises the thermodynamic properties of a certain refrigerant to absorb heat by evaporating at one pressure level and then, after compressing the refrigerant gas, rejecting the heat to the surroundings while condensing the gas. Basically the process requires four main components which are: an evaporator and a condenser, an expansion device and a compressor. These component are all used in the supermarket system as well. However, there is not only one of each to those components.

Fig. 2.1 shows a schematic drawing of a typical implementation of a refrigeration system for a supermarket. All the previously mentioned components are used, however instead of one evaporator there is a number of evaporators spread across different display cases. In addition, the system is able to provide refrigeration at two different temperature levels because of the use of two different compressor racks. That is, one compressor rack for the chilled display cases and another for the frozen display cases. The compressor racks are both controlled to maintain a certain pressure on the suction side of the compressors, which is corresponding to the desired evaporation temperature for either the frozen or the chilled display cases. The evaporation temperature determines

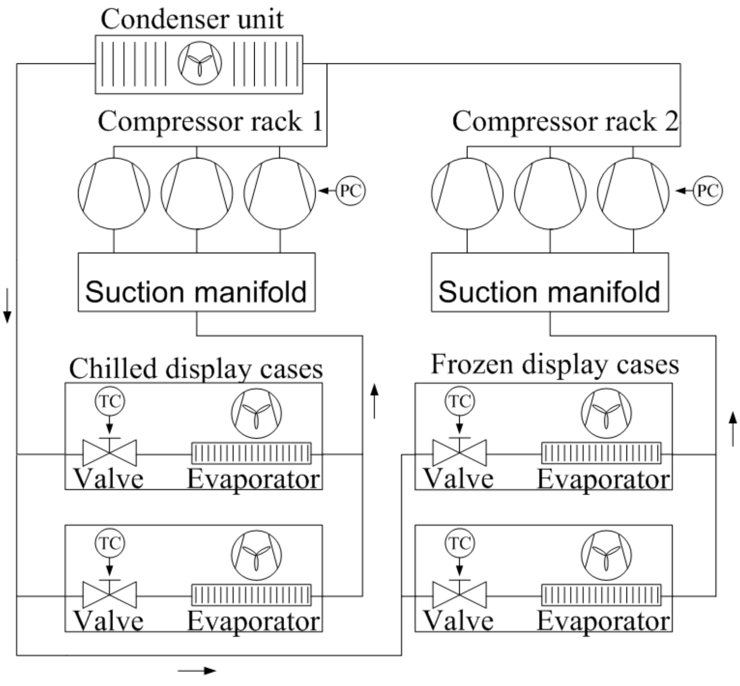

Figure 2.1: Schematic drawing of a supermarket refrigeration system with display cases for both chilled and frozen food

the temperatures that can be achieved within the corresponding display cases. The air temperature within the different display cases are controlled by a local temperature controller. The air temperature is controlled by manipulating the inlet valve and thereby the refrigerant flow through the evaporator. If the temperature is too high the controller will increase the opening of the valve and thereby increase the refrigerant flow through the evaporator. This will lead to an increase in heat transfer from the air to the refrigerant and thereby also a decrease in the temperature of the air.

The number of compressors in each of the racks and the number of display cases shown in Fig. 2.1 are chosen to show that there are more than one compressor and one display case. For example in a real life supermarket system the number of display cases is usually higher than three. The size of the system is of course dependent on the size of the supermarket.

2.2 Simulation model

The model that is presented in this section has been used throughout the project. The model is a modified version of the model presented in the paper [28]. The model illustrates a simplified version of a supermarket refrigeration system, more detailed modelling techniques for refrigeration systems can be found in [18]. The system includes a compressor rack, a condenser unit, two display cases including expansion valves and a suction manifold. Even though the model only portrays a simplified system, the relevant performance related phenomenon are captured by the model. Fig. 2.2 show a schematic of what is included in the model. The model consists of two chilled display cases, which both are temperature controlled using a PI controller to continuously manipulate the inlet valve to the evaporator, and thereby controlling the refrigerant flow through the evaporator. The compressor rack consists of two compressors and the individual sizes of the compressors are chosen based on a compressor rack from a real supermarket.

The air temperature within a display case is described by (2.1) and the load experienced as an heat transfer from the surrounding air of the display case is described by (2.2). The temperature of the stored goods is described by (2.3) and the temperature of the evaporator wall is described by (2.4). The mass of

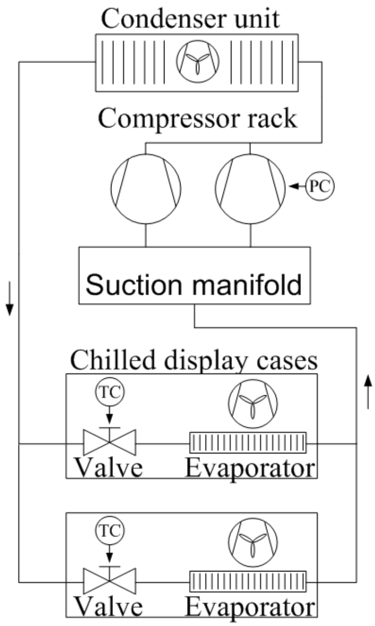

Figure 2.2: Schematic of the modelled supermarket system

refrigerant within the evaporator in a display case is described by (2.5).

$$\frac{\mathrm{d}T_{\mathrm{air,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\mathrm{goods-air,i}}(\cdot) + \dot{Q}_{\mathrm{load,i}}(\cdot) - \dot{Q}_{\mathrm{air-wall,i}}(\cdot)}{M_{\mathrm{air}} C_{\mathrm{p,air,i}}} \tag{2.1}$$

$$\dot{Q}_{\mathrm{load,i}} = UA_{\mathrm{amb}} \cdot (T_{\mathrm{amb}} - T_{\mathrm{air,i}}) \tag{2.2}$$

$$\frac{\mathrm{d}T_{\mathrm{goods,i}}}{\mathrm{d}t} = -\frac{\dot{Q}_{\mathrm{goods-air,i}}(\cdot)}{M_{\mathrm{goods,i}}\ C_{\mathrm{p,goods,i}}} \tag{2.3}$$

$$\frac{\mathrm{d}T_{\mathrm{wall,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\mathrm{air-wall,i}}(\cdot) - \dot{Q}_{\mathrm{e,i}}(\cdot)}{M_{\mathrm{wall,i}} C_{\mathrm{p,wall,i}}} \tag{2.4}$$

$$\frac{\mathrm{d}M_{r,i}}{\mathrm{d}t} = OD_i \cdot \alpha \cdot \sqrt{P_c - P_{suc}} - \frac{\dot{Q}_e}{\Delta h_{lg}} \tag{2.5}$$

The notation in (2.1) through (2.5) will be described hereafter. The air temperature of the i^{th} display case is denoted by $T_{\mathrm{air,i}}$ and the heat transfer rate from the goods to the air is denoted by is denoted by $\dot{Q}_{\mathrm{goods-air,i}}$ and the heat transfer from the air inside the display case to the evaporator wall is denoted by $\dot{Q}_{\mathrm{air-wall,i}}$. Detailed description of the terms, $\dot{Q}_{\mathrm{goods-air,i}}$ and $\dot{Q}_{\mathrm{air-wall,i}}$ can be found in [28], which are all refrigerant dependent functions and other variable, hence the notation $(\cdot)$. The heat load experienced by the display case based on heat transfer to the surrounding air is denoted by $\dot{Q}_{\mathrm{load,i}}$. The temperatures of the stored goods and the evaporator wall is denoted by $T_{\mathrm{goods,i}}$ and $T_{\mathrm{wall,i}}$, respectively. Mass and heat capacity is denoted by M and C_{p} and the corresponding medium is then denoted by the subscript. The opening degree of the i^{th} inlet valve is denoted by OD_i and α denotes the cross sectional area of the valve. The suction pressure and the condensing pressure is denoted by P_{suc} and P_c, respectively. The heat transfer rate caused by evaporation is denoted by $\dot{Q}_e$. The enthalpy difference across the two-phase region of the evaporator, i.e. where the refrigeration is a mixture of liquid in gas, is denoted by Δh_{lg}. To simplify the model it is assumed that the majority of the heat transfer is done in the two phase region of the evaporator and the heat transfer in the single phase region of the evaporator is therefore neglected.

The pressure on the low pressure side of the refrigeration system, i.e. the suction pressure, is assumed to be the same across all the display cases and the suction manifold. Thus, the dynamics of the suction pressure is describe by (2.6).

$$\frac{\mathrm{d}P_{\mathrm{suc}}}{\mathrm{d}t} = \frac{\dot{m}_{\mathrm{in-suc}}(\cdot) - \dot{m}_{comp}}{V_{\mathrm{suc}}\ \nabla\rho_{\mathrm{suc}}(P_{\mathrm{suc}})} \tag{2.6}$$

The volume of the suction manifold and the pressure derivative of the density is denoted by V_{suc} and $\nabla\rho_{\mathrm{suc}}$, respectively. The mass flow rate into the suction manifold is denoted by, $\dot{m}_{\mathrm{in-suc}}$ and is described by:

$$\dot{m}_{\mathrm{in-suc}}(M_{\mathrm{r,i}}, T_{\mathrm{wall,i}}, P_{\mathrm{suc}}) = \sum_{i=1}^{N} \frac{\dot{Q}_{\mathrm{e,i}}(\cdot)}{\Delta h_{\mathrm{lg}}(P_{\mathrm{suc}})} \tag{2.7}$$

The mass flow rate that is exiting the suction manifold and continuing into the compressor rack is denoted by, $\dot{m}_{comp}$ and described by the following:

$$\dot{m}_{comp} = \mathtt{Cap} \cdot \frac{1}{100} \cdot \eta_{vol} \cdot \dot{V}_{sl} \cdot \rho_{suc} \tag{2.8}$$

The combined volumetric efficiency of the compressor rack is denoted by, η_{vol}, and the swept volume rate of the is denoted by, $\dot{V}_{sl}$. Hence, the compressor rack is model as a single compressor, with a sufficient size, which can be controlled in steps corresponding to mimic the behaviour of a compressor rack. Transient behaviour of $\dot{m}_{comp}$ is neglected and thus (2.8) is only depended on the suction pressure through the density, ρ_{suc}. The running compressor capacity in the rack is denoted by `Cap` and is described by:

$$\mathtt{Cap} = \sum_{i=1}^{i=N} \delta_i \mathtt{Cap}_i \tag{2.9}$$

The discrete variable, (2.9), $\delta_i \in \{0, 1\}$ determines whether a compressors is or off. In (2.9) $\sum_{i=1}^{N} = 100\%$ and in the implementation of the model used in this project $N = 2$, $\texttt{Cap}_1 = 45\%$, and $\texttt{Cap}_2 = 55\%$. As mentioned before the layout of the compressor rack in this model is based on a real supermarket which has the same compressor layout. The model describes a simplified refrigeration system and formal validation of the model has therefore not been attempted. The parameters in the model have been tuned to resemble the behaviour of small size supermarket refrigeration system. Hence, some the results based on the model might only apply for smaller refrigeration systems.

In an attempt to reduce excessive switching a hysteresis band is applied around each possible combination of compressors in the rack. This particular compressor rack will have four possible states which are: the compressors are not running, compressor one is running, compressor two is running and both the compressors are running. Hence the compressor rack can only deliver 0, 45, 55 or 100 percent compressor capacity. In table 2.1 a detailed description of the compressor layout is presented. Each line in the table describes a certain step in the compressor rack. The % column describes the capacity provided by the compressor rack at the corresponding step and the columns Min and Max describes the hysteresis for the step. The two last columns, Shift up and Shift down, describes which steps should be chosen if the requested capacity exceeds either the Max or Min capacity of the current step. The model of the condenser unit contains no dynamics, the model simply defines a static condensing pressure and a static sub-cooling which implies the assumption that the condensing unit is sufficiently dimension and well controlled.

Name	%	Min	Max	Shift up	Shift down
step_off	0	0	40	step_1	step_off
step_1	45	30	50	step_2	step_off
step_2	55	40	80	step_3	step_1
step_3	100	70	100	step_3	step_2

Table 2.1: Compressor rack definition

Chapter 3

Research Contributions

The aim of this chapter is to highlight the novel contributions and present some selected results generated throughout this project. The work presented in the thesis touches upon many different research disciplines from automation and control through fault detection and diagnosis, performance assessment, performance monitoring, optimisation of dynamic systems to refrigeration. Hence, the contributions should be considered as intra-disciplinary contributions. The broad scope of the research has to some extend limited the depth of the research within the specific areas.

The main contributions of the project are within the following areas:

- Performance assessment
- Performance improvement

This chapter will elaborate on each of the contributions listed above in separate sections. These sections will also contain the state-of-the-art and related works for each contribution. In addition, the main methods and techniques that has been used throughout the project will be described.

The chapter is divided into two sections where section 3.1 presents the methods and ideas that has been used within the area of performance assessment.

The methods and ideas related to the contributions in the area of performance optimisation are described in section 3.2.

3.1 Performance assessment

The main dissemination has been done in the papers A and D however B, C, E and F mainly present different applications of performance assessment contribution. State of the art related to the area is mentioned throughout this section.

Performance assessment of control systems covers a broad range of problems and applications. The focus of this work has been to assess the performance of a complex dynamic system, which is comprissed of a number of subsystems, from a plant-wide or global perspective. Performance assessment of automated system has been given a considerable amount of attention in the literature, in particular, within the literature related to the process industry. Suboptimal performance can often increase the total cost of ownership for a process plant significantly and thereby decrease profitability of the plant. Directly measuring the total cost of ownership for any given plant is usually not a feasible tactic, and therefore the problem is often tackled by assessing some performance criteria. The relevant choice of these criteria is, of course, dependent on the specific application. However, they can usually be grouped into three different categories, which are:

Quality-related performance criteria: Criteria related to the quality of the process often relates directly to variables that can be measured directly. The task of many low-level controllers are actually to ensure that the process is delivering the desired quality. Food quality and desired temperature comfort are examples of quality-related performance criteria within the field of control of refrigeration system. In the papers [8] and [28] temperature is used, at a regulatory level, as a quality related criteria for desired temperature comfort and food quality, respectively. The quality of a process is often among the most critical performance criteria, which explains why it is usually controlled by a dedicated controller at the regulatory level.

Energy-related performance criteria: The task of high level controllers in the control hierarchy often deals with less critical control tasks with respect to the process. Hence, performance criteria related to energy consumption are usually taken care of by high level controllers. An example of the use of an energy based criterion can be seen in [32], where it is used for optimal control of a wireless communication channel.

Reliability-based performance criteria: The use of reliability related performance criteria are still not common in the process industry and neither in the refrigeration systems. However, in [12] reliability is connected to the fault-tolerance of software systems. Reliability is also considered with respect to fault tolerant control systems in [50] and [51].

3.1.1 Measuring of performance criteria

The ability to optimise any given process based on performance assessment, depends on the ability to quantify and there by measure the performance criteria and having access to manipulate relevant control inputs and parameters. Rendering a criteria measurable might include dividing the criteria into sub-criteria. As mentioned before the total cost of ownership can for example be divided into quality-, energy- and reliability-related criteria. Albeit, these sub-criteria still need to be measurable in a feasible way. References [41], [17] and [36] provide overview on different approaches toward performance measurements in the process industry. The quality of the output of a process, is an important performance criteria within the process industry, which explains the popularity of the the minimum variance benchmark within the field.

The benchmark approach toward measuring a performance criteria is typically used to compare the quality of a controller against a certain reference or benchmark for the given system. These approaches are often model based, see for example the model predictive case in [46] and the more general approaches to model based performance assessment can be found in [36], [17], [16] and [41].

3.1.2 Performance assessment of a refrigeration systems

Quality, energy and reliability is basically also the three main criteria that is considered when evaluating the performance of a refrigeration system. When if comes to refrigeration systems like the one described in chapter 2 the main concern is of course the food safety or in other words the temperature quality. Hence, the refrigeration systems ability to ensure the correct temperature is used as a quality measurement of the system. The temperature quality is closely related to the total cost of ownership because low quality temperature control will eventually destroy the stored food. Hence, the value of the stored food will have to be depreciated to zero.

The focus on performance assessment within the refrigeration community is usually on the effeciency of the vapour-comression cycle. In addition, dynamic behaviours are usually obmitted in a performance assessment analysis. The paper [42] presents a method for describing the performance of plant under some static assumptions and by comparing measurements with a theoritical calculated reference.

Energy efficiency of the refrigeration plant is another common interest with respect to performance assessment, because the energy consumption is at considerable level for a refrigeration system. Within the refrigeration industry the common way of assessing the efficiency is done by evaluating the coefficient of performance, COP, which is given by (3.1).

$$COP = \frac{\dot{Q}_{cool}[w]}{W_{ref}[w]} \ , \tag{3.1}$$

The COP is defined as the ratio between the cooling power delivered,$\dot{Q}_{cool}$ and the total electrical power, W_{ref} consumed by refrigeration system. COP is used as a performance measure in [3], [29], [26], [49] and [47]. A different approach to performance assessment of a refrigeration system is presented in [28] and [8] where compressor switching is considered in the evaluation of the performance.

3.1.3 The performance function

A method for performance assessment on a supermarket refrigeration system is presented as one of the contributions regarding performance assessment in the project. In this subsection the assessment approach will be described and the novelty will be outlined.

The focus of the performance assessment has been to provide a plant-wide performance assessment technique. Relying on a plant-wide perspective in contrast to a more local perspective with respect to performance assessment has some advantages. The local perspective provides the possibility of using performance measures that is targeting a specific performance issue locally. However, even though the local perspective does provide detailed insight to the performance of the local subsystem, the impact on the plant-wide performance might be insignificant and thereby unimportant with respect to the plant-wide performance.

Manual evaluation of the performance of the operation of a supermarket refrigeration system is a non-trivial task. The systems are, as mentioned in chapter 1, complex and distributed which renders it difficult to get an overview of the entire system. Hence, an algorithm for plant-wide evaluation of the performance is preferred.

Total cost of ownership has been divided into the three different performance criteria categories, quality-, energy- and reliability-related criteria. The quality-related criterion is chosen to be the food quality and the energy-related criterion chosen to be the cooling efficiency and as the reliability-related criterion component degradation is chosen. After choosing the criteria it has to be considered how the criteria can be measured on a refrigeration system. In this project, the criteria has been measured as follows: To represent the food quality the control errors from the controllers for each of the display cases and the compressor rack has been used. If the control errors are very high the ability of the system to provide sufficient refrigeration will not exist, which eventually will harm the food quality and thereby increase the cost of operating the system. Cooling efficiency of the plant is represented by a COP. Component degradation is represented by a switch frequency of the compressors in the system. Superfluous degradation of components will decrease the reliability of the system because it will increase the risk of component failure. One of the most crucial components in a refrigeration system is the compressor and the degradation of the compressor is increased by excessive switching.

Combining the measurements of the three criteria yields the performance function (3.2).

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}(k)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP(l)}||_R^2 + \sum_{m=1}^{M} ||f_{sw}(m)||_S^2 \tag{3.2}$$

The performance function is a sum of quadratic forms. Hence, the notation is given by (3.2).

$$||\mathbf{e}||_Q^2 = \mathbf{e}^T Q \mathbf{e} \tag{3.3}$$

In (3.2) K is the number of controllers, L is the number of suction groups and M is the number of compressors. The inverse of COP is used in the function to ensure that all the terms have the property that minimisation will be optimal.

Normalisation of each of the terms in the performance function is important to ensure scalability and re-usability of the performance function. It should be possible to employ the performance function on different refrigeration systems of different size and dimension. This scalability can be achieved through normalisation of the performance function into unitless terms. The performance function will then be established for any given supermarket by the appropriate choice of the weights. The appropriate weights will then have to account for the local regulations with respect to food quality. Moreover, the weights will have to be chosen with respect to the local energy prices and the cost of replacing compressors. As mentioned before this all requires that the terms are normalised to begin with. The normalisation procedure is explained hereafter.

By using the knowledge that temperature in the display case has a lower and an upper limit and that the reference is chosen as the mean value of the two limit, normalisation is achieved. Normalisation of the suction pressure error is reached using the same technique. The error vector, $\mathbf{e}$, can by applying then normalisation then be described by (3.4).

$$\mathbf{e} = \left| \begin{array}{c} \frac{2}{(T_{max,i} - T_{min,i})} \cdot (T_{\mathrm{ref,i}} - T_{\mathrm{air,i}}) \\ \frac{2}{(P_{suc,max} - P_{suc,min})} \cdot (P_{\mathrm{sucref}} - P_{\mathrm{suc}}) \end{array} \right| \tag{3.4}$$

The inverted COP term does not require any normalisation because the COP is unitless by definition. However, the switch frequency term does require normalisation. Dividing the measured switch frequency by the maximum allowable switch frequency provides a normalisation of the switch frequency term. All the terms, $\mathbf{e}$, $\frac{1}{COP}$ and f_{sw} are, after normalisation, in the range $\{0, 1\}$. The weights, Q, R and S in (3.2) associated with the error term, the inverted COP term and the switch frequency term will thus represent the cost, respectively. The choice of the weights Q, R and S will have to reflect the impact on the cost of operation from the corresponding term in the performance function. Hence, the choice of Q will be based on the price of significant temperature error in the display case. The weight, Q will therefore be depended on the price of the stored goods. The purpose of Q is to model that if the temperature in the display case exceeds are certain limit the stored good will be destroyed and good will have to be discarded. In addition, the supermarket could be faced with a fine from the authorities, if the failure is detected by the authorities. The energy price will have to be used as a base for choosing the weight R for the inverted COP term. The importance of the inverted COP is directly coupled to the energy price. The weight on the switch frequency term, S, is based on the price of replacing and maintaining a compressor. The cost associated with the maintenance and potential replacement of a compressors contains a significant amount of hidden cost. The imitated cost of maintenance and replacement of a compressors does for example not include the cost of down time of the refrigeration system due to the maintenance or replacement. These issues provide the performance function with significant freedom for manipulation based on intuition.

3.1.3.1 Characteristics of the performance function

The presented performance function is equipped with notable properties. The performance function is solely based on measurements and does not require detailed knowledge of the particular refrigeration system. The output of the performance function can not be considered as an absolute value even if the output can be interpreted as a cost of operation. Thus, comparing random supermarkets should be done while considering the differences of the systems. The performance function can easily be adapted to include other measurements without compromising any of the methods presented in this research project. The output of the performance function can be used for optimisation purposes as described in the papers B, C, D, E and F. However, another important potential application of the performance function is to use the output as a presentation of the performance for the end user of the refrigeration system.

Moreover, the idea of including the switch frequency in the performance function can be justified by the plots in Fig. 3.1, where the performance function is plotted for data from a real supermarket. The plots covers a time period where the supermarket closed which creates a change in operation point for the system. The blue plot is the proposed performance function whereas the red plot is omitting the contribution from the switch term, which corresponds to traditional approaches toward performance evaluation for a refrigeration system, i.e. only using the control errors and the COP to evaluate the system. It can be verified from the plots that different conclusion can be made depending on which performance measurement is used. The blue curve indicates that the performance of the system is degraded when the operation point is changed which is in contrast to what is indicated by the red curve. Thus, including the switch term contributes to a more meaningful notion of performance.

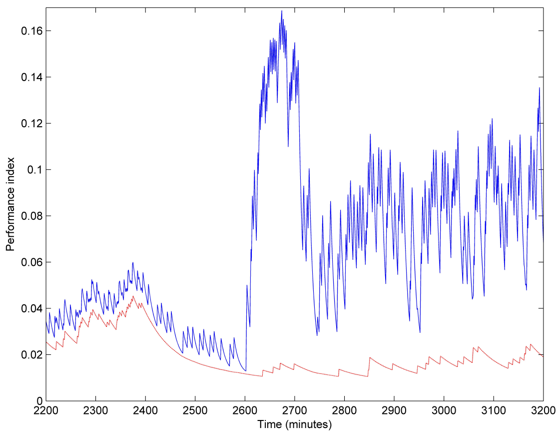

Figure 3.1: Performance index for a real supermarket system, including the transition in operation point due to the closing of the supermarket

3.2 Performance improvement

The section presents the contributions related to performance improvement. Dissemination of the contributions has mainly been done in the papers B, C, D, E and F. State of the art will be presented along with the relevant contributions.

As mentioned in Problem formulation one of the initiating problems for the project can be formulated as: How can sufficient refrigeration be provided with as low cost as possible? First of all, without any well defined notion of the performance of the refrigeration plant almost any solution can claim to improve the performance towards an optimum. Therefore, section 3.1 presents a performance assessment approach. However, even with a predefined notion about performance a solution can be reach by many different methods. The focus in this project has been to deal with performance issues that can be tackled by the control system. Hence, for example suboptimal operational performance due to a choice of an ineffective compressor has not been considered in this project. In other words, the refrigeration system is accepted as it is and the task is then to control the system to provide the best operational performance under the given circumstances.

Improving the performance of a system can be approach as an optimisation problem, since the ultimate goal is to provide optimal operation. However, techniques for optimisation through control often requires a dynamic model of the system, as in for example [7], [30]. Moreover, due to the information level existing at the time when the controllers are designed for a supermarket refrigeration system, the aim has been to avoid model-based solution method for performance. The contributions related to performance improvement has been targeting both static and dynamic performance issues. Improving the dynamical performance of the system has been targeted in the papers B, C E and F. The static performance has addressed in D and F.

Performance improvement or optimisation techniques are normally gradient based and therefore utilises the derivative of the performance function. The problems with these methods are that the gradient of the performance function might not be available and providing an analytical derived gradient might be infeasible. In addition, numerically calculating the derivative of the performance function is problematic due to noise and non-smoothness. Thus, the methods for performance improvement presented in this section can all be considered as derivative-free methods.

3.2.1 Dynamic performance improvement

Improving the dynamic performance of a supermarket refrigeration system by changing controller parameters creates the need for active performance monitoring. The reason is that even though the refrigeration system is exhibits dynamic behaviour a significant amount of the time, the majority of the time the system will be in a steady state like operation. Hence, changing a controller parameter will not affect the control error and thereby not affect the performance function. Therefore to be able to optimise the dynamic behaviour of the controllers the system needs to be excited. The idea of exciting a system originates from the field of active fault diagnosis, see for example [33, 34, 39]. The setup for active system monitoring used in this project is described by Fig. 3.2. The closed loop

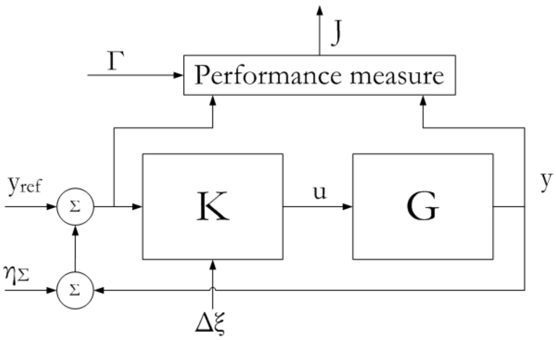

Figure 3.2: The general active system monitoring setup

under investigation for a potential performance improvement is shown in Fig. 3.2, where K denotes the controller and G denotes the system. The controller output is denoted by u and the measured output from the system is denoted by y and the reference is denoted by y_{ref}. The parameter change of the controller is denoted by $\Delta\xi$. The contributions to the performance function, from the other sub-systems, is denoted by Γ and the excitation signal is denoted by η_Σ. The novel idea is then to monitor the global performance function to evaluate current parameter setting of the controller.

3.2.1.1 Improvement clarification

The active system monitoring scheme has been used to clarify if the global performance could be improved by changing the controller parameters of a controller in a curtain closed loop. The setup used for this purpose can be seen in Fig. 3.3. The difference between Fig. 3.2 and 3.3 is that the excitation signal

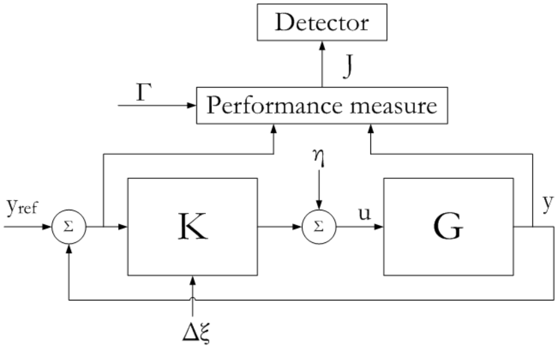

Figure 3.3: The active system monitoring setup for performance improvement clarification

has been moved and that a detector has been added. The excitation signal can in theory be placed where ever it is convenient. The placement of the excitation signal should however be taken into account when designing the signal. The idea behind the setup shown on Fig. 3.3 is to excite the closed loop and then change the controller parameters while measuring the global performance. A statistical change detector is then applied to the performance measuring to ensure that a significant change in performance has occurred. The reason for using a statistical change detector is to ensure that the amplitude of the excitation signal can be kept at a reasonable level and to limit the impact of noise. The task of the detector is basically to decide between the two which are formulated as follows:

$$\mathcal{H}_0 \quad : \quad \bar{J}_0 + w[n] \tag{3.5}$$

$$\mathcal{H}_1 \quad : \quad \bar{J}_1 + w[n] \tag{3.6}$$

In (3.5) and (3.6), $w[n]$ denotes the noise contributions and $\bar{J}_0$ denotes the initial mean value of the performance function and the estimated mean value after a change is denoted by $\bar{J}_1$. The hypothesis test will only detect a change in the mean values, and therefore to decide if the performance has been improved or degraded further analysis is required. Evaluating whether the performance has been improved or degraded is done by checking if $\bar{J}_0 > \bar{J}_1$ or if $\bar{J}_0 < \bar{J}_1$. The performance has been improved if $\bar{J}_0 > \bar{J}_1$ is true and if $\bar{J}_0 < \bar{J}_1$ is true the performance has been degrade by the parameter change in the controller. The entire procedure of performance improvement clarification can be formulated in the following steps:

1. Excite the closed loop under investigation to ensure active dynamic behaviour of the controller.

2. Apply a parameter change to the controller.

3. Detect whether a significant performance change has occurred.
4. If a change has occurred, then determine whether the performance has improved or degraded.

