Economic analysis
of air-water heat pump technologies
with a screening method

Doctoral Thesis

By

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Abstract

Early on in the process of product development a decision has to be made which technologies to focus on. Under a purely techno-economic viewpoint for a heat pump only these technologies should be considered which maximize heat pump performance for a given cost. Finding these optima is in practice far from trivial as the result is influenced by variations in operating conditions, interactions between components, model assumptions and uncertain economic data. The overall objective of this thesis is therefore to develop a screening method for the evaluation and comparison of different technologies in regard of cost and efficiency with the aim to identify optimal heat pump designs.

In practice the applicability of such a method depends on the required effort. Simple mathematical models and short computation times are as mandatory as reliable, coherent and sufficiently general results. To this end the screening method combines methods of simulation, annual performance calculation, metamodelling and optimization.

The screening method is employed for air-water heat pumps in three utilization examples. In the first example the operational costs of maldistribution in evaporators are quantified. It is shown that for air-water heat pumps increasing the evaporator size is no liable option to counteract maldistribution effects. Two alternative technologies, an adjustment of the cycle layout and a change of the superheat control method, are evaluated under the aspect of total cost of ownership. Only with the second technology noteworthy savings can be achieved compared to the baseline. In the second utilization example on/off and variable speed control for regulation of the heat pump capacity are considered. Only for colder climates variable speed control pays off for the end consumer in a reasonable time. The study shows the comparison of the two control methods to strongly depend on compressor size. In the third example four different cycle layouts are compared. It is demonstrated that the result of the evaluation depends strongly on practical operating limits and on compressor characteristics. The most promising option is a staged layout with economizer heat exchanger.

Two additional studies consider the modeling of operating conditions in more detail. In the first study the simple approach for calculating annual performance used in the screening method is compared with a comprehensive dynamic model. Optimization for both model approaches results in similar component sizes. In the second study the evaporator model is extended to include a dynamic frost growth model which is used to assess operational costs induced by frosting and defrosting of the evaporator coil. A method to reduce the number of defrosts without
negatively affecting heat pump capacity is presented and its feasibility demonstrated with experiments.

The screening method can be extended to include more optimization parameters than used in the presented examples. It can also be applied for other small scale vapor compression systems or it can be integrated in a comprehensive evolutionary optimization procedure to reduce computation time and increase numeric stability. Thus the screening method can be a valuable tool in the product development process for shortening the times until new technologies which safe energy and thereby reduce greenhouse gases are available on the market.
Sammanfattning

I ett tidigt skede i utvecklingen av en ny produkt måste beslut tas om vilken teknologi som ska användas. Ur rent teknokonomisk synvinkel, ska, för en värmpump, bara de teknologier komma i fråga som maximerar värmpumpens prestanda för en given kostnad. Att hitta dessa optima är i praktiken långt från trivialt eftersom resultatet påverkas av varierande driftvillkor, samverkan mellan komponenter, antaganden i modellerna och osäkra värden för ekonomiska faktorer. Det huvudsakliga målet med denna avhandling är därför att utveckla en gallringsmetod för utvärdering och jämförelse av olika teknologier med avseende på kostnad och effektivitet med målet att identifiera den optimala utformningen av värmpumpen.


Två andra studier behandlar modellering av driftsforhållandena mer i detalj. I den första jämför den förenklade gallringsmetoden ovan för att beräkna årsvarmeffekter med en mer detaljerad dynamisk modell. Optimering av systemets komponenter med båda modellerna ger liknande optimala komponentstorlekar. I den andra studien ersätts
förångarmodellen med en mer detaljerad modell som också dynamiskt beräknar frosttillväxt. Denna används för att beräkna kostnaderna som orsakas av påfrostning och avfrostning av förångaren. En metod för att minska antalet avfrostningscykler utan att påverka värmepumpens kapacitet negativt presenteras och verifieras experimentellt.

Gallringsmetoden kan utökas till att omfatta fler optimeringsparametrar än de som använts i dessa exempel. Den kan också användas för små kompressordrivna system, eller den kan integreras i en omfattande evolutionär optimeringsprocedur för att minska beräkningstiden och öka den numeriska stabiliteten. Gallringsmetoden kan alltså vara ett värdefullt verktyg i produktutvecklingsprocessen och därigenom korta tiden för ny teknik som sparar energi och minskar växthuseffekten att komma ut på marknaden.
Publications

This thesis is based on following papers which are appended at the end.


IV. Mader G., Palm B., Elmegaard B. 2015 "Economic potential of cycle layout changes in residential R290 air-water heat pumps.", Submitted to Energy and buildings.

V. Mader G., Tiedemann T., Palm B. 2013 "Concepts to increase the seasonal coefficient of performance of an air-water heat pump: Comparison of costs and benefits", Clima2013 - 11th REHVA World Congress, Prague, Czech Republic.


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1 Introduction

1.1 Background and problem description

In Europe around 40% of the primary energy use is related to the building sector (IEA, 2014). According to EHPA (2009) about 75% of this energy demand is needed for space heating and hot water generation. The potential to save a considerable amount of energy by establishing energy efficient heating systems is therefore high. Electrically driven vapor compression heat pumps are an attractive option due to their capacity to reduce \( \text{CO}_2 \) emissions compared to conventional heating systems (Blum et al., 2010). In the EU more than 4.5 million units have been sold between 2005 and 2011 which are saving 43.95 TWh of final energy annually. Thereby greenhouse gas emissions are reduced by 8.13 Mt (Nowak, 2012). Also the future looks promising for heat pumps due to the political willingness in Europe to increase the share of renewable energy sources (BWP, 2013) and the potential of heat pumps to work as integrated energy buffer in future ”smart grids” (Tahersima et al., 2011).

Numerous research and development studies focus on heat pumps. Various technologies, either well known from other applications or based on innovative new ideas, promise to improve heat pump performance. Some examples are optimized heat exchanger designs (e.g. Domanski and Yashar, 2007), variable speed capacity control (e.g. Madani et al., 2011), oil free compressor technologies (e.g. Schiffmann and Favrat, 2009), special expansion solutions (e.g. Elbel and Hrnjak, 2008 or Mader and Thybo, 2010), advanced control algorithms (e.g. Sakellari et al., 2006 or Li et al., 2009), different cycle layouts (e.g. Elbel and Hrnjak, 2004) or other alternative approaches (e.g. Ling, 2004). A detailed review of recent trends was given by Chua et al. (2010).

Customer demands and legislation push for increases in energy efficiency but at the same time the heat pump market is highly competitive (Nowak, 2012) and especially for small residential heat pumps consumers are very price sensitive. Also manufacturer’s resources for research and development are limited. For the sake of an efficient product development process it is therefore desirable to quickly identify among the high number of possible technical solutions those that are the most promising. This constitutes a classical multi-objective optimization problem because for the manufacturer a good solution is characterized by several criteria like high energy efficiency, low material cost, short development times, low risk and complexity or high safety.
A technology that is optimal in regard of one of these criteria will most likely be inferior to another technology in regard of another criterion.

A rigorous mathematical formulation and solution of this optimization problem is challenging. The quantification of the optimization criteria or objectives is in practice an intricate task because of strongly varying or simply lacking data and a possibly complex system behavior. Additionally various constraints may be important to take into account in the process. Resulting objective and constraint functions will often be non-linear. A comprehensive design space contains all decision or optimization variables which describe relevant technology alternatives like cycle layout variations, refrigerant alternatives, component designs and control methods. Such a design space is large and contains optimization variables of both continuous and integer type. Solving such a constrained multi-objective non-linear mixed-integer optimization problem can quickly lead to prohibitive computation times, numerical instability and convergence difficulties. Assessing the sensitivity of the solution in regard of the assumptions used in the mathematical modeling as well as in regard to small variations in the input parameters requires additional effort. Therefore strongly simplifying this cumbersome optimization problem is mandatory to make a solution possible and a procedure to find optimal heat pump designs applicable in practice.

1.2 Existing solution approaches: Literature survey

Contributions in the open literature dealing with the optimization of small scale vapor compression cycles are numerous. One feature all studies have in common is the mathematical description of the cycle and the numerical solution of the resulting equation system as described by Gordon and Ng (2000). In every other respect the existing published optimization studies vary widely. They differ in the choice of the optimization criterion (i), in regard of the selected optimization variables (ii) and the applied optimization method (iii). Just recently, methods to reduce the simulation effort are included in the optimization procedure (iv). In the following some examples are given:

(i) Many optimization criteria or objective functions are of technical nature: Granryd (1969) discussed the optimal intervals of defrosting for either maximum energy saving or maximum net cooling capacity. The secondary fluid flow rates in heat pumps were adjusted by Granryd (2010) to maximize either the unit’s coefficient of performance or capacity. A second-law thermodynamic criterion was used by Bejan (1989) who minimized the irreversibility in the system by optimal distribution
of heat transfer area between the heat exchangers. Due to possible trade-offs between efficiency and cost, solutions resulting from technical optimization may be economically unfavorable. Therefore technical and economic criteria can be combined in a techno-economic optimization as presented by Bäckström (1940). Specific annual cost as ratio of annual capital and operational costs were evaluated for finned coil heat exchangers by Granryd (1971). Other recent examples are given by Mansour et al. (2008), Waltrich et al. (2011), Najafi et al. (2011) or Sanaye and Niroomand (2011). A disadvantage of techno-economic optimization is the importance of economic parameters like interest rates or lifetime expectations while technical optimization is largely independent of these.

(ii) Often variables describing characteristics of a single component are optimized. An example was given by Schiffmann and Favrat (2012) who optimized the design of a small-scale radial compressor to achieve minimal power consumption over the heating season. Another common approach is the simultaneous optimization of several components of the vapor compression unit. Bejan (1989) optimized the allocation of heat exchange area within a refrigeration plant. Similarly, Dingeç and İleri (1999) simultaneously optimized the heat transfer areas of evaporator and condenser and the isentropic efficiency of the compressor in a household refrigerator. The search for the optimal control strategy can also be the basis for selecting optimization variables. An example was given by Manente et al. (2013) who maximized the power generation in an organic Rankine cycle by optimizing the control strategy considering off-design conditions. Optimization methods are scarcely used when the focus is on the cycle layout or refrigerant. For industrial-scale multi-stage systems Vaidyaraman and Maranas (1999) defined a superstructure representation that allowed simultaneous optimization of refrigerant choice and layout of each stage. For small scale refrigeration applications comparative studies are more common to evaluate cycle layout and refrigerant choice. Bertsch and Groll (2008) performed a theoretical screening study of several cycles for air source heat pumps to maximize energy efficiency. Hwang et al. (2004) compared the life cycle performance of a refrigeration application with different refrigerants. Component characteristics were considered in different scenarios. A rare example for applying an optimization method with a mix of these variable classifications was given by Zehnder (2004), who rigorously formulated a optimization problem which allowed the simultaneous optimization of cycle layout, refrigerant and component selection.

(iii) Objective functions for vapor compression cycle optimization are usually non-linear and include both continuous and discrete optimization variables. The solution of this type of optimization problem is numerically costly. Numerous algorithms have been developed to minimize the numeric effort and have been employed for vapor compression cycle
optimization. The following overview is not exhaustive but gives examples of different mathematical complexity and numerical effort. The mathematically simplest optimization method is a parametric study. This approach was chosen by Röyttä et al. (2009) who varied intermediate pressure and compressor rotational speed to optimize the coefficient of performance of a two stage cycle. A sequential optimization method was applied by d’Accadia and de Rossi (1998). In this study a refrigeration plant was optimized by independently optimizing each component with the objective to maximize system performance. The authors stated that this approach was only feasible if little interaction exist between the components. A gradient-based algorithm, the steepest gradient method, was employed on the system level by Lu and Goswami (2002) to maximize thermal performance of a combined heat and power generation plant.

For multi-objective optimization problems not a single but multiple solutions exist. The defining characteristic of these solutions is that one objective function cannot be improved without degrading another objective function. The group of optimal solutions forms the Pareto or trade-off curve. Negrão and Hermes (2011) used a single objective search algorithm several times with different constraint functions to identify cost-efficiency optimal designs of a household refrigerator. A more common approach for solving multi-objective optimization problems is the implementation of genetic or evolutionary algorithms. An example was given by Gholap and Khan (2007), who optimized the heat exchangers of a refrigerator in terms of minimum cost and energy consumption. Sayyaadi et al. (2009) employed an evolutionary algorithm both for a first-law and second-law thermo-economic optimization of a ground-source heat pump.

(iv) Just recently a few attempts were made to reduce the simulation effort required to optimize heating systems by using a metamodel. Eisenhower et al. (2012) used results from dynamic building and environment simulations with varying input parameters to fit an analytical model. This simplified model was then employed in the optimization procedure. Khalajzadeh et al. (2011) fitted a second-order response surface model to simulations of a computational fluid dynamics model of a ground source heat exchanger. The metamodel was then used to maximize total heat transfer and heat exchanger efficiency. Another metamodel approach was used by Gholap and Khan (2007) for the multi-objective optimization of heat exchangers for refrigerators.

Even though much work is done in the field of optimizing small vapor compression systems, several questions arise when it is attempted to develop a practical, structured and meaningful method for finding optimal heat pump designs considering optimization variables both on the system and component level:
• Is it possible to simplify the problem by splitting the search space without loss of optimality?

Splitting the search space aims at a step-wise solution of the overall optimization problem by solving several simpler ”local” optimization problems with a reduced number of optimization variables. Approaching the global optimum with this method is however only possible if the interactions between the separate parts are small. While this is obviously the case for some technology changes, other technology changes may lead to strong variations of the conditions in all parts of the system. For such ”disruptive” technology changes the local optima would then vary. Whether these variations are of secondary importance in regard of the evaluation of a technology is not necessarily clear in advance.

• Is it possible to simplify the mathematical and numerical procedures required for optimization?

One option are metamodel approaches which allow a simplified formulation of objective functions. This influences the mathematical structure of the optimization problem and therefore the choice, robustness and stability of the optimization algorithm. However, the creation of simpler models may increase model errors and therefore distort results.

• Is it possible to evaluate the heat pump efficiency for realistic operating conditions without unjustified increase of effort?

Heat pumps work in a large range of operating conditions. However, heat pump technologies are predominantly optimized and compared at a single operating point. The choice of this condition plays a major role for the result. An evaluation based on an extreme condition might overestimate the real improvement potential. Optimization for a typical condition at the other hand might lead to strong deviations in off-design operation. The opposite approach, a detailed representation of realistic operating condition variations, raises questions about the required level of detail and the generality of the results.

1.3 Aim of the study

The overall objective of this study is to develop a screening method based on mathematical modeling and simulation for finding optimal heat pump designs among a large number of alternatives. This requires a problem formulation that allows a quick and stable evaluation. To this end approaches should be investigated for simplifying the mathematical models and decoupling parts of the optimization problem with little loss.
of optimality and sufficient generality. The screening method should be
developed and tested for a selection of competing technologies which
promise to improve the heat pump performance.

1.4 Structure of the thesis

Fig. 1.1 shows how the structure of this thesis is linked with the content
of the papers on the subject published in different scientific conferences
and journals. The introduction in the first, current chapter describes
background and problem and presents related literature. On this basis
the study aim is formulated. In Chapter 2 the methodological choices
are outlined. This includes a further demarcation of the study object
and reasoning for the approaches selected to simplify different aspects
of the problem. The integration of these approaches constitutes the
comprehensive screening method which is proposed for addressing the
overall study objective. Details of the screening method and the re-
quired models are discussed in Chapter 3 which is an extended version
of those parts of Papers I and II concerned with method description.

The screening method which is formulated in Chapters 2 and 3 is utilized
in different examples with the aim to demonstrate the feasibility of the
approach. The method utilization examples are presented in Chapters
4, 5 and 6 which correspond to Papers I - V. In each example several
technologies are compared which address a specific design aspect of the
heat pump. Chapter 4 discusses methods to counteract maldistribution
in evaporators. In Chapter 5 the classical on/off capacity control is
compared with variable control of the compressor speed. In Chapter 6
cycle layouts and refrigerant choices are evaluated under an economic
viewpoint.

Two aspects concerning the simplified description of encountered oper-
ating conditions are investigated in more detail. The influence of using
generalized operating condition profiles is evaluated by a comparison
with calculations employing a complex dynamic building model. This
is presented in Chapter 7 and in Paper VI. Chapter 8, an extension
of Paper VII, discusses the effects of frost building and defrosting, a
specific topic of air-water heat pumps.

In the last chapter conclusions are drawn both regarding the developed
screening method and the potential of different technologies to improve
heat pump performance. The second part is a summary of the conclu-
sions reached in Chapters 4 - 8. Finally some general recommendations
and proposals for further research are given.
Figure 1.1: Structure of the thesis.
2 Methodology

"Completeness is rarely, if ever, achieved by a model, especially when one is dealing with complex systems”
- Weisberg (2006)

Bearing this simple truth in mind, some choices have to be made in regard of what to include and what to exclude as well as what to do and what to refrain from doing within the framework of the research. In this chapter these choices are pointed out, reasons given and consequences discussed.

2.1 The focus application

A heat pump, sketched in Fig. 2.1, extracts heat from a low temperature source and moves it to a high temperature sink where it is rejected. To this end energy has to be provided. Heat pumps vary in regard of purpose, capacity, type of heat sink and source and form of used energy. In vapor compression heat pumps the required energy is provided in form of electricity. The compressor uses this electricity to compress a fluid, the refrigerant, and circulate it to the condenser. There it changes phase from vapor to liquid, rejecting heat on the high temperature level. The liquid is then expanded to a lower temperature and pressure level and fed to the evaporator. There it takes up energy, changing back to the vapor phase before being compressed again to the high temperature level.

Figure 2.1: Sketch of an air-water heat pump.

In Europe the majority of sold heat pumps are vapor compression units in the capacity range of 2 to 20 kW for space and water heating in single family houses (Forsén, 2010). The larger part of these heat pumps are
used in retrofitting projects, meaning that in an older building the existing heating technology is replaced with a heat pump. In these projects air-water heat pumps are the most commonly used type, because today most European buildings use water to distribute heat and air is the heat source which is the cheapest and easiest to make accessible (Karls- son, 2010). With regard to sales numbers the market for European residential air-water heat pumps is a typical example for a highly cost and efficiency competitive mass market, which makes this application a suitable focus for this study.

Only air source units with direct heat transfer from the ambient air to the refrigerant as shown in Fig. 2.1 are considered. Indirect systems with a secondary fluid circuit as discussed by Granryd et al. (2009) are not taken into account.

2.2 Optimization criteria

In general multiple stakeholders have an interest in a heat pump. Each stakeholder usually has several requirements or objectives, which might be in accordance with or in opposition to other expectations by the same or other stakeholders. Thus a complex web of requirements evolves. All objectives formulated by the stakeholders are influenced by external factors. As these factors change, also the objectives will change.

External factors, stakeholders and requirements for a single family residential heat pump are visualized in Fig. 2.2. The lists make no claim to be complete but should give an impression of the complexity which has to be faced by decision makers dealing with finding optimal heat pump solutions. External factors are customs and habits, available resources, the structure of the market and many more. Stakeholders are the component and heat pump manufacturers, the installer, electric power company, the end customer and the society. Typical requirements, formulated by one or more stakeholders, are among others high safety and comfort, low risk and complexity, environmental friendliness and low costs.

In principle all requirements could be quantified and translated to mathematically treatable objective functions. However, many criteria are of subjective nature while for others simply no data exist. Methods for multi-criteria decision analysis (MCDA) exist which try to handle these difficulties. A state of the art can be found in Figueira et al. (2005). All MCDA methods have in common that the requirements are weighed and scored for several possible alternatives. Different mathematical approaches exist which transfer the weights and scores with the aim to calculate an objective ranking of alternatives and thereby find the best
one. However, independent of the nature of the mathematical algorithm, all MCDA methods rely on more or less subjective evaluations of objectives and alternatives. Dealing with this subjectivity is inevitable for decision making, but in my belief an objective mathematical treatment of the problem as a whole is not feasible. Therefore this study only focuses on criteria which can be objectively quantified.

This research project has been initiated by a component manufacturer, the funding company Danfoss, with the aim to anticipate the interest of his customers, the heat pump manufacturers, in new or improved components. Low purchase costs are mandatory for the competitiveness of a heat pump manufacturer and high heat pump efficiency is his main sales argument. Both criteria can also be objectively quantified, thus they present themselves as suitable objectives.

### 2.3 The screening method

An optimal solution is characterized by an optimization criterion, expressed as objective function, reaching a minimum or maximum. Finding this optimum requires repeated evaluation of the objective function with varying values of the optimization variables in a structured manner. Executing the required repetitions and the strict control of opti-
mization and other input parameters in an experimental setup would be extremely costly if not infeasible. Mathematical modeling of the system is therefore mandatory. However, also this method quickly becomes cumbersome for complex systems: Typically the objective function is described by a set of equations which has to be solved numerically. This is computationally costly and produces numerical instabilities. Optimization algorithms dealing with non-continuous objective functions require per se a higher number of evaluations of the objective function. This number increases dramatically with an increasing amount of optimization variables. Additionally these algorithms suffer from convergence problems and risk getting stuck in a local optimum.

The research goal is to find cost and efficiency optimal heat pump designs among many different alternatives. This requires minimizing both computational effort and convergence problems. To this end seven steps, depicted in Fig. 2.3 and described in more detail below and in chapter 3, are combined in this study to a screening method:

(i) The search space containing all optimization variables is split in a continuous and an integer part. Only continuous variables are part of a mathematical optimization. Integer variables describing step functions or yes/no choices are not subject to optimization but characterize technology alternatives.

(ii) For the heat pump unit a detailed but generic component-based thermophysical model is developed.

(iii) A performance map created from the heat pump unit model is embedded in a system and environment calculation. Both parts form the annual performance model of heat pump efficiency.

(iv) The effort required for annual performance calculation is reduced by using generalized operating condition profiles describing annual variations in the system and environment.

(v) A continuous metamodel of the annual performance model is obtained with curve fitting techniques. Continuous optimization variables form the input parameters of the metamodel. The structure of the metamodel is optimized and the effort of curve fitting minimized by employing statistical methods from the design of experiment theory. Additional metamodels for constraining parameters can be developed.

(vi) The metamodel is combined with an independent economic model. Since both parts are in the form of continuous functions, a continuous objective function is created.

(vii) Algorithms for continuous constrained optimization problems are applied to find the optimal design by varying input parameters of the objective function.
The search space is defined by all optimization variables and their upper and lower values. The overall optimization problem is characterized by a mix of continuous and integer variables, which is called a non-convex or mixed-integer problem. Examples for an integer optimization variable are the refrigerant selection or the yes/no decision for an added component which changes the cycle layout. An example for a continuous variable is a control parameter like the air flow rate induced by a fan with variable speed. Many design variables are of the discrete type, e.g. only standard sizes of tubes, plates or compressors can be chosen. In some cases such variables can be treated as "quasi-continuous", assuming that the value given by the continuous optimization can be rounded to the next discrete value without considerable loss of optimality. As Yuan et al. (2010) pointed out, for integer variables this is a reasonable approach as long as the domain of the integer variable is large. Translated to discrete variables this means that a sufficiently large amount of selections must be possible within the considered range. This is true for the different "quasi-continuous" variables chosen in this study, namely the heat transfer areas of a brazed plate condenser and a fin and tube evaporator as well as the displacement volume of the compressor. For these variables the simplification is additionally justified by the fact that the optimum is "flat", meaning that small variations around the optimal value of the optimization variable only leads to small variations in the value of the objective function.

The major advantage of the proposed splitting of the design space is that an apt transformation of the objective function allows employing gradient-based optimization algorithms for the continuous variables.
These algorithms are more stable and at the same time much faster than mixed-integer types because to converge they require a considerably reduced number of evaluations of the objective function.

