THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING

Integrated Control of Heat Pumps

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Abstract

The purpose of the work presented in this thesis is to investigate the potential and prerequisites for improved energy efficiency of heat pump systems using efficient components and proper methods for optimisation of operation. The background is that several investigations have reported on possibilities for increased energy efficiency of heat pumps if variable-speed capacity control is used instead of conventional on/off control. Further improvements are possible if variable speed pumps or fans and electronically controlled expansion valves are used as well. To take full advantage of these new controls they should be properly co-ordinated and optimised.

In this work, a measuring system consisting of three temperature sensors and two pressure sensors was used for on-line measurement of the coefficient of performance in an experimental investigation. Knowledge of the performance provides a basis for self-optimising closed-loop control. The measurement system can also be used for Fault Detection and Diagnosis (FDD) and Performance Indication and thus contribute to an integrated control system. The Nelder-Mead simplex method was applied for on-line optimisation. The method was evaluated by computer simulations and shows promising results. It was possible to find the optimal set-point, maximising COP, and the method could also handle soft constraints, such as a desired set-point for the heating capacity.

The potential of variable-speed capacity control was investigated both by laboratory tests and annual energy savings calculations. The laboratory tests were made on a brine-to-water heat pump with a variable-speed piston compressor and an electronic expansion valve. The results from the tests showed a decrease in efficiency when applying variable-speed capacity control to the compressor when compared to conventional on/off control. Analyses indicate that this is due to losses in the frequency converter and increased losses within the compressor, most likely in the electric motor. The electronic valve gave only a marginal increase in efficiency compared to a conventional thermostatic expansion valve. Using these test results as a basis, the possible annual energy savings were calculated. This analysis indicates that there exists an energy saving potential also for variable-speed controlled brine-to-water heat pumps but it requires product development mainly of the compressors. A conservative estimate indicates savings of 10 % compared to on/off control.

Keywords: variable-speed compressor, electronic expansion valve, optimisation, energy efficiency, heat pump, superheat, performance indication, control

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Preface

The work for this thesis has been carried out at SP Swedish National Testing and Research Institute, Department of Energy Technology.

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Borås March 2003

Fredrik Karlsson

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Designations

Latin letters

COP	*Coefficient of performance*
COP_1	Coefficient of performance; heating mode
COP_{hp}	Heat pump coefficient of performance; heating mode
COP_{hps}	Heat pump system coefficient of performance
COP_2	Coefficient of performance; cooling mode
COP_C	Coefficient of performance; Carnot cycle
c_p	*Specific heat; kJ/kg-K or J/kg-K*
c_{pb}	Specific heat of brine
c_{pw}	Specific heat of water
D, d	*Diameter; m or mm*
D	Outside diameter of tubes, diameter of pump wheel
d	Inside diameter of tubes
f	Thermal loss factor of compressor
f	*Frequency; Hz*
f_{comp}	Supply frequency to the compressor motor
f_{bp}	Supply frequency to the brine pump
f_{wp}	Supply frequency to the heating water pump
F	*Heat loss factor; W/K*
g	*Objective function*
h	*Specific enthalpy; J/kg or kJ/kg*
h_1	Specific enthalpy of saturated refrigerant vapour in the condenser
h_2	Specific enthalpy of the saturated refrigerant vapour in the evaporator
h_3	Specific enthalpy of the superheated refrigerant vapour in the compressor outlet
h_4	Specific enthalpy of the superheated refrigerant vapour in the compressor inlet
h_5	Specific enthalpy of the subcooled liquid refrigerant after the condenser
h_6	Specific enthalpy of the refrigerant after the expansion device
h_7	Specific enthalpy of the saturated refrigerant liquid in the condenser
h_{ml}	Intermediate result which is used in the calculations for superheated vapour (Appendix B)
h_{m2}	Intermediate result in the calculations of the superheated vapour
h_{sup}	Specific enthalpy of superheated vapour
I	*Electric current; A*

$\dot{m}$	*Mass flow rate; kg/s or kg/h*
$\dot{m}_R$	Mass flow rate; refrigerant
$\dot{m}_b$	Mass flow rate; brine
$\dot{m}_w$	Mass flow rate; heating water
n	*Rotational speed; min^{-1}*
n_{comp}	Rotational speed of compressor
p	*Pressure; Pa, kPa or bar (pressure difference is designated Δp)*
p_1	Condensing pressure
p_2	Evaporating pressure
$\dot{Q}$	*Thermal capacity; W or kW*
$\dot{Q}_1$	Heating capacity, heating load
$\dot{Q}_2$	Cooling capacity, cooling load
$\dot{Q}_{hps}$	Heating capacity of the heat pump system
$\dot{Q}_{rad}$	Heating capacity of the radiator system
$\dot{Q}_{house}$	Heat demand of the house
R	*Relative operating time*
R_{hp}	Relative operating time of the heat pump
SPF	*Seasonal Performance Factor*
SPF_{hp}	Seasonal performance factor; heat pump (excluding pumps)
SPF_{hps}	Seasonal performance factor; heat pump system
SPF_{hs}	Seasonal performance factor; heating system (including supplementary heater)
t	*Temperature; °C*
t_1	Temperature; condensing temperature
t_2	Temperature; evaporating temperature
t_3	Temperature of superheated refrigerant vapour in the compressor outlet
t_4	Temperature of the superheated refrigerant vapour in the compressor inlet
t_5	Temperature of the subcooled liquid refrigerant after the condenser
t_6	Temperature of the refrigerant after the expansion device
t_7	Temperature of the saturated refrigerant liquid in the condenser
t_{bi}	Temperature; brine inlet to the heat pump evaporator
t_{bo}	Temperature; brine outlet from the heat pump evaporator
t_f	Supply temperature to the heating system
t_r	Return temperature from the heating system
t_{wi}	Temperature; heating water inlet to the heat pump condenser
t_{wo}	Temperature; heating water outlet from the heat pump condenser
$\bar{t}_w$	Mean temperature of the heating water
t_{out}	Outdoor air temperature

t_{in} Indoor air temperature
t_{mean} Annual mean outdoor temperature

U *Electric voltage; V*

$\dot{V}$ *Volume flow rate; m³/s or m³/h*
$\dot{V}_b$ Volume flow rate; brine (cold) side of a heat pump
$\dot{V}_w$ Volume flow rate; water (hot) side of a heat pump
$\dot{V}_{hs}$ Volume flow rate; heating system
$\dot{V}_{sw}$ Volume flow rate through the compressor

V *Volume; m^3*
V_{sw} Volume of compressor

v *Specific volume; m^3/kg*

$\dot{W}$ *Power (electric); W or kW*
$\dot{W}_{wp}$ Power input to the heating water pump
$\dot{W}_{bp}$ Power input to the brine pump
$\dot{W}_{comp}$ Power input to the compressor
$\dot{W}_{hps}$ Total power input to the heat pump system

W *Work (mechanical or electric); J or kWh*
W_{hs} Energy input to the heating system
W_{hps} Energy input to the heat pump system
W_{comp} Energy input to the compressor
W_{bp} Energy input to the brine pump
W_{wp} Energy input to the heating water pump
W_{inv} Energy input to the frequency converter
W_{heater} Energy input to the supplementary heater

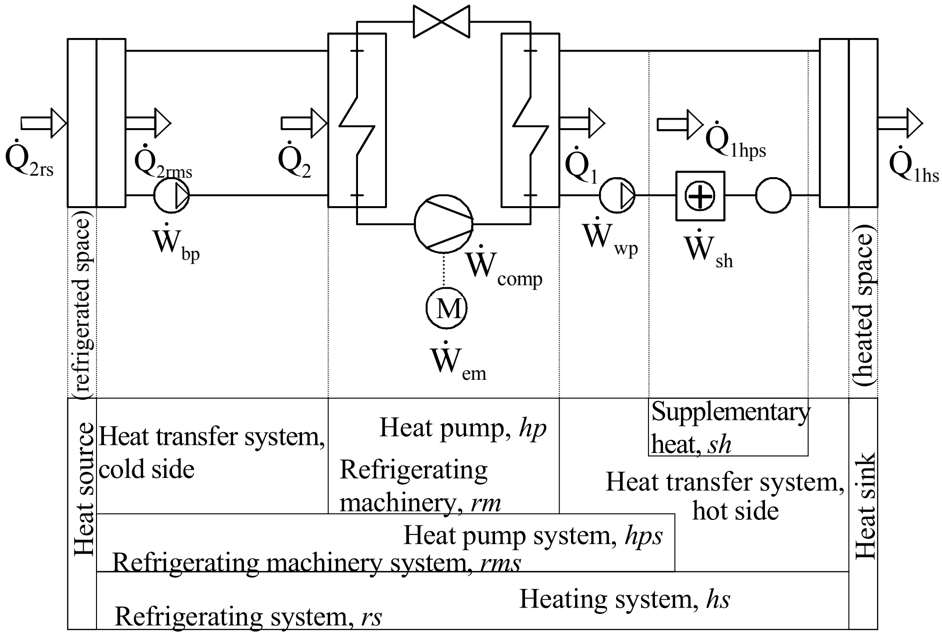
Greek letters

η	*Efficiency; -*
η_v	Volumetric efficiency
η_{is}	Isentropic efficiency
η_C	Carnot efficiency
η_{inv}	Efficiency of the frequency converter
η_{mc}	Mechanical efficiency of the compressor
$\eta_{m;el}$	Efficiency of the compressor motor
η_{mt}	Efficiency of the transmission between motor and compressor
φ	Evaporation efficiency
θ	*Temperature difference; K*
θ_c	Geometric mean temperature difference; condenser
θ_e	Geometric mean temperature difference; evaporator
θ_{rad}	Logarithmic mean temperature difference; radiator system

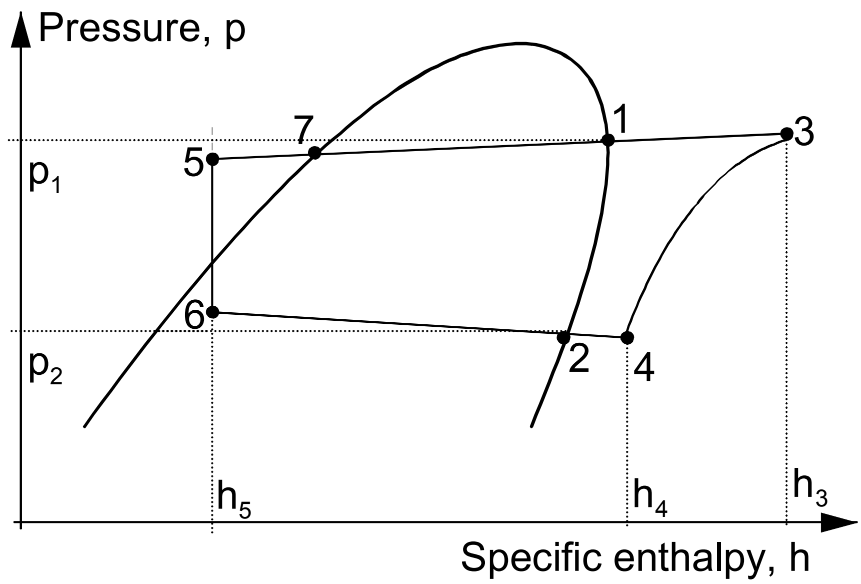
Subscripts and superscripts

hp	Heat pump
hps	Heat pump system
hs	Heating system
loss	Losses
pl	Part load
ref	Reference point
set	Set-point
ss	Steady-state

System boundaries



Designations for the points in the refrigeration cycle



1 Introduction

This introduction provides a background for the present work in terms of its relevance for a Swedish residential heating market. It also presents the purpose of the study, the methodology used and the structure of the thesis.

1.1 Background

In Sweden 22 TWh of electricity was used for heating purposes during 2001 (Svensk Energiförsörjning[66]). Of this, 94 % was produced in hydropower and nuclear power plants (Swedish Energy Agency[65]). Because of this energy mix, the electricity in Sweden has been cheap and widely used for heating purposes. However, according to the Swedish Energy Agency[65], the price of electricity for heating (excluding taxes) has increased from 0.32 SEK/kWh in 1990, to 0.43 SEK/kWh in 2001, i.e. an increase by 37 %. Furthermore there has been a substantial increase in taxation on electricity. Also, based on the decision taken in the referendum in 1980 the phase out of nuclear power from the Swedish energy system started in December 1999 by the shut down of one of the reactors in Barsebäck. This phase out will probably increase the price of electricity even further and thus raise incentives for new energy efficient techniques as well as increased energy efficiency of existing techniques.

In 1994 NUTEK (Swedish National Board for Technical Development) launched a Nordic heat pump competition with the aim of increasing their energy efficiency. This competition received quite a lot of attention and put a focus on the heat pump market again after the decrease in sales during the end of the 1980s. Since 1995 the number of sold heat pumps in Sweden has increased from 8000 units/year to > 40000 units/year in 2002, making Sweden the largest heat pump market in Europe. At the same time the need for cooling is increasing and thus refrigeration and heat pump systems will have an increased impact on the total energy use in the future. Today heat pump and refrigeration systems in Sweden use 12-15 TWh electricity annually (Fahlén[19]) and thus such a modest increase in efficiency as 10 % would save about 1.5 TWh. As a comparison, the total electricity production by wind power was 0.4 TWh during 2001, according to the Swedish Energy Agency[65]. With the increased use of heat pumps and refrigerating systems, and the need for reduced energy use, it is of vital interest to investigate what opportunities there are for increasing the energy efficiency of these systems.

Increased energy efficiency of a given supply system can be reached by using efficient components as well as by controlling the process such that it is operating at the most efficient set-point. Components showing particular promise are variable-speed compressors, pumps and fans and electronically controlled expansion valves. Investigations regarding variable-speed compressors have been reported by e.g. Poulsen[53] and Miller[45]. Jakobsen and Skovrup[37] as well as Andrade, Bullard, et al.[2] reports on variable-speed pumps and fans and investigations regarding electronically controlled expansion valves have been published by Tassou and Al-Nizari[70] and Aprea and Mastrullo[3]. Increases in energy efficiency have been reported in the range of 5-35 % depending on application and method of comparison. The dominating part of the energy saving potential relates to the compressor. Relatively few investigations have been presented on the issue of finding the optimal operation for the whole system for a certain load. Investigations on this subject have been presented by Cho and Norden[11], MacArthur and Grald[43], Braun, Klein, et al.[7] and Svensson[68]. Braun, Klein, et al.[7] and Svensson[68] present on-line model-based optimisation methods. Cho and Norden[11] used an optimisation method, without any underlying system model, on a large industrial refrigeration system.

Although several investigations have been made on the issue of optimisation of heat pump operation there is a need for investigations where a system perspective is applied in order to optimise the operation of the whole heat pump system.

1.1.1 Heat pump systems

Heat pump systems look very different depending on application. Heat pumps are used in district heating systems and in various industrial processes. However, this thesis deals with heat pump systems for domestic use. Although different in design the principles are the same. As illustrated in Figure 1.1, heat is extracted from a low-temperature heat source, Q_2. By means of mechanical work, W, the energy is lifted from a low a high temperature, which is useable for heating purposes. The efficiency of a heat pump is commonly expressed by the "Coefficient Of Performance", *COP*, defined as:

$$COP = \frac{Q_1}{W} \qquad \text{Eq. 1.1}$$

This number describes how much useful heat that is delivered in relation to the work added to the system.

For domestic systems the most common heat sources are outside air, exhaust-air and the ground. The heat sinks, Q_1, are room air, directly or indirectly via a water system, and sanitary water. In Sweden the most common heat pumps are ground-coupled brine-to-water heat pumps. They are often sold as a complete heating system including a tank for hot tap water and an electrical heater as back-up heat source.

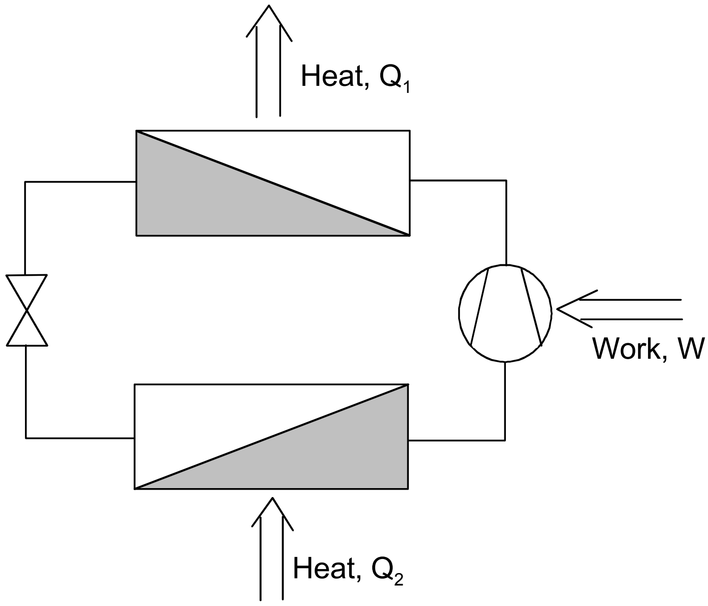


Figure 1.1 Schematic picture of a heat pump.

1.2 Purpose

The purpose of this work was to investigate how control, performance indication and fault detection and diagnosis of heat pumps and refrigerating equipments could be integrated in a common control system, Integrated Refrigeration Management (Fahlén[15]). This IRM concept requires a measurement system and the intention has been to use existing sensors as much as possible. Another intention with this work was to investigate the possibilities for increased energy efficiency by using new electronic controls, such as variable-speed compressors and electronically controlled expansion valves. This investigation was to be made both theoretically and by laboratory tests.

1.3 Methodology

The research project started with a literature survey, in which earlier investigations regarding the field of capacity control of heat pumps were analysed. With this survey as a base, an experimental heat pump was built and tested in a laboratory. The results from these tests were then used for calculations of the annual energy saving potential. Important conclusions could also be drawn regarding the efficiency of, and prerequisites for, the components in the heat pump. An on-line, direct search, optimisation method was investigated by computer simulations.

1.4 Structure of the thesis

Chapter 2 presents a state-of-the-art review on heat pump technology. This constitutes the basis for the continued work.

Chapter 3 describes the steady-state heat pump model used in chapters 5 and 6.

Chapter 4 contains a discussion on the issue of on-line optimisation and describes the optimisation method used in this work.

Chapter 5 outlines the optimisation method applied to heat pump operation. Computer simulations of this approach are presented and analysed.

Chapter 6 presents an analysis of the energy saving potential of capacity controlled heat pumps. This analysis was made both by laboratory tests and by calculations of the energy saving potential on an annual basis.

Chapter 7 provides an analysis of the possibilities for fault detection and diagnosis using the proposed measurement system. Also an analysis of the required accuracy of the intended sensors is provided.

Chapter 8, finally, summarises the major findings and concludes the thesis.

2 State-of-the-art review

This chapter presents a state-of-the-art review of the technology used in heat pumps. It intends to point out the current knowledge as well as the fields where further improvements are possible. Searching databases and contacting researchers working within the fields of interest have provided the information for this survey.

Compressors in heat pumps for domestic use are often of the single-speed type. A thermostatic expansion valve performs the evaporator superheat control and constant speed pumps or fans support the flows of the high and low temperature heat transfer media. This is a well functioning system that has been in use for many decades. Even so, the introduction of new controls and algorithms for system optimisation may increase the efficiency while maintaining the functionality. The introduction of variable-speed compressors implies possibilities for increased efficiency at part load operation, as well as reduced need for supplementary heat. This will be discussed in section 2.1.

The energy efficiency of a heat pump may be further increased if the evaporator superheat control is optimised. A thermostatic expansion valve cannot keep the superheat at the lowest possible value over a range of loads. It is tuned to give a stable operation at the evaporator design load and the superheat will thus often be unnecessarily large at lower loads (Huelle[32], Jakobs[35] and Jolly, Tso, et al.[38]). How this superheat control may be improved will be further discussed in section 2.2.

When changing the compressor capacity, and thus the refrigerant flow rate, the optimal heat transfer media flow rate will also be changed. The possibility of changing the speed of the auxiliary equipment is yet another possibility for increased energy efficiency. This will be further discussed in section 2.3.

The three factors mentioned here might be optimised one at a time. However, since they all affect each other, the best results will be reached if the actions are properly co-ordinated. Methods for doing this will be surveyed in section 2.4.

Another way of increasing the efficiency of heat pumps is to detect faults and decreasing performance at an early stage. By doing so, the lifetime of the unit may be increased and it may also operate at a higher efficiency. Such a system is called FDD (Fault Detection and Diagnosis). This FDD-system uses a number of sensors and one or several algorithms in order to detect whether a fault is present (detection) and what the cause for this is (diagnosis). FDD can be applied to all kinds of systems and many different strategies can be used. In section 2.5 investigations regarding FDD applied to vapour compression systems are presented.

2.1 Capacity control of compressors

Heat pumps, especially in domestic use, are often connected to heat loads that change over time. Various techniques can be used in order to adjust the heating capacity to the heat load. The most common techniques are, according to Qureshi and Tassou[55]: on/off control, hot gas bypass, evaporator temperature control, clearance volume control, multiple compressor control, cylinder unloading and variable-speed control. Several different investigations claim the variable-speed control to be the most energy efficient method (Holdack-Janssen and Kruse[31], Qureshi and Tassou[54], Pereira and Parise[50] and Wong and James[76, 77]). A frequency converter, often called ”inverter”, accomplishes the variable-speed control. The inverter controls the frequency supplied to the compressor motor in order to change the speed of the compressor. Changing the speed also changes the thermal capacity of the compressor. The frequency conversion can be accomplished by different methods, such as Pulse Width Modulation (PWM) and Pulse Amplitude Modulation (PAM). This topic is further discussed by e.g. Garstang[27, 28], Anderson[1] and Bose[6].

2.1.1 Improvement of performance using variable-speed compressors

The dominating control strategy for domestic heat pumps is the on/off control and thus it is interesting to compare this with the continuous variable-speed control. Miller[45] and Garstang[27] have divided the reasons why variable-speed control has the potential of being more energy efficient than the on/off control into the following categories:

- Better performance at part load
- Fewer on/off cycles
- Reduced need for supplementary heat
- Reduced need for defrosting

<u>Better performance at part load</u>: When operating at part load a fixed-speed compressor is switched on and off in order to adjust the heating capacity to the present heat load. The on and off switching causes the heat pump to work between two temperature levels as illustrated in Figure 2.1. This also causes the heat pump to work at a higher condensation temperature and a lower evaporation temperature than it would do if the compressor speed could be adjusted to meet the heat load, i.e. trace the temperature set-point.

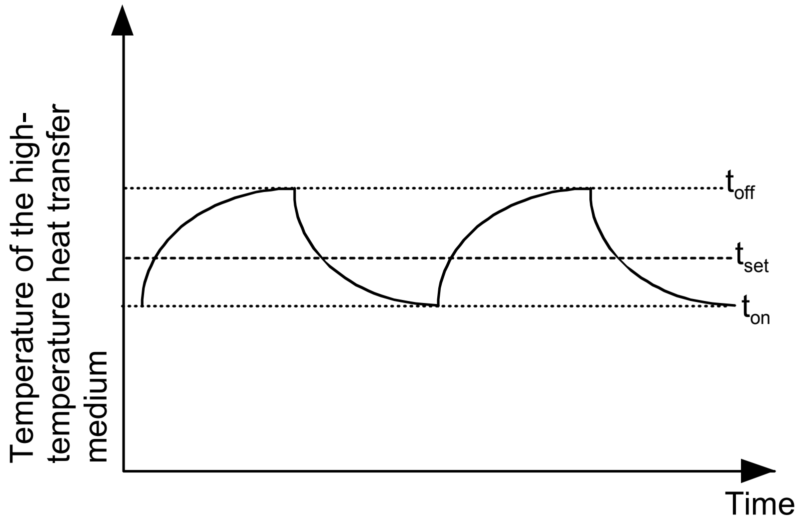


Figure 2.1 The normal way of controlling a heat pump is to switch it on and off between two limits (t_{on} and t_{off}) in order to keep the desired mean temperature (t_{set}).

Fewer cycles on/off: The possibility for adjusting the heating capacity to the heat load will decrease the number of on and off cycles. Cycling the compressor on and off causes losses and wear. Hence a reduced cycling frequency will lead to reduced losses and a longer life expectancy. How large this benefit will be depends on the frequency span within which the compressor can operate and the range of load variations.