The statistical change detector that has been used is a generalised likelihood ratio test (GLRT), see [24]. The GLRT has been chosen because it can detected unknown mean changes which is in contrast to the cumulative sum,(CUSUM), which does require knowledge about the expected change in mean value, see [4] or [5]. Moreover, the GLRT does also provide an estimate of the mean values, which is needed to determine whether the performance has improved or degraded when a change has been detected. The GLRT basically decides $\mathcal{H}_1$ if the likelihood ratio, L_G, exceeds the predefined threshold, γ in (3.7).

$$L_G(J) = \frac{p(J; \bar{J}_1, \mathcal{H}_1)}{p(J; \bar{J}_0, \mathcal{H}_0)} > \gamma \tag{3.7}$$

The probability density function in (3.7) is denoted by $p(\cdot)$ and described by (3.8).

$$p(J; \bar{J}_i, \mathcal{H}_i) = \frac{1}{2\pi\sigma^2} \texttt{exp} \left[-\frac{1}{2\sigma^2} \sum_{n=0}^{N-1} (J(n) - \bar{J}_i)^2 \right] \tag{3.8}$$

In (3.8) $i \in \{0, 1\}$ and the window size and variance is denoted by N and σ^2, respectively. To achieving the theoretical optimal performance of the GLRT, with respect to time to detect and probability of false alarms, the noise contributions, $w[n]$, in (3.5) and (3.6) has to be white Gaussian noise. The fulfilment of this requirement has not been addressed in this project. The reason is that time to detect is not critical in the setup and therefore the threshold can be chosen conservatively with only probability of false alarms in mind. Missing an alarm is not hazardous for the system and the notion of firing a false alarms can not be interpreted in the traditional way. Not detecting an alarm that should have been detected will only provide the conclusion that the parameter change did not have any effect on the global performance and therefore the parameter change can not be justified. When considering the problem of firing of false alarms it should be considered that the parameter change is imposed by algorithm and it is therefore known when to expect a change. Hence, the detector will only be active when a change is expected. The main issues when choosing the threshold is actually the probability of making the wrong decision based on an alarm is more relevant in this application of the GLRT, since it implies that a parameter change that deteriorates the performance can be interpreted as a performance improvement.

The contribution related to performance improvement clarification is utilising ideas from active fault diagnosis to clarify that whether a certain parameter change in a controller changes the plant-wide performance significantly. Instead of using a traditional residual generator the plant-wide performance measure has been used as input to the GLRT. In addition, the GLRT has been extend to become a double sided test.

Justification for the active performance monitoring scheme can be seen on Fig. 3.4, which shows a test of the performance improvement clarification setup. In the first part of the simulation external excitation is not applied to the system and therefore the change in controller parameters does not influence the plant-wide performance function. However, at 41000 seconds the excitation of

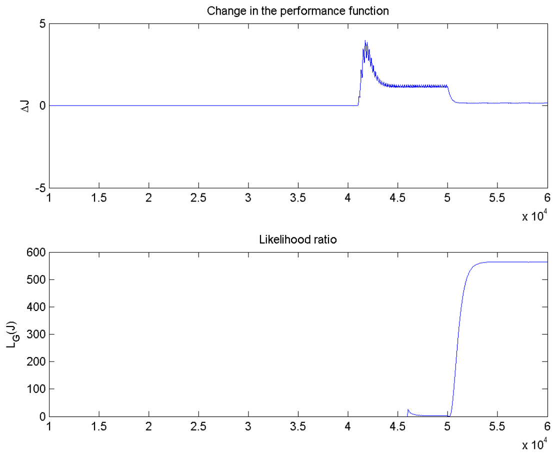

Figure 3.4: Change in the performance function, $\Delta J = J - \bar{J}_0$, likelihood ratio, $L_G(J)$. Simulation without and with excitation of the system. Controller parameters changed at 20000, 30000 and 50000 seconds. Excitation started at 41000 seconds

the closed loop system is initiated, which can be verified from plot of the air temperature in the display case, see Fig. 3.5. After the excitation has been initiated the same change of controller parameters is applied and is now affecting the plant-wide performance.

3.2.1.2 Searching the parameter space

Another approach towards improving the dynamic performance has been proposed in this project. The idea is to use a search technique which is derivative-free and searches the parameter space to find the best solution. Algorithms with

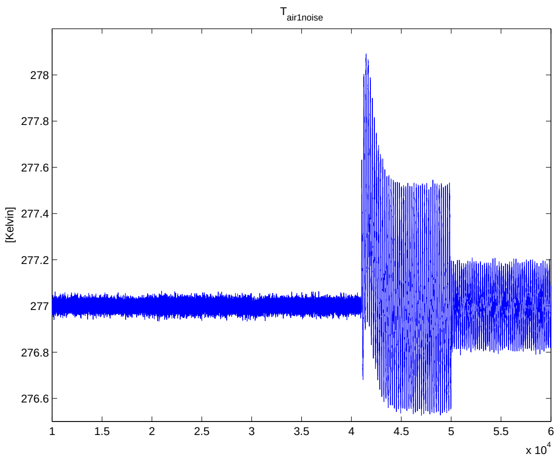

Figure 3.5: Air temperature in the display case when the parameters are changed. Simulation without and with excitation of the system. Controller parameter changed at 20000, 30000 and 50000 seconds. Excitation started at 41000 seconds

this properties has recently been used for optimisation in the literature as for example, generic algorithm [11], particle swarm optimisation [25], ant colony optimisation [9], simulated annealing [37] and tabu search [38]. These algorithms have shown high capability of searching for global minimum in different engineering applications [6, 45]. In this project, invasive weed optimisation, (IWO), has been proposed as a method for searching the parameter space, which has been introduces in [31]. The IWO algorithm is a bio-inspired numerical optimisation algorithm. The algorithm basically tries to mimic the natural behaviour of an invasive weed, when the weed is trying to colonising an area, and therefore tries to find the best positions for the weed. The idea in this project has been to utilise IWO to find the best parameters for the controllers with respect to the plant-wide performance measurement.

The general idea behind the IWO algorithm is that invasive weeds invade a field by means of dispersal and utilise the spaces left between the actual crop. The invading weed takes advantage of the unused resources in the field and develops into a flowering weed, which eventually will spread its seed and thereby continue the invasion of the field. The principle of survival of the fittest is the applied and the and only the best weeds will manage to flower and re-spread. Since the field is has limited resources this procedure will continue until the some maximum is reach. How this is translated into an optimisation algorithm will be described hereafter. To explore a parameter space random parameter sets are selected. Each of these parameter sets are then tested and evaluated. These two steps corresponds to randomly spreading the seeds of the weed and then waiting for the flowering of the weed. The survival of the fittest is also applied in the algorithm which basically means that only the parameter sets that provides good performance of the system will survive. Thus, the random selection of new parameter sets will be in the neighbourhood of the parameters that provided good performance in the last generation. The in the next generation are then chosen with in a certain radius of the parameters sets that has proven good enough to proceed. This of course corresponds to the weed spreading seeds. The algorithm will decrease the radius that the parameters are spread within in each generation. The task of the IWO algorithm has been to improve the dynamic behaviour of the system and therefore excitation of the system is also required when using the IWO algorithm. The method proposed for design of the excitation signal will be presented in 3.2.1.3.

The contribution related using IWO for optimising the global performance lies within the application of the algorithm. The IWO setup was tested on the simulation model presented in 2.2. However, in contrast to what the general idea behind the IWO algorithms is, an initial guess has been supplied as one of the seed in the first generation. This provides the algorithm with a very good staring point and the time that the system is operation suboptimal is reduced. In addition, choosing the parameter space sufficiently small also helps the IWO algorithm. With these changes to the IWO setup it is expected that it will be feasible to utilise the algorithm on a real system.

The result of using the IWO algorithm to find the parameters for a PI controller in a display can can be seen on Fig. 3.6 and 3.7. The two plots shows how the worst and best values of the controller parameters evolves. In can be verified on Fig. 3.6 and 3.7 that both the worst and the best choice of parameter converges. The parameter space is plotted on Fig. 3.8 from which the it can bee seen that the initial guess is actually close to the final result.

3.2.1.3 Design of the excitation signal

Both contributions described in subsection 3.2.1.1 and 3.2.1.2 rely on that the system dynamics are being excited. The gorverning idea is that the proposed methods should be able to run on a supermarket refrigeration system in operation. Therefore, design of the excitation signal is of course important. The combined excitation of the subsystem under investigation, η_Σ, shown on Fig. 3.2 can be described by (3.9).

$$\eta_\Sigma = \eta_{ext} + \eta_w \tag{3.9}$$

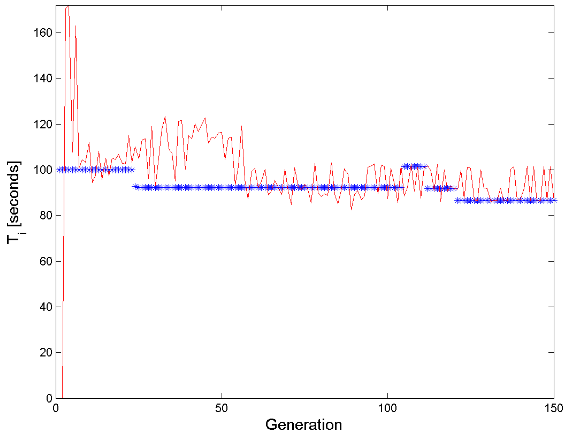

Figure 3.6: Evolution of T_i. The red line plots the evolution of the worst choice of T_i and the blue line plots the evolution of the best choice of T_i

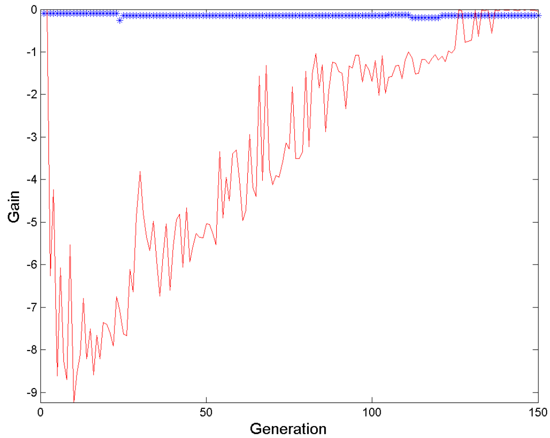

Figure 3.7: Evolution of k_c. The red line plots the evolution of the worst choice of k_c and the blue line plots the evolution of the best choice of k_c

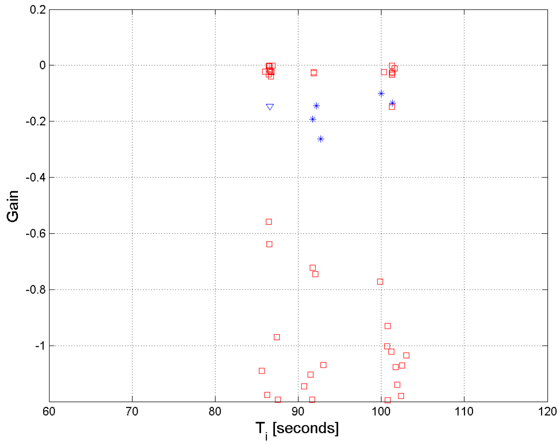

Figure 3.8: T_i versus k_c, The blue triangle indicates the end result for the parameters, $k_c = -0.1877$ and $T_i = 60.6972$.

In (3.9) the sum of all the excitation sources is denoted by η_Σ and η_{ext} denotes the externally applied excitation. The noise contribution, generated by noise sources like the other subsystems in the refrigeration plant, is denoted by η_w. The use of external excitation signal is only necessary when $\eta_\Sigma = \eta_w$ becomes insufficient as excitation. When there is a need to inject external excitation the task is then to design a signal, η_{ext}, that ensures that η_Σ has the sufficient impact in the closed loop under investigation. To generalise the abstraction presented in (3.9), η_{ext} should be considered as a set of signals, $\{\eta_1, \cdots, \eta_m\}$.

In the following design process is based on the facts listed hereafter:

1. The closed loop consists of a PI controller and a physical system, here a display case, for which the dynamics are unknown.
2. The parameters of the PI controller are known.
3. The existing controller is stabilizing the closed loop.

In addition the design process relied on these assumptions:

1. The system can be described sufficiently using a first-order model plus dead time (FOPDT).
2. The response time delay to an abrupt change in control action is known.

The first assumption can be verified by consulting [18, 43, 44] and the time delay can be measured with sufficient accuracy on the system.

Two main properties has to be chosen for the excitation signal and they are the frequency range and the amplitude. The frequency range is important to ensure that relavant dynamics of the closed loop under investigation are being excited. Secondly, the amplitude has to be chosen to ensure that the closed loop is excited significantly without compromising the operation of the closed loop significantly. Thus one of the problems is to design a set of signals, $\{\eta_1, \cdots, \eta_m\}$, which frequency range corresponds to the frequency range of closed loop system under investigation. Since the dynamic of the controlled subsystem is unknown in advance the design is not a trivial task. The knowledge with respect to the controlled subsystem is limited to a structural level. In other words gain and time constants are unknown. Thus, the design methods such as the ones introduced in [33] and [40] cannot be directly applied since they rely on model knowledge.

As mentioned before the aim is that the active performance monitoring setup should run online on a supermarket refrigeration system and thus compromising the operation significantly is unacceptable. The impact of the system should therefore be taken into account in the design phase of the excitation signal. In addition the task of choosing the amplitude for the external excitation signal is an online task, because the constraints related to the amplitude are dependent on the operation point and the actual noise level.

For the design of the excitation signal a closed loop system, which is considered a subsystem in the supermarket will be considered. In the following a display case is used as the considered closed loop. Hereafter, the notation will be described.

The dynamics of the subsystem are described by a FOPDT process model:

$$G_k(s) = \frac{k_{p_k} e^{-T_{d_k} s}}{\tau_k s + 1},$$

where T_{d_k} denote the time delay and k_{p_k} and τ_k) denotes the gain and time constant, respectively. The corresponding controller $C_k(s)$ is a PI type, i.e.

$$C_k(s) = k_c + \frac{k_c}{T_i s}.$$

The closed loop transfer function for the subsystem is then given by:

$$H_k(s) = \frac{C_k(s)G_k(s)}{1 + C_k(s)G_k(s)}.$$

For a given subsystem the task is to find an appropriate set of excitation signals $\{\eta_1(t), \cdots, \eta_m(t)\}$ and then use them to excite the subsystem and thereby providing the opportunity to improve the dynamic performance of the subsystem. The performance improvement is achieved by changing the controller parameters $\{k_{c_k}, T_{i_k}\}$ so that the performance function $J(t)$ is minimised, i.e.

$$\begin{aligned} &\underset{k_{c_k}, T_{i_k}}{\textit{minimise}} \quad J(t) \\ &\textit{subject to} \quad y_k(t) = (h_k * \zeta)(t), \quad \zeta(t) = \sum_{i=1}^{m} \eta_i(t). \end{aligned} \tag{3.10}$$

In (3.10) y_k denotes the output of the k^{th} system. '$*$' represent the convolution and represent the time domain description of $y_k(s) = H_k(s)\zeta(s)$, and the excitation signal is denoted by $\zeta(t)$.

The problem providing a solotion to (3.10) is discussed in 3.2.1.1 and 3.2.1.2 to focus in this subsection is the design of the excitation signal.

Designing the excitation signal has been spilt into two different tasks. The two tasks are, determining the frequency range and secondly the amplitude of the signal. The two task are teated separately. Thus, the choice of frequency range will be described first.

The following abstraction is introduced:

$$\zeta(t) = \sum_{i=1}^{m} \eta_i(t) = \sum_{i=1}^{m} \eta(\omega_i t) \tag{3.11}$$

In (3.11) ζ is a sum of identical functions which are operating with different frequencies ω_i. The candidate function is chosen to be a sinusoid as follow:

$$\eta_i(t) = A_i \sin(\omega_i t). \tag{3.12}$$

Utilising the fact that the controller parameters, i.e. k_c and T_i, are known, renders it possible to estimated the corresponding system parameters, i.e. τ and k_p, by assuming that the controller parameters have be designed for a desired gain and phase margins. To get an idea about what frequency range the excitation signal should lie within an estimation of the bandwidth for the closed loop under investigation is required. The phase and gain margin requirements has been used as a start point in order to provide a qualified guess on the system bandwidth. Estimating the bandwidth is based on the method for controller design presented in [19], which is a method that provides the parameters for a PID controller and ensures that the gain and phase margin specification are obeyed for a FOPDT process model. The procedure has been changed slightly to fit the problem of estimating the bandwidth based on the knowledge of the controller parameters instead of the system parameters.

The setup introduce in 3.2 has been simplified to the diagram shown on Fig. 3.9 to ease the estimation of the bandwidth based on the alternated methods from [19].

Figure 3.9: Block diagram of the simple closed loop

By manipulating the equations from [19] the time constant and the gain of the system can be calculated by (3.13) and (3.14), respectively.

$$\tau = \left(\frac{1}{T_i} + \frac{4 \cdot \omega_p^2 \cdot L}{\pi} - 2 \cdot \omega_p\right)^{-1} \tag{3.13}$$

$$k_p = \frac{\omega_p \cdot \tau}{A_m \cdot k_c} \tag{3.14}$$

The calculation of the phase crossover frequency is done using (3.15).

$$\omega_p = \frac{A_m \phi_m + \frac{1}{2}\pi A_m (A_m - 1)}{(A_m^2 - 1) \cdot L} \tag{3.15}$$

The parameters estimated in (3.13) and (3.14) can used as the best guess to compute the cutoff frequency, denoted ω_c, for the closed loop. Since the method relies on some assumption the parameters of the real system might deviate from the estimated parameters by up to 100%. Thus, to ensure a sufficient excitation of the system a set set of frequencies should be chosen. The set of frequencies are chosen based on the strategy described by (3.17) and (3.16). The frequencies are chosen by assigning ω_c in the following way:

$$\omega_{\frac{m+1}{2}} = \omega_c \tag{3.16}$$

with m chosen to be an odd number and then spreading the different frequencies using (3.17).

$$\omega_i = \frac{1}{10}\omega_{i+1} \qquad \forall i = 1, \cdots, m-1 \tag{3.17}$$

After choosing the frequency range the amplitude of the excitation signal should be chosen. The procedure for that will be described in the following.

The allowed range of a signal in the refrigeration industry is usually given by the constraints on the acceptable set-point for that particular closed loop. Therefore, by interpreting the PI controller as a sum of two terms, the P-term and the I-term. The constraints on the amplitude of the excitation signal can be described by (3.18) and (3.19). The accepted range of the set-point is given by SP_{upper} and SP_{lower}.

$$SetPoint + ||\zeta||_\infty < SP_{upper} \tag{3.18}$$

$$SetPoint - ||\zeta||_\infty > SP_{lower} \tag{3.19}$$

The function ζ is a bounded functions due to the definition in (3.11) and (3.12), and therefore extrema exists. Assuming that the closed loop is has reached steady state the contribution from the P-term in the PI controller is negligible and the I-term is the only one utilised to maintain set-point tracking. The following has been proposed as a method for choosing $||\zeta||_\infty$:

$$||\zeta||_\infty = \alpha \left| \frac{\textit{I-term}}{\hat{k}_{p_k}} \right|, \tag{3.20}$$

In (3.21) $\alpha \in \{0.01 \cdots 0.25\}$ and $\hat{k}_{p_k}$ is the estimated system gain. The choice of α should by as high as possible considering that α is constraint by 3.18, and 3.19, which cannot by violated. Choosing $\alpha = 0.15$ would be a typical choice. The correct amplitude for ζ is ensured by defining the amplitudes of the signals in (3.12) by (3.21).

$$A_i = \alpha \left| \frac{\textit{I-term}}{m\, \hat{k}_{p_k}} \right| \quad \forall k \in \{1, \cdots, m\}. \tag{3.21}$$

The design procedure for the excitation signal is not connected with the procedure for improving or optimising the performance. Thus, the choice of improvement or optimisation method can be handled after the design of the excitation signal.

3.2.2 Improving Steady State Performance

This subsection deals with the contributions related to the improvement of the steady state performance of a supermarket refrigeration is. The idea is presented in details in the papers D and F. Since refrigeration systems spends most of their operational time in steady state it is valuable to ensure optimal static performacne of the systems. Static and dynamic performance improvement rely on different parameters and therefore the different and thus different parameters will be subject to change depending on the improvement goal. When improvement of the static performance of the plant is in focus it is beneficial to find the subsystem with the most significant impact on the plant-wide performance. Thereafter, it is of course important to find the variable in that subsystem that address the condition for the operation, and they are typically the set-points.

To describe the improvement approach the performance function (3.2) has been rewritten ad:

$$J(\phi(t)) = \sum_{i=1}^{I=K+M+L} J_i(\phi(t)), \tag{3.22}$$

In (3.23) $J_i(\phi(t))$ denotes the local performance function for the i^{th} subsystem which is dependent on the set $\phi(t)$ that can be defined as:

$$\phi(t) = \left\{ Po_{ref}(t),\ \dot{Q}_{airload}(t) \right\}, \tag{3.23}$$

In (3.23) the controllable variable is Po_{ref} which denotes the suction pressure reference for the compressor rack controller. The uncontrollable variable is the heat loss the surroundings of the display cases in the sales area, which is denoted by $\dot{Q}_{airload}$. Moreover, the heat is also considered to be the main disturbance in the system.

Formulating the improvement of steady state performance as set-point optimisation can be done as follows:

$$\min_{Po_{ref}} J(\phi(t)) \qquad \forall\, \dot{Q}_{airload} \tag{3.24}$$

The focus in is to find the suction pressure $Po_{ref}(t)$ that minimises the plant wide performance, $J(\phi(t))$ independent of the heat loss/disturbance $\dot{Q}_{airload}$. To gain more knowledge about the search space for the optimisation some simulations has been carried out. The simulations are describe in the following:

Gaining knowledge of the search space is done by changing the suction pressure reference in steps from $1.0 \cdot 10^5$ $[Pa]$ to $2.5 \cdot 10^5$ $[Pa]$ with a step size of $0.1 \cdot 10^5 [Pa]$ or the equivalent of changing the evaporation temperature approximately $2°C$. In each of these steps the heat loss, $\dot{Q}_{airload}$, has been varied by changing the ambient temperature of the display cases in steps from 18 $[°C]$ to 28 $[°C]$, with a step size of $0.5°C$. Since the dynamic behaviour is not in focus only the steady state values of the data from the simulation has been used.

The results from the simulation can be seen on Fig. 3.10 where the interpolated line is added to the data from the simulation. The proposed idea is to perform the sweep of the suction pressure at two different load situations and then compute interpolated lines and use them to adjust the suction pressure reference according to the load, $\dot{Q}_{airload}$. These interpolated lines should be placed

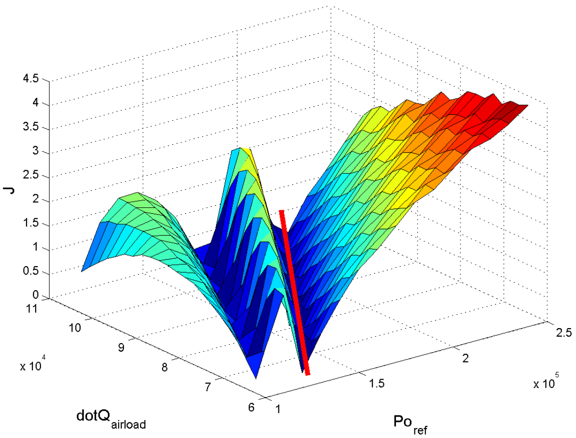

Figure 3.10: Optimal interpolated line

in the wallies of the performance function as shown on 3.10. The interpolated line can be calculated by:

$$Po_{ref} = Po_{ref,a} + (Po_{ref,b} - Po_{ref,a}) \cdot \frac{\dot{Q}_{airload} - \dot{Q}_{airload,a}}{\dot{Q}_{airload,b} - \dot{Q}_{airload,a}} \tag{3.25}$$

Hence, the optimisation problem can then be solved choosing the suction pressure reference Po_{ref} based on the a measurement of $\dot{Q}_{airload}$ and the interpolated line which has been computed based on the two sweeps. The load, $\dot{Q}_{airload}$, changes significantly between opening and closing hours of the supermarket, which enables the possibility to generate the two sweeps. Extending the algorithm to update the interpolation line when there has been change in the load due to a seasonal change might be beneficial.

Since the can be a more than one interpolated line a strategy for choosing the right line to base the adjustment of Po_{ref} on is required. The proposed method is to calculated the distance from the current operation point ($Po_{ref,current}$, $\dot{Q}_{airload,current}$) to the interpolated lines.

The method is explained in the following:

Calculation of the distance d_i between the current operation point ($Po_{ref,current}$, $\dot{Q}_{airload,current}$) and the i^{th} interpolated lines L_i : $Po_{ref} = m_i \dot{Q}_{airload} + b_i, \quad i \in \{1, 2\}$ is described by:

$$d_i = \frac{|Po_{ref,current} - m_i \dot{Q}_{airload,current}) - b_i|}{\sqrt{m_i^2 + 1}} \tag{3.26}$$

The interpolated line, L^*, that will be used to choose the new suction pressure reference, Po_{ref}, with is the one with the shortest distance to the current operation point as described by (3.26).

$$L^* = L_i, \quad \textit{such that:} \quad \forall j \neq i, \quad d_i < d_j \tag{3.27}$$

If the distances are equal, i.e. $d_i = d_j$ the line that passes through the higher suction pressures should be chosen.

The presented strategy is aim at small supermarket refrigeration systems where the rack is comprised of two compressors. However for larger system the compressor rack will be comprised of three or more compressors. Another, configuration of the compressor rack yield another surface of the performance function because the possible combinations of compressors increases. This also lead to an increase in the risk of having problems with excessive compressors switching due to the increased resolution in the compressor rack. In addition, in compressor rack with more that three compressors there is usually at least one compressor that is frequency controlled which thereby almost eliminates the switching problem. However, since the market for small and medium sized supermarket refrigeration system is of a considerable size the method has been generalised to a compressor rack comprised to three on-off controlled compressors.

The switch strategy for the compressors in the rack follows can by described by (3.28), where the three compressors are denoted by C_A, C_B and C_C, respectively.

$$\underbrace{\{C_A\}}_{Low} \longleftrightarrow \underbrace{\{C_A, C_B\}}_{Medium} \longleftrightarrow \underbrace{\{C_A, C_B, C_C\}}_{High} \tag{3.28}$$

Hence, a compressor is started to deal with the demand in the low capacity area and when the demand increases the second compressor is switched on and for higher demand all the compressors will be running.

There will therefore exist two capacity gaps for the compressor rack with three compressors i.e. one in the capacity area between $\{C_A\}$ and $\{C_A, C_B\}$ and the other one between $\{C_A, C_B\}$ and $\{C_A, C_B, C_C\}$. Thus, as in the two compressor case the performance will degrade in the capacity gaps which corresponds to an increase in the performance function. However, in the case of the three compressor rack there will exist three neighbouring interpolated lines instead of two line as in the two compressor case. The three interpolated lines can be calculated by using the same approach as for the two compressor case. Thus, the strategy presented in (3.26) and (3.27), for choosing the line to base the choice of the suction pressure reference on can be adopted for the three compressor case. The only difference is that three lines has to be taken into account, i.e. $i \in \{1, 2, 3\}$.

CHAPTER 4

Conclusion

The research work, documented in this thesis, focuses on plant-wide performance assessment, monitoring and optimisation. In this regards, the problem was formulated by three governing questions.

The first two governing questions, which are:

- *Is is possible to establish performance measures for control systems of a supermarket refrigeration system?*
- *Furthermore, is it possible to develop methods or algorithms to monitor the systems and analyse its performance by means of the established measures?*

were addressed positively due to the contributions regarding performance assessment in papers A and C. The plant-wide approach toward performance assessment were shown to be a feasible solution to the assessment problem. This approach ensured that the proposed performance improvement methods were considered at plant level. This was been shown in the papers B, D, E and F in which the performance assessment ideas were presented and utilised for performance improvement and thereby enabling a positive response to the last of the governing questions:

- *In addition is it possible to develop methods that can accommodate for a detected performance degradation of the system?*

The introduction of a plant-wide performance function for a supermarket refrigeration system was one of the important contributions in this thesis. The presented performance function covers quality-, energy-, and reliability-related criteria. Instead of evaluating a single subsystem locally the proposed method relied on assessing the plant-wide performance. In addition, the proposed performance assessment method does not rely on a model of the system and can therefore be applied to a real system relatively easy. The performance function provides the possibility of assessing the performance of a particular supermarket system over time and can therefore be used as tool for improving the performance for that particular system. Theoretically, the performance function can be used to compare two arbitrary supermarket refrigeration systems. However, that requires that the weights are chosen very carefully, which implies that the output of the performance function can then be interpreted as a cost of operation.

Performance improvement were addressed with different approaches, and ideas from different research areas has been applied in new contexts. Active performance monitoring was proposed to enable improvement of the plant-wide dynamic performance of the supermarket refrigeration system. Investigation of the properties of the output from the performance function has been used to propose a method for improving the static performance of the plant.

The generalised likelihood ration test, (GLRT), was used to test whether a parameter change, in a controller for a subsystem, creates a significant impact on the plant-wide performance. Searching for improvement of the dynamic performance has only been possible due to the proposed active performance monitoring setup, were an external excitation signal was applied to excite relevant dynamics of each subsystem. Another approach to improving the dynamic performance was to search the parameter space using the invasive weed optimisation, (IWO), algorithm with an initial guess for the parameters. A clear benefit of the IWO algorithm is that the method is derivative-free, and can therefore be applied online without the need of a system model.

