Capacity Control of Residential Heat Pump Heating Systems

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Building Services Engineering
Department of Energy and Environment
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2007
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Abstract

Heat pumps used for space and sanitary hot water heating of residential buildings are increasingly popular in northern Europe. As they compete with other heating equipment and in order to limit the environmental impact of their use, it is very important that heat pumps are energy efficient.

The scope of this thesis is to investigate the potential for increased energy efficiency of heat pumps by applying variable-speed capacity control to compressor, pumps and fans, as well as an overall optimisation strategy for online optimisation of the operation. Focus has been on ground-source heat pumps connected to hydronic heating systems but the same principles apply also to other types of heat pumps. For air-source heat pumps the defrost function adds one extra parameter to consider and this is partly investigated by the evaluation of a proposal for optimised defrost initiation.

Two prototype heat pumps with variable-speed compressors were designed and evaluated by laboratory measurements. The results from these measurements were used as input to models for calculating the energy efficiency of different heat pump systems. The models were used to compare the annual performance of the two prototype heat pumps to the performance of a state-of-the-art intermittently operated ground-source heat pump. One major result is that for the comparison between variable-speed and intermittent operation it is important to consider the transient behaviour of the heating system.

In a separate study the possible benefits of using variable-speed controlled circulators were investigated. Results show that the use of efficient circulators is of primary importance, especially for heat pumps with a variable-speed capacity controlled compressor. Of secondary importance comes the variable-speed operation and optimisation of circulator and compressor capacities. Most important in this respect is to control the capacity of the circulator for the bore hole system. Capacity control of the circulators can be implemented by a simple strategy with results very close to an optimised control.

Comparing the Seasonal Performance Factor (SPF) of an all variable-speed controlled heat pump (variable-speed pumps and compressor) with optimised pump and compressor capacities to the state-of-the-art intermittently controlled heat pump shows that the SPF could be increased by approximately 30%. However, this requires improved compressor and system design compared to the two prototype heat pumps.

Keywords: heat pump, variable-speed, capacity control, circulator, energy efficiency, performance, optimisation, ground-source, air-source, defrost, hydronic heating system
This doctoral thesis refers to research grants from two consecutive projects; “Optimal operation of heat pump systems - control of space heating and sanitary hot water production” (grant no. H22) and “Optimal operation of heat pump systems” (grant no. 22114-1). Project H22 was part of the national research program “eff-Sys – Efficient Refrigeration and Heat Pump Systems”. Both projects were financed by the Swedish Energy Agency in cooperation with SP Technical Research Institute of Sweden, Nibe, IVT, Thermia Värme, JEFF Electronics, Wilo, Grundfos, Carrier, Mare Trade, LK Lagerstedt & Krantz, Uponor, REHAU and Thermopanel
List of publications

The thesis is mainly based on the following papers:


For papers 1-3 I have gathered data (with some help from my colleagues at SP for the heat pump tests) and made the analyses. I have also written the papers. The co-authors have contributed with valuable input as discussion partners and by reviewing the papers. In paper 3, C. Markusson have also written one part of the introduction. Paper 4 is based on a master thesis by J. Erlandsson which was supervised by me, with P. Fahlén as examiner.

Papers 1-4 is the backbone of the thesis but the publications below also provide input data to the thesis and the above papers.


For the above publications where I am the first author I have made the writing and the analyses and also the data gathering. In publications 6 and 7 I have got valuable help during the laboratory tests by P. Lidbom and M. Stenlund. P. Fahlén has given valuable support in discussions and when reviewing the publications. For the publications where I am second author I have mainly served as discussion partner and reviewer of the manuscript.
Preface

The research described in this thesis has been carried out at SP Technical Research Institute of Sweden, Department of Energy Technology and Chalmers University of Technology, Building Services Engineering.

Now, when the work is finally finished (?) there is a number of persons I would like to thank. Most of all I would like to thank my supervisor Per Fahlén, who initiated this project, for support and guidance during this time. Although always busy, our meetings have been valuable and supporting. Unfortunately I still have not, and probably never will, picked up his passion for motorcycles. I still find it surprising how many connections there seem to be between motorcycles and heat pumps.

Many thanks also to all my colleagues at SP and the staff at Building Services Engineering for help with a whole lot of things, you are too many to list here. A special thanks to Torbjörn Lindholm for reading and commenting this thesis and to my co-authors Jessica Erlandsson and Caroline Markusson.

This work would not have been possible without the financial support from the Swedish Energy Agency and the knowledge of the people in the industry group listed on page IV. Thanks to all of you!

Finally, I would like to thank my fiancée Maria for her support and patience with me during the time I have worked with this thesis.

Fredrik Karlsson
Borås, October 2007
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# Designations

Latin letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area $[m^2]$</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance $[-]$</td>
</tr>
<tr>
<td>COP$_1$</td>
<td>Coefficient of performance; heating mode</td>
</tr>
<tr>
<td>COP$_2$</td>
<td>Coefficient of performance; cooling mode</td>
</tr>
<tr>
<td>COP$_C$</td>
<td>Coefficient of performance; Carnot cycle</td>
</tr>
<tr>
<td>$C$</td>
<td>Heat capacity $[J/K]$</td>
</tr>
<tr>
<td>$\dot{C}$</td>
<td>Heat capacity flow rate $[W/K]$</td>
</tr>
<tr>
<td>$d$</td>
<td>Diameter $[m]$</td>
</tr>
<tr>
<td>DOT</td>
<td>Design Outdoor Temperature $[^\circ C]$</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency $[Hz]$</td>
</tr>
<tr>
<td>$f_{comp}$</td>
<td>Supply frequency to the compressor motor</td>
</tr>
<tr>
<td>$f_{bp}$</td>
<td>Supply frequency to the brine pump</td>
</tr>
<tr>
<td>$f_{wp}$</td>
<td>Supply frequency to the heating water pump</td>
</tr>
<tr>
<td>$F$</td>
<td>Heat loss factor $[-]$</td>
</tr>
<tr>
<td>$F_{comp}$</td>
<td>Heat loss factor of the compressor, expressed as a fraction of the compressor power</td>
</tr>
<tr>
<td>$J$</td>
<td>Objective function</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy $[J/kg]$</td>
</tr>
<tr>
<td>$K$</td>
<td>Constant $[-]$</td>
</tr>
<tr>
<td>$L$</td>
<td>Length $[m]$</td>
</tr>
<tr>
<td>$\dot{M}$</td>
<td>Mass flow rate $[kg/s]$</td>
</tr>
<tr>
<td>$\dot{M}_R$</td>
<td>Mass flow rate; refrigerant</td>
</tr>
<tr>
<td>$\dot{M}_b$</td>
<td>Mass flow rate; brine</td>
</tr>
<tr>
<td>$\dot{M}_w$</td>
<td>Mass flow rate; water in the heating system</td>
</tr>
<tr>
<td>NTU</td>
<td>Number of Transfer Units $[-]$</td>
</tr>
<tr>
<td>$n$</td>
<td>Rotational speed $[\text{min}^{-1}]$</td>
</tr>
</tbody>
</table>
\( p \) Pressure [Pa] or [bar]
\( p_1 \) Condensing pressure
\( p_2 \) Evaporating pressure

\( \dot{Q} \) Thermal capacity [W]
\( \dot{Q}_1 \) Heating capacity
\( \dot{Q}_2 \) Cooling capacity

\( Q \) Thermal energy [J] or [kWh]

\( R \) Relative operating time [-] or total heat transfer resistance [K/W]
\( R_{hp} \) Relative operating time of the heat pump

\( SPF \) Seasonal Performance Factor [-]
\( SPF_{hp} \) SPF of heat pump (compressor only)
\( SPF_{hps} \) SPF of heat pump system (compressor plus pumps or fans)
\( SPF_{hs} \) SPF of heat supply system (compressor, pumps and supplementary heater)

\( t \) Temperature [°C]
\( t_r \) Return temperature from the heating system
\( t_s \) Supply temperature to the heating system
\( t_{wi} \) Inlet water temperature to the heat pump
\( t_{wo} \) Outlet water temperature from the heat pump
\( t_{woo;on} \) Outlet temperature from the heat pump when in operation
\( t_{wt} \) Temperature of supply water to the heating system taking the thermal inertia of the heating system into account

\( T \) Temperature [K]
\( TST \) Stop temperature of thermostat [°C]
\( DIF \) Thermostat dead band [°C]

\( U \) Thermal transmittance [W/(m²K)]

\( \dot{V} \) Volume flow rate [m³/s]

\( V \) Volume [m³]

\( v \) Velocity [m/s]

\( \dot{W} \) Power (electric) [W]

\( W \) Work (mechanical or electric) [J] or [kWh]

\( X \) Vapour quality [-]
<table>
<thead>
<tr>
<th>Greek letters</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha )</td>
<td>Heat transfer coefficient ([\text{W/(m}^2\text{K})])</td>
</tr>
<tr>
<td>( \Delta )</td>
<td>Difference</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>Heat exchanger effectiveness ([-] ]</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Efficiency ([-] ]</td>
</tr>
<tr>
<td>( \eta_v )</td>
<td>Volumetric efficiency</td>
</tr>
<tr>
<td>( \eta_{is} )</td>
<td>Isentropic efficiency</td>
</tr>
<tr>
<td>( \eta_C )</td>
<td>Carnot efficiency</td>
</tr>
<tr>
<td>( \eta_{inv} )</td>
<td>Efficiency of the frequency converter</td>
</tr>
<tr>
<td>( \eta_{mc} )</td>
<td>Mechanical efficiency of the compressor</td>
</tr>
<tr>
<td>( \eta_{m,el} )</td>
<td>Efficiency of the compressor motor</td>
</tr>
<tr>
<td>( \eta_{mi} )</td>
<td>Efficiency of the transmission between motor and compressor</td>
</tr>
<tr>
<td>( \eta_g )</td>
<td>Global insulation efficiency. Used for floor heating calculations</td>
</tr>
<tr>
<td>( \theta )</td>
<td>Logarithmic mean temperature difference ([\text{K}])</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>Pressure drop coefficient ([-] ]</td>
</tr>
<tr>
<td>( \mu )</td>
<td>Weighting factor ([-] ]</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density ([\text{kg/m}^3])</td>
</tr>
<tr>
<td>( \tau )</td>
<td>Time ([\text{s}]) or ([\text{h}])</td>
</tr>
<tr>
<td>( \tau_c )</td>
<td>Time constant</td>
</tr>
<tr>
<td>( \varphi )</td>
<td>Relative humidity ([%])</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Subscripts and superscripts</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a )</td>
<td>Air</td>
</tr>
<tr>
<td>( b )</td>
<td>Brine</td>
</tr>
<tr>
<td>( bldg )</td>
<td>Building</td>
</tr>
<tr>
<td>( c )</td>
<td>Condenser or cycle</td>
</tr>
<tr>
<td>( coll )</td>
<td>Collector</td>
</tr>
<tr>
<td>( comp )</td>
<td>Compressor</td>
</tr>
<tr>
<td>( d )</td>
<td>Defrost</td>
</tr>
<tr>
<td>( e )</td>
<td>Evaporator</td>
</tr>
<tr>
<td>( fan-coil )</td>
<td>Fan-coil system</td>
</tr>
<tr>
<td>( floor )</td>
<td>Floor heating system</td>
</tr>
<tr>
<td>( g )</td>
<td>Global</td>
</tr>
<tr>
<td>( h )</td>
<td>Hydraulic</td>
</tr>
<tr>
<td>( hp )</td>
<td>Heat pump – compressor only</td>
</tr>
</tbody>
</table>
\( hps \)  Heat pump system - compressor plus pumps
\( hs \)  Heat supply system - compressor, pumps and supplementary heater
\( i \)  Inlet
\( in \)  Indoors
\( int \)  Internal gains
\( is \)  Isentropic
\( m \)  Integrated mean value or exponent
\( max \)  Maximum
\( min \)  Minimum
\( n \)  Radiator exponent (=1.25 in this thesis)
\( o \)  Outlet
\( out \)  Outdoor
\( p \)  Pump
\( r \)  Return
\( R \)  Refrigerant
\( rad \)  Radiator system
\( s \)  Supply
\( set \)  Set value
\( sh \)  Supplementary heater
\( sw \)  Swept volume
\( v \)  Volumetric
\( w \)  Water

0  Start
1  Condenser, saturated refrigerant vapour in the condenser
2  Evaporator, saturated refrigerant vapour in the evaporator
3  Superheated refrigerant vapour in the compressor outlet
4  Superheated refrigerant vapour in the compressor inlet
5  Subcooled liquid refrigerant after the condenser
6  Refrigerant after the expansion device
7  Saturated refrigerant liquid in the condenser

\( \dot{i} \)  Mean value of temperature
Heat transfer system, cold side
Heat pump, hp
Supplementary heat, sh
Heat transfer system, hot side
Refrigerating machinery system, rms
Refrigerating system, rs
Heat supply system, hs

**Figure 0.1** System boundaries and designations for the heat pump heating system (NT-VVS 116[^61])

**Figure 0.2** State points in the refrigerant circuit (NT-VVS 116[^61])
1 Introduction

Heat pumping technologies are extensively used all over the world for both heating and cooling. Considering appliances for space conditioning, air conditioning for comfort cooling and dehumidification is the most common use. However, in Scandinavia heating only heat pumps is the most common application. This use is now also penetrating the central European market. Heat pumps compete with district heating and different types of boilers, in Scandinavia primarily using biomass, and this development combined with the overall goal of minimising environmental impact makes efficiency improvements of heat pumping technologies essential. This thesis focuses on ways of increasing the energy efficiency of small hydronic heat pump heating systems, i.e. heat pumps used as heat supply for water based heat distribution systems in mainly single-family houses. However, most of the principles apply also for other applications.

1.1 Background

Heat pumps for space and sanitary hot water heating are commonly used in Sweden today and more than 120 000 heat pumps were installed during 2006, making Sweden the largest market in Europe (EHPA\(^{[16]}\)). The markets in Austria and Switzerland are also established and markets in Finland, France, Norway and Germany are increasing. On average the sales of heat pumps in Europe increased by 52 % from 2005 to 2006 (EHPA\(^{[16]}\)).

According to Nowacki\(^{[60]}\) the heat pumps installed in Sweden in 2007 will, in total, supply approximately 22.5 TWh of heat. Of this, 15 TWh is free energy stored in the ground or in the air, which, according to Nowacki\(^{[60]}\) is approximately 10 % of all energy supplied to Swedish buildings. The most common heat distribution system in Sweden is the hydronic type, either with radiators, or for new houses, with under-floor heating.

![Figure 1.1 Total energy extracted and supplied per year by Swedish heat pumps (Nowacki\(^{[60]}\)).](image-url)
The use of heat pumps has an impact on the environment which can be divided into four categories:

- extraction of raw material and manufacturing
- emissions from electricity generation
- emissions from refrigerant leakage
- emissions from demolition.

The emissions due to electricity generation is for most applications considered to cause the main impact on the environment, even though it will depend on how the electricity is produced, as shown by e.g. Forsén\textsuperscript{[27]}. Thus it is vital to increase the energy efficiency, i.e. the seasonal performance factor (SPF) of heat pump heating systems.

Increased efficiency of heat pump heating systems can be achieved by making the heat pump more efficient but also by making changes to the distribution system. For example, by designing hydronic heating systems for the supply temperature +35 °C at the design outdoor temperature, DOT, instead of the more common +55 °C, will increase the SPF by approximately 40 % (using the estimate that COP increases by 2 %/K of lowered condensation temperature). Also, the use of efficient pumps and fans in both the heat pump and the distribution system is important. For the heat pump itself, the energy efficiency can be increased by improving the efficiency of the compressor (for example by variable-speed control), the heat transfer capacity of the condenser and the evaporator, the control of the expansion device and the defrost system of air source heat pumps. Further improvements are possible by controlling the heat pump heating system properly.

1.2 Purpose

The purpose of this work was to systematically investigate the possibilities for increased energy efficiency of small heat pump heating systems when applying new components and controls, such as variable-speed controlled compressors and circulators. The hypothesis was that the energy efficiency would improve by introducing new components and controls. The work was made in two steps. First the potential for improvement considering the heat pump only, was analysed and in a second step the heating system was considered together with the heat pump, i.e. the heat pump heating system.

New components in this case mean electronically controlled compressors, pumps, fans and expansion valves. The pumps and compressors are variable-speed controlled. This was applied to ground-source heat pumps in hydronic heating systems as this is the most common heat pump heating system in Sweden and as it is the application least investigated in this context before.
1.3 Methodology

Both experiments and simulations have been used during this work. First, a literature study was made. With this as a basis, two prototype ground-source heat pumps were designed and experimentally evaluated in a laboratory. Models for energy saving calculations and optimisation were then developed, and data from the laboratory tests were used as input to these models. The models were used for determining the energy saving potential on annual basis and for proposing a control and optimisation method.

1.4 Structure of the thesis

The thesis is written as a monography, but chapters 3, 4, 5 and 7 are basically papers 1-4 in the list of papers. The idea is that chapters 3 – 7 should be possible to read individually, without reading the other chapters before. Each of these chapters contains a conclusions part even though at the end of the thesis there is a general discussion and conclusions part for the whole thesis.

Chapter 1 is the introductory part of the thesis giving the background and motivation for the work.

Chapter 2 gives information to the mathematical models used in later chapters.

Chapter 3 gives results from laboratory testing on two prototype variable-speed capacity controlled heat pumps and one state-of-the-art intermittently controlled heat pump. These results are then used as input to seasonal performance calculations. Standard, non-controlled circulators are used for these calculations.

Chapter 4 further analyses the energy saving potential of variable-speed capacity controlled and intermittently controlled heat pumps, but now considers also the transient behaviour of the heating system. This makes it possible to analyse the influence of the thermal inertia of the heating system and lag times etc. Still, non-controlled standard circulators are used.

Chapter 5 analyses the potential benefits of using efficient variable-speed pumps and an optimisation method for optimising the compressor and pump speeds.

Chapter 6 analyses previous investigations and gives suggestions for implementation of methods for controlling and optimising the operation of a capacity controlled heat pump heating system.

Chapter 7 deals with the issue of when to initiate a defrost in order to optimise energy efficiency of air source heat pumps.

Chapter 8 provides indications of how the results from this thesis apply to other heat pump types.

Chapter 9 summarises the thesis and gives suggestions for further work.
1.5 Basic principle for a heat pump heating system

A heat pump can be used for cooling or heating or both depending on application. Air-to-air heat pumps often include both functionalities whereas ground source heat pumps mostly provide heating only. In this thesis a heat pump heating system consists of a heat pump for heating connected to a hydronic heating system that rejects the heat from the heat pump to the building’s occupied space.

The general principle is the same for all heat pumps; heat is extracted from a low temperature heat source ($\dot{Q}_2$) and by adding work ($\dot{W}$), heat can be rejected to the heat sink at a higher temperature level ($\dot{Q}_1$), see Figure 1.2.

![Figure 1.2 The basic principle for a heat pump](image)

For an adiabatic system the heat balance can be written as:

$$\dot{Q}_1 = \dot{Q}_2 + \dot{W}$$  \hspace{1cm} Eq. 1.1

For heat pumps for residential use the most common heat sources are outdoor air, exhaust air or the ground (horizontal or vertical collector systems). The sink is normally a hydronic system, except for the air-to-air heat pumps where the heat is rejected directly to the indoor air.

The most common hydronic system in Sweden is the radiator system, while for new construction under-floor heating systems are very common. An interesting alternative, especially when replacing the heating system in houses with direct acting electric heaters, is the hydronic fan-coil unit which is very compact in terms of heat transfer capacity per unit.
Figure 1.3  Floor heating (left), radiator (middle) and hydronic fan-coil (right). Pictures from Uponor, Rettig and Carrier respectively.

The heat pump’s capacity is normally controlled by so called curve control. This is an open-loop control where the temperature level of the heat transfer medium in the heating system is related to the outdoor air temperature, as shown in Figure 1.4. The inclination of the curve can be chosen (heating curve number) and the curve can also be moved in parallel (heating curve offset) in order to get the control system to fit the heat demand of the actual building. The heating curve can refer to the supply temperature as in Figure 1.4, or to the return temperature. In many cases there is a possibility to use a room temperature sensor as a complement. The control deviation of the room temperature is then used for changing the curve control by for example moving the curve in parallel or changing the inclination. Using a room temperature sensor gives the possibility of making the control compensate also for internal heat gains.

Figure 1.4  In the controls for the heat pump the heating curve is chosen such that it fits the demand of the actual building. In this example curve 9 is chosen and then the supply temperature of the heating system shall be 55 °C at the outdoor air temperature -20 °C (source: Nibe).

The heat pump is switched off at a certain temperature higher than the temperature given by the heating curve and is switched on again at a lower temperature. This difference, the controller dead band, can either be a fixed temperature difference or given as a heat deficit measured in degree minutes. The heat deficit is measured as the temperature difference from the set value multiplied by the time. The value for either of these are set in the heat pump’s control system. Small dead bands are
used for heating systems with high thermal inertia, such as under-floor heating systems, and higher values are used for lighter systems, such as fan-coils.

1.6  Possibilities for improved energy efficiency

The efficiency of a heat pump is expressed by the Coefficient Of Performance (COP), which is the quotient between the useful heating capacity and the power input, Eq. 1.2.

\[
COP_1 = \frac{\dot{Q}_1}{W}
\]

The theoretical upper limit for the COP of a heat pump operating between the condensation temperature \( T_1 \) and evaporation temperature \( T_2 \) is expressed by the Carnot Coefficient Of Performance, \( COP_C \). The Carnot cycle requires reversible isentropic compression and expansion and heat transfer without losses. \( COP_C \) can be expressed by the two temperatures, as shown in Eq. 1.3.

\[
COP_C = \frac{T_1}{T_1 - T_2}
\]

In a real application irreversibilities are present, such as heat losses from the compressor and friction. Furthermore, the heat pump operates between the two temperatures \( T_2 - \Delta T_2 \) and \( T_1 + \Delta T_1 \), rather than \( T_1 \) and \( T_2 \), as there must be a temperature difference between the heat source and the refrigerant, and the refrigerant and the heat sink in order to transfer heat. The actual values of \( T_1 \) and \( T_2 \), and thus COP, depend to a large extent on the present heat source temperature and the design of the heating system, i.e. parameters which are not determined by the design of the heat pump itself.