Reduced need for supplementary heat: If the heat pump is not sized to cover the maximum heat load a supplementary heat source must be installed in the heating system. This is often the case in Sweden were heat pumps for domestic use are sized to cover about 50-60 % of the maximum heat load. This will cover 85-90 % of the required annual heating demand. The possibility of widening the operating range of the compressor by variable-speed control implies a possibility to reduce or completely remove the need for supplementary heat. The benefit of this will of course depend on the operating range of the compressor and on the heat pump power coverage.

Reduced need for defrosting: The evaporating temperature increases when the compressor speed is reduced due to the reduced specific heat load of the heat exchanger. For air-source heat pumps this will lead to less frost formation and thus a reduced need for defrosting, and as a consequence the efficiency will increase.

2.1.2 The energy saving potential of variable-speed compressors

The reasons why variable-speed control can be more energy efficient than on/off control are listed above. But how much more efficient will a variable-speed heat pump be? The answer will depend on the actual application and on how the comparison is made. If the heat pump is connected to a constant heat load there will be no benefit of installing a compressor with variable-speed control. Instead it will probably cause the efficiency to decrease since the power used by the heat pump will increase due to the extra power used by the frequency converter. The result from a comparison will differ if the comparison is made for one or two operating points or if the annual energy saving potential is compared. Several different ways of comparing variable-speed and constant-speed compressors were found in the literature. Some investigations using different methods for comparison are described below.

Poulsen[52] used a rotary piston compressor in a brine-to-water heat pump and evaluated its performance at part load by laboratory tests. Testing was made both with on/off control and with variable-speed control. The heat pump was operated for two hours during each test. The tests were performed such that the same amount of heat energy was rejected at the same mean temperature during these two hours for both control methods. This means that the average heating capacity was equal for the two control methods. The thermostat was adjusted such that the heat pump performed about four on/off cycles per hour when operating in on/off control. An increase in *COP* by 13-31 % was recorded with the variable-speed control depending on the operating point. One scroll and one reciprocating compressor were also tested in the same way by Poulsen[51]. An increase in efficiency by about 10 % was recorded for both compressors.

Another investigation by the same author (Poulsen[53]) describes laboratory tests made on a brine-to-water heat pump with an inverter-controlled semi hermetic two-cylinder compressor and an electronic expansion valve. The compressor frequency could be adjusted between 15-90 Hz. The performance for a year was simulated by steady-state tests. A mean temperature both for the incoming brine temperature and the outgoing heating water temperature represented each month. The heat pump was then tested at these temperature levels and a seasonal performance factor was calculated based on these results. A seasonal performance factor of 4 for a low-temperature heating system was obtained (including circulation pumps and heating of sanitary water) and this is considered to be 18-25 % better than for conventional heat pumps available on the market.

Tests on an air-to-air split-system heat pump were made by Miller[45], both in heating and cooling mode, for different compressor speeds, air temperatures and humidities. Cycling tests at different compressor speeds were also made in order to quantify the cycling losses. Based on these test results, algorithms for the frosting/defrosting and cycling performance degradation were incorporated in a computer program, which calculated the annual energy demand for a typical residential house. The energy demand for three different cities in USA was calculated in order to determine the influence of the climate on the energy saving potential. The results show an increased *APF* (Annnual Performance Factor, corresponds to SPF_{hs}, i.e. the heating system Seasonal Performance Factor) of 15-36 % for the variable-speed heat pump when compared to an optimally controlled single-speed heat pump. The improvements are due to the following reasons:

- Modulating blower and compressor speed reduced the energy use by approximately 15-20 %
- Losses due to frosting and defrosting were reduced by 50 %
- Losses due to cycling were reduced by 50 %

It was also shown that oversizing the heat pump was beneficial. An oversize of 2.25 times the normal design showed an improvement of *APF* by 4 %.

Other investigations regarding these topics have been presented by Landé[42], Marquand, Tassou, et al.[44], Tassou, Marquand, et al.[72] and Bergman[5].

2.1.3 Comparison of dynamic performance between fixed-speed and variable-speed compressors

As mentioned before, the variable-speed control may reduce the losses due to on/off cycling just by simply making these cycles fewer. Compressors available today have a very small operating range regarding speed and thus they will operate in a cyclic manner during many of their operating hours. Hence it is interesting to see the possible differences between fixed-speed and variable-speed compressors during start-up. Tassou[69] investigated the dynamic response of an air-to-water heat pump with a variable-speed controlled reciprocating compressor and a shell-and-tube condenser. Start-up tests were performed for different compressor speeds and condenser water inlet temperatures. The author claims that changing the compressor speed or the condenser water inlet temperature will have a very small influence on the start-up losses. Another statement made is that an expansion valve that can adjust its capacity to the compressor capacity probably will increase the performance of the heat pump. This will be further discussed in section 2.2.

In another investigation made by the same author (Tassou and Qureshi[73]), a water-to-water chiller was tested with the compressor motor connected both directly to the power supply and to an inverter. The performance during start-up was analysed and compared for the two different set-ups. The peak voltage was higher when connected to the inverter compared to the direct mains connection. According to Tassou and Qureshi[73] this may lead to an increased motor winding temperature and thus a shorter motor life. Due to the slower acceleration time of the compressor when connected to the inverter, the increase in condensing pressure and drop in evaporating pressure are slower than when connected directly to the mains. Losses during start-up were analysed in periods of one minute. The result shows that the losses are highest during the first minute and then they are decreasing. After 5 minutes the operation has reached steady-state and thus the losses are zero. Start-up losses proved to be higher with the compressor connected to the inverter than if connected to the mains, 4.8 % compared to 3.9 %. This was claimed to be dependent on the soft-start used by the inverter. The compressor on-time was set to 15 minutes.

2.2 Evaporator superheat control

In heat pumps a thermostatic expansion valve is normally controlling the refrigerant flow rate out of the evaporator such that the refrigerant is superheated and thus will not cause liquid hammering and damage the compressor. The thermostatic expansion valve is a well-known component, which is widely used and performs well. However there is a limitation to its performance due to its fairly narrow operating range and hence there is a potential for increasing the performance of the superheat control both in steady-state operation and in dynamic operation of the heat pump. This is especially valid when a variable-speed controlled compressor is used, since the operating range will be widened compared to a single-speed unit. In the following sections this will be discussed, starting with the potential for increased energy efficiency during steady-state operation.

2.2.1 Energy saving potential

Both experimental (Huelle[32]) and theoretical (van der Meer and Touber[75]) analyses have shown that when the superheat is kept at the minimum value that still gives a stable control (MSS – Minimum Stable Signal, Huelle[32]) the highest energy efficiency is obtained. Lowering the superheat below the MSS will result in an unstable control and the *COP* will decrease as well, due to the unevaporated refrigerant leaving the evaporator (van der Meer and Touber[75]). In Figure 2.2 below, the relationship between cooling load and evaporator superheat is shown. To the left of the MSS-line the superheat control is unstable and the system will start "hunting" (Tassou and Al-Nizari[71]), i.e. the superheat and thus the capacity will start to oscillate. To the right of the MSS-line the superheat control is stable, but increasing superheat will lead to a decrease in *COP*. Thus the energy optimal control is to keep the superheat at the MSS-line for all cooling loads. For a thermostatic expansion valve, the static superheat is tuned such that the superheat control is stable at the maximum cooling load of the evaporator (Jolly, Tso, et al.[38]) c.f. Figure 2.2. Since the thermostatic expansion valve can only provide proportional action, the superheat will be on the stable side of the MSS-line but will be unnecessarily large. The distance between the characteristic line of the valve and the MSS-line illustrates the potential for increased energy efficiency of heat pumps by optimisation of the superheat control.

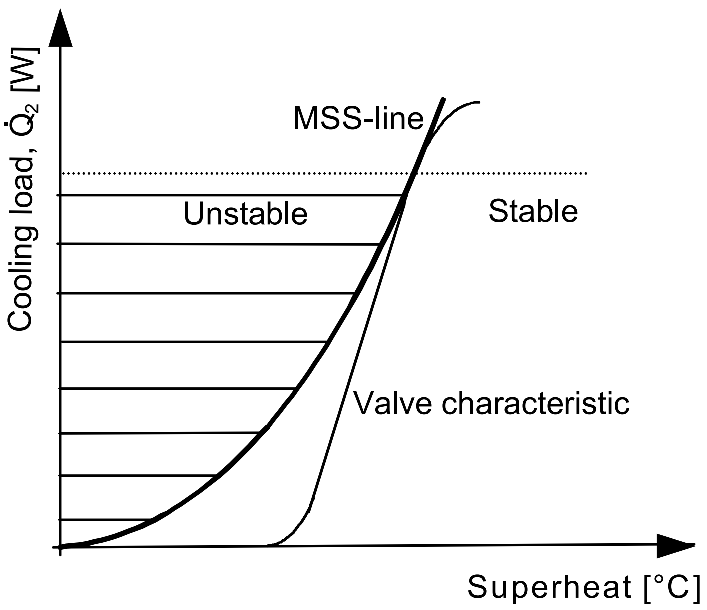


Figure 2.2 The MSS-line illustrates the limit for stable superheat control.

Using electronic expansion valves makes it possible to introduce control strategies for optimising the superheat control. Different approaches have been suggested and compared by Jolly, Tso, et al.[38], Chia, Tso, et al.[10], Outtagarts, Haberschill, et al.[49], Finn and Doyle[25] and Jakobs[35] but differences in control methods will not be further analysed here.

Instead the possibilities for increased energy efficiency when using an electronic expansion valve instead of a thermostatic expansion valve will be further discussed. Tassou and Al-Nizari[70] and Aprea and Mastrullo[3] have compared the performance of a thermostatic valve and an electronic valve at steady-state operation. In both investigations the electronic valve was set to keep the superheat at a constant value, i.e. it does not follow the MSS-line. The results show no difference in energy efficiency between the two valves. Jolly, Tso, et al.[38] tested a thermostatic valve and an electronic valve, both installed in a commercial unit for container refrigeration. The performance of the valves was compared during an evaporator pull-down (i.e. the refrigerating machine cools down the container from room temperature to the desired freezing temperature). The electronic valve was tested with two different control strategies; constant superheat (5 K) and an adaptive strategy intended to follow the MSS-line. The results show that the adaptive strategy increases the COP by 7.5 % when compared to the constant superheat strategy and by 20 % compared to the thermostatic valve. Results also show that the time for reaching the desired box temperature is significantly shortened when using the electronic valve with the adaptive control strategy.

These aforementioned investigations compare the different kinds of valves during steady-state operation or during fairly slow changes in ambient conditions. In the next section, investigations regarding the performance of valves during transients, such as compressor start-up, will be discussed. This is interesting to investigate because of the still dominating on/off control of compressors and because of the very small operating range of current variable-speed compressors.

2.2.2 Dynamic performance

Tassou and Al-Nizari[71] and Aprea and Mastrullo[3] present investigations where the performance of a thermostatic and an electronic expansion valve during transient operation were analysed. Both investigations used electronic valves with a fixed superheat set-point. The performance was analysed both during hot-start and cold-start. Cold-start means that the machine has been out of service long enough for the pressures and temperatures within the system to equalise. Hot-start means that the machine has been off for a short time and that the pressure difference between the high and low-pressure side still remains. The results from the investigations are quite similar. With the electronic valve the superheat stabilises faster and the amplitude of the oscillations is smaller, during both hot- and cold-start. This is also found in the investigation by Outtagarts, Haberschill, et al.[49]. Tassou and Al-Nizari[71] quantified the losses during on and off cycling, i.e. hot-start. To quantify the losses, the performance during start-up has been compared to the performance during steady-state. For a compressor on-time of 300 seconds the decrease in COP was 3.7 % for the electronic valve and 2.3 % for the thermostatic valve. The explanation for the smaller decrease of the thermostatic valve is that it opens faster than the electronic valve at start-up. The losses are dependent on the compressor on-time and become less important with increasing compressor on-time. For a compressor on-time of 900 seconds the losses are 1.1 % for the electronic valve and 0.7 % for the thermostatic valve.

2.3 Variable-speed pumps and fans

Pumps and/or fans are auxiliary equipment used in heat pumps in order to circulate the high and low-temperature media flows through the condenser and evaporator. They mostly operate at single-speed and their drive power thus remains fairly constant. The power used by this equipment is usually much smaller than the power used by the compressor, but when the heat pump operates at part load, the auxiliary equipment will constitute an increasing part of the total energy used by the heat pump unit. If a heat pump with a compressor of 2 kW drive power is operated at 10 % part load, its mean power usage is 200 W. This is about the same as the power used by the circulation pumps in a typical brine-to-water heat pump and thus the pumps will decrease the *COP* by 50 %. Fahlén[21] presents an investigation commenting on this issue.

Another issue regarding variable-speed pumps and fans is that increasing the flow of the heat transfer medium over the evaporator will lead to a higher mean temperature of the brine and thus a higher evaporation temperature. This may lead to an increased *COP* but on the other hand this increased flow will lead to an increased drive power of the auxiliary equipment. The same fact is valid for the condenser, where the increased flow rate will lower the condensation temperature and thus increase the *COP* but the increased flow leads to increased drive power to the auxiliary equipment. It has been shown that there exists an optimal combination of compressor speed and speed of the auxiliary equipment which will maximise the total *COP* of the heat pump unit (Andrade, Bullard, et al.[2] and

Jakobsen and Skovrup[37]), see Figure 2.3. The investigation made by Jakobsen and Skovrup[37] deals with an industrial refrigeration plant but the approach is valid also for heat pumps.

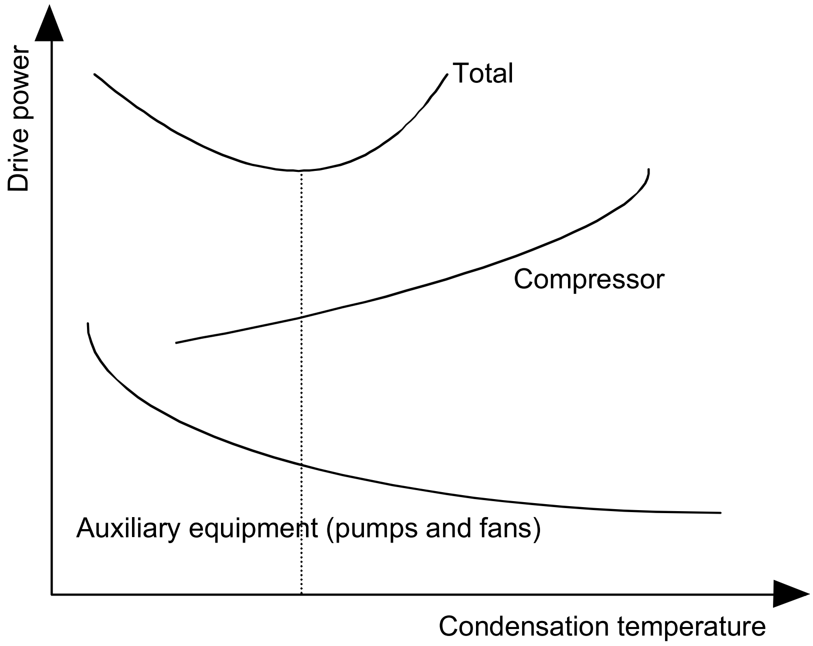


Figure 2.3 It is possible to find a combination of compressor speed and fan or pump speed such that the total drive power is minimised while the required heating capacity is maintained (Jakobsen and Skovrup[37]).

2.4 Cycle optimisation

As shown in the preceding sections, there are a number of actions that can be taken in order to increase the energy efficiency of vapour compression cycles. However, how to combine the different controls is not obvious since they all interact with each other. This section will present investigations, which intend to optimise the overall performance of vapour compression cycles. These can be divided into two categories. One where the optimal set-point for the different controllers is searched for in order to optimise the whole process. The other group tries to optimise the controlling action in order to reach the desired set-point as efficiently as possible.

2.4.1 Optimal set-point

Investigations addressing the issue of finding the optimal set-point for a vapour compression cycle are presented below.

On-line optimisation of a large chiller plant was conducted by Cho and Norden[11]. Nelder-Mead's Simplex method (Nelder and Mead[46]) was used to perform the optimisation. The optimisation criterion was to minimise the total drive power with the supplementary condition that certain process parameters were kept within their allowed operating range. The parameter constraints were handled by penalty functions. The penalty was increased by the square of the deviation. The drive power was not measured but was calculated from other measurements within the chiller plant. Apart from this, the optimisation technique does not need a model of the plant. Controlled variables were chilled water flow rate and temperature differences. When compared to the situation before the optimisation system was installed, the annual drive energy decreased by 12.5 %.

Another optimisation method, to be used in a chiller plant, was presented by Braun, Klein, et al.[7]. The objective was also in this case to minimise the drive power while fulfilling the cooling demand and keeping certain process parameters within their constraints. The strategy was to express the drive power of the chiller as a quadratic function of the cooling load and the temperature difference between the outgoing condenser heat transfer media and the outgoing evaporator heat transfer media. The drive powers of the auxiliary equipment, such as fans and pumps, were expressed as quadratic functions of their control variables. In this way the total drive power may be expressed by quadratic functions for which the optimum is easy to find. This requires detailed models of the individual components in the system. The characteristics of the process may also change over time and thus a strategy for updating model parameters may be required. The authors also suggest a "near-optimal" optimisation method where the total drive power to the chiller plant is expressed as a quadratic function of the uncontrolled variables. In this way no detailed models of the individual components are needed. Both methodologies were applied to a chiller system and the results show a very small difference between the optimal and near-optimal method over a wide range of loads and ambient wet-bulb temperatures. Both methods are claimed to be suitable for on-line optimisation.

MacArthur and Grald[43] reports on an investigation where optimal control theory was used for maximising *COP* while at the same time fulfilling a comfort index (Fanger's PMV-index, Fanger[23]). The controlled variables were the compressor speed, indoor-air flow rate and evaporator superheat. It was not intended to use a mathematical model in order to determine the optimal setting of the controlled variables. Instead the authors suggest a system identification strategy. The suggested approach was claimed to be generic, i.e. no assumptions were made about the heat pump or the environmental conditions. Performance of the method was analysed by computer simulations.

Svensson[67, 68] presents another investigation dealing with on-line optimisation of heat pump operation. Controlled variables were the condenser water outlet temperature and the water flow rates through the condenser and evaporator. The objective for the optimisation algorithm was to find the optimal set-point for these three variables such that the drive power was minimised while the heat load was satisfied. The algorithm should also handle constraints to component characteristics, such as compressor speed. The approach taken was to use a model of the heat pump and to update the parameters within this model. The parameters were updated from on-line measurements by the use of a moving horizon approach. This means that the parameters were updated using data stored during, for example, the last five minutes. Set-points for the different controllers were determined from the model. This parameter update algorithm is subjected to a steady-state criterion, i.e. the heat pump must be in steady-state operation in order for the optimisation algorithm to develop. The method was applied to a water-to-water heat pump and evaluated by laboratory tests. Two different cases were tested and for both of them the optimal set-point was found after two optimisation cycles. The computational time required finding the solution to the problem varied between 10 to 25 seconds. The results were presented as exergy losses for the different components of the heat pump.

2.4.2 Efficient Control

Having found the optimal set-point for the heat pump process, some kind of control system must perform the controlling action in order to get the system to operate at the prescribed set-point. This can be made in several different ways and the easiest way is probably to have local controllers performing the control task for each of the control loops. Then the compressor speed, pump or fan speeds and expansion valve opening have their own controllers and thus constitute local control loops with no direct interaction between each other. This kind of system with local loops and a superior coordination and optimisation level is suggested by Jakobsen, Rasmussen, et al.[36]. The controllers may then be of different kinds such as PID-controllers or fuzzy logic based controllers.

However, as shown by He, Liu, et al.[30], there exist cross-couplings between different sub-cycles within the vapour compression cycle. How the expansion valve performs will affect the compressor performance and vice versa. In order to make the cycle control faster and more efficient this should be taken into consideration when designing the controller. This was made by He, Liu, et al.[30] who proposes a model-based MIMO-controller (Multi Input Multi Output) in order to control the expansion valve opening and the compressor speed within a vapour compression cycle. The controller proved to be faster and more robust than a traditional control system with two independent PID-control loops. The proposed system may be extended to handle other HVAC or refrigeration cycles.

Another MIMO control strategy for a vapour compression cycle including a variable-speed compressor and an electronic expansion valve uses a simple model to calculate a feed-forward signal (Fredsted and de Bernardi[26]). This signal is then sent to two separate PID-control loops which control the compressor speed and expansion valve opening.

2.5 Fault detection and diagnosis

When discussing optimisation of heat pump operation it is easy to focus just on how to find the optimal set-point. Another major issue regarding optimisation is methods for fault detection and diagnosis (FDD). This is a field that will find its direct use also for today’s heat pump technology. If a heat pump is malfunctioning it may not save as much energy as supposed or its lifetime will be shortened. Either way the buyer will face increased costs for heating. By implementing functions for FDD, malfunctions may be detected at an early stage and proper action may be taken. By doing so, the lifetime of the heat pump may be prolonged or the efficiency kept at a high level. FDD has been applied in many different technological fields and the interested reader may have a look in the book “Fault Detection and Diagnosis in Engineering Systems” by Gertler[29]. A major work in the field of FDD in HVAC applications was conducted within the IEA Annex 25, "Real Time Simulation for Building Optimisation, Fault Detection and Diagnosis" (Hyvärinen and Kärki[34] and Hyvärinen[33]). Various approaches to FDD were investigated and applied in different kinds of HVAC applications. Typical faults within HVAC equipment were documented and described. However, the remaining part of this section will only deal with investigations where methods for FDD have been applied to vapour compression cycles.

A system for FDD includes the two parts fault detection and fault diagnosis. The first thing is to be able to detect whether the process is behaving normally or not, i.e. whether a fault is present or not. The next step is to determine which fault that is present, i.e. diagnosis. A third step may be to determine how severe the fault is, whether service is needed or not. Rossi and Braun[60] describe these steps in their investigation from which Figure 2.4 is taken.

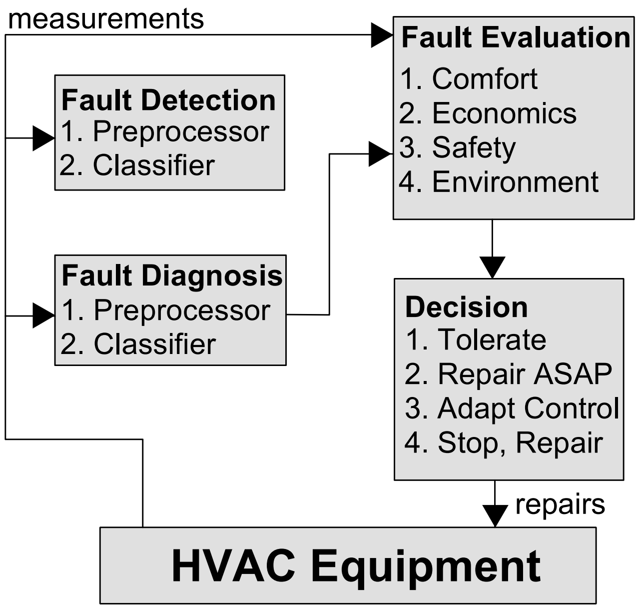


Figure 2.4 The different steps in fault detection and diagnosis (Rossi and Braun[60]).

Figure 2.5 shows the approach used for FDD by Rossi and Braun[60]. In order to detect a fault the normal operation must be defined and in this case a steady-state model defined it. A steady-state model was claimed to be sufficient since the faults to be detected develop slowly. The values of the inlet air temperature to the condenser and the inlet air temperature and humidity to the evaporator were fed to the model. Based on these inputs, the model calculated the expected values of the parameters used for the fault detection, in this case five temperatures and two temperature differences (they are listed to the right in Figure 2.5). These values were then compared with the actual values measured. The differences between the model and the measurements, i.e. the residuals were then used in a fault detection classifier. When a fault occurs the mean value and standard deviation of the residuals change. During normal operation the residuals occurring because of measurement noise and modelling errors should have a mean value of zero. The fault detection classifier identifies a fault when the difference between the measurements and the model is statistically significant. When the difference is considered to be significant was determined by a threshold value. Determining this threshold can be quite a delicate problem since it should be chosen such that the fault is detected at an early stage but at the same time the number of false alarms must be kept low. Repeated false alarms may after a while imply that alarms will be ignored.

