Optimization of Heating, Ventilation, and Air-Conditioning (HVAC) System Configurations

By

Muzaffar Ali
08-UET/PhD-ME-45

Supervised by
Prof. Dr. Mukhtar Hussain Sahir

Department of Mechanical Engineering
Faculty of Mechanical and Aeronautical Engineering
University of Engineering & Technology Taxila, Pakistan
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Author
Muzaffar Ali
08-UET/PhD-ME-45

Checked and Recommended by:

Research Monitoring Committee:

Prof. Dr. Mukhtar Hussain Sahir
Research Supervisor

Prof. Dr. Arshad Qureshi
Member Research Committee

Prof. Dr. Khalid Akhtar
Member Research Committee
National University of Science and Technology (NUST)

Dr. Irfan Gondal
Member Research Committee
National University of Science and Technology (NUST)
DECLARATION

The substance of this dissertation is original work of the author. Due reference and acknowledgement has been made, where necessary, to the work of the others. No part of dissertation has already been accepted for any degree, and it is not being currently submitted in candidature of any degree.

Author

Muzaffar Ali

08-UET/PhD-ME-45
ABSTRACT

Today, a substantial amount of world’s energy is being consumed by the building sector. In buildings, energy share of Heating, Ventilation and Air-Conditioning (HVAC) systems is dominant. Various aspects of HVAC system optimization are analyzed in several studies; however the optimization of HVAC system configurations is rarely analyzed. Moreover, no systematic approach is developed for evaluating different HVAC system design alternatives to decide the optimal configuration for a specific building load demand and climate conditions.

The thesis is aimed to develop a methodical simulation-based optimization approach for the effective and efficient evaluation of various HVAC system alternatives at the initial system design stage to result in an optimal configuration. The approach is implemented at both system design level as well as at the configuration level. It is concluded from literature review that the equation-based object-oriented (EOO) modeling and simulation approach provides a conducive environment for evaluating HVAC system configurations.

In the developed approach, two methods are proposed with respect to HVAC system modeling and simulation characteristics. The first method ‘conditional declaration of component models’ is used for automated selection of optimal system configurations. It uses the coupling between modeling tool Dymola/Modelica and the optimization tool GenOpt through an appropriate algorithm. The second method ‘redeclaration/replaceable component models’ is also proposed for HVAC system modeling in which several types of a specific component class can be varied to define various system configurations. It includes empty component models of all system component with connecting ports and without heat and mass transfer. Such multifunctional modeling enables to simulate variety of component models for performance analysis of different HVAC system configurations that can be considered in the future research. However, the method has the limitation that the component class cannot be changed automatically in the current version of Modelica language. Though, the improvement in the language is underway to incorporate such feature.

In the current work, both methods are implemented for the performance optimization of two real HVAC systems: chilled water system installed in Symantec Corporation building in South California, USA and desiccant cooling system operating in ENERGYbase building in Vienna, Austria.

The implementation of the first method established an incremental development of the methodology for chilled water system design optimization. The optimization is performed at the system design as well as configuration level through development of a dynamic chilled water system model in Dymola/Modelica capable of varying system design and configuration parameters. The approach proved efficient in terms of estimating time and labor. The overall optimization of chilled water system at system
design and configuration level with fixed building load demands resulted in 17-43% reduction of power consumption. In addition, estimated initial cost of each chilled water system configuration is also evaluated for economic considerations.

For performance analysis of desiccant evaporative cooling system, a desiccant wheel model is developed in EOO program, Dymola/Modelica. The existing wheel models in other modeling environments lack the capabilities to handle real-time control strategies with respect to wheel operational modes and sensor functionalities. Therefore, the developed model incorporates real-time control strategies for both dehumidification and enthalpy modes under the transient operating conditions that are encountered in the commercial systems. The model is calibrated and validated under the transient measurements obtained from ENERGYbase building desiccant wheel installation.

However, the other available component models of desiccant cooling system are designed for ideal behavior. Therefore all relevant component models, such as humidifier and heat wheel are modified to match the real system operation. The modified component models are calibrated and validated under the transient measurements obtained from ENERGYbase monitoring system. Afterwards, the validated component models are used to develop desiccant evaporative cooling system model. Then validation is performed at the system level based on ENERGYbase system measurements.

The second proposed method is applied on the validated desiccant cooling system model for an appropriate selection of desiccant cooling system configuration. Five system configurations consist of ventilation, standard recirculation, standard dunkle, ventilated recirculation, and ventilated dunkle cycles are evaluated in five climate zones: Vienna, Karachi, Sao Paulo, Shanghai, and Adelaid. The evaluation resulted that the ventilated dunkle cycle is the most efficient in terms of high COP system configuration in three climate zones except Karachi and Shanghai climates that are more suitable for the ventilation system configuration. Moreover, the estimated initial cost of each configuration is also provided that could be further helpful for overall configuration evaluation.

To conclude, the presented systematic simulation-based optimization approach significantly helps to handle the complex task of optimal selection of HVAC system configurations. Such development would be supportive for the HVAC design practitioners to evaluate various system alternatives and select the optimal system configuration and design parameters at the initial design stage under various building load demands and climate conditions. Moreover, the methodology signifies a step forward toward the design of software systems able to synthesize new and optimal system configurations.
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LIST OF ABBREVIATIONS

A  cross sectional area of rotor (m^2)
a  channel height of desiccant wheel (m)
A_{sp}  specific surface area of desiccant wheel (m^2/m^3)
A_{ws}  surface area of desiccant wheel (m^2/kg-rotor)
b  channel base of desiccant wheel (m)
C  Langmuir adsorption constant (---)
C_F  Freundlich adsorption constant (---)
CC  cooling capacity (kW)
C_{pc}  specific heat capacity of desiccant wheel (kJ/kgK)
C_{pw}  specific heat capacity of water (kJ/kgK)
D  diameter of desiccant wheel (m)
D_0  coefficient of diffusivity (m^2/s)
D_{sf}  surface diffusivity (m^2/s)
D_{eq}  equivalent diameter of channels (m)
E_{at}  activation energy (J/kg)
En  cooling energy delivered (kWh)
f  Fanning friction factor (---)
G  gain in the delay model (---)
HVAC  Heating, Ventilation and Air-Conditioning
H  amount of sensible heat removed per hour (kJ/h)
h  specific enthalpy of air (kJ/kg)
k  loss coefficient due to contraction and expansion of air
K_w  humidity ratio correction coefficient (---)
K_t  temperature correction coefficient (---)
L  length of desiccant wheel (m)
ṁ  mass flow rate of air (kg/h)
MSE  mean square error (kJ/kg)
MPE  mean percentage error (%)
N  rotation speed of desiccant wheel (rph)
n_F  Freundlich adsorption constant (---)
P_w  partial pressure of water vapor in moist air (kPa)
P_{ws}  pressure of saturated pure water (kPa)
dp  pressure drop (Pa)
Q  amount of water vapor absorbed onto wheel during an hour (kg/h)
q_{eq}  equilibrium amount of water absorbed (kg of moisture/kg of adsorbent)
q  amount of water vapor absorbed onto wheel (kg of moisture/kg of dry air)
R  gas constant (J/kgK)
RMSE  root mean square error (kJ/kg)
S  humidifier pump signal (%)
t_a  adsorption time (s)
T  temperature (K)
U  superficial velocity of air (m/s)
u  air velocity in the channel (m/s)
W  amount of water absorbed (kg/kg of adsorbent)
Greek letters