To make the heat pump more efficient the real cycle should of course be as close to the Carnot cycle as possible. This can be accomplished by reducing the temperature differences in the heat exchangers (\( \Delta T_1 \) and \( \Delta T_2 \)) as much as possible and by reducing losses in the compressor. Some of the means available for approaching the ideal situation are described in the following sections.

1.6.1  Capacity controlled compressors

Heat pumps used for space heating are connected to continuously varying heat loads and must be able to control its capacity to meet the load. The capacity of the heat pump can be controlled by adjusting the capacity of the compressor. The most common way for small heat pumps is intermittent control where the compressor is switched on and off by a thermostat. If applying continuous capacity control instead, such as variable-speed control, Miller\(^{58}\) and Garstang\(^{29}\) suggests that the energy efficiency of the heat pump can be improved due to:
- Better performance at part load
- Reduced need for supplementary heating
- Reduced need for defrosting
- Fewer cycles on and off

**Better performance at part load**: An intermittently controlled compressor will start to be switched on and off when the heat pump’s capacity is greater than the heat load. This causes the heat pump to operate between two temperature levels of the high temperature heat transfer medium, see Figure 1.5. This means that the compressor will operate at a higher mean supply temperature, \( \bar{t}_{\text{w0,on}} \), and consequently a higher mean condensation temperature, than it would if the capacity could be continuously adjusted and follow the mean temperature needed, \( \bar{t}_{\text{w0}} \). For the example in Figure 1.5, the difference between the cycle mean temperature and the on-mode mean temperature is 5.7 K, equivalent to a difference in COP of 11 % (assuming the COP to change by 2 %/K). Reducing the compressor capacity will also cause the evaporation temperature to increase and thus further increase COP.

![Figure 1.5](image)

**Figure 1.5** The graph shows how the supply temperature to the heating system varies over one complete on and off cycle. Note that the mean temperature during the on-time is 5.7 K higher than the mean temperature needed by the heating system (Fahlén[21]).

**Reduced need for supplementary heating**: The design of heat pumps differ between different countries. In Sweden and the rest of Scandinavia the heat pumps (except exhaust air heat pumps) are generally designed to cover approximately 60 % of the load at the DOT. The heat pump will then cover 85 – 90 % of the annual heating demand. The remaining part is covered by an electric heater which is normally integrated into the heat pump unit. This is mainly an economical design – you will have to buy a heat pump with almost doubled capacity to cover the remaining 10 – 15 % of the annual demand. In other countries it is standard to design the heat pump to cover 100 % of the demand. Full covering heat pumps may cycle on and off very frequently during the majority of the heating season, thus causing excessive wear. Also, as the heat pump’s control often limits the number of cycles per hour, the room temperature
may vary too much. Furthermore, the intermittent operation may add energy losses to the system. Using a continuously capacity controlled compressor will make it possible to design a heat pump to cover 100% of the load without the possible problems of excessive on and off cycling.

In Sweden the electricity production and distribution companies have raised concerns about the electric peak load occurring during cold winter days when many heat pumps can not provide the capacity needed and switch on their built in electric heaters. This peak load is of high capacity and short duration, thus making it expensive for these companies to provide the capacity necessary. A full covering heat pump will reduce this peak load. If capacity tariffs comes into use the economic incentive for full covering heat pumps is likely to increase.

Reduced need for defrosting: When running the heat pump at low capacities the evaporation temperature will increase thus reducing the frost formation on the evaporator of air-source heat pumps. This will reduce the need for defrosts and consequently the COP will increase.

Fewer cycles on and off: The losses due to the cycling itself, as pointed out by Bergman[7] and Karlsson[44], will be negligible for the kind of systems discussed in this thesis due to the short duration of the start-up compared to the total cycle time and the reduction of refrigerant migration by the use of thermostatic expansion valves.

1.6.2 Capacity control of pumps and fans

Pumps and/or fans drive the high and low temperature heat transfer media flows to heat pumps. The drive power to pumps and fans is at high compressor capacities a small part of the total drive power. However, reducing the compressor capacity without reducing the pump and fan capacities will cause these to constitute a larger part of the total drive power. Thus, it will be of importance to be able to vary also the pump and fan capacities for heat pumps with continuous capacity control of the compressor as the running hours will be longer than with intermittent control.

Capacity control of pumps and fans can also be used for decreasing the temperature differences $\Delta T_1$ and $\Delta T_2$ in the condenser and evaporator. An increased heat transfer medium flow will increase the total heat transfer. Keeping the inlet temperatures of the heat transfer media flows constant will then cause the outlet temperature of the high temperature heat transfer medium to be lower, and the temperature of the low temperature heat transfer medium to be higher. This lower respectively higher mean temperature of the media flows will reduce the condensation temperature and increase the evaporation temperature, which will reduce the drive power to the compressor. On the other hand the drive power to pumps and/or fans will increase. For each condition there will exist an optimal combination of flows where the total drive power is minimised while still supplying the requested capacity, see Figure 1.6.
Figure 1.6 For the heat sink side there will exist an optimal condensation temperature minimising the resulting drive powers to compressor and pump as shown e.g. by Jakobsen\cite{41}.

1.6.3 Evaporator superheat control

The refrigerant out of the evaporator is kept superheated to be sure that no liquid enters the compressor which may cause liquid hammering and damage the compressor. However, the superheat should be kept as low as possible as the single phase heat transfer capacity is lower than the phase change heat transfer. It has been shown both experimentally by Huelle\cite{36} and theoretically by van der Meer\cite{85} that the most energy efficient evaporator superheat is as low as possible while still being stable, see Figure 1.7. This limit is called the MSS-line (Minimum Stable Signal). At lower superheats the control is unstable. A thermostatic valve, which is the most commonly used throttling device, is tuned to give stable control at the design cooling load, then its characteristics will keep the superheat on the stable side. At low loads this means that the heat pump will operate at an unnecessarily high superheat, as shown in Figure 1.7. Using an electronic expansion valve make it possible to implement control strategies to keep the superheat close to the MSS-line over the full capacity range.

Figure 1.7 To the left of the MSS-line the operation of the valve is unstable, and to the right it is stable. The highest energy efficiency is obtained when following the MSS-line.
1.7 Literature survey

In the previous section the idea of improved performance by new components and controls was described. In this section the subject will be further treated by reviewing the research performed within this field. The review is based on a literature survey by Karlsson[46], with updates for new publications. In addition to the subjects dealt with in this section comes the control and optimisation of the heat pump operation. The literature review for this part can be found in chapter 6.

1.7.1 Capacity control of compressors and development in compressor technology

According to e.g Qureshi[72] there are different continuous capacity control techniques available, such as hot gas bypass, evaporator temperature control, clearance volume control, multiple compressor control, cylinder unloading and variable-speed control. Several different investigations, e.g. by Holdack-Janssen[35], Qureshi[71], Pereira[64] and Wong[88-89] claim that the variable-speed control is the most energy efficient method.

The variable-speed operation is accomplished by a frequency converter that changes the frequency of the electric supply to the motor which then changes the rotational speed of the compressor and thus the refrigerant flow. The frequency converter introduces extra losses due to losses in the converter itself, but also due to losses in the electric motor caused by the non-sinusoidal wave form. According to Qureshi[72] compressor motors for small heat pumps are mainly asynchronous motors due to low cost, reliability and availability. Obitani[62] claims that the motor efficiency can be increased by 30 % by using permanent magnet synchronous motors instead of asynchronous motors.

The most common compressor types used for small heat pumps are reciprocating, rotary and scroll compressors which all can be variable-speed controlled. Riegger[73] compared these three types of compressors in variable-speed operation. One main conclusion was that there is no simple way of determining which type is most suitable for variable-speed operation, as it depends on manufacturing costs, energy efficiency, capacity ranges and operating conditions. Thus, the compressor choice will depend on application.

Ilic[38] and Poort[65] report that compressor short cycling may be an economical alternative to variable-speed compressors. Compressor short cycling means total cycle times of on and off cycles in the order of seconds compared to normal cycling which has a time scale of minutes or hours. Thus, compressor short cycling approaches the variable-speed operation. Excluding compressor losses the short cycling reduced COP by 2.5 % at a cycle time of 10 seconds compared to ideal variable-speed operation (no frequency converter losses). Increasing the cycle time to 80 seconds causes the losses to increase to 7.5 %, mainly due to higher refrigerant heat transfer resistance and pressure drop. Normal cycling gave losses of 11.5 % with the same equipment under the same conditions.
Comparisons of the energy efficiency for heat pumps (both in heating and cooling mode) when applying variable-speed capacity control and intermittent control have been reported in many previous investigations. The comparisons are made in many ways and the results will thus also be different. Comparisons can be made by running the same compressor in both variable-speed and intermittent operation, by comparing different compressors (one variable-speed and one intermittent), by comparing the performance at specific operating conditions or for the seasonal performance. The result will also depend on the application, i.e. if it is a heat pump, air conditioner or refrigeration application.

I think the most relevant analysis is to determine the seasonal performance of a state-of-the-art intermittent controlled unit and compare this to the seasonal performance of a variable-speed controlled unit. This is what is interesting for the end user, not the result from comparisons of performance at a few conditions. Investigations where variable-speed compressors have been compared to intermittent controlled compressors in hydronic heat pump heating applications have been reported by Poulsen[66-67, 70], Tassou[81], Aprea[5], Bergman[7] and Forsén[28]. Comparison of performance at single operating points show improvement in COP by 10-30 % when using variable-speed compressors. Investigations of the seasonal performance factor show increases in performance in the order of 1-25 %. The difference in results depend on type of heat pump (air source or ground source), compressor type, operating range (speed) and of course design of capacity to heat load.

1.7.2 Variable-speed pumps and fans

Investigations on the optimal combination of heat transfer media flows and refrigerant flow have been presented by Andrade[1], Jakobsen[40, 42], Skovrup[75], Liptak[55] and Hydeman[37].

Skovrup[75] analysed the optimal condensation temperature for a large industrial cooling plant where the condensers are connected to cooling towers. He found that the optimal condensation temperature is linearly dependent on the outdoor air wet bulb temperature. The same conclusion is drawn by Liptak[55]. Hydeman[37] proposed a method where the condenser pump and cooling tower fan are controlled linearly in relation to the system load. According to simulations this is just as efficient as controlling condenser water temperature linearly to the wet bulb temperature and is claimed to be easier to implement. The simulations show this to be 10-32 % more efficient on an annual basis than conventional controls.

In another investigation, where a variable-speed pump supplies the flow to an evaporator in a cooling system with a variable-speed compressor, Jakobsen[42] proposed a method for finding the optimum combination of pump and compressor speeds, minimising the total work, by minimising the exergy losses. The method is derived by computer simulations but unfortunately no conclusion or estimation is made regarding the energy saving potential.
1.7.3 Evaporator superheat control

Superheat control concerns both energy efficiency and stable control. Both these issues can be divided into the categories of steady-state performance and transient performance. Regarding energy efficiency at steady-state conditions the investigations by Tassou\textsuperscript{[80]} and Aprea\textsuperscript{[2]} do not show any difference between electronic expansion valves (EEV) and thermostatic expansion valves (TEV). In these investigations the EEV was set to keep a constant superheat, i.e. it does not follow the MSS-line. In both investigations a solenoid valve was used. During start-up of the compressor the EEV stabilises at the target superheat faster than the TEV and with a smaller amplitude of the oscillations.

Jolly\textsuperscript{[43]} compared a TEV, an EEV with fixed superheat setting and an EEV with adaptive control during cool down of a refrigeration container. Both EEVs were solenoid valves. The EEV with fixed superheat resulted in a 10 % higher COP than the TEV. The adaptive control, aiming to follow the MSS, further increased the COP by 7.5 %. The TEV starts with a superheat of 5 K but as the cool down process proceeds and the evaporator temperature decreases the superheat increases up to 7.5 °C. The EEV with the fixed superheat keeps the superheat at 5 K which explains the higher COP. The adaptive control searches for the MSS-line and thus varies and goes down to a superheat as low as 1.3 K.

So far it seems obvious that EEVs give more stable control of the superheat at transient conditions and that the possibility to incorporate adaptive control strategies show a potential for increased energy efficiency. However, Aprea\textsuperscript{[4]} reports that when comparing the performance of an EEV (solenoid valve) to a TEV in a refrigerating machine with a variable-speed compressor supplying cold air to a cold store, the TEV resulted in an 8 % higher COP. The test was run for six hours during which the cooling load was varying. The reason for the better result for the TEV is claimed to be the smaller oscillations of the superheat compared to the EEV. This is contrary to what has earlier been reported by Tassou\textsuperscript{[80]} and Aprea\textsuperscript{[2]}, where the oscillations using an EEV were smaller than those for a TEV and the COP was also higher for the EEV than the TEV. Also, considering the theory in chapter 1.6.3, the EEV should result in higher or equal COP compared to the TEV.

Applying EEVs gives possibilities for developing different control strategies. Jolly\textsuperscript{[43]} and Chia\textsuperscript{[11]} used fuzzy logic controllers. Outtagarts\textsuperscript{[63]} compared the performance of an optimal qualitative control strategy and a PD-controller. No major difference in result between the two strategies was noted. It was found that a special start-up controller was needed as the steady-state controller resulted in large oscillations in superheat during compressor start-ups and step changes in compressor speed. Finn\textsuperscript{[26]} compared the performance of self designed PID controllers to commercially available controllers for EEVs and found it possible to get equal performance both in steady and transient state.
2 Modelling of heating systems

This section describes the general principles of the models used in chapters 3 and 4.

2.1 Outdoor climate model

Of the outdoor climatic conditions only the outside air temperature, \( t_{\text{out}} \), is considered. Wind load, solar radiation etc. are not considered. A curve fit model by Hallén\[31\] and modified by Fehrm\[25\] was used for generating the duration curve of the outdoor air temperature for Swedish conditions. The only input needed to Eq. 2.1 below is the annual mean outdoor air temperature, \( \bar{t}_{\text{out}} \). The time, \( \tau \), must be given in hours.

\[
\begin{align*}
\bar{t}_{\text{out}}(\tau) &= (\tau - 4380) \cdot (3.9 - 0.086 \cdot \bar{t}_{\text{out}}) \cdot 0.001 + \bar{t}_{\text{out}} + \\
&\quad \left[ \tau \cdot \left( \frac{8 - \bar{t}_{\text{out}}}{586} \right)^3 \right] + \left[ \frac{1550}{700 + \tau} \right]^3 + [^\circ\text{C}] \\
&\quad 1.5 \left[ \frac{\bar{t}_{\text{out}}}{8} \cdot \frac{1200}{500 + \tau} \right] \cdot \cos \left( \frac{900 - \tau}{585} \right)
\end{align*}
\]

2.2 Heating system models for steady-state conditions

The radiator system is modelled according to Fehrm\[25\]. The model is based on the energy balances of Eq. 2.2 to Eq. 2.5.

Building heat demand with:

- \( t_{\text{in}} \) = Indoor temperature [°C]
- \( \Delta t_{\text{int}} \) = Equivalent temperature rise from internal heat gains [°C]
- \( R_{\text{bldg}} \) = Heat transfer resistance through building envelope including ventilation losses [K/W]

\[
\dot{Q}_{\text{bldg}} = \frac{(t_{\text{in}} - \Delta t_{\text{int}} - t_{\text{out}})}{R_{\text{bldg}}} [\text{W}] 
\]

Eq. 2.2
Radiator system heating capacity with:

- $K =$ Design constant of the radiator system (UA-value) [W/K$^n$]
- $n =$ Radiator exponent which is specific for the radiator used. Normal values are in the range of 1.2-1.3. It is set to 1.25 in this thesis. [-]
- $\theta_{rad} =$ Logarithmic mean temperature difference for the radiator system, see Eq. 2.6. [°C]

$$\dot{Q}_{rad} = K \cdot \theta_{rad}^n \quad \text{[W]} \quad \text{Eq. 2.3}$$

Heating system supply capacity with:

- $\dot{C}_w =$ Heat capacity flow rate of the water in the heating system [W/K]
- $t_s =$ Supply temperature to the heating system [°C]
- $t_r =$ Return temperature from the heating system [°C]

$$\dot{Q}_{hs} = \dot{C}_w (t_s - t_r) \quad \text{[W]} \quad \text{Eq. 2.4}$$

At steady-state conditions the relation between heat demand, radiator system capacity and heating system supply capacity are equal.

- $\dot{Q}_{bldg; DOT} =$ Building heat demand at design outdoor air temperature [W]
- $\dot{Q}_{rad; DOT} =$ Radiator system capacity at design outdoor air temperature [W]
- $\dot{Q}_{hs; DOT} =$ Heating system supply capacity at design outdoor air temperature [W]

$$\frac{\dot{Q}_{bldg}}{\dot{Q}_{bldg; DOT}} = \frac{\dot{Q}_{rad}}{\dot{Q}_{rad; DOT}} = \frac{\dot{Q}_{hs}}{\dot{Q}_{hs; DOT}} \quad \text{Eq. 2.5}$$

Assuming a constant room temperature of $t_{in}$, the logarithmic mean temperature difference becomes:

$$\theta_{rad} = \frac{\theta_r - \theta_o}{ln\left(\frac{\theta_r}{\theta_o}\right)} = \frac{t_s - t_r}{ln\left(\frac{t_s - t_{in}}{t_r - t_{in}}\right)} \quad \text{[°C]} \quad \text{Eq. 2.6}$$
From the above set of equations it is possible to derive the return and supply temperatures as:

\[ t_r = t_s - a \cdot c \quad \text{[°C]} \quad \text{Eq. 2.7} \]

and

\[ t_s = \frac{\exp(d) \cdot (t_{in} + a \cdot c) - t_{in}}{\exp(d) - 1} \quad \text{[°C]} \quad \text{Eq. 2.8} \]

In Eq. 2.7 and Eq. 2.8, \( a, b, c \) and \( d \) are variables for a specific object according to:

\[ a = \frac{t_{in} - \Delta t_{int} - t_{out}}{t_{in} - \Delta t_{int} - \text{DOT}} \quad \text{[-]} \quad \text{Eq. 2.9} \]

\[ b = \ln \left( \frac{t_{s;DOT} - t_{in}}{t_{r;DOT} - t_{in}} \right) \quad \text{[-]} \quad \text{Eq. 2.10} \]

\[ c = t_{s;DOT} - t_{r;DOT} \quad \text{[K]} \quad \text{Eq. 2.11} \]

\[ d = a \left( \frac{1}{n} \right) \cdot b \quad \text{[-]} \quad \text{Eq. 2.12} \]

For hydronic fan-coils the set of equations are the same as the one in Eq. 2.2 to Eq. 2.5, except for Eq. 2.3 which is replaced by the expression in Eq. 2.13 below.

The expression for the capacity of the fan-coil is derived from the \( \varepsilon \)-NTU method.

- \( \varepsilon = \) Heat exchanger effectiveness [-]
- \( \dot{C}_{min} = \) Smallest heat capacity flow rate. In most cases this will be the air flow [W/K]

\[ \dot{Q}_{fan-coil} = \varepsilon \cdot \dot{C}_{min} \cdot (t_s - t_{in}) \quad \text{[W]} \quad \text{Eq. 2.13} \]
Figure 2.2 Temperature profile of the fan-coil system

The resulting expressions for the supply and return temperatures are:

\[ t_r = t_s - a \cdot c \quad [\degree C] \text{ Eq. 2.14} \]

\[ t_s = t_{in} + a \cdot (t_{s;DOT} - t_{in}) \quad [\degree C] \text{ Eq. 2.15} \]

For under-floor heating, the model by Roots\textsuperscript{[74]} is used. The equations for the building and the heat pump system are the same as before, but the energy balance model and the heating system model are changed. Because the under-floor heating system looses energy through the ground slab the energy balance equation is modified as:

\[ Q_{floor} = \dot{Q}_{bldg} = \eta_g \cdot \dot{Q}_{hr} \quad [\text{W}] \text{ Eq. 2.16} \]

The efficiency \( \eta_g \) expresses how well insulated the slab is. The more insulated the slab is the higher value of this efficiency. The efficiency value is derived by comparing the performance to a house with a heating system that is not embedded in the floor. A radiator system will thus have an efficiency \( \eta_g = 1 \). A well insulated floor gives a value of \( \eta_g \approx 0.96 \), whereas for a more normal floor heating system it is about 0.92. In this thesis the value 0.96 has been used. The heat rejected from the floor heating system is expressed as:

\[ \dot{Q}_{floor} = \frac{1}{R_{in}} \cdot \left( \frac{t_s + t_r}{2} - t_{in} \right) \quad [\text{W}] \text{ Eq. 2.17} \]

Where \( R_{in} \) expresses the heat transfer resistance between the warm water in the floor pipes and the indoor air. It can be calculated at DOT as:

\[ \frac{1}{R_{in}} = \frac{\dot{Q}_{bldg;DOT}}{t_{s;DOT} + t_{r;DOT} - t_{in}} \quad [\text{W/K}] \text{ Eq. 2.18} \]

The resulting expressions for the supply and return temperatures in the under-floor heating system become:
\[ t_r = \frac{t_s \cdot (\beta - 1) + 2 \cdot t_{in}}{\beta + 1} \] \quad [^\circ\text{C}] \quad \text{Eq. 2.19}

and

\[ t_s = \frac{\frac{R_{in}}{R_{hidg}} \cdot (t_{in} - \Delta t_{in} - t_{out}) + t_{in} \cdot \left( \frac{\beta}{\beta + 1} \right)}{\beta} \] \quad [^\circ\text{C}] \quad \text{Eq. 2.20}

with

\[ \beta = 2 \cdot \eta_g \cdot \dot{C}_w \cdot R_{in} \] \quad [-] \quad \text{Eq. 2.21}

When the heat pump is connected in parallel to the heating system, there will be a mix of the heat pump water flow and the heating system water flow. If the heat pump water flow, \( \dot{V}_{hp} \), is higher than the heating system water flow, \( \dot{V}_{hs} \), the mixing occurs in the return line. If the heating system water flow is higher than the heat pump water flow the mixing occurs in the supply line. This is not desirable because the return flow will then cool the supply flow from the heat pump. An energy balance for the right junction in Figure 2.3 gives that:

\[ t_{wi} = \frac{\left( \dot{V}_{hp} - \dot{V}_{hs} \right) \cdot t_s + \dot{V}_{hs} \cdot t_r}{\dot{V}_{hp}} \] \quad [^\circ\text{C}] \quad \text{Eq. 2.22}

In the calculations, the heat pump flow will always be higher than the heating system flow.