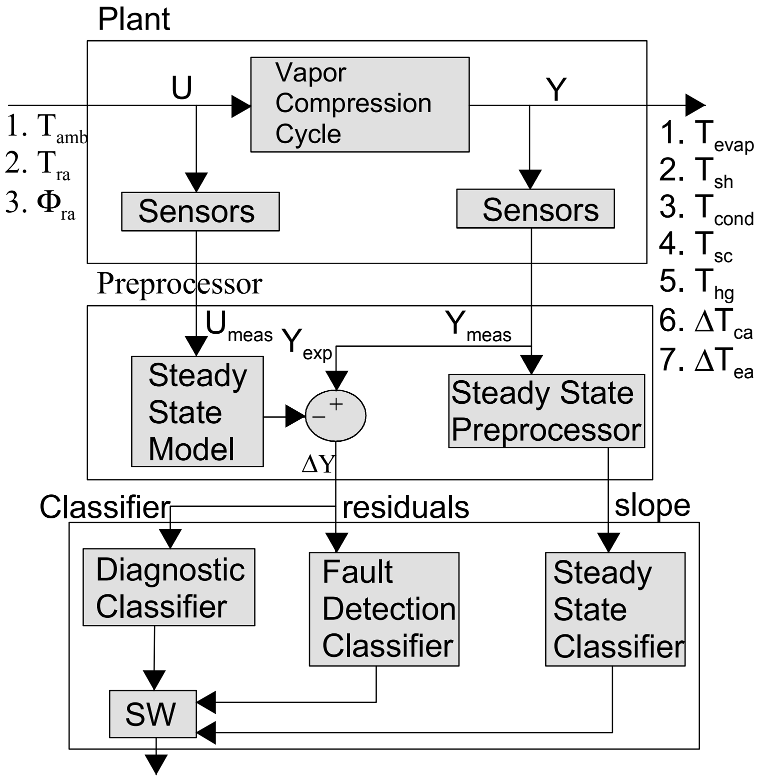


Figure 2.5 An example of a fault detection and diagnosis system (Rossi and Braun[60]; T_{amb} = ambient temperature, T_{ra} = return air temperature (inlet to evaporator), Φ_{ra} = return air relative humidity, T_{evap} = evaporating temperature, T_{sh} = superheat, T_{cond} = condensing temperature, T_{sc} = subcooling, T_{hg} = hot gas temperature, ΔT_{ca} = air temperature rise across condenser, ΔT_{ea} = air temperature drop across evaporator).

When a fault is detected, the diagnostic classifier is used for determining which fault has occurred. The classifier is rule-based and each fault has its own rule. The rules relate each fault to changes in the measurements, i.e. if the values of certain parameters are increasing or decreasing. As an example the rule for refrigerant leakage is shown in Table 2.1. The same kind of rules have been set up for the additional four faults:

- Liquid line restriction
- Leaky compressor valves
- Fouled condenser coil
- Dirty evaporator filter

Table 2.1 The rule for the detection of refrigerant leakage (Rossi and Braun[60]). evap = evaporation, sh = superheat, cond = condensation, hg = hot gas, ca = condenser air, ea = evaporator air

Fault	T_{evap}	T_{sh}	T_{cond}	T_{sc}	T_{hg}	ΔT_{ca}	ΔT_{ea}
Refrigerant leakage	↓	↑	↓	↓	↑	↓	↓

These rules have been developed from laboratory tests on an air-conditioning unit but are claimed to be generic for all similar types of air-conditioners. Using this method it was possible to detect a refrigerant leakage of 5 % of the total charge. The above described method was further evaluated by Breuker and Braun[8]. Among other things they modified and improved the classifier for compressor valve leakage.

Instead of awaiting steady-state operation, the transient characteristics of the system can be used in order to detect faults. Raeber[56] used spectral analysis in order to detect faults during start-up. Start-up curves of fault indicators differ somewhat when different faults occur. These differences become more evident when the signal is transformed into the frequency domain:

$$Y(j\omega) = \int_{-\infty}^{+\infty} y(t) e^{jwt} dt \qquad \text{Eq. 2.1}$$

Here y is the measured signal which is transformed into the frequency domain. The resulting signal is then processed in order to distinguish between different faults. This method requires a training phase where the typical operation patterns of normal and different faulty operations are determined and stored in a database. This database is then used as a reference when using the fault detection system in real operation.

Of course one can use a number of different FDD techniques in order to determine the status of a vapour compression system. Stylianou and Nikanpour[64] use different techniques during different stages of the process in order to perform FDD. The FDD-system was divided into three parts. The first one dealt with faults that are possible to detect when the heat pump is not in operation, such as faults on sensors. The second part handled faults that are more apparent during transients. Finally, the third part dealt with FDD during steady-state operation.

Faults in vapour compression systems can be of the abrupt type that requires immediate repair. Other faults develop more slowly and will show as a degradation in performance. A typical fault of this kind is heat exchanger or filter fouling. This does not require immediate action to be taken but can be optimised such that the filters are changed or cleaned when necessary. This can be addressed as an optimisation problem where the performance degradation is weighed against the cost for maintenance. Investigations addressing this issue were published by Rossi and Braun[58, 59] where the optimal maintenance interval for an air-to-air system was determined. This was claimed to reduce the maintenance costs by 11 % when compared to annual service scheduling.

The investigations described above are just a small sample of all the investigations made in the field of FDD applied to vapour compression systems. The intention is to give some ideas of the benefits that may come out of such a system. Karlsson and Fahlén[39] presents a more extensive list of relevant publications.

2.6 Conclusions

Variable-speed operation of compressors in heat pumps seems promising. However, very few investigations on brine-to-water heat pumps with plate heat exchangers as evaporator and condenser have been published. This is the most common type of heat pump in Sweden today. Only three were found, written by the same author, Poulsen[51,52,53]. The case is the same for electronic expansion valves. Several investigations have been published but they do not involve plate heat exchangers. For the investigations made, the electronically controlled valves seem promising during transients but for steady-state operation the energy saving potential seems rather small. For the operation of auxiliary equipment, such as pumps and fans, there is almost nothing published regarding capacity control of these. Especially not for small heat pumps, typical of domestic use. Regarding the overall optimisation, most of the work seems to have been made on finding fast and reliable controllers. How to find the optimal set-point for the components in the system in order to optimise the total performance has not gained as much attention.

A lot of work has been made in the field of FDD. The economic benefits and possible energy savings of such a system are difficult to estimate. However, a well functioning FDD system may increase reliability and make repairs and maintenance easier and faster.

With this information at hand, the work presented in this thesis was divided into the following parts:

- Build and perform measurements on a brine-to-water heat pump with plate heat exchangers, a variable-speed compressor and an electronic expansion valve.
- Estimate the energy saving potential of variable-speed capacity controlled heat pumps.
- Suggest a suitable method for optimisation of the internal operation of a heat pump. The method should be general and possible to apply to different kinds of heat pumps.
- Investigate the possibilities for FDD using the IRM measurement system.
- Estimate the necessary accuracy of the sensors used in the IRM concept.

3 Steady-state heat pump model

Both for the energy saving potential calculations in chapter 6 and for the optimisation calculations in chapter 5 a model of a heat pump was needed. The first thing to decide then is whether it should be a steady-state or a dynamic model. For the energy saving calculations a steady-state model is sufficient to estimate the annual energy use. Also for evaluation of the optimisation strategy a steady-state model is sufficient since the dynamics of the heat pump components are much faster than the dynamics of the heating system.

3.1 Model

The model was divided into sections describing each of the components in a heat pump, i.e. condenser, evaporator, compressor, expansion valve and circulation pumps, c.f. Figure 4.1.

3.1.1 Compressor

The compressor power is expressed as:

$$\dot{W}_{comp} = \dot{m}_R \cdot (h_3 - h_4) \qquad \text{Eq. 3.1}$$

where the refrigerant mass flow is expressed from the volume swept by the compressor according to:

$$\dot{m}_R = \frac{\left(\frac{n_{comp}}{60}\right) \cdot V_{sw} \cdot \eta_v}{v_4} \qquad \text{Eq. 3.2}$$

The enthalpy of the compressed refrigerant is calculated from the definition of the isentropic efficiency:

$$h_3 = h_4 + \frac{h_{3;is} - h_4}{\eta_{is}} \qquad \text{Eq. 3.3}$$

The following equation connects the compressor and the condenser:

$$\dot{Q}_1 = \eta_C \cdot COP_C \cdot \dot{W}_{comp} \qquad \text{Eq. 3.4}$$

The values of the three efficiencies in the equations above depend on the operating point and on the compressor. For the compressor used in the heat pump described in Appendix A the efficiencies can be expressed as:

$$\eta_{is} = 0.532 - 0.0009 \cdot \left(\frac{p_1}{p_2}\right)^2 + 0.0075 \cdot f_{comp} - 0.000079 \cdot f_{comp}^2 \qquad \text{Eq. 3.5}$$

$$\eta_C = 0.4569 - 0.0208 \cdot \left(\frac{p_1}{p_2}\right) + 0.0076 \cdot f_{comp} - 0.000073 \cdot f_{comp}^2 \qquad \text{Eq. 3.6}$$

$$\eta_v = 0.7461 - 0.0705 \cdot \left(\frac{p_1}{p_2}\right) + 0.0149 \cdot f_{comp} - 0.000155 \cdot f_{comp}^2 \qquad \text{Eq. 3.7}$$

These expressions were developed by correlating them to test data using the least square method.

When the compressor is operating with variable-speed it is connected to a frequency converter (inverter). The efficiency of the converter used in this project has been measured and can be expressed as:

$$\eta_{inv} = 0.9162 + 0.001575 \cdot f_{comp} - 0.0000098 \cdot f_{comp}^2 \qquad \text{Eq. 3.8}$$

3.1.2 Condenser

Applying an energy balance to the condenser yields the following three equations:

$$\dot{Q}_1 = \dot{m}_R \cdot (h_3 - h_5) \qquad \text{Eq. 3.9}$$

$$\dot{Q}_1 = \dot{m}_w \cdot (t_{wo} - t_{wi}) \cdot c_{p_w} \qquad \text{Eq. 3.10}$$

$$\dot{Q}_1 = (U \cdot A)_c \cdot \theta_c \qquad \text{Eq. 3.11}$$

The logarithmic mean temperature has been approximated by the geometric mean temperature:

$$\theta_c = \sqrt{(t_5 - t_{wi}) \cdot (t_3 - t_{wo})} \qquad \text{Eq. 3.12}$$

For the condenser, as well as the evaporator, the difficulty is to determine the UA-value. Since the necessary data for the plate heat exchangers used as condenser and evaporator were not available, a correlation for the heat transfer was not established. Instead a mean value of $(UA)_c$ = 0.55 kW/K, calculated from measurements, was used. The subcooling (t_7-t_5) was held constant at 3 K.

3.1.3 Evaporator

The equations for the evaporator are expressed in the same way as for the condenser.

$$\dot{Q}_2 = \dot{m}_R \cdot (h_4 - h_6) \qquad \text{Eq. 3.13}$$

$$\dot{Q}_2 = \dot{m}_b \cdot (t_{bi} - t_{bo}) \cdot c_{p_b} \qquad \text{Eq. 3.14}$$

$$\dot{Q}_2 = (U \cdot A)_e \cdot \theta_e \qquad \text{Eq. 3.15}$$

$$\theta_e = \sqrt{(t_{bo} - t_2) \cdot (t_{bi} - t_4)} \qquad \text{Eq. 3.16}$$

Also in this case the UA-value was set to a constant value, equal to 4.81 W/K. The superheat (t_4-t_2) was held constant at 4 K.

3.1.4 Expansion valve

The expansion from the high to the low pressure is isenthalpic and thus:

$$h_5 = h_6 \qquad \text{Eq. 3.17}$$

3.1.5 Circulation pumps

For the energy saving potential calculations the sizes of the circulation pumps were adjusted to the different sizes of heat pumps. By assuming uniformity and constant efficiencies the following expressions apply:

$$\frac{\dot{V}_1}{\dot{V}_2} = \frac{n_1 \cdot D_1^3}{n_2 \cdot D_2^3} \qquad \text{Eq. 3.18}$$

$$\frac{\dot{W}_1}{\dot{W}_2} = \frac{n_1^3 \cdot D_1^5}{n_2^3 \cdot D_2^5} \qquad \text{Eq. 3.19}$$

Rearranging these and setting the rotational speed constant, the following expression applies for scaling the circulation pump size to the flow:

$$\frac{\dot{W}_1}{\dot{W}_2} = \left(\frac{\dot{V}_1}{\dot{V}_2}\right)^{5/3} \qquad \text{Eq. 3.20}$$

The reference values were chosen based on measurements on the circulation pumps in the experimental heat pump described in Appendix A and are listed in Table 3.1.

Table 3.1 Data for the circulation pumps used as reference for sizing circulation pumps to the actual flow.

$\dot{W}_{1;bp}$ (W)	$\dot{V}_{1;bp}$ (m^3/h)	$\dot{W}_{1;wp}$ (W)	$\dot{V}_{1;wp}$ (m^3/h)
136.5	1.55	46.82	0.858

3.2 Model compared with test data

The output from the model was compared to data from tests made on the heat pump described in Appendix A (Karlsson and Fahlén[40]). The comparison was made to steady-state tests at 30, 50 and 75 Hz and is shown in Table 3.2-Table 3.4. The largest difference between the model and test data showed for the compressor frequency 75 Hz. Despite the differences the coefficient of performance agrees fairly well with test data. The inputs to the model for these comparisons were the outgoing heating water temperature, t_{wo}, the incoming brine temperature, t_{bi}, and the heating water and brine flows, $\dot{V}_w$ and $\dot{V}_b$.

Table 3.2 Quotient and difference (temperatures) between data from the model and data from laboratory tests at the compressor frequency 30 Hz.

	10/35	**5/35**	**0/35**	**-5/35**	**10/50**	**5/50**	**0/50**	**-5/50**
$\dot{W}_{comp}$	1.00	0.99	0.97	0.95	1.02	1.00	0.97	0.94
$\dot{W}_{hps}$	1.00	0.99	0.97	0.95	1.02	1.00	0.97	0.94
$\dot{Q}_1$	1.04	1.01	0.98	0.93	1.10	1.07	1.03	0.99
$\dot{Q}_2$	1.10	1.09	1.07	1.04	1.19	1.17	1.16	1.16
COP_{hp}	1.02	1.02	0.99	0.97	1.07	1.05	1.04	1.04
COP_{hps}	1.02	1.02	0.99	0.97	1.08	1.05	1.04	1.04
p_1	1.02	1.01	1.01	1.01	1.02	1.02	1.02	1.01
p_2	1.06	1.06	1.04	1.03	1.10	1.09	1.09	1.09
t_3	-0.2	-1.0	-1.0	-1.0	-0.1	-0.2	-0.8	-1.8
t_5	1.0	0.6	0.9	1.0	1.3	1.2	1.0	0.8
t_4	1.9	1.6	1.2	0.8	2.9	2.6	2.3	2.4

Table 3.3 Quotient and difference (temperatures) between data from the model and data from laboratory tests at the compressor frequency 50 Hz.

	10/35	**5/35**	**0/35**	**-5/35**	**10/50**	**5/50**	**0/50**	**-5/50**
$\dot{W}_{comp}$	1.08	1.05	1.03	1.01	1.11	1.09	1.07	1.05
$\dot{W}_{hps}$	1.07	1.04	1.02	1.00	1.09	1.08	1.06	1.04
$\dot{Q}_1$	1.09	1.07	1.04	1.00	1.13	1.11	1.09	1.09
$\dot{Q}_2$	1.12	1.13	1.09	1.06	1.17	1.16	1.16	1.20
COP_{hp}	0.98	0.99	0.97	0.96	0.98	0.98	0.98	1.00
COP_{hps}	1.02	1.03	1.01	0.99	1.03	1.02	1.02	1.04
p_1	1.07	1.04	1.04	1.03	1.07	1.05	1.04	1.03
p_2	1.09	1.08	1.06	1.05	1.09	1.09	1.09	1.09
t_3	2.4	0.2	-0.3	-1.3	3.8	2.5	0.9	-1.5
t_5	3.5	2.1	1.9	1.7	3.1	2.5	2.1	1.3
t_4	2.5	2.1	1.6	1.3	2.7	2.6	2.4	2.4

Table 3.4 Quotient and difference (temperatures) between data from the model and data from laboratory tests at the compressor frequency 75 Hz.

	10/35	**5/35**	**0/35**	**-5/35**	**10/50**	**5/50**	**0/50**	**-5/50**
$\dot{W}_{comp}$	1.04	1.03	1.12	1.12	1.18	1.19	1.22	1.22
$\dot{W}_{hps}$	1.03	1.02	1.10	1.11	1.17	1.17	1.20	1.19
$\dot{Q}_1$	0.99	1.00	1.12	1.12	1.18	1.18	1.22	1.24
$\dot{Q}_2$	1.00	1.02	1.17	1.17	1.21	1.21	1.26	1.33
COP_{hp}	0.95	0.97	1.00	0.99	0.99	0.98	0.99	1.01
COP_{hps}	0.95	0.97	1.01	1.00	1.01	1.00	1.01	1.03
p_1	1.06	1.02	1.05	1.04	1.08	1.06	1.05	1.02
p_2	0.99	0.99	1.08	1.05	1.09	1.08	1.07	1.07
t_3	3.4	1.3	-0.2	-1.7	5.3	3.8	1.5	-2.3
t_5	3.3	1.7	2.5	1.9	4.0	3.1	2.3	1.2
t_4	-0.2	-0.5	2.2	1.4	2.6	2.3	1.9	1.8

The uncertainty of measurement of the tests has previously been determined by Fahlén[17], and the results from that investigation are compiled in Table 3.5.

Table 3.5 Uncertainties of measurement for the tests on the heat pump (Fahlén[17]).

Measurand	**Uncertainty of measurement**
Electric power (%)	0.7
Temperature of high and low-temperature media flow (K)	0.1
Volume flow of brine (%)	0.6
Volume flow of water (%)	0.3
Specific heat of brine (%)	6
Specific heat of water (%)	0.02
Density of brine (%)	2
Density of water (%)	0.02

3.3 Conclusions

The model will be used for the investigations in chapters 5 and 6 and for those purposes it is sufficient although it differs from test data. In chapter 5 an optimisation method will be investigated and then it is not necessary that the model exactly describes an actual heat pump. Also for the energy saving calculations in chapter 6, the model is sufficient since the different alternatives are compared at the same conditions using the same model and the energy savings of the actual heat pump is not the focus for the investigation.

4 Optimisation and control strategies

When discussing optimisation of operation, a distinction must be made between optimisation and control. This may seem trivial but may be confusing when reading earlier investigations. When “optimisation” is mentioned it is important clarify whether it is the control system that, in some way, is optimal or if it is the set-point that is optimal. When the optimisation refers to the control system it often implies that the proposed control system is fast and reaches the desired set-point quickly and as a consequence of this the heat pump becomes more efficient. This thesis focuses on finding the optimal set-point. Because of this, previous investigations dealing with optimisation of operation were divided into two groups in chapter 2; “Optimal set-point” and “Efficient control”. Svensson[68] discusses the issue above and separates the optimisation task into regulatory control tasks and optimising control tasks. The regulatory control task is to keep the process variables within specified bounds or at specified set-points. The optimising control task is to get the process to operate at the most favourable operating conditions. Optimisation methods may be divided into the categories off-line and on-line methods. Off-line optimisation might be used for processes that are not influenced by disturbances. If disturbances are present, or if the process parameters change over time, on-line optimisation is needed.

This chapter starts with a discussion on on-line optimisation. In the second section the measurement and control system used in this project is described. In the last section the optimisation method used is described.

4.1 On-line optimisation

Heat pumps are influenced by continually changing operating conditions caused by changes in climate that affect the temperature level of both the heat sink and the heat source. These changes have different time scales; the outdoor air temperature varies over the day as well as with the season. Other changes in operating conditions are caused by different activities in the heated building, such as airing. Thus the disturbances affecting the process can be considered to vary over time to such an extent that an on-line optimisation method should be used. Different principles for on-line optimisation have been discussed by Svensson[68] and some of his results are presented below.

If the controlled process is subjected to rapidly varying disturbances, the optimisation problem must be solved by dynamic optimising control. If the disturbances are infrequent and if the process dynamics are fast, i.e. the process responds quickly to control actions, steady-state optimising control can be used. For many heat pump applications a steady-state optimisation technique should be sufficient since the change in heat load varies slowly compared to the process dynamics. More about this will be discussed in the next chapter.

Steady-state optimisation techniques may be divided into direct search methods and indirect methods. A direct search method performs the optimisation without using any process model. The methods are generally easy to implement. The optimisation task is expressed by an objective function containing the relevant parameters. These parameters must be possible to determine from measurements. The manipulated variables are changed in order to evaluate the objective function at different operating conditions. These changes must be made at intervals sufficiently separated to let the process reach steady-state. This makes the direct search methods slow and thus they are suitable for processes with relatively fast dynamics and with few disturbances.

Indirect methods use a process model in order to optimise the operation. A model is seldom exact and thus parameters within the model must be updated from measurements if the actual optimum is to be reached. However, the model must still be representative over the interval in which the process will operate. An algorithm for the parameter update must be determined. Svensson[68] mentions two different techniques; recursive identification and moving horizon identification. In his work the moving horizon approach is used. As a result, this method gives a moving mean value.

In his work Svensson[68] used an indirect steady-state optimisation method where the parameters were updated by a moving horizon method. The optimisation method used was Sequential Quadratic Programming. The method was applied to a water-to-water heat pump with a variable-speed compressor and two variable-speed circulation pumps. The thesis has been discussed earlier, in chapter 2 and will not be further discussed here.

Although the method described above shows promising it has one major drawback. It is an open-loop algorithm and thus it is not possible to know if the real optimum is reached. If instead a closed-loop (feed-back) method was used, it would be possible to find the optimal point even if the model does not describe the process exactly. Such a closed loop method is described in the following sections.

4.2 Integrated Refrigeration Management (IRM)

The measurement system described below has previously been proposed by Fahlén[15, 22] and is illustrated in Figure 4.1, with the corresponding points in a Mollier diagram shown in Figure 4.2. Using this system makes it possible to determine, on-line, the coefficient of performance and the heating capacity and thus use these variables for optimisation purposes. The motor coefficient of performance is defined as the ratio between the heating capacity and the drive power:

$$COP_{hp} = \frac{\dot{Q}_1}{\dot{W}_{comp}} \qquad \text{Eq. 4.1}$$

Looking at the diagram in Figure 4.2 and Figure 4.3 the heating capacity and the drive power may be expressed as:

$$\dot{Q}_1 = \dot{m}_R \cdot (h_3 - h_5) \qquad \text{Eq. 4.2}$$

$$\dot{W}_{comp} = \dot{m}_R \cdot (h_3 - h_4) + f \cdot \dot{W}_{comp} \qquad \text{Eq. 4.3}$$

The loss factor, *f*, designates the part of the electric power supplied to the compressor that is not transferred to the refrigerant but transmitted to the surroundings. It is determined by the size and type of the compressor. The resulting expression for COP_{hp} will then be:

$$COP_{hp} = \frac{h_3 - h_5}{h_3 - h_4} \cdot (1 - f) \qquad \text{Eq. 4.4}$$

Thus by measuring the high and low-pressure and the three temperatures indicated in Figure 4.1 and knowing the loss factor the suggested measurement system makes it possible to determine the coefficient of performance. If the drive power is also measured the heating capacity can be determined. This measurement system can be used for an integrated approach to performance indication, control and fault detection and diagnosis, i.e. Integrated Refrigeration Management. The necessary measurement accuracy and the demand on sensors and their installation for use in this measurement system is discussed in chapter 7. How this measurement system can be used for optimisation purposes is described in chapter 5 and the use for fault detection and diagnosis is discussed in section 7.2.