The obvious disadvantage of the search space split is that integer variables characterizing technology alternatives like refrigerant selection or cycle layout design are not subject to an automated optimization procedure. Instead a screening for the best alternative by comparing several optimized designs is required. Two approaches are proposed to integrate the continuous and integer optimization parts, without falling back to a "brute force" mixed-integer method: In principle the continuous metamodel and optimization procedure can be integrated in a superimposed integer optimization procedure. While on the first level an evolutionary algorithm deals exclusively with the integer part of the problem, on the second level the metamodeling and optimization of the continuous part is performed. Such an extension to a "two-level" optimization procedure is methodically straightforward, while technically challenging, and therefore not part of this thesis. Alternatively, a serial optimization of integer and continuous variables can be imagined. In a first step the integer variables are optimized and in a second step the optimal solution of the first step is further optimized with the continuous optimization approach. This "top-down" optimization approach could radically simplify the overall optimization procedure. However, finding a global optimum with this method requires a low level of interaction between the two parts of the optimization problem. Whether a sufficient independence exists between integer and continuous variables cannot be answered a priori but will be discussed based on the method utilization examples.

Annual performance calculations require accounting for widely varying operating conditions. To reduce the simulation effort, generalized operating condition profiles are derived based on the standard EN14825 (2010) as described in Chapter 3.3. A restriction of this choice is that these profiles are defining only space heating but not hot water heating, even though both purposes are typically integrated in European heat pumps. The effect of utilizing generalized operating condition profiles on the precision of the results are tested by a comparison to calculations with a comprehensive dynamic system and environment model, discussed in Chapter 7. This comparison is performed for a ground-water heat pump due to the availability of a suitable dynamic model, but is expected to be also valid for air-water heat pumps. Another simplification on the level of annual performance modeling is that frosting and defrosting, a specific feature of air-source heat pumps, is neglected in the optimization procedure due to the complex nature of these effects. To estimate the effect of frost and defrost on annual performance, a generic frost model is developed in Chapter 8.
A metamodel is a simplified model of a model. A curve-fitting technique from the design of experiment theory is employed to derive a second-order response surface model of the annual heat pump efficiency. With this approach, described in detail in Chapter 3.5, the optimization variables are varied by the curve fitting algorithm instead of the optimization algorithm, which considerably reduces the number of required simulations. An adequate choice of the statistical design also avoids evaluation of the simulation model at extreme values of the optimization variables, thereby reducing numerical instability. Errors induced by the metamodel are quantified in the different method utilization examples. Additional metamodels can be developed for constraining parameters like the compressor discharge temperature. These functions can be included as constraints in the optimization procedure.

The continuous metamodel of annual heat pump performance is combined with continuous component cost functions. Due to the resulting simple mathematical form of the objective functions, comparably fast and stable continuous optimization algorithms can be applied. Since heat pump efficiency is directly related to operating cost, the two objective functions efficiency and cost can even be combined to a single objective, the total cost of ownership.

### 2.4 Choice of optimization variables

The presented screening method is utilized in three different examples to demonstrate its possibilities and limitations. In these examples the potential of several up-to-date technologies to improve the performance of air-water heat pumps is quantified and compared. From a mathematical point of view these different technologies form the selection of integer variables:

A technique for counteracting maldistribution has been developed at the sponsoring company and its potential should be investigated in the framework of European air-water heat pumps. The topic is presented in Chapter 4. A spin-off, the passive defrosting of evaporator coils with the same product, is additionally investigated in Chapter 8.5. Improving heat pump performance by adjusting heat pump capacity with variable speed compressors is an obvious approach, several publications describe a promising improvement potential. Nonetheless, market introduction of the well-known technique is remarkably slow. In Chapter 5 possible reasons are delineated. The choice of the refrigerants currently found in the heat pump market is largely influenced by legislative issues. However, refrigerant phase-out projects of the European Union lead to an increased interest in alternatives. Since the feasible refrigerants strongly vary regarding their thermophysical properties, refrigerant selection is
tightly connected with cycle layout optimization. Both aspects are discussed under an economic viewpoint in Chapter 6.

For each technology several continuous variables are optimized mathematically before comparison to an alternative technology. Chosen optimization variables vary for the different utilization examples. Taken into consideration are the size of the compressor, namely the displacement volume, and the size of the heat transfer area of the evaporator, condenser and if applicable internal heat exchanger. Many other optimization variables exist on the component level and could be integrated in the procedure, e.g. tube diameters or fin spacing. However here the focus is on component sizes because they are expected to have a direct and strong effect both on annual system efficiency and purchase costs. As the main goal of the thesis is method development, it is refrained from extending the continuous optimization procedure to include more variables. Other variables like the air flow rate of the evaporator and the intermediate pressure level in two-stage cycles are in some utilization examples optimized by simple parameter variation but not included in the continuous optimization procedure. In the utilization examples the optimal values of all considered continuous parameters are presented and discussed for the different technology alternatives.

2.5 Data collection

A component-based structure of the thermophysical heat pump model is developed. For each component suitable modeling approaches are chosen which are widely used in science and industry and have been validated by different parties. An example is the ten-coefficient polynomial approach to describe compressor characteristics in dependency of suction and discharge temperature. The parameters of these component models are fitted to experimental data provided by the funding company Danfoss. For a few parameters such data was not available. Examples are exact values for the air side pressure drop or evaporator maldistribution of the focus application. In these cases experimental data published in the recent literature is used.

The economic functions for heat pump investment costs are based on data for sales prices of individual components. This data is provided by different Danfoss marketing and sales departments. Mostly data for current product series are used to derive functions which present component costs in dependency of size. Costs for technologies which at the time of the research were only sold for other applications are adjusted for the heat pump market based on the experience of the sales and application people. Examples are the two-stage compressor with suction port or the integrated expansion-distribution valve.
Component sales prices usually not only reflect material and production costs but are also affected by the price policy of the component manufacturer. While for standard components competition leads to a relatively constant market price, the margins for new technologies can be larger. However, from the viewpoint of the heat pump manufacturer who tries to maximize efficiency at minimal cost, the purchase costs of the components are the relevant measure. For the end customer on the other hand the price policy of the heat pump manufacturer may play the more important role. The difference between component sales costs and end customer prices is taken into account with simple multipliers based on practical experience. This approach of course cannot reflect possibly complex price policies of heat pump manufacturers. However, the decoupling of simulation and economic modeling allows quickly investigating the influence of varying economic framework. The procedure can also be used to calculate the cost maximum at which a technology change pays off for the end consumer. Such a reversed problem setup can help manufacturers to assess profit margins.

The benefit of data collection and parameter fitting which relies on realistic component data is that the plausibility of the results is increased. However, the generality may suffer, as other components from other manufacturers may have different thermophysical and economical characteristics. To address this issue, all evaluations include sensitivity testing of the results towards the different model assumptions and input parameters.

Experimental studies were performed in the laboratories of Danfoss and the heat pump manufacturer Danfoss Thermia. They were used both for gaining a good understanding of the characteristics and demands of air-water heat pumps today available on the European market and for verifying the transient frost growth model presented in Chapter 8.2.
3 Modeling

3.1 Model overview

The screening method implies specific system boundaries for the heat pump, visualized in Fig. 3.1. The first level is the heat pump unit, consisting of the components required for the thermodynamic cycle. This level corresponds to the classification given by Lundqvist (2011) and is reflected by the thermophysical model described in Chapter 3.2. Model outputs are heat pump coefficient of performance COP and capacity \( \dot{Q} \). The heat pump investment costs \( I \), presented in Chapter 3.4, are also formulated on this level. All variations of optimization variables, both of integer and continuous type, affect the design and therefore output of the heat pump unit model.

Figure 3.1: System boundaries of the annual performance model.

The second level, the heat pump system, is the interface between heat pump unit and building. As described in Chapter 3.3, on this level deviations between building demand and heat pump capacity are accounted for. An annual performance parameter, for example the annual work input \( W \), is calculated on this level. Opposite to the definitions by Lundqvist (2011), characteristics of the heat source and heat distribution system are not included here but combined with the building system to the level of the system environment. This last level is characterized by generalized operating condition profiles for building heat demand and temperatures of heat source and sink, described in Chapter 3.3. These profiles define the input parameters for heat pump unit simulations.
A metamodel of the chosen annual performance parameter is derived for the continuous optimization variables. As shown in Fig. 3.2, the annual performance calculation of Fig. 3.1 is repeated several times with varying values of the optimization variables. The value combinations are determined by the chosen curve fitting algorithm as discussed in Chapter 3.5. Based on the results, a continuous metamodel is derived as function of the optimization variables $\vec{Z}$. In case of single objective optimization, the metamodel is converted to express operating costs $O$ and combined with investment costs $I$. The resulting objective function is used for minimizing total cost of ownership and thereby optimizing the heat pump design as presented in Chapters 3.5 and 3.6.

![Figure 3.2: Metamodelling and optimization.](image)

### 3.2 Thermophysical model of the heat pump unit

Each steady-state cycle model consists of individual models for each of the components of the heat pump unit. In this chapter the model for a basic cycle is described, shown in Fig. 3.3, comprising a compressor, condenser, evaporator with fan and expansion device. The values for mass flow, enthalpy and pressure are communicated between the component models. This basic model is adjusted according to the specific questions of the different application examples as described in the relevant chapters.
Figure 3.3: Main components in a heat pump unit with basic cycle layout.

**Cycle equations**

The coefficient of performance COP is calculated as ratio of the capacity delivered at the condenser to the water side $\dot{Q}$ and the power consumption of the compressor $P_p$ and the air side fan $P_f$. The power consumption of the pump on the water side is neglected, as it is often not an integral part of the heat pump unit but of the hydronic system in the building.

$$\text{COP} = \frac{\dot{Q}}{P_p + P_f} \quad (3.1)$$

The overall energy balance relates the capacity of the evaporator $\dot{Q}_e$, condenser, and power consumption of the compressor.

$$\dot{Q}_e = \dot{Q} - P_p \quad (3.2)$$

**Compressor**

Compressor manufacturers typically describe nominal compressor power consumption $P_{p,0}$ and nominal evaporator capacity $\dot{Q}_{e,0}$ with ten-coefficient polynomial functions of nominal condensing temperature $T_{c,0}$ and evaporating temperature $T_{e,0}$ (Eqs. (3.3) and (3.4)). The coefficients $j_i$ and $k_i$ of these polynomials are determined experimentally for a specific refrigerant, superheat and subcooling. Eqs. (3.3) and (3.4) can be used with the same coefficients also for other refrigerants than specified by the manufacturer if refrigerant properties are similar. In this case $T_{c,0}$ and $T_{e,0}$ represent saturation temperatures of the nominal refrigerant at condensing and evaporating pressure of the refrigerant of interest.

$$P_{p,0} = j_0 + j_1 T_{e,0} + j_2 T_{c,0} + j_3 T_{e,0}^2 + j_4 T_{e,0} T_{c,0} + j_5 T_{c,0}^2$$
$$+ j_6 T_{e,0}^3 + j_7 T_{e,0}^2 T_{c,0} + j_8 T_{e,0} T_{c,0}^2 + j_9 T_{c,0}^3 \quad (3.3)$$

$$\dot{Q}_{e,0} = k_0 + k_1 T_{e,0} + k_2 T_{c,0} + k_3 T_{e,0}^2 + k_4 T_{e,0} T_{c,0} + k_5 T_{c,0}^2$$
$$+ k_6 T_{e,0}^3 + k_7 T_{e,0}^2 T_{c,0} + k_8 T_{e,0} T_{c,0}^2 + k_9 T_{c,0}^3 \quad (3.4)$$
To adjust nominal values for actual superheat and subcooling, following equations are employed:

\[
P_p = P_{p,0} \frac{\rho \eta_v \Delta h_{is} \eta_{is,0}}{\rho_0 \eta_{v,0} \Delta h_{is,0} \eta_{is}} \tag{3.5}
\]

\[
\dot{Q}_e = \dot{Q}_{e,0} \frac{\rho \eta_v (h_{e,\text{out}} - h_{e,\text{in}})}{\rho_0 \eta_{v,0} (h_{e,\text{out}} - h_{e,\text{in}})_0} \tag{3.6}
\]

Nominal refrigerant properties are also calculated at the saturation temperatures of the nominal refrigerant. The differences between \( \eta_v \) and \( \eta_{v,0} \) and between \( \eta_{is} \) and \( \eta_{is,0} \) are assumed to be small, these parameters are thus eliminated from Eqs. (3.5) and (3.6). For component based modeling it is useful to directly calculate the refrigerant mass flow rate in the compressor model instead of the evaporator capacity. For this purpose Eq. (3.6) is combined with Eq. (3.28). The equations implemented in the compressor model are thus:

\[
P_p = P_{p,0} \frac{\rho \Delta h_{is}}{\rho_0 \Delta h_{is,0}} \tag{3.7}
\]

\[
\dot{m}_r = \frac{\dot{Q}_{e,0} \rho}{\rho_0 (h_{e,\text{out}} - h_{e,\text{in}})_0} \tag{3.8}
\]

In this study an R407C compressor is employed in R290 heat pump models. The ratio of the saturation pressure of R290 to the saturation pressure of the nominal refrigerant R407C varies between 1.18 at -26°C and 0.82 at 70°C which corresponds to absolute pressure differences of 0.3 bar for minimum temperatures on the suction side and 5.9 bar for maximum temperatures on the discharge side. Since the refrigerants can be viewed as similar in respect to saturation pressures over the whole operating range, it is assumed that these corrections for refrigerant, superheat and subcooling are suitable here (Tiedemann, 2013).

If compressor speed is controlled with a frequency converter, additional losses occur. In this case, manufacturer polynomials are used to calculate correction factors for \( P_{p,0} \) and \( \dot{Q}_{e,0} \). These correction factors are functions of \( T_{c,0} \), \( T_{e,0} \) and rotational speed and take into account both losses in the inverter and motor.

Compressor isentropic efficiency \( \eta_{is} \) and volumetric efficiency \( \eta_v \) are useful to evaluate and compare compressors. They are calculated with following equations using the isentropic compressor outlet enthalpy \( h_{out,\text{is}} \), refrigerant density at compressor inlet \( \rho \) and compressor volume flow \( \dot{V}_p \). The volume flow rate depends on the volume of the compressor chamber and the rotational speed of the compressor.
\[ P_p = \dot{m}_r \frac{h_{\text{out, is}} - h_{\text{in}}}{\eta_{\text{is}}} \] (3.9)

\[ \dot{m}_r = \rho \dot{V}_p \eta_v \] (3.10)

In a simplified version of the compressor model, Eqs. (3.9) and (3.10) can replace Eqs. (3.7) and (3.8). \( \eta_{\text{is}} \) and \( \eta_v \) are then input parameters.

**Fan**

![Figure 3.4: Schematic of fan and evaporator pressure drop curves.](image)

Schematic pressure drop curves for the evaporator and fan are depicted in Fig. 3.4. Using the simple approach described by Cai (2007), the evaporator pressure drop can be described with Eq. (3.11) as a quadratic function of the air volume flow rate \( \dot{V}_a \). An empirical constant \( \psi_a \) describes the inclination. \( \psi_a \) depends on numerous parameters describing the evaporator air side geometry, e.g. fin spacing, tube spacing and number of tube rows and columns.

\[ \Delta p = \psi_a \dot{V}_a^2 \] (3.11)

The fan power consumption \( P_f \) can be expressed with the fan efficiency \( \eta_f \) as function of pressure drop and volume flow rate:

\[ P_f = \frac{\Delta p \dot{V}_a}{\eta_f} \] (3.12)

If an evaporator and a fan are combined, the operation point is described by the crossing of fan and system curve. Equating Eqs. (3.11) and (3.12) leads to the cubic equation Eq. (3.13) for the fan power. The factor \( \psi_f \) is the ratio of \( \psi_a \) and \( \eta_f \) and thus describes both evaporator and fan characteristics.
The baseline value used for $\psi_f$ in this study is 1 and is varied in sensitivity studies. $\psi_f$ is kept constant for all operating conditions and considered design options. According to Eq. (3.13) with increasing air volume flow rate the fan power consumption increases. On the other hand an increasing volume flow and thus air velocity leads to a higher air side heat transfer coefficient and thereby overall evaporator heat transfer coefficient. This effect is reflected in the evaporator model with Eq. (3.36) and the air side heat transfer correlation presented in Table 3.1. The induced increase of the evaporation temperature reduces compressor power consumption. Therefore the air flow rate can be subject to optimization as pointed out by Granryd (2010). With the value $\psi_f=1$ of Eq. (3.13) the optimization of the air volume flow rate to minimize annual power demand for a baseline fixed speed heat pump results in an annual fan power demand that accounts for about 6.7% of the total annual power demand. This corresponds well with the findings of Miara (2012) who determined in field tests typical annual fan power demands below 7% of the total power demand. Air volume flow rates are individually optimized for different technology changes in the studies presented in Chapters 5 and 6.3.

### Heat Exchangers

The physics of heat exchangers are challenging to describe mathematically. As discussed by Willatzen et al. (1998), the three-dimensional mass and energy balances as well as the Navier-Stokes equations have to be solved to completely describe the fluid mechanics. But since the computational effort for this approach is much too high, a number of various one-dimensional models to simulate transient two-phase flows have been developed. These can be generally divided, as outlined by Wang et al. (2007), into two categories, the moving boundary or lumped parameter approach, and the distributed models. The latter are generally said to be more flexible regarding geometry and maldistribution aspects (Jia et al., 1999) and give more detailed and precise results (Wang et al., 2007). However, as pointed out by Willatzen et al. (1998), they suffer from significant complexity and are computationally heavy.

Instead of using a high number of spatially equally distributed cells, moving boundary models divide a heat exchanger into a minimum number of zones according to the present state of the fluid, i.e. gas, liquid or two-phase, and provide a set of equations for each of these zones. Fluid properties and heat transfer coefficients are averaged in each zone. Hence the number of equations to be solved is reduced significantly and
Figure 3.5: Sketches of the condenser and evaporator lumped parameter model.

Simulations become much faster compared to distributed models. As the focus here is on cycle aspects, the moving boundary approach is chosen for heat exchanger modeling. This requires a strong simplification of the real heat exchanger geometries. Fig. 3.5 shows sketches of the lumped parameter heat exchanger models with temperature profiles. The model approach used here applies $\epsilon$-NTU correlations to connect inlet and outlet temperatures of refrigerant and secondary fluid in each zone. Pressure drop on the refrigerant side is mostly neglected except for the study of evaporator maldistribution in Chapter 4. Pressure drop on the water side of the condenser is neglected, pressure drop on the evaporator air side is considered with Eqs. (3.11) - (3.13).

Condenser

A brazed plate design is typical for condensers in air-water heat pumps. The flow paths of refrigerant and water are formed by the gaps between parallel plates. Both refrigerant and water flow are apportioned and distributed alternately to the different gaps, usually in counter flow arrangement. The condenser geometry can vary regarding length, width, pattern and number of parallel plates and regarding the flow paths within the heat exchanger.
This geometry is approximated with a single path counter flow structure as shown in Fig. 3.5.a. Using the lumped parameter approach it can be divided into a gas or desuperheating zone at the inlet, a two phase or condensing zone and a liquid or subcooling zone. Water flow rate \( \dot{m}_w \) and water outlet temperature \( T_{w,\text{out}} \) are input parameters provided by the system environment. The total heat transfer area \( A_c \) is also an input parameter.

The capacity \( \dot{Q} \) transferred from the refrigerant to the water, and thus the building, is composed of the heat transfer in the three zones:

\[
\dot{Q} = \dot{Q}_{c,g} + \dot{Q}_{c,2p} + \dot{Q}_{c,l}
\]

\[
\dot{Q}_{c,g} = \dot{m}_r (h_{c,in} - h_{c,dew})
\]

\[
\dot{Q}_{c,l} = \dot{m}_r c_p \Delta T_{\text{SC}}
\]

The subcooling \( \Delta T_{\text{SC}} \) is a cycle parameter. In a heat pump cycle without accumulator the main part of the refrigerant charge is distributed between condenser, compressor and evaporator. Changing evaporating and condensing pressures lead to changes in liquid and gas refrigerant densities and thus to variations in the refrigerant mass that can be stored in the different components. In residential heat pumps with dry expansion evaporator variations in the total charge in the system result in variations of the length of the subcooling zone in the condenser. The subcooling zone length is correlated with the amount of heat transferred in this zone and thus with \( \Delta T_{\text{SC}} \). The equations for modeling the correlation of refrigerant charge distribution and \( \Delta T_{\text{SC}} \) are given in Mader et al. (2010). However, charge modeling requires detailed information about internal volumes of the individual components. To keep the model simple and generic, charge modeling is not included in this study. Instead \( \Delta T_{\text{SC}} \) is an input parameter. In the most simple case, \( \Delta T_{\text{SC}} \) is zero. As described by Corberán (2012), in a real system this can be achieved by an accumulator at the outlet of the condenser. In Chapter 6.3 the influence of \( \Delta T_{\text{SC}} \) on annual performance is studied.