Figure 2.3 The heat pump is often connected in parallel with the heating system; either with a supplementary heater included in the heat pump unit (left) or with an external supplementary heater (right).
2.3 Heating system models for transient conditions

The principle is based on the work by Bergman[6]. It relies on energy balances of the heat pump and the heating system as well as of the heating system and the building. The model is illustrated in Figure 2.4. In the model thermal inertia and heat transfer capacity are separated. This was made in order to verify the model by measurements. For that a test equipment was constructed where the thermal inertia was realised by a storage tank and the heat transfer capacity was realised by a separate heat exchanger.

Figure 2.4 The figure illustrates the model developed for the calculations.

The equations given below are valid for a radiator heating system but the methodology will be the same for any other type of heating system. The energy balance for the building and the heating system is given by:

\[
\dot{Q}_{\text{bldg}} \cdot d\tau = \dot{Q}_{\text{rad}} \cdot t_{\text{rad}} - C_{\text{bldg}} \cdot dt_{\text{ai}} \quad [W]
\]

where:
- \( R_{\text{rad}} = 1/UA_{\text{rad}} \) [K/W]
- \( C_{\text{bldg}} = \text{thermal inertia of the building according to Eq. 2.24, [kJ/K] } \)

\[
C_{\text{bldg}} = \frac{Q_{\text{bldg}}}{8760 \cdot 17 - t_{\text{out}} \cdot \tau_{\text{c;bldg}}} \cdot 3600 \quad [kJ/K]
\]

where:
- \( Q_{\text{bldg}} = \text{Annual heating demand [kWh]} \)
- \( \tau_{\text{c;bldg}} = \text{Time constant of the building [h]} \)

The energy balance for the heating system and heat pump can be written as:

\[
\dot{Q}_{\text{hp}} \cdot d\tau = \dot{C}_{w} \cdot (t_{\text{wo}} - t_{\text{wi}}) \cdot d\tau = C_{\text{rad}} \cdot dt_{\text{wt}} + \frac{\dot{Q}_{\text{rad}}}{R_{\text{rad}}} \cdot d\tau \quad [J]
\]

where:
- \( C_{\text{rad}} = \text{Thermal inertia of the heat emitting system, in this case the radiator system [J/K]} \)
For steady-state conditions the energy balance is reduced to:

\[ \dot{C}_a \cdot (t_{ao} - t_{ai}) = \dot{C}_w \cdot (t_{wo} - t_{wi}) = \frac{\theta_{rad}}{R_{rad}} \quad [\text{W}] \quad \text{Eq. 2.26} \]

where:
- \( \dot{C}_a \) = Equivalent air heat capacity flow rate representing the total heat exchange \([\text{W/K}]\)
- \( t_{ai} \) = Inlet air temperature \([\circ\text{C}]\)
- \( t_{ao} \) = Equivalent air outlet temperature \([\circ\text{C}]\)

Eq. 2.23 to Eq. 2.26 make it possible to calculate the temperature change per time step of the indoor air and of the heating system. Using Eq. 2.27 to Eq. 2.31 and the information in section 2.2 makes it possible also to calculate the temperature changes in the heating system.

Temperature change in the radiator system:

\[ \Delta t_{wi} = \frac{\dot{Q}_{hp} - \theta_{rad}}{C_{rad} \cdot R_{rad}} \cdot \Delta \tau \quad [\text{K}] \quad \text{Eq. 2.27} \]

Temperature change of the indoor air:

\[ \Delta t_{ai} = \frac{\theta_{rad} - \dot{Q}_{bldg}}{R_{rad} \cdot C_{bldg}} \cdot \Delta \tau \quad [\text{K}] \quad \text{Eq. 2.28} \]

Outlet water temperature from the heat pump:

\[ t_{wo} = t_{wi} + \frac{\dot{Q}_{hp}}{C_w} \quad [\circ\text{C}] \quad \text{Eq. 2.29} \]

Warm air outlet from the radiator system:

\[ t_{ao} = t_{ai} + \frac{\theta_{rad}}{R_{rad} \cdot C_a} \quad [\circ\text{C}] \quad \text{Eq. 2.30} \]

Return water temperature to the heat pump:

\[ t_{wi} = t_{wi} - \frac{\dot{C}_w}{C_w} \cdot (t_{ao} - t_{ai}) \quad [\circ\text{C}] \quad \text{Eq. 2.31} \]
3 Testing and calculation of seasonal performance using steady-state models

Results from laboratory tests on two variable-speed controlled ground-source heat pumps are presented in this chapter. With data from the tests the seasonal performance factor of these heat pumps are calculated, according to the steady-state model in chapter 2.2, and compared to the results from the same analysis on a state-of-the-art heat pump with a single-speed compressor. This chapter is based on paper no. 1.

3.1 Introduction

Heat pumps for domestic use are connected to loads which vary continuously, and so it must be possible to adjust their capacity. Conventionally, this is done by intermittently switching the heat pump on and off. However, there are other methods of capacity control, with continuously variable-speed control being the most efficient, as described by Pereira\cite{64}, Qureshi\cite{72} and Wong\cite{88, 89}, Miller\cite{58}, Tassou\cite{82}, Halozan\cite{32} and Poulsen\cite{68}, among others, have previously investigated variable-speed control of heat pumps and compared it to conventional intermittent control. Improved energy efficiency, in the range of 10-25\% compared to intermittent operation is reported by Miller\cite{58}, Tassou\cite{81-82}, Lande\cite{52}, Marquand\cite{57} and Bergman\cite{7}. According to Miller\cite{58} and Garstang\cite{29} these savings are due to better performance at part load through heat exchanger unloading and lower supply temperature, higher compressor efficiencies, fewer on/off cycles, reduced need for supplementary heating and reduced need for defrosting. However, the majority of these investigations, except for Poulsen’s, have been concerned only with air source heat pumps, mostly air-to-air units. Ground source heat pump systems are used in buildings with a relatively large annual heat demand, and hence such systems have a greater potential for energy savings. In addition, they are usually sized to cover a large part of the total annual energy demand, and therefore operate at part load for most of the time. However, these systems do not benefit from the reduced need for defrosting, and the effect of unloading of heat exchangers will normally not be as great as for air source units, due to the use of highly efficient plate heat exchangers.

Consequently, the objective of this study was to investigate the energy-saving potential of applying variable-speed capacity control to brine-to-water heat pump systems. The work was carried out both by theoretical analysis and by laboratory tests, to create input data for calculating the seasonal performance factor.

3.2 Background

The heating demand of a building depends on the outdoor climate, indoor climate requirements and the building design. How efficiently the required heat can be supplied to the building is dependent on the design, efficiency and control strategy of the heating system and, of course, on the efficiency of the heating equipment,
in this case the heat pump. If the heat pump operates intermittently, it must supply heat at a higher temperature than necessary when in the on-mode in order to reject sufficient heat during the entire on/off cycle: see Figure 1.5. This will cause the heat pump to operate at an unnecessarily high condensing temperature. With variable-speed operation, so that the heat pump supplies exactly the capacity needed, the condensation temperature can be lowered and thus the efficiency increased, as shown below in chapter 3.3.

The ability to adjust the capacity to the load decreases the number of on/off cycles. Cycling the compressor on and off may cause refrigeration system losses and wear, and hence a reduced cycling frequency can reduce losses and increase life expectancy. However, as discussed by Bergman[7] and Karlsson[44], for the type of heat pumps discussed here, the losses due to cycling are negligible, mainly due to the short duration of the start-up compared to the time in operation, and the use of thermostatic expansion valves which reduce refrigerant migration.

If the heat pump is not sized to cover the maximum heat load, a supplementary heat source must be installed. This is often the case in Sweden, where heat pumps for domestic use are sized to cover 55-60% of the load at the design outdoor temperature (DOT), which will cover 85-90% of the annual energy demand for heating. The ability to extend the operating range of the compressor by variable-speed control provides an opportunity to reduce or to avoid the need for supplementary heat. In the calculations described in this chapter and in chapter 4, the heat pumps cover 92% of the annual energy demand when they are designed to cover 50% of the load at DOT.

3.3 Theory

The following sections outline, by theoretical analysis, why variable-speed capacity control can contribute to increased energy efficiency of heat pump systems.

3.3.1 Unloading of heat exchangers

Reducing the capacity of the compressor will cause the condensation pressure to be reduced and thus “unload” the heat exchanger. Assuming that the thermal efficiency of the condenser can be expressed by the $\varepsilon$-NTU method for a counter-flow heat exchanger we, according to Fahlén[25], have:

$$
\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{1 - \exp(-\text{NTU} \cdot (1 - R))}{1 - R \cdot \exp(-\text{NTU} \cdot (1 - R))}; \quad R = \frac{C_{\text{min}}}{C_{\text{max}}}; \quad \text{NTU} = \frac{U \cdot A}{C_{\text{max}}}
$$

where:

- $\varepsilon$ = Heat exchanger effectiveness [-]
- $U$ = Total thermal transmittance [W/(m$^2$K)]
- $A$ = Heat exchanger area [m$^2$]
- $\dot{C}$ = Heat capacity flow rate [W/K]
Furthermore, the heat capacity of the condensing refrigerant will be considered as the maximum with approximately zero temperature change. This is equivalent to an infinite heat capacity flow rate, and thus $R \approx 0$. This assumption, together with the relationship $\dot{Q} = \dot{C} \cdot \Delta t$, makes it possible to express the condensation temperature $T_1$ as:

$$T_1 = T_{wi} + \frac{\dot{Q}_1}{\dot{C}_w \cdot \left(1 - \exp\left(-\frac{U_c \cdot A_c}{\dot{C}_w}\right)\right)} \quad [K]$$  \hspace{1cm} \text{Eq. 3.2}

where:
- $T_{wi}$ = Inlet water temperature to the heat pump [K]
- $\dot{C}_w$ = Heat capacity flow rate of the water in the heating system [W/K]
- $\dot{Q}_1$ = Heating capacity [W]

Eq. 3.2 indicates approximately how much the condensing temperature $T_1$ will decrease if the output heating capacity (i.e. the heat delivered to the hydronic system) is reduced, while the return temperature from the heating system and the thermal transmittance of the condenser are assumed unchanged. Using Eq.3.1, and the same line of reasoning as for Eq. 3.2, the expression for the evaporating temperature becomes:

$$T_2 = T_{bi} - \frac{\dot{Q}_2}{\dot{C}_b \cdot \left(1 - \exp\left(-\frac{U_c \cdot A_c}{\dot{C}_b}\right)\right)} \quad [K]$$  \hspace{1cm} \text{Eq. 3.3}

where:
- $T_{bi}$ = Inlet brine temperature to the heat pump [K]
- $\dot{C}_b$ = Heat capacity flow rate of the brine in the borehole system [W/K]
- $\dot{Q}_2$ = Cooling capacity [W]

As $\dot{Q}$ decreases, so does $\dot{Q}_2$, and thus the evaporating temperature increases as the capacity is reduced. The $COP_{hp}$ will increase when the evaporating temperature increases by $\Delta T_2$ and the condensing temperature is lowered by $\Delta T_1$. This can be expressed as by Fahlén$^{[23]}$:

$$\frac{\Delta COP_{hp}}{COP_{hp}} = \left[\frac{\Delta T_2}{T_1 - T_2} - \frac{T_2}{T_1} \cdot \frac{\Delta T_1}{T_1 - T_2}\right] \quad [-]$$  \hspace{1cm} \text{Eq. 3.4}

However, the reduction in capacity must be performed such that the efficiency of the compressor is not degraded.
3.3.2 Compressor efficiency

By changing the compressor speed, \( n_{\text{comp}} \), the refrigerant mass flow rate, \( \dot{M}_R \), is changed:

\[
\dot{M}_R = \left( \frac{n_{\text{comp}}}{60} \right) \cdot V_{sw} \cdot \eta_v \cdot \rho_4 \quad [\text{kg/s}]
\]

Eq. 3.5

where:
- \( V_{sw} \) = Swept volume of the compressor \([\text{m}^3]\)
- \( \eta_v \) = Volumetric efficiency \([-\]
- \( \rho_4 \) = Density of refrigerant vapour at compressor inlet \([\text{kg/m}^3]\)

As a result of the changed compressor speed the capacity, \( \dot{Q}_l \), of the heat pump will change. The power, \( \dot{W}_{\text{comp}} \), required by the compressor for delivering this capacity can be expressed as:

\[
\dot{W}_{\text{comp}} = \frac{\dot{Q}_l}{\text{COP}_R \cdot \eta} = \dot{M}_R \cdot \frac{h_{3is} - h_4}{\eta} \quad [\text{W}]
\]

Eq. 3.6

where:
- \( \eta \) = the compressor efficiency \([-\]
- \( \text{COP}_R \) = COP for the cycle with isentropic compression \([-\]
- \( h_{3is} \) = Specific enthalpy of refrigerant after isentropic compression \([\text{J/kg}]\)
- \( h_4 \) = Specific enthalpy of refrigerant entering the compressor \([\text{J/kg}]\)

The compressor efficiency is a product of a number of efficiencies and can be expressed as:

\[
\eta = \eta_{el} \cdot \eta_{mc} \cdot \eta_{is} \cdot \eta_{m,el} \quad [-]
\]

Eq. 3.7

For hermetic compressors, the efficiency of the transmission between the compressor and the compressor motor, \( \eta_{mts} \), will hardly be affected by the variable-speed control. The mechanical, \( \eta_{mc} \), and isentropic, \( \eta_{is} \), efficiencies should increase with reduced compressor speed. How the efficiency of the electric motor, \( \eta_{m,el} \), changes with changed compressor speed will depend on the motor type, the frequency converter technique and how the frequency converter and the motor are integrated. Some of these issues have previously been noted by Qureshi\(^{72}\).
3.3.3 Matching of media flows

By differentiating Eq. 3.3 according to Fahlén\(^{[23]}\) we get:

\[
\frac{\Delta \dot{W}_{hp}}{W_{hp}} = -\frac{\Delta T_2}{T_{hi} - T_2} = \frac{\Delta Q_2}{Q_2} - \frac{\Delta C_b}{C_b} \left[ 1 + \frac{(1 - n) \cdot NTU_{e}^2}{\exp(NTU_{e}^2) - 1} \right]
\]

Eq. 3.8

The brine side heat transfer coefficient \(\alpha_b\) is assumed to vary with the flow rate according to \(\alpha_b \propto \dot{V}_b^{m_b}\). The relationship between power and flow rate for pumps is:

\[
\dot{W}_p = \frac{\dot{V} \cdot \Delta p}{\eta_p} \quad \text{[W]}
\]

Eq. 3.9

The pressure difference, \(\Delta p_b\), will depend on the flow rate according to \(\Delta p_b \propto \dot{V}_b^{m_b}\).

Differentiating Eq. 3.9 gives:

\[
\frac{\Delta \dot{W}_{hp}}{W_{hp}} = (m_2 + 1) \cdot \frac{\Delta \dot{V}_b}{\dot{V}_b} - \frac{\Delta \eta_{hp}}{\eta_{hp}} \approx (m_2 + 1) \cdot \frac{\Delta \dot{C}_b}{C_b} - \frac{\Delta \eta_{hp}}{\eta_{hp}}
\]

Eq. 3.10

Changing the capacity of the heat pump also changes the optimum flow rate of the heat transfer fluids. It will pay off to increase the flow rate as long as the decrease in drive power to the compressor is greater than the increase in drive power to the pump. At the optimum point, these changes are equal, i.e. Eq. 3.8 equals Eq. 3.10. By assuming constant efficiency, we get:

\[
\frac{\Delta \dot{C}_b}{\dot{C}_b} = \frac{\Delta \dot{Q}_2}{\dot{Q}_2} \cdot \left[ 1 + \frac{(1 - m_1) \cdot NTU_{e}^2}{\exp(NTU_{e}^2) - 1} - (m_2 + 1) \cdot \frac{\dot{W}_{hp}}{W_{hp}} \right]^{-1}
\]

Eq. 3.11

Eq. 3.11 shows that, as previously stated, a change in capacity also necessitates changes in heat transfer media flow rates in order to optimise the efficiency. The magnitude of the change will depend on the heat exchanger characteristics and on the efficiencies of the pumps and the compressor.

3.4 Evaluated heat pumps

Two heat pumps, A and B, were specially designed for this project. Both were designed as ground source heat pumps typically are in Sweden, i.e. an indirect system with plate heat exchangers as condenser and evaporator, thermostatic expansion valve and refrigerant R407C. In heat pump A an electronic expansion valve was mounted parallel to the thermostatic valve. Results from this
comparison have been reported by Karlsson\textsuperscript{[48]}. Figure 3.1 is a schematic of heat pump A and Figure 3.2 a photo of heat pump B. The component lists for these heat pumps are given in Table 3.1 and Table 3.2. To provide a reference for comparison with today’s technology, a standard single-speed ground source heat pump, available on the market, was also tested. It is called heat pump C and the component list is given in Table 3.3. A fourth heat pump, called D, was further included in the analysis. It is a fictitious heat pump having the characteristics of heat pump B, but given the same efficiency as the standard heat pump C at the nominal frequency 80 Hz.

\textbf{Figure 3.1} Schematic of heat pump A. Heat pumps B and C are designed similarly but without the EEV and for heat pump C without the frequency converter

\textbf{Figure 3.2} Photo of prototype heat pump B.
Table 3.1 Component list for heat pump A

<table>
<thead>
<tr>
<th>Component</th>
<th>Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Bristol Inertia, H25B32QDBE (piston)</td>
</tr>
<tr>
<td>Condenser</td>
<td>Cetetherm CP 415-26 plates</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Cetetherm CP 615D-26 plates</td>
</tr>
<tr>
<td>Thermostatic expansion valve</td>
<td>Danfoss TUAE</td>
</tr>
<tr>
<td>Electronic expansion valve</td>
<td>Danfoss AKV 10-7</td>
</tr>
<tr>
<td>Controller for the EEV</td>
<td>Danfoss AKC 114A</td>
</tr>
<tr>
<td>Frequency converter</td>
<td>Danfoss VLT 2830</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>R407C (1.4 kg)</td>
</tr>
</tbody>
</table>

Table 3.2 Component list for heat pump B

<table>
<thead>
<tr>
<th>Component</th>
<th>Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Mitsubishi AEV60F (scroll)</td>
</tr>
<tr>
<td>Condenser</td>
<td>CP 415-50 plates</td>
</tr>
<tr>
<td>Evaporator</td>
<td>CP 415-60 plates</td>
</tr>
<tr>
<td>Thermostatic expansion valve</td>
<td>Danfoss TCAE, 068U4325</td>
</tr>
<tr>
<td>Frequency converter</td>
<td>Danfoss VLT</td>
</tr>
<tr>
<td>Refrigerant (charge)</td>
<td>R407C (2.5 kg)</td>
</tr>
</tbody>
</table>

Table 3.3 Component list for heat pump C

<table>
<thead>
<tr>
<th>Component</th>
<th>Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Bristol H75A42QDBE (piston)</td>
</tr>
<tr>
<td>Condenser</td>
<td>Cetetherm (50 plates)</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Cetetherm (50 plates)</td>
</tr>
<tr>
<td>Thermostatic expansion valve</td>
<td>Honeywell, FLICA TLEX-3,5</td>
</tr>
<tr>
<td>Refrigerant (charge)</td>
<td>R407C (2.4 kg)</td>
</tr>
</tbody>
</table>

In summary the heat pumps are designed as follows:

A: Variable-speed piston compressor with operating range 30-75 Hz. Components are designed for the capacity at 50 Hz (considered as nominal compressor speed). Prototype heat pump.

B: Variable-speed scroll compressor with operating range 30-120 Hz. Components are designed for the capacity at 80 Hz (considered as nominal compressor speed). Prototype heat pump.

C: Standard ground-source heat pump with single-speed compressor

D: Fictitious heat pump. As B but the efficiencies are scaled such that the heat pump at 80 Hz has the same performance as C.
3.5 Method

In order to investigate how the previously mentioned advantages of variable-speed capacity control apply to ground source heat pumps, laboratory tests and seasonal performance calculations were performed on the four ground source heat pumps above.

3.5.1 Laboratory tests

Full capacity tests were performed in accordance with SS-EN 255-2\cite{76} and reduced capacity tests both for intermittent and variable-speed operation were performed in accordance with CEN/TS 14825\cite{10}. The low-temperature heat transfer medium was an ethylene glycol water solution (33 % by weight of ethylene glycol), and the high temperature heat transfer medium was water. The low temperature heat transfer medium will in this thesis be referred to as brine. For the full capacity tests the operating condition is given by the temperature of the brine entering the evaporator and the water going out of the condenser. For the reduced capacity tests the operating condition for the brine is the same as for the full capacity tests but for the water on the high temperature side the condition is given by the water entering the condenser. The inlet temperature is the same as the inlet temperature during the full capacity test. The flow rates of the heat transfer media were set according to specifications by the manufacturer who built the prototype heat pumps and were kept constant during the tests.

3.5.2 Seasonal performance

To determine the possible benefits from variable-speed capacity control it is of interest to analyse the annual performance. For this purpose, a steady-state calculation model based on the work by Fehrm\cite{25} was developed as previously described in chapter 2.2. It includes models of outdoor climate, building, heating system and the heat pump. Necessary input data are the heat transfer resistance for the building, the annual mean outdoor temperature, the temperature level of the heating system at DOT, the indoor temperature, internal heat gains and measured heating capacity and input electric power to the heat pump. Important to note here is that the return temperature from the heating system was considered to be constant within each time step of 24 hours, i.e. the dynamics of the heating system during intermittent heat pump operation was not fully considered. A correlation given by Fahlén\cite{20} between outdoor air temperature and the temperature in the borehole was used.