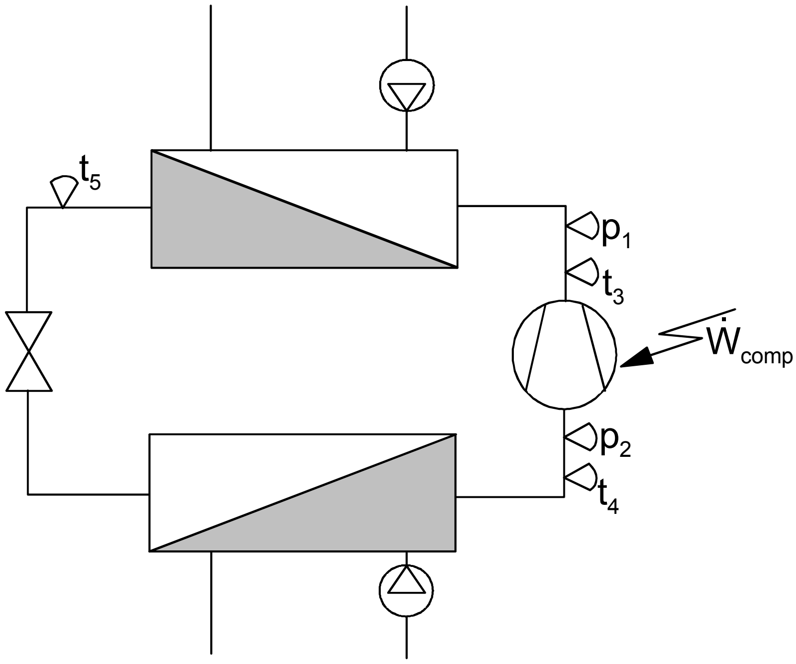


Figure 4.1 Schematic diagram of a heat pump.

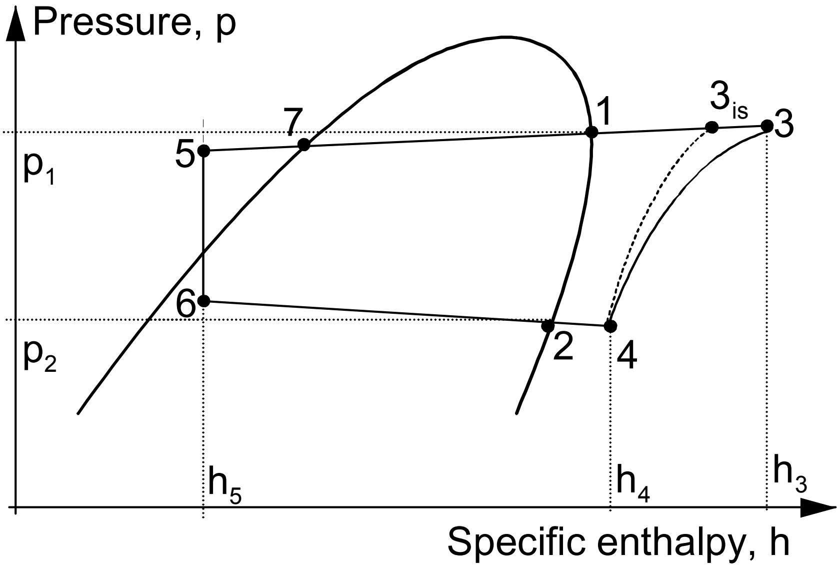


Figure 4.2 The heat pump process plotted in a Mollier diagram. The numbers correspond to the numbers in Figure 4.1 (Fahlén[18]).

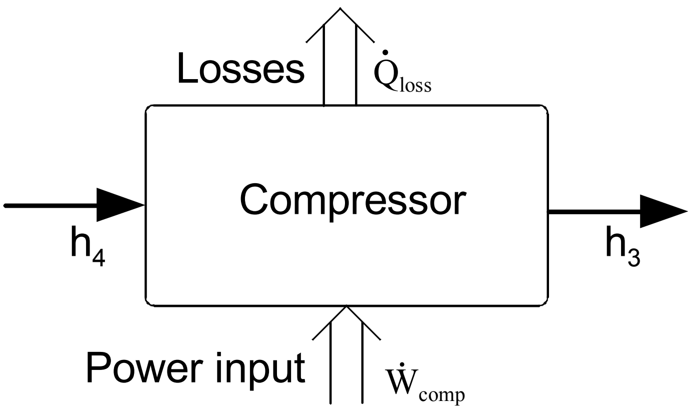


Figure 4.3 Energy balance for the compressor (Fahlén[18]).

4.3 Nelder-Mead's simplex method

The approach taken in this project for on-line optimisation was to use a general, model independent, method that can be applied to different kinds of heat pumps. As mentioned previously there are a number of such methods. Here the Nelder-Mead simplex method (Nelder and Mead[46]) is used mainly because of its simplicity. The method is described below according to Avriel[4].

The principle of the simplex method is to evaluate a number of test points, exclude the worst of them, and find a new point instead of this worst one. This process continues until the process reaches its optimum as defined by an objective function. This objective function defines the optimisation problem and the simplex method will minimise the value of this function. For a heat pump, the objective function should include the coefficient of performance as described in chapter 5. The method can be used in processes with an infinite number of variables but it gets increasingly slower and is considered to be inefficient if the number of variables exceeds 10.

A simplex consists of $n+1$ points, i.e. one point more than the number of variables, n. So, for one variable it forms a line and for two variables the simplex forms a triangle (see Figure 4.4) and so forth. The evaluation points are numbered $x_0, x_1, \ldots, x_n$. The points with the highest and lowest value of the objective function are defined as:

$$x_H = \max\{f(x_0), f(x_1), \ldots, f(x_n)\} \qquad \text{Eq. 4.5}$$

$$x_L = \min\{f(x_0), f(x_1), \ldots, f(x_n)\} \qquad \text{Eq. 4.6}$$

The centroid for all the points except x_H becomes:

$$\bar{x} = \frac{1}{n}\sum_{i=0}^{n} x_i, \quad x_i \neq x_H \qquad \text{Eq. 4.7}$$

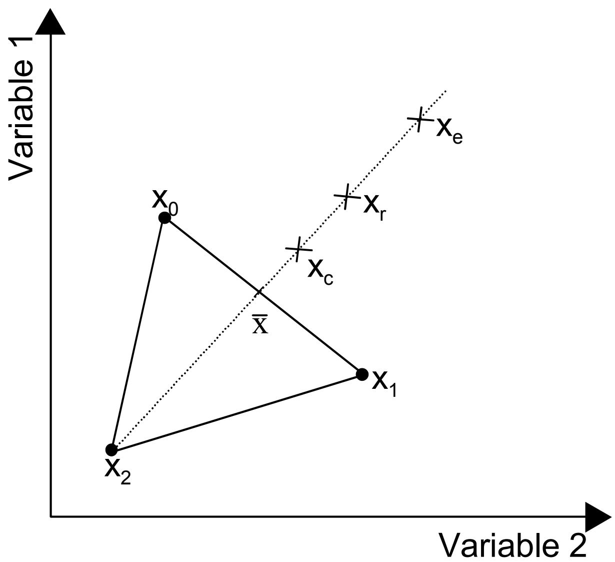


Figure 4.4 For two variables the simplex forms a triangle. In the picture the three test points and the reflection, expansion and contraction steps are indicated.

Looking at Figure 4.4, if x_2 is the point with the highest value of the objective function and x_0 gives the lowest, the centroid will be in the middle between x_0 and x_1. Now the point $x_H = x_2$ will be replaced with a new and hopefully better point. This is done by first taking a reflection step:

$$x_r = \bar{x} + \alpha \cdot (\bar{x} - x_H) \qquad \text{Eq. 4.8}$$

α is a positive constant called the reflection coefficient.

The value of the objective function is now evaluated at this point and depending on the result three different cases are possible:

1. $f(x_L) > f(x_r)$
2. $\max\{f(x_i), x_i \neq x_H\} \geq f(x_r) \geq f(x_L)$
3. $f(x_r) > \max_i\{f(x_i), x_i \neq x_H\}$

These three cases are further described below.

1. In this case the point x_r is a new minimum. For our example the objective function has a lower value at point x_r than at point x_0. This may indicate that we are moving in the correct direction. Thus a new step in the same direction is taken, i.e. an expansion step.

$$x_e = \bar{x} + \gamma \cdot (x_r - \bar{x}) \qquad \text{Eq. 4.9}$$

γ is a positive constant >1.

If $f(x_e) \geq f(x_r)$ the expansion step failed and x_H is replaced by x_r in the new simplex. If instead $f(x_e) < f(x_r)$ then the expansion step succeeded and x_e replaces x_H in the new simplex.

2. In this case the second highest value of the objective function is higher than the value at the reflection point. Then x_r will replace x_H.

3. If the value of the objective function is higher than the second highest value of the objective function in the current simplex then x_H cannot be replaced by x_r since this would then be the new point x_H in the new simplex. This is solved by defining a new point $x_{H'}$ as follows:

$$f(x_{H'}) = \min\{f(x_H), f(x_r)\} \qquad \text{Eq. 4.10}$$

After this a contraction step is taken:

$$x_c = \bar{x} + \beta \cdot (x_{H'} - \bar{x}) \qquad \text{Eq. 4.11}$$

β is the contraction coefficient, which has a constant value between zero and one ($0 < \beta < 1$).

This new point replaces x_H in the new simplex if not $f(x_c) > f(x_{H'})$. In this case all points in the simplex, x_i, are replaced by new ones as follows:

$$\hat{x}_i = x_i + \frac{1}{2} \cdot (x_L - x_i), \quad i = 0,\ldots,n \qquad \text{Eq. 4.12}$$

This step will halve the distance between the point with the lowest value of the objective function and the other points.

Following the steps above will give a new simplex that will be evaluated in the same way and thus the point that minimises the objective function is approached. This will continue forever if no suitable terminating criterion is formulated. A suggestion quoted by Avriel[4] is:

$$\left\{\frac{1}{n+1}\cdot\sum_{i=0}^{n}[f(x_i)-f(\bar{x})]^2\right\}^{1/2} < \varepsilon \qquad \text{Eq. 4.13}$$

where ε is a predetermined positive number.

5 Optimisation of heat pump operation

This chapter will show how operational optimisation of a brine-to-water heat pump may be performed. First a method of how the previously described Nelder-Mead simplex method can be applied to a heat pump system is presented in principle. In the second section, this method is applied to a brine-to-water heat pump and evaluated by computer simulations. The last two sections discuss and conclude the results from these simulations.

5.1 The Nelder-Mead simplex method used in the IRM concept

The simplex method as described in section 4.3 is valid for unconstrained optimisation. For a heat pump there are limits to the operating range of some component characteristics such as compressor speed etc. This can be handled by introducing penalty functions into the objective function as will be shown later.

Applying the simplex method to a heat pump, there are almost always four components or processes to control; the compressor speed, the speed of two circulation pumps or fans and the evaporator superheat. The evaporator superheat control is not included in the simplex method for this investigation. This is due to the fast dynamics of the superheat, which will not be possible to follow with the simplex method. The superheat control is thus left to its own local control loop, no matter if it is a thermostatic or an electronic expansion valve. As a consequence, there are three variables left to optimise, namely the speed of the compressor and the pumps or fans. This means that the simplex contains four points and will look like a tetrahedron if plotted.

Now the objective function that is to be optimised must be determined. In this case the goal is to achieve the most energy efficient operation and thus the *COP* must be maximised. As the simplex method searches for a minimum, 1/*COP* will be minimised. To get a good operation, however, the optimisation cannot focus only on *COP*. The heat pump must also deliver the required heating capacity to cover the heat load and that capacity must be delivered at a temperature level that is sufficient to keep the room temperature at the desired level. These two constraints are handled by penalty functions. This is sufficient since they are not hard constraints, i.e. no damage will occur if they are violated for a short period of time. If these constraints and the *COP* are included in the objective function it can be expressed as:

$$g_1 = \frac{1}{COP_{hp}} + \rho_1 \cdot (\dot{Q}_1 - \dot{Q}_{1;set})^2 + \rho_2 \cdot (\bar{t}_w - \bar{t}_{w;set})^2 \qquad \text{Eq. 5.1}$$

The coefficients ρ_1 and ρ_2 are constants and set the penalty on the deviation from the set-point.

The objective function g_1 is suitable if the auxiliary equipment is not included in the optimisation process. If these are to be included, the objective function should be formulated as:

$$g_2 = \frac{1}{COP_{hps}} + \rho_3 \cdot (\dot{Q}_{1hps} - \dot{Q}_{1hps;set})^2 + \rho_4 \cdot (\bar{t}_w - \bar{t}_{w;set})^2 \qquad \text{Eq. 5.2}$$

Here it is important to be aware of the following definitions:

$$\dot{Q}_{1hps} = COP_{hp} \cdot \dot{W}_{comp} + \dot{W}_{wp} \qquad \text{Eq. 5.3}$$

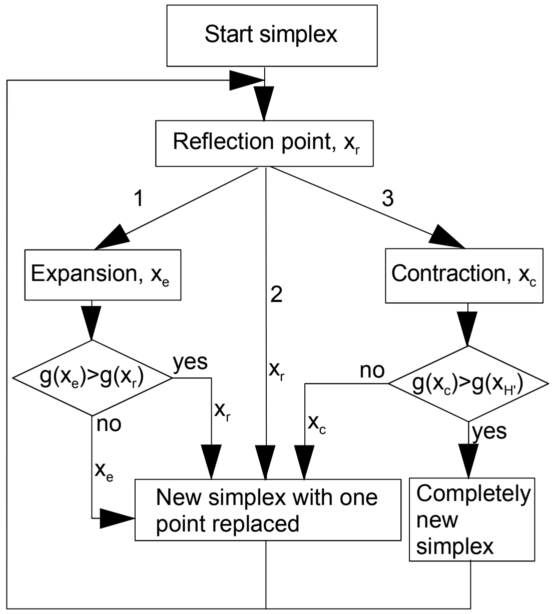
$$COP_{hps} = \frac{\dot{Q}_{1hps}}{\dot{W}_{comp} + \dot{W}_{bp} + \dot{W}_{wp}} \qquad \text{Eq. 5.4}$$

The suggested IRM measurement system makes it possible to determine $\dot{Q}_1$, using Eq. 4.1. To perform optimisation according to Eq. 5.2 the drive power of the circulation pumps must be determined. New, ”intelligent”, pumps are now available on the market where measurement of the drive energy is included in the control unit. Thus it is possible to determine also objective function g_2 correctly.

The objective functions g_1 and g_2 can be used for optimisation including soft constraints that may be violated for a short period of time. This technique cannot be used for hard constraints such as limitations in compressor speed that must not be violated. For such cases a limitation must be set to the control signal. A possible solution is that if the optimisation algorithm puts out a signal violating a constraint, the limit value is fed to the controlled component and this is also fed back to the algorithm. Then the constraint is not violated and the optimisation algorithm is updated with the actual set-point.

The different steps in the optimisation process for a heat pump would then be as follows:

1. The four starting points are run and the value of the objective function is calculated for each point.
2. Based on the results from point 1 the reflection point is calculated
3. The heat pump is operated at the reflection point and the value of the objective function is calculated.
4. If the simplex algorithm results in an expansion or contraction step the heat pump is operated at one of these points and the value of the objective function is determined. If no contraction or expansion step is taken the algorithm jumps to point 5.
5. If the contraction step fails a new simplex is formed and four new points must be run
6. A new simplex is found and the algorithm starts again at point 2.



1. $f(x_L) > f(x_r)$
2. $\max\{f(x_i), x_i \neq x_H\} \geq f(x_r) \geq f(x_L)$
3. $f(x_r) > \max_i\{f(x_i), x_i \neq x_H\}$

Figure 5.1 The different steps in the simplex method

In this work order it is assumed that it is enough to run each point just one time, i.e. operating points run in the preceding iteration must not be run again.

This optimisation algorithm also assumes steady-state operation as stated in the beginning of this chapter. Thus a steady-state criterion must be formulated. A suitable steady-state criterion is that the temperature of the heating water leaving the heat pump has been stable within $\pm\Delta t_{stable}$ °C during the last τ_{stable} minutes, i.e.:

$$t_{wo;\max} - t_{wo;\min} < \Delta t_{stable} \qquad \text{Eq. 5.5}$$

This is a simple criterion but has one major drawback, it is sensitive for sudden disturbances. In that sense it is better to formulate a criterion based on the standard deviation:

$$s = \sqrt{\frac{\sum_{i=1}^{n}\left(t_{wo,i} - \bar{t}_{wo}\right)^2}{n-1}} < \kappa \qquad \text{Eq. 5.6}$$

Suitable values of the limits will depend on the heat pump and the heating system to which it is connected.

5.2 Optimal operation of a brine-to-water heat pump

The steady-state heat pump model in chapter 3 was used for evaluation of the proposed optimisation method. The models for the single-speed pumps were changed to models for variable-speed pumps. The flow, pressure rise and drive power were expressed according to the affinity laws as:

$$\dot{V} = \dot{V}_{ref} \cdot \frac{f_p}{f_{p;ref}} \qquad \text{Eq. 5.7}$$

$$\Delta p = \Delta p_{ref} \cdot \left(\frac{f_p}{f_{p;ref}} \right)^2 \qquad \text{Eq. 5.8}$$

$$\dot{W}_p = \dot{W}_{ref} \cdot \left(\frac{f_p}{f_{p;ref}} \right)^3 \qquad \text{Eq. 5.9}$$

The reference points are the intersections of the system curves and the pump curves at the maximum pump speed. The pressure drop for the brine and heating systems are in this investigation chosen as the pressure drop for the evaporator and condenser only. The pump curves are inferred from the datasheet of a suitable commercially available variable-speed pump. Data for the pumps are given in Table 5.1.

Table 5.1 Input data to the model for the variable-speed pumps.

Brine pump		**Heating water pump**	
$f_{bp;ref}$	50 Hz	$f_{wp;ref}$	50 Hz
$f_{bp;max}$	50 Hz	$f_{wp;max}$	50 Hz
$f_{bp;min}$	12.5 Hz	$f_{wp;min}$	12.5 Hz
$\Delta p_{bp;ref}$	23.5 kPa	$\Delta p_{wp;ref}$	22 kPa
$\dot{V}_{b;ref}$	1.6 m^3/h	$\dot{V}_{w;ref}$	1.7 m^3/h
$\dot{W}_{bp;ref}$	56 W	$\dot{W}_{wp;ref}$	57 W

The efficiencies of the circulation pumps are considered to be constant. This level of accuracy of the pump data was considered to be sufficient for the purpose of investigating the potential for the Nelder-Mead simplex method.

The resulting model was implemented by means of the software EES (Engineering Equation Solver, Klein and Alvarado[41]) and if t_{bi} and t_{wi} were specified, the optimal combination of compressor speed and pump speeds could be calculated. This was used for the unconstrained optimisation in order to see if the chosen optimisation method could find the optimum.

5.2.1 Unconstrained optimisation

The simplest task would be optimisation without constraints, i.e. no limitations to the rotational speed of pumps or compressor and no criteria for the heating capacity or temperature level. Thus the objective function can be formulated as:

$$g = \frac{1}{COP_{hps}} \quad \text{Eq. 5.10}$$

Before the optimisation could start, the first simplex must be established. In this case it was arbitrarily chosen with one point in the vicinity of the optimal operating point as calculated by EES. Thus the starting simplex looks as in Table 5.2 where also the reflection, expansion and contraction coefficients are shown. The values of these are chosen according to Avriel[4].

Table 5.2 The points for the first simplex. The values of the reflection, expansion and contraction coefficients are given under the table.

	Points in the starting simplex			
Frequency	**1**	**2**	**3**	**4**
f_{comp} (Hz)	50	46	50	60
f_{bp} (Hz)	50	50	40	30
f_{wp} (Hz)	50	22	30	40

$\alpha = 1$ $\gamma = 2$ $\beta = 0.5$

The incoming heating water temperature and the incoming brine temperature were set as constant values defining the process. Setting the values for these as $t_{bi} = -1$ °C and $t_{wi} = 44$ °C gives the resulting optimisation procedure as shown in Figure 5.2.

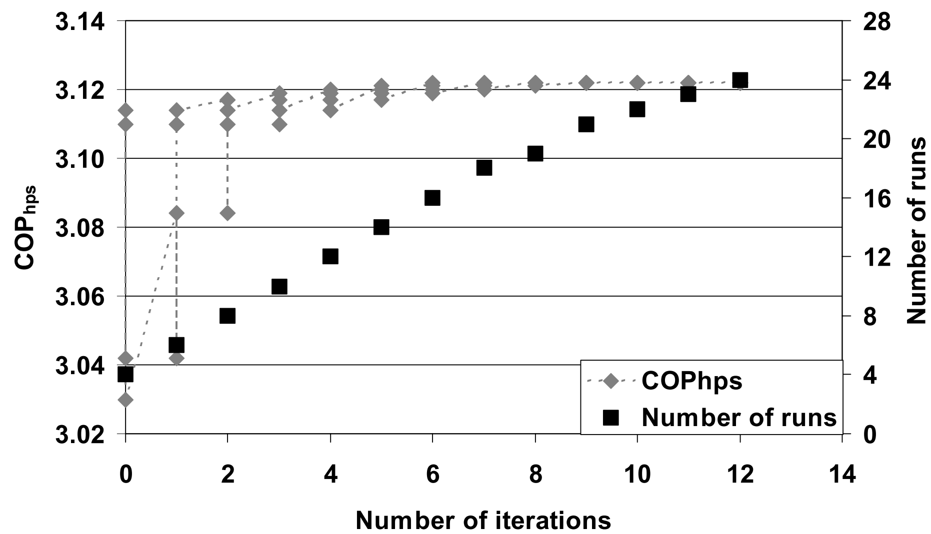


Figure 5.2 The number of iterations and number of runs (test points) for finding the optimal set-point when $t_{bi} = -1$ °C and $t_{wi} = 44$ °C.

The maximum COP_{hps} according to EES is 3.128 and this is reached with f_{comp} = 44.8 Hz, f_{bp} = 50 Hz and f_{wp} = 21 Hz. The simplex method converges towards a value of 3.122. The difference between the optimisation method used by EES and the simplex method depends on the step size chosen for the simplex method. However is the difference in COP_{hps} between the two methods much smaller than what can be measured in practice. After the second iteration and eight runs, the efficiency of the heat pump is higher than the highest value for the starting simplex and the difference between this value (3.118) and the maximum value is so small that the optimisation process could be terminated at this stage. If terminated here the operating frequencies for the compressor and the pumps would be f_{comp} = 50.2 Hz, f_{bp} = 39.2 Hz and f_{wp} = 19.5 Hz. These values differ somewhat from the ones calculated by EES and indicate that the optimum is quite flat.

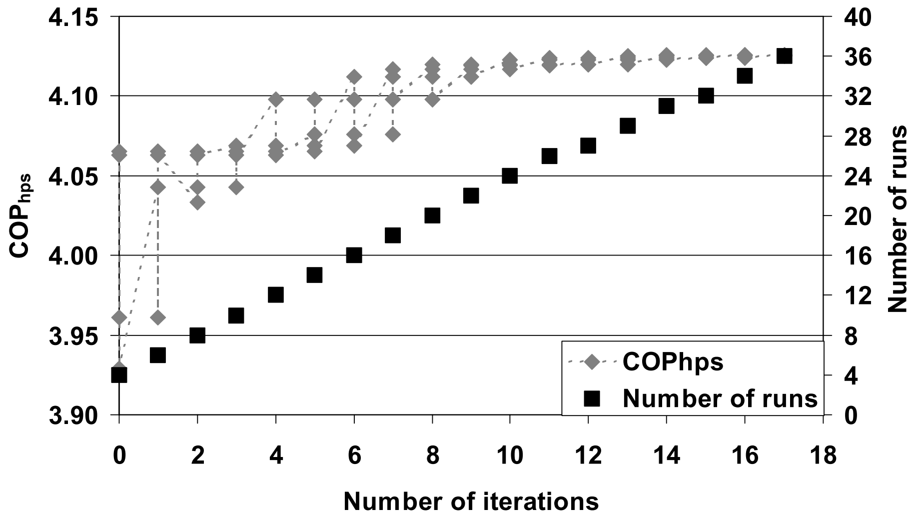


Figure 5.3 The number of iterations and number of runs (test points) for finding the optimal set-point when t_{bi} = -1 °C and t_{wi} = 25 °C.