For the secondary fluid following heat balances can be formulated with \( T_{w,1} \) and \( T_{w,2} \) as water temperatures at the outlet and inlet of the condensing zone, respectively:

\[
\dot{Q} = \dot{C}_w (T_{w,\text{out}} - T_{w,\text{in}})
\]

\[
\dot{Q}_{c,g} = \dot{C}_w (T_{w,\text{out}} - T_{w,2})
\]
\[
\dot{Q}_{c,t} = \dot{C}_w (T_{w,1} - T_{w,\text{in}})
\]  \hspace{1cm} (3.20)

In case of a single chemical or an azeotropic mixture the condensing temperature \(T_c\) is constant. This implies an infinite heat capacity rate. Therefore the capacity flow rate of the secondary fluid is always the smallest and determines the maximally transferable heat. Thus following \(\epsilon - \text{NTU}\) correlation applies:

\[
\dot{Q}_{c,2p} = \epsilon_{c,2p} \dot{C}_w (T_c - T_{w,1})
\]  \hspace{1cm} (3.21)

\[
\epsilon_{c,2p} = \left(1 - \exp\left(-\frac{L_{c,2p} U_{c,2p} A_c}{\dot{C}_w}\right)\right)
\]  \hspace{1cm} (3.22)

For the gas zone, and analogously for the liquid zone, following formulation for counterflow applies:

\[
\dot{C}_{c,g} < \dot{C}_w : \epsilon_{c,g} = \frac{(T_{c,in} - T_c)}{(T_{c,in} - T_{w,2})} \frac{\dot{C}_{c,g}}{\dot{C}_w}
\]  \hspace{1cm} (3.23)

\[
\dot{C}_w < \dot{C}_{c,g} : \epsilon_{c,g} = \frac{(T_{c,in} - T_c)}{(T_{c,in} - T_{w,2})} \frac{\dot{C}_{c,g}}{\dot{C}_w}
\]  \hspace{1cm} (3.24)

The summation condition links the zone lengths:

\[
1 = L_{c,g} + L_{c,2p} + L_{c,t}
\]  \hspace{1cm} (3.25)

The calculation of the \(U_c\) values of the different zones \(i\) varies depending on the application example. Either a constant value is used, or \(U_c\) is calculated with the heat transfer coefficients \(\alpha\) of the refrigerant and water side. The heat transfer coefficients of each zone and fluid are
calculated using the correlations for brazed plate heat exchangers given in Table 3.1. Correlation parameters are fitted to experimental data of the considered geometry. The resistance of the plate is neglected, the heat transfer areas of the refrigerant and water side in the plate heat exchanger are assumed equal.

\[
\frac{1}{U_{c,i}} = \frac{1}{\alpha_{c,i}} + \frac{1}{\alpha_w}
\]  

(3.26)

Table 3.1: Correlations for heat transfer coefficients

<table>
<thead>
<tr>
<th>Single phase</th>
<th>Fin and tube</th>
<th>Braided plate</th>
</tr>
</thead>
</table>

Evaporator

In air-water heat pumps the typical evaporator geometry consists of round copper tubes with aluminum fins. As shown in Fig. 3.6, the tubes are arranged in bundles with several horizontal rows and vertical columns. The fins are formed by thin plates connecting the tubes. The air flows across the tube bundle. Multiple tubes connected with bends form a refrigerant flow path as indicated with the blue dotted arrows. The refrigerant is usually split and distributed to several such passes. The pattern that describes which tubes are linked to form the flow paths is called the evaporator circuitry. Since the heat transfer in one tube affects the air temperature seen by tubes in flow direction of the air, this circuitry can be subject to optimization.
Reflecting the evaporator circuitry and the interaction between tubes would require a finite element model approach. Here the geometry is simplified to a simple cross flow evaporator (Fig. 3.5.b). The equation system is analogous to the one of the condenser. However, only a two phase zone and superheat zone occur, the air volume flow rate is distributed to both zones according to the zone lengths $L$.

\[
\dot{Q}_e = \dot{Q}_{e,g} + \dot{Q}_{e,2p} \tag{3.27}
\]

\[
\dot{Q}_e = \dot{m}_r (h_{e,\text{out}} - h_{e,\text{in}}) \tag{3.28}
\]

\[
\dot{Q}_{e,g} = \dot{m}_r c_p \Delta T_{SH} \tag{3.29}
\]

\[
\Delta T_{SH} = T_{e,\text{out}} - T_e \tag{3.30}
\]

The degree of superheat $\Delta T_{SH}$ is an input parameter which is either constant or depends on superheat control characteristics (Eq. (3.38)). The $\epsilon$ - NTU correlation for the two phase zone is written as:

\[
\dot{Q}_{e,2p} = \epsilon_{e,2p} L_{e,2p} \dot{C}_{a} (T_{a,\text{in}} - T_e) \tag{3.31}
\]

\[
\epsilon_{e,2p} = \left( 1 - \exp \left( \frac{-U_{e,2p} A_e}{\dot{C}_a} \right) \right) \tag{3.32}
\]

For the superheat zone a simplified $\epsilon$ - NTU correlation is used which was reported by Fösel (2008) to show satisfying accuracy for heat exchangers with parallel passes:

\[
\dot{C}_{e,g} \leq L_{e,g} \dot{C}_a : \epsilon_{e,g} = \frac{(T_{e,\text{out}} - T_e)}{(T_{a,\text{in}} - T_e)} \tag{3.33}
\]

\[
L_{g} \dot{C}_a < \dot{C}_{e,g} : \epsilon_{e,g} = \frac{\dot{C}_{e,g}}{L_{e,g} \dot{C}_a} \frac{(T_{e,\text{out}} - T_e)}{(T_{a,\text{in}} - T_e)} \tag{3.34}
\]

\[
\dot{C}_{e,g} \leq L_{e,g} \dot{C}_a : \epsilon_{e,g} = \left( 1 - \exp \left( \frac{-L_{e,g} U_{e,g} A_e}{\dot{C}_{e,g}} \right) \right) \tag{3.35}
\]

\[
L_{e,g} \dot{C}_a < \dot{C}_{e,g} : \epsilon_{e,g} = \left( 1 - \exp \left( \frac{-U_{e,g} A_e}{\dot{C}_{e,g}} \right) \right) \tag{3.36}
\]

\[
1 = L_{e,g} + L_{e,2p} \tag{3.37}
\]

The heat transfer area $A_e$ is an input parameter. For the evaporator it is important to notice that inner and outer heat transfer area differ strongly. Thus the $U_e$ value has to correspond with the chosen heat transfer area. If $A_e$ describes the inner heat transfer area, Eq. (3.36)
applies, with $r_A$ as ratio of outer to inner heat transfer area multiplied with the fin efficiency. $r_A$ is assumed to be constant with a value of 20. Heat transfer coefficients are calculated with the correlations given in Table 3.1 for fin and tube heat exchangers.

\[
\frac{1}{U_{e,i}} = \frac{1}{\alpha_{e,i}} + \frac{1}{r_A\alpha_a}
\]  

(3.36)

In the two phase zone $\alpha_{e,2p}$ is calculated as average of the heat transfer coefficients in several discrete elements. Thus the influence of varying vapor quality along the refrigerant flow path can be reflected. Based on pressure and outlet enthalpy the evaporator model can switch to a flooded evaporator representation with $L_{e,g} = 0$, $\Delta T_{SH} = 0$ and $x_{e,out} \leq 1$.

**Expansion device**

The expansion process itself is modeled to be isenthalpic:

\[
h_{x,in} = h_{x,out}
\]  

(3.37)

In refrigeration cycles with dry expansion evaporators the expansion device controls the degree of superheat by regulating the refrigerant mass flow rate. Larger opening degrees of the expansion device reduce the superheat. Evaporation temperature and therefore pressure is increasing. For a unit with fixed volume flow rate the mass flow rate then increases proportionally with the density at the compressor suction port. Thus refrigerant flow rate, evaporation pressure and superheat are coupled. In the cycle model the degree of superheat therefore can be used as an input parameter to the evaporator model.

In practice, superheat is required to protect the compressor from liquid droplets being sucked into the compression chamber and causing material failure. From an efficiency point of view, superheat should be as small as possible. First, the smaller the superheat, the bigger becomes the evaporator area which is available for evaporation. Second, the superheat that has to be created limits the possible increase of evaporating temperature. As can be seen from Fig. 3.5.b, the minimum possible temperature difference $T_h - T_e$ is $\Delta T_{SH}$. This results in the effect, that after a certain point a further increase of the evaporator area (for constant $U$ and $\dot{Q}_e$) does not result in an increase in evaporation temperature and thus has no positive effect on system performance.

However, the degree of superheat also affects the control stability of refrigeration cycles with dry expansion evaporators. This effect is described by the theory of minimum stable superheat (MSS). Lower superheat values than the minimum stable value lead to fluctuations of
the refrigerant mass flow rate. Higher values reduce system efficiency unnecessarily. Winter et al. (2012) claims that independent of evaporator design the MSS curve can be approximated with a simple linear correlation:

\[ \Delta T_{SH} = 0.65 (T_a - T_e) \]  

(3.38)

Practical experience shows this correlation to be a good approximation in many systems (Blatz, 2013). This correlation is used in the study of evaporator maldistribution in Chapter 4. In the other studies \( \Delta T_{SH} \) is kept constant for all operating conditions.

### 3.3 Annual performance calculation

The standard EN14825 (2010) describes a method for calculating the seasonal coefficient of performance SCOP of heat pumps based on measurements of COP and \( \dot{Q} \) of the heat pump unit at a small number of operating conditions. Space heating only is considered. The method for calculating annual performance described in the following is developed based on this standard.

#### Operating conditions

Three climate zones are defined: Warmer, average and colder with the reference cities Athens, Strasbourg and Helsinki. A climate zone is characterized by the number of annual operating hours \( H_j \) at each ambient temperature \( T_{a,j} \) in one Kelvin steps (Fig. 3.7). Also three water temperature levels are specified: low, medium and high. These levels are supposed to reflect floor heating, radiator heating in modern buildings and radiator heating in old buildings. The combination of three climate zones and three water temperature levels lead to nine possible system environment settings. In this study the low water temperature level is not considered, the high water temperature level is used in the framework of sensitivity studies only.

The heat pump performance, namely COP and delivered capacity \( \dot{Q} \), has to be evaluated at seven different ambient temperatures to allow calculating SCOP for all climate zones and one water temperature level. The conditions at these seven points, namely the ambient temperature \( T_a \) and the water outlet temperature \( T_{w,out} \), are used as input parameters for the simulation model. Values for these parameters are given in Table 3.2. The condenser water flow rate is constant for all operating conditions, assuming a fixed speed water pump in the hydronic circuit.
Figure 3.7: Profiles for water outlet temperature and annual operating hours, defined by EN14825 (2010).

The water flow rate is chosen such that the water temperature increase between condenser inlet and outlet is 5 K for an ambient temperature of 7°C and nominal compressor speed. At other ambient temperatures and compressor speed the temperature increase of the water is determined by the capacity delivered by the heat pump. In practice temperature curves in radiator systems may considerably differ from standard values of Table 3.2. This aspect is taken into consideration in Chapter 7.

Table 3.2: Operating conditions for annual performance calculation

<table>
<thead>
<tr>
<th>#</th>
<th>$T_a$ [°C]</th>
<th>medium $T_{w,out}$ [°C]</th>
<th>high $T_{w,out}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12</td>
<td>29</td>
<td>33</td>
</tr>
<tr>
<td>2</td>
<td>7</td>
<td>33</td>
<td>38</td>
</tr>
<tr>
<td>3</td>
<td>2</td>
<td>36</td>
<td>43</td>
</tr>
<tr>
<td>4</td>
<td>-7</td>
<td>43</td>
<td>52</td>
</tr>
<tr>
<td>5</td>
<td>-10</td>
<td>45.3</td>
<td>55</td>
</tr>
<tr>
<td>6</td>
<td>-15</td>
<td>49.2</td>
<td>60</td>
</tr>
<tr>
<td>7</td>
<td>-22</td>
<td>54.7</td>
<td>67</td>
</tr>
</tbody>
</table>
Building demand and heat pump capacity

EN14825 (2010) defines the lowest considered ambient temperatures to be 2°C, -10°C and -22°C for the warmer, average and colder climate zone, respectively. The highest ambient temperature at which the building requires heat from the heating system is 15°C in all climate zones. In this study it is assumed that the heat pump operates in each climate zone over the whole ambient temperature range. In practice also heat pumps exist that shut down operation at higher ambient temperatures than the minimum temperature of the colder climate zone.

The capacity required by the building, the building demand $\dot{Q}_d$, is assumed to be linearly decreasing with increasing ambient temperature (Fig. 3.8). The inclination of the capacity demand curve is determined by the end point (0 kW at 16°C) and the design temperature and capacity. Design temperature and capacity should reflect building characteristics like size and quality of insulation. The values usually chosen in this study are 10 kW at -10°C.

The capacity delivered by the heat pump to the water side depends on the capacity control scheme. Most heat pumps today run with a single or fixed compressor speed. The refrigerant volume flow rate is approximately constant for all operating conditions. Because the refrigerant density at the compressor suction port increases with increasing ambient temperature due to the rising evaporation pressure, the refrigerant mass flow in the cycle increases and thus the delivered capacity. At the same time building demand decreases with increasing ambient temperatures due to the reduced heat losses of the building to the ambient. As Fig. 3.8.a shows, the inclination of the building demand curve and the capacity delivered by the pump are therefore opposed. The crossing point of the two curves, where delivered capacity exactly matches building demand is denominated balance point. The ambient temperature belonging to this point is called balance point temperature $T_{a,b}$. Below $T_{a,b}$ the capacity of the heat pump is too small. Electric heat is added to cover building demand. Above $T_{a,b}$ the delivered capacity is too high. To reduce capacity the heat pump is therefore shut off periodically in the so-called on/off mode.

The balance point depends on the compressor size. Bigger sizes allow higher refrigerant flow and therefore capacity. The balance point moves to lower ambient temperatures. The temperature range requiring added electric heat is reduced, while the range of on/off mode increases. Smaller sizes result in a shift of the balance point to higher ambient temperatures.

An alternative control mode is the variable speed control. With a frequency converter the rotational speed of the compressor and there-
fore the refrigerant volume flow rate can be adjusted. By controlling the compressor speed, delivered capacity and building demand can be matched. Ideally this matching is possible at all ambient temperatures. However, often an upper and lower speed limit exists. An example for a capacity curve with an upper limit of 200% and a lower limit of 50% of the nominal speed is shown in Fig. 3.8.b. The upper speed limit occurs at the ambient temperature $T_{a, bu}$, the lower limit at $T_{a, bl}$. At ambient temperatures below $T_{a, bu}$ again electric heat has to be added to cover building demand. Above $T_{a, bl}$ on/off control is required. In between $T_{a, bu}$ and $T_{a, bl}$ the demand curve can be matched by adjusting compressor speed.

**Annual evaluation**

The values of COP simulated with the heat pump unit model has to be corrected if heat pump capacity is higher than the building demand using Eq. (3.39). Losses in the on/off control mode are taken into account in the power consumption calculation with the cyclic degradation coefficient $c$. These losses include power consumption of the unit during off mode and during start-up of the unit, if e.g. pressures in the unit have equalized during off mode. The baseline value for $c$ is 0.9 which is the standard value in EN14825 (2010). Today’s air-water heat pumps typically have higher values close to 1, therefore $c$ is varied in sensitivity studies.
\[
\text{COP}_r = \text{COP} \left( \frac{\dot{Q}_d}{\dot{Q}} \right) \quad \text{for } \dot{Q} > \dot{Q}_d
\]

\[
\text{COP}_r = \text{COP} \left( \frac{c \dot{Q}_d}{\dot{Q} + 1 - c} \right) \quad \text{for } \dot{Q} \leq \dot{Q}_d
\]

(3.39)

Values for \(\dot{Q}_j\) and \(\text{COP}_{r,j}\) at each ambient temperature \(T_{a,j}\) are linearly interpolated between the simulated values. SCOP is calculated with Eqs. (3.40) and (3.41). The auxiliary electric heat \(\dot{Q}_{el}\) is included, which has to be added on the left hand side of the balance point to cover the heat demand.

\[
\text{SCOP} = \frac{\sum_j H_j \dot{Q}_{d,j}}{\sum_j H_j \left( \frac{\dot{Q}_{d,j} - \dot{Q}_{el,j}}{\text{COP}_{r,j}} + \dot{Q}_{el,j} \right)}
\]

(3.40)

\[
\dot{Q}_{el,j} = 0 \quad \text{for } \dot{Q} > \dot{Q}_d
\]

\[
\dot{Q}_{el,j} = \dot{Q}_{d,j} - \dot{Q}_j \quad \text{for } \dot{Q} \leq \dot{Q}_d
\]

(3.41)

Similarly, the power consumption \(P_j\) which includes compressor and fan power demand and supplementary electric energy is determined accordingly with Eq.(3.42) to calculate annual operating costs:

\[
P_j = \frac{\dot{Q}_{d,j}}{\text{COP}_{r,j}} \quad \text{for } \dot{Q} > \dot{Q}_d
\]

\[
P_j = \frac{\dot{Q}_j}{\text{COP}_{r,j}} + \dot{Q}_{el,j} \quad \text{for } \dot{Q} \leq \dot{Q}_d
\]

(3.42)

Annual work \(W\) of the heat pump unit is calculated for each climate zone individually by summing power consumption \(P_j\) at each temperature step \(j\) weighted by the operating hours \(H_j\):

\[
W = \sum_j H_j \frac{P_j}{1000}
\]

(3.43)
3.4 Economic modeling

Operating and investment costs

Annual operating costs $O$ are calculated with Eq. (3.44):

$$O_y = p_{el,y} W \quad (3.44)$$

Electricity prices $p_{el}$ are taken from Eurostat (2013) data for “medium size household consumers” (annual consumption in the range of 2500-5000 kWh) and include all taxes and levies. As example the electricity prices for the first half-year of 2013 for Denmark, Sweden and France (0.3, 0.21, 0.15 € kWh$^{-1}$) are used. These prices are common consumer prices. Prices for customers with higher annual consumption are usually slightly lower but would also be in the range of 0.12 - 0.3 € kWh$^{-1}$ for the three countries.

Costs for the main components, namely the evaporator, condenser and compressor, were obtained for different component sizes. For the heat exchangers the reference size is the heat transfer area on the refrigerant side $A$. For the compressor the reference size is the physical displacement volume at nominal rotational speed $V_p$. The fin-and-tube evaporator data is based on information from different suppliers. The brazed plate condenser data is based on the Danfoss H62L-CX geometry with plates with a special dimpled design to minimize pressure drop. The compressor data are from the Danfoss HHP series, a compressor developed especially for heat pumps. Additional component costs for different technologies considered in this study are also based on Danfoss internal information.

The costs for the individual components were obtained in this study in the form of OEM prices, valid for purchases by another company (purchase quantity of 1000 pieces, year 2013). End customer investment costs $I$ considerably deviate from OEM prices. A current estimate by Saar and Sullivan (2013) gives for the considered application and components a factor of five for the expected price increase of a component between manufacturer and end customer if taxes are included. This factor is about twice as high as assumed 30 years ago by Marquand et al. (1984) but close to the 20/80 rule considered to be a general approximation for the purchase price increase (Madani, 2014). Thus in all optimization examples in this study that take the viewpoint of the end customer a constant factor of five is used to adjust OEM prices.

Total end customer investment costs $I$ for an air water heat pump unit of the investigated type and size, including required accessories but without installation costs, are in the range of 13.000 - 16.000 € (Heizungs-
The relative component costs used in this study, including the price increase factor for end customers, are presented graphically in Fig. 3.9. Minimum and maximum component sizes used in the graph are determined by the available cost data. Under the given assumptions and in the considered size range compressor investment costs account for about 10 - 17% of the total investment costs. The condenser accounts for about 3.5 - 9%, the evaporator for about 2 - 7%. Optimal component sizes as determined in the different utilization examples lie within these ranges.

**Payback time and total cost of ownership**

One parameter to evaluate the economic impact of changes in heat pump technology for the end customer is the simple payback time $\tau$, calculated with Eq. (3.45). $\tau$ compares an alternative technology to a baseline heat pump unit by considering the differences in investment $\Delta I$ and operating cost $\Delta O$.

$$\tau = \frac{\Delta I}{\Delta O} \quad (3.45)$$

Alternatively, the total cost of ownership (TCO) of a heat pump system is calculated following the net present value method described by Bejan et al. (1996). In the classical net present value method expenses are taken into account with a negative sign while returns have a positive sign. A heat pump owner normally gets no monetary return from the investment in a heating system. Therefore, in the context of TCO the
positive sign is used for expenses. In this study only expenses for the investment $I$ and operating costs $O$ are considered. TCO for a heat pump unit is thus calculated with Eq. (3.46) for an annual effective discount rate $i$ and an economic lifetime $a$.

$$\text{TCO} = I + \sum_{y=1}^{a} O_y \frac{1}{(1 + i)^y}$$  \hspace{1cm} (3.46)

Today’s heat pumps are expected to run for 20 years or more (Energy Saving Trust, 2014) with yearly maintenance costs of 50 - 100 € (Heizungsfinder, 2014). However, no studies regarding changes of reliability and expected lifetime for alternative heat pump technologies were available. Hence maintenance costs are neglected. This is an appropriate assumption because the focus of interest is on TCO changes with changing technologies, but not on absolute TCO. The economic lifetime $a$, which is defined as the expected average system lifetime in the field, is varied between 10 and 20 years.

In the calculation of total cost of ownership the changing value of money with time is accounted for with the effective discount rate $i$. The effective discount rate describes the amount of interest earned or paid at the end of an annual period and includes corrections of the nominal interest rate due to inflation and varying compounding periods. Eq. (3.46) implies that the money required to cover all expenses related to the heat pump during the expected lifetime has to be provided at the time of investment. This is visualized in Fig. 3.10. Fig. 3.10.a shows the present value of a future value, here as example a future annual cost of 1000 €. If the changing value of money with time is not considered...
(i=0%), present and future value is equal. The total cost of ownership (Fig. 3.10.b, with an investment of 10000 €) accordingly increases linearly with time as constant annual costs are summed up. For effective discount rates \(i > 0\) future annual costs have a smaller present value. This present value is the amount of money which has to be deposited at time of investment with the given effective discount rate to yield enough money to cover the future costs. Both with increasing discount rates and costs shifted into the future less money has to be provided at time of investment. With increasing discount rates the TCO after 10 years, calculated with the value of money at time of investment, becomes smaller. As shown in Fig. 3.10.b, varying annual costs, e.g. due to increases in the electricity price, have a similar effect on TCO as the discount rate. In the following only the effective discount rate is varied in sensitivity studies while the electricity price is assumed constant with time.

### 3.5 Metamodelling

A metamodel, also called a response surface, is a simple model of a model. Metamodels can be derived by curve-fitting techniques. In this study metamodels are derived from the annual performance calculations. In the following description the annual work \(W\) is used as annual performance parameter, the procedure for other parameters like SCOP is equivalent.

The goal of the metamodelling is to express \(W\) as continuous function \(W_q\) of \(k\) (quasi-)continuous optimization variables \(z_1, z_2, \ldots, z_k\) which form the vector \(\vec{Z}\). NIST/Sematech (2012) stated that quadratic models are sufficient in most practical cases, thus in this study a second-order response surface is chosen. For \(k\) optimization variables this quadratic model has following general form (Kleppmann, 2003):

\[
W_q(\vec{Z}) = \beta_0 + \sum_{n=1}^{k} \beta_n z_n + \sum_{i=1}^{k} \sum_{j=i}^{k} \beta_{ij} z_i z_j
\]  

(3.47)

For a vector \(\vec{Z}\) with two optimization variables Eq. (3.47) contains six coefficients \(\beta\). For three variables the quadratic function is described with ten coefficients \(\beta\). These coefficients have to be fitted to simulation data. In this study a least squares fit is used.

Several simulation runs with varying values of the optimization variables \(\vec{Z}\) are required to provide a data set for curve fitting. To minimize this number of simulations, statistical methods from the design of experiment theory (DoE) are applied. This theory provides matrices which describe how to efficiently vary the values of the optimization variables.
in the different simulation runs. In the context of DoE the optimization variables are called factors. Varying values of the selected continuous optimization variables correspond to varying factor levels. The matrices describing how to vary factor levels are called experimental designs.

As pointed out by NIST/Sematech (2012), the central composite design is the most common experimental design for deriving second-order quadratic models. This design is thus employed in this study and depicted in Fig. 3.11.a for two factors. Each of the nine points I-IX correspond to specific values of the optimization variables for nine simulation runs. In the dimensionless normalized representation of Fig. 3.11.a the nine points form a characteristic pattern. This pattern consists of a center point IX with normalized factor levels \( N = 0 \) for both factors. Around this center four design points I-IV are grouped in the shape of a square at the normalized levels \( N = 1 \) and \(-1\). In the four star points V-VIII one factor is kept at the factor level \( N = 0 \) while the other factor is changed to \( N = \sigma \) or \(-\sigma\). Varying the simulation input parameters in correspondence with this pattern allows minimizing the number of simulation runs required to derive a model which captures a quadratic system response adequately. In two dimensions this minimum is nine simulation runs, for three factors or optimization variables 15 simulations are required. With increasing dimension the number of minimally required simulations increases further.
The upper and lower normalized factor levels $\sigma$ and $-\sigma$ correspond to upper and lower limits for the optimization variables $z_i$. The quadratic model is only valid within these boundaries. Therefore the minimum and maximum values for all optimization variables have to be chosen in advance such that the optimum can be expected to occur inside the factor region. Eq. (3.48) allows to calculate the values of the optimization variables at the normalized factor levels $N = -1, 0, 1$ with the chosen minimum and maximum sizes $z_{i,-\sigma}$ and $z_{i,\sigma}$. Based on statistical theory, a value of 1.414 is chosen for $\sigma$ (NIST/Sematech, 2012) for two factors and a value of 1.682 for three factors.

$$z_i = z_{i,-\sigma} + \frac{N + \sigma}{2\sigma} (z_{i,\sigma} - z_{i,-\sigma}) \quad -\sigma \leq N \leq \sigma$$  (3.48)

In this study the heat exchanger areas of the evaporator $A_e$ and condenser $A_c$ are (among others) chosen as continuous optimization variables. However, it is known a priori that a saturation effect occurs for the annual work input $W$ with increasing $A_e$ and $A_c$. Such an effect cannot be adequately reflected with a quadratic function. To nonetheless describe this system behavior with a second-order response surface, a variable transformation is performed (Kleppmann, 2003). In Eqs. (3.47) and (3.48) the optimization variables $z_i$ corresponding to the size of evaporator and condenser are replaced by the inverse $1/z_i$. The resulting general correlation between values of an optimization variable $z_1$ (which represent $A_e$ or $A_c$ or any other optimization variable producing a system response with a saturation effect) and the corresponding factor levels of the experimental design is presented in Fig. 3.11.b.