\( \rho \) density \((\text{kg/m}^3)\)
\( \alpha \) Heat coefficient (---)
\( \beta \) fractional area of each zone (---)
\( \gamma \) Mass co-efficient (---)
\( \omega \) air humidity ratio \((\text{kg of moisture/kg of dry air})\)
\( \phi \) relative humidity \((0-1)\)
\( \epsilon \) effectiveness (---)
\( \tau \) time constant \((\text{s})\)
\( \lambda \) thermal conductivity \((\text{W/m}^2\text{K})\)

Subscripts

a air
ad adsorbent
at activation energy
avg average
c cold
cr corrected
dw desiccant wheel
d distributed
db dry bulb
eq equilibrium
h hot
in inlet
l local
lt latent
mon monitored
out outlet
opt optimal
pc heat capacity of air at constant pressure
pw heat capacity of water at constant pressure
reg regeneration air
s process air
sn sensible
sf surface
sp specific
sim simulated
tot total
wb wet bulb
ws wheel surface
Chapter 01

1 INTRODUCTION

Heating, Ventilation, and Air-Conditioning (HVAC) systems control the indoor environment throughout the year to ensure comfortable conditions in homes, offices, and commercial facilities. Beside the fact that HVAC systems are making human life healthier and more productive but various products could also be produced faster, better, and more economically in an appropriately controlled environment. Almost each residential, institutional, commercial, and industrial building has a year-round controlled environment in the developed countries of the world.

1.1 Motivation

The awareness of the importance of living place and indoor air quality has increased for both health and comfort. The growing requirements for closely controlled working environments in laboratories, hospitals, and industrial facilities have rapidly increased overall energy demands. Therefore the HVAC systems designers are challenged as never before to come up with the most energy efficient systems. One way of addressing the challenge is by optimizing conventional setups and introducing efficient innovative systems utilizing renewable energy resources [1].

1.1.1 Energy demand in buildings

Energy consumption of the world market is continuously increasing and projected to upswing by nearly 50% from 2009 through 2035. Most of the growth is occurring in evolving economies outside the Organization for Economic Cooperation and Development (OECD), especially in non-OECD Asia. Total non-OECD energy use rises by 84%, compared to a 14% increase in the developed OECD nations over the projected period. China and USA are leading countries with 20% and 19% share of world energy consumption in 2010 as shown in Figure 1.1 [2].
Figure 1.1: World primary energy consumption in 2010

Total energy consumption can be fragmented into four main sectors: industrial, transportation, commercial, and residential. Buildings energy consumption has significant contribution and its share is the greatest in most regions of the world depending on income levels, natural resources, climate, and available energy infrastructure.

The share of buildings sector accounted about 21% of world delivered energy consumption in 2008 as shown in Figure 1.2 [2]. Energy use in the residential sector accounted for about 14% of world delivered energy consumption, while the non-residential sector accounted for 7% of total delivered energy consumption.

Figure 1.2: World energy consumption by sectors

In 2009, the total energy consumed in residential and non-residential buildings is around 40% of total energy requirements in Europe [3]. In the perspective of all the end-use sectors, buildings represent the leading sector, followed by transport with 33% as
shown in Figure 1.3. The consumption for 1990 to 2009 shows two main trends: a 50% increase in electricity and gas use while a reduction in use of oil and solid fuels by 27% and 75% of final energy consumption, respectively. Overall, the energy use in buildings shows a upward tendency with an increase from 400 Mtoe to 450 Mtoe over the last 20 years.

In the building sector, European households were responsible for 68% of the energy use in 2009. The significant increase in use of appliances in households is also evident through the steady 38% increase in electricity consumption over the last 20 years. Similarly during the same duration, electricity consumption in non-residential buildings has increased by remarkable 74%. Offices, wholesale and retail trade buildings represent more than 50% of energy use. Education and sports facilities account for a further 18% of the energy use while other buildings account for about 6%.

![Energy consumption by sectors in EU](image)

**Figure 1.3: Energy consumption by sectors in EU**

In USA building sector alone accounted for 7% of global energy consumption and about 41% of primary energy consumption in 2010. The consumption is 44% more than the transportation and 36% more than industrial sector as shown in Figure 1.4 [4].

Primary energy consumption in the residential sector is 54% of consumption in the buildings sector and 22% of total primary energy consumption. Commercial buildings represent just under one-fifth of the USA energy consumption, with office space, retail space, and educational facilities representing about half of commercial sector energy consumption. In aggregate, commercial buildings are representing 46.0% of building energy consumption and 18.9% of the USA energy consumption.
In underdeveloped countries such as Pakistan, building sector share was 44% of final energy consumption in 2006-07. Residential buildings are the key contributor with 41% as shown in Figure 1.5. The amount of energy consumption in residential sector is projected to almost double in year 2030 and would be the largest component of final energy use in 2030. But it may drop to 32% of total final energy in 2030 [5].

Growth in population, enhancement of comfort requirement, global climate change, and time spent inside buildings (about 90% of our whole life) predicted the upward trend of energy consumption in building sector. Residential energy use is predicted to grow at an average rate of 1.1% per year from 2008 to 2035. Similarly, the growth in commercial sector is expected to increase at an average rate of 1.5% per year from 2008 to 2035.
Therefore, due to increasing energy demands, price, and environmental issues, the developed countries are focusing on buildings sector as the greatest potential for energy savings [3].

1.1.2 Energy Demand of HVAC systems

The buildings energy demands are the main source of substantial increase of the electricity consumption due to rising space heating, cooling, ventilation, and refrigeration requirements [6]. Energy consumption in buildings is directly related with energy demands of HVAC systems. HVAC is the largest energy end use both in the residential and non-residential sector. Studies indicate that air-conditioning is responsible for 10% to 60% of the total building energy consumption, depending on the building type [7]. In developed countries, HVAC systems are the most energy consuming devices, accounting for about 10–20% of final energy use [8].

In the USA, buildings account for 50% of the site energy consumption for combined space heating and cooling as shown in Figure 1.6 [4]. The share of HVAC in the residential sector is 61% of site energy consumption and 41% of primary energy consumption as shown in Figure 1.7. The geographical distribution of new housing has contributed to greater electricity consumption in the residential sector from 53% of total primary energy in 1980 to 69% in 2009 as more homes come with electricity-intensive heating and cooling equipment installed. The percentage of new single-family homes with air conditioning has increased from 62% in 1980 to 79% in 1995 and 88% in 2010. Recently, heat pump heating systems have also gained market share, from 23% in 2001 to 38% in 2010. Similarly, in non-residential sector, space heating, lighting, and space cooling are the top three end uses representing around half of site energy consumption as shown in Figure 1.8.