When the heat pump could not supply the capacity needed, an electric resistance heater (efficiency 95 %) provided the extra capacity. In addition, when the heat pump operates intermittently, the pumps for the brine side and heating system media flows were switched on and off simultaneously with the compressor. The pump for the radiator system was in continuous operation. The tested heat pumps have different capacities, but were scaled to equal capacities, as were also the water flows through the condenser and the capacity of the pumps. Furthermore, the water flow through the heating system and the heat pump were not the same.
The heat pump was connected to the heating system as shown in Figure 2.3 (right).

The calculations were made for a typical domestic building with an annual heating demand of 25 000 kWh. No domestic hot water demand was considered. The seasonal performance factor (SPF) was calculated for all three heat pumps, each of them sized to cover 50-100 % of the heating demand at DOT. This was calculated for two different heating systems; one Swedish standard system with temperature levels 55/45 °C at DOT, and one low-temperature system with temperatures 35/28 °C at DOT (supply/return temperatures).

### 3.6 Results and discussion

The results are based on measurements performed in SP’s accredited laboratory. The uncertainties of these measurements are derived according to EAL[14] and ISO GUM[39] with a coverage factor $k = 2$, which gives a confidence level of approximately 95 %. The following uncertainties apply:

\[
U_\varnothing = 1 \pm 2 \%, \ U_t < 0.1 K \text{ (water and brine)}, \ U_\Delta < 0.05 K \text{ (water and brine)}, \\
U_\varphi < 1 \% \text{ and } U_\varpi < 0.5 \%
\]

The uncertainty of measurement for the compressor efficiency, defined in Eq. 3.7, is shown in Figure 3.3 and Figure 3.4. As shown in the figures, the difference in compressor efficiency between different measurements is larger than the uncertainty of measurement thus making it possible to distinguish this as real changes in efficiency. Note that it is the reproducibility that should be analysed when comparing different measurements within the same measurement series. The reproducibility is lower than the uncertainties stated in Figure 3.3 and Figure 3.4 and thus the change in compressor efficiency is more significant than it may look.

### 3.6.1 Results from the laboratory tests

As shown in Table 3.4 and Table 3.5, the results from the laboratory tests of the two variable-speed heat pumps differ quite a lot. For heat pump A, with the conventional compressor, efficiency decreases when operated in a variable-speed mode, compared to intermittent operation. However, in the case of heat pump B, with a compressor designed for variable-speed operation, efficiency increases when operated in a variable-speed mode, when compared with intermittent operation. The operating points at part load are defined by the inlet temperatures of the heat transfer media to the evaporator and condenser. These temperatures are the same as for the full capacity tests, as previously described in chapter 3.5.1. The relative capacity of 63 % was chosen because it corresponds to the lowest allowable compressor speed of 30 Hz.
Table 3.4 The change in $COP_{hp}$ for variable-speed drive (vsd) compared with on/off control, for heat pump A.

<table>
<thead>
<tr>
<th>Operating point $t_{bi}/t_{wi}$ [°C]</th>
<th>5/26.5</th>
<th>0/27.7</th>
<th>0/27.7</th>
<th>5/42.7</th>
<th>0/44.0</th>
<th>0/44.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative capacity [%]</td>
<td>63</td>
<td>75</td>
<td>63</td>
<td>63</td>
<td>75</td>
<td>63</td>
</tr>
<tr>
<td>$COP_{hp}$(vsd)/$COP_{hp}$(on/off)</td>
<td>1.04</td>
<td>0.99</td>
<td>1.01</td>
<td>0.97</td>
<td>0.96</td>
<td>0.95</td>
</tr>
</tbody>
</table>

Table 3.5 The change in $COP_{hp}$ for variable-speed drive (vsd) compared with on/off control, for heat pump B.

<table>
<thead>
<tr>
<th>Operating point $t_{bi}/t_{wi}$ [°C]</th>
<th>5/26.6</th>
<th>0/27.7</th>
<th>0/27.7</th>
<th>5/42.1</th>
<th>0/43.1</th>
<th>0/43.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative capacity [%]</td>
<td>50</td>
<td>75</td>
<td>50</td>
<td>50</td>
<td>75</td>
<td>50</td>
</tr>
<tr>
<td>$COP_{hp}$(vsd)/$COP_{hp}$(on/off)</td>
<td>1.22</td>
<td>1.13</td>
<td>1.20</td>
<td>1.17</td>
<td>1.08</td>
<td>1.10</td>
</tr>
</tbody>
</table>

Eq. 3.2 and Eq. 3.3 imply that when operating at part load by variable-speed control, the condensing temperature should be lower and the evaporating temperature higher than when operating in intermittent part load operation. Measurements also show that this happens for both heat pumps, and so $COP_{hp}$ should increase by approximately 5-9 % for heat pump A and by 7-15 % for heat pump B. The fact that heat pump A shows a decrease in efficiency in variable-speed operation is explained by a decrease in the compressor efficiency, defined in Eq. 3.7. Test data show that the compressor efficiency decreases by 3-4 % for heat pump A at part load, while for heat pump B the corresponding compressor efficiency increases by 4-11 % as shown in Figure 3.3 and Figure 3.4.

![Figure 3.3](image.png)

Figure 3.3 The diagram shows the ratio between the compressor efficiency for heat pump A at part load operation and the efficiency at full load.
Following the line of argument in Section 3.3.2, I believe that the efficiency of heat pump A falls due to reduced efficiency of the compressor motor when connected to a frequency converter, i.e. the variable-speed drive. Such results have previously been reported by Qureshi\cite{Qureshi72}. It is worth noting that the absolute values of $COP_{hp}$ at the design speed hardly differ between the two variable-speed heat pumps. However, the conventional heat pump shows better efficiency, see Figure 3.5. The higher $COP_{hp}$ for the standard heat pump is caused by several factors. Firstly, the test points in Figure 3.5 show the performance at full load, when there is no benefit from variable-speed control. Instead, the performance is reduced due to the energy loss in the inverter. Secondly, the technique for standard, single-speed, compressors is optimised for its application. The two variable-speed controlled compressors do not represent an optimised product. The inverter used in these tests was not developed specifically for use with these compressors and the combined performance of inverter and compressor motor can probably be improved.

**Figure 3.5** The $COP_{hp}$ for heat pumps A, B and C at their nominal speeds.
3.6.2 Results from seasonal performance calculations

Considering the results from previous investigations, described in the introduction, it was expected that the variable-speed heat pumps would be more efficient than the standard heat pump. However, the results from the seasonal performance calculations, see Figure 3.6 and Figure 3.7, show that improvements of the variable-speed heat pumps are necessary in order for them to be more efficient than the standard heat pump. Heat pump C gives a \( SPF_{hs} \) which is 14 – 19 % higher than that of heat pump B. Furthermore, B does not give better annual efficiency than A, despite the better performance in part load operation. This result depends on a too low COP at the nominal speed, see Figure 3.5, and on the increased operating times for B. Long operating times lead to large energy inputs to pumps as their time in operation will also be longer, see Figure 3.8 and Figure 3.9. For the fictitious heat pump D, the seasonal performance is the same as for the standard heat pump C. The reason for not being better than C is obvious when comparing Figure 3.6 and Figure 3.7 - the energy input to pumps reduces the SPF more for D than for C. The two heat pumps have equal \( SPF_{hs} \) whereas \( SPF_{hp} \) for D is 5 – 8 % higher than for C. It has previously been shown that the pumps should be controlled when used together with a capacity controlled compressor. This will be made later in chapter 5 and the reason for not doing it already here is to separate the different effects occurring when applying capacity control.

![Figure 3.6](image)

**Figure 3.6** This diagram shows the seasonal performance factor, \( SPF_{hs} \), for the heat pump heating system for the four different heat pumps for increasing size at DOT. Valid for the 55/45 heating system.
Figure 3.7 This diagram shows the seasonal performance factor, $SPF_{hp}$, for the four different heat pumps for increasing size at DOT. Valid for the 55/45 heating system.

As outlined in the background section, it is common to use heat pumps in bivalent systems, and thus to use an additional heat source. One way of eliminating the need for supplementary heat is of course to size heat pumps to cover the maximum load. Variable-speed compressors offer a way of doing this without the drawbacks of short operating times and frequent switching on and off. A comparison of the heat pumps, all sized to cover the full load, gives the results shown in Table 3.6. Still, the intermittently operated standard heat pump (C) has the highest $SPF_{hs}$; even though the difference from D is very small. Considering $SPF_{hp}$, D is still 5 – 6% more efficient than C. However, the real capacity controlled heat pumps (A and B) still have lower efficiencies.

Table 3.6 SPF for the heat pumps when covering the full load.

<table>
<thead>
<tr>
<th>Heat pump</th>
<th>SPF$_{hs}$ [-]</th>
<th>SPF$_{hp}$ [-]</th>
<th>Power coverage at DOT (nominal frequency) [%]</th>
<th>Heating system (supply/return temperatures at DOT) [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>2.7</td>
<td>3.2</td>
<td>90</td>
<td>55/45</td>
</tr>
<tr>
<td>B</td>
<td>2.7</td>
<td>3.3</td>
<td>70</td>
<td>55/45</td>
</tr>
<tr>
<td>C</td>
<td>3.2</td>
<td>3.8</td>
<td>100</td>
<td>55/45</td>
</tr>
<tr>
<td>D</td>
<td>3.1</td>
<td>4.0</td>
<td>70</td>
<td>55/45</td>
</tr>
<tr>
<td>A</td>
<td>3.4</td>
<td>4.2</td>
<td>87</td>
<td>35/28</td>
</tr>
<tr>
<td>B</td>
<td>3.5</td>
<td>4.4</td>
<td>71</td>
<td>35/28</td>
</tr>
<tr>
<td>C</td>
<td>4.0</td>
<td>4.9</td>
<td>100</td>
<td>35/28</td>
</tr>
<tr>
<td>D</td>
<td>3.9</td>
<td>5.2</td>
<td>71</td>
<td>35/28</td>
</tr>
</tbody>
</table>
3.6.3 Effect of auxiliary equipment

For a complete analysis of the heat pump system, the auxiliary equipment must be included, as indicated in the previous section. In this case, the auxiliary equipment is the electric resistance heater and the pumps used for circulating the high and low temperature heat transfer media, and the radiator pump. Eq. 3.12, derived by Fahlén\cite{23}, shows how changes in condensing and evaporating temperatures (via Eq. 3.4) and drive powers to pumps affect the overall $COP_{hps}$. From the last two terms in the equation, it can be concluded that the pumps will not normally affect the $COP_{hps}$ much at high capacities. However, as the capacity is decreased, the influence from the pumps will increase if they are not suitably controlled.

\[
\Delta COP_{hps} \approx \left[1 - \frac{\dot{W}_{wp} + \dot{W}_{bp}}{W_{hps}}\right] \cdot \Delta COP_{hp} \cdot \frac{COP_{hp} - 1}{COP_{hps}} \cdot \frac{\Delta \dot{W}_{wp}}{W_{hps}} - \Delta COP_{hp} \cdot \frac{\Delta \dot{W}_{bp}}{W_{hps}}
\]

Eq. 3.12

Figure 3.8 and Figure 3.9 show the total energy used by the heat pump system, divided among the individual components for heat pumps B and C. The share used by the pumps increases with increased size of the heat pump, due to the longer duration of part-load operation. Even when the heat pump is not in operation, the circulation pump for the radiator system is in continuous operation and thus degrades the energy efficiency. The share is also larger for heat pump B than for heat pump C, due to the variable-speed operation.

**Figure 3.8** The diagram shows the percentage of the drive energy over a full heating season that is used by the different components of heat pump B. The pumps are of the single-speed type, i.e the speed is not changed as the compressor speed is changed.
Figure 3.9 The diagram shows the percentage of the drive energy that is used by the different components of heat pump C.

The degradation of the heat pump system efficiency due to the pumps can be reduced in two ways: by increasing the pump and pump motor efficiencies or, for the variable-speed case, by adjusting the pump speed when the compressor speed is changed, as previously outlined in Eq. 3.11. Typically, pumps in these applications have efficiencies around 10%. If the efficiency is doubled, the $SPF_{hs}$ for heat pump B would increase by 4-9%, which is considerable. It is also larger than for C, which would only gain 2-3%. To double the efficiency may seem a tough challenge, but such pumps are available from the pump manufacturers. The market demand, however, is low according to Fahlén[21].

### 3.7 Conclusions

Two heat pumps, with the compressors of each being operated in variable-speed mode and in intermittent operation (on/off) mode, have been evaluated by laboratory tests. The results differed between the two units. The theory outlined in this chapter, plus the measured temperatures and pressures in the refrigerant circuit, indicate that variable-speed operation should increase the $COP_{hp}$ by 5-9% for heat pump A, and by 7-15% for heat pump B, in comparison with intermittent operation. However, the measurements reveal that $COP_{hp}$ is instead reduced for heat pump A, whereas for heat pump B, $COP_{hp}$ increases as expected. The difference in results probably depends on whether the compressor motor was designed for variable-speed operation or not.

The seasonal performance calculations, however, show that despite this improvement in COP at part load the annual efficiency is still lower for the variable-speed operated heat pumps than for the intermittently operated heat pump. Even if the variable-speed heat pump at its nominal speed was given the efficiency of the standard heat pump, the seasonal efficiency is still not higher than for the single-speed heat pump. The main reasons for this are the efficiency of the variable-speed compressors, and the efficiency and control of the pumps for the ground collector system and the heating system. Considering the seasonal...
performance of the heat pump only, heat pump D (variable-speed with improved baseline efficiency), had a higher efficiency than the intermittently operated heat pump. Furthermore, it was shown that the drive energy for pumps is higher for the variable-speed heat pumps than for the intermittent heat pump. Thus increased efficiency and control of the pumps are important in order to improve system efficiency.

This investigation was made on prototype heat pumps, and higher efficiency may be achieved by further development of compressors, compressor motors, motor controls and pumps. Variable-speed operation also provides the opportunity to remove the supplementary heat source commonly used with Swedish heat pumps.

Although the focus for this chapter was on ground source heat pumps the results are valid also for air source heat pumps, where further benefits are possible in reduced needs for defrosting of the evaporator. Finally, the theory presented in this chapter provides a method of quickly approaching the optimum relation between compressor speed and pump speed in systems with automatic capacity control. This is further discussed in chapter 6.
4 Impact of design and thermal inertia of the heating system on the energy saving potential – dynamic models

In this chapter the analysis from chapter 3 is further treated by applying a model that also considers the dynamics of the temperature changes in the heating system. Using this transient model, described in chapter 2.3, the effect of thermal inertia and controller dead band etc. on seasonal performance is analysed. This chapter is based on paper no 2.

4.1 Introduction

Continuous capacity control of heat pumps connected to continually varying loads gives possibilities for improved energy efficiency compared to the conventional intermittent capacity control. For domestic heat pump systems variable-speed control have shown promising results as described e.g. by Pereira\cite{64}, Qureshi\cite{72} and Wong\cite{88}; Wong\cite{89} and is also commonly used, even though other techniques such as compressor short-cycling also show promising results according to Ilic\cite{38} and Poort\cite{65}. Variable-speed control of heat pumps has previously been investigated and compared to conventional intermittent control in several investigations, and efficiency increases in the range from zero to twenty percent and above are reported by Miller\cite{58}, Tassou\cite{81}, Halozan\cite{32} Poulsen\cite{68} and Karlsson\cite{47}. According to Miller\cite{58} and Garstang\cite{29} these savings are due to better performance at part load through heat exchanger unloading and lower supply temperature, higher compressor efficiencies and reduced need for supplementary heating and reduced need for defrosting. In the majority of studies where the seasonal performance of heat pump systems is calculated, the dynamics present in a hydronic heating system is not considered. Instead the supply and return temperatures of the heating system are considered constant in each calculation step, i.e. a heating system with infinite thermal inertia. Using this approach the performance of an intermittently controlled heat pump system is overestimated because excluding the dynamics causes the supply and return temperatures to be lower than in reality. Thus, in order to get a more fair comparison, a method for calculating the seasonal performance considering system dynamics was developed.

The purpose of this study was to investigate the energy saving potential of variable-speed capacity controlled heat pumps in hydronic heating systems and to compare with intermittently controlled heat pumps. The influence of thermal inertia, lag time and connection principles etc. on the results are also investigated.
4.2 Methodology

The methodology used was to first measure the performance of three ground-source heat pumps, see section 3.4, and then to use these performance data as input to a model where the seasonal performance of the entire heat pump heating system was calculated.

The method for calculating the seasonal performance is based on a model by Fehrm\textsuperscript{[25]} to which the transient model by Bergman\textsuperscript{[6]} was added, see chapter 2. The reason for choosing the model by Bergman was mainly because it has been compared to laboratory measurements with good agreement. The heating system is considered as one single unit instead of, as in reality, consisting of several units. It makes the analysis easier and as the intention of this paper is to investigate the general principles rather than to design a specific system, the complexity level is sufficient. Bergman’s model only considers radiators and thus models for floor heating and fan-coils were added.

Energy losses due to ventilation are included in the building’s total heat transfer resistance. But when considering thermal inertia only the inertia of the building is included, not the much smaller inertia of the ventilation air flow. Thus the indoor air temperature drops more slowly than it would in reality and thus the number of on and off cycles of the heat pump will be underestimated. Radiator and fan-coil systems were designed according to manufacturer specifications and thermal inertias of these systems were calculated from catalogue data (Carrier\textsuperscript{[9]} and Thermopanel\textsuperscript{[84]}). For floor heating systems it is more difficult to estimate the thermal inertia as it will depend on cycling frequency of the heat pump. For short times in operation only a small part of the floor construction around the pipe will be heated and thus be included in the thermal inertia. For longer operating times of the heat pump a larger part of the floor will be heated and thus be part of the thermal inertia. The situation will be more complicated if also considering different types of floor materials. In order not to make the model unnecessarily complex a first estimate was that the entire slab was part of the heating system’s thermal inertia.

The dynamics of the heat source system, i.e. the bore hole system, was not considered. The temperature of the brine entering the evaporator is considered constant for each calculation step of 24 hours. The annual temperature profile, correlating outdoor air temperature and brine temperature, follow the measurements by Fahlén\textsuperscript{[20]}.

4.3 Calculation model

The model was developed based on the work by Bergman\textsuperscript{[6]}, and is further described in chapter 2. It was used for calculating the seasonal performance of heat pump heating systems for three different types of system; radiators, floor heating and fan-coils. Two temperature levels of the heating system were evaluated; 55/45 and 35/28 (supply/return temperatures at DOT). For radiators
and convectors both 55/45 and 35/28 were evaluated, whereas for floor heating only 35/28 was evaluated.

To calculate the intermittent operation, the controller dead band, designated DIF, of the thermostat must be decided as well as the stop temperature, TST. In Figure 4.1 the controller dead band is 5 K. This is an input to the calculation procedure. The stop temperature is not fixed as an input. It is calculated iteratively by the program, as it must be chosen such that the energy balance between the heating system and the building is fulfilled, i.e. the heating system supplies the energy demanded for space heating. In real operation the TST value is set by curve control as explained in section 1.5.

Figure 4.1 DIF designates the controller dead band and TST the stop temperature.

### 4.4 Results and discussion

The results and discussion section is divided in three parts, where the first compares the results from the calculation method presented in this chapter to results from the calculation method in chapter 3. The second part covers different effects which are linked to the design of the heating system and the third part covers effects linked to the design of the heat pump and its controls.

If nothing else is stated the following conditions apply for the calculations:

- A space heating demand is present until the outdoor air temperature exceeds +11 °C.
- The calculations were made in steps of 24 hours under which the outdoor climatic conditions were considered constant.
- The building has an annual space heating demand of 25 MWh per year and the average annual outdoor air temperature is +6 °C.
- 35/28 and 55/45 designate the supply and return temperatures of the heating system at the design outdoor air temperature (DOT).
• During each 24 hours and during intermittent operation the calculations were made in steps of 5, 10 or 30 seconds depending on the inertia of the system
• The heat pump is connected in parallel (see Figure 4.5) to the heating system. The pumps of the heat pump system, i.e. the pump for the bore hole and the circulator for the space heating side, are running simultaneously with the compressor. The pump for the space heating system is in continuous operation throughout the heating season
• When the heat pump can not match the load, the lacking capacity is supplied by an electric resistance heater with an efficiency of 95 %.
• The evaluated heat pumps are in reality of different capacities but are scaled to equal capacity at DOT. Heat transfer flows and power used by pumps are scaled with the heat pump capacity such that all heat pumps with the same capacity at DOT have pumps of the same capacity
• The controller dead band for radiators and fan-coils is 5 K and for 35/28 radiator system and floor heating it is 2 K
• The thermal inertia of the condenser is neglected

The chosen combination of space heating demand and start of heating season does not resemble the average situation for a typical Swedish building. For the heating demand 25 MWh the start of the heating season should have been +17 °C to resemble the average. The chosen combination results in a larger portion of the heating demand occurring at lower outdoor air temperatures. However, this situation is the same for all cases evaluated in this thesis and will thus not affect the main results and conclusions.

4.4.1 Comparison to the results of the steady-state model

Figure 4.2 shows the SPF_{hs} for all four heat pumps and for the two calculation methods for a 55/45 radiator system. In the previous investigation, reported in chapter 3, the supply and return temperatures in the heating system were considered constant in each time step. As shown in Figure 4.2, regarding heat pump C, the steady-state model used in chapter 3 overestimates the seasonal performance factor for the intermittent operation mode. This derives from the higher supply temperature needed when operating intermittently compared to continuous operation as previously described in Figure 1.5. Figure 4.4 clearly shows that the temperature, t_s, needed by the heating system (radiators in this case) is considerably lower than the average temperature, t_{\text{ave, on}}, supplied by the heat pump during operation. If the heat pump had a variable-speed compressor it could have adjusted its capacity to meet exactly t_s and thus improve the efficiency.