With the same starting simplex as above and instead setting t_{wi} = 25 °C the optimisation process will look as in Figure 5.3. Compared to the first example, 18 more runs must be made in order to reach the best COP_{hps}. This is a significant increase indicating that the same starting simplex cannot be used for the entire operating range. Other simulations were made where the values of α, β and γ were changed to see whether this could make the optimisation converge faster (α and γ were decreased, β was both increased and decreased). These simulations were made for the case in Figure 5.2. The optimisation proved to be unstable but could reach a higher value of COP_{hps} (3.128). This is equal to the optimum calculated by EES. This could be reached due to the smaller step size used for these calculations compared to the calculations showed in Figure 5.2. Important to notice is that it is possible to refine the search by changing the step size but the actual difference in COP_{hps} showed here (0.006) is too small to be of practical interest.

5.2.2 Constrained optimisation

Previously the only goal was to maximise the coefficient of performance. Now the objective is to maximise the coefficient of performance including the constraints that the heating capacity and the temperature level in the condenser are to be kept at certain prescribed levels. Thus the objective function will be formulated according to Eq. 5.2.

The set value for the heating capacity and the mean temperature were set to 5 kW and 44 °C respectively. The values of α, β and γ were set to values according to Table 5.2. The coefficients ρ_1 and ρ_2 are used for weighting the importance of the constraints relative to the main goal, which in this case is to maximise the energy efficiency. For this simulation $\rho_1 = 0.1$ and $\rho_2 = 0.01$ and the input data defining the process were set as $t_{wi} = 40$ °C and $t_{bi} = -1$ °C. How the optimisation process proceeded is shown in Figure 5.4 - Figure 5.7. The objective function stabilises after 8 iterations or 23 runs. The increased number of runs compared to the unconstrained optimisation is not surprising since two more parameters are added, but this large number of runs implies some practical drawbacks that will be further discussed in the next section. It can also be seen that even though the objective function, the coefficient of performance and the heating capacity, have stabilised, the mean temperature of the heating water keeps fluctuating. Increasing the weighting coefficient, ρ_2 may solve this problem. The resulting settings, after 8 iterations, for the compressor and the pumps are: $f_{comp} = 41.3$ Hz, $f_{bp} = 30.7$ Hz and $f_{wp} = 14.4$ Hz.

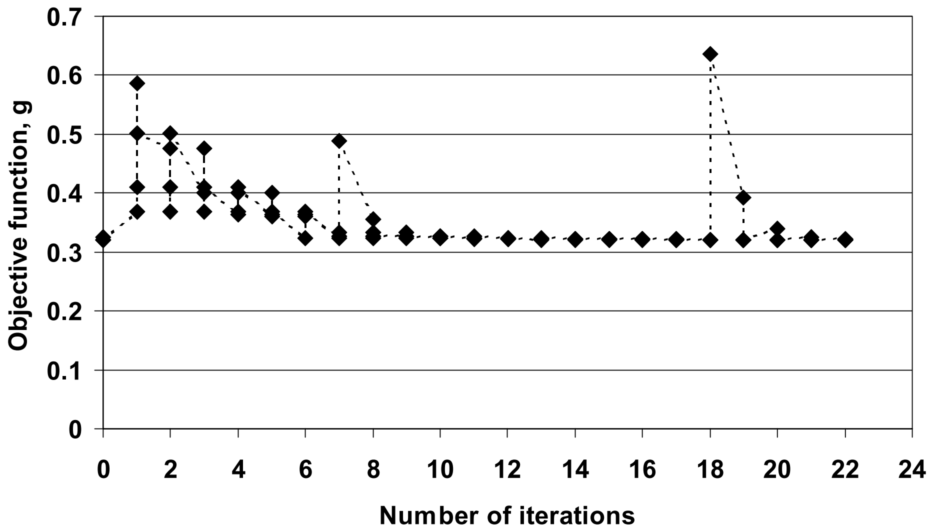


Figure 5.4 The value of the objective function during the optimisation procedure.

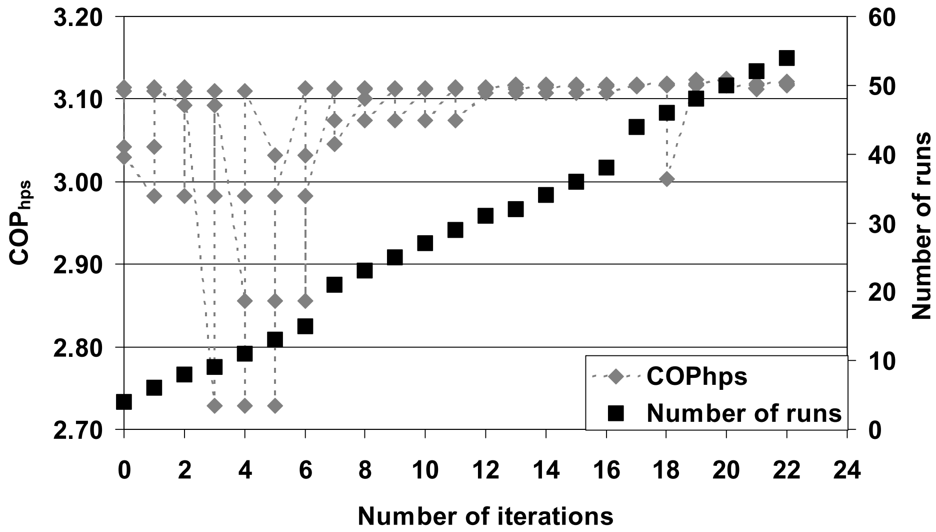


Figure 5.5 The value of the coefficient of performance and the number of runs necessary to reach the optimal set-point.

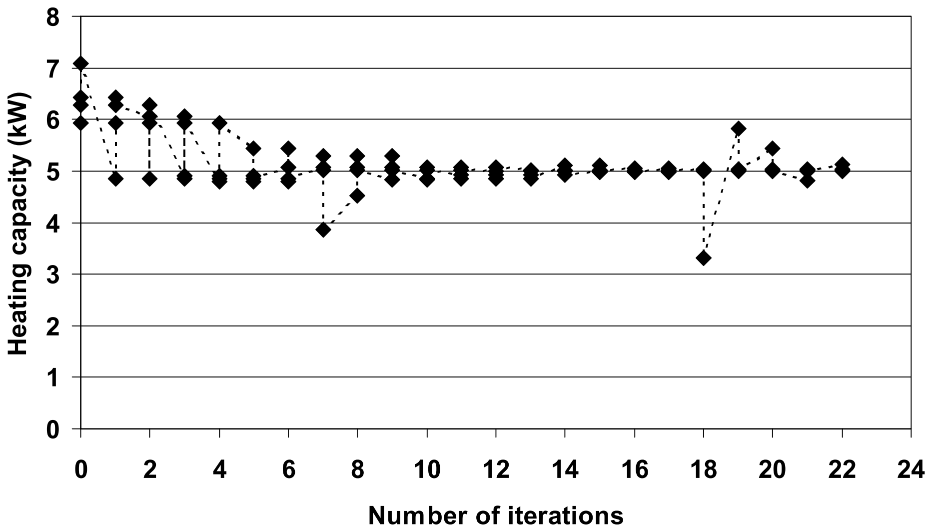


Figure 5.6 The change of the heating capacity as the optimisation proceeds.

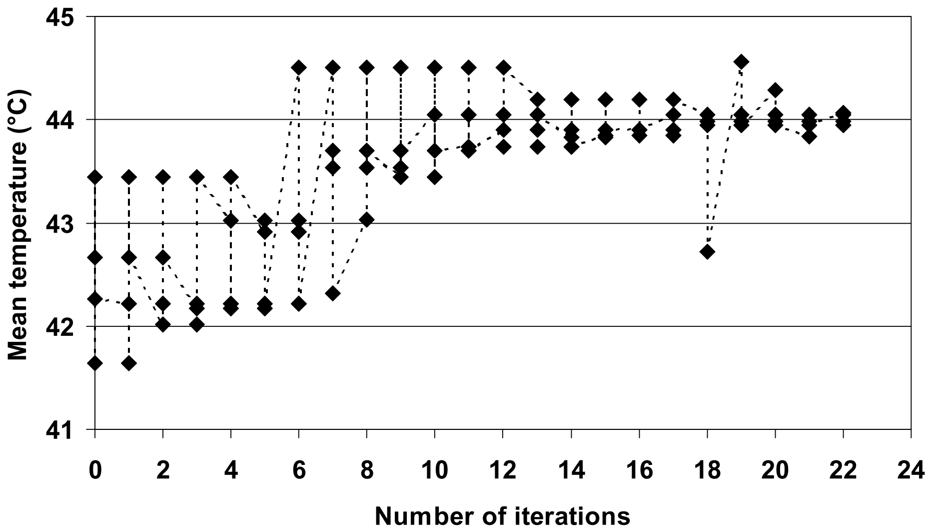


Figure 5.7 The change of the heating water mean temperature as the optimisation algorithm proceeds.

5.3 Discussion

The suggested optimisation method seems promising; it can maximise the coefficient of performance and it is also possible to handle constraints by expressing them as penalty functions and include them in the objective function. A prerequisite for this method is that the frequency of disturbances is much slower than the dynamics of the process. It is also necessary that the process is allowed to reach steady-state operation between the changes in order to perform the necessary measurements. The possibilities for fulfilling these criteria will be different for different kinds of heat pumps and heating systems. A mini-water system will respond faster to changes and thus the time required to reach steady-state will be shorter and thus the time for reaching the optimal set-point may be shorter. On the other hand the thermal mass of this system is quite small and thus it is more sensitive for disturbances. The opposite of this system is a floor-heating system that responds slowly to changes but on the other hand are less sensitive to disturbances. Disturbances in these cases can for example be airing. The primary parameter affecting the variations of a heating system is the change of the heating demand. This changes with the climatic changes which are slow and is further damped out by the thermal mass of the building. Typically the time constant for a building is of the order of 24 hours. Thus this should not present any major difficulties.

Another issue is the limitation to the rotational speed of mainly the compressor. This speed cannot be varied in an unlimited range and for low heat loads it will start to operate on and off. A possible solution then would be to save the settings of the process and let the process start with these parameters when activated again. If the cycle period of the on/off operation would be short it is possibly better to terminate the optimisation algorithm.

As indicated in Figure 5.5 the suggested optimisation method can be rather slow and if it is considered too slow for a certain application it can still be used if combined with a model-based method. Then a fairly simple model may be used in order to find an operating point in the vicinity of the optimal point. Then the simplex method is used for finding the true optimum. This will most likely make the process faster.

5.4 Conclusions

The suggested optimisation seems promising and should be possible to use in order to perform on-line optimisation of heat pump operation. If considered too slow it should be possible to use it together with a model, which is used for finding a set-point in the vicinity of the optimal point. Then the suggested method would find the actual optimum. Still further evaluation is needed by laboratory tests or field trials.

6 Energy saving potential

As described in chapter 2, the energy efficiency of heat pumps may be increased if variable-speed compressors, pumps and fans as well as electronic expansion valves are used instead of the traditional single-speed units and thermostatic expansion valves. An analysis of the benefit of using a variable-speed compressor and an electronic expansion valve in a brine-to-water heat pump will be presented in this chapter. First results from laboratory tests will be presented and discussed followed by an analysis of how the seasonal efficiency changes with compressor efficiency and the heat pump size in relation to the heat load.

6.1 Laboratory tests

In order to see how variable-speed capacity control works in practice for a brine-to-water heat pump the prototype described in Appendix A was built and tested in a laboratory. Steady-state tests at different compressor speeds were performed as well as comparative tests between on/off and variable-speed part load control of the compressor. Finally, comparative tests between an electronic expansion valve and a thermostatic expansion valve were performed. If not explicitly stated otherwise, in this chapter the compressor power includes the power used by the inverter.

6.1.1 Steady-state tests

To determine the performance of the heat pump over the allowed operating range (30-75 Hz) of the compressor, tests were performed at 30, 50 and 75 Hz for the operating points in Table 6.1.

Table 6.1 Temperature levels for the steady-state tests.

Temperature of brine inlet to evaporator, t_{bi} (°C)	Temperature of heating water outlet from condenser, t_{wo} (°C)
- 5	+ 35
± 0	
+ 5	
+ 10	
- 5	+ 50
± 0	
+ 5	
+ 10	

Tests were performed according to the European test standard for heat pumps and air-conditioners, SS-EN 255-2[63]. Figure 6.1 shows that the efficiency of the compressor has a peak around the compressor frequency 50 Hz. This result is not surprising since the compressor was designed for single-speed operation. Similar results have earlier been reported by Poulsen[51,52] and Riegger[57]. The efficiency of this experimental heat pump is in the same range as commercially available brine-to-water heat pumps with the same heating capacity. The values at t_{wo} = 35

°C are slightly lower than for the best units on the market but for t_{wo} = 50 °C the efficiency is at the same level.

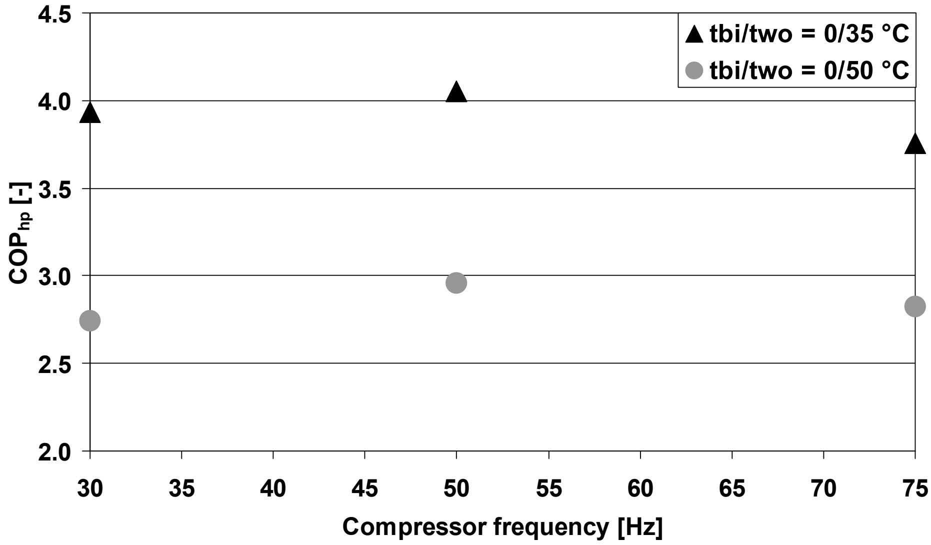


Figure 6.1 Heat pump coefficient of performance as a function of the compressor frequency for two of the operating points in Table 6.1.

6.1.1.1 Comparative tests between an electronic and a thermostatic expansion valve

Both the electronic and the thermostatic expansion valves were tested for the operating points in Table 6.1 for the three compressor frequencies 30, 50 and 75 Hz. The electronic expansion valve keeps the superheat at a lower level than the thermostatic valve does (approximately 2 °C at the most), but this hardly shows on the COP_{hp}, the largest difference in COP_{hp} is 4 %. This is in accordance with the investigations by Tassou and Al-Nizari[70] and Aprea and Mastrullo[3]. However, in both those investigations the electronic expansion valve was set to keep a constant superheat. For the tests presented here, the valve was controlled using an adaptive control algorithm that follows the MSS-line, i.e. keeps the superheat at the most energy efficient level. Despite this, the heat pump coefficient of performance shows no or quite a small increase when compared to the thermostatic valve.

6.1.2 Part load tests

Part load tests were made in order to investigate the energy saving potential of the variable-speed control. The performance of the heat pump using variable-speed control was compared to that of the same heat pump with conventional on/off control. The tests were performed according to a draft for a new European technical specification for part load testing of heat pumps and air-conditioners, CEN/TC 113 N 366[9]. The incoming brine temperature to the evaporator, t_{bi}, and the incoming water temperature to the condenser, t_{wi}, defined the operating points. The incoming water temperatures were set to the levels achieved in the steady-state tests. The steady-state tests at 50 Hz were defined as full load, i.e. 100 %

load. The part loads were then defined by the heating capacity in relation to the heating capacity at these full loads. In other words, if the heating capacity at 50 Hz, full load, was 7 kW then a part load of 50 % means that the heating capacity equals 3.5 kW.

For the variable-speed control, the heating capacity was adjusted by the compressor frequency. The part loads 75 % and 63 % were tested (63 % corresponds to 30 Hz, the lowest permissible frequency). For the on/off control, the part loads were set by time control, e.g. 50 % means 30 minutes in operation and 30 minutes turned off when the cycle time was one hour. For this investigation the part loads 75, 63 and 50 % were tested. Because of the dynamics during the starts and stops, the measurements lasted for two hours with a sampling interval of 5 seconds. For the variable-speed control, the measurements lasted 30 minutes with a sampling interval of 30 seconds. The test points for the part load tests are compiled in Table 6.2.

Table 6.2 Test points for part load tests. The tests at 50 % load were run only with the compressor operated by on/off control.

	Load (%)		
Test point t_{bi} / t_{wi} (°C)	**Variable-speed**	**On/off**	**Corresponding test point at full load t_{bi} / t_{wo} (°C)**
5 / 26.5	63	63 50	5 / 35
0 / 27.7	75 63	75 63 50	0 / 35
5 / 42.7	63	63 50	5 / 50
0 / 44.0	75 63	75 63 50	0 / 50

The results are illustrated in Figure 6.2 to Figure 6.5. As expected the coefficient of performance decreased with deceasing load when the compressor operated on/off, even though the decrease was small. The decrease was less than 3 % between 100 % and 50 % load. One explanation for this low decrease in efficiency is that the circulation pump for the brine was switched on and off simultaneously as the compressor and thus it was only the circulation pump for the heating water that was in operation during the periods when the compressor was not in operation. The electric power for this pump is low compared to the rest of the system and had no great impact on the system efficiency. For these tests the cycle time was one hour. In order to see whether the cycle time has an impact on the results, additional tests were made at the test point 0/27.7 ˚C and 50 % load (see Table 6.2) for the cycle times 30 minutes and 15 minutes. No difference in efficiency was observed from these tests. As reported in chapter 2, several earlier investigations have reported that the on/off cycling leads to increased losses. Most likely these heat pumps have been equipped with capillary tubes instead of a thermostatic expansion valve. Thus the losses for those heat pumps originate from refrigerant migration. This migration will be low for a heat pump with a shutting valve such as a thermostatic expansion valve and heat exchangers with small volumes such as plate heat exchangers.

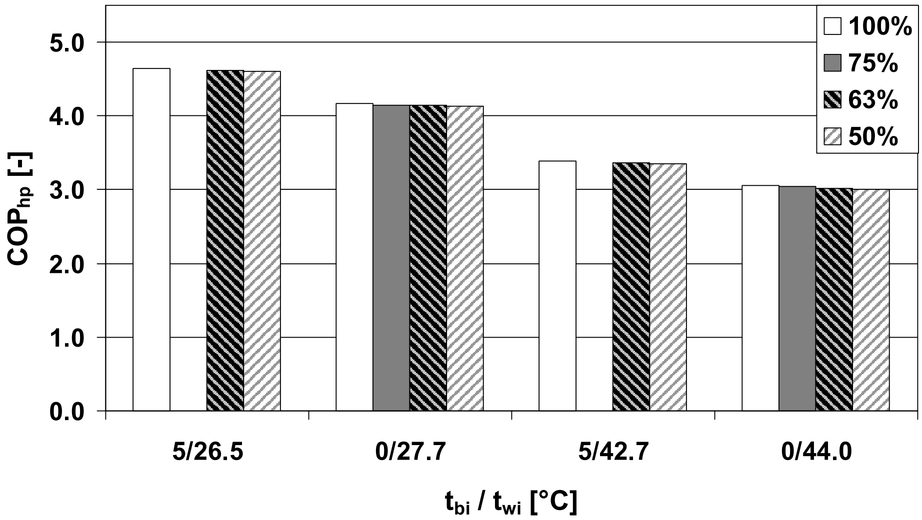


Figure 6.2 Heat pump coefficient of performance for on/off control (pumps excluded). The 75 % part load has not been tested for the operating points 5/26.5 and 5/42.7.

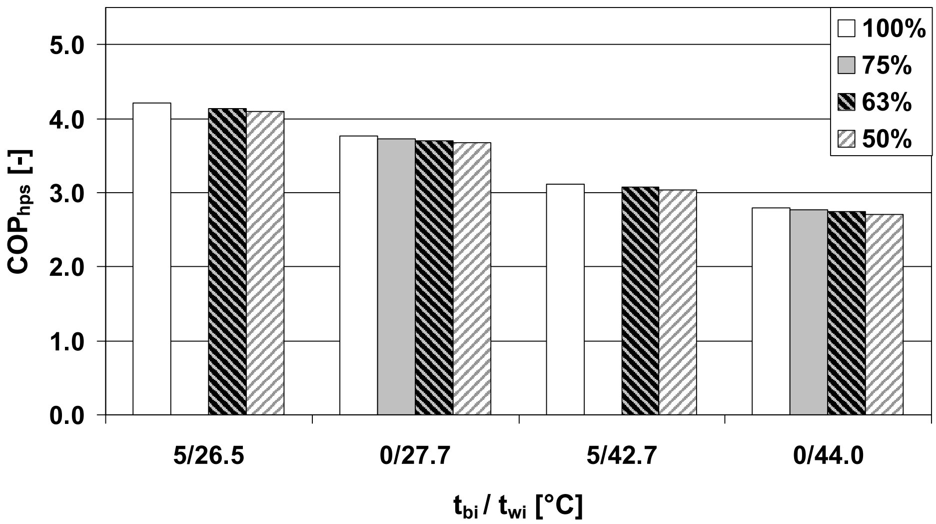


Figure 6.3 Heat pump system coefficient of performance for on/off control (pumps included). The 75 % part load has not been tested for the operating points 5/26.5 and 5/42.7.

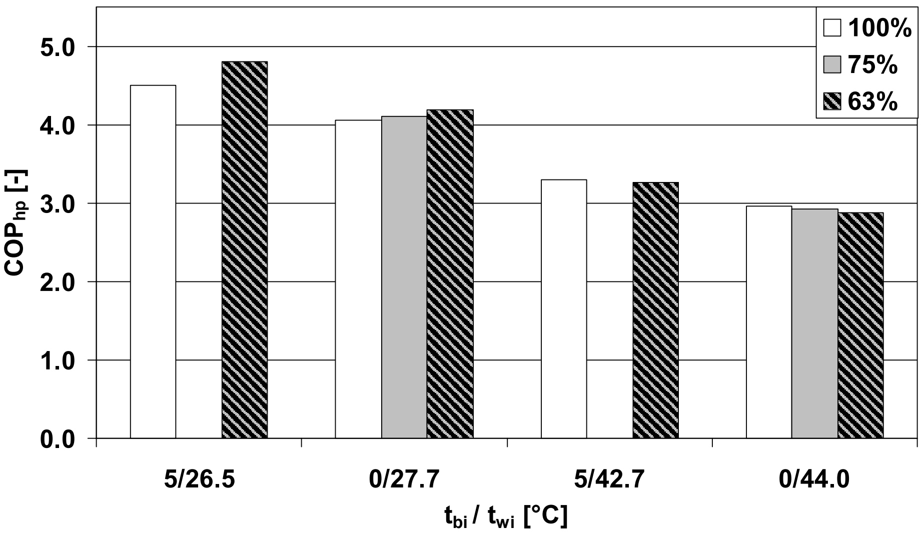


Figure 6.4 Heat pump coefficient of performance for variable-speed control (pumps excluded). The 75 % part load has not been tested for the operating points 5/26.5 and 5/42.7.