![Energy Consumption Breakdown](image-url)
Figure 1.6: Building site energy consumption by end use in the USA

Figure 1.7: Residential site energy consumption by end use in the USA

Figure 1.8: Non-residential site energy consumption by end use in the USA

The breakdown of cooling and heating energy use by equipment type in the USA for commercial buildings [9] is shown in Figure 1.9. Packaged air conditioning systems are leading with about 54% share of cooling energy. Central systems are mostly representing the remaining segment of cooling energy with centrifugal and reciprocating chillers. Similarly, the equipment breakdown for heating energy is shown in Figure 1.10. Packaged units along with boilers, furnaces, and unit heaters are the key elements for heating energy.
In EU electricity consumption has continued to grow in the last decade despite numerous energy efficiency guidelines and programs at EU and national level. Total electricity consumption in the residential sector in the EU-25 has grown by 10.8 % in the period 1999-2004 [10]. Electricity demand is increasing due to various reasons. One of the key factors is increase in use of electric appliances maintaining comfort conditions in the buildings. More single family houses, each with some basic electrical appliances, alongside larger houses and apartments results in more lighting, more heating and cooling. This adds with the older population demand for higher indoor
temperatures with all-day heating in winter and cooling in summer. The main drivers in electricity consumption and electricity peak demand in the southern EU countries is a fast growth of small residential air conditioners and their extensive use during the summer months. Total residential electricity consumption in air-conditioning for EU-25 in year 2005 was estimated between 7 -10 TWh/yr. Breakdown of electricity consumption in residential sector during 2004 is also shown in Figure 1.11. In 2007, HVAC systems were estimated to account for about 313 TWh of electricity, approximately 11% of the total 2800 TWh of electricity consumed in Europe during same year. The share of air conditioning units and chillers, fans in ventilation systems, pumps, and space and hot water heating is 0.75%, 3.34%, 1.81%, and 5.23% of total EU energy use in 2007, respectively [11].

Figure 1.11: Breakdown of electricity consumption among residential end-use equipment in EU-15, yr. 2004

The demand for air conditioning in UK buildings is also growing rapidly in response to more intensive building use, increased demands for comfort by occupants, business and market pressures and the expectation of a warmer climate. It is anticipated that about 40% of commercial floor-space will be air-conditioned by 2020 compared with 10% at the end of 1994. Annual electricity consumption by 2020 is expected to increase four times to nearly 64 TWh. Energy consumed by air conditioning systems in commercial buildings is expected to double from current levels by 2020. In a typical office, air conditioning can account for over 30% of annual electricity consumption [12].

In Pakistan, domestic consumption of electricity raised to about 30% in 2005-06. The key cause behind it was remarkable increase of air conditioning by domestic consumers. Market survey revealed that the sale figures during 2005-08 for air conditioning appliances were around two millions, adding 4000MW of load. The huge demand of air conditioning is one of the major causes of electricity crisis in Pakistan [13].
1.1.3 HVAC equipment demand

World demand for HVAC equipment is projected to rise 6.2% per year, reaching a total of $93.2 billion in 2014 as shown in Figure 1.12 [14]. Cooling equipment will continue to overtake heating equipment worldwide with increasingly hot climatic conditions.

It is also predicted that demand in the Asia-Pacific region will outpace the average global demand, rising 7.1% yearly through 2014. China will be the fastest growing national market, comprising 40% of global demand. China has become a leading producer in the room air conditioner segment, exporting products to the United States, Western Europe, and throughout Asia.

In addition, the USA and Japan are also considered major producers, each with annual shipments of more than $5.5 billion during 2007. Demand in the USA for HVAC equipment is expected to rise 8.1% annually through 2014. The HVAC equipment annual growth demand is projected at 3.0% and 4.9% from 2009 to 2014 in Western Europe and other regions, respectively [14]. In Europe, Germany and Italy were leading with more than $2 billion shipments in 2007. In EU annual sale of HVAC equipment was more than $2.4 Billion during 2009 [15].

![Figure 1.12: HVAC equipment demand and annual growth](image)

Statistical data presented for different regions of the world showed that building sector is a key contributor towards energy consumption. Keeping in view the significance of buildings sector, the European Council in 2007 adopted energy and climate objectives for 2020. The goal of saving 20% energy consumption through energy efficiency is a crucial part and the main focus is buildings sector. The minimum energy savings in
buildings is expected to generate 60-80 Mtoe/year reduction in final energy consumption by 2020.

In buildings, share of HVAC systems is dominant for the total energy consumption. For a new building, about 30% to 50% of total cost is related to HVAC systems in non-residential buildings, and approximately 5% to 10% in case of residential buildings [16]. Therefore, HVAC systems would be the vital element towards world energy savings and economics. In spite of various efforts for energy use reduction, space conditioning remains a very large portion of the total energy use and still provides substantial prospect for energy use reduction.

HVAC systems comprise on wide range of different components to define their configuration. No single manufacturer provides all components of whole system. Therefore, for efficient HVAC system, it is essential that all components perform effectively and efficiently when operate in an overall system. Thus, decisions regarding the choice and design of HVAC systems are of the utmost importance for overall energy savings.

### 1.2 Problem Statement

#### 1.2.1 Problem Background

The HVAC system configuration is a conceptual design of HVAC system including the active components, airflow set-up, and the control strategies with set points. Selection of HVAC system configuration is typically decided in the early stage of the design process. However the system configuration design has substantial impacts on the performance of the final system. The maximum opportunities for energy efficiency exist at the design stage for HVAC systems in a new building facility. It is generally cheaper to install energy efficient HVAC equipment during the building construction compared to the up gradation of an existing building with an efficient system [17].

HVAC system design is a process of decision making. Normally, it consists of four sequential stages depending on the level of detail. In conceptual design, general HVAC system selection is involved. In preliminary stage, schematic design with preliminary component sizing is completed. Detailed design includes load calculation, ductwork and pipe work layout, detailed component sizing, and control system design. Fully detailed layout drawings are finalized with co-ordination of architectural and allied services layouts during engineering design stage. Impact of decisions diminishes with the progress of each design stage and on contrary, the cost of modification increase sharply for the latter design stage along with the level of knowledge as shown in Figure 1.13. Thus, it is more difficult to make decisions in the beginning. Therefore, it is very critical to make appropriate decisions in the early stages for the overall success of a HVAC system. Good decisions in the design and operation of buildings can substantially improve their energy performance. HVAC system configuration selection during earlier
stages of design would cost less and have a more impact on final system performance [18].

Figure 1.13: Impact of decisions and cost for HVAC system design

Selection of appropriate HVAC system configuration is one of the most important tasks in the early stages of HVAC system design and extensively affects the overall building performance. The configuration of HVAC system includes the decision of system type, selection of components type and their number and the choice of control strategies. Economic and environmental aspects also need to be considered. Determination of optimal values of HVAC design and control parameters also play their roles for overall energy savings. Optimized building and HVAC design can help in savings 5% to 30% in annual energy consumption for lighting, cooling and heating [19] [20] [21]. In another study, nine measures were identified related to HVAC system that contribute 75% of total cumulative energy savings including revise control sequence (21%), reduce equipment runtime (15%), optimize air side economizer (12%), optimize supply air temperature (8%), add variable frequency drive to pump (6%), reduce coil leakage (4%), reduce/reset duct static pressure (4%), optimize start/stop (3%), and add/optimize condenser water supply temperature reset (2%) [22].