Improving the static performance of a refrigeration plant has been done by analysing how to chose the set-point for the predominant control loop. The best static performance is achieved if the suction pressure reference is chosen with respect to the actual load on the refrigeration system. This result was achieved by analysing data from a simulation model of a supermarket refrigeration system.

4.1 Perspective

Ongoing research within the area is still needed to ensure that proposed performance function covers the relevant phenomenon. The proposed plant-wide performance assessment approach has been treated for a specific application and it would therefore be recommendable to generalise the idea and thereby create a general framework. A general framework would be easier to adapt for new objectives for the refrigeration system, e.g. heat recovery or smart grid interaction. In addition, the problem of performance assessment for a distributed control system which most of the time operates in steady state can be found in many applications in the process industry. Hence. the approach would also beneficial in other application in process industry.

Furthermore, a framework description of the plant-wide performance approach could be used to choose the performance improvement methods based on the properties of the framework. This could then ensure that the proposed performance improvement methods provides a general solution to the problem.

The proposed performance assessment method and the performance improvement methods presented in this thesis rely on some assumption and prerequisites that could be relaxed or investigated in future research. The assumptions that could be relaxed will be discussed in the following:

Weights for the performance function: The use of the proposed performance function requires a set of weights for each term. The underlying assumption in this work has been that reasonable weights exist. However, formal guidelines and design methods has only been minor attention. Thus, formal methods for choosing or designing weights remains an open research topic that could be pursue in the future.

PI controller: The assessment setup and the proposed performance improvement methods both assumes that the controllers in the subsystems are PI based. Reevaluation of the methods for different controller types could therefore be interesting.

First order plus dead time: For the design of the excitation signal for the active system monitoring setup it is assumed that the subsystem can be modeled using a first order plus dead time model. Creating design methods for the excitation signal that does not rely on this assumption could also be an opportunity to pursue.

Paper A

On the choice of performance assessment criteria and their impact on the overall system performance – The refrigeration system case study

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Abstract: The aim of this paper is to illuminate the impact of the choice of a system's performance criteria on the quality of the corresponding monitoring system's assessment results. Special attention is given to the performance issues that are caused by or can be solved by control actions. The compressor capacity gap issue in the supermarket refrigeration systems is used as a case study to elaborate on the problem through employment of both real life field data as well as simulation data. A performance function that can capture the compressor capacity gap problem is presented in the paper and used to evaluate both data from the real supermarket system and the data generated by the simulation model.

A.1 Introduction

The aim of many published scientific achievements has been to detect faults, identify them and, if possible, to accommodate them to the extend that the main functionality of the considered system is maintained and a minimum level of acceptable performance is obtained. The dominating reason has been that faults are the main causes for degradation of the system performance. Seen from an industrial point of view, an equivalently interesting issue that has a significant impact on the total system performance is the way different subsystems are devised to interact. Depending on the actual operating point, a combination of subsystems may force the other subsystems to operate in a less optimal fashion, which results in a decrease in the overall system performance. In this case no actual faults/malfunctions have occurred. So, in practice, a degradation in the overall system performance may not be caused by a fault but rather an inappropriate change in the operating conditions of the system. To accommodate for such situations there is a need for methods to carry out monitoring (and diagnosis) not only at the subsystem level but also at the system level. In this regard, establishing measures by which the system performance can be assessed becomes essential.

This paper contributes to this subject by providing an overall overview on the used/proposed measures that are commonly used. Moreover, the paper underscores the importance of the use of appropriate performance measures by providing an example from the supermarket refrigeration systems that illustrates the impact of the chosen performance measures/functions on the ability of detecting degradation of the overall system performance.

The paper starts with a general description of performance assessment in section A.2. Section A.3 describes the controller structure for a supermarket refrigeration system, and includes an introduction of a model of a refrigeration plant. In section A.4 the compressor capacity gap problem is introduced and described. The developed performance indicators are described in section A.5. Different scenarios with compressor capacity gap problems are illustrated with data from a real supermarket system and simulation in section A.6. A conclusion is provided as the last section of the paper.

A.2 Performance assessment in general

Performance assessment is considered to be essential in the process industry – Suboptimal performance can lead to a decrease in the quality of the products or/and superfluous energy consumption. Generally speaking, the criteria that are used to assess the performance of a system can be classified into three categories. These are:

Quality-related performance criteria Such criteria typically involve system variables with direct relation to the product quality. Many low-level controllers are actually based on this performance category. For example within control of refrigeration systems temperature is used at the regulatory level as a performance indicator of food quality and as a control parameter, see [8] and [28].

Energy-related performance criteria are often used at a higher level of the control hierarchy compared to the quality-based performance criteria which is more often used at regulatory level in the control hierarchy.

Reliability-based performance criteria are seldom used in the process industry (or refrigeration systems). However, new ideas are emerging that connect fault-tolerance to the total reliability of the system.

A.2.1 Performance measures

In the process industry several measures have been proposed/used to assess the considered system's performance. An overview of the most commonly used techniques Å¡can be found in [41] and [17]. Both papers discuss the minimum variance benchmark method and some of the extensions of the method. The minimum variance benchmark is popular because many processes have some sort of quality performance criteria which are related to a low variance of the process output. The notion of performance assessment in the control community is usually focused on investigating the control performance in comparison with some benchmark as described in [46] when dealing with model predictive control (MPC), and in [36], [17], [16] and [41], when dealing with the more general cases.

A.2.2 Performance assessment within refrigeration systems

The most commonly used parameter in the supermarket is the temperature in the display cases. However, when the task is to evaluate the plant-wide performance it becomes more difficult. Although the temperature is a good performance index with respect to food safety, it is not a useful index with respect to energy efficiency. To evaluate a change of performance with respect to certain changes in the plant the performance index has to be chosen correctly.

In the refrigeration community the focus is on the performance of the entire plant. However, the main interest is usually the static performance of the refrigeration plant meaning that dynamic issues are not considered. Paper [42] presents a way of describing the performance of a refrigeration system using static assumptions when calculating a theoretical reference and comparing it to measurements.

A common performance indicator in the refrigeration industry is the coefficient of performance, COP, which is defined as follows:

$$COP = \frac{Q_{cool}[w]}{W_{ref}[w]} \ , \tag{A.1}$$

where Q_{cool} denotes the heat removed from the cold reservoir and W_{ref} denotes the power consumed by the refrigerant system. In the papers [3], [29], [26] [49] and [47] COP is used as a performance indicator. An alternative method for performance assessment is presented in [28] where the number of compressor switches over time is used as a performance measure. This performance measure is interesting because switching of compressors is neither beneficial with respect to energy efficiency nor with respect to compressor life time (reliability).

A.3 Controller structure for a Supermarket refrigeration system

The layout for a supermarket refrigeration system is shown in Fig. A.1. The

Figure A.1: Simplified supermarket refrigeration layout

compressor rack supplies the flow in the system, and the control objective of the compressor controller is to maintain a specified suction pressure by switching the compressors on or off. The task of the condenser fans are to keep the condensing temperature at the correct level, which is done by switching fans on or off. The main task of the display cases is to maintain the temperature within the cabinet at a desired level. Ideally the capacities of the compressors are chosen so that the common operation points of the refrigeration plant can be handled with a fixed number of compressors running. In practice the compressor rack does not fit the operation points exactly. Therefore, the compressors will have to switch to satisfy the refrigeration load. Thus, the choice of compressors is a compromise and is usually in favour of a certain operation point. The result is that a refrigeration plant can experience higher switch frequency of the compressors at operation point where the composition of the compressor rack is not aligned with the cooling demand. This phenomenon is known as the capacity gap problem.

A.3.1 Modelling the refrigeration system

To illustrate the capacity gap problem a slightly modified version of the supermarket refrigeration system model from [28] has been adopted. The model has been changed so that the injection valve can be controlled continuously and thereby render a better model to illustrate the capacity gab problem. The model features a refrigeration system with two display cases, a suction manifold, a compressor rack and a condenser. The temperature in each of the display cases is described by (A.2), where (A.3) describes heat flow from the surroundings and into the display case. The other terms in (A.2) are described in detail in [28]. The ambient temperature of the display case, T_{amb}, is assumed constant since it corresponds to the indoor temperature of the supermarket. To enable the possibility of a continuously controlled refrigerant flow the mass flow into the evaporator is modelled using (A.4).

$$\frac{\mathrm{d}T_{\mathrm{air,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\mathrm{goods-air,i}}(\cdot) + \dot{Q}_{\mathrm{load,i}}(\cdot) - \dot{Q}_{\mathrm{air-wall,i}}(\cdot)}{M_{\mathrm{air}} C_{\mathrm{p,air,i}}} \tag{A.2}$$

$$\dot{Q}_{\mathrm{load,i}} = UA_{\mathrm{amb}} \cdot (T_{\mathrm{amb}} - T_{\mathrm{air,i}}) \tag{A.3}$$

$$\frac{\mathrm{d}M_{r,i}}{\mathrm{d}t} = OD_i \cdot \alpha \cdot \sqrt{P_c - P_{suc}} - \frac{\dot{Q}_e}{\Delta h_{lg}} \tag{A.4}$$

$$\frac{\mathrm{d}P_{\mathrm{suc}}}{\mathrm{d}t} = \frac{\dot{m}_{\mathrm{in-suc}}(\cdot) - \dot{m}_{comp}}{V_{\mathrm{suc}}\ \nabla\rho_{\mathrm{suc}}(P_{\mathrm{suc}})} \tag{A.5}$$

In (A.4) the opening degree of the expansion valve is denoted by OD, P_c denotes the condensing pressure, $\dot{Q}_e$ denotes the heat removed by evaporation and the enthalpy difference across the two-phase region is denoted by Δh_{lg}. For details about the modelling of $\dot{Q}_e$ and Δh_{lg} see [28]. The suction pressure is the only common state for all of the display cases, the suction manifold and the compressor, and its dynamics can be described by (A.5). In (A.5) the refrigerant density is denoted by ρ_{suc} and $\nabla\rho_{\mathrm{suc}}$ denotes the pressure derivative of the refrigerant density.

The mass flow rate into the suction manifold, $\dot{m}_{\mathrm{in-suc}}$ is described by:

$$\dot{m}_{\mathrm{in-suc}}(M_{\mathrm{r,i}}, T_{\mathrm{wall,i}}, P_{\mathrm{suc}}) = \sum_{i=1}^{N} \frac{\dot{Q}_{\mathrm{e,i}}(\cdot)}{\Delta h_{\mathrm{lg}}(P_{\mathrm{suc}})} \tag{A.6}$$

The temperature, T_e, denotes the evaporation temperature. As explained in [28] Δh_{lg}, ρ_{suc} and $\nabla\rho_{\mathrm{suc}}$ are all refrigerant specific functions.

The compressor rack is described by the following equation:

$$\dot{m}_{comp} = \mathtt{Cap} \cdot \frac{1}{100} \cdot \eta_{vol,i} \cdot \dot{V}_{sl,i} \cdot \rho_{suc} \tag{A.7}$$

In (A.7) `Cap` denotes the running compressor capacity of the rack, the volumetric efficiency is denoted by η_{vol}, and the swept volume flow rate is denoted by $\dot{V}_{sl}$. The condenser model contains no dynamic, it only defines a static condensing pressure and a static sub-cooling, which suggests an assumption that the condenser is controlled well enough to keep a constant condensing pressure and a constant sub-cooling.

A.3.2 Controller setup

The control setup is comprised of a temperature controller for each of the display cases which manipulate the opening degree of the expansion valve, OD_i, and a suction pressure controller that manipulates the running compressor capacity. The temperature controllers and the suction pressure controller are implemented as PI controllers. However, to emulate a rack of compressors a discretization is required. This is achieved by defining the compressor rack as being comprised of two compressors. Generally the $\texttt{Cap}$ in (A.7) has the following form:

$$\texttt{Cap} = \sum_{i=1}^{i=N} \delta_i \texttt{Cap}_i \tag{A.8}$$

with $\delta_i \in \{0, 1\}$ and $\sum_{i=1}^{N} \texttt{Cap}_i = 100\%$. In our application $N = 2$, $\texttt{Cap}_1 = 45\%$ and $\texttt{Cap}_2 = 55\%$. To avoid excessive switching of the compressors a hysteresis band is applied around each compressor step. The layout of the compressor rack is based on a real supermarket system which has the same layout of the compressor rack.

A.4 Compressor capacity gap problem

The compressor rack is comprised of n parallel-coupled compressors. Each of the compressors can supply an individual amount of cooling capacity, w_i, while operating. Hence, the total cooling capacity of the compressor rack is given by:

$$\Psi_{rck} = W \cdot u_\delta \tag{A.9}$$

The cooling capacity generated by the compressor rack, Ψ_{rck}, is described as the vector product in (A.9), where W is a row vector given as $W = [w_1, w_2 \ldots w_n]$ and u_δ is a column vector given as $u_\delta = [\delta_1, \delta_2 \ldots \delta_n]^T$, where $\delta_i \in \{0, 1\}$ denotes the switch states of the compressors. Hence, Ψ_{rck} is quantised and can only assume a number of values defined by the compressor sizes and the allowed combinations of compressors which are depending on the chosen switch pattern. Therefore to satisfy the required cooling capacity, Ψ_{req}, the controller will have to select the optimal compressor combination. Since Ψ_{req} is not measurable the controller will have to rely on other measurements as feedback. Assuming $\tilde{\Psi}_{req}$ can be used as an indirect measure of Ψ_{req}, the control law, for the compressor rack with two compressors, can be described as follows:

$$\Psi_{rck}(n) = \begin{cases} \psi_1 & \texttt{if } \tilde{\Psi}_{req} < \frac{\psi_2 - \psi_1 - h}{2} \\ \psi_2 & \texttt{if } \tilde{\Psi}_{req} > \frac{\psi_2 - \psi_1 + h}{2} \\ \texttt{else} & \Psi_{rck}(n-1) \end{cases} \tag{A.10}$$

The possible compressor combinations are denoted as ψ_i, and h denotes the hysteresis threshold. The values of Ψ_{rck} are given by the set of possible compressor combinations $S = \{\psi_1, \psi_2 \dots \psi_p\}$, i.e. $\Psi_{rck} \in S$, hence the compressor capacity gap problem will arise when $\Psi_{req} \notin S$. The use of an indirect measurement does also generate part of the problem. The assumption of $\tilde{\Psi}_{req}$ being an indirect measurement of Ψ_{req} does not hold under all circumstances. The indirect measurement can be described by (A.11),

$$\tilde{\Psi}_{req} = \Psi_{req} + \epsilon \tag{A.11}$$

where ϵ denotes the disturbances in the system. In refrigeration systems where both $\epsilon \geq h$ and $\Psi_{req} \notin S$ the control law in (A.10) will be compromised. The compressor capacity gap problem exits when the demanded cooling capacity cannot be satisfied by a single combination of the compressors. The problem is then to decide when to switch to ensure optimal operation. If the refrigeration system only has one display case the control problem becomes more simple since the compressors then can be controlled based on the temperature in that display case. However, when the system has more than one display case it will not be feasible to base the control for the compressors on a single temperature measurement. A common approach is to use the suction pressure as control variable, because it is a common variable for all of the display cases, as shown in Fig. A.1. Another argument for using the suction pressure is that it can be used as an expression of the load on the refrigeration system. However, the correlation between load and the suction pressure only holds in situations where the controller for the compressor is in steady state. Hence, the controller will not be able to maintain steady state when the actual load of the refrigeration system is in a capacity gap. Thus, the common control approach is not suitable in these situations because using the suction pressure as control variable will generate excessive switching under these circumstances. The control problem in a capacity gap therefore calls for another control strategy.

A.5 Performance indicators

For the supermarket refrigeration system the main performance criteria are food quality and power consumption. In addition, component wear is an important issue as it can reduce the component life time or increase the required service for the component.

The food quality criteria can be addressed by the temperature error in the display cases and supported by the error for the suction pressure. To address the remaining performance criteria COP can be considered as a performance indicator. However, neither the COP nor the temperature and pressure control errors cover the reliability criteria which are needed to address the compressor capacity gap problem. One way of capturing the problem arising from the quantisation in the compressor rack is to measure the switch frequency of the combined compressor rack. The switch frequency will increase if the desired compressor capacity lies within a capacity gap of the rack.

The switch frequency is not sufficient to describe the performance of the refrigeration system. Thus, the control error for the temperature controller will also be used as a performance indicator. To ensure a sufficient temperature difference the controller error for the suction pressure is also used as performance indicator.

To capture the performance problems for a refrigeration system the performance index will have to be based on a number of performance indicators. The performance criterion with respect to food quality is represented by the control error for the temperature controllers. In addition, the food quality performance is also represented in a more indirect way by the control error for the suction pressure. To address the second performance criterion, which is power consumption, COP is used. The COP is used as a performance indicator instead of the power consumption because the COP is also considering the generated cooling capacity. However, since the COP will increase when the relative cost is low it is the inverse of the COP that will be used in the performance function. The switch frequency is used to address the performance criteria of compressor wear.

The control errors are gathered in a vector, $\mathbf{e}$, where the elements are defined as follows:

$$e_1 = T_{ref,1} - T_{air,1} \tag{A.12}$$

$$e_2 = T_{ref,2} - T_{air,2} \tag{A.13}$$

$$e_3 = P_{sucref} - P_{suc} \tag{A.14}$$

The COP and the switch frequency can not be considered as control errors, they are however still included in the performance function.

$$J(t) = \sum_{i=1}^{N} ||\mathbf{e}(i)||_Q^2 + \sum_{i=1}^{N} ||\frac{1}{COP(i)}||_R^2 + \sum_{i=1}^{N} ||f_{sw}(i)||_S^2 \tag{A.15}$$

In (A.15) Q, R and S are the weights on the different terms. Choosing the weights is not a trivial task and it has to be done considering the importance of each term in the specific application. In addition, it has to be considered what the outcome of the performance function is used for. If the outcome of (A.15) is used for supervisory control of the plant, it will be important to choose the weights so that the first term in (A.15) is not given too much importance. The reason is that the task of minimizing the control errors should not be the main task of a supervisory controller, and in addition the task is already maintained by local controllers. Thus, choosing the weights to favour the second and third term in (A.15) is recommendable. Choosing the ratio between the weights R and S can also be a difficult task which can only be based on knowledge of the plant. Since neither COP nor the switch frequency are controlled elsewhere, sufficient weights are necessary if a high COP and a low switch frequency are desired.

A.6 Scenarios for the compressor capacity gap problem

This section presents results that describe different scenarios for the compressor capacity gap problem. The purpose of this section is to illustrate the compressor capacity gap problem. To justify the existence of the compressor capacity problem, the first scenario will be based on data from a real supermarket refrigeration system. The following scenarios are based on simulation results.

A.6.1 Scenario one

The first scenario, which is based on field data from a supermarket system in operation, shows how a difference in operation condition can change the switching behaviour of the compressor rack. The data are captured around closing time of the store and the reason for the change in operation point is that night covers are placed on the display cases to reduce energy consumption outside opening hours of the supermarket. The top plot of Fig. A.2 shows the suction pressure with a blue curve and the reference with a dashed red curve. The plot clearly shows that there is a significant change in the operation point for the compressor rack controller around 2600 minutes. The bottom plot of Fig. A.2 shows the running compressor capacity of the entire compressor rack, and it can be seen that the average capacity drops significantly around 2600 minutes. However, since the compressor rack is comprised of two compressors which are representing 45 % and 55 % of the compressor rack's full capacity, respectively, the change in operation point causes an increase in compressor switching.

To illustrate the compressor capacity gap more clearly the data from the supermarket has been used to calculate the performance using (A.15) with small modifications. The term in (A.15) which penalises the control error is only comprised of the suction pressure control error due to lack of temperature data. The lack of the temperature control errors is not considered an issue, since it is a fair assumption that the temperature errors will be kept constant in this scenario. Fig. A.3 shows two performance indices for the data from a real

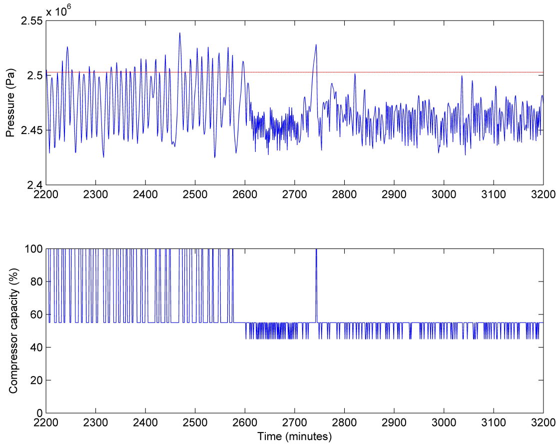

Figure A.2: Pressure and running compressor capacity, including the transition in operation point due to the closing of the supermarket

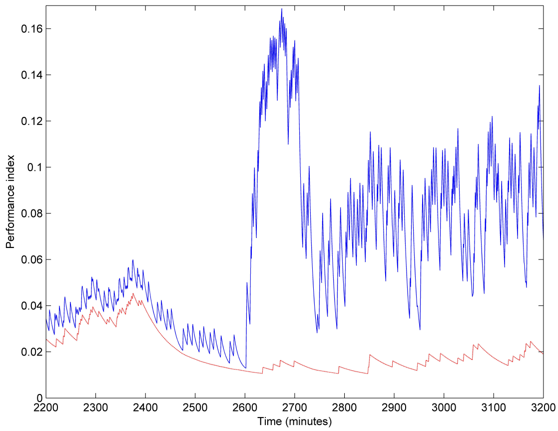

Figure A.3: Performance index for a real supermarket system, including the transition in operation point due to the closing of the supermarket

supermarket. The blue curve is generated by using the performance function (A.15) and the dashed red curve shows the results for the same performance function but with omitting switch frequency term. The latter corresponds to the traditional evaluation of the performance of a refrigeration plant. From Fig. A.3 it can be seen that the dashed red curve experiences a small decrease around the change of operation point. However, the blue curve increases significantly after the change of operation point. Hence, introducing the switch frequency term in the performance index can change the conclusion regarding the overall performance significantly.

A.6.2 Scenario two

The purpose of the second scenario is to emulate the effect of applying night covers to the display cases, as described using real data in A.6.1, by using the simulation model. Thus, the UA_{amb} is stepped for each of the display cases from 4000 [$^J/_{s\cdot m}$] to 3500 [$^J/_{s\cdot m}$], which corresponds to a reduction in the load. Hence, a decrease in compressor capacity is expected which can be verified in Fig. A.4 where the top plot shows the suction pressure and the bottom plot shows the running compressor capacity. Although the negative step in UA_{amb} corresponds

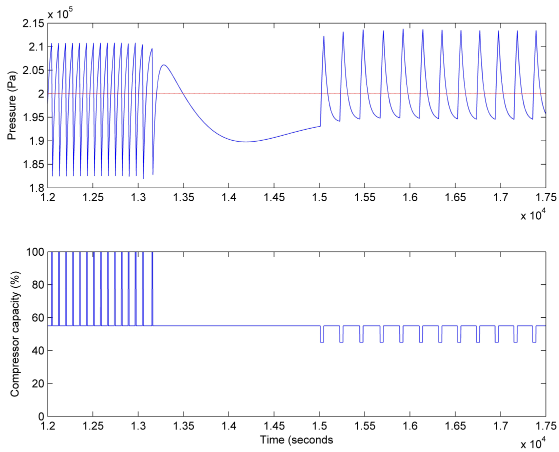

Figure A.4: Sunction pressure and compressor capacity with compressor capacity gap problem, and step in UA_{amb} at 13000 seconds

to a reduction in the load, it cannot be expected to change the outcome of the cost function significantly except from the change that can be achieved from a change in switch frequency. Fig. A.5 shows the two cost functions, where the dashed red curve is the cost function without the switch frequency term, and the blue curve is the cost function, where the switch frequency is included. As expected there cannot be seen a significant change on the dashed red curve after the step in UA_{amb}. However, a small increase in cost can be seen from the blue curve which is due to an increase in switch frequency of the compressor rack.

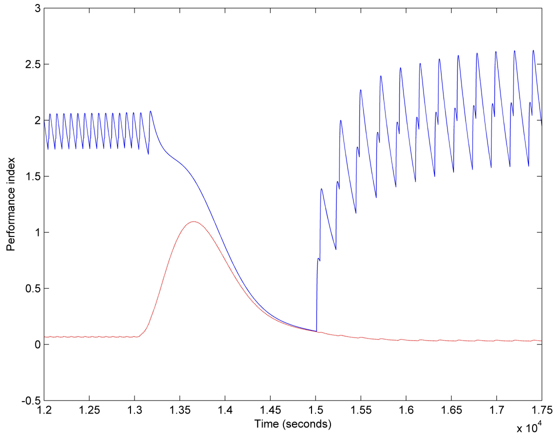

Figure A.5: Cost function with capacity gap problem, and step in UA_{amb} at 13000 seconds

A.6.3 Scenario three

In the third scenario the suction pressure reference is increased which should give an increase in COP and lower the required running compressor capacity. Hence, intuitively a decrease of the performance function is expected from the simulation. However, because the change in set point for the suction pressure, the system is forced into another operation point for the compressor rack controller, which results in an increase in switch frequency of the compressors. Fig. A.6 shows the simulation results from the third simulation. As expected the running compressor capacity is reduced significantly when the suction pressure reference is changed from $2e5$ [Pa] to $2.42e5$ [Pa] which corresponds to a change in evaporation temperature from 263.15K to 268.15K for the refrigerant R134a. The COP of the system is increased when the suction pressure is increased and the condensing pressure is kept constant. However, changing the operation point of the compressor rack to a less fortunate operation point increases the switch frequency. Fig. A.7 shows the two cost functions where the dashed red curve

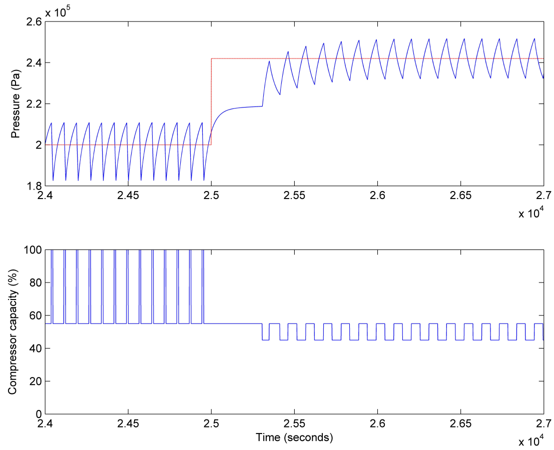

Figure A.6: Suction pressure and compressor capacity with compressor capacity gap problem, and step in the suction pressure reference at 25000 seconds

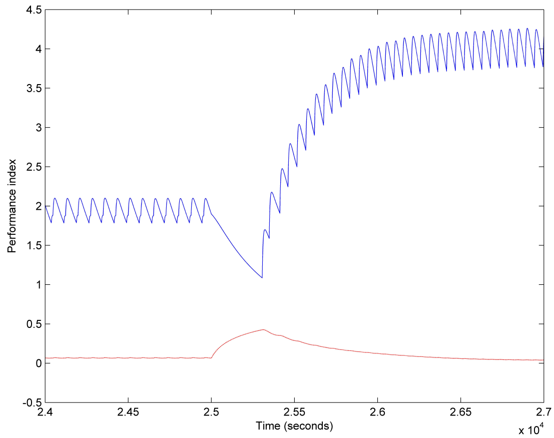

Figure A.7: Cost functions with the capacity gap problem, and step in the suction pressure reference at 25000 seconds shows the cost function, where the switch frequency term is omitted, and the blue curve shows the cost function, where the frequency term is included. It can be seen that the two curves will lead to two different conclusions based on the change of the suction pressure reference. The dashed red curve clearly implies that raising the suction pressure reference will decrease the cost of the operation. The blue curve, on the other hand, implies that the cost of the operation is indeed increased when the suction pressure reference is increased.

A.7 Conclusion

The main purpose of this paper was to highlight the importance of the choice of appropriate performance measures as these have a profound impact on the quality of corresponding monitoring system's assessment results. To elaborate on this topic the compressor capacity gap problem from the supermarket refrigeration systems has been employed. It is shown, through the use of both real-life field data as well as a simulation model, that choosing the appropriate performance measures is essential in order to capture the potential performance degradation. The paper introduced performance measures that covers quality, energy and to some extend reliability. The quality measure was constructed through the use of temperature deviations from the reference, and COP was used as an indicator for energy consumption. The reliability measure was partially covered by including the compressor switch frequency as a performance indicator. The three performance indicators where used in a performance function to show the extend of the compressor capacity gap problem.

PAPER B

Performance improvement clarification for refrigeration system using active system monitoring

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Abstract: This paper addresses the problem of determining whether a refrigeration plant has the possibility of delivering a better performance of the operation. The controllers are well-known but detailed knowledge about the underlying dynamics of the refrigeration plant is not available. Thus, the question is if it is possible to achieve a better performance by changing the controller parameter. An approach to active system monitoring, based on active fault diagnosis techniques, is employed in order to evaluate changes in the system performance under operation.

Keywords: Performance assessment, Performance optimisation, Active system monitoring, Refrigeration systems

B.1 Introduction

The operation quality of a supermarket refrigeration system is crucial to ensure the profit of the supermarket because the system enables sale of refrigerated goods and furthermore the running cost of the system is considerably high. Thus, ensuring high operation quality of the refrigeration plant is important. Achieving optimal performance requires performance measurement, assessment and possibly adjustment of the control system. In addition, determining a possible improvement potential is required. [13] presents an adequate performance function which captures the main features of the system. To utilise the results for reconfiguration of the controllers the improvement potential for the particular plant has to be evaluated. This paper describes a method that can be used to determine the improvement potential that the control system will be able to deliver, with respect to the operation quality.

The usual way to evaluate the operation performance of a given plant is to compare the achieved performance against a predetermine benchmark. These techniques require a model of the benchmark and are therefore hard to handle in applications where the knowledge of the system under assessment is restricted. This situation is exactly what the industry for supermarket systems are faced with. The design of control software is usually done without exact knowledge of the layout of the refrigeration plant.