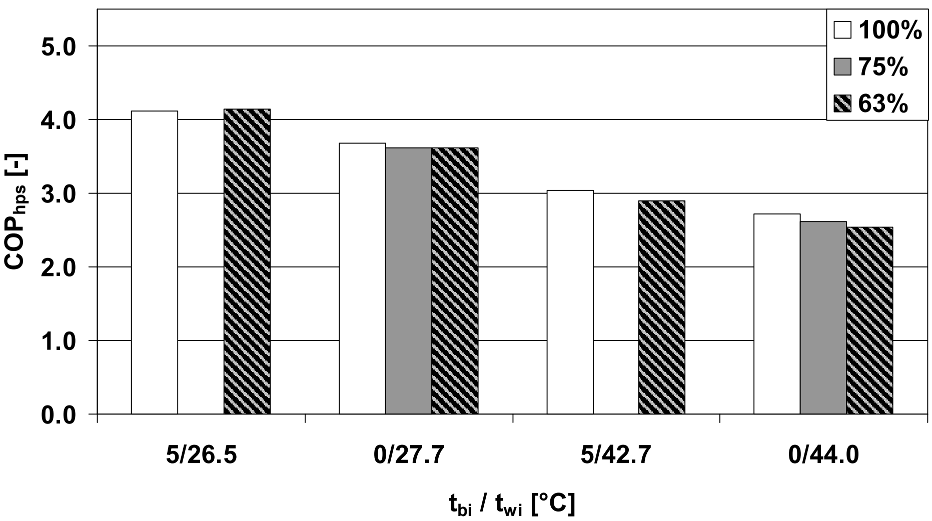


Figure 6.5 Heat pump system coefficient of performance for variable-speed control (pumps included). The 75 % part load has not been tested for the operating points 5/26.5 and 5/42.7.

For the variable-speed tests the results were contrary to the expected. Based on the review in chapter 2, the coefficient of performance was expected to increase for decreasing loads. But the results show that the COP_{hp} increases only for the operating points with low heating water temperature and then only slightly. When the circulation pumps are included in the analysis the coefficient of performance decreases for all operating points. The COP_{hps} decreases because the circulation pumps are in constant operation and will thus become an increasing part of the total used power when the load is decreasing. Apart from this, the efficiency of the heat pump should be increasing but it is not. Table 6.3 shows the ratio between the coefficients of performance for the two operation principles. From the table it is obvious that the variable-speed control is less efficient than the on/off control.

Table 6.3 Comparison between variable-speed control and on/off control.

Operating point t_{bi} / t_{wi} (°C)	5 / 26.5	0 / 27.7	0 / 27.7	5 / 42.7	0 / 44.0	0 / 44.0
Load (%)	63	75	63	63	75	63
variable-speed control / on/off control						
COP_{hp}**(vsd)/** COP_{hp}**(on/off)**	1.04	0.99	1.01	0.97	0.96	0.95
COP_{hps}**(vsd)/** COP_{hps}**(on/off)**	1.00	0.97	0.98	0.94	0.95	0.93

Table 6.4 Expected increase in COP_{hp}, estimated from measurements.

Operating point t_{bi} / t_{wi} (°C)	5 / 26.5	0 / 27.7	0 / 27.7	5 / 42.7	0 / 44.0	0 / 44.0
Load (%)	63	75	63	63	75	63
Increase in COP_{hp} **due to raised evaporation temperature (%)**						
	3	3	3	2	1	2
Increase in COP_{hp} **due to lower condensation temperature (%)**						
	6	3	5	5	4	5
Total increase (%)						
	9	6	8	8	5	7

The results are even more surprising when the measurements within the refrigeration circuit were analysed. The pressure and temperature measurements imply that the efficiency should increase. Assuming that an increase in evaporation temperature, t_2, will increase the COP_{hp} with approximately 2 %/K and that a decrease in condensation temperature, t_1, also increases the COP_{hp} with close to 2 %/K, will give the results of Table 6.4. From this calculation, the COP_{hp} is expected to increase by 5-9 % when the load is decreased. Instead we get much smaller increases or even decreases in COP_{hp}. How can this be?

Possible reasons are:

- Large evaporator superheat
- Losses in the frequency converter
- Losses within the compressor

The superheat is not the reason for the losses since in this case the measured superheat was lower for part load operation than for operation at full load.

To be able to determine whether the frequency converter is the reason for the losses, its efficiency was measured for different frequencies and operating points. Figure 6.6 shows the results from these measurements, and they indicate that the efficiency is dependent on the compressor frequency but not on the operating point. The efficiency changes very little between the different frequencies, from 95.5 % at 30 Hz to 98 % at 75 Hz. However, it can be concluded that the frequency converter contributes to the loss in coefficient of performance. Recalculating the values in Table 6.3 gives the results in Table 6.5. Comparing these values with the expected increase in Table 6.4 reveals that the losses in the frequency converter are not the only reason for the decrease in COP_{hp}. There is still up to 7 % missing. This remains to be explained by increased losses within the compressor due to the variable-speed operation.

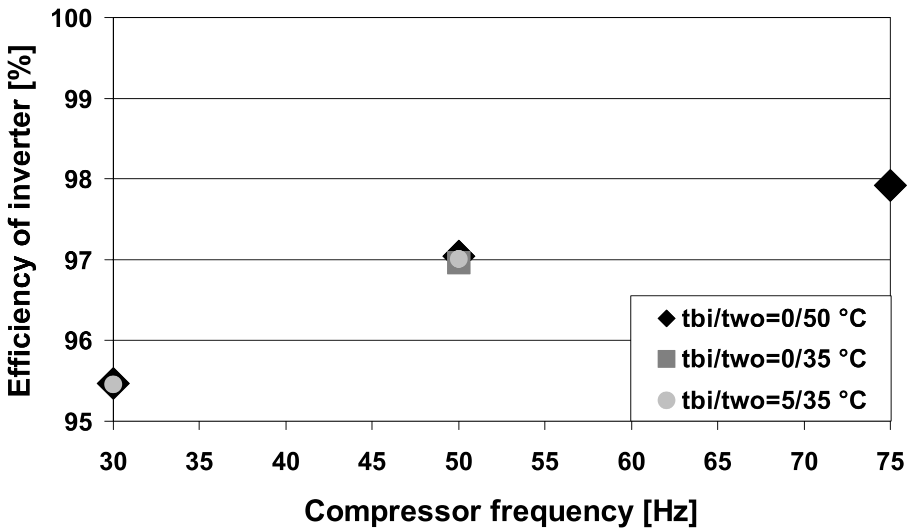


Figure 6.6 The efficiency of the frequency converter depends on the compressor frequency but not on the operating point.

Table 6.5 Quotient between heat pump coefficients of performance with the frequency converter excluded.

Operating point t_{bi} / t_{wi} (°C)	5 / 26.5	0 / 27.7	0 / 27.7	5 / 42.7	0 / 44.0	0 / 44.0
Load (%)	63	75	63	63	75	63
variable-speed control / on/off control						
***COP_{hp}*(vsd)/ *COP_{hp}*(on/off)**	1.09	1.03	1.06	1.01	1.00	1.00

The losses in the compressor appear in two ways, heat losses and non-isentropic compression. With the measurement set-up used for these experiments, and the fact that the compressor is hermetic, the real isentropic efficiency cannot be determined. The efficiency which is possible to calculate is a product of several efficiencies as given by Eq. 6.1.

$$\eta_{is;meas} = \varphi \cdot \eta_{is} \cdot \eta_{mc} \cdot \eta_{mt} \cdot \eta_{m,el} \qquad \text{Eq. 6.1}$$

This total efficiency is calculated as the quotient between the isentropic enthalpy difference and the real enthalpy difference over the compressor (for designations see Figure 4.2).

$$\eta_{is;meas} = \frac{\Delta h_{is}}{\Delta h} = \frac{h_{3is} - h_4}{h_3 - h_4} \qquad \text{Eq. 6.2}$$

The diagram in Figure 6.7 shows the values of this efficiency for full and part load at different operating points. At part load the efficiency drops 2-3 % compared to the full load operation. The evaporation efficiency, φ, and the efficiency of the mechanical transmission between motor and compressor, η_{mt}, should be unaffected, or increase, when the compressor frequency is reduced. According to Tassou and Qureshi[74] and Scalabrin and Bianco[62], the isentropic efficiency, η_{is}, increases with reduced frequency for a reciprocating compressor. This leaves the mechanical efficiency, η_{mc}, of the compressor and the efficiency of the electric motor, $\eta_{m,el}$, to explain the decrease in efficiency. The mechanical efficiency of the compressor should increase with decreasing compressor speed and thus remains only the efficiency of the electric motor to explain the decrease in efficiency. A decrease in efficiency of the electric motor when the compressor speed is reduced has earlier been reported by Scalabrin and Bianco[61] and Riegger[57]. The remaining losses not explained by the discussion above should be referred to heat losses from the compressor shell.

In conclusion it can be stated that instead of an expected increase in COP_{hp} by approximately 10 % when applying variable-speed control to the compressor at part load, the tests showed only very small increases or even decreases by up to 5 % due to losses in the frequency converter and probably in the electric motor of the compressor.

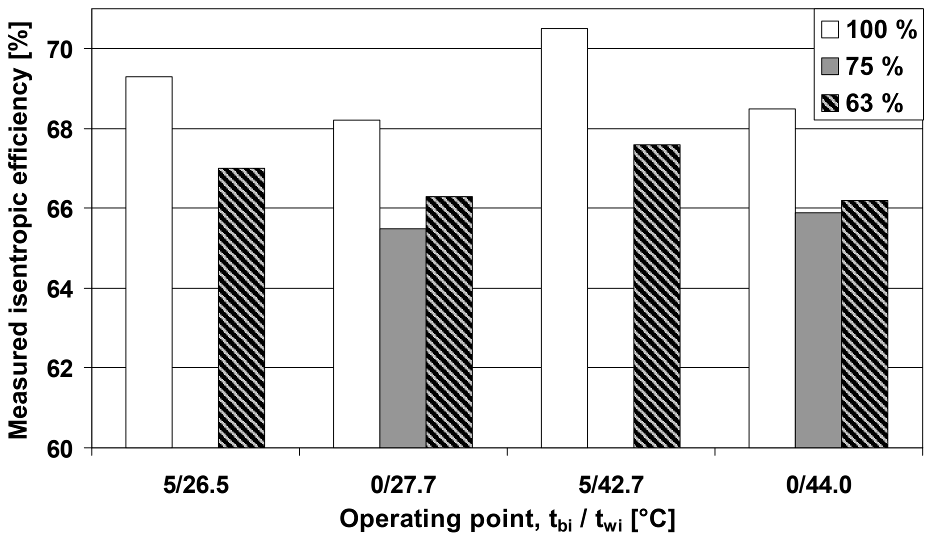


Figure 6.7 The efficiency of the compressor for different loads and operating points. Tests were not performed at the part load 75 % for the operating points 5/26.5 and 5/42.7.

6.1.2.1 Influence from circulation pumps

The drive powers to the circulation pumps are quite low compared to the drive power of the compressor when the heat pump is operating at full load. When operating at part load the drive power to the pumps will constitute an increasing part of the total drive power. This increased influence is indicated by the quotient between COP_{hps} and COP_{hp} as shown in Figure 6.8 and Figure 6.9. For the on/off control the decrease is very small since the brine pump was switched off as the compressor was shut down. The decrease was then only due to the heating water pump. For the variable-speed controlled heat pump the decrease due to the circulation pumps is more evident since the compressor as well as the brine and heating water pumps operate continuously and at constant speed. By adjusting the speed of the circulation pumps and thus the brine and heating water flows it is possible to reduce the decrease in coefficient of performance. An optimal combination of compressor speed and pump speeds is possible to find as shown in chapter 5.

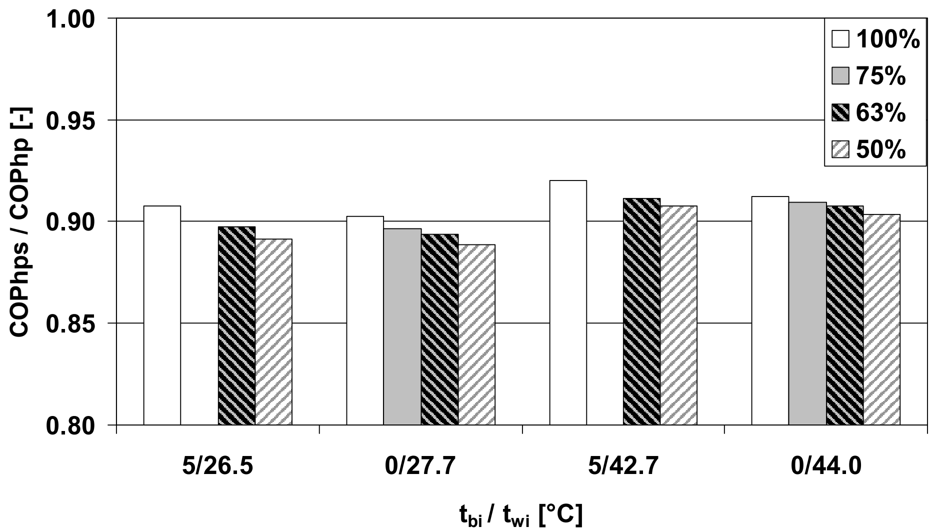


Figure 6.8 The decrease in coefficient of performance due to the circulation pumps when the compressor is operating on and off. Tests were not performed at the part load 75 % for the operating points 5/26.5 and 5/42.7.

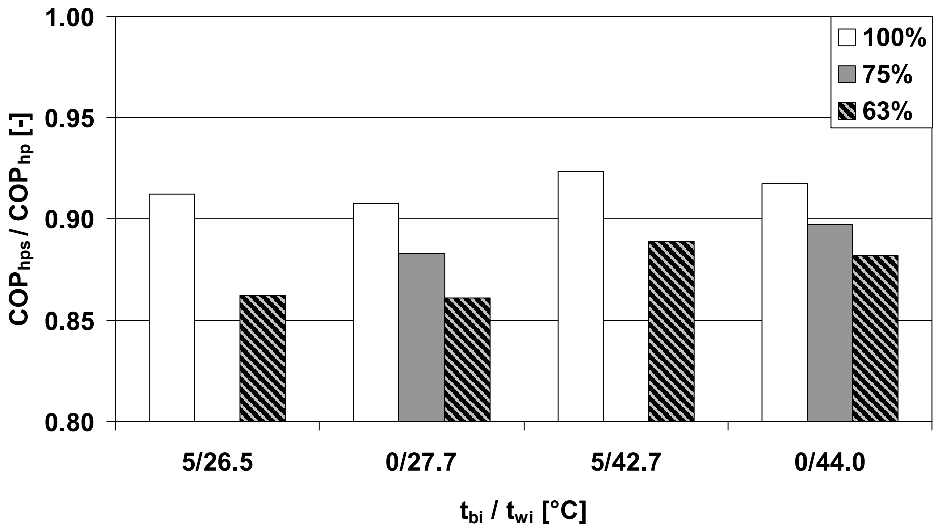


Figure 6.9 The decrease in coefficient of performance due to the circulation pumps when the compressor is in variable speed operation. Tests were not performed at the part load 75 % for the operating points 5/26.5 and 5/42.7.

Even at full load the influence from the circulation pumps decreased the coefficient of performance with about 8-9 %. Is it possible to decrease this influence by using more efficient pumps? In order to answer this question, the efficiency of the circulation pumps installed in the heat pump described in Appendix A was measured. The pumps are wet motor pumps typically used in brine-to-water heat pumps with three different speeds that are set manually. The measurements were performed at speed 2. The efficiency of the heating water pump proved to be 8-9 % (flows between 0.7 and 1.0 m^3/h) and the brine pump showed an efficiency of 15-16 % (flows between 1.2 and 1.9 m^3/h). These results are typical for these kinds of pumps. More efficient pumps are available on the market, also pumps with variable-speed control. If the efficiency of both pumps could be increased by 50 % (from 8 to 12 % for the heating water pump) the decrease in COP_{hps} would be 3-4 % lower at full load.

6.2 Seasonal performance

In the previous section, variable-speed operation of a compressor did not show any increase in efficiency compared to on/off operation. To further analyse the possibilities with variable-speed operation the heat pump model described in chapter 3 was used for calculations of the seasonal performance of both a variable-speed compressor and a single-speed compressor. Thus it will be possible to see whether, despite the lower efficiency at specific operating points, the seasonal performance factor may be higher due to the reduced need for supplementary heat when using variable-speed capacity control.

6.2.1 Calculation model

For this analysis a climate model, a house model and a model of the heating system were needed. The duration curve for the outdoor air temperature was modelled according to Fehrm and Hallén[24]:

$$t_{out}(h) = (h - 4380) \cdot (3.9 - 0.086 \cdot t_{mean}) \cdot 0.001 +$$
$$t_{mean} + \left[\frac{h \cdot (1 + \frac{8 - t_{mean}}{586})}{8300} \right]^{38} - \left[\frac{1550}{700 + h} \right]^{3} + \qquad \text{Eq. 6.3}$$
$$1.5 \cdot \left[\frac{t_{mean}}{8} \cdot \frac{1200}{500 + h} \right]^{2} \cdot \cos\left(\frac{900 - h}{585} \right)$$

Here, h, stands for the number of hours per year during which the temperature t_{out} or lower is present and t_{mean} designates the mean outdoor temperature for the whole year. The heating system was modelled according to Fehrm and Hallén[24] where the heating output from the radiators is expressed as:

$$\dot{Q}_{rad} = C_2 \cdot \theta_{rad}^{n} \qquad \text{Eq. 6.4}$$

Here, n, is a constant which is specific for the radiator used (in this investigation set to 1.25) and θ_{rad} represents the logarithmic mean temperature difference between the radiators and the surrounding air. The heat needed to keep the indoor air temperature at the desired level was expressed as:

$$\dot{Q}_{house} = F \cdot (t_{in} - \Delta t_0 - t_{out}) \qquad \text{Eq. 6.5}$$

where Δt_0 represents the equivalent temperature rise due to internal heat gain (set to 3 K). The heating capacity of the heat pump system including supplementary heat can be expressed as:

$$\dot{Q}_{hs} = C_3 \cdot (t_f - t_r) \qquad \text{Eq. 6.6}$$

Using Eq. 6.3 to Eq. 6.5 together with the fact that

$$\frac{\dot{Q}_{house}}{\dot{Q}_{house;DOT}} = \frac{\dot{Q}_{rad}}{\dot{Q}_{rad;DOT}} = \frac{\dot{Q}_{hs}}{\dot{Q}_{hs;DOT}} \qquad \text{Eq. 6.7}$$

makes it possible to express t_f and t_r as

$$t_f = \frac{e^{d} \cdot (a \cdot c + t_{in}) - t_{in}}{e^{d} - 1} \qquad \text{Eq. 6.8}$$

$$t_r = t_f - a \cdot c$$ Eq. 6.9

The coefficients a-d are given by the following expressions:

$$a = \frac{t_{in} - \Delta t_0 - t_{out}}{t_{in} - \Delta t_0 - t_{out;DOT}}$$ Eq. 6.10

$$b = \ln\left(\frac{t_{f;DOT} - t_{in}}{t_{r;DOT} - t_{in}}\right)$$ Eq. 6.11

$$c = t_{f;DOT} - t_{r;DOT}$$ Eq. 6.12

$$d = a^{\left(1-\frac{1}{n}\right)} \cdot b$$ Eq. 6.13

So, by specifying *DOT (Design Outdoor Temperature)*, $t_{f;DOT}$, $t_{r;DOT}$, t_{mean}, F, t_{in} and Δt_0 it is possible to determine how the temperatures in the radiator system vary over the year. For this particular investigation input data according to Table 6.6 were used and the variation of the temperatures in the radiator system is shown in Figure 6.10.

Table 6.6 Input data for the calculations of the seasonal performance factor. The value of F is chosen such that the total heat demand is 17000 kWh/year (tap water heating excluded).

DOT	- 20 °C
F	169 W/K
t_{mean}	+ 6 °C
$t_{wo;DOT}$	55 °C
$t_{wi;DOT}$	45 °C
Δt_0	3 K
t_{in}	20 °C

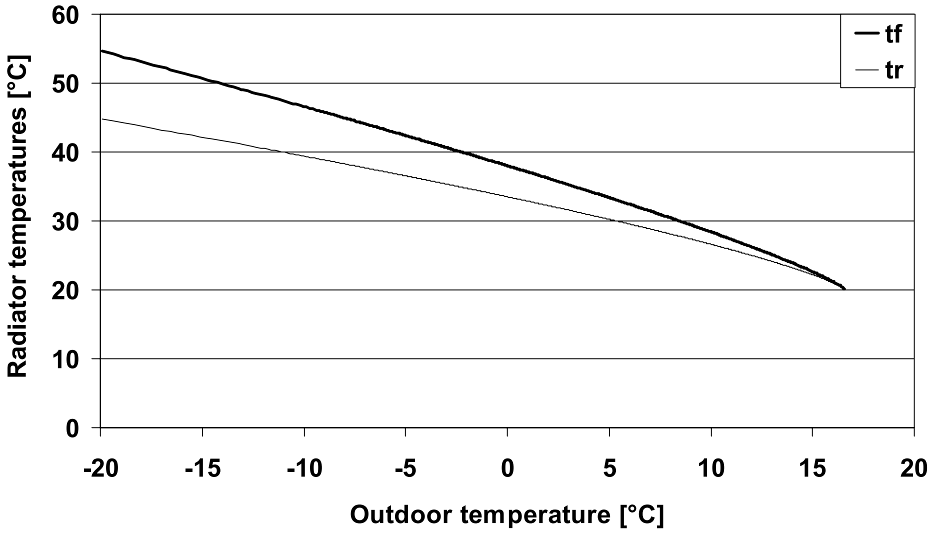


Figure 6.10 Radiator temperatures as a function of the outdoor temperature for the chosen 55/45 heating system.

The calculations of the radiator temperatures also determined the flow rate of the heating water in the radiator system, which in this case was lower than the internal flow in the heat pump. Thus the incoming temperature to the heat pump is not the same as the return temperature from the radiator system. By establishing an energy balance over the connecting point marked in Figure 6.11 it is possible to determine the incoming temperature to the heat pump.

$$\left(\dot{V}_f - \dot{V}_{hs}\right)\cdot \rho(t = t_f)\cdot t_f + \dot{V}_{hs}\cdot \rho(t = t_r)\cdot t_r = \dot{V}_{wi}\cdot \rho(t = t_{wi})\cdot t_{wi} \quad \text{Eq. 6.14}$$

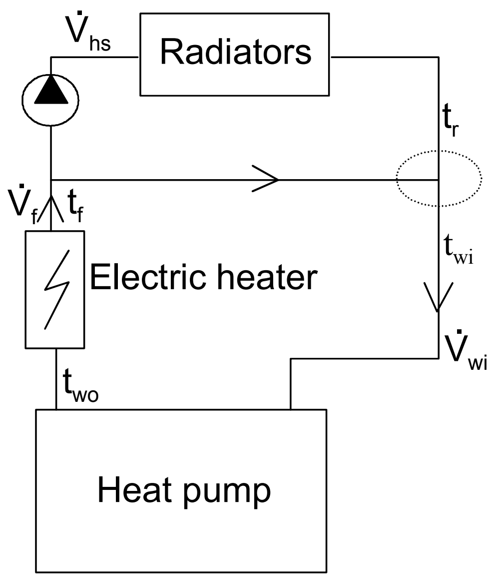


Figure 6.11 The connection of the heat pump to the heating system. The designations for the different temperatures in the system are also marked.

The calculations for the seasonal performance factor were divided into three parts:

1. The outdoor temperature is below the balance temperature and the supplementary heat must be used.
2. The heat pump operates continuously and adjusts its heating capacity to the heat load.
3. The heat load is lower than the heating capacity of the heat pump at the lowest possible frequency. The heat pump must be operated on/off.

For part 1 the supply and return temperatures, the volume flow in the heating system and the heat load were given as inputs. The drive powers for the compressor, pumps, inverter and the electric heater were calculated as well as the outgoing heating water temperature from the heat pump, t_{wo}. The difference between the heat load, $\dot{Q}_{house}$, and the heating capacity of the heat pump constitutes the power that must be supplied by the electric heater. The efficiency of the electric heater was set to 95 %. For part 2 the same inputs are given as for part 1 except for the supply temperature that is not given. In part 3, using the same inputs as part 2, the powers used by compressor, brine pump and inverter were reduced from their steady-state values in proportion to the relative operating time R_{hp}, i.e.:

$$\dot{W}_{pl} = \dot{W}_{ss} \cdot R_{hp} \qquad \text{Eq. 6.15}$$

$$R_{hp} = \frac{\dot{Q}_{house}}{\dot{Q}_{1;ss}} \qquad \text{Eq. 6.16}$$

The heating water pump was set to operate continuously.