In general, the HVAC systems are mechanical systems providing and maintaining an artificial environment for either operational demand or health and comfort of the occupants. HVAC system components are connected in a certain topology to define its configuration that ensure thermal energy to be properly transferred and air to be conditioned and distributed. Many types of HVAC system configurations have been developed over the last century. The ASHRAE handbook defines typical HVAC system configurations based on different zoning, media, component arrangements, air transportation, and operation strategies [23]. However, it does not include innovative
technologies and configurations utilizing components for heat recovery, evaporative cooling, and desiccant dehumidification. Significant energy saving potential is associated with these configurations. All these component and control strategies make selection of an efficient configuration more complex. HVAC system designer is expected to select an appropriate configuration for certain design perspective to match the requirements of building load demands in a specific climate. The ultimate objective of the selected configuration is to control indoor comfort conditions with minimum cost and environment impacts. In early design stage, various alternatives can be identified and the final choice is only made among the identified solutions.

New emerging technologies based on alternative and renewable energy resources are also penetrating in the HVAC sector. The main reasons are high energy demand in building sector, increasing energy prices, and environmental issues due to CO2 emissions, and ozone depleting mediums in conventional systems i.e. CFCs, and HCFCs. Several types of solar air conditioning systems are proving relatively efficient alternatives in terms of energy saving especially in hot climate conditions. These alternative systems make the task of HVAC system selection more complex.

In practice, HVAC system configuration and design parameter’s selection is decided based on knowledge, experience, and skill of system designer through a traditional procedure. The expertise could be of an individual or group of practitioners. But they have limited knowledge, and experience based on the type of system frequently designed by them. It could be an inappropriate way in terms of time, effort and labor and may lead to sub-optimal configuration. Modern HVAC system designs and components are emerging all the time. Therefore, the task of frequently updating knowledge, experience, and skills is not simple especially when many alternatives are needed to be evaluated for same task. New HVAC system designers have to spend extensive amount of time before achieving trustworthiness. Meanwhile, knowledge sharing from experienced designers is quite tricky because experts are frequently unaware of the knowledge they have and of the way in which they use it. Additionally, experts may have different and sometimes conflicting sets of rules [7]. Therefore a systematic approach could be very helpful in this respect.

1.2.2 Problem Definition

Selection of an appropriate HVAC system configuration along with design parameters requires detailed performance analysis of diverse configurations at the initial design stage. Ideally, the selection process should consist of a detailed performance analysis for various systems and the system with the best overall performance is the logical choice. However, detailed analysis and selection of HVAC system is time consuming and complex task. On the other hand, HVAC system designers are facing exceptional pressure to complete designs quickly. Therefore, designers are increasingly turned to computer design tools and approaches to help them in this situation. Consequently,
numerous design applications have been developed for this purpose. For HVAC system configuration, simulation and optimization tools could offer the possibility to effectively carry out automated system selection.

Researchers think that it is possible to attain 70% savings by using new and better design techniques and tools during design stages [7]. Most of the savings are based on an integrated system design approach through finding the optimum interaction between various components and their constraints. Therefore, the current study proposes an automated simulation-based optimization approach to effectively and efficiently deal with the complex task of optimal HVAC system configuration and design parameters selection at the initial design stage.

Physical-modeling and simulation can play a crucial role throughout the system lifetime of a HVAC system. For instance, at (1) initial design stages, when the control strategies are decided, (2) validation of system performance, (3) commissioning, and (4) operation support. Modeling and simulation activities can be very cost-effective, both during the design and commissioning stage. At design stage, HVAC system configuration and design parameters are decided which would result in later costly interferences, and during commissioning stage, when the savings in terms of reduced down-time can be significant [24].

The significance of modeling and simulation studies is especially high when designing innovative systems, like solar air conditioning. In such cases past experience is of modest or no value in understanding the operational performance. Therefore, in view of complexity of the task involving evaluation of many configurations, modeling, and simulation is a reasonable choice for HVAC system configuration selection at the initial design stage.

An appropriately implemented optimization method along with modeling and simulation could be able to find the optimal or near-optimal result. The design of HVAC system configuration is formulated as an optimization problem to automatically find the optimal configuration. Optimization is typically implemented on the detailed design to find optimum values of the identified parameters for a given system. Control optimization is one such example where the objective is to find optimal parameters and set points of a particular controller or control strategy in a given system. In a more complex example of air conditioning system sizing, both dimensions and mechanical structure of the coils and the fan, and the control set points like water flow rate and fan speed can be taken as problem variables [25].

The key difference of the configuration problem to the sizing problem is that the configuration of the HVAC system is not predefined. Once the configuration is decided, the typical sizing variables associated to the particular configuration can be identified and searched. Therefore, the configuration problem is a multi-level optimization, at system configuration level and design level. Iterative and simultaneous approaches are
being used for solving multi-level problems. The iterative approach deals each level of problem with a dedicated search method. The overall design problem is solved by nested optimization loops. The approach could use different optimization algorithms at each level that results better performance in term of convergence and robustness in finding an optimal solution. However, overall progress of the approach is generally slow for multiple configurations. On the other hand, the simultaneous approach uses a combined optimization process to search in both configuration and parameter space at the same time. The advantage of the simultaneous search in the two spaces may result in faster process than searches performed in each space separately [18]. In brief, simulation-based optimization approach can be used to solve HVAC system configuration problem.

1.3 Scientific Approach

1.3.1 HVAC system modeling and simulation

Keeping in view of problem complexity and advantages of modeling and simulation, the current study uses an equation-based object-oriented approach. In this approach, physical models of HVAC systems are developed. The overall system model comprises of different sub-models. Each sub-model represents a certain physical component of HVAC system e.g. a chiller, a pump, or cooling tower. Model of each component is described by a set of algebraic, differential, and event-triggered difference equations [24]. The equations are based on thermodynamics principles depending on physical component type to describe its behavior under various input conditions. The interaction of a component with other components is defined using ports, called connectors. Aggregate HVAC model(s) are defined by connecting sub-models through their connectors. For example, fluid connectors carry pressure, flow rate, and specific enthalpy, and heat connectors carrying temperature and heat flux. Any two components with compatible connectors are connected together, regardless of their internal details. The component models are reused properly to form an overall HVAC system capable of evaluating various alternative systems. Overall HVAC model accounts for the load profile and weather data as inputs.

Overall HVAC system is modeled to perform optimization at both system configuration and design level. Optimization at system design level involves variation of few key design parameters of sub-components along with system configurations at the same time. For example, mass flow rate, temperature difference across chiller evaporator and condenser, and cooling tower fan speed. This would provide the optimal system configuration along with optimal values of design parameters for overall optimal system design.

It is ensured that the developed model meets the building load demand which is provided as input to the overall model. Sub-component models of overall system model are conditionally added or removed based on building load profile to simulate various
HVAC system alternatives. Initial values of design parameters for sub-components are based on international standards defining typical values for specific components. For example, Air-Conditioning and Refrigeration Institute (ARI) standard 550/590-2003 defines typical design values for air and water cooled centrifugal chiller [26]. Similarly, Cooling Tower Institute (CTI) describes typical design values for cooling towers [27].

For design optimization, the key system design parameters are also incorporated in the complete model with suitable variation space. The overall developed HVAC system model is adequately constrained to simulate various feasible system configurations and design parameters.

1.3.2 Automated Simulation-based Optimization

An automated simulation-based optimization with simultaneous approach is implemented through coupling HVAC system model with an optimization algorithm. Appropriate selection of optimization algorithm is very important for the effective interaction with overall HVAC system model. Generally types of problem variables are the deciding factors to choose optimization algorithm. Few problem variables related with system configurations are discrete in nature, for instance number of chillers or cooling towers, and others are continuous, for instance temperature difference across chiller evaporator or condenser, and cooling tower fan speed.