Assessing performance based on a model of the system under assessment has been done in a large number of publications. In, [16], [41], [22], [17] and [36], the minimum variance benchmark is used to assess the performance and even though the minimum variance benchmark is considered to be a data driven method in the literature it does require a benchmark model. In [46] a model-predictive control benchmark is introduced. Common for these publications are that they all assume that enough information about the system is present, at the design phase for the assessment scheme, to derive a model.

This paper shows that the need of a model can be circumvented by utilising the fact that full knowledge about the controller is given at the design phase for the assessment scheme. In addition, the question that should be answered is whether the performance can be improved by changing controller parameters for a particular plant operating under a particular set of conditions. Answering that question has much higher value than determining whether an achieved performance is worse than some theoretical level.

In section B.2 the problem is formulated and described and the new approach is described in B.3. The basic supermarket refrigeration system is described in B.4. An illustrative example is presented in B.5 along with a description of the model used for simulating the supermarket refrigeration system. Section B.6 presents results from different test scenarios and the paper ends with a discussion in B.7.

B.2 Problem formulation

The main problem is to assess whether a certain closed loop subsystem has the parameter freedom to improve the overall performance of the entire system. Therefore, to address this problem a performance measure for the entire system is needed along with a method for determining the parameter freedom. A refrigeration system for a supermarket will most of the time run in steady state, hence not all parameter changes will be observable from the measurements or the performance measure. Thus, to ensure that the parameter changes will affect the behaviour of the measurements and thereby also the performance measure, excitation of different closed loop controlled subsystems is required.

A refrigeration system for a supermarket can be divided into a number of subsystems. The different subsystems influence the operation of each other in both positive and negative direction. Therefore, the evaluation of both a local and global performance is important, because evaluation of the local performance is not the same as evaluating the global performance. Changing the parameters in the controller for a specific closed loop might deteriorate the local performance. However, that does not necessarily imply that the global performance will also deteriorate.

The evaluation of the performance should ideally be done using as little knowledge about the system as possible. That is due to the fact that the structure of the control system is usually designed without any specific knowledge about the particular refrigeration system, i.e. detailed knowledge about the underlying system dynamic is basically non-existent. However, full description of all the utilised controllers are available. Therefore, basing the performance evaluation on the knowledge about the nominal controllers and omitting a reference model of the refrigeration system would be the ideal solution.

B.3 New Setup

The aim is to assess whether it is possible to improve the overall system performance. The approach to determine the improvement potential will be based on the performance measure introduced in [13]. As mentioned earlier a refrigeration system is usually operating in steady state conditions and will therefore not be affected by a change in a controller parameter unless the dynamics of the system is excited. A simulation model will be used to select a probe signal that can be used to evaluate the performance change created by a parameter change in the controller. The probe signal has to be designed to generate enough variations on the performance measures to enable an evaluation of a parameter change, but without compromising the operation of the refrigeration plant.

A block diagram of the setup can be seen on fig. B.1, which illustrates a closed loop under assessment plus the global performance measure and the detector. The parameter change of the controller K is denoted by $\Delta\xi$ and η denotes the excitation signal. The control input to the system is denoted by u and J denotes the performance measure which is directly used as residual by the detector to decide whether a given parameter change, $\Delta\xi$, has changed the overall performance measure. The input to the performance measure, from the other subsystems, is denoted by Γ.

B.3.1 Performance indicators

The performance indicators are basically chosen to cover three performance criteria which are food quality, energy efficiency and actuator life time. The indicator for food quality are the control errors, of the temperature controller in the display cases and the suction pressure controller, which are gathered in a vector described in (B.1).

$$\mathbf{e} = \begin{vmatrix} T_{\mathrm{ref,i}} - T_{\mathrm{air,i}} \\ P_{\mathrm{sucref}} - P_{\mathrm{suc}} \end{vmatrix} \tag{B.1}$$

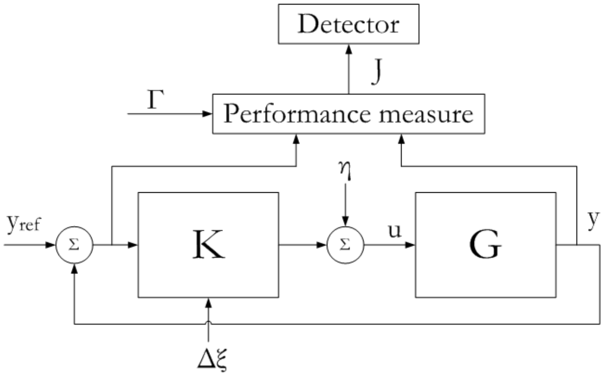

Figure B.1: Active performance assessment setup

The coefficient of performance, COP, which is described by (B.2), is used as an indicator for energy efficiency.

$$COP = \frac{Q_{cool}[w]}{W_{ref}[w]} \, , \tag{B.2}$$

In (B.2) the delivered cooling capacity is denoted by Q_{cool} and the electrical energy supplied to the refrigeration plant is denoted by W_{ref}. The indicator with respect to actuator life time is the switch frequency of the compressors in the compressor rack, which is denoted by f_{sw}. It is used as a simple indicator since switching of compressors reduces the life time of the compressors. Combining the three different performance indicators in a single performance function yields

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}(k)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP(l)}||_R^2 + \sum_{m=1}^{M} ||f_{sw}(m)||_S^2 \tag{B.3}$$

where Q, R and S denotes the weights that determine the impact of each term on the performance function. Since the COP is a number that will increase with efficiency the inverse is used to ensure that minimising the performance function is still the target. The choice of weights has to be done with respect to the application under investigation. The choice of weights will have a significant influence on which parameter changes can be detected using reasonable size of both the excitation signal and the parameter changes. In other words, the weights determine the definition of optimality for a particular system and should therefore be chosen with care. Even though the weights are usually chosen based on empirical knowledge they should not be considered as tuning parameters for the detector.

$$||\mathbf{e}||_Q^2 = \mathbf{e}^T Q \mathbf{e} \tag{B.4}$$

B.3.2 Active performance assessment

To ensure an adequate result from the excitation of the system some considerations have to be done to design the excitation signal η. Since the task of the excitation signal is to excite the dynamics of the controller and the system the frequency range of the signal has to be chosen with care. If the excitation signal is too fast the effect will not be visible on the output because of the low pass filtering effect of the closed loop system. In contrast if the frequency is chosen too slow it will not be possible to detect the change within reasonable time. In addition, the impact on the operation of the system should be minimal since the excitation is carried out under normal operation of the system. Hence, the operation quality must not be compromised. In this paper a sinusoid signal has been chosen, however any periodic signal with appropriate frequency and amplitude properties can be used. The excitation signal is given by:

$$\eta = A \cdot \mathtt{sin}(\omega t) \tag{B.5}$$

The choice of the frequency, ω, for the excitation signal is important for the method to work. It is worth mentioning that the frequency of the excitation signal should lie within the bandwidth of the closed loop both before and after the parameter change to ensure that the influence of the signal is not filtered away by the closed loop. However, since the bandwidth of the closed loop is unknown a conservative choice of both the excitation signal frequency and the size of the parameter change is recommended.

B.3.3 Detector

To ensure that a significant change in the performance measure has occurred a statistical change detector has been used. In addition, a statistical method is applied to ensure that a decision can be made with confident and without the need of unreasonable amplitude of the excitation signal η. The detector has to decide between two hypothesis which can be formulated as,

$$\mathcal{H}_0 \quad : \quad \bar{J}_0 + w[n] \tag{B.6}$$

$$\mathcal{H}_1 \quad : \quad \bar{J}_1 + w[n] \tag{B.7}$$

where $w[n]$ denotes a noise contribution and $\bar{J}_0$ and $\bar{J}_1$ denoted the initial mean value of the performance function and the estimate of the unknown change in mean of the performance function respectively. The outcome of this simple hypothesis test will only be that a change has occurred. To evaluate whether the performance has been improved or degraded a separate test will have to be applied. The test will be to check if $J_0 > J_1$ or $J_0 < J_1$ is true. If $J_0 > J_1$

is true the performance of the system has improved and an flag will be set to 1. In contrast if $J_0 < J_1$ true the performance has been degraded by the parameter change and the alarm state will therefore be set to -1. When the system dynamics is excited and a parameter in the controller has been changed then the detector has to

a) detect whether a significant change in the performance measure has been created by the parameter change and

b) determine whether the performance has been improved or degraded.

To fulfill the detection task stated above a statistical test method is needed due to the presence of noise. A well known test that can cope with detecting an unknown change in mean value is the generalised likelihood ratio test, (GLRT), see [24]. Oppose to the cumulative sum, see [5], which requires knowledge about the mean value after the change has occur, that knowledge is not required by the GLRT. In addition, a bi-product of the GLRT is an estimate of the mean values, which can be used to determine if an improvement or a degradation has been detected. Basically the GLRT decides $\mathcal{H}_1$ if the likelihood ratio, L_G, in (B.8) crosses the threshold, γ.

$$L_G(J) = \frac{p(J; \bar{J}_1, \mathcal{H}_1)}{p(J; \bar{J}_0, \mathcal{H}_0)} > \gamma \tag{B.8}$$

In (B.8) $p(\cdot)$ denotes the probability density function. The probability density function is given by:

$$p(J; \bar{J}_i, \mathcal{H}_i) = \frac{1}{2\pi\sigma^2} \mathtt{exp} \left[-\frac{1}{2\sigma^2} \sum_{n=0}^{N-1} (J(n) - \bar{J}_i)^2 \right] \tag{B.9}$$

where $i \in \{0, 1\}$ and N, σ^2 denotes the window size and the variance, respectively. To ensure the theoretical performance of the detector it is necessary to assume that ω in (B.6) and (B.7) is white Gaussian noise to fulfill the conditions for the GLRT. The fulfilment of that condition has not been treated in this paper. In addition, it is not considered to be an issue in this application of the GLRT, since the performance of the detector is non-critical. The choice of the threshold, γ, is usually a tradeoff between time to detect and probability of false alarms when the detector is used in a normal fault detection application. The threshold, γ, can be computed to fit a desired probability of false alarms by utilisation of a right-tail function, for details see [24] chapter 6. However, the use of the detector proposed in this paper requires another way of thinking of the quality of the detector. In the proposed setup missing an alarm is not hazardous and the notion of firing a false should also be reconsidered. Missing an alarm is not an issues in this setup because it is known when the parameter change in the controller happens. Hence the detector will only be active when a change in the performance measure is expected. On the other hand firing a false alarm is not a problem because as mentioned before the detector will only be active when a change is expected. Even though the normal tradeoff problem does not apply, it is of course still the target to chose γ so that the noise level in the performance function cannot trigger an alarm. In addition, it will still be desirable to chose a γ that enables the detector to fire an alarm within reasonable time of the parameter change that the detector is trying to detect. The threshold can be chosen relatively defensive with respect to the probability of false alarms because time to detect is not critical. The probability of making the wrong decision based on an alarm is more relevant in this application of the GLRT, since it implies that a parameter change that deteriorates the performance can be interpreted as a performance improvement.

B.4 Supermarket refrigeration setup

The supermarket refrigeration setup, which can be seen on figure B.2, is comprised of a number of display cases, a compressore rack and a condensing unit. The display cases are where the stored good are refrigerated and the condensing unit is where the heat removed from the stored goods is emitted to the surroundings. The compressors generates the flow of refrigerant and the control task for the compressore rack is to maintain a certain saturation temperature for the refrigerant. The saturation temperature determins the cooling capacity of the display cases. The temperature in each of the display cases is control by the inlet valve. The condensing pressure is controlled by fans to ensure that the heat can be transfered to the surroundings. The control of the saturation temperature is achieved by switching the compressors on or off. Hence, to avoid excess switching of the compressors their sizes should ideally be chosen to match the common load of the system with a fix number of compressors. However, the compressors cannot be chosen to fit the load exactly in practice. Thus, the compressors will have to switch on and of to accommodate the required refrigeration load. The supermarket refrigeration can be abstracted as a number of subsys-

Figure B.2: Simplified supermarket refrigeration layout

tems. The division into subsystems can be done as follows. Each display case including its control comprises one subsystem. The compressor rack including control is another subsystem and the condenser unit include the fans and the control is the third subsystem.

B.5 Illustrative example

To illustrate the performance assessment problem introduced in section B.2 a model of a refrigeration system is needed. The model setup given in [13], which is based on the model presented in [28], will be described shortly. In this model a refrigeration system comprised of; two display cases with individual inlet valve and temperature control, a compressor rack containing two compressors, and a condensing unit which is assumed to be able to keep the condensing pressure perfectly stable. The air temperature in the display cases can be described by (B.10), where the heat flow from the display case and to the surrounding air can be expressed by (B.11). The refrigerant mass flow rate into the i^{th} display case is model by (B.12), where OD, P_c, P_{suc}, $\dot{Q}_e$ and Δh_{lg} denotes the opening degree of the valve, the condensing pressure, the suction pressure, the heat removed by the evaporator and the enthalpy difference across the two-phase region, respectively. For details about the calculation of the unmentioned terms see [28].

$$\frac{\mathrm{d}T_{\text{air,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\text{goods-air,i}}(\cdot) + \dot{Q}_{\text{load,i}}(\cdot) - \dot{Q}_{\text{air-wall,i}}(\cdot)}{M_{\text{air}} C_{\text{p,air,i}}} \tag{B.10}$$

$$\dot{Q}_{\text{load,i}} = UA_{\text{amb}} \cdot (T_{\text{amb}} - T_{\text{air,i}}) \tag{B.11}$$

$$\frac{\mathrm{d}M_{r,i}}{\mathrm{d}t} = OD_i \cdot \alpha \cdot \sqrt{P_c - P_{suc}} - \frac{\dot{Q}_e}{\Delta h_{lg}} \tag{B.12}$$

$$\frac{\mathrm{d}P_{\text{suc}}}{\mathrm{d}t} = \frac{\dot{m}_{\text{in-suc}}(\cdot) - \dot{m}_{comp}}{V_{\text{suc}}\ \nabla\rho_{\text{suc}}(P_{\text{suc}})} \tag{B.13}$$

The common state between the display cases, the suction manifold and the compressors, is the suction pressure, P_{suc},for which (B.13) describes the dynamics. The refrigerant density and the pressure derivative of the refrigerant density is denoted by ρ_{suc} and $\nabla\rho_{suc}$, respectively in (B.13). The mass flow rate into the suction manifold, $\dot{m}_{\text{in-suc}}$ is described by:

$$\dot{m}_{\text{in-suc}}(M_{\text{r,i}}, T_{\text{wall,i}}, P_{\text{suc}}) = \sum_{i=1}^{N} \frac{\dot{Q}_{\text{e,i}}(\cdot)}{\Delta h_{\text{lg}}(P_{\text{suc}})} \tag{B.14}$$

The mass flow rate generated by the compressor rack is describe by

$$\dot{m}_{comp} = \texttt{Cap} \cdot \frac{1}{100} \cdot \eta_{vol,i} \cdot \dot{V}_{sl,i} \cdot \rho_{suc}, \tag{B.15}$$

where `Cap` is the total running compressor capacity of the rack and η_{vol} and $\dot{V}_{sl}$ is the volumetric efficiency and the swept volume flow rate, respectively. No dynamics of the condenser is modelled, it simply defines the high pressure as being static. Hence, the condenser is assumed to be able to maintain a constant pressure and a constant sub-cooling.

The controllers for the display cases and the compressor rack are PI controllers and can be described by the following equations,

$$OD(t) = k_{c,od} \cdot (T_{\text{ref,i}}(t) - T_{\text{air,i}}(t)) + \frac{1}{T_{i,od}} \cdot \int_0^t (T_{\text{ref,i}}(\tau) - T_{\text{air,i}}(\tau))d\tau \quad \text{(B.16)}$$

$$\mathtt{Cap}_{req}(t) = k_{c,comp} \cdot (P_{\text{sucref}}(t) - P_{\text{suc}}(t)) + \frac{1}{T_{i,comp}} \cdot \int_0^t (P_{\text{sucref}}(\tau) - P_{\text{suc}}(\tau))d\tau \quad \text{(B.17)}$$

where $k_{c,od}$, $k_{c,comp}$ and $T_{i,od}$, $T_{i,comp}$ denotes the proportional gain and the integration time respectively. In (B.16) the desired temperature in the i^{th} display case is denoted by $T_{\text{ref,i}}$ and the measured air temperature is denoted by T_{air}. The reference and the measured suction pressure is denoted by P_{sucref} and P_{suc} respectively. The requested compressor capacity is in (B.17) denoted by $\mathtt{Cap}_{req}$ which is used by the distributer algorithm to decide the compressor combination by determining the values of $\delta \in \{0, 1\}$ in (B.18).

$$\mathtt{Cap} = \sum_{i=1}^{i=N} \delta_i \mathtt{Cap}_i \quad \text{(B.18)}$$

In (B.18) N is the number of compressors in the rack and $\mathtt{Cap}_i$ denotes the capacity of the i^{th} compressor.

After introducing the system model an illustrative example will be presented. Passive detection methods relay on a change in the system that causes a change in the residual.However, in the performance assessment case it is not guaranteed that a change in a controller parameter will affect the performance indicators since a refrigeration system usually operates in steady state. In addition, due to a substantial amount of measurement noise the detectors have to be robust with respect to the noise level. Without any excitation of the refrigeration system the affect on the performance measure becomes undetectable, which is illustrated by Fig. B.3, where the top plot is the cost function, the middle plot is the test statistics and the bottom plot is the controller parameter $k_{c,od}$ for one of the display case controllers. The same parameter change is carried out at 20000 and 50000 seconds and it is clear to see that the performance change is not visible without excitation. Fig. B.4 is a plot of the temperature in the display case for which the controller parameters are changed. The signal in the figure shows that the parameter change in the controller is visible only in the case when the system is excited. To ensure that a parameter change is detectable and can be distinguished from influences on the performance measurement, in presence of significant noise, excitation of the system will be required, as well as employment of appropriate statistical detection methods.

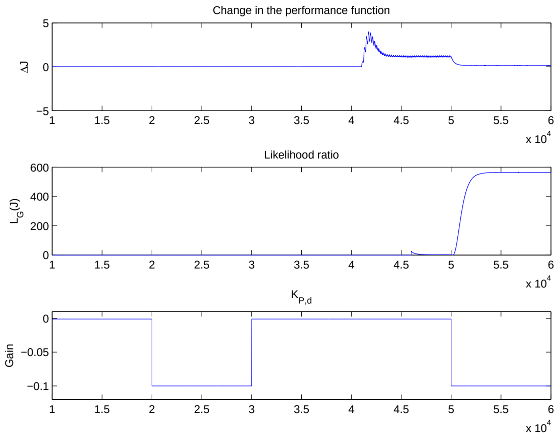

Figure B.3: Change in the performance function, $\Delta J = J - \bar{J}_0$, likelihood ratio, $L_G(J)$, and controller gain $K_{c,od}$. Simulation without and with excitation of the system. Controller parameters changed at 20000, 30000 and 50000 seconds. Excitation started at 41000 seconds

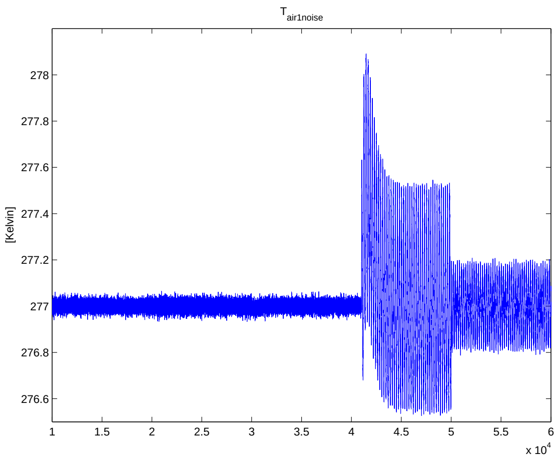

Figure B.4: Air temperature in the display case when the parameters are changed. Simulation without and with excitation of the system. Controller parameter changed at 20000, 30000 and 50000 seconds. Excitation started at 41000 seconds

B.6 Test scenarios for the active performance assessment

In this section two different scenarios will be shown to illustrate the use of active performance assessment for supermarket refrigeration systems. In the first scenario the parameters, $k_{c,od}$ and $T_{i,od}$, of one of the display case controllers has been changed to improve the operation performance of that particular display case. On Fig. B.5 it can be seen that the variations of the temperature declines significantly after the parameters have been changed in the controller. Since the control error of the temperature controller in the display case are represented in the performance measure a visible change can be expected. Fig. B.6 shows

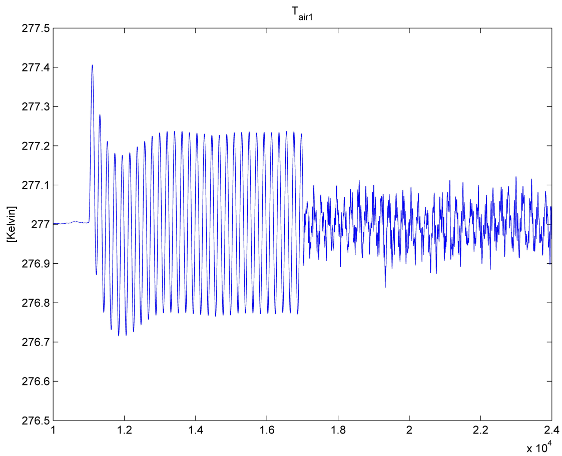

Figure B.5: Air temperature from a display case when the parameters of the controller is changed at 17000 seconds improve the performance.

change in the performance measure on the top plot, the middle plot shows the likelihood ratio and the bottom plot is the flag state. It is clear to see that a change in mean value of the performance measure has occurred. In addition, from the flag state it is possible to verify that the change in performance is an improvement because the flag changes from 0 to 1, a degradation of performance will result the flag state going to -1.

In the second scenario the parameters are changed to degrade the performance to illustrate that the detector can also handle deterioration of the performance. On Fig. B.7 it can be seen that the temperature variation clearly increases when the controller parameters are changed at 17000 seconds. Fig.

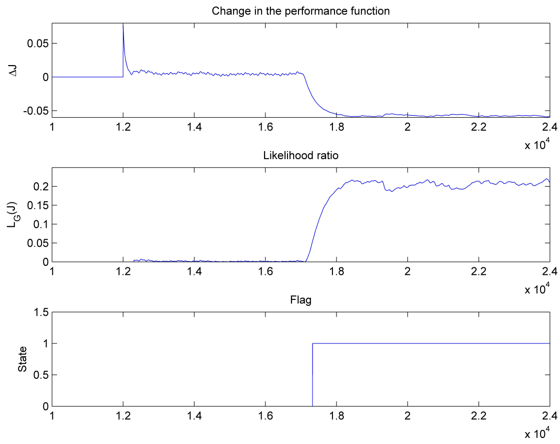

Figure B.6: Change in the performance function, $\Delta J = J - \bar{J}_0$, likelihood ratio, $L_G(J)$, and the flag when the controller parameters are changed at 17000 seconds to improve the system performance.

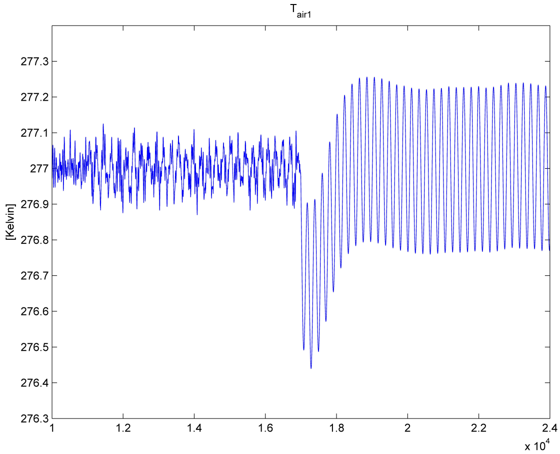

Figure B.7: Air temperature from a display case when the parameters of the controller is changed at 17000 seconds to degrade the performance.

B.8 shows the change in the performance on the top plot, the second plot is the likelihood ratio and the bottom plot is the flag state. It is clear to see that a change in mean value of the performance measure has occurred. In addition from the flag state it is possible to verify that the change in performance is a degradation of performance because the flag changes from 0 to -1.

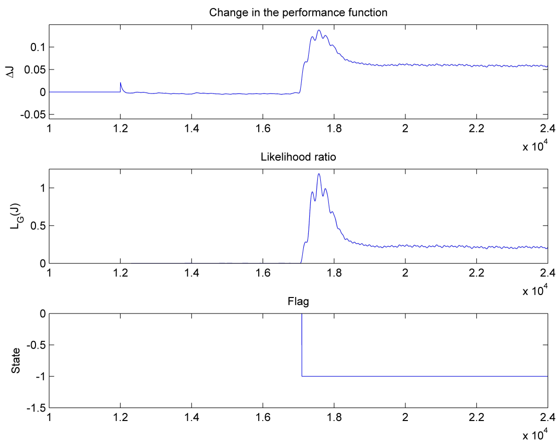

Figure B.8: Change in the performance function, $\Delta J = J - \bar{J}_0$, likelihood ratio, $L_G(J)$, and the flag when the controller parameters are changed at 17000 seconds to degrade the system performance.

B.7 Discussion

The paper proposed an approach for active performance assessment, based on active fault diagnosis techniques, in order to online optimize the overall system performance under operation. The approach is appealing for industrial applications where the knowledge about the underlying system dynamics is not available. The approach was justified by a illustrative example and the new setup was tested on different parameter changes of closed loop controlled subsystem of a supermarket refrigeration system. The approach presented in the paper governs the use a performance function introduced in [13] and the used of a statistical test method for detection of a change in the performance index. The method solely based on measured signals and does not depend on the existence of a model of the system under assessment. Future research will focus on utilising the method for systematic optimisation of the performance of a supermarket refrigeration plant.

Paper C

Design of Excitation Signal for Active System Monitoring in a Performance Assessment Setup

Torben Green, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Design of Excitation Signal for Active System Monitoring in a Performance Assessment Setup. Published in *Proceedings of the 9th European Workshop on Advanced Control and Diagnosis*. Budapest 2011.

Abstract: This paper investigates how the excitation signal should be chosen for a active performance setup. The signal is used in a setup where the main purpose is to detect whether a parameter change of the controller has changed the global performance significantly. The signal has to be able to excite the dynamics of the subsystem under investigation both before and after the parameter change. The controller is well known, but there exists no detailed knowledge about the dynamics of the subsystem.

Keywords: Performance assessment, Performance optimisation, Active system monitoring, Refrigeration systems

C.1 Introduction

In the fault diagnosis literature active fault diagnosis schemes has been introduced in the papers [33] and [39]. The governing idea in these papers is that a probe signal is used to excite the system

In previous papers like [33], [39], [35], and [21] only rough guideline are given about the design of the excitation signal.

The main problem is to assess whether a closed loop subsystem has the parameter freedom to improve the overall performance of the entire system. Hence, global performance measure is needed along with a method that can explore the parameter freedom.

A refrigeration system for a supermarket will usually run in steady state. Thus, the performance measurement will not be sensitive to all parameter changes. Therefore, excitation of the different closed loop controlled subsystem will be required to ensure that the changes in the controller parameters can be observed from the performance measure.

In this paper a design method for the excitation signal is presented. The idea is to present a consistent way for choosing the frequency for excitation signals for active fault diagnosis system, especially when the techniques are used in a performance assessment.

In following section the active fault diagnosis setup used for performance assessment will be described followed by a problem formulation. Thereafter the problem of designing a excitation signal for the setup will be described. Section C.5 explains the proposed design procedure. Simulation results using different excitation signals will be presented and at the end of the paper a discussion is presented.

C.2 New Setup

In this section the new setup for active system monitoring will be described. The aim is to assess whether it is possible to improve the overall system performance. The approach to determine the improvement potential will be based on the performance measure introduced in [13] and is elaborately described in [15]. The performance measure is described by (C.1)

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}(k)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP(l)}||_R^2 + \sum_{m=1}^{M} ||f_{sw}(m)||_S^2 \qquad \text{(C.1)}$$

The first term is representing the food quality by the control errors in K controllers and the second term is a measure of the efficiency of the plant using the inverted coefficient of performance, COP, where L denotes the number of suction groups in the refrigeration plant. The last term is the switch frequency of the compressors, where M denotes the number of compressors. The switch frequency is included to avoid excessive wear on the compressors. However before the new setup is explained, a small introduction to the supermarket system will be given.

The common setup for a supermarket refrigeration system can be seen on Fig. C.1. The system in Fig. C.1 is comprised of a number of display cases, where in

Figure C.1: Simplified supermarket refrigeration layout

the food is stored. These display cases are connected to a compressor rack which changes the pressure of the refrigerant and thereby enables the condenser unit to reject the heat to the surroundings. Each display case is fitted with a controller that ensures the correct temperature inside the display case by controlling the expansion valve and thereby the flow of refrigerant into the evaporator of the display case. The compressor rack is also fitted with a controller which ensures that the refrigeration system is operation at a specific evaporation pressure.

As mentioned earlier a refrigeration system is usually operating in steady state conditions and will therefore not be affected by a change in a controller parameter unless the dynamics of the system is excited. A simulation model will be used to select a probe signal that can be used to evaluate the performance change created by a parameter change in the controller. The probe signal has to be designed to excite the closed loop under investigation sufficiently to enable an evaluation of a parameter change, but without compromising the operation of the refrigeration plant.

To test whether a closed loop system can affect the global performance measure, the controller parameters are changed during operation. The task of the detector is to detect the change in the global performance and to decide if the performance has improved or degraded. To ensure that an excitation signal will impact the performance measurement even after a parameter change in the controller the excitation signal has to be designed with respect to the expected bandwidth of the closed loop.