When including the thermal inertia of the system, the optimal capacity of the intermittently controlled heat pump C is more pronounced and also lower. For the variable-speed heat pumps the change in result is very small, with the larger change for the heat pump with the smaller operating range. Considering the real variable-speed heat pumps they are, as in the previous investigation, not as efficient as the single-speed heat pump even though the difference now is smaller.
Table 4.1 gives the seasonal performance factor for the four heat pumps when covering 100% of the load at DOT with their maximum compressor speed. On a system level heat pump D is just 3% more efficient than C, but when excluding the pumps from the analysis D is 14% more efficient than C. The difference depends on the longer time in operation for the variable-speed heat pump and the consequently larger amount of energy used by the pumps. The total annual energy used by the pumps for heat pump C is 1200 kWh and for heat pump D it is 1800 kWh. This corresponds to 16% and 24% respectively of the total electricity use. This further emphasises the importance of using efficient pumps. Obviously, with continuous capacity control of the compressor the pumps should also be controlled.

Table 4.1  Seasonal performance factors for the heat pumps when covering the design load at maximum compressor speed. For heat pump D, this corresponds to 70% of the design load at nominal speed.

<table>
<thead>
<tr>
<th>Heat pump</th>
<th>SPF_{hs} [-]</th>
<th>SPF_{hp} [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>2.6</td>
<td>3.1</td>
</tr>
<tr>
<td>B</td>
<td>2.7</td>
<td>3.3</td>
</tr>
<tr>
<td>C</td>
<td>3.0</td>
<td>3.5</td>
</tr>
<tr>
<td>D</td>
<td>3.1</td>
<td>4.0</td>
</tr>
</tbody>
</table>

Figure 4.2  Comparison to the steady-state (SS) model of chapter 3. The capacity is valid at the nominal compressor speed. A, B and D are variable-speed. C is single-speed.
4.4.2 Impact of heating system design

The following sections deal with different aspects on how the design of the hydronic heating system affects the energy efficiency of heat pump systems and the comparison between intermittently and variable-speed controlled systems.

4.4.2.1 Thermal inertia

The thermal inertia of the heating system will affect the cyclic behaviour of the heat pump and the temperature level in the heating system. If the heating system has no thermal inertia and no lag time the response to changes will be immediate and the heat pump will be running at the design supply temperature. The other extreme is a system with infinite thermal mass and for this system the return temperature from the heating system will not change, and thus the heat pump will supply a constant temperature determined by the temperature lift of the heat pump.

Real heating systems will have different thermal inertias depending on design and type of system. For example, a floor heating system has a much larger thermal inertia than a fan-coil system. However, in this section only systems of the same type will be considered in order to compare only the different thermal inertias and not mix the results with different heat transfer characteristics.

Designing a radiator system with the temperature level 55/45 at DOT, according to manufacturer specifications (Thermopanel[^84]), the thermal inertia of the nominal system becomes approximately 543 kJ/K (only the radiators and their volume of water is considered, piping is excluded). This system is designated ‘middle’ below. This middle system is compared to two other systems with the same characteristics in all other aspects but the thermal inertia. ‘Light’ has a thermal inertia of one fifth of the middle system. The ‘heavy’ system is five times heavier than the middle system.

As shown in Figure 4.3 the thermal inertia has negligible influence on $SPF_{hs}$ for the variable-speed system. For the intermittently operated system the light system has a slightly higher $SPF_{hs}$ (3 %) than the other systems, and the point of optimal capacity is changed towards higher capacities. The better performance is due to a lower supply temperature during compressor operation as shown in Figure 4.4. The drawback of the light system is the frequent on and off switching, which for the 100 % capacity heat pump are as many as eleven starts per hour which is a lot more than what is normally allowed by the manufacturers. The lower average temperature of the light system depends on the curve form for the supply temperature that is not linear. After the first steep part the gradient decreases before the heat pump is shut off by the thermostat, see Figure 4.1. For a light system the last part will be shorter in time and the average temperature in operation is lower.

A similar analysis as the one previously described was also made for floor heating systems with the same conclusions.
Figure 4.3  Impact of thermal inertia on seasonal performance. D is variable-speed, and C is single-speed. The triangles for heat pump C in the “heavy” system does not show in the diagram, they lie behind the triangles for the “middle” system.

Figure 4.4  The figure shows the difference in mean supply temperature when the compressor is in operation. The figure is for heat pump C covering 100 % of the demand. The curves are jagged because the tolerance in the calculation procedure allows small temperature variations.
4.4.2.2 Connection principles

In domestic systems the heat pump is usually connected to the hydronic heating system in one of the two configurations, parallel and serial, shown in Figure 4.5. The parallel system requires two pumps, one in the heat pump and one in the heating system. The pump in the heat pump is switched off when the compressor is switched off. The serial connection requires only one pump, but it will be in constant operation throughout the heating season as it needs to provide flow to the heating system also when the compressor is not in operation.

Installing a storage tank in the by-pass line of the parallel design makes it possible to keep the flows at their optimum both for the heat pump and the heating system. The same principle applies also without the storage tank but then the return temperature to the heat pump will be affected by the flow configuration. If the flow through the heat pump is larger than the flow through the heating system the return temperature will be higher than when in the serial connection. If the flow through the heating system is larger than the flow through the heat pump the supply temperature to the heating system will be lower. Especially in retrofit installations this is hard to control and the relation between the flows will depend on the capacity of the circulator in the heat pump in relation to that of the heating system (a low capacity heat pump has a lower optimal flow than a high capacity heat pump).

The serial connection requires only one circulator and there are no flow short cuts. However, the flow in the heating system will be a compromise between the heat pump and the heating system, and the circulator in the heat pump must generally have a larger capacity than in the parallel case.

The impact on the seasonal performance factor will depend on how the circulators are controlled and the design of the heat pump capacity to the load. Using variable-speed circulators that are controlled to keep the optimum flow will very likely make the system performance less dependent on design.

![Figure 4.5 Two common connection principles, parallel (left) and serial (right)](image-url)
4.4.2.3 Type of heating system

Hydronic systems for domestic use are dominated by radiators, even though for new construction, especially in combination with heat pumps, floor heating is very common. A third alternative is hydronic fan-coils which are very compact. Fan-coils are used to increase the heat transfer capacity when retrofitting e.g. an oil burner with a heat pump. However, they are also interesting when changing the heating system in houses with direct-acting electric room heaters. Fahlén[23] made a comparison between two radiators and one fan-coil considering the linear capacity density (UA/L, [W/K/m]). The fan-coil had a value of 160 W/K/m whereas the radiator values were 15 W/K/m and 27 W/K/m. Thus the convector in this sense is 5-10 times more compact. The thermal inertia for the fan-coil and the radiators were presented as a time constant. For the fan-coil the time constant was 1 minute and for the radiators it was 23-26 minutes, showing the large difference in thermal inertia. Floor heating systems have a lot higher thermal inertia than the other systems. However, it depends on how it is laid out and how the heat pump is connected to the system, as discussed in the methodology part of this paper.

Figure 4.6 shows the result from calculations where heat pump B has been operating in all three systems designed for the temperature level 35/28. The radiator and fan-coil systems have been designed from catalogue data, and thus thermal inertias are different. For the floor heating system it has been assumed that the entire slab takes part in the process. The thermal inertia data are given in the figure caption.

![Figure 4.6](image)

**Figure 4.6** Seasonal performance for heat pump B in three different heat pump systems designed for 35/28 at DOT. The thermal inertias are 1900 kJ/K, 38 kJ/K and 32 800 kJ/K respectively for radiators, fan-coils and floor heating

The radiator system gives an SPF_{hs} that is approximately 2 % lower than for the other systems. The difference is almost constant and does not depend on capacity or thermal inertia. The difference in result derives from the fact that the heat transmission coefficient drops more for radiators than for fan-coils (almost constant heat transfer coefficient) or floor heating (large part of the heat transfer
via radiation) when the supply temperature is reduced. Radiators transfer heat to the room both via natural convection and radiation, the transfer coefficients which both decrease with temperature. This causes the heat transfer capacity to change more than for fan-coils when the temperature level of the heating system is changed. The change in heat transfer capacity can be illustrated by comparing the steady-state supply and return temperatures required by the different heating systems for the same conditions. It shows that the radiator system require higher temperatures than the fan-coil and floor heating system, see Figure 4.7. The difference between radiators and fan-coils and floor heating also increases as the temperature in the heating system decreases.

![Graph showing supply and return temperatures in radiator (full line) and fan-coil (dotted line) heating systems. The temperatures for floor heating is not shown as they coincide with the temperatures of the fan-coils.](image)

**Figure 4.7** Supply and return temperatures in radiator (full line) and fan-coil (dotted line) heating systems. The temperatures for floor heating is not shown as they coincide with the temperatures of the fan-coils.

If including also the fan power of the fan-coil in the above calculations the $SPF_{hs}$ for the fan-coil system drops by 2 % and then there is no difference in energy efficiency between radiators and fan-coils.

### 4.4.2.4 Lag time

Lag time means the time it takes for the water to travel once through the heating system. The major part of this transport time is due to the volume of the heat emitters, i.e. radiators and fan-coils, even though the piping of course also plays a role. The lag time is added to the model as situated after the heating system. In reality the lag time is distributed but for simplicity it is considered to be situated in one spot. No lag time is considered in the by-pass line.

Figure 4.8 shows the resulting $SPF_{hs}$ for heat pumps B and C for a lag time of 10 minutes in a 55/45 radiator system. For the variable-speed heat pump the lag time has no effect on the seasonal performance, but for the intermittently controlled heat pump the lag time has a positive effect on the SPF. The effect of the lag time is increasing for increasing capacities, and for the 100 % capacity the
lag time increases the $SPF_{hs}$ by approximately 5 %. There is a practical limit to the lag time considering the length of piping etc., and the 10 minutes used here is in the upper end. Changing the lag time to 7 minutes, a reduction by 30 %, only reduces the $SPF_{hs}$ by 1 % for the 100 % capacity heat pump. It is not possible to change the lag time more downwards as the volume of the radiators corresponds to an approximate lag time of 6 minutes. The $SPF_{hs}$ increases with increasing lag time because the average supply temperature is reduced during the compressor on time as shown in Figure 4.9 and Figure 4.10.

![Figure 4.8](image)

**Figure 4.8** Influence of lag time on the seasonal performance factor. The lag time was fixed to 10 minutes.

![Figure 4.9](image)

**Figure 4.9** Comparison of average supply temperature for a radiator system with no lag time and 10 minutes lag time. The figure is for heat pump C, covering 100 % of the demand.
In Figure 4.10 the operation of the heat pump stops at step 85 for the system with lag time. Because of the lag time the return temperature from the heating system continues to increase until 10 minutes after the stop (step 155). That is the explanation for the two peaks of the supply temperature for the system with lag time. Also the plateau in the beginning of the cycle is due to the lag time, the heat pump will receive an almost constant return temperature during these 10 minutes.

4.4.3 Impact of heat pump control

Domestic heat pumps often use what is known as curve control, where the desired temperature level of the heating system is correlated to the outdoor air temperature. This is an open-loop control where the gradient can be altered to fit the building characteristics. Instead of using a fixed controller dead band the heat deficit, often measured in degree minutes, can be used for controlling when to start and stop the heat pump. The deficit is measured from the selected curve as shown in Figure 1.4, and when it reaches a limit the heat pump starts. Once the deficit is zero the heat pump stops.

No matter which of the above methods that is used the limit values should be chosen such that the efficiency is as high as possible without deteriorating equipment life. To protect the compressor from too frequent starts and stops, the compressor is prohibited by the controls to restart before a certain time has elapsed since the latest start and/or stop. The effect of this will also be analysed below.

4.4.3.1 Controller dead band

In this thesis only the method of using a fixed temperature difference as controller dead band will be analysed. A very small controller dead band will make the
system approach a continuous capacity control, but will have an unrealistic intermittency with too frequent on and off switching. $SPF_{hs}$ increases with decreasing controller dead band even though the increase is very small, but as shown in Figure 4.11 the number of starts and stops per hour increases.

![Figure 4.11](image)

**Figure 4.11** Number of cycles per hour depending on the chosen controller dead band. Valid for heat pump C with 100% capacity at DOT in a 55/45 radiator system.

Calculations were made for controller dead bands down to 2 K, which was considered as the practical limit due to the inaccuracy of thermostats. The $SPF_{hs}$ improves by only 1% when changing from 5 K to 2 K controller dead band for this high-capacity intermittently controlled heat pump. It is a very small change which will be even smaller for a variable-speed heat pump as it only runs part of its operating hours intermittently. Thus the effect would be almost negligible for these heat pumps.

### 4.4.3.2 Restart time limitation

In addition to the controller dead band, the number of starts of the compressor is often controlled by a dedicated start frequency limiter. A common limitation for domestic ground-source heat pump systems is that 20 minutes must elapse between each start and 10 minutes between a stop and the next start. Inserting this limitation in the program without any other start limitations or lag times has no effect on the 55/45 radiator system, as the frequency was only approximately 2.5 cycles per hour anyway (see Figure 4.11 above). Even for the fan-coil system at 55/45, there is no difference in seasonal performance when introducing the restart time limitation. However, the room temperature drops as the system has a very low thermal inertia. The restart time limitation will not affect variable-speed heat pumps with reasonable operating range of the compressor speed.
4.5 Conclusions

Four heat pumps have been evaluated regarding their seasonal performance using a calculation method which considers the dynamic effects due to thermal inertia, lag time and controller dead band etc. The results from this model were compared with the results from a previous investigation by Karlsson\cite{karlsson2014}, see chapter 3, where a conventional steady-state model was used. This comparison shows that the performance of the intermittently controlled heat pump decreases by 0.5-7 % when using this new model, whereas the performance of the variable-speed controlled heat pumps is only affected by 0-4 %. For the heat pump with the compressor having the largest operating range the performance is not affected at all by this new calculation method. As a consequence, conventional methods will overestimate the performance of intermittently controlled heat pumps. The benefit from using compressors with variable-speed control will not be fully shown.

The effects from changes in thermal inertia, inclusion of lag time, and alternative controller dead band changes the result only in the order of a few percent. These parameters only affect the intermittently controlled heat pumps and will not change the conclusion about the difference in calculation methods.

Differences in performance due to connection principles will need a separate investigation before putting numbers to them. However, in principle the parallel connection should be less sensitive to design of the system than the serial connection. Parallel connection with a storage tank in the by-pass line will make it possible to independently optimise the heat transfer flow of the heating system and the heat pump. Using variable-speed circulators with proper control should also make the performance less sensitive to system design.

There was no great difference in result for the different heating systems. The radiator system gave a slightly lower efficiency than the other systems for the variable-speed heat pump. For intermittently operated systems the somewhat higher SPF for light systems will influence the results. As a consequence, with normal design, the radiator system will probably be as efficient as the floor heating system.

The temperature level will of course have a great impact on the SPF, irrespective of the complexity of the model. Changing the system from 55/45 to 35/28 increases the $SPF_{hs}$ by 30-35 % and thus the temperature level of the heating system is a much more important parameter than any of the others.

To conclude, the major result is that to introduce the system dynamics is very important to get a fair comparison between heat pumps with variable-speed and single-speed compressors. The comparison will differ depending on system chosen, connection and dead band etc. but these differences will, within reasonable limits, not change the main conclusions.
5 Variable-speed pumps in hydronic heat pump systems – energy saving potential and considerations for control

It is possible to find an optimal combination of compressor and fan or pump speeds that minimises the total drive power. In this chapter this theory is applied to a ground-source heat pump. This chapter is based on paper no. 3.

5.1 Introduction

The use of continuous capacity control of compressors, pumps and fans in heat pump systems add degrees of freedom for the control system and give opportunities for increased energy efficiency. The use of variable-speed compressors in heat pumps connected to varying loads can increase the energy efficiency due to increased performance at part load, reduced need for supplementary heat and reduced need for defrosts. However, adding variable-speed compressor control without changing from standard constant-speed circulators and applying a proper control strategy can diminish the gain of the variable-speed compressor as previously described in chapter 3. This is because the time in operation is longer for heat pumps with continuous capacity control compared to intermittently controlled heat pumps. Thus the drive energy for pumps and fans will be a greater part of the total energy use.

It is possible to find an optimal combination of circulator and compressor speeds that maximises energy efficiency while still fulfilling the operational objective (e.g. a certain room temperature). Jakobsen\[40\] analysed such a case for a cold storage where the condenser was connected to a cooling tower. In this case the speed of one compressor, two fans and one pump could be varied. He found that the optima are quite flat but that the relation between deviations from the optimum and increase in total power use is progressive. In another investigation where a variable-speed pump supplies the flow to an evaporator in a cooling system with a variable-speed compressor, Jakobsen\[42\] proposed a method for finding the optimum by minimising the exergy losses. It is claimed to be easier to implement than to optimise directly on energy use.

Skovrup\[75\] analysed the optimal condensation temperature for a large industrial cooling system where the condensers are connected to cooling towers. He found that the optimal condensation temperature is linearly dependent on the outdoor air wet bulb temperature. The same conclusion is drawn by Liptak\[55\]. Hydeman\[37\] made a similar analysis for a system with several chillers connected to cooling towers. Based on computer simulations he proposed a control method where the pump and fan capacities are controlled as a linear function of the load. The method is claimed to give the same energy efficiency as the method of controlling the fan speed by the wet bulb temperature. Karlsson\[45\] investigated the character of the optimum for the heat sink side of a ground-source heat pump at separate operating points. It was found that the optimum was asymmetric as the COP decreased less when the condensation temperature was higher than optimal than it...
did for condensation temperatures lower than optimal. The heat transfer resistance on the refrigerant side was neglected in this investigation.

Previous investigations, except Jakobsen\[^{42}\] and Karlsson\[^{45}\], consider large systems with somewhat different components than what is normally used for small heat pump heating systems, even though the general principle applies. Furthermore, there is no analysis made on the energy saving potential of applying an optimising control. Consequently, the focus of this chapter is to apply optimisation methods for finding the optimal combination of compressor and circulator speeds in a domestic ground-source heat pump heating system. This is made separately for the heat sink and heat source systems. The energy saving potential for an optimising control method is analysed by comparing the seasonal performance for an optimised system to a standard system. Another aim of this chapter is to determine the characteristics of the optimum, as this will be important for how to perform the optimisation and control in practice.

### 5.2 Background

Below, the general background of the investigation is presented. This chapter only considers ground-source heat pumps in heating mode operation, connected to a hydronic heating system. The general principles are the same for other heat pump types and for heat pumps in systems with air distributed heat and for cooling mode operation but the boundary conditions will be different.

#### 5.2.1 Heat pumps and variable heat transfer media flows

An increased heat transfer medium flow through a heat exchanger increases the heat transfer coefficient and reduces the required mean temperature difference. With the same inlet temperature and an increased flow rate the required mean temperature difference can be maintained with a lower condensation temperature. For the condenser in a heat pump this will mean a lower average temperature level and thus a lower condensing pressure and reduced compressor work. For the evaporator an increased flow will result in a higher evaporation pressure and consequently a reduced compressor work. The penalty for the increased flows is increased pump work. As illustrated in Figure 5.1 the resulting curve will have an optimum where the drive power is minimal. The location and form of this optimum will depend on the boundary conditions and characteristics of the components, i.e. outdoor climate, desired indoor climate and type of heating system, volumetric and isentropic efficiencies of the compressor, efficiencies and capacity of the pumps and heat transfer in the evaporator and condenser. Limits to the optimisation will be set by the operating range of the compressor and pumps, stable operation of expansion device and secured oil return etc.
5.2.2 Methods for varying the heat transfer media flow

In a domestic ground-source heat pump heating system pumps exist not only in the heat pump but also in the hydronic heating system. Balancing valves and control valves are usually introduced into the system to control the heat transfer media flow. Unfortunately, this strategy leads to pressure drops that increase the power demand of the pumps. For example, when the thermostats in a radiator system are closing (due to solar radiation, internal heat sources, etc.) the system curve will be steeper resulting in a pressure drop increase. Energy savings can be made by applying a variable speed pump. The pump can then be set to keep a constant differential pressure forcing the motor speed to decrease, see Figure 5.2, reducing the load on the motor and consequently the energy use. Usually variable speed pumps can also be set to keep a proportional differential pressure to match pressure with flow, see Figure 5.2. An even more energy efficient solution would be if all pressure drops introduced in the system to control it can be replaced by variable speed pumps with direct flow control, as suggested by Fahlén[24].

For further improvement of the energy efficiency, attention to the pump design is important. The pump efficiency is the combined efficiency of the motor, the hydraulic part and the frequency converter. Of these parameters the largest saving potential lies in the motor design. Today, the most commonly used motor in small pumps is the asynchronous motor. Recently, the more efficient permanent magnet (PM) motor has become commercially available in small pumps. By using a PM motor the pump efficiency is improved drastically due to no losses for switching the electromagnetic field. In PM motors the frequency converter is built in which means that no extra hardware is needed for the variable speed drive, as in the asynchronous motor. Furthermore, PM motors have a wider working range with maintained high efficiency.
5.3 Methodology

The heat pump simulated is a ground-source heat pump with a variable-speed compressor, variable-speed pumps, plate heat exchangers and the refrigerant R407C. The heat pump model described below was implemented in Matlab (The Mathworks Inc\(^{[83]}\)), and refrigerant properties were obtained from Refprop (Lemmon\(^{[54]}\)), which was linked to Matlab. The optimisation was performed by using a Matlab routine for constrained optimisation called fmincon. The heat transfer data (U-values) and data on pressure drop for the evaporator and condenser were available from a software supplied by a plate heat exchanger manufacturer (SWEP\(^{[79]}\)). Data on pump efficiencies were obtained from manufacturers’ data (Grundfos\(^{[30]}\) and Wilo\(^{[87]}\)). Compressor efficiency data were obtained from laboratory measurements reported by Karlsson\(^{[47, 49]}\).

Below, first the boundary conditions and system boundaries are described, followed by a description of the model.