An example for the values of different optimization variables on the five factor levels is given in Table 7.1 for a three-dimensional design. An example for a two-dimensional quadratic model of $W$ as function of the two optimization variables $A_e$ and $A_c$ is presented in Eq. (3.49). The coefficients $\beta$ have to be fitted to the simulations for each technology and for each variation of assumptions in the annual performance model.

$$W_q(\vec{Z}) = \beta_0 + \beta_1 \frac{1}{A_e} + \beta_2 \frac{1}{A_c} + \beta_3 \left( \frac{1}{A_e} \right)^2 + \beta_4 \frac{1}{A_eA_c} + \beta_5 \left( \frac{1}{A_c} \right)^2$$  (3.49)

Equivalent metamodels can also be derived for any other output parameter of the simulation model. Of special interest are parameters which have to be limited in practice to prevent system failure, e.g. too high compressor discharge temperature or too low suction pressure. In the formulation of the optimization problem the metamodels for such parameters can be converted to constraint functions.

Within this study both at the design points and at the optimized values deviations between annual work calculated with Eq. (3.43) and with
the quadratic model (Eq. (3.49)) are typically below 0.5% for a two dimensional quadratic model. For three dimensions, the error increases to about 1.5%. Kleppmann (2003) claims that the quadratic regression approach is mostly used for up to five and only very seldom for more than six factors.

An alternative to this metamodel approach would be to create a parameter map for \( W \) to be used by the optimizer. Also with this approach the optimization procedure would be fast and stable and no additional simulations would be required for variations in the economic framework. With the parameter mapping approach errors are introduced due to linearization between the mapped values. For a simple estimation of the numerical effort of the metamodel and the mapping approach it is assumed that simulations should be performed for at least six different values of each variable to keep the linearization error in the same range as the error introduced by quadratic modeling. For three dimensions then 6x6x6 map points are required. As described above, for annual performance calculation each map point requires eight simulations with varying operating conditions. Assuming that one simulation needs a computation time of one second, the creation of the performance map would need 29 minutes. The 15 points required for metamodeling need two minutes. The metamodel approach reduces computation time by about 93%. With the software implementation as described in Chapter 3.7 the computation time for a complex cycle model is up to 35 seconds per simulation. Computation times are then 70 minutes for the metamodel and 16.8 hours for the parameter map.

### 3.6 Continuous optimization

As discussed in Chapter 2, the search space of all input or optimization variables is split in a continuous and a discrete part. The optimization problem is formulated for the continuous part alone. Here a problem formulation with a single objective function and two variables is presented as an example. As objective function the total cost of ownership TCO is chosen, the vector of optimization variables \( \tilde{Z} \) contains the heat transfer areas of the evaporator \( A_e \) and condenser \( A_c \). Based on \( W_q \) the annual operating costs can be directly expressed as a continuous function \( O_q = O(\tilde{Z}) \) (Eq. (3.44)). Since also the investment costs are given as continuous functions of the component sizes, applying Eq. (3.46) leads to a continuous function for the total cost of ownership TCO = TCO(\( \tilde{Z} \)). The upper and lower limits for the optimization variables \( \tilde{Z}_\sigma \) and \( \tilde{Z}_{-\sigma} \), introduced for metamodel regression, form linear inequality constraints. Another non-linear inequality constraint is formulated for the compressor discharge temperature \( T_{p,\text{out}} \) which is supposed to be be-
low $T_{p,out,max}$. Thus following continuous constrained single objective minimization problem can be formulated:

$$\begin{align*}
\min \quad & \text{TCO}(\tilde{Z}) \\
\text{subject to} \quad & \tilde{Z}_- \sigma \leq \tilde{Z} \leq \tilde{Z}_\sigma \\
& T_{p,out}(\tilde{Z}) \leq T_{p,out,max}
\end{align*}$$

(3.50)

The objective is to minimize TCO by finding optimal variable values $\tilde{Z}$, feasible solutions fulfill all constraints. Due to the relatively simple form of the the function TCO$(\tilde{Z})$ which is a combination of linear and quadratic functions, solving Eq. (3.50) directly converges to the global minimum.

![Figure 3.12: General representation of the Pareto front of optimal solutions for a bi-objective optimization problem.](image)

Also multi-objective problem formulations can be solved with continuous optimization algorithms. As shown in Fig. 3.12 for the objectives SCOP and investment cost, for each multi-objective optimization problem exist not one but multiple solutions. A solution is characterized by the fact that one objective cannot be improved without deteriorating another. All solutions form the Pareto front or trade-off curve. This curve can be traced by solving several single objective optimization problems:

$$\begin{align*}
\min \quad & I(\tilde{Z}) \\
\text{subject to} \quad & \tilde{Z}_- \sigma \leq \tilde{Z} \leq \tilde{Z}_\sigma \\
& T_{p,out}(\tilde{Z}) \leq T_{p,out,max}
\end{align*}$$

(3.51)

$$\begin{align*}
\max \quad & \text{SCOP}(\tilde{Z}) \\
\text{subject to} \quad & \tilde{Z}_- \sigma \leq \tilde{Z} \leq \tilde{Z}_\sigma \\
& T_{p,out}(\tilde{Z}) \leq T_{p,out,max}
\end{align*}$$

(3.52)
\[
\begin{align*}
\text{max} & \quad \text{SCOP}(\tilde{Z}) \\
\text{subject to} & \quad \tilde{Z}_{-\sigma} \leq \tilde{Z} \leq \tilde{Z}_{\sigma} \\
& \quad T_{p,\text{out}}(\tilde{Z}) \leq T_{p,\text{out,\max}} \\
& \quad I(\tilde{Z}) = I_{\text{const}}
\end{align*}
\] (3.53)

Eq. (3.51) searches within the space of feasible solutions for the combination giving minimal investment cost. Because this correlates with the solution giving minimal SCOP, the solution marks the lower left end of the Pareto front, shown in Fig. 3.12. Accordingly, the solution of Eq. (3.52) marks the upper right end. In Eq. (3.53) an equality constraint is added to maximizing SCOP for a constant value \(I_{\text{const}}\) of the investment cost which lies between the minimum and maximum value. By repeatedly solving Eq. (3.53) for different \(I_{\text{const}}\), the shape of the Pareto front can be traced.

Figure 3.13: Schematic sketch of the relation between single objective and bi-objective optimization.

A special relation exists between the single and bi-objective optimization problems formulated in this study. This relation is sketched schematically in Fig. 3.13. The function for TCO includes both the function for the investment \(I\) and for the annual work \(W\). Annual work is directly related to SCOP as shown in Eq. (3.40) - (3.43). Therefore, SCOP and \(I\) of an optimal solution that minimizes TCO according to Eq. (3.50) will lie on the Pareto front of optimal solutions determined with Eqs. (3.51) - (3.53). The exact position of this optimum on the Pareto front is governed by the economic parameters \(p_{el}, a\) and \(i\). Lower values for \(p_{el}\) and \(a\) as well as higher values for \(i\) will shift the optimum which minimizes TCO towards smaller values of SCOP and \(I\), reflecting a preference to reduce investment cost. Hence the shape of the Pareto curve of the bi-objective problem can also be traced by varying the economic parameters of the single-objective optimization problem. From
a practical point of view it is easier to find a solution to Eq. (3.50) because it is less constrained and thus converges easier.

### 3.7 Implementation

The component based model of the heat pump unit is implemented in the dynamic modeling language Modelica. Thermophysical properties of the refrigerant are obtained from the Refeqns package by Skovrup (2009). The Dymola (2010) software is used to create an executable file of each model, which is used to perform batch simulations, a repeated solving of the equation set with varying input parameters. Matlab (2013) is used for manipulating the input parameter text file, starting the executable simulation file, checking for convergence and errors, and saving the simulation results. Data processing for annual performance calculation, metamodel regression, cost calculation and solving of the continuous optimization problem is also done in the Matlab (2013) environment. Curve fitting functions from the Matlab (2013) Statistics toolbox are used to derive the quadratic regression model, for solving the optimization problem an interior-point algorithm from the Matlab (2013) Optimization toolbox is implemented.
4 Method utilization example: Countering maldistribution in evaporators

This chapter is a summary of Paper I and Paper II appended at the end of the thesis.

4.1 Introduction

Maldistribution in evaporators leads to different degrees of superheat at the outlet of parallel flow channels. This is known to reduce the overall heat transfer coefficient, evaporation temperature and heat pump performance. Different causes for non-uniform superheat exist. Uneven flow or temperature distribution on the air side can be induced by heat exchanger, fan or casing design, or by dirt accumulation. On the refrigerant side uneven allotment of gas and liquid phase can occur in the distributor and differences in length or diameter of parallel flow paths lead to uneven flow restrictions. The impact of these effects varies with heat exchanger size, geometry and operating conditions and may be outbalanced or aggravated by a combination of different effects.

Maldistribution in round tube-and-fin evaporators with a distributor at the inlet is mostly investigated by artificially inducing non-uniform flows, either experimentally or by simulation. Lee and Domanski (1997) simulated effects of various air and refrigerant distribution patterns on heat pump performance for cross-counter flow evaporators with different flow circuitry. Kim et al. (2009a) systematically investigated effects of feeder tube geometry, air flow and void fraction distribution utilizing a lumped parameter evaporator model. An R410A air conditioning unit was studied in a similar manner by Kærn et al. (2011b). The same types of non-uniformity were considered employing a finite element evaporator model. In the experimental investigation of Choi et al. (2003), non-uniform air flow was induced by partly blocking the flow path. Refrigerant maldistribution was induced by controlling superheat at parallel flow paths individually, evaporation pressure was kept constant. Comparisons of the severity of effects of different maldistribution types as well as losses in capacity and COP reported in these studies largely depend on the severity of maldistribution initially chosen by the authors.

A quantitative study of gas/liquid phase non-uniformity in a distributor was performed by Yoshioka et al. (2008). A maximum deviation of
30% between distributor inlet quality and quality at a single distributor outlet was observed. Performing an experimental study on an air-air heat pump with clean evaporator coil, Bach et al. (2012a) reported an increase of COP and capacity of 1% each if maldistribution was eliminated. Results of a similar study on a cooling unit (Bach et al., 2012b) showed a recovery of 4 and 6% of COP and capacity.

The simplest approach to counteract maldistribution is to increase evaporator size (Kærn et al., 2011a). The refrigerant circuitry of the evaporator influences maldistribution effects and has been subject to numerical and experimental analysis (Liang et al., 2001) and optimization (Domanski and Yashar, 2007). Nakayama et al. (2000) experimentally tackled non-uniform distribution of refrigerant quality by improving distributor design. Quality distribution was also addressed by a cycle setup with flash gas bypass, proposed in an experimental study by Tuo and Hrnjak (2012). Smart refrigerant distribution has been discussed by Payne and Domanski (2002) in a numerical and experimental study. This idea has been put into practice with a so-called hybrid method by Kim et al. (2009b) and an alternative method for individual superheat control by Kærn et al. (2011a).

The objective of Paper I is to quantify the operating cost increase caused by flow maldistribution for an air-water heat pump. For this purpose effects of non-uniform air flow and refrigerant phase distribution on evaporator, heat pump and annual system performance are analyzed at varying operating conditions encountered during a year. The use of a simulation model ensures comparable conditions and allows a better understanding of occurring effects. A lumped parameter approach (Mader et al., 2011) is proposed for modeling the evaporator and verified by comparison to results from the literature. Annual operating costs are calculated for three climate zones. The possibility to compensate for maldistribution-induced performance losses by increasing evaporator size is discussed.

The goal of Paper II is an economic analysis of flash gas bypass cycle and individual superheat control applied in an air-water heat pump system. Performance of all cycle layouts is assessed on an annual level. The ability of the two technologies to counteract maldistribution and their cost-effectiveness is investigated and compared in a total cost of ownership analysis.

4.2 Methods

Distribution parameters

The influence of maldistribution on operating costs is expected to vary with the geometry and size of the heat exchanger. In order to keep
the simulation model generic, robust and quick to solve, the evaporator geometry is simplified to two main parallel channels aligned in one row. They are connected at the inlet by a distributor. Fig. 4.1 shows that each of the two channels is subdivided into six parallel passes to represent a realistic flow division of a fin-and-tube evaporator of the considered capacity. No maldistribution effects occur between the sub-channels.

\[ f_x = -\frac{x_{2,\text{in}}}{x_{\text{in}}} + 1 \quad 0 \leq f_x \leq 1 \]  

(4.1)

The phase distribution parameter represents malfunctioning of the distributor which can occur in case of improper design or installation. If \( f_x \) is zero, distribution of phases is equal, hence inlet quality of both passes is equal to quality at distributor inlet. If \( f_x \) is unity, only saturated liquid is fed to the second pass. The inlet quality of the first pass is determined by mass and energy conservation equations for the distributor.

\[ f_a = 2 \frac{V_{a,1}}{V_a} - 1 \quad -1 \leq f_a \leq 1 \]  

(4.2)

The air flow distribution parameter represents unequal distribution of air volume flow rate \( \dot{V}_a \) over different parts of the evaporator. This may occur due to the design of the housing or because of differences in coil

Figure 4.1: Sketch of the basic heat pump cycle and evaporator model.
face area and area swept by the fan. If \( f_a \) is zero, the air flow rate over both passes is equal. For \( f_a = 1 \) all air is led to the first channel, for \( f_a = -1 \) all air flows to the second channel. Both distribution parameters are defined such that the value zero represents the best possible case (even distribution) and the value one a worst case maldistribution.

To quantify costs caused by maldistribution, upper limits of 0.3 and 0.2 are chosen for the phase and air flow distribution parameter, respectively. Based on practical experience and the values reported by Bach et al. (2012a) and Yoshioka et al. (2008), it is assumed that no stronger maldistribution will occur in a reasonably designed and maintained heat pump unit. Worse effects may occur during long-term operation, e.g. due to uneven blockage with dirt, which is however not considered here. Three maldistribution cases, defined in Table 4.1, are discussed in more detail. The case of equal distribution of refrigerant phase and air flow, \( f_x = f_a = 0 \), denominated \( F_0 \), is used as baseline.

### Table 4.1: Maldistribution cases

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<tr>
<td>( f_x )</td>
<td>0</td>
<td>0.15</td>
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<tr>
<td>( f_a )</td>
<td>0</td>
<td>0.1</td>
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**Selected technologies to counteract maldistribution**

The basic heat pump cycle (BC) of Fig. 4.2.a consists of a compressor, condenser, expansion device and an evaporator. Superheat at the outlet of the evaporator is controlled by the expansion device. In this study technologies addressing maldistribution by changes in the system design and control are considered. The first is the flash gas bypass cycle (FGB) with internal heat exchanger (Fig. 4.2.b). From the outlet of the condenser the refrigerant is subcooled in an internal heat exchanger. The refrigerant is then expanded into a flash tank which separates gas and liquid. The bypass mass flow rate is controlled with a solenoid valve either based on information of a liquid level sensor or on a strategy specified by the manufacturer. Costs for a level sensor are not included in this study. In the ideal case, the refrigerant is separated into saturated liquid which is fed to the evaporator and saturated vapor which is bypassed. Again superheat is controlled at the evaporator outlet. The two streams are mixed after the evaporator outlet which reduces the superheat of the mixed stream compared to the superheat created in the evaporator. The refrigerant then flows through the internal heat exchanger where again superheating takes place before entering the compressor.
The second alternative technology is the individual superheat control (ISC) of parallel evaporator passes. The upstream control with balancing valves (Kim et al. (2009b)) and the combined expansion and distribution valve (Kærn et al. (2011a)) both adjust the mass flow rates in individual parallel passes. While in practice different components and control algorithms are required, from a thermophysical point of view these approaches are identical. In this study the investment costs of the combined expansion and distribution valve are used in the economic analysis of the individual superheat control technology. For the given application of European type air-water heat pumps eight to sixteen parallel refrigerant passes on the evaporator are common. Therefore it seems more likely that manufacturers employ a single component controlling all passes than installing such a high number of valves.

Fig. 4.2.c shows the ISC cycle where the expansion valve of the BC cycle is replaced by a device combining expansion and distribution as described in Mader and Thybo (2010) and Chapter 8. The distributing valve allows controlling the superheat in different passes individually by adjusting the refrigerant mass flow rate through each channel.

The study considers a heat pump with single speed control and constant compressor displacement volume (9.92 m$^3$ h$^{-1}$). Annual operating costs are calculated with constant heat exchanger area sizes $\overline{A_{IX}}$ ($A_e=2m^2, A_c=0.875 m^2$). The total cost of ownership analysis is performed with individually optimized heat exchanger sizes. Flow rates of the secondary fluids are kept constant.
4.3 Summary of results

Costs of maldistribution

Figure 4.3: Selected parameters vs. $f_x$ (for $f_a = 0$) and vs. $f_a$ (for $f_x = 0$).
The effect of maldistribution on evaporator $U$ value is shown in Fig. 4.3.a for different ambient temperatures. Phase maldistribution influences the total evaporator $U$ value mainly due to changes of the total dry out area. An increase of the dry out zone length in the pass that receives more gaseous refrigerant can only be partly compensated by the reduction of the dry out zone length in the second pass. Therefore, with increasing $f_x$ the total dry out area increases and thus the $U$ value decreases. For high ambient temperature and therefore high refrigerant flow rate and $U$ value the internal compensation of dry out zone lengths works better than at low ambient temperatures: The total dry out area is reduced less, as a result also the total evaporator $U$ value is reduced less at 12°C than at -7°C or -15°C. For air side maldistribution the $U$ value is additionally affected by changing heat transfer coefficients on the air side due to the flow rate variations. This effect is stronger with increasing ambient temperatures.

The resulting approach temperature difference between air inlet temperature and evaporation temperature is shown for the different maldistribution cases in Fig. 4.3.b. The air flow rate not only affects the evaporator $U$ value but according to Eqs. (3.31) and (3.31) also directly the energy balances of the evaporator. Thus the effect of air maldistribution on evaporation temperature is stronger than could be expected from 4.3.a. Resulting heat pump COP and heating capacity are shown in Fig. 4.3.c and d.

Fig. 4.3.e shows the percentage change of power consumption, including direct electric heat and on/off losses. The strongest increase in total power consumption is found at $T_a = -7°C$. While absolute power consumption is higher at lower $T_a$, at -15°C the percentage increase induced by maldistribution is smaller. Interestingly, for $T_a = 12°C$ power consumption is reduced with increasing maldistribution. At this ambient temperature the capacity delivered by the heat pump exceeds building demand which results in on/off operation. A decrease in heat pump capacity reduces on/off losses. In regard of total power consumption, this effect compensates the maldistribution-induced increase in compressor power consumption during the on period.

Fig. 4.4.a shows the resulting percentage increase of annual operating costs for different distribution parameter values compared to the baseline with even distribution $F_0$. Even though the range of ambient temperatures varies largely for the three climate zones, the percentage change of costs induced by maldistribution is similar. The different effects on total power consumption for different ambient temperatures seem to outbalance each other seen from an annual perspective.

The variation in absolute cost increase between climate zones shown in Fig. 4.4.b is mainly due to the difference in operating hours. In the warmer climate zone operating cost rise is below 30 € yr$^{-1}$ for all
selected maldistribution cases. Not negligible is the cost increase in
the colder climate zone, where maldistribution causes an annual cost
increase of up to 210 € yr\(^{-1}\).

Fig. 4.5 shows the potential of FGB and ISC technology to recover the
yearly operating cost increase induced by maldistribution in the aver-
geage climate zone. The cost increase is again compared to BC with even
distribution (with annual operating costs of 1140 €). Constant heat
exchanger area sizes (\(A_{IX}\)) and the electricity price of Sweden (0.21
€ kWh\(^{-1}\)) are used for this comparison. For BC operating costs in-
crease up to 50 € with non-uniform refrigerant quality distribution $f_x$. Combined with maldistribution on the air side $f_a$, annual costs for this system increase by up to 95 €.

Since saturated liquid only is fed to the evaporator in the FGB cycle, no sensitivity of the technology towards changes in $f_x$ exists. Therefore money can be saved compared to a BC system with refrigerant maldistribution. However, air side maldistribution cannot be compensated, operating costs increase up to 60 €. Hence, if air flow non-uniformity occurs, operating costs of the FGB technology are only smaller than operating costs for the BC system, if the quality distribution parameter $f_x$ is bigger than 0.15.

The ISC technology on the other hand can largely recover losses in performance induced both by air and refrigerant side maldistribution and is therefore less costly in operation than the BC cycle. Only a slight cost increase can be observed compared to the uniform distribution case. This is caused by a slightly reduced overall U value if individual flow rates in the parallel channels vary.

Component sizing

Fig. 4.6 shows COP versus heat transfer area for $F_0$ and $F_2$ at $T_a = -7^\circ C$ for the basic cycle layout BC. For small heat exchangers, a COP reduction due to maldistribution can be recovered by increasing the heat exchanger size, as depicted for an inner heat transfer area of 1 m². Reductions occurring in larger heat exchangers however cannot be counteracted by a size increase. The same graph shows the approach temperature difference $T_a - T_e$ for $F_0$. Air-water heat pumps are typically operated with approach temperature differences below 7-8 K at the considered ambient temperature.
Optimized evaporator heat transfer area sizes versus economic lifetime and corresponding approach temperature differences are presented in Fig. 4.7 for BC. The shown area sizes result in minimum TCO for the given conditions. Considering only short time periods below five years, optimal area size for the warmer climate is below the lower limit of 1.25 m$^2$ used in this optimization study. For a lifetime of 15 years the optimal heat exchanger size would be about 45% higher than this minimum. The corresponding approach temperature differences at the lower evaporator size limit are not constant because optimum condenser sizes vary. Optimal heat exchanger sizes for the average and colder climate are considerably higher. Also the expected effective discount rate during the lifetime of the heat pump considerably influences the heat exchanger size giving lowest possible TCO as shown for the average climate zone. Optimal evaporator size for 15 years of operation varies by 8% considering a discount rate of either 0% or 3%. The trends for optimal condenser sizes are similar.

Fig. 4.8 shows relative optimal evaporator and condenser areas and corresponding temperature differences for all considered cycles and maldistribution cases for an economic lifetime of ten years, the colder climate zone, 0.21 € kWh$^{-1}$ and 3% discount rate. $\Delta T_{cw}$ depicts the temperature difference $T_c - T_{w,in}$. Baseline sizes are optimal areas for BC with no maldistribution ($A_e = 3.9$ m$^2$, $A_c = 1.8$ m$^2$). Absolute values depend strongly on the chosen heat exchanger geometries and heat transfer correlations and cannot be generalized, while the trends seen in the graph are expected to be similar also for other coil geometries. Interestingly, for the BC cycle with increasing maldistribution optimum evaporator size decreases. To offset the negative effect of maldistribution on oper-
Figure 4.8: Relative optimal heat exchanger sizes and corresponding temperature differences (colder climate, 0.21 € kWh$^{-1}$, 3% effective discount rate, 10 years).

For the FGB cycle with no maldistribution optimal heat exchanger sizes are bigger compared to the BC cycle. The trend to reduce evaporator size with increasing maldistribution is even stronger. Only for the ISC technology, which is able to largely compensate maldistribution losses, it is preferable to increase evaporator size slightly with increasing maldistribution. Hence for the ISC technology optimal evaporator sizes are up to 20% higher compared to the other technologies if maldistribution occurs.

**Total cost of ownership analysis**

Differences in total cost of ownership, compared to the basic cycle with the same non-uniform distribution, are shown in Fig. 4.9 and Fig. 4.10 for the FGB and ISC technology, respectively. Considered is an economic lifetime of ten years for varying electricity prices and discount rates. For the less severe maldistribution case $F_1$, the FGB technology is more costly for the end consumer both in warmer and average climate.
Figure 4.9: FGB technology: Differences in TCO compared to BC for economic lifetime of 10 years.

Figure 4.10: ISC technology: Differences in TCO compared to BC for economic lifetime of 10 years.

for all considered business conditions. Also for the colder climate zone FGB pays off only for higher electricity prices and low discount rate. For the more severe maldistribution case $F_2$, FGB is economic for some business conditions in the average climate zone and for all considered conditions in the colder climate zone. In the latter up to 650 € can be saved during ten years of operation compared to the basic cycle with the same maldistribution.