Fig. C.2 shows a block diagram of the setup, which illustrates a closed loop under assessment, plus the global performance measure and the detector. The parameter change of the controller K is denoted by $\Delta\xi$. The control input to the system is denoted by u and J denotes the performance measure which is directly used as residual by the detector to decide whether a given parameter change, $\Delta\xi$, has changed the overall performance measure. The input to the performance measure, from the other subsystems, is denoted by Γ. The overall excitation of the subsystem under investigation, η_Σ, can be abstracted by the following formulation:

$$\eta_\Sigma = \eta_{ext} + \eta_w \tag{C.2}$$

where η_Σ is the sum of all the excitation sources and η_{ext} denotes the externally applied excitation. The last term, η_w, denotes the excitation contribution generated by noise sources like the other subsystems in the refrigeration plant. In some situations the excitation created by the other subsystems will be sufficient i.e. $\eta_\Sigma = \eta_w$ because applying η_{ext} becomes unnecessary. Thus, the task is to choose η_{ext} so that η_Σ has the sufficient impact in the closed loop under investigation.

C.3 Problem Formulation

Utilising active fault diagnosis methods requires design of dedicated excitation signals. The excitation signal can be both deterministic and stochastic. If the knowledge about the excitation frequency can be exploited as in [39] a periodic signal would be preferable. If a periodic signal is chosen the frequency of the excitation signal has to lie within the bandwidth of the closed loop system under investigation. Choosing that signal is not a trivial problem because the dynamics of the system under investigation is unknown. In addition, since the scheme relies on changing the controller parameters, the bandwidth of the closed

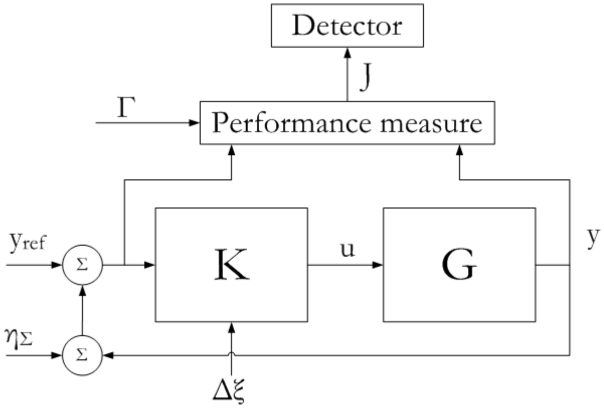

Figure C.2: Active performance assessment setup

loop will change over timer. Hence, it has to be ensured that the excitation signal will be within the bandwidth of the closed loop all the time.

The main topic of this paper is design of the externally applied excitation signal η_{ext} which is used in the in the new setup for performance assessment. The task of the signal is to excite the dynamics of the subsystem enough to assess whether a certain parameter change will affect the global performance measure. Since the aim is that the active performance assessment setup should run online on a supermarket refrigeration system, compromising the operation of the system will not be acceptable. Hence, the excitation signal cannot be chosen without considering the impact on the operation of the plant. In an industrial application, as the supermarket refrigeration system, their sometimes exists switching phenomenon that can render an internal generated excitation. When designing the external excitation signal the properties of the possible internal excitation will have to be taken into account. In addition, choosing the amplitude for the external excitation signal is an online task, because the internal generated excitation is dependent on the operation point. That is, in some situation it might not be necessary to apply external excitation. Fault diagnosis without external excitation is normally called passive fault diagnosis. However, in this case there is practically no distinction between active and passive fault diagnosis because the classification relies on an online decision.

Detailed model knowledge regarding the refrigeration plant cannot be used in the design approach for the excitation signal because it cannot be expected that there exist a sufficient model for the refrigeration plant. The knowledge regarding the system under investigation is limited to a structural level. Meaning that time constants and gain of the subsystems are unknown. This is in contrast with the assumptions required for the design methods introduced in [33] and [39].

C.4 Design considerations

When utilising the proposed new setup for performance assessment it is important to note that design of the excitation signal and design of the performance function are two separate design problems and should be treated separately. The design of the performance function should focus on expressing the relevant performance criterion in the system. Hence, the performance function should be designed with a global system perspective and with expressing the global system performance as the goal. The design of the excitation signal should be done with focus on the local closed loops under investigation. In addition, the design of the excitation signal can actually, as is the case in the refrigeration example, be constrained by restrictions on the system states. As argued earlier the decision to apply an external excitation signal, and thereby utilise active fault diagnosis techniques, has to be made online. This paper will focus on design issues for the excitation signals. Knowledge about the bandwidth of the closed loop under investigating cannot be assumed. Therefore, a reasonable estimate will be used as basis for the design of the excitation signal.

C.5 Excitation signal design approach

The design of the excitation signal has been split into two different tasks. The first task is to determine what frequency range will be appropriate for the excitation signal. The second task is determining the amplitude for the excitation signal. The two tasks have to be treated with different approaches. The focus in this paper will be to solve the first task. Solving the first task requires knowledge about the bandwidth of the closed loop under investigation and about the noise that the closed loop is imposed with. However, the bandwidth of the closed loop subsystems is not known in advance. Hence, the bandwidth of the closed loop system has to be estimated based on accessible knowledge. The estimation of the closed loop bandwidth shall be based on the following facts:

1. The closed loop consists of a PI controller and a physical system for which the dynamics are unknown.

2. The parameters of the PI controller are known.

3. The controller is stabilising the closed loop.

In addition, some assumptions are needed before we can utilise an alternated version of the method presented in [19]. The assumptions are:

1. The system can be described sufficiently using a first-order model plus time delay (FOPTD).

2. The closed loop is stable with the chosen PI parameters.

3. The time delay in the system is assumed to be known.

It is experimentally verified that the system dynamics can be sufficiently described by a FOPTD process model around an operating point. The non-linearities and un-modelled dynamics have an insignificant impact on the closed loop performance around the usual operating point. Since the PI controller is stabilising the closed loop the system is a member of the set of systems that the controller can stabilise. Independent on the chosen criteria any system that is stabilised by this controller is hence a member of this set. In the following, we utilise phase and gain margin requirements as start point in order to provide a qualified guess on the system bandwidth. The original expectation is that the system closed loop bandwidth does not vary significantly with respect to the system parameter variations. This will be investigated in the next section.

Assuming that the system is described by a FOPDT process model, [19] proposes a method that provides the parameters for a PID controller, which fulfils a given set of specifications for gain and phase margin of the closed loop system. The proposed tuning method has been used in a slightly alternated way in this paper as described in the following: Knowing the parameters of the PI controller combined with the assumption that the system time delay is known the method in [19] is utilised to derive the remaining system parameters. This is elaborated in the following:

Estimating the bandwidth based on the idea from [19] the setup introduced in Fig C.2 need to be simplified as shown on Fig. C.3. The system transfer

Figure C.3: Block diagram of the simple closed loop

function $G(s)$, the controller $K(s)$, and the corresponding closed loop transfer function $H(s)$ are given as:

$$G(s) = \frac{k_p \cdot e^{-sL}}{1+s\tau}, \quad (C.3)$$

$$K(s) = k_c + \frac{k_c}{T_i s}, \quad (C.4)$$

$$H(s) = \frac{KG}{1+KG}. \quad (C.5)$$

After rewriting of the equations from [19] we can calculate the time constant, τ, and the gain of the system, k_P, using (C.6) and (C.7) respectively. The crossover frequency is denoted by ω_p and the time delay in the system $G(s)$ is denoted by L. The integration time and the controller gain is denoted by T_i and k_c respectively. The method also requires desired phase margin and gain margin which is denoted by ϕ_m and A_m respectively.

$$\tau = \left(\frac{1}{T_i} + \frac{4 \cdot \omega_p^2 \cdot L}{\pi} - 2 \cdot \omega_p\right)^{-1} \quad (C.6)$$

$$k_p = \frac{\omega_p \cdot \tau}{A_m \cdot k_c} \quad (C.7)$$

The phase crossover frequency is calculated using (C.8)

$$\omega_p = \frac{A_m \phi_m + \frac{1}{2}\pi A_m (A_m - 1)}{(A_m^2 - 1) \cdot L} \quad (C.8)$$

Using (C.6) through (C.8) the method has been tested with various values of the time delay, the phase margin, the amplitude margin. The test has been done to clarify the dependency of the estimated bandwidth with respect to these parameters.

In the first scenario the time delay of the system has been varied to see how sensitive the estimated bandwidth would be on variation of the time delay. The controller parameters k_c and T_i and the desire phase and gain margin are kept the same for all the different time delays. For each time delay a new estimate of the parameters for the system, $G(s)$, has been calculated and the thereafter the closed loop transfer function has been calculated. The bode plot of the closed loop transfer function can then be seen on Fig. C.4. Fig. C.4 shows that there is a high dependency on the estimated closed loop bandwidth and the difference in time delay. However, the time delay itself can be measured with high precision by simply applying a small step to the system and then measuring the time before a response can be seen on the output. Thus, even if

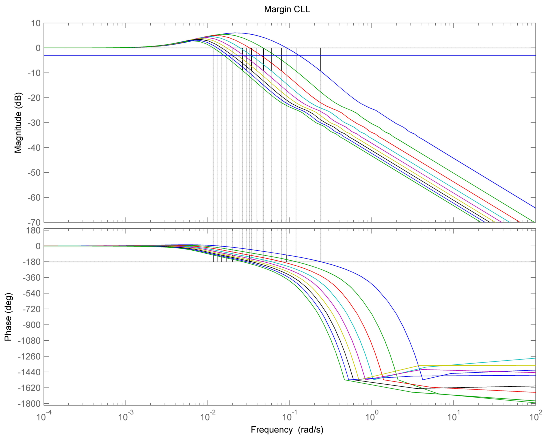

Figure C.4: Bode plot of the closed loop with different time delay

there is a significant influence on the estimated closed bandwidth from the time delay, it is not considered an issue.

In the second scenario the desired phase margin has been changed to see the effect on the estimated closed loop bandwidth of the system. The controller parameters k_c and T_i are kept constant and the time delay is also chosen to be the same. The desired gain margin is also kept at a constant value. For each of the chosen values of the phase margin a new set of parameters has been estimated for the system $G(s)$. The closed loop transfer function has been calculated and the bode plots can be seen on Fig. C.5

Fig. C.5 shows the bode plot for different phase margins. It can be seen that the choice of phase margin does have an influence on the estimated bandwidth for the closed loop system. However, the difference in estimated bandwidth due to different choices of phase margin is negligible. In the third scenario the desired gain margin has been changed to see the effect on the estimated closed loop bandwidth of the system. The controller parameters k_c and T_i are kept constant and the time delay is also chosen to be the same. The desired phase margin has been kept constant in this particular scenario.

The difference in closed loop bandwidth due to different choice of gain margin can be seen on Fig. C.6. The larges estimated bandwidth is approximately three times larger than the smallest estimate. However, the smallest bandwidth originates from the most defensive choice of gain margin.

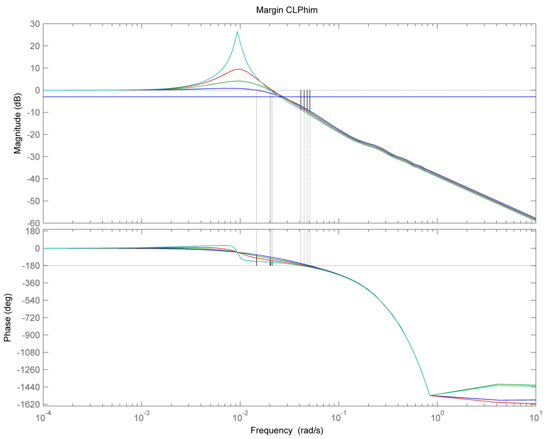

Figure C.5: Bode plot of the closed loop with different choice for the phase margin

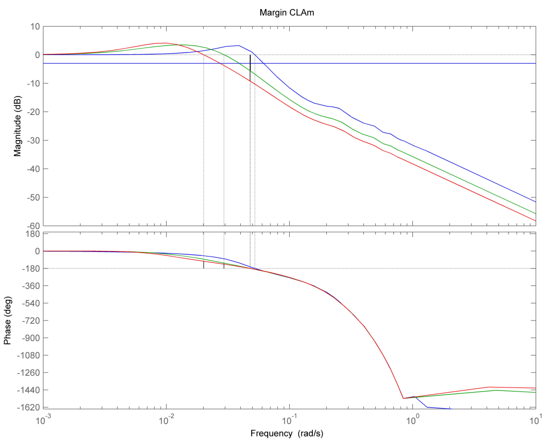

Figure C.6: Bode plot for the closed loop with diffent choice of the gain margin

The task is to design the excitation signal η_{ext}, and based on the three scenarios it can be seen that the proposed method can be used to estimate a reasonable bandwidth for the closed loop under investigation. Now the question is how this bandwidth estimate can be used to choose the frequency content of the excitation signal. Since the task is to design a signal that can be used in a active performance monitoring setup as the one proposed in the paper [15]. Since the aim of the active performance monitoring setup is to optimise the operation of the global performance the excitation signal has to be chosen within the closed loop bandwidth. That is because that will ensure that the excitation signal will still be visible on the performance measure. In addition choosing the frequency around the $-3dB$ for the closed loop will ensure that the excitation of the closed loop is in the frequency range where the performance has to be improved. The results of the bandwidth estimation is summaries in table C.1, which suggests that a choice of 0.03 $[rad/sec]$, as the frequency for the excitation signal, is reasonable.

	Min	Max	Min bandwidth	Max bandwidth
L	6	54	0.0157	0.1297
ϕ_m	15	60	0.0252	0.0273
A_m	2	4	0.0273	0.0615

Table C.1: Results of the estimated bandwidth

C.6 Test results

In this section some test results will be presented. The tests have been done using the setup from [15], where the excitation signal has been applied and after that a change in the controller parameters has been applied. The global performance is then measured and a detector decides if a significant change has occurred on the global performance. If that is the case the state variable output from the detector will go to zero or one. The state variable is discrete and take the value 0 if no change is detected and 1 if the detected change is a performance improvement and -1 if the detected performance change is a degradation. The tests have been done on a simulation model of a supermarket refrigeration system. The subsystem that has been under investigation is one if the display cases. The excitation signal that is used is described in (C.9)

$$\eta_{ext} = A \cdot \mathtt{sin}(\omega t) \tag{C.9}$$

In the first test the combined knowledge gathered from examining the three scenarios from section C.5 a frequency of 0.03 $[rad/sec]$ has been chosen. Since this frequency is chosen based on this novel design principle it is expected that the active performance monitoring setup will work without any problems. The results of the first test can be seen on Fig. C.7. Figure C.7 shows that the

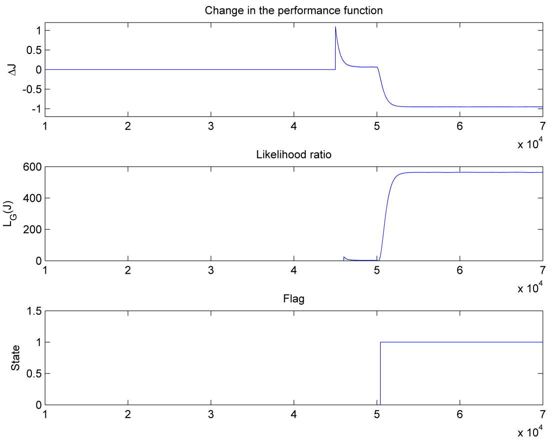

Figure C.7: Testing the active performance setup using an excitation signal with the frequency of 0.03 $[rad/sec]$

detector is working as expected with the designed frequency for the excitation signal. The upper plot in Fig. C.7 shows the change in the performance measure J and the second plot show the likelihood ratio which is the output of generalised likelihood ration test. The bottom plot is the state variable. Fig. C.8 shows the effect on the temperature in the display case. It is clear to see that there is a change in the performance for the better. In addition, the influence of the excitation signal is quite noticeable.

In the second test the same procedure as in the first test has been followed. The only thing that has been changed is the frequency of the excitation signal. In the second test the frequency of the excitation signal is $0.06[rad/sec]$. The result from the second test can be seen on Fig. C.9. The purpose of this second test is to justify the importance of choosing the correct excitation signal. Since the chosen signal is double the frequency of the first test the detector is unable to detect any changes in the performance. The change becomes too small for the detector setup. On Fig. C.10 the air temperature in the display case for test two can be seen. It is clear that the system is of course still experiencing a performance improvement since it is the same parameter change as in test one. However, the important thing to notice is that by comparing Fig. C.8

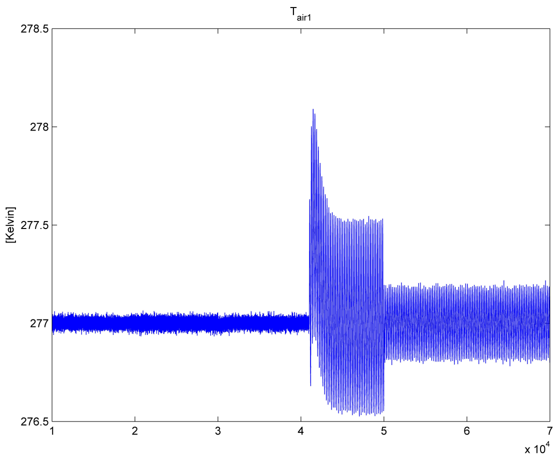

Figure C.8: Air temperature in the display case using an excitation signal with the frequency of 0.03 $[rad/sec]$

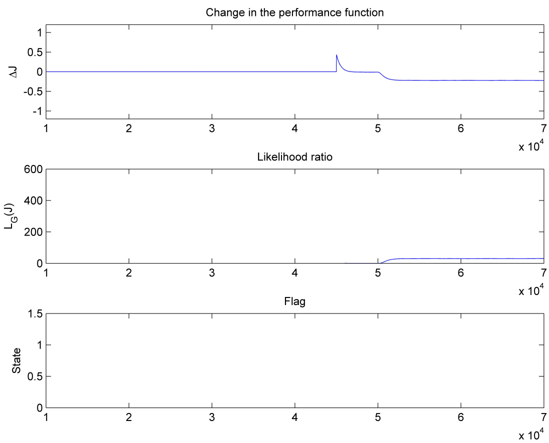

Figure C.9: Testing the active performance setup using an excitation signal with the frequency of 0.06 $[rad/sec]$

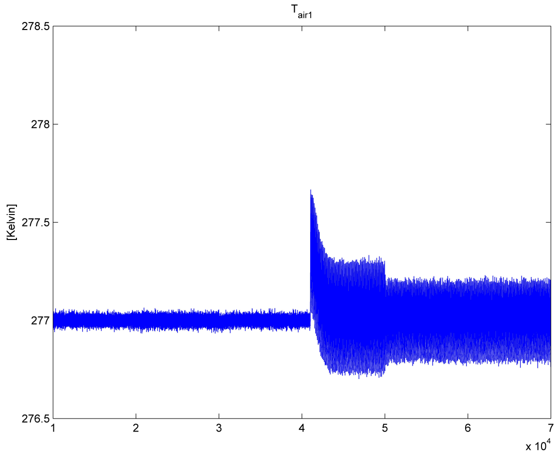

Figure C.10: Air temperature in the display case using an excitation signal with the frequency of 0.06 $[rad/sec]$

and Fig. C.10 it is clear to see that the impact of the parameter change is less visible. Hence, the impact on the global performance measure will also be less and therefore more difficult to detect.

C.7 Discussion

A structured approach for choosing the frequency content for the excitation signal for the active system monitoring setup has been presented. The method calculates the system parameters for a FOPTD based on the knowledge of the parameters of the PI controller and the time delay of the system. The system parameters are then used to give a qualified estimate of the closed loop bandwidth. The knowledge of the bandwidth is then used to determine the frequency of the excitation signal. The proposed method has been verified by testing the results on a simulation of a supermarket refrigeration system. It has been shown that the method can be used to choose a reasonable signal for the excitation frequency.

Paper D

Optimising performance in steady state for a supermarket refrigeration system

Torben Green, Michel Kinnart, Roozbeh Razavi-Far, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Optimising performance in steady state for a supermarket refrigeration system. Accepted for presentation at the *20th Mediterranean Conference on Control and Automation*. Barcelona 2012. IEEE

Abstract: Using a supermarket refrigeration system as an illustrative example, the paper postulates that by appropriately utilising knowledge of plant operation, the plant wide performance can be optimised based on a small set of variables. Focusing on steady state operations, the total system performance is shown to predominantly be influenced by the suction pressure. Employing appropriate performance function leads to conclusions on the choice of set-point for the suction pressure that are contrary to the existing practice. Analysis of the resulting data leads to a simple method for finding optimal pressure set-point for given load situations.

D.1 Introduction

In a competitive and global business environment plant-wide performance assessment and optimisation in the process industry have increasingly become important issues as they have direct impact on the operational costs, energy and environmental issues. Supermarket refrigeration systems are no exception: One of the larger operational costs of a supermarket is the refrigeration plant. In a supermarket, the refrigeration system accounts for 40% to 60% of annual electrical energy consumption, see [10]. Furthermore, there are substantial costs related to component replacement and unscheduled maintenance. In many industrial systems, it is customary to use over-dimensioned components that provide excess capacity in order to guarantee that the system functionality is provided under all conditions, which are in particular caused by non-optimal operational conditions. For instance, the compressor rack in supermarket systems is typically made of compressors that can provide up to 50% more capacity than the supermarket system is actually designed for. Non optimal operation not only affects the cooling efficiency and food quality but also have direct impact on the operational lifetime of the components. Proper optimisation tools/methods can be used to optimise the performance of an operating system. It can also assist the design engineering group to choose appropriate components of right dimensions/capacities at much more suitable costs in future plants. An appropriate performance function for plant-wide operation should include contributing terms that describe the quality of products, system efficiency, as well as the operating lifetime of the subsystems. In section D.2.1, a performance function fulfilling these requirements is proposed. Optimisation of refrigeration system has been attempted in other papers such as [18] where multi variable control is used to get at better performance of a vapour compression system. An energy optimal control approach for refrigeration system is introduced in [23] and in [27] an online steady state energy minimisation is presented, where the minimisation is relying on a model of the refrigeration system. In this paper, we utilise an appropriate performance function, introduced in [13], to assess the system performance. Since, supermarket refrigeration systems operate in steady state in the majority of their operating time, as many other plants in process industry, it is reasonable to separate the steady state condition from the transient one and then explore the ways to asses and optimise the performance. By employing appropriate performance function and choosing suction pressure as the dominating variable it is shown that the suggested operating set-points for the suction pressure differ significantly from the ones that can be obtained by using performance function that is used in common practice. Furthermore, the knowledge based on investigating the resulting performance under different load conditions is used to suggest a simple procedure for identifying the pressure set-point that leads to optimal operation in steady state conditions. In section D.2 the problem is defined and the performance function is introduced as well as the optimisation problem. Thereafter, in section D.3 the simulation setup is presented and simulation results are presented in D.4. The method for set-point optimisation is then described in D.5 and the paper ends with a conclusion.

D.2 Problem statement

The supermarket refrigeration system operates in steady state in the majority of time and it is therefore of high importance to always use the optimal reference for the controllers. The compressor control loop is one of the important control loops because of the relatively high energy consumption of the compressors. The control task of the compressors is to maintain a desired suction pressure. However, choosing a proper suction pressure reference, in refrigeration system, is critical to provide a certain temperature level of refrigeration. To fully understand the problem an overview of a supermarket refrigeration system will be introduced hereafter. In Fig. D.1, a simple diagram for a supermarket refrigeration system is presented. The compressors are connected in parallel with display cases and the condenser unit. The controller structure for the refrigeration systems is implemented in a distributed setup where each of the display cases has a controller that controls the temperature by manipulating the inlet of refrigerant into the evaporator of the display case. To ensure a sufficient temperature difference in all the display cases, a common evaporation temperature is achieved by controlling the compressors, to deliver a common suction pressure. The control of the compressor rack is discrete which means that the compressors can only be switched on or off. Excessive switching of the compressors is not desirable because it creates superfluous energy consumption and excessive wear on the compressors. The commonly used strategy is to choose the suction pressure reference as high as possible. The reason for choosing that strategy is that a higher suction pressure requires a lower compressor capacity. However, that strategy does not take the switching phenomenon into account.

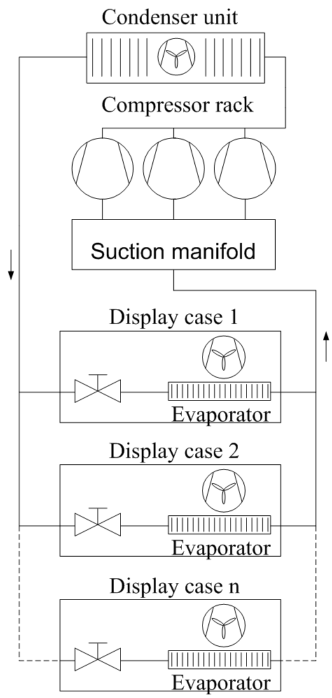

Figure D.1: Simplified supermarket refrigeration layout

D.2.1 Performance function

The quality of the solution for optimal performance is highly dependent on which performance criteria are included in the performance function. The relevant performance criteria for supermarket refrigeration systems are food quality, energy efficiency and reliability. Monitoring of the food quality is achieved by including the control errors of the temperature controllers and the suction pressure controller in the monitoring setup. The energy efficiency is monitored by using the coefficient of performance, COP, in the setup. The COP is defined as the delivered cooling power delivered divided by the electrical power consumed. The reliability of the plant is monitored by including the switch frequency of the compressors in the setup. Excessive switching of the compressors will generate excessive wear on the compressors and thereby increase the need for maintenance and decrease the lifetime of the compressors. The combined effect on the system is a lower operational reliability. Hence, minimisation of the switch frequency is desirable.

To ensure that the operation of the plant is optimal with respect to all three criteria the following performance function, which was introduced in [13], has been used in this work:

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}(k)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP(l)}||_R^2 + \sum_{m=1}^{M} ||f_{sw}(m)||_S^2 \tag{D.1}$$

The performance function (D.1) is a sum of quadratic terms where the notation is given by (D.2).

$$||\mathbf{e}||_Q^2 = \mathbf{e}^T Q \mathbf{e} \tag{D.2}$$

The first term in (D.1) is the control errors of K controllers. The second term is the inverted COP of L refrigeration cycles and the third term is the switch frequency of M compressors.

D.2.2 Normalisation

Since the main objective is to be able to employ the performance function in various supermarkets (with different subsystems of different sizes and dimensions), the performance function should be made of the scalable terms that can be easily adapted for a given supermarket. The scalability can be achieved through normalization of the terms in the performance function. For any given supermarket the corresponding performance function will then be established through the choice of appropriate weights that reflect the regional regulations on safety requirements as well as local operational expenses. In the following the normalisation procedure will be explained.

The error term is normalised using the knowledge that the temperature in a display case has a lower and an upper limit and the reference is chosen as the mean value of the temperature limits. The same argument is applied for the suction pressure error which also has an upper and a lower limit and the reference is then chosen as the mean value of the two limits.

$$\mathbf{e} = \left| \begin{array}{c} \frac{2}{(T_{max,i}-T_{min,i})} \cdot (T_{\text{ref,i}} - T_{\text{air,i}}) \\ \frac{2}{(P_{suc,max}-P_{suc,min})} \cdot (P_{\text{sucref}} - P_{\text{suc}}) \end{array} \right| \tag{D.3}$$

The switch frequency term is normalized by dividing the measured frequency by the maximum allowable frequency of compressor switching in a compressor rack. This frequency is given by the compressor manufacture

$$f_{sw} = \frac{f_{meas}}{f_{max}} \tag{D.4}$$

As mentioned before the $\frac{1}{COP}$ is unit less and does therefore not need any normalisation. After normalisation each of the terms $\mathbf{e}$, $\frac{1}{COP}$ and f_{sw} will then be in the range between 0 and 1 and the weights will therefore represent the cost associated with each term. Each term in (D.1) contains a quadratic term which includes the weight matrices Q, R and S. These weights represent the costs, in terms of economic penalties or lost profits, which are associated with the performance of each subsystem.

Choosing Q, R and S will be done based on each of the terms impact on the price of running the refrigeration plant.

The weight Q, which is for the control errors, is based on the price that is associated with temperature requirements for the display cases in the supermarket. If the temperature gets too high the stored goods will have to be destroyed. In addition, if the failure is detected by the authorities the supermarket could be faced with a fine. The weight R, which is for the inverted COP, is based on the energy price since the inverted COP is effectively an efficiency of the refrigeration cycle. The weight R, which is for the switch frequency of the compressors, is based on the price of replacing and maintaining a compressor in the refrigeration system. The cost of maintaining and replacing a compressor contains many hidden costs, such as the price of the system being out of operation. Contributions like that renders significant freedom for manipulating the weight based on intuition. The simulation data presented in this paper is scaled.

Thus, the absolute values of the performance function do not have any physical interpretation.

D.2.3 Optimisation formulation

In this section an optimisation problem for the considered system will be defined. As previously mentioned the supermarket refrigeration systems operate in steady state most of the time , i.e. at over 80% of the time. Therefore, it makes sense to first focus on optimising the system performance in steady state operation, henceforth denoted static performance optimisation. Depending on whether the focus is on static performance optimisation or on optimising the dynamic behaviour of the system, different set of parameters will be subject to optimisation. When the main objective is to optimise the static performance one should look for which subsystems have the highest impact on the total performance of the plant. For these systems the variables, which are used to address the conditions and objectives for the operation, are typically the set-points.

The performance function in (D.1) can be written in a more abstract way as:

$$J(\phi(t)) = \sum_{i=1}^{I=K+M+L} J_i(\phi(t)), \tag{D.5}$$

where $J_i(\phi(t))$ is the local performance function for the i^{th} subsystem that is dependent on the set $\phi(t)$ which is defined as:

$$\phi(t) = \left\{ Po_{ref}(t),\ \dot{Q}_{airload}(t) \right\}, \tag{D.6}$$

where Po_{ref}, denotes the suction pressure reference for the compressor rack controller which is a controllable variable, and the uncontrollable variable which is the heat loss to the surroundings and the main disturbance in the supermarket, is denoted by $\dot{Q}_{airload}$.