In the optimization algorithm the cost function and constraints are defined in terms of limits on the system configurations and design parameters. Cost function can be the total system energy consumption, system efficiency, or system annual operation cost. The optimization algorithm starts the simulation of HVAC system model based on problem variables that will define all possible variations with respect to configurations and design parameters. The value of objective function is determined at the end of each simulation of a specific configuration. The optimization algorithm provides the optimized configuration and design parameters at the same time using all the possible alternative configurations of overall HVAC system model. The system configuration and design parameters with minimum value of cost function are considered as optimal. Then, optimal system configuration is decided based on the optimal values of problem variable and cost function. The implemented approach is depicted in Figure 1.14.
1.4 Research Aims and Objectives:

The objective of the research is to develop and validate the methodology of automated simulation-based optimization for design optimization of HVAC system configurations. The aims of the work are:

- To define the concept of configuration for HVAC systems; to give the overview of various HVAC system configurations including conventional and innovative technologies; to analyze the impacts of configuration design on the energy performance (Chapter 03);
- To identify the criteria for automatic system selection; to identify the feasible simulation and optimization tools for the research approach; to give overview of tools and methods used for the research approach; to identify criteria for suitable selection of optimization algorithms for solving configuration problems; to identify optimization variables for the configuration problem; to define the optimization objective function and constraints (Chapter 04);
- To develop a desiccant wheel component model capable to estimate the wheel performance in dehumidification and enthalpy modes for commercial
applications; to calibrate and validate the model under the transient measurements (Chapter 05);

- To develop an incremental simulation-based optimization methodology for chilled water system design optimization; to validate the proposed approach for the selection of optimal HVAC system configuration through a real chilled water system (Chapter 06);

- To modify the ideal component models of desiccant cooling system to match with real system operation; to validate the modified component models; to develop a desiccant cooling system model through validated component models based on ENERGYbase system design; to validate the system model under the transient measurements (Chapter 07);

- To develop multifunctional modeling approach for desiccant cooling systems; to develop desiccant cooling system models for different configurations; to evaluate five configurations of the desiccant cooling system in five different climate zones; to select an appropriate system configuration in a certain climate zone (Chapter 08);

- To summarize the research, draw conclusions, and provide directions for the future research (Chapter 09);

### 1.5 Scientific Contribution

The scientific contribution is the development of a systematic simulation-based optimization approach for the effective and efficient evaluation of various HVAC system alternatives to automatically select the optimal configuration at the initial design stage. The approach could be implemented at both system design and configuration level. Additionally, the approach would minimize the limitations of knowledge and experienced-based system selection in terms of designer’s expertise for specific systems under certain climate conditions.

In the thesis, equation-based object-oriented (EOO) modeling and simulation approach is used to explore its prospects in the analyses of HVAC system configurations. Two methods are presented in the developed approach with respect to HVAC system modeling in Dymola/Modelica environment. The method of conditional declaration of component models can be used for automated selection of optimal system configurations by coupling system modeling tool with the optimization tool through an appropriate algorithm. While the in the method of redeclaration/replaceable component models empty component models are introduced with only connection ports and without heat and mass transfer that would enable multifunctional model-based performance analysis of different HVAC system configurations.

The developed methods are implemented for optimization of a real chilled water system installed in Symantec Corporation building in South California, USA and desiccant cooling system operating in ENERGYbase building in Vienna, Austria. The first
method implementation presents an incremental approach for chilled water system design optimization that proved efficient in terms of time and labor for evaluating system configuration. The overall optimization of chilled water system at system design and configuration level with fixed building load demands resulted with the power consumption reduction of 17-43% by selecting an optimal system configuration.

The thesis also contributes at the component level in which a desiccant wheel model is developed for performance analysis of commercial system operation. The existing models lack the capabilities to handle real-time control strategies with respect to wheel operational modes and sensor functionalities. Therefore, the developed model is modified to incorporate real-time control strategies for both dehumidification and enthalpy modes under the transient operating conditions that are encountered in the commercial systems. The model is validated in comparison to the published data and measurements from the ENERGYbase building desiccant wheel installation.

In the current work, the second method is implemented for an appropriate selection of desiccant cooling system configuration. However, the available component models related to desiccant cooling system are designed for ideal behavior. Therefore all relevant component models, such as humidifiers and heat wheel are modified to match the real system operation. The modified component models are calibrated and validated under the transient measurements obtained from ENERGYbase monitoring system. Afterwards, the validated component models are used to developed system model. Then validation is performed at the system level.

Afterwards, the validated system model is used for performance analysis of desiccant cooling system configuration. In the study, five system configurations consist of ventilation, standard recirculation, standard dunkle, ventilated recirculation, and ventilated dunkle cycles are evaluated in five climate zones: Vienna, Karachi, Sao Paulo, Shanghai, and Adelaide. The evaluation resulted that the performance of ventilated dunkle system configuration is much better in three climate zones except Karachi and Shanghai climates that are more suitable for the ventilation system configuration.

Finally, the developed approach presented the concept of ‘experience beyond the boundaries’ in HVAC system design and considerably ease the complex task of optimal selection of HVAC system configurations. Such systematic approach would be supportive for the HVAC design practitioners to evaluate various system alternatives and select the optimal system configuration and design parameters at the initial design stage under various building load demands and climate conditions. Additionally, the approach is a step forward toward the development of software systems able to synthesize new and optimal system configurations.
1.6 Thesis Outline

The thesis is composed of nine chapters. A brief summary of all chapters is presented here.

Chapter 01 gives an overview of basic idea of the research in terms of motivation based on the statistical data related to energy consumption of HVAC systems in buildings sector. Additionally, the research approach is also briefly discussed along with the basic objective and scientific contribution.

Chapter 02 provides the literature review of HVAC system optimization through numerous research studies.

Chapter 03 is related to HVAC system configurations. It defines the various types of HVAC system both at component and system level. Moreover, an overview of alternative and innovative HVAC system configurations is also presented.

Chapter 04 is focused on the various aspects of research methodology. It highlights the significance of the proposed equation-based object-oriented (EOO) modeling and simulation approach. Two methods of HVAC system modeling and simulation are discussed through examples. Finally, an overview of simulation-based optimization approach is presented with respect to coupling between the simulation and optimization tools.

Chapter 05 presents the thesis contribution at the component level in which a desiccant wheel model is developed capable of operating in both dehumidification and enthalpy modes under the real control strategies. The model is calibrated and validated through real transient operating conditions obtained from a real system installation.

Chapter 06 is a proof of concept of the first proposed methodology for automated optimal selection of chilled water system configurations. The approach is implemented on system design and configuration level on a real chilled water system operating with three chillers and five cooling towers. Three different strategies are implemented and validated for performance optimization of chilled water system.

Chapter 07 presents model development of a desiccant cooling system based on the real system installation. The available component models of desiccant cooling system are modified to match with the real system operation. The modified component models are calibrated and validated under the transient operating conditions of a real system. Afterwards, the developed model is validated at the system level using the validated component models.

Chapter 08 is related to the performance analysis of desiccant cooling system configurations. The model desiccant cooling system is developed based on the second proposed modeling and simulation approach for appropriate system selection. Such
approach can be implemented for multifunctional system modeling and simulation. Five configurations of desiccant cooling system are analyzed in five different climate zones.