Also the ISC technology investment costs cannot be amortized within ten years if the heat pump is running in the warmer climate zone. Deficits are in the same size of order as for the FGB technology. In the average climate zone ISC pays off for most of the considered economic conditions.
cases, except for very low electricity prices and high discount rates for less severe maldistribution. In the cold climate ISC is economic for all considered business conditions. Especially for the more severe maldistribution the benefit is considerable, varying between 700 and 2300 €. This accounts for 5 - 7% reduction in TCO compared to the BC cycle as shown in Fig. 4.11. Here the relative TCO changes refer only to costs of the components which are altered, added or exchanged for the different layouts.

4.4 Concluding remarks

Two types of maldistribution have been considered: uneven distribution of refrigerant phases at the inlet of the evaporator and uneven distribution of the air flow. The resulting absolute annual cost increase is small for warmer climate zones but can become, depending on the severity of maldistribution, considerable for the average and especially colder climate zone.

Only restricted knowledge exists about the types and extent of typically occurring maldistribution and its variation for different heat pump and evaporator designs. Based on values reported in the literature, it is assumed that the considered maldistribution cases represent somewhat realistic scenarios. Other types of maldistribution like uneven distribution of the air inlet temperature in different parts of the evaporator coil or maldistribution induced by differences in the geometry of parallel evaporator channels are neglected. Also the evaporator geometry is simplified, neglecting the often complex circuiting of real coils. Resulting
maldistribution effects are not only difficult to quantify in reality, but also will differ strongly for different evaporator and heat pump design. However, the results of this study allow identifying the critical operating conditions that should be considered when investigating maldistribution in a specific heat pump.

Cost information, both for investment and operation, varies with time, location and often for each individual case. The results presented here are therefore valid in a specific economic framework. However, the results allow stating that economic conditions have to deviate strongly from those assumed here to make technologies to counteract maldistribution economically feasible for warmer climates and economically unattractive for colder climates.

For two technologies, flash gas bypass (FGB) with internal heat exchanger and individual superheat control (ISC), the ability to counteract maldistribution and the economic feasibility are investigated. While costs induced in the basic cycle by non-uniform distribution of the refrigerant phases can be well recovered by both technologies, only the ISC technology can also largely compensate air flow maldistribution. For the current economic framework, after ten years of operation investment in FGB pays off only for the colder climate. The economic benefit is relatively small. ISC is cost-effective for operation in average and colder climate. Savings realized by counteracting both refrigerant and air side maldistribution are considerable in the colder climate zone.

It is important to notice that, purely from a thermophysical point of view, an increase in evaporator size in the BC layout is not necessarily an effective means to counteract heat pump performance reductions induced by maldistribution.

Both hot water production and frost and defrost is neglected in this study. Maldistribution will in both cases increase operating costs additionally, rendering counteracting technologies more attractive than illustrated in this study.

In this application example it is shown that both the effect of maldistribution and the potential of counteracting maldistribution vary strongly with the evaporator size. Results comparing different counteracting technologies hence strongly depend on the chosen baseline component sizes. Optimizing these sizes is a feasible way to prevent biasing the results. It is also documented that differences occur between optimal heat exchanger sizes for the different technologies, here mainly the ISC technology stands out with a strongly increased evaporator heat transfer area. It is interesting to notice that increasing the condenser size is economically more optimal than increasing the evaporator size if maldistribution occurs. These findings confirm that in an analysis of the performance improvement potential of different technologies various component and system characteristics play a major role.
5 Method utilization example: Capacity control

This chapter is a summary of Paper III appended at the end of the thesis.

5.1 Introduction

Changes in the control scheme can reduce emissions associated with air source heat pumps operating over a wide range of conditions by more than a third (Cooper et al., 2014). Today, when building demand is lower than the capacity delivered by the unit, on/off control is the most common way of balancing capacities. The compressor and sometimes secondary loop fans and pumps are intermittently switched on and off to match building demand. Also the opposite effect of a too low heat pump capacity occurs, because in practice heat pumps are often designed to cover only a part of the annual building heat demand. Therefore at low ambient temperatures the building demand is often higher than the heat pump capacity. This gap between capacity demand and delivery is typically closed by an electric backup heater.

An alternative control scheme to match demand and capacity delivered by the heat pump is to adjust the compressor speed. At high ambient temperatures the compressor speed is reduced to deliver less capacity. Increasing the speed range at lower ambient temperatures leads to higher capacities. Thus both the need for on/off control and for direct electric heat can be reduced or even eliminated by varying the compressor speed.

Marquand et al. (1984) presented an economic comparison between variable speed (VS) and fixed speed on/off (FS) control for air-water heat pumps. They reported payback times of about five years for the increased investment costs of a VS unit. Karlsson and Fahlén (2007) listed several researchers who found efficiency improvements of 10 - 25% comparing variable compressor speed to fixed speed on/off control for air source units. For air-air systems Adhikari et al. (2012) reported energy savings of about 13% with VS control. However, savings for air-water units were shown to be closer to the savings gained in groundwater systems of about 9%. For buildings with small heating loads Liu and Hong (2010) show a VS air source unit to be nearly as efficient as a FS ground source heat pump while for larger loads the FS ground source heat pump is still more efficient. In an experimental study of
a ground-water system by Karlsson and Fahlén (2007) both COP and seasonal performance for the variable speed unit was found to be lower than for the fixed speed unit. Also for a ground source system Madani et al. (2011) demonstrated that seasonal performance improvement by VS control strongly depends on the nominal capacity of the baseline fixed speed unit. The performance of the FS unit with electrical backup heat covering less than 5% of the annual demand was the same as the performance of the VS unit.

Jakobsen et al. (2000) emphasized the importance of adjusting not only the compressor speed but also the flow rates of secondary fluids to exploit the full benefit of variable speed control. Granryd (2010) discussed the same aspect by showing the strong dependency of efficiency and capacity on secondary flow rates. Madani et al. (2010) showed that flow rate optimization of the brine side of ground source heat pumps can increase heat pump capacity at low ambient temperatures.

The objective of Paper III is to apply the screening method to quantify the total cost of ownership of variable speed (VS) and fixed speed (FS) control for different climate zones. Madani (2012) pointed out that comparing the two control methods with an inconsistent component selection is a possible reason for the different estimates of the potential of variable speed control found in the literature. To prevent biasing the comparison, in this study the air flow rate, compressor displacement volume and heat exchanger sizes are optimized both for the FS and VS before comparing total cost of ownership of the two control schemes.

5.2 Methods

The metamodel for the continuous optimization is derived for the heat exchanger areas of the evaporator $A_e$ and condenser $A_c$. In this study the compressor displacement volume is not a parameter of the metamodel but individual efficiency characteristics of the different compressors are taken into account. The compressor selection is therefore treated as an integer optimization variable. Seven hermetic scroll compressors of the Danfoss HHP-T4 R407C series, which are optimized for heat pump operation, are modeled; simulations are repeated for each one. The HHP-T4 R407C series is built for fixed speed operation. For variable speed control an upper and lower speed limit is assumed at 200% and 50% of the nominal speed of 50 Hz, respectively.

The air flow rate is also not included in the continuous optimization but individually optimized for on/off and variable speed control. The simulations required for quadratic model regression are repeated for different air flow rates between 0.3 - 1.4 m$^3$/s. In the fixed speed model
the air flow rate is kept constant for all operating conditions. The flow rate minimizing the annual work is selected. For the variable speed system the air flow rate is optimized to give minimum power input for each operating condition individually. This represents variable air flow control.

For the economic modeling the electricity prices for the first half-year of 2013 for Denmark, Sweden and France (0.3, 0.21, 0.15 EUR per kWh) are chosen. Investment costs $I$ of compressor, evaporator and condenser with varying sizes were based on OEM prices as described in Chapter 3.4. At the time of the study no current prices for speed control of compressor and fan were available to the author. Hence it is assumed that the price for the inverter and control units required for fan and compressor speed control is still similar to the price of two averagely sized heat exchangers as reported by Marquand et al. (1984). It is also assumed that the investment cost of the fan is independent of the air flow rate. Investment costs for all other components like expansion device or filter dryer are neglected since they do not differ for the considered options and therefore are not relevant in the comparison.

5.3 Summary of results

Annual operating costs

To evaluate the distribution of annual operating costs between different ambient temperatures, the heat pump power consumption at each ambient temperature step is multiplied with the electricity price (0.21 EUR per kWh) and with the operating hours $H_j$ for this temperature step according to Fig. 3.7 for the three climate zones. Differences between fixed and variable speed in the resulting cost distribution are presented in Fig. 5.1 for identical component sizes in both FS and VS system. Values below zero mean operating a VS system is more costly at the given condition. Variable speed control reduces operating costs of the heat pump below the balance point temperature $T_{a,b}$ of the FS system. However, above $T_{a,b}$ the operating costs for the VS system become higher than for on/off control. Only for ambient temperatures higher than 6°C VS is again less costly to operate. Below $T_{a,b}$ the reduction of direct electric heat with VS control leads to the observed savings. Above $T_{a,b}$ differences between heat exchanger $U$ values and compressor isentropic efficiencies dominate the performance comparison between FS and VS.

Fig. 5.2 shows the summed up yearly operating cost differences between fixed and variable speed system. For the colder climate zone a sizable economic benefit of VS control can be observed. However, the
benefit gets small for the average and warmer climate zone, because the fixed speed heat pump often operates above $T_{a,b}$ with the given component selection. Especially for the average climate zone reduced $U$ values at medium ambient temperatures and increased losses in motor and inverter at reduced compressor speed jeopardize variable speed performance. Simulations are repeated with constant values for overall heat transfer coefficients $U$ in both heat exchangers and constant total compressor efficiency for all operating conditions. If negative effects of $U$ and $\eta$ at medium ambient temperatures are thus eliminated, savings achieved by VS control increase considerably, especially in the average climate zone.
Component sizing

Figure 5.3: Influence of balance point temperature at 2900 RPM on annual operating costs ($0.21 \text{ € per kWh}$).

In Fig. 5.3 operating costs for different balance point temperatures $T_{a,b}$ are presented for base case heat exchanger sizes and air flow rate and nominal compressor speed of 2900 RPM. Increasing balance point temperatures are related to decreasing compressor sizes. Since each considered compressor has individual efficiency characteristics, curves are not smooth. For FS operation the optimum balance point temperature increases when switching from colder to average and warmer climate zone. By shifting balance point temperatures to higher ambient temperatures the amount of required backup electrical heat is increased. At the same time however approach temperature differences in the heat exchangers are reduced due to smaller refrigerant flow rates. The latter is more relevant for average and warmer climate zones.

Table 5.1 presents the optimal component selection which minimizes TCO of both fixed and variable speed unit. Resulting nominal balance point temperatures and approach temperatures in the heat exchangers are also presented. With lower minimum operating temperatures of the climate zones, optimal compressor sizes increase. Optimal compressor sizes for VS control are smaller than for FS control. Only in the warmer climate the smallest available compressor is optimal for both control modes. Optimal evaporator sizes also are smallest for warmer and largest for colder climate zone, with smaller optima for the VS system. Interestingly, for the VS system the ratio of optimal evaporator to condenser size is smaller than for FS, investment in bigger condensers seems more economical.
Table 5.1: Optimization results for 0.21 € kWh$^{-1}$, effective discount rate 3%, economic lifetime of 10 years and warmer (w), average (a) and colder (c) climate zone.

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<th>$V_p$</th>
<th>$A_e$</th>
<th>$A_c$</th>
<th>$T_{a,b}$ (at 2900 RPM)</th>
<th>$\Delta T_{ae}$ (at $T_{a,b}$)</th>
<th>$\Delta T_{cw}$ (at $T_{a,b}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[m$^3$/h]</td>
<td>[m$^2$]</td>
<td>[m$^2$]</td>
<td>[°C]</td>
<td>[K]</td>
<td>[K]</td>
</tr>
<tr>
<td>w</td>
<td>FS</td>
<td>5.9</td>
<td>1.1</td>
<td>0.9</td>
<td>2.1</td>
<td>5.8</td>
</tr>
<tr>
<td></td>
<td>VS</td>
<td>5.9</td>
<td>1.0</td>
<td>1.1</td>
<td>2.2</td>
<td>6.2</td>
</tr>
<tr>
<td>a</td>
<td>FS</td>
<td>9.9</td>
<td>1.9</td>
<td>1.2</td>
<td>-3.25</td>
<td>5.7</td>
</tr>
<tr>
<td></td>
<td>VS</td>
<td>8.0</td>
<td>1.6</td>
<td>1.6</td>
<td>-0.75</td>
<td>5.7</td>
</tr>
<tr>
<td>c</td>
<td>FS</td>
<td>14.3</td>
<td>2.7</td>
<td>1.1</td>
<td>-8.1</td>
<td>6.0</td>
</tr>
<tr>
<td></td>
<td>VS</td>
<td>9.9</td>
<td>2.3</td>
<td>1.7</td>
<td>-3.25</td>
<td>5.6</td>
</tr>
</tbody>
</table>

Total cost of ownership analysis

Figure 5.4: Averaged annual cost of ownership vs. economic lifetime (0.21 EUR per kWh, effective discount rate 3%).

Fig. 5.4 shows the development of minimal total cost of ownership divided by the years of operation for different economic lifetime. FS and VS control are compared with an electricity price of 0.21 EUR per kWh and an effective discount rate of 3%. The values presented in the graph include only investment costs of the considered components and are not
put in relation to total heat pump investment cost. With increasing lifetime the initial investment costs are of decreasing significance while operating costs are increasingly dominating TCO. Thus in the short run the FS control is the cheaper solution while with increasing lifetime VS control becomes economically more attractive. The crossing points of FS and VS curves represent the break even points. The according discounted payback times for changing from a VS to a FS system are 2.4 years for the colder climate, 6.5 years for the average climate and, not shown in the graph, 38.5 years for the warmer climate zone.

The sensitivity of the TCO difference between FS and VS to different economic scenarios is presented in Fig. 5.5 for economic lifetimes of 10 and 15 years. After 10 years, in the warmer climate zone VS control is more costly than FS control for all considered scenarios. In the worst considered case financial loss is about 800 EUR. In the average climate zone VS control is profitable for scenarios with higher electricity prices and low discount rate. Up to 1000 EUR can be saved by choosing a variable speed system. The VS system is cost-efficient for the colder climate zone, independent of the considered economic scenario. Differences in TCO of up to 3300 EUR can be observed.

After 15 years VS control is still uneconomic in the warmer climate, in the average climate it pays off for all considered scenarios. In the colder climate savings realized by operating a variable speed system increase up to about 5000 EUR.

Figure 5.5: Difference in TCO of FS and VS: Sensitivity towards electricity prices and discount rate.
5.4 Concluding remarks

Variable speed control lowers operating costs mainly by reducing electricity consumption at low ambient temperature. At medium ambient temperatures the economic comparison between fixed and variable speed is very sensitive both towards system immanent parameters (e.g. variations in heat transfer coefficients) and towards external parameters (e.g. electricity prices). From an economic viewpoint variable speed control in air-water heat pumps is therefore most promising for a heat pump operating in colder climate zones. Variable speed units optimized for colder climates are also slightly more cost efficient when run in the average climate than fixed speed units optimized for colder climates. However, in the warmer climate zone the most cost efficient option is a fixed speed unit optimized for the conditions encountered there. Several aspects may lead to an under- or overestimation of the TCO difference between fixed speed and variable speed control in this study:

- The choice and accuracy of correlations for heat transfer and compressor efficiency has an effect on the comparison of fixed and variable speed control. Simulations performed with fixed $U$ values and compressor efficiencies show higher benefits for the VS control. The general trend for differences in TCO is however not reversed: In the warm climate zone VS control is too costly, in the average climate zone the benefit depends on the economic framework and variable speed control pays off in the colder climate.

- No up to date values for prices of speed control of compressor and fan were available. 50% higher costs than assumed in this study increase payback time to 4 years for colder and 11 years for average climate (0.21 EUR per kWh, discount rate 3%).

- Time periods of 10 and 15 years are used in the analysis. Longer time periods favor the variable speed system by shifting the emphasis from investment costs to operating costs. However for longer time periods maintenance costs should be taken into account, but to the knowledge of the author no long term reliability studies comparing fixed and variable speed control are available.

- EN14825 (2010) defines for each climate zone three water temperature levels: 35°C (floor heating), 45°C (radiator heating) as considered in this study and 55°C (refurbishment projects). Simulations show that while operating costs are increased with higher water temperatures, operating cost differences between fixed and variable speed control vary only little with water outlet temperature, Fig. 5.5 will hence not differ much.
- In additional simulations the influence of the cycle degradation coefficient during on/off control is tested. An increase from 0.9 to 0.95 decreases the TCO difference between optimized systems by about 20%.

- In this study for the fixed speed unit a fixed speed fan is considered: However, as discussed by Granryd (2010), at temperatures below the balance point temperature increasing fan speed could increase heat pump capacity and therefore reduce the need for direct electric heat. Combined with a variable speed fan the FS unit performance could thus be improved. The increase in investment cost would be small compared to the costs of variable compressor speed control.

- Increasing the frequency range of the variable speed compressor to reduce or eliminate the regions of electric backup heat and on/off control would improve VS performance.

- Design temperature and design building capacity determine the slope of the building demand curve to represent the building characteristics as described in Chapter 3.3. The smaller the inclination, the smaller becomes the deviation between the capacity curve of a fixed speed heat pump and the building demand curve: less electric heating and less on/off control is required. VS systems are therefore less beneficial for small and well insulated buildings.

- Building demand not only depends on ambient temperature as assumed in EN14825 (2010), but also on other factors like user behavior, solar radiation and building dynamics. As shown by Madani et al. (2011), the building demand curve changes to a cloud with some demand points below the simplified linear demand curve and some points above. It can be assumed that effects of points above and below the linear curve will outbalance each other in an economic comparison of fixed and variable speed control.

- The effect of tap water heating on annual performance is neglected in this study. This effect depends on the ratio of seasonal capacity used for space and tap water heating. It also differs for different system configurations on the building water side like the hot water tank geometry. It is expected that the benefit of VS control is higher if tap water heating is taken into account. Further research is required to quantify this effect with respect to FS and VS control.

Optimal component sizes differ for fixed and variable speed control. Obviously a comparison of the two control methods with equally sized components could easily lead to wrong conclusions. Particularly for the
colder climate zone the optimal compressor displacement volume for the FS unit is considerably bigger than for VS control. The optimized VS unit still requires electric backup heating since otherwise losses at high ambient temperatures induced by reduced $U$ values and increased motor and inverter losses would become too large. The ratio of optimal evaporator to condenser area is lower for variable speed control.
Method utilization example:
Cycle layout and refrigerant selection

This chapter is a summary of Paper IV and Paper V appended at the end of the thesis.

6.1 Introduction

A multitude of different cycle layouts have been developed with the aim to improve performance of the vapor compression system. Many of those are interesting in the context of heat pump applications as shown in the overview given by Minh et al. (2006).

The layout with internal heat exchanger was discussed in detail in the theoretical study of Klein et al. (2000). Performance improvement was shown to depend on the chosen refrigerant, operating conditions and internal heat exchanger effectiveness. At -20°C evaporation and 40°C condensation temperature and maximum effectiveness of one, cooling capacity was improved over the baseline cycle by 21% for R507A and by 12% for R290. Tambovtsev (2007) proposed an alternative control method for a cycle with internal heat exchanger. Superheat was controlled at the suction port of the compressor. In this control mode a two phase mixture occurred at evaporator outlet. To keep the system stable an electronic expansion valve was required, a co-flow arrangement of the internal heat exchanger and a size limit that allowed maximally 20% of liquid to be evaporated in the internal heat exchanger. In the experimental setup this layout led to an increase of the evaporation temperature by up to 2 K.

Domanski (1995) showed in a theoretical comparison of the internal heat exchanger layout with a two stage economizer cycle higher system performance improvements with the more complex cycle layout. The economizer layout was further investigated experimentally for a residential air-water heat pump by Ma and Chai (2004) in a setup with an economizer heat exchanger and a single compressor with injection port. At -25°C evaporation and 45°C condensation temperature capacity was reported to increase by 20% and COP by 14% over the basic cycle. Using a similar layout in an air-air heat pump, Wang et al. (2009) experimentally showed improvements of capacity and COP of up to 30% and 20% respectively for an outdoor and indoor temperature of -18°C.
and 21°C. Bertsch and Groll (2008) experimented with a cycle layout with economizer heat exchanger and two independent staged compressors in a residential air-water heat pump application. At -15°C ambient and 50°C water temperature COP was reported to improve by 18%, heating capacity by a factor of 2.3.

In the discussed literature cycle layout changes are predominantly investigated under the aspect of improving COP and capacity at challenging operating conditions at low temperature. *Paper IV* focuses therefore on the economic assessment of the economizer layout and two internal heat exchanger layouts in air-water heat pumps. *Paper V* includes the comparison of two refrigerants using a bi-objective optimization problem formulation.

### 6.2 Total cost of ownership optimization

Methods

Three alternative cycle layouts, sketched in Fig. 6.1, are compared to the basic cycle (BC) for the refrigerant R290. The internal heat exchanger (IX) and flooded evaporator (FE) layouts require only a small adaptation of the basic cycle while the economizer (EC) layout is more complex regarding structure, component design and control. Heat pumps with IX and EC layouts are commercially available, in contrast to the FE layout.

Some special assumptions are used in the modeling of the different cycle layouts: For the EC cycle the two compression stages are modeled as individual compressors. Mixing losses at the injection port are neglected. For the economic modeling additionally a version with a single stage compressor with injection port (EC, injection) is taken into account. The bypass flow is shut off if the power consumption with single stage compression is lower. The high to low stage compressor displacement
volume ratio in the EC layout is optimized to give minimum TCO. The internal heat exchanger in the IX layout is sized such that maximum discharge temperature at compressor outlet is 140°C. This limit is given by the compressor manufacturer and is reached in the given application at the lowest ambient temperature. According to Tambovtsev (2007) also in the FE layout the heat exchanger size is limited due to a minimum evaporator outlet quality of 0.8 which is required for control stability. Again this limit is reached at lowest ambient temperature. For both layouts the size of the internal heat exchanger is chosen to be suitable for the cold climate. The resulting size is also used for calculations for the warmer and average climate zone.

Fig. 6.2 shows relative changes of the end consumer investment costs of the three cycles. The cost of the BC layout is assumed to be 13,000 €. The impact of component size variation on total investment cost is represented by the purple vertical line in Fig. 6.2. The upper end of this line represents investment costs with all components sizes at the upper end of the size range. The lower end of the purple line represents costs with minimum component sizes.

Additionally to the heat transfer area sizes of evaporator and condenser in this study the displacement volume of the compressor is included in the continuous optimization. The metamodel thus has three dimensions. To this end it is assumed that the coefficients of the polynomials describing the compressor characteristics do not change with changes in the compressor displacement volume.

Five model scenarios listed in Table 6.1 are investigated in the economic analysis. Two compressor models (characterization with ten-coefficient
Table 6.1: Scenarios for total cost of ownership optimization

<table>
<thead>
<tr>
<th>Scenario</th>
<th>$\eta_{is}$</th>
<th>Capacity control</th>
<th>$T_{w,\text{out}}\text{-level} / \dot{Q}_{d,\text{ds}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$S_0$</td>
<td>$f(p_{p,\text{in}},p_{p,\text{out}})$</td>
<td>FS</td>
<td>medium /10 kW</td>
</tr>
<tr>
<td>$S_3$</td>
<td>$f(p_{p,\text{in}},p_{p,\text{out}})$</td>
<td>VS</td>
<td>medium /10 kW</td>
</tr>
<tr>
<td>$S_{10,11}$</td>
<td>$f(p_{p,\text{in}},p_{p,\text{out}})$</td>
<td>FS</td>
<td>high / 11 kW</td>
</tr>
<tr>
<td>$S_1$</td>
<td>0.73</td>
<td>FS</td>
<td>medium /10 kW</td>
</tr>
<tr>
<td>$S_{1,3}$</td>
<td>0.73</td>
<td>VS</td>
<td>medium /10 kW</td>
</tr>
</tbody>
</table>

polynomials versus constant isentropic efficiency), two capacity control methods (fixed speed FS versus variable speed VS) and different system environment parameters (design building demand and water temperature level) are tested. These parameters showed the strongest impact on performance differences between the cycle layouts in a detailed sensitivity analysis.