Set-point optimisation of the supermarket is defined as:

$$\min_{Po_{ref}} J(\phi(t)) \qquad \forall\, \dot{Q}_{airload} \tag{D.7}$$

The optimisation problem is to focus on finding the optimal value for $Po_{ref}(t)$ that minimises $J(\phi(t))$ for any given disturbance, $\dot{Q}_{airload}$. To solve the optimisation problem some primary simulations have been performed to get a deeper knowledge about the search space. The simulations have been carried out as follows:

To get an overview of the optimisation space, a series of simulations has been carried out. In the simulations, the suction pressure reference has been changed in steps from $1.0 \cdot 10^5$ $[Pa]$ to $2.5 \cdot 10^5$ $[Pa]$ with a step size of $0.1 \cdot 10^5 [Pa]$ or the equivalent of changing the evaporation temperature approximately $2°C$. For each of the steps in the suction pressure reference the disturbance, $\dot{Q}_{airload}$, and thereby also the load of the system has been changed by changing the ambient temperature of the display cases in steps from 18 $[°C]$ to 28 $[°C]$, with a step size of $0.5°C$. In these simulations, steady state values of various measurements have been used. Therefore changes, both in the suction pressure reference and ambient temperature, are step like. The dynamic behaviour has not been the focus.

D.3 Simulation setup

The simulation setup is based on a simplified model of a supermarket refrigeration system. The model contains two display cases, a compressor rack comprised of two compressors and a condensing unit. Each of the display cases are fitted with a PI controller that controls the air temperature in the display case. The compressor rack is fitted with a PI controller and a step controller that maintains the desired suction pressure, and thereby ensures a sufficient temperature difference to enable a heat transfer. The layout for a supermarket refrigeration system is shown in Fig. D.1.

The control task of the display case is to ensure that the temperature is maintained at the desired level within the display case. This is normally done by controlling the opening degree of the inlet valve to the evaporator. The capacities of the compressors should ideally be chosen so that the common operation points of the refrigeration plant can be handled with a fixed number of compressors running. However, in practice the compressor rack will not fit the operation points exactly. Therefore, the compressors will have to switch to satisfy the refrigeration load. Thus, the choice of compressors is a compromise and is usually in favour of a certain operation point. Hence, switching of the compressors are unavoidable but should be kept at a minimum to avoid excessive wear on the compressors and superfluous energy consumption.

D.3.1 Modelling the refrigeration system

The model used for simulation in this paper is a slightly modified version of the supermarket refrigeration system model presented in [28]. The model has been changed so that the injection valve can be controlled continuously and thereby providing a model where the temperature in the display case can be controlled continuously.

The model features a refrigeration system with two display cases, a suction manifold, a compressor rack and a condenser. The temperature in each of the display cases is described by (D.8), where (D.9) describes heat flow from the surroundings and into the display case. The other terms in (D.8) are described in detail in [28]. The ambient temperature of the display case, T_{amb}, is assumed constant since it corresponds to the indoor temperature of the supermarket. To enable the possibility of a continuously controlled refrigerant flow the mass flow into the evaporator is modelled using (D.10).

$$\frac{\mathrm{d}T_{\text{air,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\text{goods-air,i}}(\cdot) + \dot{Q}_{\text{load,i}}(\cdot) - \dot{Q}_{\text{air-wall,i}}(\cdot)}{M_{\text{air}}C_{\text{p,air,i}}} \tag{D.8}$$

$$\dot{Q}_{\text{load,i}} = UA_{\text{amb}} \cdot (T_{\text{amb}} - T_{\text{air,i}}) \tag{D.9}$$

$$\frac{\mathrm{d}M_{r,i}}{\mathrm{d}t} = OD_i \cdot \alpha \cdot \sqrt{P_c - P_{suc}} - \frac{\dot{Q}_{e,i}}{\Delta h_{lg}} \tag{D.10}$$

$$\frac{\mathrm{d}P_{\text{suc}}}{\mathrm{d}t} = \frac{\dot{m}_{\text{in-suc}}(\cdot) - \dot{m}_{comp}}{V_{\text{suc}}\ \nabla\rho_{\text{suc}}(P_{\text{suc}})} \tag{D.11}$$

In (D.10) the opening degree of the expansion valve is denoted by OD, P_c denotes the condensing pressure, $\dot{Q}_e$ denotes the heat removed by evaporation and the enthalpy difference across the two-phase region is denoted by Δh_{lg}. For details about the modelling of $\dot{Q}_e$ and Δh_{lg} see [28].

The suction pressure is the only common state for all of the display cases, the suction manifold and the compressor, and its dynamics can be described by (D.11). The refrigerant density is denoted by ρ_{suc} and $\nabla\rho_{\text{suc}}$ denotes the pressure derivative of the refrigerant density.

The mass flow rate into the suction manifold, $\dot{m}_{\text{in-suc}}$ is described by:

$$\dot{m}_{\text{in-suc}}(M_{\text{r,i}}, T_{\text{wall,i}}, P_{\text{suc}}) = \sum_{i=1}^{N} \frac{\dot{Q}_{\text{e,i}}(\cdot)}{\Delta h_{\text{lg}}(P_{\text{suc}})} \tag{D.12}$$

The temperature, T_e, denotes the evaporation temperature. As explained in [28] Δh_{lg}, ρ_{suc} and $\nabla\rho_{\text{suc}}$ are all refrigerant specific functions.

The compressor rack is described by (D.13).

$$\dot{m}_{comp} = \texttt{Cap} \cdot \frac{1}{100} \cdot \eta_{vol,i} \cdot \dot{V}_{sl,i} \cdot \rho_{suc} \tag{D.13}$$

In (D.13) $\texttt{Cap}$ denotes the running compressor capacity of the rack, the volumetric efficiency is denoted by η_{vol}, and the swept volume flow rate is denoted by $\dot{V}_{sl}$. The condenser model contains no dynamic, it only defines a static condensing pressure and a static sub-cooling, which suggests an assumption that the condenser is controlled well enough to keep a constant condensing pressure and a constant sub-cooling.

D.3.2 Controller setup

The control setup is comprised of a temperature controller for each of the display cases that manipulates the opening degree of the expansion valve, OD_i, and a suction pressure controller that manipulates the running compressor capacity in the compressor rack. The temperature controllers and the suction pressure controller are implemented as PI controllers. Emulation of the discrete behaviour of the compressor rack is achieved by defining the compressor rack as being comprised of two compressors. Generally the $\texttt{Cap}$ in (D.13) has the following form:

$$\texttt{Cap} = \sum_{i=1}^{i=N} \delta_i \texttt{Cap}_i \tag{D.14}$$

with $\delta_i \in \{0, 1\}$ and $\sum_{i=1}^{N} \texttt{Cap}_i = 100\%$. In our application $N = 2$, $\texttt{Cap}_1 = 45\%$ and $\texttt{Cap}_2 = 55\%$. To avoid excessive switching of the compressors a hysteresis band is applied around each compressor step. The layout of the compressor rack is based on a real supermarket system which has the same layout of the compressor rack.

D.4 Simulation results

The simulation results presented in this section will be used as a basis for the analysis for the set-point optimisation presented in section D.5. On Fig. D.2 the top plot shows the performance function and the remaining plots each of the terms from the performance function versus Po_{ref} at three different loads, "Low, Medium and High", which in the simulation corresponds to a change in the ambient temperature of the display cases. As shown in Fig. D.2, by increasing the load, the minimums are shifted to the right on the Po_{ref} axis. The inverted COP term is not changed in different loads. The performance function is highly correlated with the switch frequency and this term mostly shapes the behavior of the performance function.

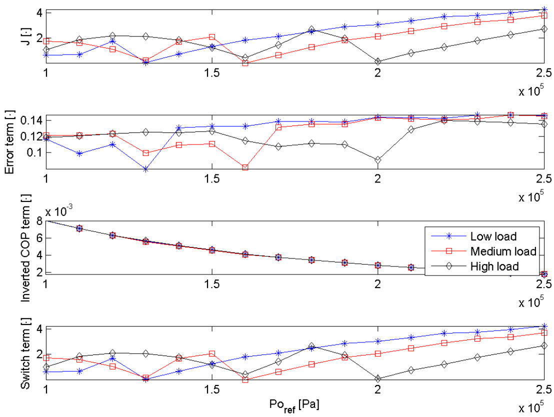

Figure D.2: Performance function and each of the terms plottet versus Po_{ref} at different load levels

Fig. D.3 shows the performance function plotted versus Po_{ref} and the load, $\dot{Q}_{airload}$ and it can be seen that the reference for the suction pressure is dependent on the load, $\dot{Q}_{airload}$. Increasing load clearly calls for a higher suction pressure reference if optimal operation should be maintained. The same conclusion can be made by looking at Fig. D.4 and Fig. D.6. The contribution from the inverted COP term is shown in Fig. D.5, which represents the curve form that will usually be used for optimising a refrigeration plant. Hence, using the performance function (D.1) as shown on Fig. D.3 for choosing the optimal suction pressure will present a better set-point than solely basing the choice on the inverted COP.

D.5 Set-point optimisation

The set-point optimisation has to be based on the available information in the system. The optimisation method can not assume that there exists a sufficiently detailed model of any given refrigeration plant. Thus, it will not be possible to predict or estimate the performance for any given set-point. In addition, the method cannot use a benchmark data set, because it does not exist, and it is not feasible to assume the existence of such a data set for any given supermarket. However, it is feasible to online generate a limited data set.

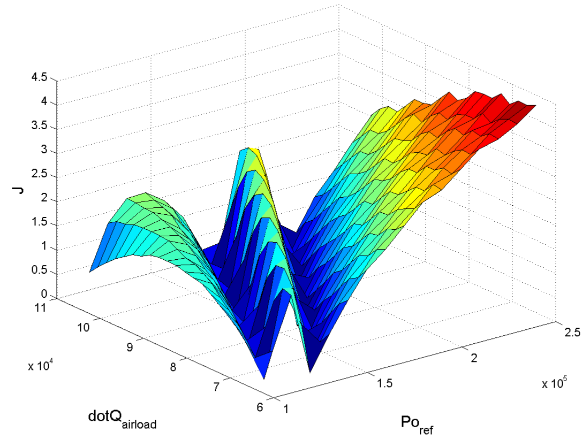

Figure D.3: Performance function versus Po_{ref} and $\dot{Q}_{airload}$

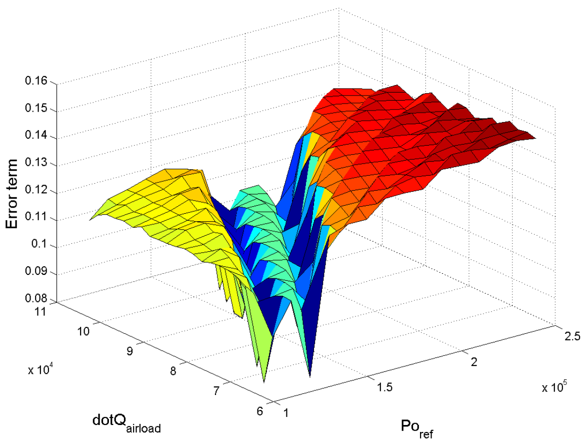

Figure D.4: Error term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

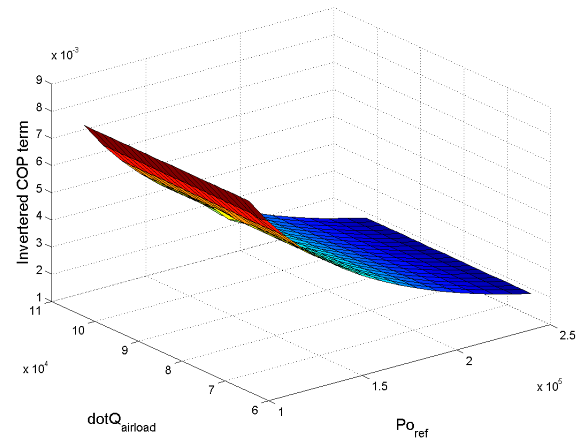

Figure D.5: Inverted COP term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

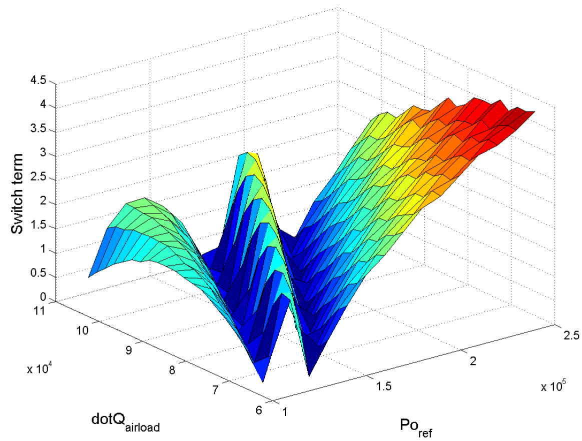

Figure D.6: Switch term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

Considering the above simulation results and problem statement, interpolation is chosen as a method to find the optimal set-pont.

D.5.1 Interpolation

Interpolation is a technique used to estimate unknown values that lie between known values, see chapter 6 in [7]. A linear interpolation of the following form is been used:

$$Po_{ref} = Po_{ref,a} + (Po_{ref,b} - Po_{ref,a}) \cdot \frac{\dot{Q}_{airload} - \dot{Q}_{airload,a}}{\dot{Q}_{airload,b} - \dot{Q}_{airload,a}} \tag{D.15}$$

More sophisticated interpolations are available and often applied to datasets with irregular spacing. However, in this case, the linear interpolation technique is used for set-point optimisation purpose because of the speed of the method, accurate response and the linear behaviour of the system performance with respect to the change on load and set-point pressure, see Fig. D.7. Linear interpolation requires at least two sweeps of the suction pressure reference Po_{ref} at different values of the disturbance $\dot{Q}_{airload}$. According to the previous analyses, optimal set-points have been shifted to the right on the Po_{ref} axis, see Fig. D.2 or D.3. Analysis of Fig. D.3 through D.6, shows that the optimal point is changing linearly with respect to the change in the disturbance, $\dot{Q}_{airload}$. Therefore, by means of three or at least two optimal set-points, an optimal set-point line at different values of $\dot{Q}_{airload}$ can be interpolated D.7. Therefore, the optimisation problem could be solved by determining the value of $\dot{Q}_{airload}$ at each instance and then set Po_{ref} to the corresponding value of $\dot{Q}_{airload}$ base on the knowledge gain by the two sweeps and the interpolation. Since the interpolation approach is based on the ability to do at least two sweeps at different values of $\dot{Q}_{airload}$, which is a disturbance, the method relies on that the disturbance changes over time and thereby renders sweeping at different load situations possible. The proposed algorithm can be described as follows:

1. For a given $\dot{Q}_{airload}$ sweep Po_{ref} and save the values of $J(t)$
2. Find the minimum value $J(t)$ and use the corresponding Po_{ref} as the optimal set-point for the given $\dot{Q}_{airload}$
3. When $\dot{Q}_{airload}$ changes significantly, repeat step 1 and 2 and establish the linear interpolation as suggested in (D.15).
4. Use the interpolated line to chose the optimal suction pressure reference, Po_{ref}, for future $\dot{Q}_{airload}$.

The load change between opening and closing hours of the supermarket system will be sufficient to provide a good interpolation results. However, extending the algorithm to update the interpolation when there has been a seasonal change might be a good idea. The proposed method will provide a close to optimal

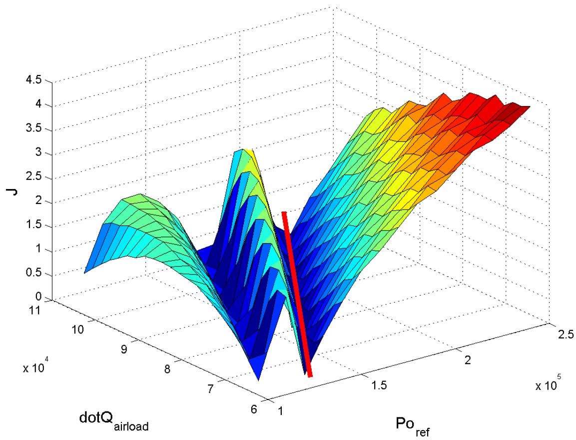

Figure D.7: Optimal interpolated line

set-point based on the interpolation.

D.6 Conclusion

Optimising the global steady state performance of a supermarket refrigeration system has been the focus of this paper. This has been done by concentrating the optimisation effort on the choice of set-point for predominate controller for the plant, which is the compressor controller. The paper presents a method for handling the problem of choosing the optimal suction pressure reference with minimum knowledge of the refrigeration plant. The problem is solved by utilising the general knowledge gained by examining the simulation results based on model of a supermarket refrigeration system. The results clearly indicates that the optimal set-point is dependent on the load on the system, which is a disturbance in the system. The proposed method indicates that the set-point should be adjusted when the load changes significantly. The proposed method does not rely on the existence of a detailed model of the system or the availability of benchmark data.

Paper E

Plant-wide performance optimisation – The refrigeration system case

Torben Green, Roozbeh Razavi-Far, Roozbeh Izadi-Zamanabadi and Henrik Niemann. Plant-wide performance optimisation – The refrigeration system case. Accepted for presentation at the *2012 IEEE Multi-Conference on Systems and Control*. Dubrovnik 2012.

Abstract: This paper investigates the problem of plant-wide performance optimisation seen from an industrial perspective. The refrigeration system is used as a case study, because it has a distributed control architecture and operates in steady state conditions, which is common for many industrial applications in the process industry. The paper addresses the fact that dynamic performance of the system is important, to ensure optimal changes between different operation conditions. To enable optimisation of the dynamic controller behaviour a method for designing the required excitation signal is presented. Furthermore, invasive weed optimisation is used to find the optimal parameters for local controllers based on the plant wide performance measure.

E.1 Introduction

Achieving performance optimisation in plants with multiple interconnected subsystems in practice is far from being a trivial task. To illustrate the complexity of such tasks a routine installation and commissioning process in supermarket refrigeration systems is described in the following: An installer connects the control instrumentation into the display cases and uses the parameter set that he has used in a similar system (that can actually be significantly different from the one he is working on) and starts the control system. If the controller stabilises the system and can track the set point to a certain degree then he will be content and no more effort is spent on optimising the controller. If there is a need for more optimisation it will primarily be carried out on local subsystems.

As the predominant used controllers in the process industry are of PI(D) types there exist a number of optimisation/ tuning methods that can be utilised to modify the controller parameters in order to achieve improved performance (see for instance, [1, 2, 48]. However, these techniques/ methods are developed in order to achieve local performance optimisation.

When considering supermarket refrigeration systems, a challenging issue appears, which is due to the fact that refrigeration systems are usually operating in steady state conditions and, therefore, will not be affected by a change in controller parameters unless the dynamics of the system is excited. This leads to the problem of designing auxiliary signals for a given subsystem that can be used for optimisation without affecting the performance of the subsystem. The problem is accumulated by the fact that there is little or no á priori knowledge of the underlying system dynamics.

In complex plants, such as large supermarket systems, when the subsystems are coupled and interact dynamically, local tuning of individual controllers for each of the subsystems does not guarantee that an optimal plant-wide performance can be achieved. Furthermore, due to the lack of knowledge of the underlying dynamics, it is very difficult to use the same techniques/methods that are typically used for local control tuning / subsystem optimisation. The optimisation techniques in these methods are often gradient based and utilise the derivative of a performance function. The commonly stated problems with these *derivative-based* optimisation methods are:

- Gradients may not be available or it can be impractical to calculate analytic gradients,
- Noise or non-smoothness in the performance function can makes finite differences inaccurate,

An alternative is utilisation of *derivative-free* search algorithms, which use the performance function and constraints to steer towards the optimal solution. Since derivative information is not used, the direct search methods are typically slow, requiring many function evaluations for convergence. Recently, genetic algorithm [11], particle swarm optimisation [25], ant colony optimisation [9], simulated annealing [37] and tabu search [38] have been extensively used for optimisation and have shown high capability of searching for global minimum in different engineering applications [6, 45]. Evaluation of the algorithms have not been the focus in this work and the reader is therefore encouraged to consult the literature for a comparison of the different algorithms. In this work, invasive weed optimisation algorithm, IWO, is employed for plant-wide performance optimisation. IWO, which is introduced in [31] for the first time, is a bio-inspired numerical optimisation algorithm that simply simulates natural behaviour of weeds in colonizing and finding suitable place for growth and reproduction. The idea behind this paper is to plant-wide optimise the performance of the refrigeration plant, using invasive weed optimisation to find the parameters, k_c and T_i, for the PI controllers.

E.2 Problem formulation

The governing problem is to globally optimise the operation of the supermarket refrigeration system. To achieve global optimisation a performance function is required and a formalised method to search for the solution is required. In addition, an excitation signal is required to ensure that the effect of changes in the controller parameters are visible from the performance function. The excitation signal is required because refrigeration systems are operating in steady state most of the time. To give a better insight to the problem an short description of a supermarket refrigeration system will be presented hereafter. The refrigeration system is presented as simple diagram on Fig. E.1. In a supermarket the compressors are usually placed in a machine room and then parallel connected to the display cases in the sales area. The condenser unit, which is usually places outside the supermarket, receives compressed gas from the compressors and condenses the gas into liquid by rejecting the heat to the surroundings. The outlet of the condenser is parallel connected to the display cases and can therefore supply liquid refrigerant to them.

The control task of for each of the display cases is to maintain a predefined air temperature inside the cabinet. That is done by measuring the air temperature and then using a PI controller manipulating the inlet valve to the evaporator. The suction/evaporation pressure is chosen to deliver a sufficient temperature difference in the display cases to create sufficient heat transfer. The control task for the compressor rack is then to deliver the chosen suction pressure. The compressor rack control is based on a PI controller followed by a step controller. The PI controller calculates the required compressor capacity which is used by the step controller to decide how many compressors that should be running. System wide optimisation of a supermarket refrigeration system can of

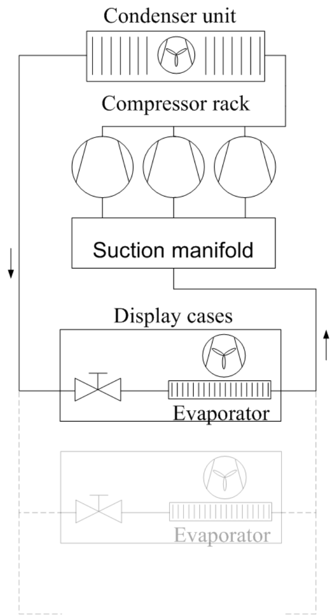

Figure E.1: Simplified supermarket refrigeration layout

course only be achieved with a predefined performance measure. In this paper, the performance function, (E.1), presented in [13] will be used to assess the operational performance.

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}_k(t)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP_l(t)}||_R^2 + \sum_{p=1}^{P} ||f_{sw_p}(t)||_S^2 \tag{E.1}$$

The first term in (E.1) represents the food quality by the control errors in K controllers and the second term is a measure of the efficiency of the plant at different suction pressures using the inverted coefficient of performance, COP, for L suction levels, and the last term is the switch frequency of the P compressors which is used to avoid excessive wear on the compressors. Q, R and S are cost weights with appropriate size.

As mentioned previously, in order to assess and optimise the system performance there is a need for dedicated auxiliary signals. The problem is to design a set of signals, $\{\eta_1, \cdots, \eta_m\}$, that lie in the appropriate frequency range of their corresponding closed loop systems. Such design is not a trivial task as the dynamics of the controlled subsystems is not known in advance. The knowledge regarding the system under investigation is limited to a structural level. Meaning that time constants and gain of the subsystems is unknown. Thus, the design methods such as the ones introduced in [33] and [40] cannot be directly applied since they rely on model knowledge.

Since the aim is that the active performance assessment setup should run online on a supermarket refrigeration system, compromising the operation of the system will not be acceptable. Hence, the excitation signal cannot be chosen without considering the impact on the operation of the plant. In addition, choosing the amplitude for the external excitation signal is an online task, because the internal generated excitation is dependent on the operation point.

In this work the subsystems that the method will be applied on are the display cases in the supermarket refrigeration system. In order to describe the problem in mathematical terms a number of facts and assumptions are introduced in the sequel:

1. The closed loop consists of a PI controller and a physical system, here a display case, for which the dynamics are unknown.
2. The parameters of the PI controller are known.
3. The existing controller is stabilising the closed loop.

Assumptions:

1. The system dynamics can be described by a first order differential equation with time delay.

2. The response time delay to an abrupt change in control action is known.

The first assumption can be verified by consulting literature such as [18, 43, 44]. The time delay in the second assumption can be measured online.

Let us consider the k^{th} subsystem (here a display case). Now the following notation are introduced:

The subsystem dynamics is described by a first-order-plus-dead-time (FOPDT) process model::

$$G_k(s) = \frac{k_{p_k} e^{-T_{d_k} s}}{\tau_k s + 1},$$

where T_{d_k} denotes the time delay and k_{p_k} and τ_k denotes the gain and time constant, respectively. The corresponding controller $C_k(s)$ is of PI type, i.e.

$$C_k(s) = k_{c_k} + \frac{k_{i_k}}{s}.$$

The closed loop transfer function is then given by:

$$H_k(s) = \frac{C_k(s)G_k(s)}{1 + C_k(s)G_k(s)}.$$

The performance measure for the total system is defined as given in (E.1). For a given subsystem k, find an appropriate set of excitation signals $\{\eta_1(t), \cdots, \eta_m(t)\}$ and then use them to tune the controller parameters $\{k_{c_k}, k_{i_k}\}$ so that the performance function $J(t)$ is minimised, i.e.

$$\begin{aligned} & \underset{k_{c_k}, k_{i_k}}{minimise} && J(t) \\ & subject\ to && y_k(t) = (h_k * \zeta)(t), \quad \zeta(t) = \sum_{i=1}^{m} \eta_i(t). \end{aligned} \tag{E.2}$$

where y_k denotes the output of the k^{th} system. '$*$' represent the convolution and represent the time domain description of $y_k(s) = H_k(s)\zeta(s)$. The excitation signal is represented by $\zeta(t)$.

There are hence two main problems to solve: 1) How to design the excitation signals?, and 2) what is the suitable strategy to optimise the controller parameters in a time efficient manner? The ensuing sections address these two problems.

E.3 Design of the excitation signals

Following choice/simplification is been made:

$$\zeta(t) = \sum_{i=1}^{m} \eta_i(t) = \sum_{i=1}^{m} \eta(\omega_i t) \tag{E.3}$$

It means that ζ is a sum of identical functions which are operating with different frequencies ω_i. We choose sinusoidal functions as a candidate, i.e.

$$\eta_i(t) = A_i \sin(\omega_i t). \tag{E.4}$$

The signal is added to the closed loop as shown on Fig. E.2. In the sequel the

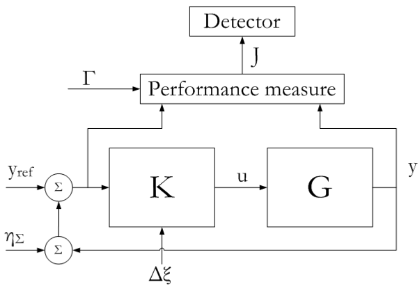

Figure E.2: Active performance assessment setup with excitation added to the measurement

process of choosing frequencies and the magnitude of each excitation signal will be described.

E.3.1 Frequency

When the controller parameters, i.e. k_c and k_i, are known, the corresponding system parameters, i.e. τ and k_p, can be estimated based on the desired gain and phase margins, see [14]. The estimated system parameters can be used as the best guess to compute the cutoff frequency of the resulting closed loop transfer function, denoted by ω_c. In practice, the real system parameters may deviate from the estimated ones by up to 100%. Therefore, in order to make sure that the resulting excitation signal has the proper impact on the system dynamics a set of frequencies should be chosen. Following strategy can be adopted:

$$\omega_i = \frac{1}{10}\omega_{i+1} \qquad \forall i = 1, \cdots, m-1 \tag{E.5}$$

where

$$\omega_{\frac{m+1}{2}} = \omega_c$$

and m is chosen to be an odd number (a good choice would be 5 or 7).

E.3.2 Magnitude

When the choice of the excitation signal is based on the bandwidth of the closed-loop transfer function and as the assumption is that the current controller is stabilising the system then one can choose the magnitude of excitation signals in accordance with the following practice: For industrial system a boundary is typically defined, within which the set-point can vary. Lets denote the boundaries as, SP_{upper} and SP_{lower}. Furthermore, consider the PI controller as a sum of two terms, the P-term and the I-term. The magnitude of the excitation signal should hold the following conditions:

$$SetPoint + ||\zeta||_\infty < SP_{upper} \tag{E.6}$$

$$SetPoint - ||\zeta||_\infty > SP_{lower} \tag{E.7}$$

Since according to (E.3) and (E.4), ζ is a sum of bounded continuous functions then it is bounded and hence has extremum (here maximum/minimum). Lets assume that the controller has been tracking a non-zero set-point and the system has reached its steady-state conditions. In this state, due to the insignificant tracking error, the contribution of the P-term in the PI controller is negligible. Hence the I-term is the only term which is utilised in order to maintain the set-point tracking. A suggestion for a choice for $||\zeta||_\infty$ is the following:

$$||\zeta||_\infty = \alpha \left| \frac{\textit{I-term}}{\hat{k}_{p_k}} \right|, \tag{E.8}$$

where $\alpha \in \{0.01 \cdots 0.25\}$ and $\hat{k}_{p_k}$ is the estimated system gain. α should be chosen as high as possible as long as conditions given by (E.6), and (E.7) are not violated. A typical choice would be $\alpha = 0.15$. In order to realize ζ with correct magnitude the amplitude of the signals in Eq. E.4 are given by:

$$A_i = \alpha \left| \frac{\textit{I-term}}{m\,\hat{k}_{p_k}} \right| \quad \forall k \in \{1, \cdots, m\}. \tag{E.9}$$

The next step, after determining the characteristics of the excitation signals for each subsystem, is to employ an appropriate method to optimise the plant-wide performance.