Chapter 09 presents the summary of key conclusions and directions for the future research activities for further enhancement.
Chapter 02

2 LITERATURE REVIEW

Today, modeling and simulation are established techniques for solving design issues in several engineering and other disciplines. Wide range of tools is available in field of design, analysis, and optimization of system performance. Design, test, operation, and management of HVAC systems rely progressively on modeling and simulation techniques. Such techniques together with model-based analysis of HVAC systems provide an important tool facilitating the users to carry out thorough tests of the systems by emulating their performance on a computer. Similarly, numerous optimization programs are also being practiced in HVAC design problems.

2.1 HVAC system modeling and simulation

Modeling of HVAC systems is rapidly gaining more interest for system performance evaluation. Especially, at the conceptual design stage, that often requires evaluating various system alternatives to decide the best system configuration. Modeling and simulation tools for HVAC system design and analysis could be categorized with respect to the problems they are meant to deal with. For example, tools for pipe/duct design, equipment sizing and selection, energy performance analysis, system optimization, and control analysis and optimization. Each tool has its own limitations and could only be applied for a certain range of applications. Therefore, available tools are not fully suited for modeling and simulation of all relevant aspects and possible design analyses. In general, HVAC modeling approaches can be categorized into three classes as, modeling approaches for HVAC components, modeling approaches for HVAC control and modeling approaches for HVAC systems. Different HVAC system modeling approaches required different levels of user skills, modeling resolutions, and user customization capabilities. To solve different models, simulation tools have also been seen as promising solutions for establishing the baseline performance prediction which can be used during initial design stages of HVAC systems. Solution techniques for HVAC system simulation model can also be classified as, simultaneous modular solution, independent modular solution, and equation-based solution using manipulation [28].

However, HVAC modeling and simulation is relatively complex from a user and developer point of view. For a user, the complications grow with the level of explicitness due to increasing requirement of user knowledge of HVAC system and the number of system definition parameters. But the availability of data pertaining to those parameters from manufacturer is decreasing and analyses have become more complicated. Similarly, the difficulties increase with the explicitness and detail for a developer. This is due to the interactions among the components of the HVAC system or HVAC system with the building. It is important that when system simulation is used
for building performance evaluation, the building aspects in terms of load demands and climate conditions should be taken into account by modeling the overall system using an integrated approach [16].

Despite of frequent improvements in HVAC system modeling and simulation tools/approaches, researchers believe that a lot of work is still to be done. New modeling techniques are being introduced enabling re-use of component models and concurrent coupling of programs at run-time level.

In the recent years, various modeling and simulation approaches have been extensively used in different research activities for HVAC system performance analyses. Several studies were related to HVAC modeling at component, control, and system levels. At component level, models of air and water cooled chillers were developed in TRNSYS to analyze their performance with various control strategies [29] [30]. Similarly, simplified models of cooling coil unit [31] and cooling tower [32] were developed for control and optimization of HVAC systems. In another study, component models of axial fan, air filter, and duct for a ventilation unit were developed in Simulink to analyze the performance of the constant airflow control scheme [33]. At system level, a combined building-HVAC system model was presented including models of building zone and HVAC equipment. The model was developed in Engineering Equation Solver (EES) and showed its utility for the energy audit of commercial buildings [34].

Modeling and simulation approaches are also quite supportive for performance analyses of innovative and alternative HVAC systems utilizing low grade energy such as solar air conditioning and desiccant cooling systems. Desiccant wheel is the key component of desiccant cooling systems. Therefore, several desiccant wheel models were developed for performance evaluation of desiccant and solar cooling systems. A desiccant wheel psychrometric model was developed to predict the performance of three types of desiccant wheels manufactured by using different kind of solid desiccants [35]. Moreover, a desiccant wheel modeling and simulation approach was presented in Simulink for parametric study of desiccant wheel. The modeling solutions were used to develop simple correlations for the outlet air conditions as a function of physically measureable input variables [36]. In another study, simulation models of hybrid HVAC system were presented combining conventional vapor compression system and desiccant cooling system. The models were used to analyze the electricity reduction potential of hybrid system [37].

Currently, solar cooling and heating is one of the most effective alternative technologies for HVAC systems. Till 2007, there were 81 installed large-scale solar cooling systems with a cooling power over 20kW around the world, 73 in Europe, 7 in Asia (China is particular) and 1 in the USA (Mexico) [38] [39]. Around 60% of the installations are dedicated to office buildings, 10% to factories, 15% to laboratories and education centers, 6% to hotels and others at buildings with different final use (hospitals, canteen, sport center, etc.). Absorption chillers are used at 56 different installations, 10 chillers
are adsorption based and around 17 Desiccant Evaporative Cooling (DEC) systems. Amongst all DEC installations only two systems use a liquid regenerator. The overall cooling capacity of the solar thermally driven chillers amounts to 9 MW out of which 31% is installed in Spain, 18% in Germany and 12% in Greece [39]. Numerous research activities involved modeling and simulation approaches for performance analysis and optimization of different solar air conditioning systems. In this respect, a computational model of solar ejector cooling system was developed in TRNSYS to perform hourly simulation. However, the TRNSYS library does not include an ejector cooling cycle component; therefore the component model was developed in EES and then coupled with TRNSYS for system analyses [40]. Furthermore, a modular dynamic simulation model of solar cooling system was developed and validated with real data. The model can be adapted in other solar cooling systems [41]. Similarly, dynamic modeling and simulation approach was used for evaluation of new design of solar air conditioning prototype [42]. Matlab-Simulink platform was also used for modeling and simulation of solar-powered desiccant regenerator for open absorption cooling cycles [43]. In another study, TRNSYS was used as a design tool to develop an integrated transient simulation program for simulating the Iraqi solar cooling system. The simulation is modeled for other virtual solar cooling systems [44]. One widespread application of a solar-powered system is for absorption cooling which is an alternative approach to cooling that is largely thermally driven and requires slight external work. Several absorption solar cooling systems were also modeled and simulated with TRNSYS for performance investigation [45] [38] [46] [47]. Beside solar energy, utilization of wind energy for space heating and cooling had also proved feasible at small scale by coupling wind turbine with HVAC system [48].