Summary of results

Table 6.2 lists optimum sizes of compressor, evaporator, condenser and internal heat exchanger and the associated balance point and approach temperatures. Optimum condenser heat transfer area is similar for all cycle layouts; optimum evaporator heat transfer area is about 8% larger for the FE layout and 8% smaller for the EC layout compared to the basic cycle layout.

Table 6.2: Optimization results for $S_0$, 0.21 € kWh$^{-1}$, effective discount rate 3%, 10 years of operation, colder climate

<table>
<thead>
<tr>
<th></th>
<th>BC</th>
<th>IX</th>
<th>FE</th>
<th>EC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{V}_p$ [m$^3$h$^{-1}$]</td>
<td>14.0</td>
<td>14.1</td>
<td>13.7</td>
<td>9.2 (11.0)</td>
</tr>
<tr>
<td>$A_e$ [m$^2$]</td>
<td>3.6</td>
<td>3.6</td>
<td>3.9</td>
<td>3.3</td>
</tr>
<tr>
<td>$A_c$ [m$^2$]</td>
<td>2.1</td>
<td>2.1</td>
<td>2.0</td>
<td>2.1</td>
</tr>
<tr>
<td>$A_{hx}$ [m$^2$]</td>
<td>-</td>
<td>1.0</td>
<td>1.1</td>
<td>1.0</td>
</tr>
<tr>
<td>$T_{a,b}$ (at 2900 RPM) [°C]</td>
<td>-7.2</td>
<td>-7.4</td>
<td>-7.2</td>
<td>-11.1</td>
</tr>
<tr>
<td>$\Delta T_{ae}$ (at $T_{a,b}$) [K]</td>
<td>6.2</td>
<td>6.2</td>
<td>5.3</td>
<td>7.0</td>
</tr>
<tr>
<td>$\Delta T_{cw}$ (at $T_{a,b}$) [K]</td>
<td>10.8</td>
<td>10.3</td>
<td>10.9</td>
<td>13.1</td>
</tr>
</tbody>
</table>

Fig. 6.3.a and 6.3.b show capacity and relative COP of the different cycle layouts. Absolute values of COP for the BC layout vary between 2 and 7.1 at lowest and highest ambient temperature respectively. For high ambient temperatures the differences between BC, IX and FE layout are small. At lowest ambient temperature for the IX cycle an im-
Improvement of about 5% over BC can be observed both for $\dot{Q}$ and COP. This result is similar to the data presented by Klein et al. (2000), who showed variations of cooling capacity improvement for a R290 cycle between 12 and 2.5% for different heat exchanger effectiveness of 1 and 0.2. The FE cycle improves $\dot{Q}$ by about 11% and COP by 8% at the lowest ambient temperature. A large performance improvement at low ambient temperatures can be observed for the EC cycle with about 84% higher capacity and 50% higher COP. At high ambient temperatures though, both capacity and COP of the EC cycle are below the BC cycle. The main reason is the low isentropic efficiency of the compression process. Pressure differences between suction and discharge port is small for both compressors, thus the compressors operate far from their design conditions.

![Diagram](image)

**Figure 6.3:** Capacity and relative COP of the four cycle layouts for different ambient temperatures. ($S_0$, optimized)

Annual savings achieved with the alternative layouts are shown in Fig. 6.4. For the colder climate zone savings amount to 2, 3 and 22% of the BC operating costs for the IX, FE and EC layouts, respectively. Percentage values are smaller for the average and warmer climate.

Total cost of ownership for the heat pump unit running in the colder climate is presented in Fig. 6.5 for an economic lifetime of 10 years, an electricity price of 0.21 € kWh$^{-1}$ and 3% interest. For the EC layout both costs of the version with two staged compressors and of the version with vapor injection compressor are used. While improvements of capacity and COP of EC over BC calculated with model scenario $S_0$ are slightly more positive than reported in various publications, results for $S_1$ are less positive. Thus $S_1$ is assumed to underestimate the improvement potential of an EC version with two independent compressor
Figure 6.4: Annual operating cost differences ($S_0$, 0.21 € kWh$^{-1}$, BC costs in three climate zones 372, 1082 and 2316 € yr$^{-1}$).

stages. Therefore for scenarios $S_1$ and $S_{1,3}$ only the costs for the vapor injection version are taken into account.

In the first three scenarios the order of economical attractiveness of the different cycle layouts is constant, independent of capacity control mode and system environment. TCO decreases between BC, IX, FE and EC layout. In $S_0$ with the EC layout the total cost of ownership can be reduced by 5.5%. If the reference layout is a BC unit with variable speed capacity control, switching to an EC layout with VS control decreases TCO by only 2.5%. Higher building demand and water temperature level in the hydronic cycle increases the economic attractiveness of the EC layout with savings of about 8%.

Figure 6.5: Total cost of ownership for different model scenarios (10 years, 0.21 € kWh$^{-1}$, 3% interest).
The picture changes if compressor efficiency is assumed to be constant. The savings potential of the EC layout is strongly reduced. In the IX layout the discharge temperature is reduced due to higher isentropic efficiency at low ambient temperature. Therefore the internal heat exchanger area can be increased, the savings potential increases. Under this assumption IX and EC layout are similarly attractive from an economic viewpoint; the IX layout is even preferable in VS control mode. However, maximum achieved savings in TCO after 10 years are still below 3%.

For the average climate the effect of cycle layout change on TCO is small, under most conditions a switch to the EC layout increases TCO in 10 years. For the warmer climate any cycle layout change increases TCO compared to the basic cycle, independently of the model scenario.

The economic attractiveness of variable speed capacity control is less affected by the assumption about compressor efficiency. Fig. 6.5 shows that savings of about 5 - 6% can be achieved, both when comparing $S_3$ to $S_0$ and $S_{1,3}$ to $S_1$. Whether it is more attractive to replace a standard BC unit by a variable speed BC unit or by changing to the EC layout depends on the compressor efficiency of the two stage concept. A combination of EC layout and variable speed control reduces TCO further in both cases, the additional savings are however relatively small. Varying economic parameters do not change the general trends.

### 6.3 Pareto optimization

#### Methods

*Paper V* presents a bi-objective optimization with the objectives to maximize SCOP and minimize investment cost. Following modeling assumptions differ from those of the study presented in *Paper IV*:

- Subcooling in the condenser is taken into account. The influence on the seasonal coefficient of performance (SCOP) of the BC cycle is investigated in a sensitivity study. For the optimization a fixed, constant value is used for all operating conditions.

- The model of a reciprocating compressor, presented by Granryd et al. (2009), is used. The isentropic efficiency is a function of the pressures at compressor inlet and outlet as well as inlet temperature.

- Only variable capacity control is considered. However no upper or lower limit on the speed is taken into account. Thus the building
demand is matched over the whole range of ambient temperatures. As changes of the efficiency with varying frequency are neglected, the size of the compressor displacement volume is irrelevant.

- Both a fixed and variable speed fan are modeled. For the fixed speed option the air volume flow rate is calculated which maximizes SCOP. For the variable speed fan the air volume flow rate is adjusted to minimize the combined power input of fan and compressor at each operating condition.

- The economizer heat exchanger of the EC is replaced by a separator or flash tank, resulting in the open economizer cycle OC. In this flash tank ideal separation of the liquid and gas phase takes place. The saturated liquid is fed to the evaporator, the saturated gas into the bypass. In different publications both types of economizer cycles showed similar performance improvements, however Wang et al. (2009) judged the version with separator to be less flexible and less easy to control. Accordingly in this study it is assumed that the bypass is never closed. For the intermediate pressure the geometric mean of evaporation and condensing pressure is used.

- The heat transfer area size of the internal heat exchanger in the IX cycle is an additional input parameter for the continuous optimization problem. A constraint equation takes into account the discharge temperature limit.

- The cost criterion is the sum of the OEM prices of the different components. Only prices of components that are adjusted or exchanged are taken into consideration. Costs are presented relative to the cheapest feasible solution. Following the argumentation of Hwang et al. (2004) for C290 a price surcharge of 20% is assumed for all components except the heat exchangers compared to non-flammable refrigerants.

**Summary of results**

Fig. 6.6 shows the changes in SCOP for the colder climate zone with variations of different input parameters. The strongest influence is observed for the air volume flow rate. For this parameter an optimal fixed volume flow rate $\dot{V}_{a,\text{opt, fix. speed}}$ exists. The additional improvement on SCOP gained by optimizing the air volume flow rate individually for each operating condition ($\dot{V}_{a,\text{opt, var. speed}}$) is relatively small. Assuming that the subcooling $\Delta T_{SC}$ could be controlled and optimized at each operating condition, the improvement of SCOP is in the same size of order of individual fan speed control. If both parameters are optimized
at each operating condition, an annual performance improvement of about 5% can be gained in this example.

![Diagram](image)

Figure 6.6: Changes in SCOP with percentage variations and optimization of different input parameters.

The Pareto fronts of different technology combinations for colder climate and medium water temperature level are presented in Fig. 6.7. The conditions are closest to scenario S<sub>3</sub> discussed in Paper IV. The increase of SCOP between BC and IX for medium heat exchanger sizes is about 3% and thus comparable to the results presented above. However the improvement achieved with the OC cycle layout is smaller than for the EC layout. The main reason is the difference in cycle control at high ambient temperatures: Under these conditions in the EC layout the bypass is shut off and the cycle works as the basic cycle. Performance of the OC cycle however is considerably worse than of the basic cycle. The reason is the small pressure differences at compressor inlet and outlet ports in high ambient temperature conditions and therefore low isentropic efficiencies of the staged compressors.

The Pareto fronts of the IX cycle working with R410A are not presented because for all heat exchanger sizes in the considered range the discharge temperature constraint is violated. For the same reason the Pareto front of the R410A BC cycle is not continued to lower SCOP and cost: With smaller heat exchangers the discharge temperature increases above the given limit. The improvement potential of the OC layout does not differ strongly for R410A and R290. The different potentials of the IX cycle can be observed for the average climate zone, not presented here. There it becomes obvious that the R410A cycle benefits less from the internal heat exchanger than the R290 cycle.
Figure 6.7: Pareto curves for different cycle layouts, refrigerants, fan speed control (colder climate).

6.4 Concluding remarks

The influence of following aspects on the results of this study should be noted:

- Especially the improvement potential of the internal heat exchanger cycle layouts is severely limited by practical operation requirements. Here it is postulated that these limits have to be fulfilled in all occurring operating conditions. The lowest ambient temperature condition showed to be the limiting factor which restricts the internal heat exchanger sizes for both layouts. At moderate ambient conditions these internal heat exchanger sizes are small in relation to the refrigerant mass flow rate and thus inefficient. Since these moderate conditions have a much stronger impact on annual performance, the overall economic potential is small. In a real fixed compressor speed unit the effectiveness of the internal heat exchangers might decrease less with increasing ambient temperature than calculated in this study, because increasing mass flow rates would also lead to increasing U values. This effect is neglected here. Internal heat exchangers are not equally beneficial for all refrigerants or operating conditions, the given results for this layout are valid for R290 only.
• For the FE layout it is important to notice that this study assumes a superheat setting of 6 K for all operating conditions, which is similar to the settings used in the experimental and theoretical study by Tambovtsev (2007). The sensitivity study shows that only slightly higher required superheat settings lead to a higher improvement potential of the FE layout. However, control stability of a real unit with FE layout might also depend on superheat settings. Further research is required for this cycle layout.

• The evaluation of the annual performance of two-stage cycles in comparison to single stage cycles is very sensitive to compressor and control characteristics. Even for the colder climate zone annual performance is not improved, or even negatively affected, if the bypass flow is not shut off at high ambient temperatures.

• Hot water production is not considered in this study. In the IX and FE layout the size of the internal heat exchangers would need to be reduced further due to the higher required water temperatures. The potential to reduce operating costs in space heating mode is thus reduced further. The economic attractiveness of the EC cycle however increases considerably.

In general for all layouts and all model assumptions the TCO savings in the colder climate are quite small compared to the strong improvement of COP and capacity observed at low ambient temperatures. The explanation is given by the heating hour profiles which show that the unit is operating a large part of the time at modest ambient temperatures where the improvement potential of alternative cycle layouts is small if practical limits are considered.
7 System and environment modeling: Comparison of two approaches

This chapter is a summary of Paper VI appended at the end of the thesis.

7.1 Introduction

In most publications dealing with the optimization of heat pump components, annual operating costs are calculated based on electricity costs at a single operating point multiplied by the annual operating hours. Such an approach was used e.g. by Sanaye and Niroomand (2011) for a first-law thermo-economic optimization of a ground coupled steam ejector heat pump and by Sayyaadi et al. (2009) in a classical second-law analysis of a ground source heat pump.

The influence of varying climatic conditions and building demand is more commonly considered for control strategy optimization. E.g. Wang and Jin (2000) and Fong et al. (2006) optimized control setpoints of air conditioning systems to minimize annual energy consumption with a dynamic building model and genetic optimization algorithms. Likewise, Wright et al. (2002) employed a transient building model with the two objectives of minimizing energy cost and thermal discomfort. In this study the optimization variables were formed both from the control system and the size of HVAC components. The optimization was performed for three typical days. Again, a genetic algorithm was used for solving the optimization problem. In these studies a massive effort for design computations and the optimization procedure was required.

The current study aims at optimizing the sizes of the main components (compressor, evaporator and condenser) of an on/off controlled ground-water heat pump unit in a residential building to minimize total cost of ownership. Unlike most of the previous studies considering component optimization, the heat pump performance should be evaluated for varying weather and demand conditions occurring throughout a year. The main goal of the study is to minimize the required simulation effort. For this purpose a performance map is derived from the heat pump unit model and a second-order metamodel from the dynamic building model. The possibility of further reducing the simulation effort is investigated by comparing the method of dynamic system and environment modeling with a simplified profile method which weighs different operating conditions according to EN14825 (2010).
7.2 Methods for system and environment modeling

Dynamic method

Figure 7.1: Conceptual representation of the thermophysical model with (a) heat pump unit model and (b) dynamic system and environment model with submodels and relationships.

The conceptual representation of the dynamic system and environment model with all submodels and their relationships is shown in Fig. 7.1. The heating system contains a thermal storage tank with direct connection of heat sink and source in a configuration often used in systems providing space heating and domestic hot water simultaneously. The data for climatic conditions of Stockholm comprise the ambient temperature, effective sky temperature, ambient relative humidity and solar irradiation. For the building a Swedish single family house with an area of 300 m\(^2\) in one floor and windows covering one third of the northern and southern walls is modeled. The heat load is calculated including heat losses through external walls, windows, floor and roof, infiltration losses, the ventilation load, internal gains and solar gains. Indoor set point temperature is constant at 21\(^\circ\)C. Annual and peak heating demand of the considered building are 146.6 kWh m\(^{-2}\) yr\(^{-1}\) and 15.6 kW.
respectively. In the heat distribution system hot water from the storage tank is supplied with a fixed speed water pump to radiators. At design outdoor temperature of -18°C the nominal supply and return temperatures are 55°C and 45°C respectively. In the supply line between radiators and storage tank a mixing valve is included which is connected with a tempering valve in the return line of the radiators. With this bypass return water can be mixed into the supply line to reduce the radiator supply temperature if necessary. The auxiliary heater is situated between thermal storage tank and mixing valve. The thermal storage tank is a vertical stratified tank with three temperature levels. Water from the lowest layer is supplied to the heat pump, hot water returning from the heat pump is fed to the upper layer. Water supplied to the radiators is taken from the middle layer and the return water enters in the lowest temperature layer of the storage tank. The heat pump unit is represented in form of performance maps for capacity delivered at the condenser and power input to the compressor at different combinations of brine and water inlet temperatures. The performance maps are created with a detailed model for a ground source heat pump unit with brazed plate condenser and evaporator. In the dynamic simulation mapped values are linearly interpolated to calculate unit performance at all operating conditions encountered throughout the year. A fixed speed pump circulates the water between storage tank and heat pump condenser. The heat source contains a U-type ground heat exchanger installed in a 260 m deep ground-water filled borehole. Thermal conductivity of several ground layers are taken from the study of Acuña et al. (2008). A mixture of ethanol and water with 16% ethanol is pumped between the bore hole and the heat pump evaporator with constant flow rate.

A strategy of constant hysteresis to control the radiator supply temperature is employed. The strategy, described in detail by Madani et al. (2013) determines the required supply temperature as function of indoor and ambient temperature. The central control unit controls the heat pump unit, the electrical heater and a tempering valve in the radiator loop. If the indoor temperature exceeds 21°C, the heat pump unit is shut off. The bypass between radiator return and supply line is activated whenever the inlet temperature of the mixing valve exceeds the set point supply temperature.

Profile method

Generalized operating condition profiles for brine-water heat pumps of EN14825 (2010) are used for comparison with the dynamic method. The inclination of the linear building demand curve is chosen such that the annual building heat demand is the same as in the dynamic method.
The brine temperature at evaporator inlet is kept constant at the standard value of 0°C. Required water supply temperatures at condenser outlet for varying ambient temperatures are determined with the same constant hysteresis control strategy as in the dynamic method. The operating hour profile of the colder climate zone is used.

**Metamodeling**

Both methods are used for simulations at 15 design points with varying component sizes to derive quadratic models of the annual electricity input required by compressor and auxiliary heater to fully cover the building demand for space heating. The component sizes as related to the five factor levels according to the design of experiment theory described in Chapter 3.5 are given in Table 7.1.

<table>
<thead>
<tr>
<th>Factor level</th>
<th>( \dot{V}_p ) ([m^3 h^{-1}])</th>
<th>( A_e ) ([m^2])</th>
<th>( A_c ) ([m^2])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum (-\sigma)</td>
<td>4.48</td>
<td>0.56</td>
<td>1.08</td>
</tr>
<tr>
<td>(-1)</td>
<td>4.90</td>
<td>0.60</td>
<td>1.16</td>
</tr>
<tr>
<td>Center</td>
<td>6.83</td>
<td>0.95</td>
<td>1.80</td>
</tr>
<tr>
<td>(1)</td>
<td>8.76</td>
<td>2.22</td>
<td>3.99</td>
</tr>
<tr>
<td>Maximum (\sigma)</td>
<td>9.18</td>
<td>3.11</td>
<td>5.40</td>
</tr>
</tbody>
</table>

**7.3 Summary of results**

Fig. 7.2 depicts the influence of varying balance point temperature on SCOP for minimum, center and maximum heat exchanger sizes according to Table 7.1. In each curve a maximum in SCOP occurs which is marked with a circle. For smaller compressor sizes respective higher balance point temperatures the required amount of auxiliary electric heat increases and thereby SCOP is reduced. For lower balance point temperatures the negative effect of increasing approach temperature differences between refrigerant and secondary fluids in the heat exchangers on compressor power input prevail. With increasing heat exchanger sizes SCOP increases and the maximum SCOP shifts to lower balance point temperatures. This makes it obvious that component sizes should be optimized together. In the profile method the influence of the compressor size on SCOP and thus operating costs is stronger than in the dynamic method. Especially reducing the compressor size leads to a higher demand of auxiliary electric heat. The maxima in SCOP are
therefore shifted to higher compressor displacement volumes or lower balance point temperatures. The more detailed dynamic method shows a dampening of this effect.

![Figure 7.2: Comparison of component influence with dynamic and profile method for system and environment modeling.](image)

Economically optimal sizes of all three components are given in Table 7.2 both for the dynamic and profile method. The end consumer investment costs of the three components optimized with the dynamic method amount to about 3310 €. This accounts for 28% of the average capital cost for a ground source heat pump excluding drilling costs determined by Blum et al. (2011). With the profile method negative effects of smaller evaporators are slightly stronger. A possible reason is the difference in brine inlet temperatures between both approaches. Positive effects of larger condensers are weaker. Most likely the broad controller dead bands for the temperature in the storage tank lead in the dynamic method often to larger water temperature differences between condenser inlet and outlet than in the profile method. This makes a larger condenser more profitable.

In the dynamic method with economically optimized component sizes the auxiliary heater covers 0.9% of the annual building heat demand. With the profile method this coverage is 1.2%. The design capacity of the heat pumps optimized with the dynamic and the profile method are 88% respective 92% of the building peak capacity. So even though the detail level of system and environment in both modeling approaches is very different, the resulting economically optimal heat pump unit sizes for space heating are comparable.
Table 7.2: Optimization results for an electricity price of 0.21 € kWh\(^{-1}\), effective discount rate 3% and economic lifetime of 15 years.

<table>
<thead>
<tr>
<th>method</th>
<th>( \dot{V}_p ) [m(^3)h(^{-1})]</th>
<th>( A_e ) [m(^2)]</th>
<th>( A_c ) [m(^2)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic method</td>
<td>8.1</td>
<td>1.9</td>
<td>5.1</td>
</tr>
<tr>
<td>Profile method</td>
<td>8.4</td>
<td>2.1</td>
<td>4.7</td>
</tr>
<tr>
<td>Deviation [%]</td>
<td>3.7</td>
<td>10.5</td>
<td>-7.8</td>
</tr>
</tbody>
</table>

7.4 Concluding remarks

The method of metamodeling with a second-order response surface model proves to be applicable also with a considerably more complex method for system and environment modeling. Maximum deviation between annual electricity input calculated with the dynamic method and the joint metamodel at the 15 simulation points used for regression is 1.1%.

A comparison of the dynamic method with a simple profile method shows similar results for optimal component sizes and heat pump capacity. Thus, the profile method might yield as accurate results as a detailed dynamic method regarding the evaluation of optimal component sizes and SCOP.
8 Frost and defrost

This chapter includes content of Paper VI appended at the end of the thesis.

8.1 Introduction

In air-water heat pumps under some operating conditions frost can grow on the outer evaporator surface. This effect occurs if the air contains water vapor and the evaporator surface temperature is below the freezing point of water. The evaporator surface temperature increases with increasing air temperature. Therefore an upper temperature limit occurs for frost formation which depends on evaporator size and superheat control. On the other hand the amount of water vapor contained in the air decreases with decreasing air temperature which reduces frost growth for low temperatures. Measurements by Flach-Malaspina (2004) showed frost to play a significant role for air-water heat pumps between approximately -10 and 7°C. Also below this temperature range frost can form on the evaporator, especially in supersaturated conditions as described by Granryd (2005). The effects on heat pump performance however diminish noticeably.

Thick frost layers both insulate the evaporator surface and block the air flow path. To enable continuous operation, frost has to be removed regularly. Typically a heating interval with growing frost is followed by a defrosting interval. Defrosting however requires energy which penalizes the system performance.

In Chapter 8.2 the effect of growing frost on system performance is discussed based on findings in the literature. Several frost growth models from literature are presented and compared. A simple model is developed based on the analogy of heat and mass transfer and validated with experimental data for frost formation on an evaporator in an air-air heat pump. The model is then implemented in the heat pump cycle model to compare frost growth in different cycle layouts. Defrosting is discussed in Chapter 8.3. A quantification of operating costs induced by frost and defrost is given in Chapter 8.4. A new defrost method is presented in Chapter 8.5 and results both from simulations and experiments are discussed.
8.2 Frost growth

Air inlet temperature, relative humidity, air mass flow rate and evaporator surface temperatures are identified to strongly influence the frost formation. Lee and Kim (1999) used the parameters frost thickness and averaged energy transfer resistance as indicator for frost formation under varying air conditions. For increasing air temperatures between 4°C and 10°C and a constant humidity ratio, both frost thickness and energy transfer resistance decreased continuously, indicating that with higher temperatures thinner but denser frost layers were formed. With increasing relative humidity both frost thickness and energy transfer resistance increased, indicating thicker, less dense frost formation. With increasing inlet air velocity however, frost thickness increased while energy transfer resistance decreased, indicating both thicker and denser frost layers. These observations were explained with the interaction of two different frost growth mechanisms: 1) frost growth at the surface, which increases frost thickness and the insulating effect of the layer and is induced by high mass transfer and 2) vapor diffusing into the frost layer, which increases frost density and thereby reduces the insulating effect.