E.4 Invasive Weed Optimsation (IWO)

The IWO is inspired from the phenomenon of colonization of invasive weeds in nature and became a powerful optimsation algorithm by capturing the natural properties of the invasive weeds [31]. Weeds invade a cropping field by means of dispersal and occupy opportunity spaces between the crops. Each invading weed takes the unused resources in the field and grows to a flowering weed and creates new weeds, separately. The number of newly produced weeds by each flowering weed is subject to the fitness of that flowering weed in the colony. Those weeds that take more unused resources and have better adoption to the environment can rapidly grow and produce more seeds. Consequently, the newly produced weeds are randomly dispread over the field and grow to flowering weeds. Due to the limited resources, this procedure carries on until the maximum number of weeds on the field is reached. Now, only those weeds with higher fitness can survive and produce new weeds. This competitive evolution among the weeds causes their well adaption to the environment and improvement over the time.

E.4.1 IWO Algorithm

At the beginning, prior to explaining the IWO algorithm, the key terms are introduced here as follow: Each individual or agent in the colony, so called seed, contains a value of each optimsation variable. A value represents the goodness of the solution for each seed is called fitness. Each seed grows to a flowering plant in the colony; so the meaning of a plant is one individual after evaluating its fitness. Colony indicates the entire agents or seeds. The number of plants in the colony is called population size. Therefore, growing a seed to a plant corresponds to evaluating an individual's fitness or performance.

The following steps are considered to simulate the colonizing behaviour of weeds [31]:

1. Search space definition: Initially, the number of parameters that need to be optimised has to be defined, hereafter denoted by D. Next, for each parameters in the D-dimensional search space, a minimum and maximum value are assigned.

2. Population initialisation: A limited number of seeds are being randomly dispread through the defined search space. Alternatively, each seed catches a random position in the D-dimensional search space. Each seed's position is an initial solution, including D values for the D parameters, of the optimsation problem.

3. Fitness estimation: Initial seeds grow up to flowering plants. A fitness value for each seed is returned by the fitness function, defined to represent the goodness of the solution.

4. Ranking and reproduction: The flowering plants are firstly ranked based on their assigned fitness values prior to generate new seeds. Then, flowering plants can reproduce new seeds depending on their rank in the colony. In other words, the number of seeds produced by each plant can increase from the minimum possible seeds production S_{min}, to its maximum, S_{max} based on plant's fitness values and their ranking. Hereafter, the plants with higher fitness which are more adapted to the colony can produce more seeds that solve the problem better. In this step, it is permitted for all plants to participate in the reproduction competition which adds an important property to the IWO algorithm in searching a global optimum.

5. Dispersion: Here, the seeds are being randomly dispread through the search space by using normally distributed numbers with mean equal to the position of the producing plants and varying standard deviations. The standard deviation σ at the current time step can be explained by:

$$\sigma_{iter} = \frac{(iter_{max} - iter)^n}{(iter_{max})^n} (\sigma_{initial} - \sigma_{final}) + \sigma_{final} \tag{E.10}$$

   where $\sigma_{initial}$ and σ_{final} denote to initial and final standard deviations, respectively. $iter_{max}$ indicates the maximum number of iterations and n is the nonlinear modulation index. The σ can be reduced from the $\sigma_{initial}$ to the σ_{final} with different velocities in accordance with the chosen nonlinear modulation index, n. Initially, the whole search space can be explored by the IWO algorithm due to the high value of initial standard deviation $\sigma_{initial}$. Then, the standard deviation σ is gradually reduced by increasing the number of iterations, to focus the search around the local minima or maxima to find the global optimum.

6. Competitive exclusion: Afterwards all seeds have been placed in their positions in the search space; the new seeds grow up to the flowering plants and subsequently, they are all ranked along with their parents according to their fitness. Then, weeds with lower fitness are eliminated to reach the predefined criterion of maximum number of plants in the colony, P_{max}. Apparently, the number of fitness evaluations, the population size, is more than the maximum number of plants in the colony. This 'survival of the best fit' mechanism, inspired from the theory of natural selection [20], provides a chance to plants with lower fitness to reproduce and survive in their seed's existence, if their seeds have good fitness.

7. Termination Condition: Survived plants can produce new seeds based on their fitness rank in the colony. The whole process is continues until either

the maximum number of iterations has been reached or the fitness criterion met.

E.5 Simulation setup and Results

This section presents the results obtained by utilising the invasive weed optimisation for plant-wide optimisation of the dynamic operation of the supermarket refrigeration system. However, before presenting the results the simulation setup will be described.

E.5.1 Simulation setup

The optimisation approach has been tested on a simulation model of a supermarket refrigeration system. The simulation setup is described in detail in [15]. The model is describing a supermarket refrigeration setup that is comprised of two identical display cases and a compressor rack with two compressors and an ideal and static condensing unit.

The optimisation algorithm has been used to optimise the gain, k_c, and the integration time, T_i, of the PI controller for the display cases. Since the feasible set for the two parameters are not overlapping constraints are required on each of the variables. The search space has been constrained based on the knowledge about the controller behaviour. In addition, to allow the IWO algorithm to have at least one good plant, a single seed has been defined by an initial guess of values for k_c and T_i. Thus, ensuring that at least one plant will produce a reasonable performance.

The algorithm was started with the following input parameters:

- Maximum number of generations: $iter_{max} = 150$
- Maximum number of plants: $plants_{max} = 30$
- Maximum number of seeds per plant: $seed_{max} = 5$
- Number of initial plants: $plants_{init} = 5$

The maximum number of evaluations of the simulation, $simrun_{max}$, is given by (E.11).

$$simrun_{max} = (1 + seed_{max}) \cdot plant_{init} + (iter_{max} - 2) \cdot plants_{max} \tag{E.11}$$

With the chosen input parameters for the IWO algorithm $simrun_{max} = 4470$. In this setup the simulation runs for 5000 seconds to ensure that the performance function is in steady state with the given set of controller parameters, k_c and T_i.

E.5.2 Results

After 150 generations of the weed optimisation algorithm produces the results shown on Fig. E.3 On Fig. E.3 the minimum, maximum and mean fitness or

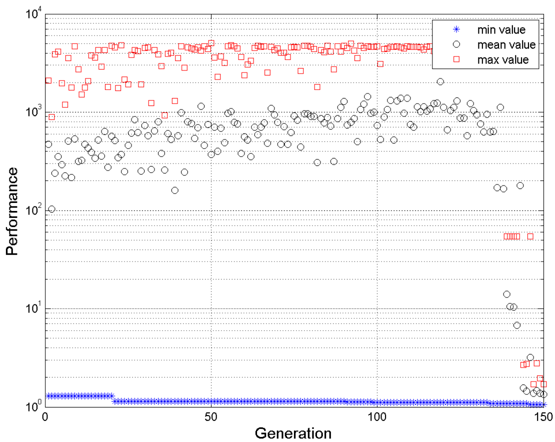

Figure E.3: Generations versus the performance function J. The best achived performance after 150 generations is 1.0546

performance of the refrigeration system is plotted for each generation of the IWO algorithm. It can be seen that the maximum, the mean and the minimum performance converges. Thus, the results is accepted as reasonable.

Fig. E.4 shows the evolution of the best and worst T_i versus the generations of the optimisation algorithm, and it is clear to see that both the worst and the best parameter result of each generation converges toward the same region in the search space.

Fig. E.5 shows the evolution of the best and worst k_c versus the generations of the optimisation algorithm. In addition, Fig. E.5, shows that both the worst and the best parameter result of each generation converges toward the same region in the search space.

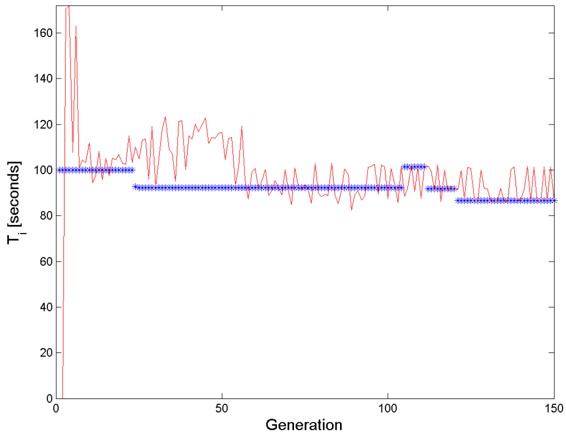

Figure E.4: Evolution of T_i, The red line plots the evolution of the worst choice of T_i and the blue line plots the evolution of the best choice of T_i

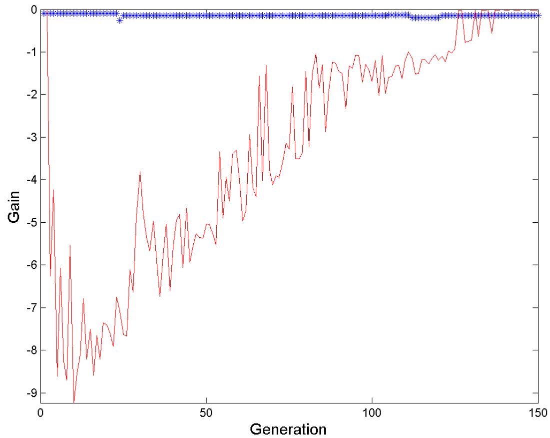

Figure E.5: Evolution of k_c.The red line plots the evolution of the worst choice of k_c and the blue line plots the evolution of the best choice of k_c

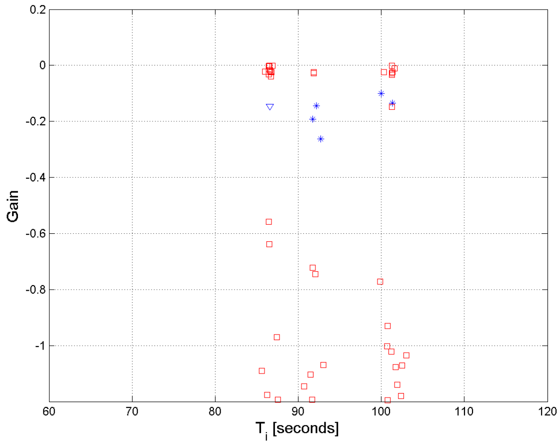

Figure E.6: T_i versus k_c, The red diamond indicates the end result for the parameters, $k_c = -0.1877$ and $T_i = 60.6972$.

Fig. E.6 shows that all the results of the best parameter combination, which is indicated by blue triangles, lies within a small area close to the end result, which is indicated by a red diamond. The parameter values proposed by the IWO algorithm for the controller are, a gain of $k_c = -0.1877$ and an integration time at $T_i = 60.6972$, which can be seen on Fig. E.6. The results presented in Fig. E.3 through E.6 indicate that the approach is feasible for optimising the controller parameters. In addition, the results indicate that the use of initial values for the IWO algorithm is a clear benefit. Especially if the algorithm has to run on a system instead of a simulation model. Narrowing the search space and using initial values for the optimisation parameters would make it possible to decrease the number of generations and thereby reducing the time the algorithm uses to generate a result.

E.6 Conclusion

This paper considered plant-wide performance optimisation for industrial processes that operate in steady state conditions most of the time. A simulated version of a supermarket refrigeration system, which is an example of such plants, was used as a test bench.

The challenge that has been addressed is, that under steady state conditions the performance of the system will not be affected by change in parameter values of a controller of an underlying subsystem, unless the dynamics of the corresponding subsystem is excited. A method of characterising the excitation signal under realistic conditions was presented in the paper. The dynamic performance has been addressed to ensure that the performance of the system will be optimal even when the system is changing between operation conditions.

Furthermore, an optimisation method, based on invasive weed optimisation (IWO), was utilised to optimise the dynamic behaviour of the local controllers based on the defined global performance function. The simulation results showed a clear improvement. The choice of the initial values for the IWO algorithm had a significant impact on convergence speed of the algorithm toward the global minimum, and hence, need be considered with care.

This ensures that the system can maintain optimal operation when the operation conditions changes from one operation point to another.

Paper F

Plant-wide Dynamic and Static Optimisation of Supermarket Refrigeration Systems

Torben Green, Roozbeh Izadi-Zamanabadi, Roozbeh Razavi-Far and Henrik Niemann. Plant-wide Dynamic and Static Optimisation of Supermarket Refrigeration Systems, Submitted to the *International Journal of Refrigeration*

Abstract: Optimising the operation of a supermarket refrigeration system under both dynamic as well as steady state conditions is addressed in this paper. For this purpose an appropriate performance function that encompasses food quality, system efficiency and also component reliability is established. Depending on whether the focus is on optimising the system performance under steady state or dynamic conditions different set of parameters will be subject to optimisation. Focusing on steady state operations the total system performance is shown to predominantly be influenced by the suction pressure. Employing appropriate performance function leads to conclusions on the choice of set-point for the suction pressure that are contrary to the existing practice. The dynamic optimisation requires use of dedicated excitation signals. A method for designing such signals under realistic operational conditions is been suggested. A derivative free optimisation technique based on invasive weed optimisation (IWO) has been utilised to optimize the parameters of the controllers in the system. Simulation results have been used to substantiate the suggested methodology.

F.1 Introduction

In a competitive and global business environment plant-wise performance assessment and optimisation in the process industry have increasingly become important issues as they have direct impact on the operational costs, energy and environmental issues. Supermarket refrigeration systems are no exception: One of the larger operational costs of a supermarket is the refrigeration plant. In a supermarket, the refrigeration system accounts for 40% to 60% of annual electrical energy consumption [10]. Furthermore, there are substantial costs related to component replacement and unscheduled maintenance.

In many industrial systems, it is customary to use over-dimensioned components that provide excess capacity in order to guarantee that the system functionality is provided under all conditions, which are in particular caused by non-optimal operational conditions. For instance, the compressor rack in supermarket systems is typically made of compressors that can provide up to 50% more capacity than the supermarket system is actually designed for. Non-optimal operation not only affects the cooling efficiency and food quality but also have direct impact on the operational life-time of the components. Proper optimization tools/methods can not only be used to optimize the performance of an operating system but also assist the design engineering group to choose appropriate components of right dimensions/capacities at more suitable costs in future plants. An appropriate performance function for plant-wise operation should include contributing terms that describe the quality of products, system efficiency, as well as the operating life-time of the subsystems. Optimisation of refrigeration systems have been attempted in other references such as [18] where multi variable control is used to get a better performance of a vapour compression system. [23] introduces an energy optimal control approach for refrigeration system and in [27] an online steady state energy minimisation is presented, where the minimisation is relying on a model of the refrigeration system.

The strategy toward system-wide performance assessment and optimisation, proposed in this paper, can be described by using Fig. F.1. In Fig. F.1 the

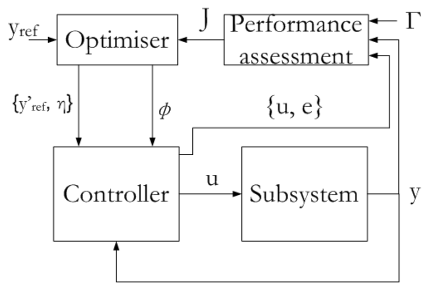

Figure F.1: Block diagram illustrating the system-wide performance assessment and optimisation approach.

optimiser adjusts the behavior of the controller for the subsystem. This is achieved by passing an alternated reference signal $y^{'}_{ref}$ and excitation signal η, or by changing the parameters, ϕ. The output from the optmiser is based on the performance function, J which is the output from the performance assessment block. The performance assessment is base on the control error,e, the output from the controller, u, the output from the subsystem, y, and contribution from the other subsystems, Γ. The contributions from the other subsystems ensures that the system-wide perspective is preserved.

A supermarket refrigeration systems, like many other plants in the process industry, operate in steady state conditions in the majority of their operating time it is reasonable to separate the optimisation task in two parts; the first one will be addressing the plant-wide optimisation in steady state conditions and the second one will be focusing on optimising the local subsystems' dynamic behaviour. To perform the optimisation task an appropriate performance function is proposed in section F.2.3. Optimisation in steady state is typically formulated in terms of set-point optimisation. For considered supermarket systems the suction pressure is chosen as the dominating variable. In section F.4.1 it is shown that the suggested operating set-points for the suction pressure differ significantly from the ones that can be obtained by using performance function that is used in common practice. The knowledge based on investigating the resulting performance under different load conditions is used in section F.4.3 to suggest a simple procedure for determining the pressure set-point that leads to optimal operation for any given load condition. In section F.4.4 the approach is generalised to the case where a compressor rack with three compressors is employed.

Carrying out the dynamic optimisation task will be a challenge as refrigeration systems usually operate in steady state conditions and, therefore, will not be affected by a change in controller parameters unless the dynamics of the local subsystem is actively excited. This leads to the problem of designing/choosing auxiliary signals for a given subsystem that can be used for optimisation without affecting the performance of the subsystem. The problem is accumulated by the fact that there is little or no ťa priori knowledge of the underlying system dynamics. In section F.5.1 the design of appropriate excitation signals will be discussed and a design method will be suggested.

In complex plants, such as large supermarket systems, when the subsystems are coupled and interact dynamically, local tuning of individual controllers for each of the subsystems does not guarantee that an optimal plant-wide performance can be achieved. Furthermore, due to the lack of knowledge of the underlying dynamics, it is very difficult to use the same techniques/methods that are typically used for local control tuning/ subsystem optimisation. As the predominant used controllers in the process industry are of PI(D) types there exist a number of optimisation/ tuning methods that can be utilised to modify the controller parameters in order to achieve improved performance (see for instance, [1, 2, 48]). However, these methods are developed in order to achieve local performance optimisation. The optimisation techniques in these methods are often gradient based and utilise the derivative of a performance function. An alternative is utilisation of derivative-free search algorithms, which use the performance function and constrain values to steer towards the optimal solution. Recently, genetic algorithm [11], particle swarm optimisation [25], ant colony optimisation [9], simulated annealing [37] and tabu search [38] have been extensively used for optimisation and have shown high capability of searching for global minimum in different engineering applications [6, 45]. Invasive Weed Optimisation (IWO), which is introduced in [31] for the first time, is a bio-inspired numerical optimisation algorithm that simply simulates natural behaviour of weeds in colonizing and finding suitable place for growth and reproduction. In this work, invasive weed optimisation algorithm, IWO, is employed for plant-wide performance optimisation by finding the most suitable parameters for the local controllers. A short description of the IWO scheme is presented in section F.5.2. Simulation setup and results are presented (and discussed) in sections F.5.3 and F.5.4. Finally, concluding remarks will be provided in section F.6.

F.2 System description

The considered refrigeration system dealt with in this paper is a simplified supermarket refrigeration system for which a skectch can be seen on Fig. F.2. The refrigeration system is comprised of two display cases, a compressor rack which is comprised of two compressors and a condenser unit. Each of the display cases is fitted with an air temperature controller which adjusts the temperature by manipulating the opening degree of the inlet valve to the evaporator. The control task of the compressor rack is to ensure a certain suction pressure, by switching the compressors on or off, to fit the demand. In this work the condenser unit has been considered to be ideal and is therefore not explained further.

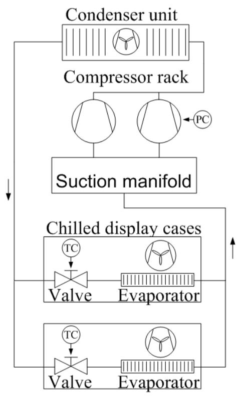

Figure F.2: Schematic of the modelled supermarket system

F.2.1 Model of the simplified refrigeration system

A slightly modified version of the model, for a supermarket refrigeration system, presented in [28] has been adopted. The inlet valves have been modified to enabled the possibility of continuous control which renders it possible to control the air temperature in the corresponding display cases continuously. The mathematical model features two display cases, a suction manifold, a compressor rack and a condenser. The air temperature within each of the display cases is described by (F.1). The heat flow from the surroundings and into the display case is described by (F.2) and is considered to act as a disturbance.

The other terms in (F.1) are described in detail in [28].

$$\frac{\mathrm{d}T_{\mathrm{air,i}}}{\mathrm{d}t} = \frac{\dot{Q}_{\mathrm{goods-air,i}}(\cdot) + \dot{Q}_{\mathrm{load,i}}(\cdot) - \dot{Q}_{\mathrm{air-wall,i}}(\cdot)}{M_{\mathrm{air}} C_{\mathrm{p,air,i}}} \tag{F.1}$$

$$\dot{Q}_{\mathrm{load,i}} = UA_{\mathrm{amb}} \cdot (T_{\mathrm{amb}} - T_{\mathrm{air,i}}) \tag{F.2}$$

$$\frac{\mathrm{d}M_{r,i}}{\mathrm{d}t} = OD_i \cdot \alpha \cdot \sqrt{P_c - P_{suc}} - \frac{\dot{Q}_{e,i}}{\Delta h_{lg}} \tag{F.3}$$

$$\frac{\mathrm{d}P_{\mathrm{suc}}}{\mathrm{d}t} = \frac{\dot{m}_{\mathrm{in-suc}}(\cdot) - \dot{m}_{comp}}{V_{\mathrm{suc}}\ \nabla\rho_{\mathrm{suc}}(P_{\mathrm{suc}})} \tag{F.4}$$

In (F.3) OD denotes the opening degree of the expansion valve and P_c represents the condensing pressure. The heat removed by evaporation is denoted by $\dot{Q}_{e,i}$, and the enthalpy difference across the two-phase region is denoted by Δh_{lg}. For details about the modelling of $\dot{Q}_{e,i}$ and Δh_{lg} see [28].

The suction pressure is assumed to be the same across the low pressure section of the refrigeration system and is therefore modelled as a common state for all of the display cases in the suction manifold. The dynamics of the suction pressure is modelled by (F.4), where ρ_{suc} and $\nabla\rho_{\mathrm{suc}}$ denote the refrigerant density and the pressure derivative of the refrigerant density. The mass flow rate into the suction manifold, $\dot{m}_{\mathrm{in-suc}}$ is described by:

$$\dot{m}_{\mathrm{in-suc}}(M_{\mathrm{r,i}}, T_{\mathrm{wall,i}}, P_{\mathrm{suc}}) = \sum_{i=1}^{I} \frac{\dot{Q}_{\mathrm{e,i}}(\cdot)}{\Delta h_{\mathrm{lg}}(P_{\mathrm{suc}})}, \tag{F.5}$$

which is a sum over the mass flow contributions from the I display cases. The terms Δh_{lg}, ρ_{suc} and $\nabla\rho_{\mathrm{suc}}$ are all refrigerant specific functions which are explained in detail in [28] The mass flow rate created by the compressor rack is described by:

$$\dot{m}_{comp} = \texttt{Cap} \cdot \frac{1}{100} \cdot \eta_{vol,i} \cdot \dot{V}_{sl,i} \cdot \rho_{suc}, \tag{F.6}$$

where the running compressor capacity of the rack is denoted by `Cap` and η_{vol} and $\dot{V}_{sl}$ denotes the volumetric efficiency and the swept volume flow rate, respectively. No dynamics of the condenser is modelled. Hence, the condenser simply defines a static condensing pressure and a static sub-cooling which indicates that the condenser is considered ideal.

F.2.2 Controller structure

The control structure is comprised of PI controller for each of the display cases, which controles the air temperature by manipulating the opening degree of the expansion valve, OD_i. In addition the suction pressure is controlled by a PI controller and switch logic that manipulates the running compressor capacity in the compressor rack. The discrete behaviour of the compressor rack is achieved by describing the delivered compressor capacity, $\texttt{Cap}$ in (F.6) by:

$$\texttt{Cap} = \sum_{i=1}^{i=N} \delta_i \texttt{Cap}_i \tag{F.7}$$

In (F.7) $\delta_i \in \{0, 1\}$ and $\sum_{i=1}^{N} \texttt{Cap}_i = 100\%$. In this particular application two compressors are used i.e. $N = 2$, $\texttt{Cap}_1 = 45\%$ and $\texttt{Cap}_2 = 55\%$. A hysteresis band is applied around each compressor step to avoid excessive switching of the compressors. The layout of the compressor rack is based on a real supermarket system which has the same layout of the compressor rack.

F.2.3 Performance function

Improving the performance of any given plant requires a predefined notion for the performance. Evaluating the performance based on total cost of ownership, TOC, can be seen as the optimal solution. Direct measurement of the TOC is a difficult task and it can therefore be beneficial to identify certain performance criteria for the process and then use them as a performance measure. This idea has been presented in [13] where the following has been identified as relevant performance criteria for a supermarket refrigeration system:

- Food quality
- Energy efficiency
- Reliability

The food quality is monitored by measuring the control errors for the temperature controller and the suction pressure controller. The energy efficiency is measured by using the coefficient of performance, COP, in the setup. The COP is defined as the delivered cooling power divided by the total electrical power consumed. The reliability of the system is measured by the switch frequency of the compressors because it gives an indication of the wear of the compressors. Excessive switching of the compressors will lead to unnecessary wear of the compressors and thereby increase the need for maintenance and thus decreases the general reliability of the refrigeration system.

Collecting all of the three performance criteria in one performance function to provide overview was introduced in [13], and has been used in this work:

$$J(t) = \sum_{k=1}^{K} ||\mathbf{e}(k)||_Q^2 + \sum_{l=1}^{L} ||\frac{1}{COP(l)}||_R^2 + \sum_{m=1}^{M} ||f_{sw}(m)||_S^2 \tag{F.8}$$

The first term in (F.8) is the control errors of K controllers. The second term is the inverted COP of L refrigeration cycles and the third term is the switch frequency of M compressors. The inverted COP is used to ensure that the term has the same properties as the two other terms in connection with performance optimisation. The performance function (F.8) is a sum of quadratic terms where the notation is given by:

$$||\mathbf{x}||_A^2 = \mathbf{x}^T A \mathbf{x}, \tag{F.9}$$

where x is a vector and A is a weight matrix for the particular term.

F.2.4 Normalisation

The performance function should be made of scalable terms that can be easily adapted for a given supermarket, because the main objective is of course to be able to employ the performance function and the related algorithms in various supermarkets (with different subsystems of different sizes and dimensions). To achieve scalability property the performance function has to be comprised of scalable terms. Normalisation of the terms in the performance function provides precisely this scalability. The choice of the weights Q, R and S for the performance function (F.8) will have to reflect the regional regulations on safety requirements as well as local operational expenses, for a particular supermarket .

The normalisation of the terms will be described hereafter. Each display case has a lower and an upper limit and the reference is chosen as the mean value of the temperature limits and this knowledge is then used to achieve the normalisation. By using the same argumentation on the suction pressure controller the normalisation of the error term can be described as:

$$\mathbf{e} = \begin{vmatrix} \frac{2}{(T_{max,i}-T_{min,i})} \cdot (T_{\mathrm{ref,i}} - T_{\mathrm{air,i}}) \\ \frac{2}{(P_{suc,max}-P_{suc,min})} \cdot (P_{\mathrm{sucref}} - P_{\mathrm{suc}}) \end{vmatrix} \tag{F.10}$$

Normalisation of the switch frequency term is achieved by dividing the measured switch frequency with the the maximum allowable switch frequency of the compressors in the rack. Hence, the switch frequency term is normalised in the following way:

$$f_{sw} = \frac{f_{meas}}{f_{max}}, \tag{F.11}$$

where f_{meas} denotes the measured switch frequency and f_{max} denotes the maximum allowable switch frequency of the compressors.

Due to the definition of the COP the term $\frac{1}{COP}$ is unit less and does therefore not need any normalisation. The range of all of the three terms $\mathbf{e}$, $\frac{1}{COP}$ and f_{sw} will after normalisation be in the interval between 0 and 1 and the weight will therefore represent the cost associated with each of the terms. The weights Q, R and S represent the costs, in terms of economic penalties or lost profits, which are associated with the performance of each subsystem. The choice of the weights has to be done based on the impact on the total operation cost from each of the terms in the performance function.

The price associated with temperature requirements for the display cases in the supermarket should be used as a base for the weight in the error term, Q. Too high or low temperature will destroy the stored food and thereby provide a financial loss for the supermarket. The cost of electrical energy is the base for the weight on the inverted COP term, R, since it is basically an efficiency. The weight on the switch term, S, has to be based on all the cost associated with replacing and maintaining a compressor in the refrigeration system. Because some of the contributions to the cost are hard to define a significant freedom for manipulating the weights based on intuition is maintained. The simulation data presented in this paper is scaled. Thus, the absolute values of the performance function do not have any physical interpretation.

F.3 Optimisation formulation

The formulations of the optimisation problem will be presented in this section. For the majority of the operation time the supermarket refrigeration system is in steady state. However, the quality of the operation is still important under dynamic behaviour. Thus, the problem of optimising the general operation of the plant has been split into two separate tasks, where the first task is the optimisation of the steady state performance and the second tasks is optimising the dynamic performance. From here on they will be referred to as static optimisation and dynamic optimisation, respectively. The static and dynamic optimisation will be described in detail in F.4 and F.5, respectively. Depending on whether the optimisation focus is on static or dynamic performance, different parameters will be subject to optimisation. For the static optimisation case the proposed strategy is to identify the predominant subsystem, with respect to the performance measure for the entire refrigeration plant. The optimisation parameters should then be chosen to be the parameters with highest influence on the performance. Parameters with these characteristics are usually the set-points for the controllers. In the case of the supermarket refrigeration system the predominant subsystem is the suction pressure control loop and the optimisation parameter is the suction pressure reference. In the dynamic optimisation case the controller parameters will be the optimisation variables. The common optimisation problem can be formulated by rewriting the performance function in (F.8) as:

$$J(\phi(t)) = \sum_{i=1}^{I=K+M+L} J_i(\phi(t)), \tag{F.12}$$

where $J_i(\phi(t))$ is the local performance function for the i^{th} subsystem which is depending on the parameter set $\phi(t)$.