It can be concluded from above cited research activities that TRNSYS is effectively used as modeling and simulation tool for energy performance analyses of HVAC systems. It is generally acknowledged as one of the most accurate and reliable tool for the simulation of thermal systems. In fact, its library consists of a large number of validated component models for simulating most of the common thermal equipment [49]. However, the selection of a suitable modeling and simulation approach is not a simple task. Various factors are needed to be considered in the selection process. Especially detailed knowledge about the HVAC system is required in terms of model parameters for a comprehensive analysis. In addition, the computational requirements for such analyses are becoming more intensive with the analysis of the results getting more complicated. In general simple modeling approaches are currently used in most of the design analyses that do not require detailed system modeling and simulation. However for evaluating different control strategies and comparing HVAC system alternatives detailed HVAC system models are required. In the system-based modeling approach, the speed of system alternatives evaluation is much higher compared to the component based modeling approach, however the investigation of innovative technologies is limited [28].
System-based modeling approaches are categorized as: pure conceptual system modeling approach, system-based modeling approach, component-based modeling approach, and component-based multi-domain system modeling approach. Comprehensive analysis of these approaches and the tools based on these is presented in the study [28]. At the system level, the simulation method normally comprises solving a set of nonlinear equations that can be organized in different ways. Equation-based object-oriented simulation and sequential/procedural modular simulation are the two leading methods used in engineering. The system as a collection of modules is represented in the sequential modular simulation. Thus, the equations, representing each subsystem or piece of component, are coded so that a module may be used in isolation from the rest of the system. TRNSYS and DOE-2 are based on sequential approach [50]. The equation-based object-oriented simulation approach combines all the equations in a large system of non-linear equations providing flexibility in implementing an efficient optimization algorithm. The equation-based system modeling approach has evolved from the need to improve the existing building performance tools. This modeling approach represents the case where a system is represented by a basic modeling unit that is practically “smaller” than a component and in the form of an equation or a low-level physical process model. Examples of equation-based object-oriented tools include SPARK [51], EKS [52], NMF [53], IDA [54], SimScape [55], and Modelica [56]. Models developed with these tools cannot be executed directly. Therefore, a model needs to be transferred into a programming language where it can be complied. Different techniques are employed to reduce the dimensionality of the linear and non-linear systems defined in the model in order to increase the execution efficiency of the complied program. The features associated with equation-based object-oriented simulation tools are: input-output free, modular, hierarchical, universal, separation of physics from numerical solution algorithms and faster developments of simulation models [28]. Therefore, the tools are being acknowledged as a suitable alternative for modeling and simulation of HVAC systems as compared to traditional tools. But still, all capabilities of this approach are not yet explored. A model of an autonomous solar desiccant cooling system was developed in SPARK for parametric investigation [57]. In another study, Modelica/Dymola was used to develop desiccant air conditioning model due to limitations of other tools to handle discontinuities in the sorption isotherms [58]. Moreover, object-oriented dynamic modeling library was also implemented to develop absorption refrigeration systems with Modelica/Dymola. It was determined that object-oriented modeling techniques allowed considerable reduction of the model development time through model reuse [59].

2.2 HVAC design optimization

Optimization techniques have been extensively studied and practiced on HVAC design problems. The automatic simulation-based optimization, especially at HVAC system configuration level is a new concept.
HVAC design optimization problems can be classified into two types. The first type is optimization of static design parameters and the second type is optimization of the dynamic input variables, which usually comprise control scheduling and set points. The static variables are generally system design parameters that are fixed in each simulation. Design of building envelopes, HVAC system and components, ductwork and hydraulic systems, lighting is included in such type of problems [18].

HVAC model-based optimization approach is extensively studied during the last decade. An optimization of thermal performance of a building with ground source heat pump system was performed through TRNSYS to reduce building heating and cooling energy costs [60]. The developed models of cooling coil unit and cooling tower were also used for real time control and optimization of HVAC systems [31] [32]. Similarly, global optimization of overall HVAC systems was performed using developed component models [61]. Another optimization study was performed for optimal water-cooled chiller and cooling tower combination. Cooling tower approach and design wet bulb was determined as key parameters along with condenser water flow rate for optimal performance and to improve system life cycle costs [62].

Optimization research activities were mainly focused to the second type of optimization involving optimization of HVAC control strategies. Different tools and strategies were introduced for control optimization. For example, in a study, optimization of chiller-water condenser-pump system was performed in two steps. In the first step design conditions were optimized and in second stage control logic was optimized to take full advantage of equipment in the chiller system. It was shown that 5% to 8% improvements can be obtained by switching the basic controls to advance controls [63]. A tool called QuickControl was applied to perform the complex control simulations and 60% savings were predicted [64]. In addition, an integrated dynamic HVAC simulation process was proved as a viable practice for control optimization to improve the thermal management of buildings through efficient system control [65]. In another study, a parametric analysis technique was presented to optimize the control sequence of chilled water plant. Theoretical Optimum Plant Performance (TOPP) model was applied to find the optimal control [66]. Similarly, a load-based speed control was introduced for the cooling tower fans and condenser water pumps to achieve optimal system performance. The control strategy resulted in about 5.3% savings of annual system electricity use and 4.9% of operating cost [30].

Genetic algorithms (GA) are also extensively used in simulation-based optimization techniques for HVAC systems. A comprehensive review of basics of GAs and their applications for optimization of design and control of HVAC system is presented [67]. GA was developed to optimize building envelope and HVAC system parameters. The aim was minimization of life cycle cost of a detached house [68]. Similarly, a district cooling system was optimized using GA. The case studies showed that the method was effective to give optimal or near-optimal solutions [69]. A simulation-based
optimization approach using robust evolutionary algorithm also provided effective system optimization for HVAC energy management [70][71]. In another study, a modified GA was proposed for optimization of condenser water loop of centralized HVAC systems [72] and global optimization of overall HVAC systems [73]. The global optimization model for the overall control of air conditioning system was developed using decomposition-coordination algorithm for minimization of energy consumption [74]. A multi-objective optimization methodology was presented based on combination of Artificial Neural Network (ANN) and multi-objective GA. Results of the optimizations showed significant reduction in term of energy consumption as well as improvement in thermal comfort [75]. Similarly, ventilation system design and operation was optimized by simulation-based optimization approach using computational fluid dynamics (CFD) techniques in conjunction with GA [76]. A concept of integrating neural network (NN) and GA was presented for the optimal control of absorption chiller system [77].

In another study, a systematic methodology, termed complete simulation-based sequential quadratic programming (CSB-SQP), was presented for determining the optimal control of HVAC systems [50]. An optimization was performed on five different control variables to minimize system power consumption. The analysis was based on a combination of a realistic simulation of a direct expansion air conditioning system and direct numerical optimization technique. Significant savings were anticipated over conventional control strategies used in packaged direct expansion method [78]. An integrated energy optimization model of a HVAC system was developed by a data-driven approach and solved by a particle swarm optimization algorithm. The optimization results showed 7% reduction of energy consumption [79]. Another system optimization with dynamic neural network resulted in 30% of energy savings [80].

Simulation-based optimization approach is also being instigated for solar air conditioning systems. A simulation-based optimization approach was used for a solar assisted desiccant cooling system using evolutionary algorithm [81]. In another approach, central composite design method was used for system optimization. The optimization method was applicable to different types of solar cooling systems [38]. Optimization of absorption based solar cooling system with interior energy storage was performed by developing a model in TRNSYS [82]. In another study, optimal values of collector slope angle, pump flow rate, boiler thermostat setting, storage tank size, and collector area of a LiBr solar system were determined [46]. A review of various stochastic techniques used for the optimization in solar systems was summarized in a study [83].

Optimization tools could also be used effectively in conjunction with HVAC modeling and simulation tools. In the combination, a set of parameters is optimized according to a given cost/objective function in multiple simulation runs. GenOpt, generic optimization
program is an example of such tools. Few research activities utilized the approach by coupling a simulation and optimization tools for HVAC system optimization. A combination of the building energy simulation software EnergyPlus and GenOpt was efficiently applied for optimization of energy consumption in buildings with hydronic heating system [84]. In another study, building performance simulation program IDA ICE was coupled with GenOpt to find optimized values of five design parameters of building and HVAC system for minimization of life cycle cost. The optimization resulted 23-49% reduction in the space heating energy [85].