Tassou and Marquand (1987) investigated the effects of frost formation on heating capacity and COP for air temperatures between 5°C and -3°C at a constant relative humidity of 60%. The degradation on both heating capacity and COP is found to be increasingly larger for higher air temperatures. Also increasing relative humidity at constant air temperature was found to lead to faster degradations of capacity and efficiency. Votsis et al. (1989) as well as Yan et al. (2003) used the air pressure drop over the coil as an indication of the frost formation. Both varied the air temperature with constant relative humidity of 82% and 70% respectively. Both found a maximum in pressure drop increase, Votsis et al. (1989) at 0°C, Yan et al. (2003) at 5°C. One reason for the different observations might be that different relative humidity was adjusted and that humidity ratios varied strongly with constant relative humidity for varying temperatures. Furthermore significantly different coolant temperatures are chosen in the different studies which also can contribute to the discrepancies.

These differences notwithstanding, for a wide range of operating conditions a characteristic pattern for changes of capacity and COP with increasing frost growth can be described: During the initial frost growth period an increase in heat transfer rate and evaporator capacity might be observed, as reported among others by Lee and Kim (1999) and Guo et al. (2008). It is attributed to an increase in surface roughness and area, while frost layers are not yet thick enough to have an insulating effect or block the free flow area. This effect can however neither be seen
in the results of Kondepudi and O’Neal (1989) nor in those of Xia et al. (2006). Both investigate louvered fins, pointing out that frost blocks the louvers first, thus destroying their positive area and turbulence enhancement effects. This cannot be compensated by the frost roughness effect. When frost thickness increases further, first a gradual and then a fast drop in capacity can be observed (Tassou and Marquand, 1987; Guo et al., 2008). Both kept the fan speed constant during the experiments, thereby reflecting a conventional heat pump operating under frosting conditions. The capacity decrease occurs partly due to the insulating effect of the frost and mainly due to the blockage of the free flow area. The latter leads to a strong decrease of the air flow rate followed by a drop in evaporation temperature and pressure as well as in COP.

**Frost growth modeling**

Heat transfer under frost conditions involves complex interacting processes of heat transfer, mass transfer and diffusion. Various modeling approaches exist as depicted in Fig. 8.1. These approaches can be combined to a model for predicting the performance of finned tube heat exchangers under frosting conditions. Several such models can be found in the open literature.

Kondepudi and O’Neal (1993) presented a quasi-steady frost layer model which takes the two mechanisms of increasing frost thickness and increasing frost density into account. It uses a diffusion model to describe
heat and mass transfer in the frost layer. The pressure drop over the coil is calculated based on the reduction in the free flow area as frost is growing. Seker et al. (2004) applied the model of Kondepudi and O’Neal on a fin-and-tube geometry using various heat transfer and pressure drop correlations to include geometrical effects. An adaption to the original model was provided by Yao et al. (2004) who treated the water vapor at the frost surface as ideal gas to derive an advanced equation for the frost density increase. Yang et al. (2006) modeled frost growing on the tube and on the fin surface separately using a simplified water vapor diffusion equation. An additional dimension was added by Tso et al. (2006) who used the original model approach of Kondepudi and O’Neal and took variations of frost thickness along the fins into account. A lumped parameter model was developed by Xia and Jacobi (2010) to predict the performance of folded louvered fin heat exchangers. The mass flow of freezing water vapor attributing to the densification of the frost was accounted for by a constant absorption factor. In the frost layer only heat conduction was considered. This allowed calculating the frost surface temperature based on the exact solution to steady state heat conduction in a frosted fin between two flat tubes. Frost surface density was calculated based on that frost surface temperature. Experimentally derived curves for air side heat transfer coefficients and time dependent Reynolds number were used to compare the model with experimental data.

As the objective here is to illustrate generic system level effects of frost growth, a simple and stable quasi-steady mathematical model is required which can be integrated with the lumped parameter evaporator model described in Chapter 3.2. The new model is based on the analogy of heat and mass transfer as described in the following.

**Analogy of heat and mass transfer**

The mathematical description of heat and mass transfer is very similar, as depicted in Fig. 8.2. While the driving potential for heat transfer $d\dot{Q}$ is a temperature difference $\Delta T$, the driving potential for a molar mass flow $d\dot{J}$ is a concentration difference $\Delta \chi$. In the special case of humid air, the concentration of water can be expressed as product of air density $\rho_a$ and specific humidity $X$ to directly calculate the water mass flow rate $d\dot{m}_w$.

For a constant temperature on one side the local heat transfer can be described with Eq. (8.1) and Eq. (8.2).

$$d\dot{Q} = -\dot{m}_a c_{p,a} dT_a^*$$  \hspace{1cm} (8.1)

$$d\dot{Q} = \alpha_a dA (T_a^* - T_s)$$  \hspace{1cm} (8.2)
Equating (8.1) and (8.2) and separating of variables gives Eq. (8.3). Integration leads to Eq. (8.4).

\[
\int_{T_{a,in}}^{T_{a}} \frac{dT_a}{T_s - T_s} = -\frac{\alpha_a}{\dot{m}_a c_{p,a}} \int_0^A dA \\
\frac{T_s - T_s}{T_{a,in}} = \exp \left(-\frac{\alpha_A A}{m_a c_{p,a}} \right)
\]  

(8.4)

Inserting Eq. (8.4) in Eq. (8.2) and integrating along A finally gives the heat transfer equation:

\[
\int_0^A d\dot{Q} = k (T_{a,in} - T_s) \int_0^A \exp \left(-\frac{\alpha_A A}{m_a c_{p,a}} \right) dA \\
\dot{Q} = \dot{m}_a c_{p,a} (T_{a,in} - T_s) \left[ 1 - \exp \left(-\frac{\alpha_A A}{m_a c_{p,a}} \right) \right]
\]  

(8.5)

(8.6)

Analogous Eq. (8.7) and Eq. (8.8) can be formulated for the mass transfer. Following the same procedure as above, Eq. (8.9) can be derived.

\[
d\dot{m}_w = \dot{m}_a dX^*  
\]  

(8.7)

\[
d\dot{m}_w = -\beta \rho_a dA (X^* - X_s)  
\]  

(8.8)

\[
\dot{m}_w = \dot{m}_a (X_{in} - X_s) \left[ 1 - \exp \left(-\frac{\beta \rho_A A}{\dot{m}_a} \right) \right]
\]  

(8.9)

The mass transfer coefficient \( \beta \) follows from the assumption that the formulation of heat and mass transfer problems is similar when expressed with adequate dimensionless numbers. Nusselt number \( Nu \) and Prandtl
number $Pr$ used for heat transfer problems are replaced by Sherwood number $Sh$ and Schmidt number $Sc$ for mass transfer problems. The division of Schmidt and Prandtl number results in the Lewis number $Le$.

\begin{align*}
    Nu &= bRe^n Pr^m \\
    Sh &= bRe^n Sc^m \\
    \frac{Sh}{Nu} &= \left( \frac{Sc}{Pr} \right)^m = Le^m
\end{align*}

(8.10) \quad (8.11)

For humid air it can be assumed that the Lewis number is one. With the definition of the Nusselt and Sherwood number the mass transfer coefficient $\beta$ can be expressed as function of the heat transfer coefficient $\alpha$:

$$\beta = \frac{\alpha_a}{\rho c_p}$$

(8.12)

**Lumped parameter frost model**

Eqs. (8.6), (8.9) and (8.12) are integrated with the lumped parameter model of Chapter 3.2. For this purpose it is assumed that frost builds up only in the two phase zone. Along the length of this zone both refrigerant temperature $T_r$ and evaporator surface temperature $T_s$ are assumed to be constant. Neglecting the heat transfer resistance of the tube wall, the evaporator surface temperature $T_s$ can be calculated using the total amount of heat transferred from the outer evaporator surface to the refrigerant:

$$\dot{Q}_{2p} = \alpha_r A_r (T_s - T_r)$$

(8.13)

The total heat transfer in the two phase zone is the sum of sensible and latent heat:

$$\dot{Q}_{2p} = \dot{Q}_S + \dot{Q}_L$$

(8.14)

Eq. (8.6) describes the sensible heat transfer $\dot{Q}_S$ in the two phase zone, while (8.9) is used in Eq. (8.15) to express the latent heat transfer $\dot{Q}_L$ with the enthalpy change from water vapor to ice $\Delta h_{vi}$:

$$\dot{Q}_L = \dot{m}_w \Delta h_{vi}$$

(8.15)

For the sake of simplicity the change in the air side heat transfer coefficient due to frost growth is defined such that it incorporates the effect of increasing thermal resistance of the growing frost layer. Thus Eqs.
(8.1) - (8.12) employ the evaporator surface temperature $T_s$ according to Eq. (8.13) also during frost formation.

Due to the growing frost also the air side cross flow area decreases. Since the fan speed is typically kept constant, this reduces the mass flow rate and thus velocity of air through the evaporator which induces an additional reduction of the heat transfer coefficient of the air side. Xia et al. (2006) showed that the relation between Jason-Colburn factor and frost thickness for microchannel heat exchangers of different geometries can be described with polynomials which vary only slightly with differences in fin pitch and also with different air inlet conditions. Correlations for heat exchangers with round tubes and different fin geometry can however be expected to differ. To calculate the actual mass flow rate for a certain frosting state typically a pressure drop model is applied taking into account the area decrease and combined either with a fan curve or with a plenum method where a virtual fixed air volume is treated as ideal gas. However, as shown by Xia and Jacobi (2010), the variation of friction factors with increasing frost layer thickness is extremely sensitive towards small changes in the fin geometry. Besides, experimental data suffer from dramatic uncertainties for heavily frosted evaporators. This challenges the modeling and the accuracy of model predictions. For want of unambiguous pressure drop and heat transfer correlations for frosted heat exchangers the model instead describes the reduction of air mass flow rate and heat transfer coefficient with simple functions of the relative free cross flow area $\tilde{F}_a$ with Eqs. (8.16) and (8.17). The coefficients $\xi$ are fitted to experimental data for frost growth on a fin and tube evaporator as described below and are given in Table 8.1.

$$\frac{\dot{m}_a}{\dot{m}_{a,0}} = e^{(\xi_1(1-\tilde{F}_a))}$$  \hspace{1cm} (8.16)

$$\frac{\dot{m}_a}{\dot{m}_{a,0}} = 1 + \xi_2(1-\tilde{F}_a) + \xi_3(1-\tilde{F}_a)^2$$  \hspace{1cm} (8.17)

Table 8.1: Coefficients for decrease of air side heat transfer coefficient and air mass flow rate.

<table>
<thead>
<tr>
<th>$\xi_1$</th>
<th>$\xi_2$</th>
<th>$\xi_3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>-2.5</td>
<td>-0.06</td>
<td>-0.94</td>
</tr>
</tbody>
</table>

The relative free cross flow area $\tilde{F}_a = F_a/F_{a,0}$ describes the reduction of the air side cross flow area. It is assumed that the geometry between two tubes and two fins can be described as a rectangle with tube pitch $\pi_t$ and fin pitch $\pi_{fin}$ as sketched in Fig. 3.6, where the frost layer grows at the sides:
\[
\bar{F}_a = \frac{(\pi_t - 2d_\phi)(\pi_{fin} - 2d_\phi)}{\pi_t \pi_{fin}}
\] (8.18)

The thickness of the frost layer \(d_\phi\) is calculated with a constant frost density \(\rho_\phi\):

\[
d_\phi = \frac{\dot{m}_w}{\rho_\phi A_a}
\] (8.19)

Figure 8.3: Experimental data and simulation results for frost growth on the evaporator surface. \((T_a = 1.67^\circ C)\)

Fig. 8.3 shows experimental data for a microchannel evaporator with a fin pitch of 1.5 mm and tube pitch of 9 mm. R410A is used as refrigerant. Ambient temperature is constant at 1.67\(^\circ\)C. The approach temperature difference \(\Delta T_{ae}\) increases over time with frost growth as the free flow area on the air side is increasingly reduced. The evaporator capacity is reduced accordingly. The data for a relative humidity of 81\% is used to fit the model coefficients \(\xi\). The model then is able to reasonably predict both initial capacity and the capacity drop for a test run with an increased humidity of 86\% for the same refrigerant volume flow rate and fan speed.

**Cycle comparison**

The frost growth evaporator model is implemented in the basic cycle BC, the flooded evaporator cycle FE and the economizer cycle EC as described in Chapter 6.2. Fixed compressor speed is assumed. Heat transfer coefficients in both evaporator and economizer cycle are calculated with the according correlations. Isentropic and volumetric compressor efficiency is kept constant. Wet bulb temperatures for different ambient temperatures are taken from EN14825 (2010). For the BC
layout optimal sizes for evaporator, condenser and compressor displacement volumes are used ($A_e = 3.6 \text{ m}^2$, $A_c = 2.1 \text{ m}^2$, $\dot{V}_p = 14 \text{ m}^3 \text{ h}^{-1}$). The same evaporator and condenser size is kept for FE and EC layout. The compressor displacement volume is adjusted for FE and EC layout such that condenser capacity with clean coil is equal for all cycle layouts.

Fig. 8.4 shows for the basic cycle layout the decrease of evaporation temperature and heating capacity under the influence of growing frost for two air inlet temperatures. The decrease is much stronger for the higher air inlet temperature. After 1.5 hours the amount of frost on the evaporator surface is less than half for the colder condition. This is due to the higher humidity ratio at higher ambient temperature which is in accordance with results from the literature as described above. As the capacity at lower ambient temperature is lower for fixed compressor speed, the negative effect of decreasing evaporator performance on system performance is also reduced.

In Fig. 8.5 the results for the three cycle layouts are compared. In accordance with results from Chapter 6.2, the evaporation temperature with clean evaporator coil is higher for the FE layout and lower for the EC layout in comparison to BC. However, the frost growth effect varies only little. Neglecting cycle degradation effects, the total power consumption of the unit is increased by 7%, 8% and 7% after 1.5 hours of operation for the BC, FE and EC layout. After 2.5 hours power consumption is increased by 26.7%, 31% and 28.6% for the three layouts. Averaged power increase over a heating period of 1.5 hours compared to operation with dry air is approximately 2.5% for all cycle layouts.
For a heating period of 2.5 hours the averaged power increase is 7.6%, 8.6% and 7.8% for BC, FE and EC layout.

Since the capacity delivered by the heat pump unit is higher than the building demand, the unit is operated in on/off mode. A decrease in heating capacity due to growing frost thus actually reduces the numbers of switching on and off required by the room temperature control. Therefore losses induced by this switching are reduced. Assuming a cyclic degradation factor of 0.9, the averaged power consumption increase within a heating interval of 1.5 hours is only 1.9% for the BC cycle.

### 8.3 Defrost

As illustrated by the capacity decrease in Fig. 8.4.b, at a certain point defrosting of the evaporator is necessary to ensure long-term operation of the heat pump under frost conditions. Typically, the shorter the heating interval, the less frost grows and the smaller the average performance losses during heating. However, also each defrost interval penalizes the heat pump efficiency because energy is used and no capacity is delivered. Balancing the length of the heating interval and the number of defrost intervals is therefore an optimization problem which attracted a significant research effort. The method of defrosting, the influence of heat pump type, evaporator geometry, capacity and fan control on frost and defrost characteristics, and the determining and sensing of the optimal defrost initiation condition, defrost operation time and intervals between defrosts are issues of interest. Basic equations for optimizing
interval lengths are given by Granryd (2005).

The most basic defrost method is the compressor shut down which passively allows melting of the frost for ambient temperatures above 0°C. For residential air-water heat pumps the typical on/off capacity control method thus serves under some frosting conditions also for defrosting. If the fan is kept running in the off period, defrosting is more efficient. Practical limits of this "self-defrosting" are given by Granryd (2005). Typical active methods which remove frost from the evaporator also at lower ambient temperatures are electric resistance heating, hot gas bypass and reverse cycle defrosting. The electric resistance heating is the least complex from a component point of view, however the direct use of electricity strongly reduces system efficiency and requires long defrost periods as claimed by Wenju et al. (2011). In the hot gas bypass method the hot gas from the compressor outlet is directly passed into the evaporator. In the method most commonly found in residential units, the reverse cycle defrost, the refrigerant flow in the unit is reversed. The frosted evaporator becomes the condenser; heat is taken from the water side.

Table 8.2 shows percentage losses induced by reverse cycle defrosting at different air temperatures, as measured by Flach-Malaspina (2004) for an air-water heat pump. Capacity and COP in a full cycle of heating and defrosting is compared to the capacity and COP measured during the heating interval only.

<table>
<thead>
<tr>
<th>$T_a$ [°C]</th>
<th>-7</th>
<th>-3</th>
<th>0</th>
<th>3</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative humidity [%]</td>
<td>88</td>
<td>87</td>
<td>86</td>
<td>81</td>
<td>76</td>
</tr>
<tr>
<td>COP change [%]</td>
<td>14</td>
<td>17</td>
<td>17</td>
<td>16</td>
<td>0</td>
</tr>
<tr>
<td>Capacity change [%]</td>
<td>13</td>
<td>15</td>
<td>15</td>
<td>13</td>
<td>5</td>
</tr>
</tbody>
</table>

### 8.4 Annual cost assessment

Annual operating costs are affected both by performance losses in the heating interval with growing frost (Chapter 8.2) and by losses during defrosting (Chapter 8.3). For estimating the increase in annual operating costs here a reference dry air case is chosen, namely the basic cycle with on/off capacity control presented in Chapter 6.2. To quantify the costs induced by frost growth the average power consumption increase of 2.5% for a heating period of 1.5 hours (Chapter 8.2) is assumed to be constant for all operating conditions between -7 and 7°C. In practice the speed of frost growth will differ for the different conditions, however the
length of the heating period can be adjusted accordingly. To quantify also the defrosting costs, simulated COP and capacity including losses by frost growth are reduced according to the values given in Table 8.2.

Resulting annual operating cost increases are presented in Fig. 8.6 for the warmer, average and colder climate. In this example the cost increase induced by defrosting is big compared to the losses induced by growing frost on the coil. However, the ratio between both can be varied to optimizing the balance between the length of the heating interval and number of defrosts for minimal operating cost increase. Obviously for air-water heat pumps the effect of frost and defrost on annual costs is substantial for operation in average and especially colder climates.

### 8.5 A method for passive defrosting

A new method to prolong the frost growth period and thereby reduce the number of defrosts is proposed here. It is based on shutting or cycling off individual parallel passes of a multipass evaporator. By using a new distributing valve (EcoFlow) as illustrated in Figure 8.7, control of the flow of refrigerant through the different passes is made possible. This is achieved by rotating the left disc shown in Figure 8.7 such that refrigerant is pulsed through one valve outlet at a time. The desired distribution is specified in form of a distribution vector and is implemented as a dwell fraction in front of each outlet. The defrost off-cycling is achieved by specifying a distribution vector with zero dwell-time for the off-cycled pass. While the other passes are operating normally, no refrigerant is fed to the off-cycled pass. Since the air continues to flow
over the closed pass the surface temperature rises and for air temperatures above 0°C the frost melts and evaporates. For air temperatures below the freezing point the flow of the air leads to a compacting of the frost layer; thereby heat transfer is increased and capacity can be partly recovered. Each pass is alternately cycled off; this cycling is continuously performed under frosting conditions.

Figure 8.7: The internal parts of the EcoFlow valve showing the distribution of refrigerant to the individual evaporator circuits.

Passive defrost modeling

Transient models for a reversed cycle hot gas defrost are presented amongst others in Dopazo et al. (2010) and Liu et al. (2003). However, there are no models available describing the effect of passive defrosting by air blowing over the evaporator coil. Hence for modeling the defrosting period here only some basic assumptions are formulated.

In the defrost period initially the liquid refrigerant remaining in the channel evaporates after closing the valve. The time until the channel is evacuated depends on the mass of liquid refrigerant at the time of closing, the suction pressure and the heat transfer rate. During this time the frost layer continues to grow. As soon as all liquid refrigerant has evaporated, the temperature of the pass and the frost surface starts to increase until both reach the temperature of the inlet air. This whole period is modeled as a constant during which the transferred heat decreases linearly to zero and the frost surface temperature increases accordingly to the air inlet temperature. As soon as the humidity ratio at the frost surface exceeds the inlet humidity ratio, water starts to sublime from the frost layer surface, this sublimation mass flow rate can again be described by Eq. (8.9). A second process of frost removal starts as soon as the frost temperature increases beyond the freezing temperature: the frost starts to melt. The liquid then starts to drain
both under the influence of gravity and the shear stress of air flowing over the surface. To maintain the model as simple as possible, again a correlation between the drainage mass flow rate $\dot{m}_\phi$ and the relative cross flow area of the evaporator $\tilde{F}_a$ is formulated using a factor $\xi$ to adjust to experimental results:

$$\dot{m}_\phi = \xi e^{\left(-2.5(1-\tilde{F}_a)\right)}$$  \hspace{1cm} (8.20)

### Simulation results

The on/off operation of one of the evaporator passes with silent defrost is illustrated in Figure 8.8. After 720 seconds the opening degree of the valve is set to zero, the pass inlet is closed. The liquid refrigerant remaining in the pass is evaporated during the initial part of the off-cycle period. Frost continues to grow, decreasing further the relative free cross flow area $\tilde{F}_a$ for this pass. When the surface temperature increases subsequently, the residual frost formation starts melting, $\tilde{F}_a$ starts to increase. If the off-cycle period is long enough, the residual frost formation on the off-cycled evaporator pass is totally melted. This pass becomes frost free, $\tilde{F}_a$ equals one. The figure illustrates an off-cycle period which exceeds the melting period. Obviously, it would be ideal to switch the pass on just after the residual frost has been fully melted. Applying a shorter off-cycle time however would cause a residual frost layer to remain on the coil for the next on period of the cycle. This would lead to a slow frost build-up of the coil until a reverse cycle defrost is eventually needed.

![Figure 8.8: Air side free flow area on one pass.](image-url)

Two parameters can be adjusted to optimize the performance of the heat pump under this defrost operation scheme: The cycle period, hence the time between repeated shutting off of the same pass, and the defrost...
ratio which describes the ratio between the sum of all off-periods in a cycle to the total cycle period. For a defrost ratio of one, single passes are switched off one after the other with no time in between where all passes are fed; a defrost ratio of zero describes normal operation without cycling off of passes. The choice of cycle period and defrost ratio depends both on the evaporator geometry and the operating conditions. Different fin spacing, pass number and inner volume of the evaporator require different settings. As described in the introductory part the frost growth depends on air temperature, humidity and velocity and coil surface temperatures. These conditions however also influence the frost melting efficiency in the off-period of a pass, making the adjustment of cycle period and defrost ratio necessary if an optimal performance should be reached.

![Normalized capacity vs. Defrost Ratio](image)

Figure 8.9: Influence of defrost ratio on capacity during operation.

Figure 8.9 illustrates the changes in capacity over time for different defrost ratios. Shown is the time period until under normal operation the capacity is reduced by 20% compared to the clean coil and defrosting would be necessary. For a small defrost ratio of 0.2 the off period of an individual pass is shorter than needed to evaporate the liquid refrigerant in the pass, no defrosting occurs. Because part of the heat transfer area is regularly deactivated, evaporation temperatures and thus the capacity are lower than in normal operation. With an increased defrost ratio of 0.6 the time averaged capacity in the beginning of the frost growth period is even lower. The off period of an individual cycle is still too short to fully defrost the surface, frost is increasingly growing. The period until a regular defrost is necessary is nonetheless already prolonged compared to the normal operation. For a defrost ratio of one the frost is fully melted off the individual passes, frost layers stay thin, a regular defrost is not necessary. Averaged over the whole time period, the capacity as well as the COP is higher than under normal operation. Generally speaking longer off-periods improve defrost of the
off-cycled pass, however evaporation temperatures are reduced and in the meantime thicker frost layers build on the remaining passes, thus hampering defrost efficiency due to reduced air flow.