5.3.1 Boundary conditions

The system boundaries are illustrated in Figure 5.3. When optimising the sink system, the input data are the return temperature from the heating system, the evaporation temperature (-4.5 °C, obtained from measurements), the evaporator superheat (4 K, used in laboratory tests), the required capacity and the temperature difference \(t_s-t_w=0.5\) K (derived from measurements). If nothing else is stated the heat pump is considered to be connected in parallel to the heating system and thus the circulator in the heat pump only services the condenser and the by-pass line, see Figure 4.5. Similarly, when optimising the heat source system the input data are the return temperature from the bore hole, the temperature of the sub-cooled refrigerant leaving the condenser (\(t_w-0.5\)), the condensing pressure and the evaporator superheat (4 K).

For the annual performance optimisation, the input regarding required capacity and heating system return temperatures are taken from chapter 3 (heat pump B covering 70 % of the required capacity at DOT at the nominal compressor speed (80 Hz), connected to a 55/45 radiator system), see Figure 5.4. The compressor is
an inverter controlled hermetic scroll compressor, heat pump B, see chapter 3.4. The condensing pressure obtained in the optimisation of the heat sink system is used as input to the heat source calculations. The bore hole return temperature is correlated to the outdoor air temperature according to measurements by Fahlén[20]. The duration curve for the outdoor air temperature is calculated from the annual mean outdoor air temperature (+6 °C) and the curve fit model by Fehrm[25], see Eq. 2.1. To keep the simulation time at a reasonable level the annual simulations are made in steps of one week.

![System boundaries](image1.png)

**Figure 5.3** System boundaries for the optimisation of the heat sink system (left) and source system (right)

![Actual return temperature](image2.png)

**Figure 5.4** Actual return temperature from the heating system (left), and requested capacity (right). Inputs to the annual optimisation procedure.

### 5.3.2 Model

The isentropic efficiency, $\eta_{is}$, is calculated from test data by Karlsson[49] and the expression in Eq. 5.1 is derived by linear regression. The result from the equation fits test data within ±3 %:
\[
\eta_{\text{tr}} = 0.926 - 0.0823 \cdot \left( \frac{p_1}{p_2} \right) + 0.00352 \cdot \left( \frac{p_1}{p_2} \right)^2 + 0.00294 \cdot f_{\text{comp}} - 0.00022 \cdot f_{\text{comp}}^2 \quad \text{[-]} \quad \text{Eq. 5.1}
\]

where:
- \( p_1 \) = Condensing pressure [bar]
- \( p_2 \) = Evaporating pressure [bar]
- \( f_{\text{comp}} \) = Compressor frequency [Hz]

The volumetric efficiency, \( \eta_v \), was also calculated from test data, and was then correlated using an expression given by Poulsen[69]. Adding terms for the variable-speed operation gives the following equation:

\[
\eta_v = A \cdot \frac{p_1}{p_2} + B + C \cdot f_{\text{comp}} + D \cdot f_{\text{comp}}^2 \quad \text{[-]} \quad \text{Eq. 5.2}
\]

\[
A = 0.0012 \cdot t_1 - 0.088
\]
\[
B = 1.1262 - 0.0045 \cdot t_1
\]
\[
C = 0.0039
\]
\[
D = -0.000025 \quad \text{Eq. 5.3}
\]

where:
- \( t_1 \) = Condensation temperature [°C]

The model fits test data within ±3 %.

The heat transfer in the evaporator and condenser was modelled by the \( \varepsilon \)-NTU method:

\[
\varepsilon_e = \frac{t_{\text{wo}} - t_{\text{wi}}}{t_3 - t_{\text{wi}}} = 1 - \exp\left( - NTU_e \right) \quad \text{[-]} \quad \text{Eq. 5.4}
\]

where:
- \( t_{\text{wo}} \) = Temperature of outgoing water from the heat pump [°C]
- \( t_{\text{wi}} \) = Temperature of incoming water to the heat pump [°C]
- \( t_3 \) = Temperature of refrigerant after compression [°C]

\[
\varepsilon_e = \frac{t_{\text{bo}} - t_{\text{bi}}}{t_{\text{bi}} - t_6} = 1 - \exp\left( - NTU_e \right) \quad \text{[-]} \quad \text{Eq. 5.5}
\]

where:
- \( t_{\text{bo}} \) = Temperature of outgoing brine from the heat pump [°C]
- \( t_{\text{bi}} \) = Temperature of incoming brine to the heat pump [°C]
- \( t_6 \) = Temperature of refrigerant after expansion device [°C]
\[ NTU = \frac{U \cdot A}{C} \]  \[ \text{Eq. 5.6} \]

The software for calculating the heat transfer was not linked to the optimisation routine. Instead a number of test cases were supplied to the software and U-values for different operating conditions were obtained. The result was then fitted to input data by linear regression. The following types of expressions were used for modelling the U-values for the evaporator and condenser respectively:

\[
U_e = k_1 + k_2 \cdot \dot{M}_R + k_3 \cdot \dot{M}_R^2 + k_4 \cdot t_{hi} + k_5 \cdot t_{hi}^2 + k_6 \cdot f_{comp} + k_7 \cdot f_{comp}^2 + k_8 \cdot p_2 + k_9 \cdot p_2^2 + k_{10} \cdot \dot{V}_b + k_{11} \cdot \dot{V}_b^2 + k_{12} \cdot X_6 + \quad \text{[W/(m}^2\text{K)]]} \quad \text{Eq. 5.7}
\]

\[
k_{13} \cdot X_6^2 + k_{14} \cdot \dot{M}_R \cdot X_6 + k_{15} \cdot f_{comp} \cdot \dot{V}_b + k_{16} \cdot p_2 \cdot X_6
\]

where:
- \( \dot{M}_R \) = Refrigerant mass flow rate [kg/s]
- \( \dot{V}_b \) = Volume flow rate of brine [m\(^3\)/s]
- \( X_6 \) = Quality of refrigerant vapour [-]

\[
U_e = k_{17} + k_{18} \cdot \dot{M}_R + k_{19} \cdot \dot{M}_R^2 + k_{20} \cdot t_{wi} + k_{21} \cdot t_{wi}^2 + k_{22} \cdot f_{comp} + k_{23} \cdot f_{comp}^2 + k_{24} \cdot p_1 + k_{25} \cdot p_1^2 + k_{26} \cdot \dot{V}_w + \quad \text{[W/(m}^2\text{K)]]} \quad \text{Eq. 5.8}
\]

\[
k_{27} \cdot \dot{V}_w^2
\]

where:
- \( \dot{V}_w \) = Volume flow rate of water through the condenser [m\(^3\)/s]

The correlations fit the data with an \( R^2 \)-factor greater than 97 \%. It was decided to use this kind of approach as it was hard to find easy-to-use expressions for the heat transfer in plate heat exchangers. Also, scientifically developed expressions only predict the refrigerant heat transfer within 15-25 \% (Claesson\(^{13}\) and Yan\(^{90}\)). Consequently, the above expressions were considered as sufficient for the purpose of this study.

The drive power to the compressor, \( \dot{W}_{comp} \), was calculated as:

\[
\dot{W}_{comp} = \dot{M}_R \cdot \frac{h_3 - h_4}{1 - F_{comp}} \quad \text{[W]} \quad \text{Eq. 5.9}
\]

where:
- \( h_3 \) = Specific enthalpy of refrigerant after compression [kJ/kg]
- \( h_4 \) = Specific enthalpy of refrigerant before compression [kJ/kg]
- \( F_{comp} \) = Heat losses from the compressor [-]

The pressure drops in the heat exchangers are calculated from the SSP CBE software (SWEP\(^{79}\)). For the heat source system the pressure drop of the collector
system was added. The length of the collector was calculated from the design heat extraction rate of 6 kW and a maximum specific extraction rate of 50 W/m. This results in a bore hole of 120 m, which for a single U-tube configuration means a collector length of 240 m. Adding also the length from the bore hole to the building, the assumed collector length was 260 m. The pressure drop, $\Delta p_{\text{coll}}$, was then calculated as (VDI[86]):

$$\Delta p_{\text{coll}} = \lambda \cdot \rho_b \cdot \frac{L}{d_h} \cdot v_b^2; \text{ laminar flow } \lambda = \frac{64}{Re_b}$$

Eq. 5.10

turbulent flow $\lambda = 0.32 \cdot Re_b^{-0.25}$

where:
- $\lambda$ = Pressure drop coefficient [-]
- $L$ = Collector length [m]
- $\rho_b$ = Density of brine [kg/m$^3$]
- $d_h$ = Hydraulic diameter of the collector [m]
- $v_b$ = Velocity of brine flow [m/s]
- $Re_b$ = Reynold’s number for the brine flow [-]

The efficiency of the circulators was calculated for the specific system curve resulting from the used heat exchangers and systems connected to them. The efficiency was expressed as polynomials with the heat transfer flow as parameter. With the pressure drop, efficiency and flow rate known, the pump power could easily be calculated. The COP used for the optimisation was then calculated as:

$$COP = \frac{\dot{Q}}{W_{\text{comp}} + W_p} \text{ [-]}$$

Eq. 5.11

The optimisation routine minimise the specified objective function, $J$, considering the specified boundary conditions. The objective function for the current optimisation problem is thus:

$$J = \min(-COP)$$

Eq. 5.12

The constraints are the limits in compressor and pump speeds.

5.4 Results and discussion

This section is divided into three parts. Two deal with variable-speed compressors and pumps and focus on the heat sink system and heat source system respectively. In the third section, both the heat source and heat sink systems are treated but for systems with single-speed compressors and variable-speed pumps. First the character of the optimum is shown followed by the results from the annual optimisation.
5.4.1 Heat sink system

To see the character of the optimum for the heat sink system, inputs to the model are the same as specified in section 5.3.1 plus the pump speed. Output from the model is the compressor speed and the COP. This was made for three weeks during a year. The circulator is a Wilo Stratos Eco 30/1-5 which operates in the range of 40 – 100 % of its maximum speed. The results in Figure 5.5 show that the optimum changes with changing boundary conditions but that it is very flat. The flat optimum implies that for optimisation purposes it will be sufficient to be in the vicinity of the optimum.

Figure 5.5 The diagram shows the relative change of COP when deviating from optimal frequency. Note the difference in scale compared to Figure 5.7.

Performing the optimisation for each week during a year gives the result in Figure 5.6, showing the optimal circulator speed to the optimal compressor speed for three different circulators. They all show the same curve form and considering the flat optimum it should be possible in a practical application to set the circulator speed as a linear function of the compressor speed, without degrading seasonal efficiency.
To determine whether it is feasible to implement a method for on-line optimisation it is of interest to see the impact on the seasonal performance factor (given by equation Eq. 5.13).

$$SPF = \frac{Q_1}{W_{\text{comp}} + W_p} \quad [-]$$  \hspace{1cm} \text{Eq. 5.13}

Table 5.1 shows the result from annual optimisation simulations for variable-speed (no. 1-3), single speed (no. 5-7), and one variable-speed pump at constant speed (no. 4).

**Table 5.1** Seasonal performance for different pumps in the heat sink system. Pumps 1-4 have PM motors. $\eta_{\text{mean}}$ designates the average efficiency for the current conditions.

<table>
<thead>
<tr>
<th>Pump</th>
<th>SPF [-]</th>
<th>$W_{wp}$ [kWh]</th>
<th>$W_{\text{comp}}$ [kWh]</th>
<th>$W_{\text{tot}}$ [kWh]</th>
<th>SPF/SPF$\text{no.1}$ [-]</th>
<th>$\eta_{wp=wp_{\text{max}}}$ [%]</th>
<th>$\eta_{\text{mean}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Stratos Eco 30/1-5</td>
<td>3.3</td>
<td>190</td>
<td>7520</td>
<td>7710</td>
<td>1.00</td>
<td>32</td>
<td>27</td>
</tr>
<tr>
<td>2 Magna 25/60</td>
<td>3.3</td>
<td>220</td>
<td>7510</td>
<td>7720</td>
<td>1.00</td>
<td>32</td>
<td>25</td>
</tr>
<tr>
<td>3 Stratos 30/1-6</td>
<td>3.3</td>
<td>200</td>
<td>7500</td>
<td>7700</td>
<td>1.00</td>
<td>35</td>
<td>30</td>
</tr>
<tr>
<td>4 Stratos Eco fixed speed STAR RS 25/4</td>
<td>3.2</td>
<td>280</td>
<td>7490</td>
<td>7770</td>
<td>0.99</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td>5 STAR RS 25/4</td>
<td>3.2</td>
<td>280</td>
<td>7630</td>
<td>7910</td>
<td>0.97</td>
<td>10 (step2 of 3)</td>
<td>10</td>
</tr>
<tr>
<td>6 TOP-S 25/7</td>
<td>3.0</td>
<td>890</td>
<td>7410</td>
<td>8300</td>
<td>0.93</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>7 TOP-S 25/5</td>
<td>3.1</td>
<td>650</td>
<td>7480</td>
<td>8100</td>
<td>0.95</td>
<td>15</td>
<td>15</td>
</tr>
</tbody>
</table>
The three variable-speed pumps show almost identical results. The optimisation only marginally influences the seasonal efficiency as the same pump runs at a fixed speed (no. 4) shows almost the same result as the optimised operation. Compared to the single-speed pumps, the variable-speed pumps increase the SPF by 3-7 %. Most important is thus to use an efficient pump and as a second stage comes optimisation of pump speed.

5.4.2 Heat source system

As before, we start by analysing the character of the optimum for the heat source system. Inputs to the model are as specified in section 5.3.1 plus the pump speed. Outputs are the compressor speed and the COP. This was done for three different weeks during a year. The circulator is a Wilo Stratos 30/1-12, which operates in the speed range of 42 – 100 % of its maximum speed. As shown in Figure 5.7, the optimum is more pronounced than for the pump in the heat sink system. Thus it will be more important to be close to the optimum in order to increase system efficiency.

![Figure 5.7](image)

**Figure 5.7** The diagram shows the relative change of the COP when deviating from the optimum frequency

The heat source circulator should also be possible to control as a linear function of the compressor speed, as the lines in Figure 5.8 have the same character as those of Figure 5.6. As indicated by the more pronounced optimum the gain of optimised control is larger for the brine pump than for the heat sink circulator. In Table 5.2 pumps I and II are variable-speed controlled, pump IV-V are single-speed and pump III is a variable-speed pump running at constant speed. Table 5.2 also shows that the benefit of using a variable-speed pump is greater for the source system than for the sink system. This is according to expectations as the brine pump has larger capacity and thus uses more energy. Comparing pump I with pump IV shows an increase in SPF by 18 %. The optimisation gives an increase in SPF of 5 % (compare pumps II and III).
Figure 5.8 Optimal brine pump speed as a function of the optimal compressor speed

Table 5.2 SPF for the heat source system. Pumps I to III have PM motors. $\eta_{\text{mean}}$ designates the average efficiency for the current conditions.

<table>
<thead>
<tr>
<th>Pump</th>
<th>SPF</th>
<th>$W_{\text{bp}}$ [kWh]</th>
<th>$W_{\text{comp}}$ [kWh]</th>
<th>$W_{\text{tot}}$ [kWh]</th>
<th>SPF/SPFno.I</th>
<th>$\eta_{\text{fbp=fbp,max}}$ [%]</th>
<th>$\eta_{\text{mean}}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>I Stratos 30/1-6</td>
<td>3.7</td>
<td>190</td>
<td>6560</td>
<td>6750</td>
<td>1.00</td>
<td>33</td>
<td>28</td>
</tr>
<tr>
<td>II Stratos 30/1-12</td>
<td>3.7</td>
<td>260</td>
<td>6570</td>
<td>6830</td>
<td>0.99</td>
<td>32</td>
<td>25</td>
</tr>
<tr>
<td>III Stratos 30/1-12</td>
<td>3.5</td>
<td>740</td>
<td>6370</td>
<td>7110</td>
<td>0.95</td>
<td>33</td>
<td>31</td>
</tr>
<tr>
<td>fixed speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>IV TOP-S 30/10</td>
<td>3.0</td>
<td>1930</td>
<td>6320</td>
<td>8250</td>
<td>0.82</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>V TOP-S 25/10</td>
<td>3.1</td>
<td>1730</td>
<td>6330</td>
<td>8060</td>
<td>0.84</td>
<td>24</td>
<td>24</td>
</tr>
</tbody>
</table>

5.4.3 Single-speed compressor

Also a heat pump with a single-speed compressor may increase its performance if variable-speed circulators and an optimisation method are used. Using the same model as above and keeping the compressor frequency at 120 Hz (i.e. the heat pump covers the maximum load) gives the results in Figure 5.9 and Figure 5.10. The use of the optimisation hardly has an impact on the seasonal performance factor. Instead the use of efficient circulators are more important also in this case and increases the SPF by 3 % for the sink system and 6 % for the source system.
5.5 Conclusions

The main conclusion is that there is a potential for applying an optimisation method for keeping the compressor and pump speeds at the optimal combination. The heat source system shows the largest potential as the increase in SPF was approximately 5% when applying the optimisation method compared to the same case without optimisation. For the heat sink system the energy saving potential for the optimisation was small. However, the radiator pump was not considered in this chapter and applying capacity control also to that pump will further improve the energy efficiency of the total system. Another important result is that applying the best pump technology available is very important. By simply changing from a standard pump to the best technique available the SPF increased by 14%. The results from this chapter will have an impact of the comparison between standard heat pumps and variable-speed capacity controlled heat pumps as discussed in chapter 4. This will be further treated in chapter 9.
When extrapolating the results of this chapter to a national level also the very small change for the heat sink system can have a considerable effect. Applying a pump with a variable-speed PM motor instead of a standard pump (no. 4 and 6 in Table 5.1), the difference in energy use is 530 kWh. If all ground-source heat pump systems in Sweden (approximately 300,000) would reduce their energy use by 530 kWh it would result in an energy saving of 0.16 TWh. The same analysis for the source side results in an energy saving of 0.39 TWh, and the total savings would be 0.55 TWh. As a comparison the total energy produced by wind energy in Sweden during 2005 was 1 TWh (Swedish Energy Agency[77]).

Regarding the control of this system, it was shown that the optimum for the heat sink system was very flat and thus it is not critical to find the exact optimum. The optimum for the heat source system was more pronounced even though it is also flat (a deviation from the optimal pump speed by 30% changes the COP by approx. 3%). Considering this, it seems possible to set the pump speed as a linear function of the optimal compressor speed, which is similar to the approach suggested by Hydeman[37]. The relation, however will change depending on what kind of system the heat pump is connected to and thus it will be advantageous to apply an on-line optimisation method.
6 On-line optimisation of heat pump operation

Applying electronic controls of variable-speed pumps, compressors and electronic expansion valves provides an opportunity of using on-line methods for continuous optimisation of the process. This chapter gives the basic principles of optimisation, reviews previous work and gives suggestions for how to implement an on-line optimisation method for heat pump heating systems. The content is partially based on a licentiate work by Karlsson[44], especially sections 6.2 and 6.3.2.

6.1 Background

To optimise the operation of a process, a strategy for determining the optimal set values for the controlled variables at given conditions must be applied. Furthermore, a control system is needed to change the controlled variables from the current state to the new, optimal state. The two tasks of optimisation and control can be more or less integrated, which will be further discussed.

The optimisation task must satisfy at least one objective function which is subjected to certain constraints, see Eq. 6.1. Constraints can be equality constraints and/or inequality constraints, and these constitute the model of the process to optimise. As specified by Edgar[15], the optimal solution does not violate the constraints and fulfils the objective function(s) to a certain degree of precision. The inequality constraints can, according to Svensson[78], be defined as either hard or soft constraints. The hard constraints must never be violated whereas the soft constraints temporarily can be violated without causing damage or major process disturbances.

\[
\begin{align*}
\text{Minimise:} & \quad J(x) \\
\text{Subject to:} & \quad h(x) = 0 \quad \text{equality constraints} \\
& \quad g(x) \geq 0 \quad \text{inequality constraints}
\end{align*}
\]

Eq. 6.1

There are several choices to be made regarding which optimisation strategy to apply, see Figure 6.1. The first decision to make is whether the optimisation strategy should be implemented and used on-line or if the optimisation can be made off-line. If the process is well known, not subjected to disturbances and is not likely to change performance over time, the optimisation can be made off-line. The optimisation is then made at the design stage and implemented with no feedback of real performance. If the process is subjected to disturbances or time varying performance an on-line optimisation method would be preferable.

If the decision has been to use on-line optimisation, the choice is between dynamic optimisation methods or steady-state optimisation. Steady-state
optimisation can be used if the disturbances are infrequent and the process responds quickly to control actions. For processes with very frequent disturbances the process will be in more or less continuous transition and a dynamic optimisation method will then be the best choice. Trying to use a steady-state model in this case may, according to Svensson[78], cause a continuous search, or hunting, for the optimum without ever reaching it.

Figure 6.1 There are several choices to make when choosing an optimisation strategy.

Steady-state optimisation methods can be divided into two groups; direct search methods and indirect, model based methods. Direct search methods can be applied without an explicit process model. Instead the strategy is to change set points of controlled variables directly in the process and measure the impact on the objective function. Examples of direct search methods are the Simplex method (Nelder[59]) and different gradient methods (Edgar[15]). These methods are easy to understand and implement but may require quite a lot of measurements for finding the optimum and can thus become too slow, especially if the number of controlled variables is high (Svensson[78]). Model based methods uses a mathematical model of the process to find the optimum, and should thus be faster. However, the model must be a very accurate representation of reality if the true optimum is to be found. As this can be too challenging one alternative is to update parameters in the model from measurements in the process. The update can be made either in steady-state or dynamic state, depending on the parameter.

For the control task in relation to the optimisation task different approaches can be chosen. One option proposed by Jakobsen[40], Svensson[78] and Leducq[53] is a multilayer decomposition, which implies individual controllers for each controlled variable and an overall optimiser which gives the set point for the individual controllers, see the example in Figure 6.2.

Dynamic optimising control can be applied more or less regardless of the disturbance frequency and dynamics of the process, thus being a very general approach. Applying optimal control theory makes it possible to integrate the control and optimisation tasks. A dynamic model of the plant to control and
optimise is needed, but then the optimisation can be performed at each time step. In one example by MacArthur\textsuperscript{[56]} the COP is maximised while at the same time fulfilling a criterion for good indoor climate. Other investigations on dynamic optimisation control are presented by He\textsuperscript{[33]} and Leducq\textsuperscript{[53]}.