For the calculations of an on/off controlled heat pump, part 2 does not apply.

Calculations were made for three different cases:

1. Single-speed compressor with efficiencies according to chapter 3.
2. Variable-speed compressor with efficiencies according to chapter 3.
3. Variable-speed compressor with efficiencies as at 50 Hz.

Each of these cases was analysed for three different capacities of the heat pump corresponding to 100, 75 and 50 % (A, B and C) power coverage at the design outdoor temperature (*DOT*). This coverage was determined at the compressor frequency 50 Hz for all three cases. The swept volume of the compressor adjusted the size of the heat pump and was calculated for case 2 and then kept at the same value for case 1 and 3 as well. The heating water and brine flows were chosen such that the temperature difference between the incoming and outgoing heat transfer media was 7.3 K for the condenser and 3.3 K for the evaporator when t_{wo} = 35 °C and t_{bi} = 0 °C. These values were chosen according to tests of the heat pump described in Appendix A. The size of the compressor and the heat transfer media flows for the different compressor sizes are compiled in Table 6.7. The sizes of the circulation pumps were scaled to the flow as described in section 3.1.5.

Table 6.7 Swept volume of the compressor and the heating water and brine flows for the different sizes of heat pumps.

Size	V_{sw} (cm^3)	$\dot{V}_b$ (m^3/h)	$\dot{V}_w$ (m^3/h)
A (100 %)	5.638	1.76	0.96
B (75 %)	4.178	1.34	0.72
C (50 %)	2.762	0.89	0.48

6.2.2 Results

The results from the calculations described above are put together in Table 6.8 below. As pointed out in the table the highest SPF_{hs} was achieved for the heat pump with efficiencies independent of compressor speed, sized for 50 % power coverage at *DOT* (case 3, size C).

Table 6.8 The electric drive energy (in kWh) and the seasonal performance factor for the three different cases described in the text.

Case	**1 (on/off)**			**2 (vsd, test)**			**3 (vsd, theor.)**		
Size	*A*	*B*	*C*	*A*	*B*	*C*	*A*	*B*	*C*
W_{hs}	5775	5400	5579	6300	5695	5302	6007	5504	5051
W_{comp}	4943	4693	4175	4970	4715	4426	4734	4554	4346
W_{bp}	401	334	238	668	510	307	622	488	298
W_{wp}	431	265	134	431	265	134	431	265	134
W_{inv}	-	-	-	230	203	156	220	198	154
W_{heater}	0	107	1032	0	1	279	0	0	119
SPF_{hs}	2.97	3.17	3.07	2.72	2.99	3.22	2.85	3.11	3.38
SPF_{hps}	2.97	3.22	3.55	2.72	2.99	3.35	2.85	3.11	3.44
SPF_{hp}	3.38	3.57	3.84	3.36	3.56	3.77	3.52	3.70	3.87

Cases 1 and 3 were compared with each other in order to evaluate the difference in energy saving potential between heat pumps with variable-speed compressors and heat pumps with single-speed compressors. Starting at system level, and comparing heat pumps of equal size, the variable-speed heat pump has a SPF_{hs} that is from 4 % lower to 10 % higher than the single-speed heat pump. The increase relates to the smallest size and is due to the reduced need for supplementary heat. For the largest size, the variable-speed heat pump system has a lower efficiency since the opportunity for operation at low speed is used only to a small extent and does not outbalance the extra energy used by the brine pump and the inverter. Worth noting here is that the power losses in the frequency converter are of the same magnitude as the power used by the heating water pump. Thus the benefit of variable-speed operation must outbalance one “extra” circulation pump compared to the on/off control. For SPF_{hp} the variable-speed heat pump is always more efficient, but for the smallest size the increase is negligible. This reduced benefit can be explained by the pressure ratio between the high and low-pressure sides. Figure 6.12 to Figure 6.14 illustrates the pressure ratios for the three different sizes and cases in Table 6.8. The figures show that for size A the pressure ratio is always higher for the single-speed compressor than for the variable-speed compressor, and thus the variable-speed compressor has a higher SPF_{hp}. For size C, the pressure ratio for the variable-speed compressor is higher than the pressure ratio for the single-speed compressor during approximately 20 % of the year and thus the increase in SPF_{hp} will be lower. The reason why the pressure ratios intersect is that the increased compressor speed for the variable-speed unit increases the pressure ratio above values possible for the single-speed unit. This will make the efficiency worse when looking at the heat pump unit only, but when considering the entire system the variable-speed unit is more efficient since less supplementary heat is needed.

If the best alternatives of each control-strategy are compared, the variable-speed heat pump is 6 % better. Comparing heat pumps of size C, the variable-speed heat pump is 10 % more efficient than the single-speed unit. In this context it should be mentioned that the efficiency of the compressor was considered to be independent of the rotational speed. For efficient variable-speed compressors the isentropic efficiency increases with decreasing compressor speed and thus the total efficiency will increase even further. Another issue is that single-speed circulation pumps were used. If variable-speed pumps were used their speed and power use could be reduced at low compressor capacities.

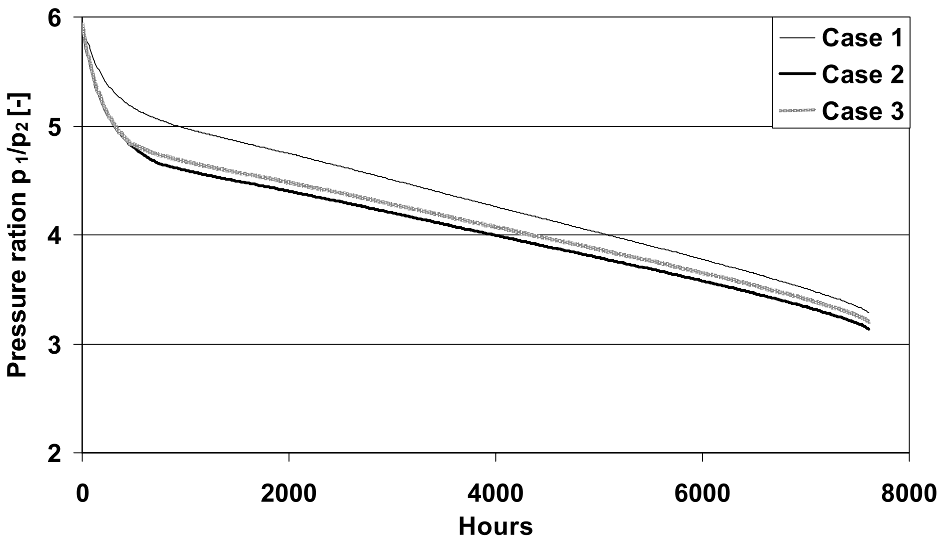


Figure 6.12 The ratio between condensation and evaporation pressure during the heating season for heat pump size A.

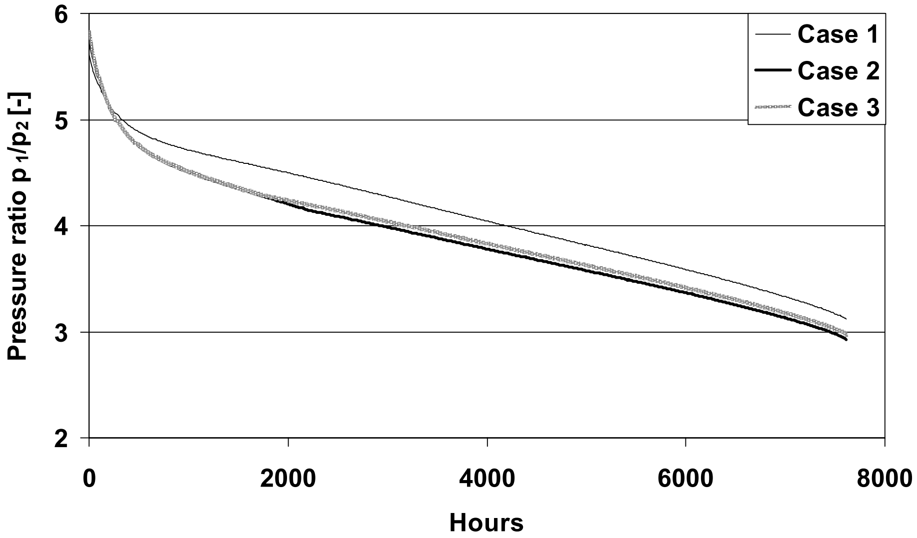


Figure 6.13 The ratio between condensation and evaporation pressure during the heating season for heat pump size B.

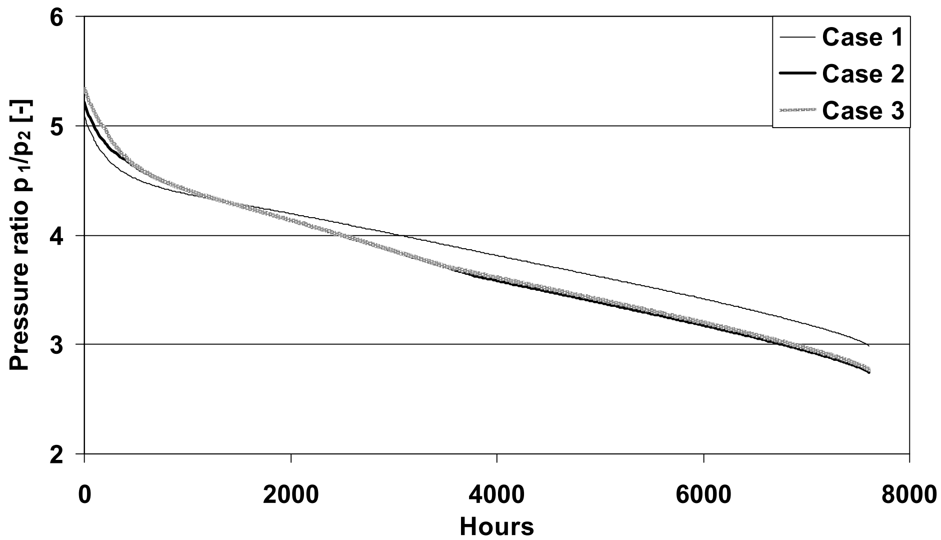


Figure 6.14 The ratio between condensation and evaporation pressure during the heating season for heat pump size C.

6.3 Conclusions

The results show that variable-speed capacity control of compressors has the potential of being more energy efficient than conventional on/off control also for brine-to-water heat pumps. However, this requires that compressors suitable for brine-to-water heat pumps are designed for variable-speed operation. Applying variable-speed control to a conventional compressor proved to decrease the efficiency.

The major reason why the variable-speed compressor proved to be more efficient was that for the same compressor size less supplementary heat was needed.

Furthermore, the increase in energy efficiency is low when using electronically controlled expansion valves instead of conventional thermostatic valves. It seems more promising to apply new more efficient circulation pumps. Improvement in pump efficiency should be quite easy to achieve since the pumps normally used have efficiencies lower than 20 %.

7 Performance indication, control and supervision

The measuring system described in section 4.2 can be used for general information, performance indication, control and operational aids, such as fault detection and diagnosis and safety functions. In this chapter, demands on the sensors used will be discussed with reference to these various applications. The last section describes the possibilities for fault detection and diagnosis with the proposed IRM concept.

7.1 Sensors for performance indication

This measurement system, where the *COP* is inferred from refrigerant states, has earlier been described and evaluated by Fahlén[15, 22]. The method was evaluated by laboratory tests on heat pumps where the results from this method was compared with the results from a test method described in the Swedish standard SS 2095 which in most aspects is similar to the European standard EN 255. Both these standards prescribe measurement of temperatures and flows on the secondary side in order to measure the performance of heat pumps during steady-state conditions. When there was sufficient space for a correct installation of the sensors, the motor coefficient of performance (COP_{hp}) could be determined within ±5 % from the COP_{hp} measured according to the Swedish standard, SS 2095. Larger deviations than ±5 % derived from one or more of the following causes (Fahlén and Johansson[22]):

- It was not possible to make a proper installation of the sensors due to limited space.
- A proper temperature measurement was not possible due to large temperature gradients along the copper pipes.
- The pressure sensors were mounted such that the readings included large pressure drops between the heat exchangers and the sensor.
- The evaporator superheat was too small and thus liquid refrigerant entered the compressor
- The sub cooling was too low to make all refrigerant condense in the condenser.
- The heat exchange between the compressor and the surrounding air differed from normal conditions
- The temperature sensor on the compressor discharge pipe was badly insulated. Since the calculation of the *COP* largely depends on the discharge temperature it is vital to measure this temperature correctly. This is quite difficult because of the large temperature difference between the discharge pipe temperature and the temperature of the surrounding air.

In order to determine with what accuracy each measurement must be performed in order to reach a certain total uncertainty of measurement the uncertainty propagation for this measurement method must be derived. This was made according to the Nordtest method NT-VVS 116[48]. Thus deriving Eq. 4.4 gave the following expression for the uncertainty of the *COP* measurement:

$$\left|\frac{\Delta COP}{COP}\right| = \left|B_1 \cdot \frac{\Delta p_1}{p_1}\right| + \left|B_2 \cdot \frac{\Delta p_2}{p_2}\right| + \left|B_3 \cdot \Delta t_3\right| + \left|B_4 \cdot \Delta t_4\right| + \left|B_5 \cdot \Delta t_5\right| + \left|B_6 \cdot \frac{\Delta f}{f}\right| \quad \text{Eq. 7.1}$$

where:

$$B_1 = \frac{-(h_4 - h_5) \cdot p_1}{(h_3 - h_4) \cdot (h_3 - h_5)} \cdot \frac{\partial h_3}{\partial p_1} \, [-] \quad \text{Eq. 7.2}$$

$$B_2 = \frac{p_2}{(h_3 - h_4)} \cdot \frac{\partial h_4}{\partial p_2} \, [-] \quad \text{Eq. 7.3}$$

$$B_3 = \frac{-(h_4 - h_5)}{(h_3 - h_4) \cdot (h_3 - h_5)} \cdot \frac{\partial h_3}{\partial t_3} \, [K^{-1}] \quad \text{Eq. 7.4}$$

$$B_4 = \frac{1}{(h_3 - h_4)} \cdot \frac{\partial h_4}{\partial t_4} \, [K^{-1}] \quad \text{Eq. 7.5}$$

$$B_5 = \frac{-1}{(h_3 - h_5)} \cdot \frac{\partial h_5}{\partial t_5} \, [K^{-1}] \quad \text{Eq. 7.6}$$

$$B_6 = -\frac{-f}{1-f} \, [-] \quad \text{Eq. 7.7}$$

The installation effects tend to contribute to an overestimation of the *COP*. On the other hand the uncertainty propagation in Eq. 7.1 estimates the maximum uncertainty and is thus quite conservative. The derivatives in Eq. 7.2 – Eq. 7.7 may be evaluated using property tables etc. In NT-VVS 116[48] the derivatives are calculated from equations of state derived by Cleland[12, 13], where the equations of state are expressed mainly as polynomials in order to be easy to implement and use in dynamic simulations. In NT-VVS 116[48] the derivatives have been calculated for the refrigerants R12, R22, R114, R502 and R717. Since R717 is the only of these refrigerants that may be used in Sweden today, the equations were derived for R407C. This refrigerant is widely used in domestic heat pumps in Sweden and was also used in the experimental heat pump described in Appendix A. The equations of state are given in Appendix B together with the expressions for the derivatives.

The allowed uncertainty of measurement for each of the measured parameters in Eq. 7.1 can now be determined by means of the equations in Appendix B. In Table 7.1 below, the total uncertainty of *COP* is set to three different levels, 5, 10 and 15 %. These three levels are stated in a Nordtest method directed towards field-testing of heat pumps, NT-VVS 115[47]. If the total uncertainty is divided equally on each measurement the following uncertainties of measurement apply (valid for the operating conditions according to Table 7.2):

Table 7.1 The allowed uncertainty of measurement for the individual parameters when the uncertainty contribution is equally divided (2.5 %, 1.67 % and 0.83 %).

	Operating point								
Measurand	**1**			**2**			**3**		
ΔCOP/COP (%)	**15**	**10**	**5**	**15**	**10**	**5**	**15**	**10**	**5**
$\Delta p_1/p_1$ (%)	18.4	12.3	6.1	43.1	28.8	14.3	25.8	17.2	8.6
$\Delta p_2/p_2$ (%)	23.4	15.7	7.8	33.0	22.0	10.9	20.4	13.6	6.8
Δt_3 (K)	1.7	1.2	0.6	2.5	1.7	0.8	1.9	1.2	0.6
Δt_4 (K)	1.6	1.1	0.6	2.3	1.5	0.7	1.8	1.2	0.6
Δt_5 (K)	3.4	2.3	1.1	3.1	2.1	1.0	3.0	2.0	1.0
$\Delta f/f$ (%)	47.5	31.7	15.8	47.5	31.7	15.8	47.5	31.7	15.8

Table 7.2 The operating conditions used for the calculations in Table 7.1 (Karlsson and Fahlén[40]). They correspond to the temperatures 0/35, 0/50 and 10/50 respectively (t_{bi}/t_{wo}).

Measurand	Operating point 1	Operating point 2	Operating point 3
p_1 (bar)	14.92	21.44	21.19
p_2 (bar)	3.93	3.86	5.22
t_3 (˚C)	71.5	93.6	85.3
t_4 (˚C)	-0.2	0.1	9.6
t_5 (˚C)	30.7	44.5	44.9
f (%)	5.0	5.0	5.0

In NT-VVS 116[48], which prescribes this measurement method, the total uncertainty of measurement should be a maximum of 15 %. Will that be possible to achieve for the operating conditions given and the refrigerant R407C? In order to answer that, typical uncertainties of measurement for the different parameters must be known. Fahlén and Johansson[22] give ranges of possible uncertainties for each parameter see Table 7.3.

Table 7.3 Typical uncertainties of measurement according to Fahlén and Johansson[22].

Measurand	Typical uncertainty of measurement
p_1	±1-5 %
p_2	±1-5 %
t_3	-1 to –5 K
t_4	± 1 K
t_5	0 to –1 K

The loss factor, f, must be known in advance or be calculated based on for example the surface temperature of the compressor shell. Only few investigations report on typical values for the loss factor. According to Fahlén and Johansson[22] and Fahlén[20], the loss factor of single-speed hermetic compressors can be expected to be in the range of 4-10 %. Measurements on a hermetic non-insulated piston compressor at SP showed a loss factor of 4-6 % (Karlsson and Fahlén[40]).

Because of the lack of publicly available information regarding its magnitude, the loss factor together with the discharge pipe temperature, t_3, are considered to be the most difficult parameters to measure or estimate. The loss factor may be more difficult to estimate for capacity controlled units. Since its magnitude depends on the temperature difference between the compressor shell and the surrounding air, and not on the compressor frequency, the loss factor, if expressed in percent, will increase with decreasing compressor speeds.

One way to increase the allowed uncertainty of measurement for t_3 and f is to tighten the demands on one or more of the other measurands. The most obvious measurand to choose are the temperature t_5 and the pressures p_1 and p_2 since the current demands are easy to achieve. The new demands on t_5, p_1 and p_2, as well as the new permissible uncertainty of t_3, are stated in Table 7.4. The demands on the pressure measurements are still quite modest, thus allowing the use of simple pressure sensors. If the demands on the pressure measurements are set to ± 5 % and the other parameters are kept as in Table 7.4 the allowed uncertainty of t_3 will be 5 K and thus equal to the maximum uncertainty according to Table 7.3. So the answer to the question asked in the beginning of this passage is yes; it is possible to determine the coefficient of performance with an uncertainty less than 15 %. Still, however, this requires a proper installation of the sensors.

Table 7.4 Permissible uncertainty of measurement for the individual measurements in order to get a maximum uncertainty for *COP* of 15 or 10 %.

Measurand	**Uncertainty**	
ΔCOP/COP (%)	15	10
$\Delta p_1/p_1$ (%)	10	5.0
$\Delta p_2/p_2$ (%)	10	5.0
Δt_3 (K)	4.2	2.3
Δt_4 (K)	1.6	1.6
Δt_5 (K)	2.0	1.5
Δf/f (%)	50	35

Regarding the other two uncertainty levels it should be possible to reach the 10 % level if the uncertainties in Table 7.4 apply. The 5 % level, however, will be hard to guarantee mainly because of the demands on the temperature measurements. Even if the uncertainty of the loss factor is as low as 5 %, the uncertainties of the temperatures must be about 1 K. This is hard to achieve if surface mounted sensors are used. It may, however, be possible if the sensors are mounted in thermometer wells.

7.1.1 Installation errors in temperature measurements

To illustrate the importance of a proper installation of the temperature sensors an example will be given (Fahlén[16]).

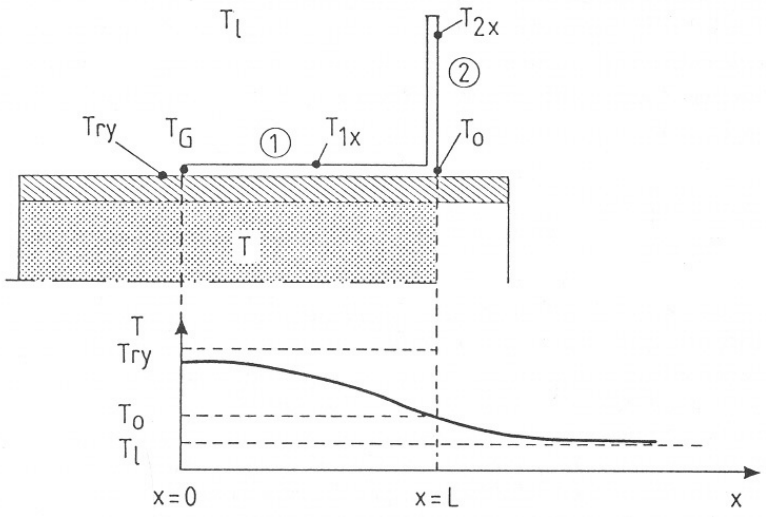


Figure 7.1 The figure shows the temperature profile along a surface mounted temperature sensor (Fahlén[16]).

The measurement error when measuring surface temperatures may be expressed as (Fahlén[16]):

$$\Delta t = t_G - t_{ry} = \frac{t_l - t_{ry}}{(\frac{R_2}{R_1})^{1/2} \cdot \sinh(K_1 \cdot L_1) + \cosh(K_1 \cdot L_1)} \qquad \text{Eq. 7.8}$$

$$R_i = \frac{1}{U_i \cdot A_{iL}} \quad i = 1,2 \qquad \text{Eq. 7.9}$$

$$K_1 = \sqrt{\frac{L_1}{\lambda_1 \cdot A_1 \cdot R_1}} \qquad \text{Eq. 7.10}$$

The subscripts 1 and 2 designate the sensor and the connecting wire respectively as indicated in Figure 7.1. Assuming the sensor to be a thermocouple of type T (copper-constantan) with the sensor length $L_1 = 0.1$ m and the following data:

$\lambda_1 = \lambda_2 = 380$ W/m/K (the heat conductance in the constantan thread is neglected)
$A_1 = 0.52\ mm^2$
$R_1 = 3.5$ K/W
$R_2 = 2$ K/W

Eq. 7.8 yields the following result:

$$\Delta t = \frac{t_l - t_{ry}}{2.95} \qquad \text{Eq. 7.11}$$

For a pipe temperature of 70˚C and a surrounding air temperature of 20˚C the difference between the sensor temperature and the pipe temperature in this example would be 17 K! This very large fault depends on the fact that the sensor in this example had a fairly thick copper conductor, was uninsulated and applied to the pipe without thermally conductive paste. If a paste is used and the sensor is properly insulated, the temperature difference between the pipe and the sensor can be expressed as (Fahlén and Johansson[22]):

$$\Delta t = 0.075 \cdot (t_l - t_{ry}) \qquad \text{Eq. 7.12}$$

Using this expression for the example above gives a temperature difference between sensor and pipe of 3.8 K. This is 13 K lower than for the case with an un-insulated sensor and this illustrates the importance of a proper installation of the temperature sensors. In this example it was assumed that the pipe surface has the same temperature as the medium inside the pipe. To obtain this it is important to insulate the entire pipe a long distance before and after the point were the sensor is mounted.