**Experimental Results**

In the experimental setup a reversible residential air conditioning unit with a nominal cooling capacity of 10.5 kW is used. The split system consists of an indoor heat exchanger installed in an air duct and an outdoor heat exchanger which is operated as evaporator in heating mode. The outdoor heat exchanger is a louvered fin microchannel heat exchanger. In the inlet header baffles are inserted in a way that a number of parallel flat tubes comprise one pass as schematically illustrated in Fig. 8.10.a. The heat exchanger is divided into six such passes in total.

![Schematic illustration and frost build up](image)

**Figure 8.10:** Microchannel evaporator: a) Schematic illustration and frost build up after 100 min operation with b) TXV and c) EcoFlow.

The test procedure follows AHRI-Standard:210/240 (2010) for heating mode test conditions for units having a variable speed compressor. Frost and defrost behavior is tested at intermediate compressor speed with an indoor temperature of 21.1°C and an outdoor temperature of 1.67°C with a relative humidity of 81.8%. Accuracy of the measurements follows the values given by the standard. Comparisons are made between a setup with a thermostatic expansion valve (TXV) mounted on the outdoor heat exchanger and one with the distributing valve, both are tested under the same operating conditions. Due to the possibility of balancing the refrigerant mass flow in the different passes with the distributing valve the superheat at the outlet of the heat exchanger could be reduced to 3 K compared to 5 K for the TXV maintaining a stable superheat control. Since the outdoor heat exchanger is designed
for cooling purposes mainly it is oversized for heating mode; hence the evaporation temperature could be increased by 2 K accordingly when operating the system with the distributing valve. With the TXV tests are run until the capacity is dropped by 20% from the initial value which occurs at approximately after 100 minutes of operation. For the setup with the distributing valve a cycle period of 360 seconds with a defrost ratio of one is chosen.

![Figure 8.11: Normalized capacity and COP for a heat pump with microchannel heat exchangers.](image)

Results of both test runs are presented in Fig. 8.11. The values for capacity and COP are normalized with values of the TXV setup for a clean coil at the beginning of the heating interval. The graph illustrates that by utilizing the EcoFlow valve in passive defrost mode a continuous operation can be maintained without a drop in capacity or COP. After 100 min operation time at the end of each pass off period the respective pass is still fully defrosted. A visual comparison of the frost state at that time is given in Fig. 8.10.b and c for operation with a TXV and the distributing valve respectively. Already at the frost free state with the EcoFlow valve an increase both in COP and capacity can be observed. This can partly be explained by the increase in evaporation temperature gained by reduced superheat temperatures. However also increasing heat transfer coefficients with higher refrigerant velocities might play a role when only a part of the heat exchanger is in operation. The main effect leading to an improvement in system efficiency is however that strong buildup of frost layers on the coil can be prevented under the given conditions. Averaged over 100 minutes operation time (until defrosting would be necessary for the system with the TXV) an increase in COP of approximately 15% can be achieved for the given system.

The tested microchannel heat exchanger is a prototype design with vertically oriented tubes. However the passive defrost procedure is also possible for the more common horizontal orientation. Previous tests
showed that the distribution of liquid and gas phase in the header section is not affected by orientation: The momentum induced by the pulsation due to the rotation of the disk dominates the flow pattern. However, the defrost routine needs adaptations in case of a horizontal orientation of the passes. For evaporator geometries with round tubes and plate fins it was observed that passes in the lower part of the heat exchanger have to be cycled off for longer periods than passes in the upper part since some of the melted frost flows downwards and freezes on the lower passes. This should be equally valid for microchannel geometries.

8.6 Concluding remarks

In cold and average climate zones frost and defrost has a significant effect on the annual performance of an air-water heat pump. Since especially the defrosting power consumption is high, reducing the number of active defrosts thus is a promising approach to improve system performance. A new method of defrosting heat pump evaporators by cycling off individual evaporator passes is presented. Based on a simple mathematical model the principles of the method as well as optimality considerations for adjusting the parameters of cycle time and defrost ratio are discussed. The feasibility of the method and the potential for improving system efficiency is demonstrated on an experimental setup with a microchannel heat exchanger in a system using variable capacity control. Comparing two similarly sized systems utilizing either a microchannel or a fin and tube evaporator Shao et al. (2010) found that the system with a microchannel evaporator needs to operate with much shorter frosting periods both due to maldistribution effects and reduced fin spacing. The continuous frost removal of the proposed method prolongs the frosting period notably and therefore reduces the number of necessary standard defrosts or makes them even superfluous under many operating conditions. Hence the proposed method might be especially suitable for microchannel evaporators, though other tests not presented here were also conducted successfully for fin and tube geometries. The method requires to equip the system with a new type of valve and to adjust the parameters for the control routine. The evaporator circuitry for microchannel heat exchangers has to be adapted by sectioning the inlet header. Round tube with plate fin geometries can either be used directly without changes or the circuitry can be optimized for operation with the new defrost routine. No other changes in component design or system setup are necessary.

The heat exchanger used in the defrost experiment (Fig. 8.11) shows a much stronger capacity decrease with growing frost than observed for other designs (Fig. 8.3). Severe maldistribution can be identified as
one cause. Thus, technologies to counteract maldistribution will under frost conditions show higher performance improvements than presented in Chapter 4. Simulations for three different cycle layouts showed very similar frost growth behavior. Because therefore also defrosting power consumption will be similar, results of the screening method utilization example presented in Chapter 6 will not be strongly affected by frost effects. More complex is the situation for the comparison of on/off and variable speed capacity control, presented in Chapter 5. Within the ambient temperature range of occurring frost growth the capacity delivered by the fixed speed heat pump is typically above building demand, the unit is therefore operated in on/off mode. So on the one hand in variable speed mode the frost growth will be reduced as less refrigerant is evaporated due to the reduction in capacity compared to fixed speed control in on-mode. On the other hand, at ambient temperatures above 0°C, on/off control offers the possibility for natural defrosting in the regular off period. The efficiency of this immanent defrost depends on complex ambient conditions like frost structure, wind, snow and solar radiation. For variable speed control however, dedicated off periods for defrosting have to be scheduled, thus reducing system efficiency. Whether positive or negative aspects of variable speed control under frosting conditions prevail cannot be answered without further research.
9 Conclusions

"Even so, it might be difficult to persuade the purchaser to afford the optimum plant."

- Brendeng and Aflekt (1980)

As it is nonetheless worth trying, in this study a screening method for finding optimal heat pump designs among various alternatives is developed by combining several approaches to simplify the optimization problem. The method which can be applied for different problem statements enables to gain a good understanding of each considered technology in a structured and computationally inexpensive way. This allows a fair comparison of competing alternatives and therefore to identify the solution with highest potential to increase heat pump efficiency at reasonable cost. Also, maybe even more importantly, technical challenges for the successful implementation of a new technology can be pointed out. Thus the presented structured approach for thoroughly testing new technologies in an early development phase can help to effectively use research resources. This can contribute to a faster market launch of innovative designs which are economic, reliable and reduce the carbon footprint of the consumer.

9.1 The screening method

In Chapter 1 three questions are formulated regarding the development of a structured method for finding optimal heat pump designs: Is it possible to simplify the problem by splitting the search space without loss of optimality? Is it possible to simplify the mathematical and numerical procedures required for optimization? Is it possible to evaluate the heat pump efficiency for realistic operating conditions without unjustified increase of effort? Here some answers should be given based on the work presented in this thesis.

The different utilization examples show that for small heat pumps the interactions between components, control, cycle layout and refrigerant selection are strong. Effects of all these aspects both on investment cost and efficiency of the heat pump unit are often in the same size of order. Chapter 6.3 shows that optimization of air flow rate and subcooling has a similar or even stronger effect on annular performance than changing the cycle layout. The size of the evaporator determines which approach for counteracting maldistribution is more promising as
shown in Chapter 4.3. Chapter 5.3 makes it obvious that the compressor size chosen by the engineer in a simulation model or experimental setup can determine which of two capacity control methods turns out to be economically preferable. Chapter 6.2 shows that differences in the capacity control method and the characteristics of the compressor can change the order of economic attractiveness of different cycle layouts. Thus a splitting of the overall search space by serial solution of smaller optimization problems which treat only individual aspects - e.g. a "top-down" approach of optimizing integer variables like cycle layout and then optimizing continuous variables like component sizes - is not possible without risking a serious loss of optimality.

Typical approaches to eliminate this risk are either a split of the search space and optimization combined with a thorough sensitivity analysis for all excluded parameters or a comprehensive formulation of the optimization problem for the full search space. The screening method presented in this thesis can be described as a hybrid of these approaches to avoid numerical problems induced by the comprehensive optimization and to reduce the simulation effort required for a thorough sensitivity analysis. To this end the search space is split: integer parameter changes describe technology alternatives, (quasi-)continuous parameters are included in an optimization routine and the sensitivity analysis is limited to those parameters and assumptions not covered in either of the two parts.

The metamodeling plays a central role in the screening method. It allows for the continuous parameters to decouple simulation of the heat pump unit, evaluation of annual performance and economic optimization. Thereby continuous optimization algorithms can be employed and the optimization can be repeated for varying economic parameters or strategies without additional simulation effort. As discussed in Chapter 3.5, compared to a simpler alternative offering the same benefits the computation time is reduced by 93%. To keep the errors induced by metamodeling low, a careful selection of an adequate model structure is required. The wider the value range of the optimization variables the higher is the risk that the system behavior cannot be reproduced satisfyingly. In the examples discussed in this thesis errors introduced by metamodeling are below 0.5 - 1.5%.

Effects of strongly varying operating conditions encountered by residential heat pumps throughout a year are addressed by the simple profile method described in Chapter 3.3. Based on eight simulations at varying conditions the annual performance in three climate zones can be calculated. A comparison of the profile method with a dynamic model, Chapter 7, shows comparable optimization results. Thus, with the profile method parameters on the level of the system environment can obviously be included in a generic but representative manner. The simula-
tion effort is only slightly increased compared to an approach that tries to find a representative or average operating condition for each climate zone.

To summarize, the results of this study show that with the screening method

- characteristics of components, control, cycle layout and refrigerant on the heat pump unit level can be described in sufficient detail,
- important parameters on the level of the system environment can be included in a generic but representative manner,
- the computational effort and convergence problems of the optimization algorithm can be reduced by the metamodel approach,
- varying economic parameters or strategies can be easily tested without additional simulation effort,
- and thereby a fair, comprehensive and comprehensible comparison of different technologies for varying demands and economic scenarios is enabled.

### 9.2 Potential for performance improvement

In Chapters 4 - 6 several technologies to improve energy efficiency of air-water heat pumps are discussed under the aspect of total cost of ownership. It is remarkable to notice that all technologies only lead to a considerable reduction of total cost of ownership for heat pumps running in the colder climate zone. Within ten years, assuming a building heat demand of 10 kW at -10°C, a medium water temperature level around 45°C, an electricity price of 0.21 € kWh\(^{-1}\) and an effective discount rate of 3%, with variable speed capacity control about 2000 € can be saved compared to an on/off controlled unit with basic cycle layout. Compared to the same reference unit, savings achieved with a two stage economizer cycle are about 1800 €. Savings achieved by counteracting air and refrigerant side maldistribution in the evaporator with individual superheat control in parallel passes are in the size of order of 500 - 1500 €, depending on the severity of maldistribution faced.

In the average climate zone these savings in total cost of ownership are reduced by roughly two thirds. The likelihood increases that variations in the economic framework or also less positive component performances than implied in the model will set these savings off in practice. For heat pumps running in the warmer climate zone the annual savings in operating costs achieved by all technologies are so small that they can
usually not compensate the increase in investment costs induced by a technology change. For these climatic conditions the basic cycle layout with standard on/off control seems the most economical solution.

The benefit of the two most promising solutions, variable speed control and two stage cycle layout, results largely from the reduction of direct electric heat at low ambient temperatures. However, their performance is severely challenged at high ambient temperatures. These conditions are characterized by low pressure differences between the compression stages in the economizer cycle and by a severe reduction of compressor speed in case of variable speed control. The total efficiency of the compression process is considerably reduced if the characteristics of typical heat pump compressors are assumed and the influence of motor and inverter losses is factored in.

The individual superheat control which is dedicated to counteract evaporator maldistribution induces a considerable increase in control complexity in comparison to the competing approach to optimize evaporator circuitry for that purpose. At the same time possible operating cost reductions are lower compared to variable speed control and two stage cycle layout. However, with little additional effort also performance under frosting conditions can be improved with this technology. The increase in annual operating costs induced by defrost requirements is only slightly smaller than the possible savings achieved with variable speed control. Thus the potential of the individual superheat control strategy to reduce operating costs is increased. In comparison, the two stage layout shows a similar frost growth behavior as the basic cycle. The improvement potential of this technology will not considerably change if frost is factored in. As discussed in Chapter 8.6, the effect of frost on the economic comparison of fixed and variable speed control cannot be quantified with the employed models.

The sensitivity analysis addressing those assumptions and parameters on the unit level which are in this study neither described by technology alternatives nor by optimization shows no significant effects on the results of the technology comparisons. Also variations in the economic framework within a realistic range do not change the observed trends. However, differences in the assumptions about the system environment show a larger influence. Thus, the observed savings potential both of variable speed control and two stage cycle layout is affected by variations in the system environment. Higher required water outlet temperatures will considerably increase the benefit of two stage compression while the performance improvement gained with variable speed control is not strongly affected. Higher annual building demands increase especially the benefit of variable speed control as the inclination of the demand curve steepens but also make the two stage cycle layout economically more attractive. This means in reverse that the smaller the building,
the better the insulation and the more modern the hydronic system, the less economically attractive changes to the standard on/off controlled basic cycle become.

In general it can be observed that the savings potential of all considered solutions is smaller than may have been expected from various publications which discuss the different technologies under the aspect of improving COP and capacity at a few operating conditions. The wide range of operating conditions, a strong emphasis on conditions with mild ambient temperatures and practical limitations encountered with all alternatives make a drastic economic improvement by changes of the heat pump unit design difficult.

9.3 General recommendations

The effects on investment and operating costs of refrigerant selection, cycle layout, control and component characteristics are very tightly connected. Each intervention in the design of the heat pump unit with a new or altered component or technology therefore should involve careful adjustments on all levels. Only an integrated optimization of the whole unit or thorough sensitivity testing allows preventing unforeseen negative interactions and thus exploiting the full potential of a new solution. In practice this requires a tighter cooperation between component and heat pump manufacturers as component and system design have to be harmonized. It also requires a comprehensive view on the development process as a system optimization problem and therefore an intensified use of computer-aided design and optimization methods.

For the optimization of an application like air-water heat pumps a holistic evaluation is indispensable. The units operate under widely varying operating conditions and at the same time variations in the system environment have a sometimes stronger effect on optimality than changes of intrinsic characteristics of the unit or the economic framework. This can be taken into account by quantifying the annual performance for different scenarios. The evaluation of annual performance instead of COP and capacity at selected conditions has the added benefit that a single performance criterion - annual work or annual operating costs - can be used. Thus different technologies can be more easily and objectively compared. Calculating annual operating costs is useful even if investment costs are unknown. In a reversing of the problem formulation it can be quantified how much a new technology is allowed to cost to be economically promising.

To pick low hanging fruit first, the annual performance of heat pumps running in colder climates can be improved by reducing the amount of
direct electric heating at low ambient temperatures required to meet the building demand. Different technologies are available for this purpose, in this study variable capacity control and two stage layout of the refrigerant cycle are discussed in detail. Also an increase of the design capacity of on/off controlled heat pumps would lead to this goal. However, for each of these solutions the risk exists that unit performance at high ambient temperatures is negatively affected. Seen from the annual perspective in colder climate zones this reduces the gains achieved at low temperatures. The impact on the performance in average or warmer climate may even become negative. Therefore for technologies which reduce electric heat a careful testing at high ambient temperatures is necessary. Options like bypassing a compressor stage or a combination of speed reduction and on/off control at high ambient temperatures could be considered.

It is however obvious that heat pumps optimized for operation in colder climate zones are economically unfavorable for customers in the average and especially the warmer climate zone. Even if careful design avoids performance reductions at high ambient temperatures, the increase in investment costs is hardly justifiable. On the other hand, compromises which reduce investment costs and therefore help customers in average and warmer climates lead to unnecessarily high operating costs in cold climates. From a consumer point of view the most cost-effective and from a reliability point of view the least risky approach is to design heat pumps individually for the varying climate zones. The optimal sizing and design of the components for the different demands offer a strong improvement potential without requiring a change to more complex technologies for heat pumps which mainly operate in mild ambient temperatures. This is not only beneficial regarding investment costs but also regarding the reduced potential for errors during installation and maintenance. However a diversification of the product portfolio challenges procurement, development and sales departments of heat pump manufacturers. Also the prices of specialized components may increase due to the reduced sales numbers.

\section*{9.4 Possibilities for further investigations}

In this study only space heating is considered. However, the description of the system environment can be extended to include the demand for domestic hot water production. Again, it should be accounted for the fact that hot water has to be produced throughout the year at very different outdoor conditions. If representative data could be collected, air-water heat pumps can be optimized for the combined demand of space heating and hot water production with the screening method.
The presented optimization studies only consider components of the heat pump unit. The heat distribution in the building is represented with rudimentary temperature profiles. From an end customer point of view aspects affecting these profiles like pump speed, dimensioning of radiators, tanks or other components in the hydronic cycle should also be subject to optimization.

Frost and defrost remains an interesting field for improving air-water heat pump performance. A simple transient frost model as presented in this study can be integrated in the heat pump unit model and thus included in the screening method. However it should be anticipated that for different technologies the optimal lengths of heating and defrosting periods vary. This is therefore an optimization problem in itself. Also thorough model development for both active and passive defrosting as well as validating with experimental data is required.

The screening method can be used for many other problems of air-water heat pump unit design. However, the approach may also be suitable for other small vapor compression systems with strong interactions between components. This may not only include other types of heat pumps but also air conditioning units, bottle coolers or transport cooling systems.

In the continuous optimization problem other relevant input parameters could be included like fan size or ratio of evaporator face area to evaporator size. This requires an adequate detailing of the heat pump unit model and a careful choice of the metamodel structure. The accuracy of the metamodel should be cautiously tested.

The screening method applies mathematical optimization algorithms only for continuous optimization variables. In all utilization examples each variation of integer variables is treated as an individual technology alternative. The benefit of this approach is that thereby a thorough understanding of each technology can be gained. However, in a ”two-level” approach the screening method can also be integrated with an optimization algorithm for the integer variables. The number of simulations required to solve classical mixed-integer optimization problems increases exponentially with the number of optimization variables. Therefore a considerable reduction of the simulation effort can be achieved if the continuous optimization variables are treated independently in the way demonstrated here.
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Tiedemann, T. Personal communications, Danfoss, Offenbach, Germany. 2013.


## Nomenclature

### Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>BC</td>
<td>basic cycle layout [-]</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance [-]</td>
</tr>
<tr>
<td>COP&lt;sub&gt;r&lt;/sub&gt;</td>
<td>corrected coefficient of performance [-]</td>
</tr>
<tr>
<td>EC</td>
<td>economizer cycle layout [-]</td>
</tr>
<tr>
<td>IX</td>
<td>internal heat exchanger cycle layout [-]</td>
</tr>
<tr>
<td>FE</td>
<td>flooded evaporator cycle layout [-]</td>
</tr>
<tr>
<td>FGB</td>
<td>flash gas bypass cycle layout [-]</td>
</tr>
<tr>
<td>FS</td>
<td>fixed speed on/off capacity control [-]</td>
</tr>
<tr>
<td>ISC</td>
<td>individual superheat control [-]</td>
</tr>
<tr>
<td>IX</td>
<td>internal heat exchanger cycle layout [-]</td>
</tr>
<tr>
<td>MSS</td>
<td>minimum stable superheat [-]</td>
</tr>
<tr>
<td>OC</td>
<td>open economizer cycle layout [-]</td>
</tr>
<tr>
<td>OEM</td>
<td>original equipment manufacturer [-]</td>
</tr>
<tr>
<td>SCOP</td>
<td>seasonal coefficient of performance [-]</td>
</tr>
<tr>
<td>TCO</td>
<td>total cost of ownership [€]</td>
</tr>
<tr>
<td>TXV</td>
<td>thermostatic expansion valve [-]</td>
</tr>
<tr>
<td>VS</td>
<td>variable speed capacity control [-]</td>
</tr>
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</table>
Roman

\(A\) heat transfer area \([m^2]\)

\(a\) economic lifetime \([yr]\)

\(\dot{C}\) heat capacity rate \([J \, s^{-1} \, K^{-1}]\)

\(c\) cyclic degradation coefficient \([-\]\)

\(c_p\) specific heat capacity \([J \, kg^{-1} \, K^{-1}]\)

\(d\) diameter, thickness \([m]\)

\(F_a\) free air side cross flow area \([m^2]\)

\(F_0, F_1, F_2\) maldistribution cases

\(f_a\) air flow distribution parameter \([-\]\)

\(f_x\) phase distribution parameter \([-\]\)

\(H\) heating hours per year \([h \, yr^{-1}]\)

\(h\) specific enthalpy \([J \, kg^{-1}]\)

\(I\) investment costs \([\text{\euro}]\)

\(i\) effective discount rate \([-\]\)

\(\dot{J}\) molar mass flow rate \([mol \, s^{-1}]\)

\(L\) zone length \([-\]\)

\(\dot{m}\) mass flow rate \([kg \, s^{-1}]\)

\(N\) normalized factor level \([-\]\)

\(O\) operating costs \([\text{\euro} \, yr^{-1}]\)

\(P\) power \([W]\)

\(p\) pressure \([Pa]\)

\(p_{el}\) electricity price \([\text{\euro} \, kWh^{-1}]\)

\(\dot{Q}\) heat transfer rate \([W]\)

\(r_A\) ratio of outer to inner heat transfer area \([-\]\)

\(S_i\) model scenarios \([-\]\)

\(T\) temperature \(^[\circ C]\)
\( \Delta T_{ae} \)  
approach temperature difference between air inlet and evaporation [K]

\( \Delta T_{cw} \)  
approach temperature difference between condensation and water inlet [K]

\( \Delta T_{SC} \)  
degree of subcooling [K]

\( \Delta T_{SH} \)  
degree of superheat [K]

\( U \)  
overall heat transfer coefficient [W m\(^{-2}\) K\(^{-1}\)]

\( 
\dot{V} \)  
volume flow rate [m\(^3\) h\(^{-1}\)]

\( W \)  
annual work input [kWh yr\(^{-1}\)]

\( X \)  
specific humidity [-]

\( x \)  
quality [-]

\( y \)  
year [yr]

\( \vec{Z} \)  
vector of optimization variables [-]

\( z \)  
optimization variable [-]

**Greek**

\( \alpha \)  
heat transfer coefficient [W m\(^{-2}\) K\(^{-1}\)]

\( \beta \)  
mass transfer coefficient [m s\(^{-1}\)]

\( \epsilon \)  
effectiveness [-]

\( \eta \)  
efficiency [-]

\( \pi \)  
pitch, distance [m]

\( \rho \)  
density [kg m\(^{-3}\)]

\( \tau \)  
payback time [yr]

\( \sigma \)  
normalized factor limit [-]

\( \psi \)  
pressure drop constant [-]

\( \chi \)  
concentration [mol m\(^{-3}\)]

\( \xi \)  
frost coefficients [-]
### Subscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>nominal, baseline</td>
</tr>
<tr>
<td>2p</td>
<td>two phase</td>
</tr>
<tr>
<td>a</td>
<td>air, ambient</td>
</tr>
<tr>
<td>b</td>
<td>capacity balance point</td>
</tr>
<tr>
<td>bu</td>
<td>upper capacity balance point</td>
</tr>
<tr>
<td>bl</td>
<td>lower capacity balance point</td>
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$q$  continuous quadratic model
$r$  refrigerant
$S$  sensible
$s$  surface
$\sigma$  at upper design limit
$-\sigma$  at lower design limit
$t$  tube
$v$  volumetric
$vi$  sublimation
$x$  expansion
$y$  annual