![Diagram of multilayer decomposition of control and optimisation task](image)

**Figure 6.2** An example of a multilayer decomposition of the control and optimisation task proposed by Jakobsen\textsuperscript{[40]}.

### 6.2 Previous work

Cho\textsuperscript{[12]} used the Nelder-Mead simplex direct search method for steady-state optimisation of the performance of an industrial chiller plant. The objective was to minimise energy use with inequality constraints to keep certain process parameters within allowable limits. As the simplex method in its original form does not handle constraints, the inequality constraints were introduced as penalty functions into the objective function (see the example in Eq. 6.2). As described by Edgar\textsuperscript{[15]} the penalty functions transform the constrained optimisation into an unconstrained optimisation problem. The system checks if the plant is in steady-state operation before the optimisation is carried out. Controlled variables were chilled water flow and the temperature difference over the evaporator. After implementation the energy use was reduced by 12% without any substantial changes to the cooling load.

Braun\textsuperscript{[8]} presents an investigation where the objective was to minimise the drive power while satisfying the constraints imposed by cooling load and keeping certain process parameters within their allowable limits. The strategy was to express the drive power of the chiller, condenser pumps and cooling tower fans as quadratic functions. 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. A “near-optimal” optimisation method is also suggested where the total drive power to the plant is expressed as a quadratic function. In this way no detailed models of the individual components are needed. Both methodologies were applied to a chiller plant 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.

Svensson[78] have made an extensive work on steady-state indirect 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, thus controlling the speed of two pumps and one compressor. The
objective for the optimisation was to minimise total drive power with the
constraints of supplying the heat load and not violating parameter limits. 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, at
steady-state, by the use of a moving horizon algorithm. 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 for solving the optimisation
problem varied between 10 to 25 seconds.

The above investigations applied steady-state optimisation methods. Leducq[53]
and MacArthur[56] used dynamic optimisation methods instead. Leducq[53] used a
predictive method which uses an explicit process model for predicting the future
response of the process. The method is applied by using a global optimiser and
local controllers. The method was applied to a refrigeration plant, where two
pump speeds and one compressor speed was controlled. The plant also has an
electronic expansion valve but that is controlled by its own control loop. The
objective function in this case was to maximise $COP$ and to supply a specified
cooling capacity, $Q_{set}$. These two objectives were combined into one function as:

$$
J = \int_{\tau} \left( \mu_0 \cdot (\hat{Q}_e - \hat{Q}_{set})^2 + \mu_1 \cdot \left( \frac{1}{COP} \right)^2 + \mu_2 \cdot \left( \frac{dU}{d\tau} \right)^2 \right) d\tau 
$$

Eq. 6.2

The last term penalises too rapid input changes. Constraints to pump and
compressor speeds were introduced as extra terms in Eq. 6.2, expressed in the
same way as the first term of the equation. The coefficients must be tried out in
order to find a good balance between the different criteria. The method was found
to work properly, optimised COP and could handle disturbances. Also, the model
of the plant must be simple enough to be computed many times per second. In this
case the controller was implemented in a PC.

MacArthur[56] suggested the use of optimal control theory to control and optimise
the operation of an air-conditioner. The objective function was to maximise COP
and fulfil an indoor climate criterion (Predicted Mean Vote index). He[53] used a
model-based linear-quadratic designed controller for controlling compressor
speed and evaporator superheat in an air conditioner. This MIMO (Multiple Input
Multiple Output) controller is claimed to be more efficient and faster than
multiple SISO (Single Input Single Output) controllers as suggested by
Jakobsen[40], Figure 6.2. This is due to the strong cross-couplings between the
individual processes. It is claimed that the approach could be extended to include
also fan speeds and thus control the total operation of the air conditioner. Also this
controller was implemented in a PC.
6.3 Application to a small heat pump heating system

In this chapter suggestions will be given regarding how to optimise and control a small heat pump heating system. The approach is to keep the complexity as low as possible and to have the method as general as possible. The system to control and optimise is a ground-source heat pump with two variable-speed pumps, one variable-speed compressor and an electronic expansion valve.

The heat pump is connected in parallel to a hydronic heating system, as shown in Figure 6.3. The parameter to optimise is the energy efficiency which should be as high as possible, and thus the COP must be maximised or the drive power minimised. This objective is valid when the heat pump’s capacity is greater than the load, i.e. the heat pump operates at conditions above the balance point. If operating below the balance point the “free” heat extracted from the ground should be maximised and thus the cooling capacity $Q_c$ should be maximised.

However, this chapter will only deal with the issue of maximised COP. The constraints are the capacity of the heat pump that must meet the current load, and that pump and compressor speeds must be kept within the allowed range. Other constraints such as limits put on temperature of refrigerant exiting the compressor, minimum allowable superheat etc. may apply but the exact constraints to consider must be determined for each design and application. For this example the optimisation task can be formulated according to Eq. 6.1 as:

$$J_1 = \min \left( \frac{1}{COP} \right) \quad \text{alternatively} \quad J_1 = \min \left( \dot{W}_{\text{tot}} \right)$$

$$h(x) = \dot{Q}_{hp} - \dot{Q}_{bldg} = 0$$

$$g(x) = \begin{bmatrix} f_{hp,min} - f_{hp} \\ f_{hp} - f_{hp,max} \\ f_{wp,min} - f_{wp} \\ f_{wp} - f_{wp,max} \\ f_{comp,min} - f_{comp} \\ f_{comp} - f_{comp,max} \end{bmatrix} \leq 0$$

Eq. 6.3

The equality constraint(s), $h(x)$, represent the set of equations needed for describing the heat pump system and determine COP or drive power.
6.3.1 Optimisation using a direct search method

Heat pump systems for space heating are subjected to continuously varying conditions caused by changes in outdoor climate, which affect the heat load and temperature of the heat source, as well as varying internal gains, which again affect the heat load. These variations have different time scales as the outdoor air temperature varies both during one day and with the seasons. For these reasons, an on-line optimisation method would be the preferable choice. The refrigerant circuit responds quickly to control actions whereas the response from the heating system is slower. The variations in indoor and outdoor conditions are also rather slow. Furthermore, for a capacity controlled heat pump system there are no fast transients occurring from on and off switching of components. For these reasons a steady-state, direct search optimisation method should be sufficient. A dynamic optimisation and control strategy as suggested by He, MacArthur or Leducq would probably also be a good choice. However, in order to keep things simple, and also as the results in chapter 5 suggest that a fairly simple approach can be taken, dynamic optimisation will not be further considered here. Instead the method that will be investigated is the Nelder-Mead simplex method (Nelder) as previously suggested by Cho. The method is fairly simple to understand and does not require extensive modelling and knowledge about the plant. In the context for this thesis the choice of model is mainly used for
demonstrating what is important to consider. The exact model to apply can of course be different in a real application.

As the dynamics of the refrigerant circuit are much faster than the dynamics of the heating system the control of the evaporator superheat is kept out of the optimisation process and thus the electronic expansion valve is given an independent local controller. The expansion valve must respond quickly to changing conditions in order to protect the compressor. Also the circulator for the heating system is given a separate control loop. Modern variable-speed controlled circulators often include a local control system which control the pump speed in relation to the pressure drop in the system as previously discussed in chapter 5. The control of this circulator is left out of the optimisation to keep the number of variables as low as possible. This approach also separates the optimisation of the flow in the heating system from the optimisation of the heat pump operation.

Hence, there remain three controlled variables, which are the three rotational speeds of two pumps and one compressor. Considering the results from chapter 5 the circulator for the warm heat transfer medium could also be left out of the optimisation routine but will be included in the analysis below. As the simplex method does not treat constraints, the objective function must be changed in order to handle limits such as speed ranges. This can be made by the use of penalty functions or barrier functions (Edgar[15]). Applying quadratic penalties, the objective function can be written as:

\[
J = \mu_1 \cdot \left( \frac{1}{COP} \right)^2 + \mu_2 \cdot (t_{in} - t_{in, set})^2 + \mu_3 \cdot \left( \text{max}(0, f_{bp} - f_{bp,max}) \right)^2 + \\
\mu_4 \cdot \left( \text{max}(0, f_{wp} - f_{wp,max}) \right)^2 + \mu_5 \cdot \left( \text{max}(0, f_{comp} - f_{comp,max}) \right)^2 + \\
\mu_6 \cdot \left( \text{max}(0, f_{comp,min} - f_{comp}) \right)^2
\]

Eq. 6.4

The coefficients for exceeding and going below the speed limits are the same but these can of course be set individually. The room temperature is used as indicator of the load. Room temperature or return temperature of the heating system are common indicators of the capacity in relation to load.

When using a direct search, steady-state method, each suggestion for a new operating point must be evaluated by measurement, which takes time. A good first guess is a way to speed up the process. The methodology proposed in chapter 5, where the pump speeds are controlled in direct relation to the compressor speed, is one option for getting a start value. As there are uncertainties related to the modelling, the combination of a design stage control design and an on-line optimisation with feed-back would be a good solution for a strategy that will find the real optimum without too long search time. Eq. 3.8 to Eq. 3.11 is also an option to utilise in order to find a start value for the optimisation routine.
6.3.2 Applying the simplex method

Figure 6.4 shows the simplex for a two dimensional optimisation. The number of points in the simplex is one more than the number of variables. The method can in theory be used for an infinite number of variables but it gets increasingly slower and should not be used if the number of variables is more than ten according to Avriel\cite{5}. The three points \( x_0, x_1 \) and \( x_2 \) form the start simplex. The idea of the method is to evaluate each of these points and then to exclude the worst of them. The disregarded point is then mirrored in the other two. Presuming that the evaluation results in that point \( x_2 \) in Figure 6.4 being the worst, it is excluded and mirrored in the two other points. This new point, \( x_r \), is called the reflection point. Now we have a new simplex to evaluate. However, the Nelder-Mead simplex method contains more steps. The value of the objective function at \( x_r \) is evaluated and depending on the result in relation to the result of the evaluation at \( x_0 \) and \( x_1 \), the following steps are taken:

- The step is expanded to \( x_e \)
- Stays at \( x_r \)
- The step is contracted to \( x_c \)
- All points in the simplex are replaced by new ones, forming a smaller simplex

After these steps a new simplex is formed and the process starts all over again. This process goes on forever if no stop criterion is formulated. Avriel\cite{5} suggests to stop the optimisation on the standard deviation of the objective function, as described in Eq. 6.5 where \( \sigma \) is a predetermined positive value.

\[
\sqrt{\frac{1}{n+1} \sum_{i=0}^{n} (J(x_i) - J(\bar{x}))^2} < \sigma
\]

Eq. 6.5

**Figure 6.4** The simplex forms a triangle for a 2-dimensional optimisation. In the figure the points for reflection, expansion, and contraction are indicated.
In the licentiate work by Karlsson\cite{44}, the Nelder-Mead simplex method was applied to a mathematical model of a heat pump, in order to estimate the usability of the method. The mathematical model is similar to the one presented in chapter 5, but with constant U-values and pump efficiencies. Two pump speeds and one compressor speed was optimised, subjected to the following objective function:

$$J = \frac{1}{COP_{bps}} + \mu_f \cdot \left( \dot{Q}_{bps} - \dot{Q}_{bps, \text{set}} \right)^2 + \mu_{t} \cdot \left( t_w - t_{w, \text{set}} \right)^2$$  \hspace{1cm} \text{Eq. 6.6}$$

As there are three parameters, the simplex will form a three-dimensional tetragon. The pumps operated between 25-100 % of their maximum speed and the compressor operated between 40-100 % of its maximum speed (30-75 Hz). However, these limits were not introduced as constraints. One of the performed optimisations had the following constraints:

- $\dot{Q}_{bps, \text{set}} = 5000 \, \text{W}$
- $t_{w, \text{set}} = 44 \, ^\circ \text{C}$

With $t_{w, \text{set}} = 40 \, ^\circ \text{C}$ and $t_{h, \text{set}} = -1 \, ^\circ \text{C}$, and the starting simplex of Table 6.1, the optimisation proceeded as in Figure 6.5 and Figure 6.6. The optimisation narrows the range for COP and after seven iterations there is no point of continuing the optimisation as the change in $COP_{bps}$ after that is very small and cannot be measured in practice. Unfortunately the number of runs is quite large and would thus take a long time. However, if continuing the design process with proper values for the coefficients of the penalty functions and a good model for start values it seems possible to use this method.

**Table 6.1** The points for the first simplex

<table>
<thead>
<tr>
<th>Frequency</th>
<th>Points in the starting simplex</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f_{\text{comp}}$ (Hz)</td>
<td>1</td>
</tr>
<tr>
<td>50</td>
<td>46</td>
</tr>
<tr>
<td>$f_{\text{bp}}$ (Hz)</td>
<td>50</td>
</tr>
<tr>
<td>$f_{\text{wp}}$ (Hz)</td>
<td>50</td>
</tr>
</tbody>
</table>
**Figure 6.5** The figure shows how the COP evolves during the optimisation and how many runs that are necessary for the optimisation.

**Figure 6.6** Value of the objective function during the optimisation process

### 6.3.3 Measurement system for a direct search method

A measurement system is needed for evaluating the objective function. The design of the measurement system will depend on how the objective function is designed. Two different approaches were proposed in Eq. 6.3. If using the COP for optimisation, a measuring system as the one of Figure 6.7 is needed. The COP is then calculated according to Eq. 6.7. The heat loss factor for the compressor must be obtained from measurements or from a manufacturer’s data.
Figure 6.7 Measurement system needed for determining COP.

\[
COP_{hp} = \frac{h_3 - h_5}{h_3 - h_4} \cdot \left(1 - F_{comp}\right)
\]  
Eq. 6.7

Figure 6.8 A Mollier diagram with the measuring points corresponding to the ones in Figure 6.7 (NT-VVS 116[44]).

In a commercial application surface mounted temperature sensors would most likely be applied. Then a measurement accuracy of 10% would be possible to achieve in steady-state (Karlsson[44]). The uncertainty of the temperature measurement must be less than 1 K if a total uncertainty of measurement of 5% shall be reached. This will be hard to accomplish with surface mounted sensors, especially for the temperature measurement of the refrigerant exiting the compressor. To determine also the capacity of the heat pump, the electric power used by the compressor must be measured.

However, with the proposed measurement method only the heat pump COP, i.e. \(COP_{hp}\), will be used for optimisation purposes. But what is wanted to minimise is the total power used by the unit, i.e. to maximise \(COP_{hp_{ps}}\), and then the COP must be determined as in Eq. 6.8. This may be considered as a lot of measurements but
the suggested measurement system can also be used for fault detection and diagnosis purposes.

\[
COP_{bps} = \frac{\text{COP}_{bp} \cdot \dot{W}_{em}}{\dot{W}_{em} + \dot{W}_{epu} + \dot{W}_{epb}}
\]

Eq. 6.8

A system with less measurements would be to measure only the total power used by the unit. This would work as long as the operation is over the balance point. However, this will not be possible to use for fault detection and diagnosis.

6.4 Concluding discussion

As previously discussed in this chapter there are several methods to use for optimisation and control of heat pump operation. Considering the fast dynamics of the refrigeration circuit in relation to the heating system it should be possible to use a steady-state optimisation method. The choice of a direct search method or a model based method will depend on the time needed for the direct search method to find the optimum in comparison to how frequently the operating conditions change. The direct search method will be faster if the number of variables is low. The system proposed in this chapter can be further simplified if the circulator for the warm heat transfer medium is left out of the optimisation routine, which would be possible with only minor changes in efficiency as shown in chapter 5. Further, the compressor speed could be controlled by the return temperature of the heating system utilising so called curve control as illustrated in Figure 1.4. Then the only parameter left to optimise is the speed of the circulator for the cold heat transfer medium. In this way it is possible to keep the optimisation routine quite simple and fast. However, the efficiency may be penalised depending on how well the relation between outdoor air temperature and heating system return temperature is established. This relation can be improved if also a room temperature sensor is included.
7 Optimised control of air-source heat pumps – defrost control

The models and principles applied in previous chapters apply also for air source heat pumps. In addition comes the optimisation and control of the defrost function. This chapter is based on paper no. 4, and deals with methods for determining the optimal time to start a defrost. For a complete optimisation also the method of defrosting must be optimised. This will not be investigated here.

7.1 Introduction

Air-source heat pumps (in this thesis air-source implies the use of outdoor air as heat source) are widely used all over the world, mostly for cooling appliances but also for heating. When operating in a cold climate, frost and ice will form on the evaporator surface and eventually block air-flow and heat transfer such that the efficiency is reduced and the unit will not be able to provide the capacity needed. If not designing the evaporator or air cooler such that no frost forms on the surface (oversizing of coil, surface treatment etc.), or operating the system such that no frosting occurs (e.g. capacity control), a function for defrosting is needed. Even though needed, defrosts mean reduced efficiency and thus they should be performed efficiently and not more often than needed. Current methods for determining when to defrost often rely on combinations of demand control and time, and it is not rare that the time control overrides the demand control. The demand controls often rely on an indirect measurement of the frost growth such as temperature differences, changes in air flow or pressure drop etc. These indirect demand control systems are difficult to design as they must handle all combinations of climate conditions and variations due to manufacturing tolerances etc. Previous investigations by Fahlén[22] and Karlsson[50] also show that defrosts are often performed too frequently. Fahlén[19] proposed a method for optimized defrost control where the optimal moment for initiation of defrost is controlled on the \( \text{COP} \) or the cooling capacity, which are measured on-line. Erlandsson[17] applied this method on data from earlier tests and showed that \( \text{COP} \) could be improved by 6-12 %. In this chapter the proposed optimal method is experimentally evaluated on two air-to-water heat pumps; one direct and one indirect system.

7.2 Theory

Fahlén[19] has reviewed and discussed a number of different defrost controls. Based on this, he proposed an alternative method, described below. To maximise energy efficiency often means to maximise \( \text{COP} \). Thus the mean value of the \( \text{COP} \) over a complete cycle including heating and defrost must be as high as possible, i.e. the optimal moment is when:

\[
\frac{\partial \text{COP}_m}{\partial \tau} = 0
\]

Eq. 7.1
Fahlén[19] showed that when the integrated mean value has a maximum (Eq. 7.1), the instantaneous value for \( COP \) and the integrated mean value, \( COP_m \), are equal. The same principle applies also to capacity. In Figure 7.1 below the moment when \( COP \) intersects \( COP_m \) is the moment for initiating the defrost. In this particular case the heat pump continues to run and thus the \( COP \) for the whole cycle is lowered.

![Figure 7.1 Instantaneous COP and integrated mean value COP_m for one cycle](image)

When implementing this in a time-discrete system Eq. 7.2 must be rewritten. If using a fixed time interval between each sample Erlandsson[17] showed that it can be rewritten as:

\[
COP_m(\tau_{t+1}) = \frac{\sum_{\tau=\tau_0}^{\tau_{t+1}} \dot{Q}(\tau) + \dot{Q}(\tau_{t+1})}{\sum_{\tau=\tau_0}^{\tau_{t+1}} W(\tau) + \dot{W}(\tau_{t+1})}
\]

Eq. 7.3

In this way only the sums need to be updated in each time step. The corresponding equation for optimizing capacity can, according to Fahlén[19], be written as:

\[
\dot{Q}_m(\tau_{t+1}) = \left( \frac{\tau_i - \tau_0}{\tau_{t+1} - \tau_0} \right) \cdot \dot{Q}_m(\tau_i) + \left( \frac{\tau_{t+1} - \tau_i}{\tau_{t+1} - \tau_0} \right) \frac{\dot{Q}(\tau_i) + \dot{Q}(\tau_{t+1})}{2}
\]

Eq. 7.4

Eq. 7.4 shows that the added values will affect the integrated mean value less as the cycle time increases because the time quotient in the last term will continually decrease.
7.3 Methodology

The above described method was evaluated on two air-to-water heat pumps, one direct system (X) and one indirect (Y), see Figure 7.2 and Figure 7.3. The direct system uses a hot gas defrost (reversed refrigerant flow). Heat pump Y defrosts by letting the warm brine in the defrost tank flow through the air cooler and then the brine passes through the evaporator before flowing back into the tank.

For comparison, the manufacturers’ own controls were also evaluated at the same conditions as the optimized control. Measurements were performed at the outdoor conditions +2 °C at 85 % and 90 % relative humidity, and at -7 °C and 70 % relative humidity. The tests were made in SP’s accredited laboratory according to the former European test standard SS-EN 255-2[76] where three complete cycles, including heating operation and defrosts, are evaluated for each test. The uncertainties of measurement are derived according to Fahlén[18] and ISO GUM[39] with a coverage factor k = 2, which gives a confidence level of approximately 95 %. The following uncertainties apply: $U_{i_a} < 0.2\ K$, $U_{i_{ab}} < 0.3\ K$, $U_{\Delta t} < 0.05\ K$ (water and brine), $U_{\psi} < 1\ %$ and $U_{\psi_w} < 0.5\ %$. The uncertainty in COP is below 2 %. Measurements were made every 10 seconds. Table 7.1 summarises the tests performed.

Figure 7.2 Direct evaporation outdoor air-source heat pump, designated X.
Figure 7.3 Indirect outdoor air-source heat pump, designated Y. The dashed lines show the flow when in defrost operation.

Table 7.1 Test conditions for the two heat pumps. All conditions were run with both the manufacturer’s control and the proposed optimized control.

<table>
<thead>
<tr>
<th>Heat pump</th>
<th>Test no</th>
<th>( t_{ui} ) [°C]</th>
<th>( \varphi_{ai} ) [%]</th>
<th>( t_{wo} ) [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>X1</td>
<td>+2</td>
<td>85</td>
<td>35</td>
</tr>
<tr>
<td>X</td>
<td>X2</td>
<td>+2</td>
<td>90</td>
<td>35</td>
</tr>
<tr>
<td>X</td>
<td>X3</td>
<td>-7</td>
<td>70</td>
<td>35</td>
</tr>
<tr>
<td>Y</td>
<td>Y1</td>
<td>+2</td>
<td>85</td>
<td>35</td>
</tr>
<tr>
<td>Y</td>
<td>Y2</td>
<td>+2</td>
<td>90</td>
<td>35</td>
</tr>
</tbody>
</table>

Stopping of the defrosts was controlled by the manufacturer’s control system and thus the stop procedure was the same for both the manufacturer’s control and the optimized control.