7.1.2 Installation errors in pressure measurements

Regarding the pressure sensors the errors due to installation effects should be very small and the main part of the uncertainty of measurement should be due to the sensor itself. The installation errors originate from two factors, pressure drop in the pipes and dynamic pressure in relation to static pressure. The pressure drop in the pipes may become important if there is a long distance between the compressor outlet/inlet and the sensor. The dynamic pressure relates to the fluid velocity and thus becomes significant at high velocities. An estimate of the two will be given below. A third installation effect that may cause errors in measurements occur if the pressure sensor on the high-pressure side or its connecting line is mounted at the bottom of the pipe. The refrigerant may condense and fill the sensor and the connecting pipe with liquid refrigerant. This refrigerant column will influence the pressure measurement, but normally this influence will be low and can be neglected. In order to avoid this, the pressure sensors should be attached on the upper side of the pipe.

Incompressible flow was assumed to estimate the pressure drop and thus the Bernoulli equation for flow in pipes could be used. The pipes were considered to be smooth. For the high-pressure side the maximum influence from pressure drop is approximately 0.1 % per meter pipe between the compressor and the pressure transducer. For the low-pressure side the maximum influence is approximately 1 % per meter pipe between the pressure transducer and the compressor. The influence from the dynamic pressure is so low that it can be neglected (< 0.1 %).

In conclusion the only source of installation errors worth considering for pressure measurement in a heat pump is the pressure drop in the pipes (at least for the suction pipe) and the error of the sensor itself. For small sized heat pumps for domestic use the distances between sensors and compressor are small and thus the influence from pressure drop in the pipes will be small and only worth considering for the low-pressure side. In the analysis smooth pipes without bends were considered. Also the connections to the pressure sensors were considered to be smooth without burrs. If the pipe between sensor and compressor has many bends or if the connecting pipe to the pressure sensor is poorly constructed the aforementioned errors will increase. The analysis is made based on data from laboratory tests on the heat pump described in Appendix A.

7.1.3 Demand on the measurement of electric power

To determine the heating capacity, the electric power used by the compressor is measured. The error in measurement for the heating capacity may then be expressed as:

$$\frac{\Delta \dot{Q}_1}{\dot{Q}_1} = \sqrt{(\frac{\Delta \dot{W}_{comp}}{\dot{W}_{comp}})^2 + (\frac{\Delta COP_{hp}}{COP_{hp}})^2} \qquad \text{Eq. 7.13}$$

This equation states the demand on the measurement uncertainty of electric power when the demand on the heating capacity measurement is known.

Measuring the electric power can be quite expensive due to the cost of the power transducer. Another technique one may use is to measure one of the parameters in Eq. 7.14 and estimate the others. The electric power used by the compressor is expressed as:

$$\dot{W}_{comp} = \sqrt{3} \cdot U \cdot I \cdot \cos\varphi \qquad \text{Eq. 7.14}$$

Differentiating Eq. 7.14 gives:

$$\frac{\Delta \dot{W}_{comp}}{\dot{W}_{comp}} = \sqrt{(\frac{\Delta U}{U})^2 + (\frac{\Delta I}{I})^2 + (\frac{\Delta \cos\varphi}{\cos\varphi})^2} \qquad \text{Eq. 7.15}$$

The current, I, is a good measure of the used electric power. In order to measure only the current, the uncertainty of estimating the other parameters must be analysed. The voltage, U, is quite stable. An estimate based on measurements on heat pumps at SP is that the voltage varies less than 2 %. The phase angle, $cos\varphi$, was measured by Karlsson and Fahlén[40] and varied from approximately 0.5-0.8 depending on compressor frequency. This is a quite broad range and cannot be used as an estimate. But if the variation of the phase angle with frequency is known, then the variation around each of these points is much smaller. The maximum deviation from the mean value was measured for operation at 30 Hz and it was 0.04 units, which is equal to 7.2 % (Karlsson and Fahlén[40]).

The uncertainty of the heating capacity will be larger than the uncertainty of the *COP*. To give an example the uncertainty of the heating capacity is set to be less than 20 %, with the uncertainty in *COP* less than 15%, the uncertainty in compressor power must be less than 13 % (Eq. 7.13). This value together with the previously stated values for the variation in voltage and phase angle gives (using Eq. 7.15) that the current must be determined with an accuracy of at least 11 %.

7.2 Fault detection and diagnosis (FDD) using IRM

When discussing energy efficiency and how it may be increased for heat pumps it is easy to focus on how to find the best set-point or the fastest control system etc. Another important issue, however, is to find methods for detecting faults and performance degradations at an early stage. This will enhance the energy efficiency and the availability of the heat pump over its life. For heat pump systems with supplementary heat it may take a long time for the user to realise that the heat pump unit is put out of operation due to a malfunction and that the heat is supplied only by the back-up system if no alarm function is activated. More difficult to detect is performance degradation. Then the heat pump operates but with a reduced efficiency. Such a fault will take a long time to detect without an on-line detection system.

Regarding fault detection and diagnosis these terms should be defined. Detection means becoming aware of that something is wrong. The diagnosis part determines where and why the fault occurs. There are several investigations made in this field using different techniques. One thing they have in common is that normal operation must be defined. In an investigation by Rossi and Braun[60] the normal operation was determined by a steady-state model of the heat pump. Measurements during operation were then compared with the model. The residuals, i.e. the differences between the measurements and the model were then used in a statistical rule-based classifier which determined if the operation was normal or not and also determined what fault was present, if any. Other investigations use the process characteristics during transients in order to perform FDD. More about investigations on FDD was presented in chapter 2.

The rest of this section will focus on the possibilities for FDD with the measuring system suggested in section 4.2. The figure below shows the sensors that are normally installed in a heat pump. For a standard heat pump the pressures are not measured with sensors providing an output signal. Instead pressure switches are used as safety functions. The temperature sensors are often thermistors installed on the surface of the pipes. This can also be used for the IRM concept. The idea of the IRM is to use the sensors that are normally used and in that way minimise the number of extra components.

Fault detection and diagnosis can be performed at different levels of accuracy depending on which faults are to be detected and how soon after they have occurred they should be recognised. The first and most basic form of FDD is to see whether some sensors indicate values that the experience says are well outside the normal limits. This can be very coarse and will not detect the faults at a very early stage. On the other hand it is quite simple and easy to use. Some examples of this are shown below, first stating what is wrong and then giving some possible explanations (Fahlén and Johansson[22] and ETM Mätteknik AB[14]):

1. Superheat is outside the normal interval (4-10 °C)

    - Refrigerant leakage
    - Clogged drying filter
    - Malfunctioning expansion valve
    - Too small expansion valve
    - Other restrictions in the refrigerant circuit

2. Subcooling outside the normal interval (1-30 °C)

    - Refrigerant leakage

3. Compressor efficiency outside the normal interval (50 – 75 %)

    - Malfunctioning compressor
    - Leakage between high and low pressure side

4. Difference between condensation temperature and temperature of the outgoing heat transfer medium is outside the normal interval (1-10 °C)

    - Clogged condenser
    - Pressure drop in the refrigerant circuit
    - Too much refrigerant in the system
    - Wrong flow of high-temperature heat transfer medium

5. Difference between evaporation temperature and temperature of the incoming heat transfer medium is outside the normal interval (5-12 °C)

    - Clogged evaporator
    - Evaporator full of ice or frost

6. Carnot efficiency outside the normal interval (40 – 60 %)

    - Malfunctioning compressor
    - Low superheat

7. Unstable conditions

    - Unstable conditions in the heating system
    - Expansion valve stuck in fully open position

Those were some examples making it possible to determine that something is wrong, i.e. detection, and also giving some suggestion to why, diagnosis. This is, as mentioned before, a quite coarse method for FDD. A way of improving the diagnosis part and be able to say exactly which fault that is present and that also makes it possible to sooner detect a fault is to combine the above given indication into a rule-based identifier as described by Rossi and Braun[60]. For example, the rule for refrigerant leakage, which is a fault that should be detected early, looks as:

Fault	t_2	θ_{sup}	t_1	θ_{sub}	t_3	Δt_w	Δt_b
Refrigerant leakage	↓	↑	↓	↓	↑	↓	↓

If the trend for the given temperatures follow this pattern and deviates more than a certain threshold value from their normal values the FDD algorithm will indicate that there is a refrigerant leakage in the heat pump. This rule can be used in the IRM concept with the sensors shown in Figure 7.2. According to the same authors (Rossi and Braun[60]) the following faults can also be detected in the sensor set up but with other rules:

- Compressor valve leakage
- Restriction in the refrigerant pipe after the condenser
- Clogged condenser
- Clogged evaporator

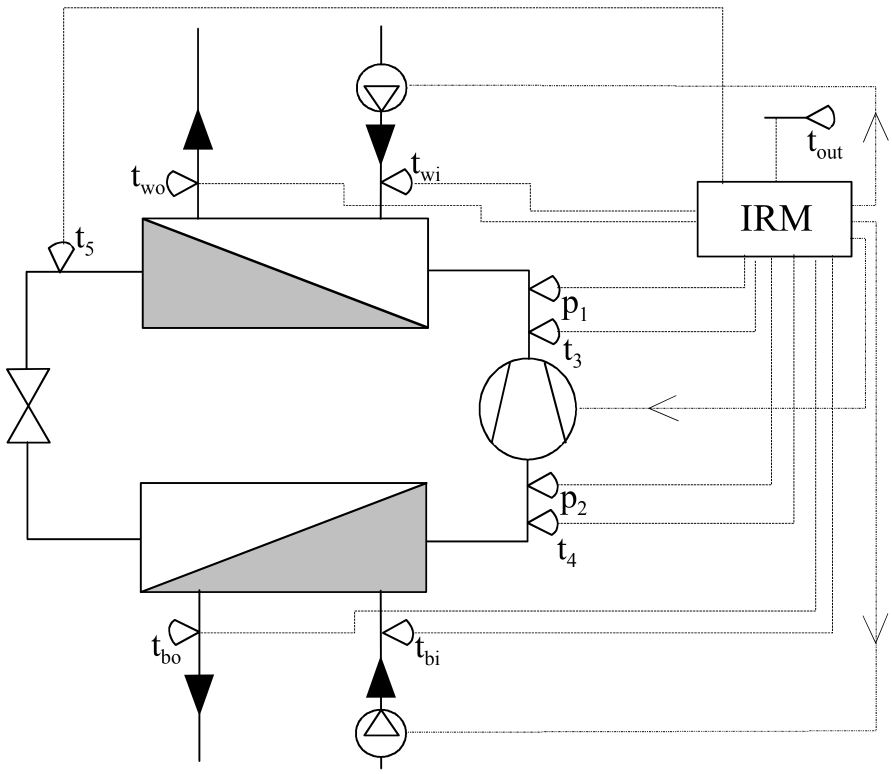


Figure 7.2 The temperatures and pressures that are normally measured in a heat pump are indicated in this layout. The arrowed dotted lines indicate the signals sent to the controlled components.

7.3 Conclusions

The suggested measurement system proved to be able to measure the coefficient of performance with a total uncertainty of measurement of 15 %. Probably an uncertainty of 10 % is also possible with proper installation of the sensors. It was also shown that the proposed IRM concept has the possibility to handle fault detection and diagnosis.

8 Discussion

This thesis has shown that it is possible to design an on-line optimising control system for heat pumps with feedback control and fault detection and diagnosis. The work also indicate substantial gains are possible by means of variable-speed drive electric motors for compressors, pumps and fans but that components must be specifically designed this. Below the most important results are further discussed and summarised.

8.1 Potential for increased energy efficiency

The results from the part load tests (section 6.1) showed no increase in energy efficiency for the variable-speed capacity controlled compressor when compared to a conventional on/off control. These results are quite to the contrary of what has been presented in earlier investigations by Marquand, Tassou, et al.[44], Tassou and Qureshi[74] and Miller[45]. The most probable reason for this is losses in the compressor motor due to the frequency conversion. The compressor used in these tests was not designed for variable-speed operation. There is also one very apparent difference between this investigation and those mentioned previously and that is that this investigation is made on a brine-to-water heat pump where plate heat exchangers are used as condenser and evaporator. The investigations mentioned above dealt with air-source heat pumps where finned-tube heat exchangers were used. The plate heat exchangers are oversized regarding the heat-transfer surfaces since the pressure drop across them is the dimensioning factor. For finned-tube heat exchangers it is the heat transfer surface that is dimensioning and not the pressure drop. Because of the large heat transfer capacity, the condensing and evaporating temperatures will not change as much for the plate heat exchangers as they will for finned-tube heat exchangers when the compressor capacity is reduced. Also, air source heat pumps experience the problems with frosting which degrades the efficiency. This problem can be reduced if variable-speed capacity control is applied. These two factors may explain why the benefits using variable-speed capacity control on air-source heat pumps are larger than for ground-source heat pumps. Another issue is that for small air-to-air heat pumps the compressor is designed for variable-speed operation. Such compressors are not yet available for ground-source units.

Still there is, as shown in section 6.2, a potential for increased energy performance for variable-speed brine-to-water heat pumps on an annual basis. This is mostly due to the decreased need for supplementary heat. The variable-speed control makes it possible to design a heat pump covering the total annual heat load without using supplementary heat. This can of course also be made with a conventional on/off controlled heat pump but the performance will be worse due to higher pressure ratio and a high frequency of on/off switching during many operating hours at part load. The results from this investigation also show the importance of considering also small losses and drive powers that has many operating hours. For example the power losses in the inverter are of the same magnitude as the drive power to the heating water pump. The total power used by the inverter, the heating water pump and the brine pump, although small in relation to the compressor, together have an accumulated effect on the total seasonal performance factor (SPF_{hs}) by approximately –12 % (case 3, size C).

8.2 On-line optimisation

The on-line optimisation method suggested in section 4.3 showed promising results during the computer simulations. It was possible to find the optimal function value and it was also possible to include soft constraints, i.e. constraints that may be violated for a short period of time. The major advantage with this method is that it is a direct search method and thus does not need a process model to perform the optimisation. The process can perform optimisation directly from measurements using the IRM integrated control concept. Thus it can be applied to different kinds of heat pump systems. This major advantage is also its major drawback. This direct searching makes the optimisation rather slow. Thus the best solution is probably to combine this method with an indirect model-based optimisation method. Thus a fairly simple model can be used and the Nelder-Mead simplex method can then be used for finding the true optimal set-point.

8.3 The IRM (Integrated Refrigeration Management) concept

The IRM concept was shown to be possible to use for control, performance indication and fault detection and diagnosis. It can also be used for the optimisation method described in the sections 4.3 and 5.1. The results show that it is possible to determine the coefficient of performance with an uncertainty of measurement of 10 %. The IRM system aims at using sensors that are normally used in heat pumps and this is possible for the temperature measurements. For the pressure measurements the normally used pressure switches must be replaced by pressure sensors providing an output signal. Also the power used by the compressor and the pumps must be able to measure. The power can be determined if the current is measured, presuming that the power factor and the voltage are stable. This current measurement is often included in frequency inverters and in new “intelligent” pumps. Thus it should be possible to realize the IRM concept.

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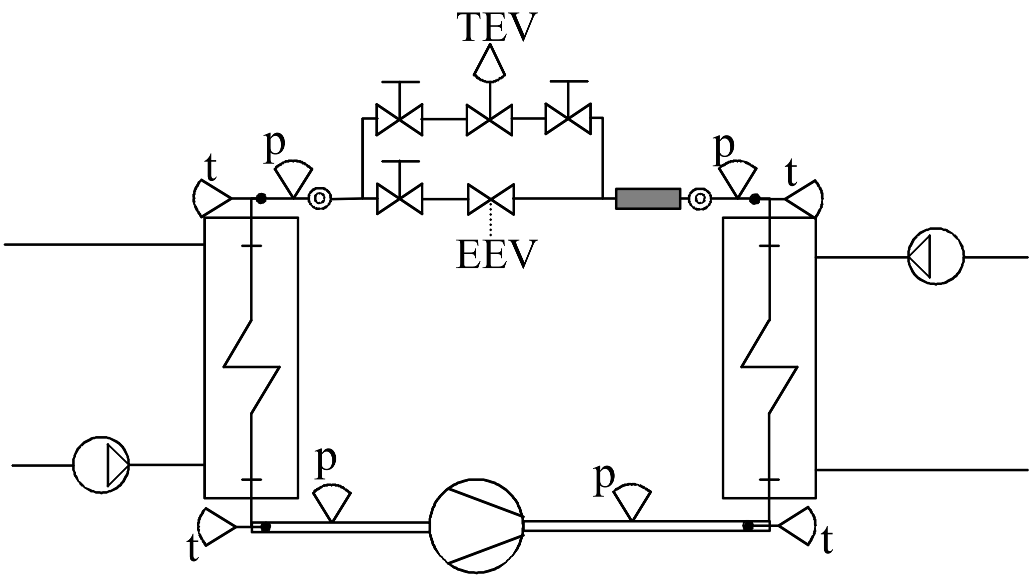
Appendix A

Test facility

To evaluate the energy saving potential of variable-speed compressors and electronically controlled expansion valves the heat pump shown in the picture below was built. The compressor was a standard compressor designed for single-speed operation. Two expansion valves were installed, one thermostatic and one electronic, with possibilities for operating one at a time. The electronic valve is of the modulating type, i.e. it is either fully open or fully closed. The refrigerant is thus sprayed into the evaporator. This particular valve had a period time of six seconds. The circulation pumps were of the type with three different speeds. The speed is set manually.



Condenser	Cetetherm CP 415-26
Evaporator	Cetetherm CP 615D-26
Thermostatic expansion valve (TEV)	Danfoss TUAE
Electronic expansion valve (EEV)	Danfoss AKV 10-7
Controller for the EEV	Danfoss AKC 114A
Compressor	Bristol Inertia (H25B32QDBE)
Frequency converter	Danfoss VLT 2830
Refrigerant	R407C (1.4 kg)

Desired flows and temperatures were provided by a test rig at SP used for capacity tests of ground-source heat pumps. Pt-100 resistance thermometers were used for measuring temperatures both in the refrigeration circuit and in the brine and heating water circuits. Pressure transmitters, type Danfoss AKS 32, were used for measuring the pressures in the refrigeration circuit.

Appendix B

B1 Equations of state for R407C

In order to calculate the derivatives in Eq. 7.2 - Eq. 7.7 explicit equations of state were derived for the refrigerant R407C. The equations of state were expressed in the same form as in the investigations by Cleland[12, 13].

Relation between temperature and pressure for the saturated refrigerant vapour:

$$t_s = \frac{-a_2}{\ln(p_s) - a_1} - a_3 \qquad \text{Eq. B.1}$$

Enthalpy of the liquid refrigerant:

$$h_5 = a_4 + a_5 \cdot t_5 + a_6 \cdot t_5^2 + a_7 \cdot t_5^3 \qquad \text{Eq. B.2}$$

Enthalpy of the saturated vapour:

$$\begin{aligned} h_{ml} &= a_8 + a_9 \cdot t_s + a_{10} \cdot t_s^2 + a_{11} \cdot t_s^3 + a_0 \cdot t_s^4 \\ h_s &= h_{ml} + a_{12} \end{aligned} \qquad \text{Eq. B.3}$$

This equation differs from the equation suggested by Cleland[12] which only used terms up to the power of three. In order to get a good accuracy also for R407C the last term (t^4) had to be added.

Enthalpy of the superheated vapour:

$$h_{m2} = h_{ml} \cdot g \qquad \text{Eq. B.4}$$

$$\begin{aligned} g = 1 &+ a_{13} \cdot \theta_{\sup} + a_{14} \cdot \theta_{\sup}^2 + a_{15} \cdot t_s + a_{16} \cdot \theta_{\sup}^2 \cdot t_s \\ &+ a_{17} \cdot \theta_{\sup} \cdot t_s^2 + a_{18} \cdot \theta_{\sup}^2 \cdot t_s^2 \end{aligned} \qquad \text{Eq. B.5}$$

$$h_{\sup} = h_{m2} + a_{12} \qquad \text{Eq. B.6}$$

These equations were fitted by least square solutions to the data for R407C generated by EES Klein and Alvarado[41]. Values of the constants a_0-a_{18} are stated in Table B.0.1. They are stated in the same way as in NT-VVS 116[48]. The equations above are valid in the range –60 to +60 ˚C with a maximum superheat of 60 ˚C. When used in this specified range they are correct within ±0,6 % when compared with data from EES. This is not valid for Eq. B.1 which is only correct within ±0,8 K (when using p_s as input) or ±3 % (when using t_s) as input.

Table B.0.1 Coefficients in Eq. B.1 - Eq. B.6.

Coefficient	
a_1	21,58155
a_2	2083,2326
a_3	243,65
a_4	200540
a_5	1439,64
a_6	2,5693
a_7 (10^{-3})	10,151
a_8	266453
a_9	566,4119
a_{10}	-1,62045
a_{11} (10^{-3})	22,2519
a_0 (10^{-3})	0,1596
a_{12}	146290
a_{13} (10^{-3})	3,09846
a_{14} (10^{-7})	24,6495
a_{15} (10^{-6})	103,222
a_{16} (10^{-8})	7,69973
a_{17} (10^{-8})	13,1101
a_{18} (10^{-10})	-7,57619

B2 Derivatives

The derivatives used in Eq. 7.2 - Eq. 7.7 are given below.

$$\frac{\partial h_3}{\partial p_1} = \frac{a_2}{(\ln(p_1) - a_1)^2} \cdot \frac{1}{p_1} \cdot [g_1 \cdot (a_9 + 2 \cdot a_{10} \cdot t_1 + 3 \cdot a_{11} \cdot t_1^2$$
$$+ 4 \cdot a_0 \cdot t_1^4) + h_{ml;1} \cdot (-a_{13} - 2 \cdot a_{14} \cdot \theta_{\mathrm{sup};1} + a_{15} +$$
$$a_{16} \cdot \theta_{\mathrm{sup};1} \cdot (\theta_{\mathrm{sup};1} - 2 \cdot t_1) + a_{17} \cdot t_1 \cdot (2 \cdot \theta_{\mathrm{sup};1} - t_1) +$$
$$2 \cdot a_{18} \cdot \theta_{\mathrm{sup};1} \cdot t_1 \cdot (\theta_{\mathrm{sup};1} - t_1) \qquad \text{Eq. B.7}$$

$$\frac{\partial h_4}{\partial p_2} = \frac{a_2}{(\ln(p_2) - a_1)^2} \cdot \frac{1}{p_2} \cdot [g_2 \cdot (a_9 + 2 \cdot a_{10} \cdot t_2 + 3 \cdot a_{11} \cdot t_2^2$$
$$+ 4 \cdot a_0 \cdot t_2^4) + h_{ml;2} \cdot (-a_{13} - 2 \cdot a_{14} \cdot \theta_{\mathrm{sup};2} + a_{15} +$$
$$a_{16} \cdot \theta_{\mathrm{sup};2} \cdot (\theta_{\mathrm{sup};2} - 2 \cdot t_2) + a_{17} \cdot t_2 \cdot (2 \cdot \theta_{\mathrm{sup};2} - t_2) +$$
$$2 \cdot a_{18} \cdot \theta_{\mathrm{sup};2} \cdot t_2 \cdot (\theta_{\mathrm{sup};2} - t_2) \qquad \text{Eq. B.8}$$

$$\frac{\partial h_3}{\partial t_3} = h_{ml;1} \cdot (a_{13} + 2 \cdot a_{14} \cdot \theta_{\sup;1} + 2 \cdot a_{16} \cdot \theta_{\sup;1} \cdot t_1 + a_{17} \cdot t_1^2 + 2 \cdot a_{18} \cdot \theta_{\sup;1} \cdot t_1^2) \qquad \text{Eq. B.9}$$

$$\frac{\partial h_4}{\partial t_4} = h_{ml;2} \cdot (a_{13} + 2 \cdot a_{14} \cdot \theta_{\sup;2} + 2 \cdot a_{16} \cdot \theta_{\sup;2} \cdot t_2 + a_{17} \cdot t_2^2 + 2 \cdot a_{18} \cdot \theta_{\sup;2} \cdot t_2^2) \qquad \text{Eq. B.10}$$

$$\frac{\partial h_5}{\partial t_5} = a_5 + 2 \cdot a_6 \cdot t_5 + 3 \cdot a_7 \cdot t_5^2 \qquad \text{Eq. B.11}$$