F.4 Static performance Optimisation

As supermarket refrigeration systems operate in steady state conditions most of the time (i.e. over 80% of the time) it makes sense to first focus on optimising the system performance for steady state operations, henceforth denoted static performance optimisation. When the main objective is to optimise the static performance one should look for the subsystems that have the highest impact on the total performance of the plant. For these systems, the conditions and objectives of the operation, are predominantly realized through the choice of appropriate set-points.

$\phi(t)$ in (F.12) is the set of optimisation variables which, in the static case, is idefined as:

$$\phi(t) = \left\{ Po_{ref}(t),\ \dot{Q}_{airload}(t) \right\}, \tag{F.13}$$

where Po_{ref}, denotes the suction pressure reference for the compressor rack controller, which is the controllable variable and $\dot{Q}_{airload}$ denotes the heat loss to the surroundings in the supermarket sales are, which is considered to be the main disturbance in the system.

Set-point optimisation of the supermarket is defined as:

$$\min_{Po_{ref}} J(\phi(t)) \qquad \forall\, \dot{Q}_{airload} \tag{F.14}$$

The optimisation problem is to find the optimal value for $Po_{ref}(t)$ that minimises $J(\phi(t))$ for any given disturbance $\dot{Q}_{airload}$. To solve the optimisation problem it is required to obtain deeper knowledge about the profile of the performance function $J(\phi)$ over the relevant search space ϕ. This can be done by either carrying out comprehensive field tests or simulating the case by using appropriate models of the plant.

F.4.1 Simulation setup and Results

The simulation model presented in section F.2.1 has been used for the purpose. Hence, the simulated refrigeration system is comprised of two display cases and a compressor rack comprised of two compressors. Simulations under varying loads and suction pressures have been performed to get a deeper knowledge about the performance function over a realistic search space. Investigation of simulation results leads to a proposed procedure for choosing the optimal set-point under any given load conditions.
In the simulations, the suction pressure reference has been changed in steps from $1.0 \cdot 10^5$ $[Pa]$ to $2.5 \cdot 10^5$ $[Pa]$ with a step size of $0.1 \cdot 10^5 [Pa]$ or the equivalent of changing the evaporation temperature approximately $2°C$. For each of the steps in the suction pressure reference the disturbance, $\dot{Q}_{airload}$, and thereby also the load of the system has been changed by changing the ambient temperature of the display cases in steps from 18 $[°C]$ to 28 $[°C]$, with a step size of $0.5°C$. In these simulations, steady state values of various measurements have been used.

On Fig. F.3 the top plot shows the performance function and the remaining plots each of the terms from the performance function versus Po_{ref} at three different loads, *Low, Medium* and *High*, which in the simulation corresponds to a change in the ambient temperature of the display cases. As shown in Fig. F.3, by increasing the load, the minimums are shifted to the right on the Po_{ref} axis. The inverted COP term is not changed in different loads. The performance function is highly correlated with the switch frequency and this term mostly shapes the behavior of the performance function.

Fig. F.4 shows the performance function plotted versus Po_{ref} and the load, $\dot{Q}_{airload}$ and it can be seen that the reference for the suction pressure is dependent on the load, $\dot{Q}_{airload}$. Increasing load clearly calls for a higher suction pressure reference if optimal operation should be maintained. The same conclusion can be made by looking at Fig. F.5 and Fig. F.7. The contribution from the inverted COP term is shown in Fig. F.6, which represents the curve form that will be used for optimising a refrigeration plant. Hence, using the performance function (F.8) as shown on Fig. F.4 for choosing the optimal suction pressure will present a better set-point than solely basing the choice on the inverted COP.

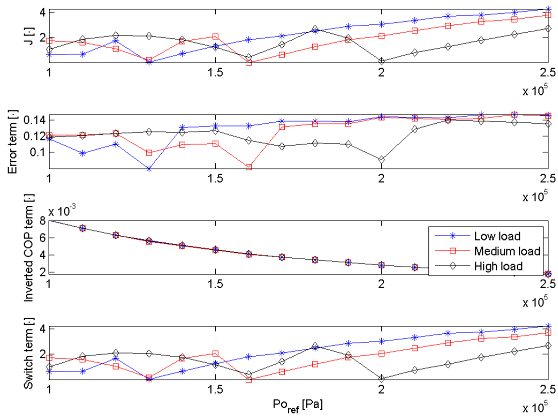

Figure F.3: Performance function and each of the terms plotted separately versus Po_{ref} at different load levels. The top plot is the performance function and the following plots are the error term, the inverted COP term and the switch term of the performance function.

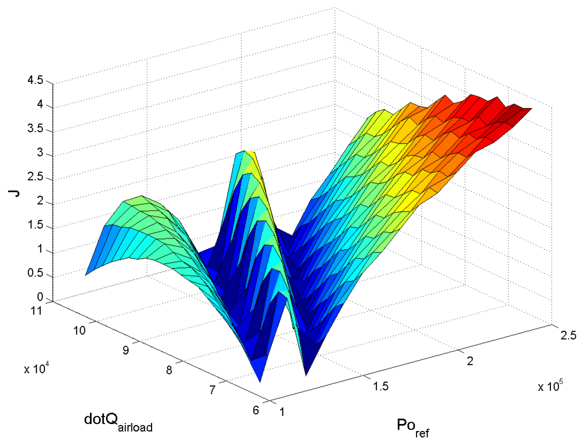

Figure F.4: Performance function versus Po_{ref} and $\dot{Q}_{airload}$

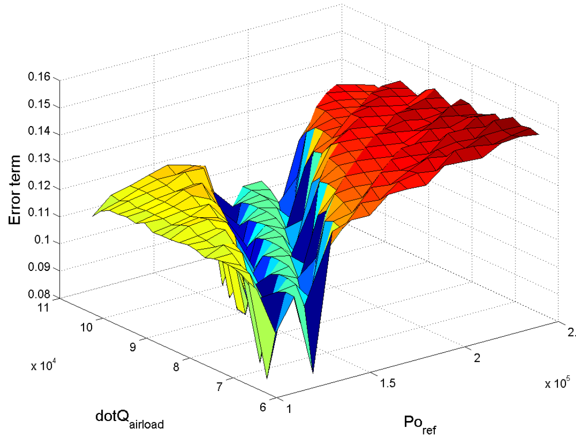

Figure F.5: Error term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

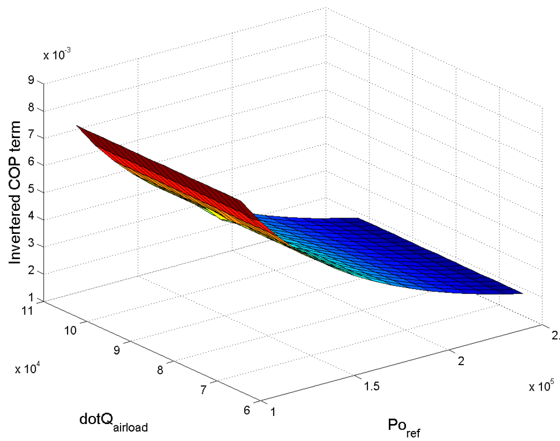

Figure F.6: Inverted COP term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

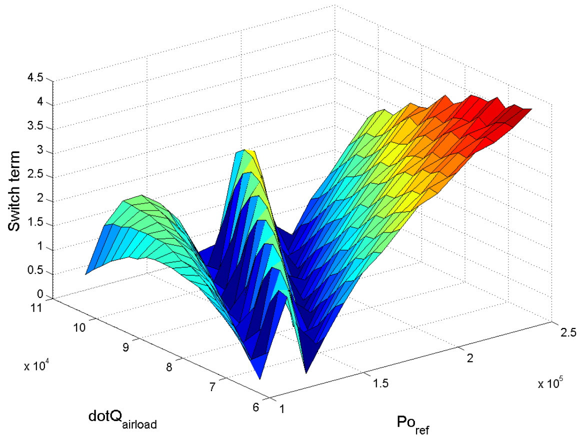

Figure F.7: Switch term from the performance function versus Po_{ref} and $\dot{Q}_{airload}$

F.4.2 Set-point optimisation

Under realistic conditions generating complete set of data is not a viable solution. Nor it is normally possible to use a sufficiently detailed model of a given (arbitrary) refrigeration plant. However, it is feasible to online generate a limited data set. Taking these constraints into considerations and studying the simulation results lead to an interesting observation; the optimal set-points for suction pressure lie along two lines on the search space, see Fig. F.8. Analysis of Fig. F.4 through F.7, shows that the optimal point is changing linearly with respect to the change in the disturbance, $\dot{Q}_{airload}$. Therefore, by means of three or at least two optimal set-points, an optimal set-point line at different values of $\dot{Q}_{airload}$ can be interpolated F.8. These lines can be sufficiently characterised by a linear interpolation of the following form:

$$Po_{ref} = Po_{ref,a} + (Po_{ref,b} - Po_{ref,a}) \cdot \frac{\dot{Q}_{airload} - \dot{Q}_{airload,a}}{\dot{Q}_{airload,b} - \dot{Q}_{airload,a}} \tag{F.15}$$

Therefore, the optimisation problem could be solved by determining the value of $\dot{Q}_{airload}$ at each instance and then set Po_{ref} to the corresponding value of $\dot{Q}_{airload}$ based on the knowledge gained by the two sweeps and the interpolation. Since the interpolation approach is based on the ability to do at least two sweeps at different values of $\dot{Q}_{airload}$, the method relies on the changes of the disturbance over time and thereby renders sweeping at different load situations possible. The load change between opening and closing hours of the supermarket system will be sufficient to provide a good interpolation results. However, extending the algorithm to update the interpolation when there has been a seasonal change might be a good idea.

F.4.3 Choosing strategy of the optimal set-point

For a given/estimated load the optimal set-point for the suction pressure need be identified. This can be done based on calculation of the minimum distance from the current operating conditions, i.e. $(Po_{ref,current},\ \dot{Q}_{airload,current})$ to the interpolated lines. The procedure is described in the following:

The distance d_i between the point $(Po_{ref,current},\ \dot{Q}_{airload,current})$ and the i^{th} line $\lambda_i: \quad Po_{ref} = m_i\dot{Q}_{airload} + b_i, \quad i \in \{1,2\}$ is given by the following formula:

$$d_i = \frac{|Po_{ref,current} - m_i\dot{Q}_{airload,current}) - b_i|}{\sqrt{m_i^2+1}} \tag{F.16}$$

The distance d_i between the point $(Po_{ref,current},\ \dot{Q}_{airload,current})$ and the i^{th} line $\lambda_i: \quad Po_{ref} = m_i\dot{Q}_{airload} + b_i, \quad i \in \{1,2\}$ is given by the following formula:

$$d_i = \frac{|Po_{ref,current} - m_i\dot{Q}_{airload,current}) - b_i|}{\sqrt{m_i^2+1}} \tag{F.17}$$

The candidate line, λ^* , will be the one that lies within the shortest distance from the working point, i.e.

$$\lambda^* = \lambda_i, \quad \textit{such that:} \quad \forall j \neq i, \quad d_i < d_j \tag{F.18}$$

$$\lambda^* = \lambda_i, \quad \textit{such that:} \quad \forall j \neq i, \quad d_i < d_j \tag{F.19}$$

if $d_i = d_j$ then the candidate line should be chosen as the one that goes through the higher suction pressure.

The proposed method will provide a close to optimal set-point based on the interpolation.

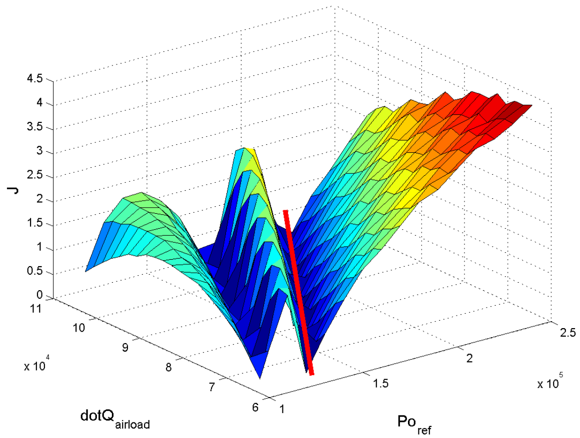

Figure F.8: Performance function versus Po_{ref} and $\dot{Q}_{airload}$ with the optimal interpolated line

F.4.4 Generalisation of the method

The considered system corresponds to a small supermarket refrigeration system where the compressor rack consists of two compressors. The compressor rack for larger supermarkets contains three or more compressors. Use of more compressors implies that it is possible to combine the compressors so that the risk of possible gaps in the delivered capacity is minimised. Furthermore, in compressor racks with more than three compressors it is customary to use at least one frequency controlled compressor. This will virtually remove any risk of gap in the delivered compressor capacity. However, as there is a large segment in the (small-medium) supermarket refrigeration systems where the compressor rack consists of three on-off compressors it is appropriate to generalise the method in order to cover this case as well.

The compressors in the compressor rack is denoted by C_A, C_B and C_C. The corresponding control strategy always pursues the following switching pattern:

$$\underbrace{\{C_A\}}_{Low} \longleftrightarrow \underbrace{\{C_A, C_B\}}_{Medium} \longleftrightarrow \underbrace{\{C_A, C_B, C_C\}}_{High},$$

i.e. start a compressor to deal with demands in low capacity area, then switch the second one when the demand increases to medium level. All compressors are then started in order to meet high capacity demand. There are two possible gaps in the compressor capacities, i.e. one in the capacity area between

$\{C_A\}$ and $\{C_A, C_B\}$ and the other one between $\{C_A, C_B\}$ and $\{C_A, C_B, C_C\}$. Similar to the two-compressor case, the performance of the system within each gap degrades, i.e. the value of the performance function (the cost) increases. Correspondingly, the profile of the performance function will look similar to the two-compressor case. The difference is that in stead of having two interpolated lines that represent the optimal set-points for the suction pressure for given external load, now we have three non-overlapping interpolated lines. The area between two neighboring lines corresponds to a gap in compressor capacities due to switches between two compressor configurations, for instance $\{C_A\}$ and $\{C_A, C_B\}$.
To establish the three interpolated lines it is sufficient to perform a sweep over the suction pressure range for two different load conditions as it is prescribed in section F.4.2. Similarly, for a given load $\dot{Q}_{airload,current}$ the strategy of calculating the corresponding optimal set-point for the suction pressure follows exactly the procedure presented in section F.4.3. The only difference is that there are three interpolated lines, i.e. $i \in \{1, 2, 3\}$, that need be considered.

F.5 Dynamic Performance Optimisation

As stated before optimising the dynamic behaviour of a supermarket refrigeration system with the controller parameters as optimisation variables, creates the need for active performance monitoring. The systems are usually in steady state for a significant amount of time, and a change in controller parameters will not affect the error and therefore not influence the plant-wide performance. Hence, excitation of the dynamics is required to optimise the dynamic behaviour of the system. The setup for active system monitoring is shown on Fig. F.9, as a block diagram of the closed loop under investigation. The controller is

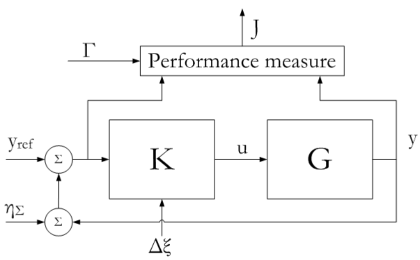

Figure F.9: The general active system monitoring setup

denoted by `K` and the system is denoted by `G`. The input to the system and the measured output is denoted by u and y, respectively and the reference to the controller is denoted by y_{ref}. The applied parameter change is denoted by $\Delta\xi$ and the contributions to the plant-wide performance function, J, is denoted by Γ. The excitation signal is denoted by η_{Σ}. The novel idea is in essence to excited relevant dynamics and then evaluate the current parameter setting with respect to J.

F.5.1 Design of Excitation Signal

A prerequisite for the active system monitoring scheme is the design of an excitation signal. Excitation of relevant dynamics of the system is the purpose and the problem is therefore to design a set of signals, $\{\eta_1, \cdots, \eta_m\}$, that lie in the appropriate frequency range of their corresponding closed loop systems. Since the dynamics of the subsystem in the closed loop under investigation is unknown the design problem is non-trivial. The knowledge regarding the closed loop is limited to a structural level. The use of design strategies from [33] and [39] is therefore restricted.

The intention is that the active performance monitoring setup should run on a supermarket refrigeration system in operation, and it is therefore important that a excitation signal does not compromise the operational performance significantly. In other words, the impact of on the system should be taken into account when the excitation signal is designed.

The air temperature control loop in the display case of a supermarket refrigeration system has been used show case for the proposed method in this work. The following facts are used as base for the design process hereafter:

1. The closed loop consists of a PI controller and a physical system, here a display case, for which the dynamics are unknown.
2. The parameters of the PI controller are known.
3. The existing controller is stabilizing the closed loop.

Furthermore, the following assumptions are used in the design process:

1. The system can be described sufficiently using a first-order model plus dead time (FOPDT).
2. The response time delay to an abrupt change in control action is known.

Verification of the first assumption can be obtained by consulting [18, 43, 44] and the time delay can be measured with sufficient accuracy on the system. The following notation is introduced fot the k^{th} subsystem wich is described as a first-order-plus-dead time (FOPDT) process model:

$$G_k(s) = \frac{k_{p_k} e^{-T_{d_k} s}}{\tau_k s + 1},$$

where T_{d_k} denote the time delay and k_{p_k} and τ_k) denotes the gain and time constant, respectively. The corresponding controller $C_k(s)$ is a PI type, i.e.

$$C_k(s) = k_c + \frac{k_c}{T_i s}.$$

The closed loop transfer function for the subsystem is then given by:

$$H_k(s) = \frac{C_k(s)G_k(s)}{1 + C_k(s)G_k(s)}.$$

For the chosen subsystem k, the task is then to find an appropriate set of excitation signals $\{\eta_1(t), \cdots, \eta_m(t)\}$ which excites relevant dynamics of the system and therefore assists in the task of finding the optimal controller parameters $\{k_{c_k}, T_{i_k}\}$ with respect to the plant-wide performance as describe in (F.8). The optimisation problem can be formulated as:

$$\begin{gathered} \underset{k_{c_k}, T_{i_k}}{minimize}\ J(t) \\ subject\ to: \\ y_k(t) = (h_k * \zeta)(t), \quad \zeta(t) = \sum_{i=1}^{m} \eta_i(t). \end{gathered} \tag{F.20}$$

where y_k denotes the output of the k^{th} system. '$*$' denotes the convolution function and represents the time domain description of $y_k(s) = H_k(s)\zeta(s)$. The excitation signal is represented by $\zeta(t)$. Hence, there exists two main problems that needs to be solved. Firstly, the excitation signal has to be designed and secondly a suitable optimisation strategy has to be applied. The two problems will be dealt with in the following.

The design of the excitation signal has been split into two different tasks. The first task is to determine the frequency content of the signal and the second task is to determine a reasonable amplitude for the signal. To assist the design process the following simplification has been made:

$$\zeta(t) = \sum_{i=1}^{m} \eta_i(t) = \sum_{i=1}^{m} \eta(\omega_i t) \tag{F.21}$$

Thus, ζ is essentially a sum of identical functions with different frequency ω_i. The candidate function is chosen to be a sinusoidal function described by:

$$\eta_i(t) = A_i \sin(\omega_i t). \tag{F.22}$$

The excitation signal is then injected to the system as shown in Fig. F.9.

The following will describe how the frequencies and the magnitude for the excitation signal will be described. Based on the knowledge of the controller parameters, k_c and T_i the corresponding system parameters, i.e. τ and k_p, can be estimated based on the desired gain and phase margins, see [14]. The cutoff frequency ω_c of the closed loop can then be estimated based on the transfer function of the closed loop. The estimated parameters of a real system might deviate from the estimation by up to 100%. Hence, to ensure that the resulting excitation signal provides the proper impact on the dynamics a set of frequencies should be used.

Choosing the frequencies can be done based on the strategy presented in (F.24) and (F.23)

$$\omega_{\frac{m+1}{2}} = \omega_c \tag{F.23}$$

where m is chosen to be an odd number and hence the frequencies will be spread using (F.24).

$$\omega_i = \frac{1}{10}\omega_{i+1} \qquad \forall i = 1, \cdots, m-1 \tag{F.24}$$

After the frequency content has been chosen the amplitude of the signal should be determined. The method for choosing the amplitude for the excitation signal will be described hereafter. Since the choice of the frequency content relies on the closed loop transfer function the assumption is that the current controller is able to stabilise the closed loop and the choice of amplitude can therefore be done by using the following method. The boundaries for a controllable signal is usually provided as constraints on the set-point which can be denoted by SP_{upper} and SP_{lower} and can therefore be used in the following way:

$$SetPoint + ||\zeta||_\infty < SP_{upper} \tag{F.25}$$

$$SetPoint - ||\zeta||_\infty > SP_{lower} \tag{F.26}$$

Extrema for ζ does exists due to the definition in (F.21) and (F.22), which defines ζ as a bounded function. Furthermore by assuming that the closed loop has reached steady state the contribution from the P-term of the PI controller can be neglected and the I-term is the only contributer that enables the controller to keep set point tracking. Based on the discussion above, it is proposed to choose $||\zeta||_\infty$ using the following equation:

$$||\zeta||_\infty = \alpha \left| \frac{\textit{I-term}}{\hat{k}_{p_k}} \right|, \tag{F.27}$$

In (F.27) $\alpha \in \{0.01 \cdots 0.25\}$ and $\hat{k}_{p_k}$ is the estimated system gain. The value of α should be chosen as high as possible without violating (F.25), and (F.26). A reasonable choice is $\alpha = 0.15$. The correct amplitude for ζ is the sum of amplitudes of the sinusoidal function in (F.22) which are given by:

$$A_i = \alpha \left| \frac{I\text{-}term}{m\, \hat{k}_{p_k}} \right| \quad \forall k \in \{1, \cdots, m\}. \tag{F.28}$$

After the design of the excitation signal a proper method for optimisation with respect to the plant-wide performance has to be employed.

F.5.2 Invasive Weed Optimisation (IWO)

The use of traditional optimisation techniques as those presented in [7] is hard in complex industrial settings like the supermarket refrigeration system. The methods usually rely on the existence of a process model and secondly the methods are derivative-based. Meaning that they rely on calculating the derivative of the performance function. To avoid the problem of dealing with online calculation of the derivative of the performance measurement the aim has been to employ a derivative-free optimisation scheme. The invasive weed optimisation, which was introduces in [31], is a derivative-free optimisation scheme that in this paper has been used as a method for searching the parameter space.

The bio-inspired IWO algorithm basically tries to mimic the way an invasive weed colonises an area to find the best position for the weed. The idea proposed in this paper is to utilise the IWO algorithm to find the best parameters for the controllers with respect to the plant-wide performance measurement. The IWO algorithm searches the parameter space by randomly choosing an initial set up parameters in the predefined search space and then evaluating the performance of each parameter set. Then by applying the survival of the fittest principle only the best parameter sets are selected. In the next generation of the algorithm there is then randomly selected new parameter sets in a radius to the surviving parameter sets from the last generation. The radius is reduced after each generation of the algorithm and after a predefined number of generations a solution is provided. This process corresponds to an invasive weed that colonises a field with crops by randomly spreading seeds and then allowing them to grow into plants where the best plants will be able to spread their seeds until there is no more space between the crops.

F.5.3 Simulation setup and Results

This section presents the simulation setup results for the use of the IWO algorithm for plant-wide optimisation of the dynamic performance of the supermarket refrigeration system. The basic idea is to employ the approach online on a real system and the method is therefore tested on the simulation model. The simulation model used in the setup is described in section F.2. The IWO algorithm has been used to find the optimal parameter set for the PI controller in a display case which is the controller gain, k_c, and the integration time, T_i. The size of the search space has been chosen based on the knowledge about what reasonable ranges can be accepted for the different optimisation variables. Furthermore, to assist the IWO algorithm an initial guess has been used in the first generation of the algorithm to ensure that at least one parameter set is reasonable.

In this setup the simulation runs for 5000 seconds to ensure that the performance function is in steady state with the given set of controller parameters, k_c and T_i.

F.5.4 Results

The result produced by the IWO algorithm over 150 generation can be seen on Fig. F.10, where the minimum, the mean and the maximum values of the performance function is plotted for each generation each corresponding to a different parameter set. On Fig. F.11 the evolution of the worst and the best integration time T_i is plotted and it can be seen that even the worst performing converges toward the solution. The worst and the best controller gain k_c is plotted on Fig. F.12. In addition, it also shows that the worst performing choice of k_c is on the edge of the search space for many generations i.e $k_c = -0.001$. However, despite that even the worst performing choice of k_c converges toward the best solution. On Fig. F.13 the integration time T_i is plotted versus the controller gain k_c and it shows that all the best performing parameter sets from all the generation are within a narrow area of the solution. Another important point is that the initial guess of the controller parameter, $k_c = -0.1$ and $T_i = 100$, has never been the best performing parameter set. The initial parameter set is chosen based on empirical knowledge about the sýstem. However, the solution provided by using the IWO algorithm is a controller gain of $k_c = -0.1716$ and an integration time of $T_i = 55.84$.

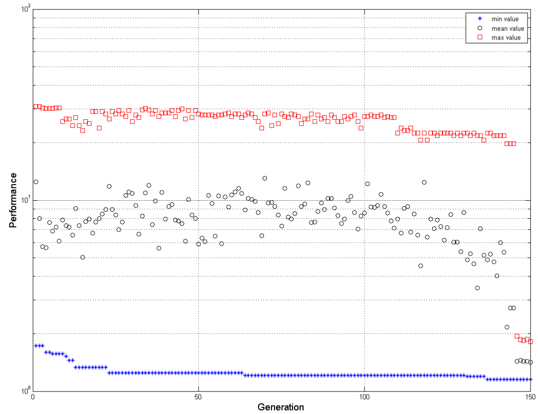

Figure F.10: Generations versus the performance function J. The best achived performance over 150 generations is 1.1512

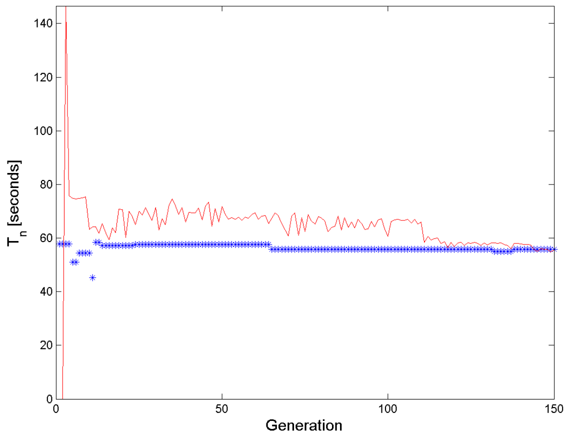

Figure F.11: Evolution of T_i. The red line plots the evolution of the worst choice of T_i and the blue line plots the evolution of the best choice of T_i

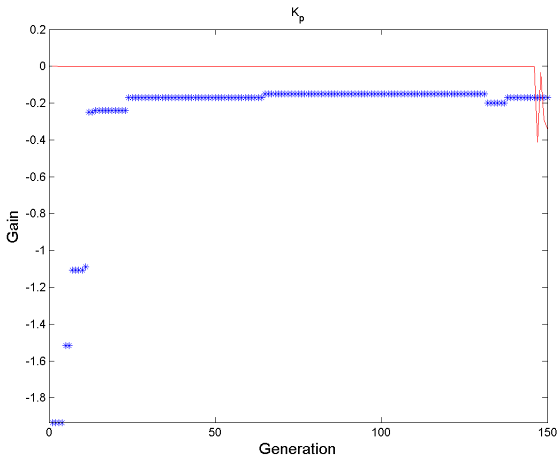

Figure F.12: Evolution of k_c. The red line plots the evolution of the worst choice of k_c and the blue line plots the evolution of the best choice of k_c

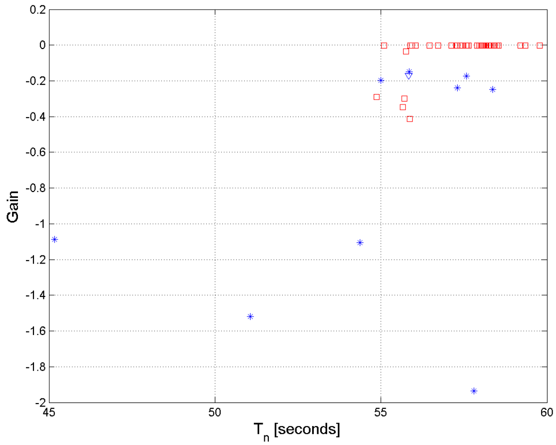

Figure F.13: T_i versus k_c, The blue triangle indicates the end result for the parameters, $k_c = -0.1716$ and $T_i = 55.84$.

F.6 Conclusion

The focus of this paper was on achieving optimised system performance from a plant-wide point of view. To enable performance assessment and optimisation an appropriate performance function, which encompasses food quality, energy efficiency, and system reliability, was introduced. Due to the fact that the supermarket refrigeration systems operate the majority of time under steady-state (i.e. static) conditions, it was appropriate to consider the performance optimisation case under two different conditions; static (steady state) conditions and dynamic (transient) conditions. The static performance optimization was realised through set-point optimisation. The simulation results on a supermarket refrigeration system with a compressor rack, consisting of two on-off compressors with different capacities, lead to a strategy for choosing set-point that is contrary to the existing practice. A generalisation of the method to a compressor rack with three compressors was also provided. As the system operates in steady state conditions most of the time, it is necessary to generate auxiliary signals in order to sufficiently excite the local subsystems to enable parameter optimization of their corresponding controllers. The paper suggested a design strategy for these signals that also takes the realistic operational conditions into considerations. A derivative-free optimisation strategy, based on invasive weed optimisation method, was employed to search for optimal parameters of the local controllers. Simulation results were used to discuss and propose a strategy for appropriate employment of the optimisation method.

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