7.4 Results

For the tests, the optimization routine for \textit{COP} was implemented and the results will mainly focus on this. However, when the heating demand exceeds the heat pump capacity then the use of the heat source, i.e. the cooling capacity, should be maximised to reduce the use of supplementary heating. Based on the tests for \textit{COP} the optimization of cooling capacity is also discussed.

7.4.1 Optimization of COP

For all the test conditions the optimized control results in longer cycle times and also in longer defrost periods, as shown in Table 7.2 below. For heat pump Y, the differences in \textit{COP} between the two control methods are small and of the same order as the uncertainty of measurement, even though the cycle times differ quite a lot. This indicates that the optimum is very flat.
For heat pump X the differences are larger. For all tests except X3 the manufacturer’s control gives higher $COP$ than the optimized control. The explanation for this can be found in the longer defrost times. As the optimal control lets the heat pump run longer before initiating a defrost it also allows more frost to form on the evaporator surface, which will cost more energy to remove.

For condition X3, the manufacturer’s control initiates defrosting just as often as for condition X1 and X2. As the air contains less water at X3 the time between defrosts should be longer. This is also the case for the optimized control where the cycle time is actually twelve times longer than for the manufacturer’s control and with a higher $COP$. The low water content in the air is also reflected in the defrost time which is just slightly longer for the optimized control than for the manufacturer’s control despite the much longer cycle time.

### Table 7.2 Results from the tests on the two heat pumps

<table>
<thead>
<tr>
<th>Test no</th>
<th>Control</th>
<th>$COP$</th>
<th>$COP_{hp}$</th>
<th>$\tau_c$ (min)</th>
<th>$\tau_d$ (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>X1</td>
<td>Manufacturer</td>
<td>3.2</td>
<td>3.5</td>
<td>55</td>
<td>1.7</td>
</tr>
<tr>
<td>X1</td>
<td>Optimized</td>
<td>3.0</td>
<td>3.3</td>
<td>131</td>
<td>5.3</td>
</tr>
<tr>
<td>X2</td>
<td>Manufacturer</td>
<td>3.2</td>
<td>3.4</td>
<td>56</td>
<td>2.2</td>
</tr>
<tr>
<td>X2</td>
<td>Optimized</td>
<td>3.0</td>
<td>3.2</td>
<td>127</td>
<td>6.0</td>
</tr>
<tr>
<td>X3</td>
<td>Manufacturer</td>
<td>2.5</td>
<td>2.8</td>
<td>55</td>
<td>1.2</td>
</tr>
<tr>
<td>X3</td>
<td>Optimized</td>
<td>2.6</td>
<td>3.0</td>
<td>665</td>
<td>1.8</td>
</tr>
<tr>
<td>Y1</td>
<td>Manufacturer</td>
<td>2.8</td>
<td>3.2</td>
<td>230</td>
<td>5.5</td>
</tr>
<tr>
<td>Y1</td>
<td>Optimized</td>
<td>2.7</td>
<td>3.2</td>
<td>278</td>
<td>5.9</td>
</tr>
<tr>
<td>Y2</td>
<td>Manufacturer</td>
<td>2.7</td>
<td>3.2</td>
<td>196</td>
<td>5.6</td>
</tr>
<tr>
<td>Y2</td>
<td>Optimized</td>
<td>2.7</td>
<td>3.1</td>
<td>237</td>
<td>5.8</td>
</tr>
</tbody>
</table>

Even though the manufacturer’s control gives a better $COP$ than the optimized control it can still be improved as shown in Figure 7.4, where the theory for finding the optimal $COP$ has been applied to the data from the measurements at condition X1 with the manufacturer’s control. The figure shows that the optimal point is not reached before a defrost is initiated. For condition Y1 the $COP$ values hardly differ between the two control methods, which is also shown in Figure 7.5 where the lines for $COP$ for the two cycles are almost identical. The manufacturer’s control initiates a defrost a little earlier than optimal, but as the curve is quite flat the early initiation will hardly affect the result.
Figure 7.4 Instantaneous COP and integrated mean value, \(COP_m\), for one cycle, test condition X1. ‘opt’ indicates the test with the optimized control and the other two graphs show the result with the manufacturer’s control.

Figure 7.5 Instantaneous COP and integrated mean value, \(COP_m\), for one cycle, test condition Y1. ‘opt’ indicates the test with the optimized control and the other two graphs show the result with the manufacturer’s control.

### 7.4.2 Optimization of cooling capacity

When running below the balance point, i.e. when supplementary heat is needed, then the heat source should be used to a maximum. Thus the cooling capacity should be maximised. Applying Eq. 7.4 on the measurements gives the optimized cooling capacity given the defrost energy used.
Table 7.3 The table shows the optimized cooling capacity, the measured cooling capacity and the cycle time for the optimized cycle. A hyphen, –, denotes that the measured cycle was too short to determine the optimized cycle.

<table>
<thead>
<tr>
<th>Test no</th>
<th>Control</th>
<th>( \hat{Q}_{2,\text{opt}} ) [kW]</th>
<th>( \hat{Q}_2 ) [kW]</th>
<th>( \tau_{c,\text{opt}} ) (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>X1</td>
<td>Manufacturer</td>
<td>-</td>
<td>3.7</td>
<td>-</td>
</tr>
<tr>
<td>X1</td>
<td>Optimized</td>
<td>3.4</td>
<td>3.3</td>
<td>114</td>
</tr>
<tr>
<td>X2</td>
<td>Manufacturer</td>
<td>-</td>
<td>3.7</td>
<td>-</td>
</tr>
<tr>
<td>X2</td>
<td>Optimized</td>
<td>3.3</td>
<td>3.3</td>
<td>114</td>
</tr>
<tr>
<td>X3</td>
<td>Manufacturer</td>
<td>-</td>
<td>2.4</td>
<td>-</td>
</tr>
<tr>
<td>X3</td>
<td>Optimized</td>
<td>2.5</td>
<td>2.5</td>
<td>608</td>
</tr>
<tr>
<td>Y1</td>
<td>Manufacturer</td>
<td>-</td>
<td>4.1</td>
<td>-</td>
</tr>
<tr>
<td>Y1</td>
<td>Optimized</td>
<td>3.9</td>
<td>3.9</td>
<td>243</td>
</tr>
<tr>
<td>Y2</td>
<td>Manufacturer</td>
<td>-</td>
<td>4.0</td>
<td>-</td>
</tr>
<tr>
<td>Y2</td>
<td>Optimized</td>
<td>3.9</td>
<td>3.8</td>
<td>208</td>
</tr>
</tbody>
</table>

The results in Table 7.3 are similar to those for COP optimization in the sense that the manufacturer’s control gives higher cooling capacities than the optimizing method, except for case X3. However, comparing the cooling capacity achieved when optimizing COP to the one achieved when actually optimizing the cooling capacity, the latter results in higher values. The differences are very small and lie within the uncertainty of measurement and thus these maxima are flat. Comparing the cycle times for Table 7.2 and Table 7.3 also shows that when optimizing the cooling capacity the optimal cycle time is shorter than when optimizing COP.

7.5 Discussion and conclusions

The suggested method for optimized defrost control does not give the best COP for a heat pump at given conditions. The results show that the optimizing defrost control prolongs the cycles and thus more frost will form on the evaporator surface, forcing the unit to do longer defrosts. These longer defrosts take more operating time to recover and thus the cycle will be even more prolonged. This process will balance but not necessarily on the best solution. The method gives the maximum COP for a heat pump under given conditions and the given defrost performed, thus the method only partly optimizes the process. However, without any trimming of the controls, the method resulted in COPs which are close to the ones resulting from the use of the manufacturer’s factory tuned control. Thus there should be a possibility to use the proposed method if including also a routine which can tune the method such that the unit is not operating in the heating mode too long before defrost starts.

The results above are based on laboratory measurements where flow rate and temperature measurements are used for determining the capacity. Alternatively one can measure temperatures and pressures in the refrigerant circuit to measure the capacity and COP. As described by Fahlén[19], the COP can be determined by measuring temperature and pressure at the inlet and outlet of the compressor and the temperature of the refrigerant exiting the condenser. Then points 3, 4 and 5 in Figure 6.8 are known and COP can be determined according to Eq. 6.7, where
$F_{\text{comp}}$ designates the heat losses from the compressor as a fraction of the drive power input. By measuring the electric power or current the capacity can also be determined. This measurement set-up could quite easily be installed when producing the unit, and could then also be used for optimizing the operation of the entire heat pump system as described in chapter 6.
8 Application to other types of heat pump system

This chapter aims to generalise the previous results to heat pump systems other than hydronic ground-source systems.

8.1 Outdoor air source heat pumps

Outdoor air is a very common heat source for heat pumps, also in cold climates. As already discussed in chapter 7 the defrost function is an extra consideration beside the aspects of variable-speed control of compressors and circulators as discussed before. The energy saving potential for outdoor air-source heat pumps should be larger than for ground-source systems as the evaporation temperature, with standard design of the evaporator, increases more for air-source heat pumps when the compressor capacity is reduced than it does for ground-source systems. The greater temperature change is due to the normally smaller heat transfer capacity of standard designs of finned-tube coils compared to standard designs of plate heat exchangers. Also the frosting will be less and thus there will be less need for defrosts and the efficiency will increase further because of this.

Considering the optimisation of pump, fan and compressor speeds the same principles apply as described in chapters 5 and 6. The power use of the evaporator fan generally is about the same as the power use of the ground-source circulator. Hence, it should be more important to control and optimise the evaporator fan than the condenser fan or pump.

8.2 Exhaust air heat pumps

Exhaust air heat pumps recover heat from the ventilation exhaust air flow and rejects it back to the building, normally via a hydronic heating system, and/or to sanitary hot water, see Figure 8.1. Due to the limited heat source, these heat pumps normally cover a smaller part of the total heating demand than do ground-source systems. In a test from 2001, reported by Lagergren\[51\], the four heat pumps evaluated covered 67 – 82 % of the annual heating demand for a building with a total demand for space and domestic hot water heating of 15 MWh. The capacity of the heat pump can be increased by increasing the ventilation air flow or by increasing the capacity of the compressor. However, increasing the air flow will also increase the heating load, as described by Eq. 8.1, where transmission and ventilation losses are separated.

\[
Q_{bldg} = \left(\frac{1}{R_{bldg}} + \dot{C}_a\right) \cdot \left(t_{in} - \Delta t_a - t_{out}\right)
\]  

Eq. 8.1

The design compressor capacity will be limited by the evaporation temperature and the penalty paid by reduced efficiency when the heat pump operates under frosting conditions. As the indoor air is the heat source for the exhaust air heat
pump, the source temperature is almost constant throughout the year. Thus, if the heat pump is designed such that frosting occurs on the evaporator during the coldest days (when the heat demand is high) frosting may occur throughout the year. Using a variable-speed compressor in the heat pump makes it possible to increase the capacity of the heat pump without the draw-back of frosting during the major part of the year. As in ground-source systems the efficiency of the heat pump system will increase when operating at reduced capacity when the heat pump does not operate intermittently.

Considering defrosting, there is no need for a special function as the high temperature heat source can be used for melting the frost. It is sufficient to just stop the operation of the compressor and let the warm exhaust air melt the frost.

![Diagram of exhaust air heat pump](image)

**Figure 8.1** Schematic diagram of exhaust air heat pump rejecting heat to a hydronic heating system. Normally these systems also reject heat to a sanitary hot water tank.

As the drive power of the compressor in an exhaust air heat pump for domestic use is rather small (approximately 500 W) the efficiency and operation of fans and pumps is even more important than it is for ground-source heat pumps. Thus, the potential for applying efficient variable-speed pumps and fans and applying an on-line optimisation routine should have a greater potential than for ground-source heat pump systems.

The optimisation of fan, pump and compressor speeds will have more boundary conditions to consider than is the case for outdoor air and ground-source systems. The exhaust air flow is limited in the lower end by demands on the indoor air quality, and higher flows will be limited by the extra capacity needed to heat the outside air entering the building. Thus Eq. 8.1 must be incorporated in the optimisation criterion.
To summarise, there seem to be a clear potential for improved energy efficiency of exhaust-air heat pumps by applying variable-speed compressors, pumps and fans. The potential lies in both increased energy coverage and improved performance at part load.

### 8.3 Other heat sinks

The methods and principles presented in this thesis should apply also if the heat pump is connected to warm-air distributed systems or larger hydronic heating systems in for example office buildings. For larger systems there is the possibility to apply sequencing of several compressors instead of applying variable-speed capacity control. One option could be to have a number of single-speed compressors and use only one variable-speed compressor for adapting the capacity to the load. A steady-state on-line optimisation method should also be possible to apply, if having the distribution pumps controlled by their inherent control algorithms, for example proportional pressure difference. For fan-coil systems the control of the fan speed should also be included in the optimisation process as they otherwise will reduce system efficiency at reduced capacities in the same way as previously discussed for the circulators.

Sanitary hot water is another common heat sink for hydronic heat pump heating systems. The most common solution is that the heat pump is used both for space heating and sanitary hot water heating. Normally, the hot water is stored in a double-walled storage tank, where the high-temperature heat transfer medium coming from the condenser is heating the hot water inside the tank, see Figure 8.2. The heat transferred from the high-temperature heat transfer medium to the hot water is limited by the heat transfer capacity of the double-wall. Thus, a number of on and off cycles are normally needed for heating up the water. Using a variable-speed capacity controlled compressor instead, it will be possible to heat the tank to the set temperature in fewer cycles and also get a higher temperature of the sanitary hot water. This is possible because the capacity of the heat pump can be adapted to the heat transfer capacity of the heat exchanger (double wall). As the temperature of the water in the double-wall increases the capacity of the heat pump can be reduced, thus transferring heat at a lower capacity. This allows more heat to transfer into the storage tank before the temperature of the water in the double-wall gets too high and the operation is shut off. It is, of course, possible to heat sanitary hot water with a low capacity heat pump and in this way achieve the high temperature level, but then it will take a long time to heat up the water in the storage. Using a variable-speed capacity controlled compressor makes it possible to combine both high capacity, thus fast reheat, and low capacity and high temperature. There are other possibilities of heating the sanitary hot water but generally the better the heat transfer capacity of the heat exchange, the less important should be the possibility of adapting capacity.
Figure 8.2 The figure shows the parts and flows of a typical ground-source heat pump. The high-temperature heat transfer medium can either be used for space heating or for sanitary hot water heating. The use is controlled by the three-way valve. (Source: Nibe)
9 Summarising discussion, conclusions and suggestions for further work

The purpose of this work was to systematically investigate the possibilities for increased energy efficiency of small heat pump heating systems when applying new components and controls, such as variable-speed controlled compressors and circulators. The work has focused on ground-source heat pump systems as this application is most common in Sweden and it is also the least investigated application in this context. It has been shown that efficiency gains are possible when applying variable-speed compressors and pumps but care must be taken to choose efficient components and to control the speed of the circulator for the heat source system. Below these aspects are summarised and discussed in more detail.

9.1 Discussion

Previous investigations have shown the advantage of applying variable-speed capacity control of compressors in heat pumps. The work presented in this thesis partially confirms this but to be successful the compressor efficiency must still be high and efficient circulators must be used. An extra increase in efficiency is possible if applying an on-line optimisation method for optimising the combination of pump and compressor speeds to achieve the highest COP.

It has also been shown that the dynamics of the heating system, primarily the thermal inertia, must be considered in order to get a fair comparison between variable-speed and intermittent capacity control. It is not common to consider the dynamic effects in calculation programs used for calculating the seasonal performance of heat pump systems. When the dynamics are not considered the seasonal performance of intermittently controlled heat pumps is overestimated and will thus seem better when compared to a variable-speed system.

One central question is of course how much there is to gain by applying variable-speed capacity control. The question is not easy to answer as it depends on which system you use as base-line and on what technologies that are available. In this thesis I have chosen to compare the prototype heat pumps constructed during this work with the performance of a state-of-the-art heat pump available on the market. I do this to get as fair an impression as possible of the possible gains. However, this will change as technological development proceeds. Combining the results from chapter 4 and 5 make it possible to compare the efficiency of an all variable-speed controlled heat pump (variable speed compressor and pumps) to the efficiency of an intermittently controlled heat pump. For the heat pump with a single-speed compressor the application of variable-speed pumps result in a 10 % (1-0.97·0.93) increase in SPF compared to using standard single-speed pumps.

Using variable-speed pumps in a heat pump with a variable-speed compressor increases the SPF by 19 % (1-0.97*0.84) to 24 % (1-0.93·0.82) compared to using standard single speed pumps. Applying these results to the SPF for heat pumps B, C and D in chapter 4, see Table 4.1, give the results shown in Table 9.1.
Comparing the standard heat pump C to the prototype heat pump B with variable-speed pumps show that the latter is 10 % more efficient. With improved compressor efficiency (heat pump D) the SPF will be 27 % higher for the all variable-speed heat pump compared to the standard heat pump. However, currently the Swedish heat pump manufacturers have started to use more efficient pumps in their heat pumps and then the difference will be lower.

Table 9.1 Comparison of SPF for heat pumps with standard circulators and heat pumps with variable-speed circulators applying on-line optimisation

<table>
<thead>
<tr>
<th>Heat pump</th>
<th>SPF&lt;sub&gt;hs&lt;/sub&gt; standard circulators [-]</th>
<th>SPF&lt;sub&gt;hs&lt;/sub&gt; variable speed circulators with optimised control [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>2.7</td>
<td>3.2 - 3.3</td>
</tr>
<tr>
<td>C</td>
<td>3.0</td>
<td>3.3</td>
</tr>
<tr>
<td>D</td>
<td>3.1</td>
<td>3.7 - 3.8</td>
</tr>
</tbody>
</table>

In the investigation by Nowacki\textsuperscript{[60]}, referred to in the introduction, the heat pumps in Sweden are estimated to use 7.5 TWh of electricity per year and to deliver 22.5 TWh of heat. Using these data as a basis, if the efficiency of these heat pumps could be increased by approximately 20 % it would result in a decrease in electricity use by 1.5 TWh. This is 50 % more than the total electricity production by Swedish wind energy plants in 2006 (Swedish Energy Agency\textsuperscript{[77]}). Nowacki\textsuperscript{[60]} has assumed an average seasonal performance factor of 3 which must be considered as high, and as such the estimate of the potential savings is conservative. A lower SPF will mean more use of electricity and thus the potential savings would be even larger.

For on-line optimised operation of heat pump systems there are several different methods to choose between, such as direct search methods, model-based methods, steady-state or dynamic optimisation. Considering the results from chapter 5, that there is little to gain from optimising the operation of the circulator of the warm heat transfer medium, a rather simple control system should be possible to use. One approach could be to have the capacity of the heat pump, i.e. the compressor speed, controlled by the heating system return temperature. The set value of the return temperature should then be controlled by a combination of curve control, where the return temperature is related to the outdoor air temperature, and a room temperature sensor. The only parameter left to optimise is the capacity of the brine pump, which then preferably is optimised on-line or set as a linear function of the compressor speed as shown in chapter 5. If choosing an on-line optimisation method, the start value for the method, e.g. during start-up or large changes in conditions, could be the linear relationship to the compressor speed or the method proposed in chapter 3. As the number of variables now are down to one, it would be possible to use the Nelder-Mead simplex method, proposed in chapter 6. One benefit for a heat pump manufacturer by using an on-line direct search method for optimising the heat pump operation is that the method can be made general and thus be applied to different types of heat pumps without large changes to the controller design.
The methods and results in this thesis also apply to air-source systems. The energy saving potential of these systems should be larger as the frosting of the evaporator can be less when reducing the compressor capacity. Furthermore, as finned-tube heat exchangers are used, the effects of heat exchanger unloading will be larger, thus improving COP at reduced capacity more than was the case for the ground-source heat pumps investigated in this thesis. The method proposed for optimised defrost initiation did not give the highest COP for the given heat pump at give conditions but gave the highest COP for the given heat pump at given conditions and the defrost performed. However, as the results were quite close to the results obtained using the manufacturer’s control, the method should be possible to include in an overall optimisation procedure which then needs to include some kind of tuning parameter for the defrost control.

9.2 Conclusions

- It is important to control the capacity of the circulator of the heat source system if applying continuous capacity control to the compressor.

- For the heat pumps evaluated in this work the seasonal performance factor can be improved by 10% compared to state-of-the art ground-source heat pumps when applying all variable-speed capacity control. If improving the compressor efficiency of the variable-speed compressor up to the efficiency of the single-speed compressor the SPF can be improved by 27%.

- Using efficient pumps instead of standard pumps in the heat pump heating system can save as much as 0.5 TWh electricity in Sweden.

- The efficiency of the heat pump heating system is only slightly affected by the choice of heat emitters. Most important is the design of the system and the design supply temperature.

- Preferably an on-line optimisation method is used for controlling the operation of the heat pump system.

- The dynamics of the heating system and intermittent operation should be considered in energy efficiency calculations to achieve a good comparison between intermittent control and continuous capacity control.

9.3 Further work

As the compressor efficiency and integration of compressor motor and frequency converter is very important, further efforts should be made for optimising these. There are efficient compressors available, at least for the small air-to-air heat pumps, and a raised demand for efficient variable-speed compressors for the application of ground-source heat pumps should drive the technology forward. The efficiency of small circulators have increased in the last few years as permanent magnet motors have been introduced. Applying permanent magnet
motors also in compressors will increase also their efficiency. Further work is also needed for the on-line optimisation. It would be interesting to see a comparison of performance of a direct-search method and a model-based method. The method proposed for optimised defrost initiation needs to be further investigated. The use of capacity controlled compressors in heat pumps for heating sanitary hot water is also an area which should be investigated.
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