2.3 Modeling, simulation, and optimization of HVAC system configurations

Optimizing HVAC system configuration is always an interest of system designers. However, the configuration problem for HVAC system design is a multi-level problem that includes both structural and design parameters. Selection of optimal configuration of HVAC systems, especially in automatic way, is seldom studied. The task is reasonably complex in terms of time and effort because it requires evaluation of different system alternatives at the initial design stage. The emerging modeling, simulation and optimization techniques could help to handle the task.

In a study, a simplified model was developed for evaluating chiller system configurations. A multiple chiller system consisted of two-ten equally sized chillers was analyzed. It was concluded that the energy efficiency of multiple-chiller system increases with a higher number of chillers, and the maximum saving was estimated to be 9.5% [86]. An air-cooled chiller system model with maximum six chillers was developed in TRNSYS for performance assessment. Four design options in terms of the number and size of chiller were evaluated. The valuation showed that electricity savings of 10.1% can be achieved with six chillers of different sizes instead of four equally sized chillers [87]. In another study, four configurations for the heating system with a heat pump, condensing boiler, conventional boiler, and solar collector were analyzed. Solar collector-based heating system showed highest energy performance for the considered case study [88]. An optimization of solar cooling system was performed considering three different configurations in terms of number of heat pumps with different cooling powers [82]. Similarly, three different configurations of solar heating and cooling system with LiBr-H2O absorption chiller and evacuated tube collector were investigated. The first configuration was designed for the maximum cooling load using an electric chiller as auxiliary cooling system. The second configuration was similar to first but the absorption chiller and the solar collector area were sized to balance only a fraction of the maximum load. Finally, in the third configuration, no electric chiller was used and a gas-fired boiler was used as back-up. The simulation model was developed in TRNSYS for the detailed optimization of their energy performance. The results of the optimization suggested that the first configuration was able to achieve the best energy performance [89].
One of the most comprehensive studies was conducted to find optimal HVAC system configurations by evolutionary algorithm [18]. However the scope of the study was limited to the configurations of secondary HVAC systems. In the study, three significant factors were identified towards the energy aspects of secondary HVAC system configurations. These are: (1) the ability to minimize outside air load, (2) the ability to eliminate simultaneous cooling and heating, (3) the availability of inter-zonal air flow. The performance of ten two-zone system configurations were analyzed for a number of operational conditions including single duct, dual duct, and fan-coil-based systems. The configurations were modeled as a set of ideal psychrometric processes linked by airflows. It was found that fan-coil-based configurations perform better than other configurations [90]. In continuation of the study, a model-based optimization procedure was developed using GA and afterwards, applied on the two-zone optimization problem. It was determined that the search algorithm was not fully reliable especially when the component set, system topology, and system operation were simultaneously optimized. However if the component set is fixed as a boundary condition, the algorithm was able to find a feasible HVAC system configurations with 81% probability [91]. The experimental results of the proposed approach revealed that it was possible to synthesize near-optimum HVAC system configurations. However, in order to find reliable solutions, the algorithm requires multiple runs [92]. In another study, a method was proposed for choosing the best possible HVAC systems in new and existing buildings. The method used a combination of multi-criteria decision-making tool and building simulation tool to compare the six different HVAC systems. HVAC system was modeled by coupling TRNSYS and COSMIS simulation tools. Multi-criteria method Electre III was applied for decision making [93].

2.4 Summary

It can be concluded from the literature review that several modeling and simulation tools based on various approaches are being implemented for performance analyses of HVAC systems. Available tools and approaches have their own prospects and constraints. Procedural modeling environment is mostly used in various studies. Equation-based object-oriented is an emerging modeling and simulation approach. Only few studies have utilized the approach for HVAC system simulation and optimization. However optimization of HVAC system configurations has not yet been achieved. Therefore, there is need to apply the equation-based object-oriented approach because it offers significant benefits in terms of model development time, model reuse, and hierarchical model construction while handling the complexity of large systems. The model development duration often dominates the time that is ultimately spent for conducting numerical experiments.

It can be noticed that most of the HVAC systems optimization studies were focused towards control strategies or optimization of a specific system or a component. Optimization algorithms, mostly genetic algorithms, were extensively used. Only few
research activities were related to the optimization of HVAC system configurations. Therefore, it is important to further investigate the optimization of HVAC system configurations, especially at the initial design stage.

The main focus of the current research is to develop a systematic approach for automated optimal selection of HVAC system configuration. Here equation-based object-oriented modeling and simulation approach is used to further explore its prospects in the analyses of HVAC system configurations. This work also focuses on the coupling between simulation and optimization tools for automatic selection of optimal HVAC system configuration.
Chapter 03

3 HVAC SYSTEM CONFIGURATIONS

The HVAC system provides the year-round control of the indoor environment to maintain specified conditions for a certain application. The configuration of HVAC system is a schematic design that determines the type, number, arrangement of different components and the operational strategy to meet the building comfort requirements. The arrangement includes the connection between the components, supply of external sources, and assumed control scheme. The topology of how the components are connected is crucial for the system performance. Each component in the configuration is dedicated to perform a certain task of the overall system. However, the design of HVAC system is also driven by a number of different criteria, such as costs, energy consumption, indoor air quality, and thermal comfort.

3.1 Classification of HVAC systems

HVAC systems can be classified in two major categories, such as conventional and innovative/alternative systems. Conventional systems are generally electrical driven and based on CFCs and HCFCs refrigerants. In view of increasing energy demands and environmental problems associated with conventional systems, innovative/alternative systems operated with renewable and low grade energy resources are proved as obvious choice for achieving comfort conditions. In general, HVAC systems can be classified in different ways. For example, velocity-based classes of HVAC systems are: Low-velocity (low pressure) systems, and High-velocity (High pressure) systems. In low-velocity system, the air flow velocity in ducts is approximately between 2-8 m/s and pressure drop (external) is about 500-2000Pa. Duct shape in such systems are normally rectangular with aspect ratio of 1:2 to 1:4.5. These systems are generally useful in hotels, theaters, museums, concert halls etc. Similarly, in high-velocity (high pressure) system, the air flow velocity in ducts is 10-30 m/s and pressure drop is about 1500-3500Pa. Round duct shape is normally used in these systems. These systems are normally used for business and office buildings and also in the buildings having limited space for ducts [94].

Mostly, HVAC systems are categorized based on the fluid media used in the thermal distribution system. Three types of such HVAC systems are:

- All-air HVAC systems
- Air-water HVAC systems
- All water HVAC systems
- Refrigerant-based HVAC systems (Unitary systems)
Each type of the systems has certain technical and economic benefits. Few are better than others for specific applications. The generic classification for centralized systems is shown in Figure 3.1 [95].

![Diagram of Generic classifications of centralized air-conditioning systems](image)

**Figure 3.1: Generic classifications of centralized air-conditioning systems**

### 3.1.1 All-air systems

An all-air system delivers both sensible and latent cooling, and possibly humidification. The systems are often used in the buildings that require individual control of multiple zones, for example, hotels, hospitals, schools, and office buildings. These systems can be further classified as: single/ double duct systems, and multi-zone systems with constant or variable air volume. Single duct normally consumes less energy than dual duct system. However, variable volume systems are more efficient than constant volume systems. The major benefit of these systems is the greatest potential for use of outside air for ‘free’ cooling. Heat recovery systems can also be easily integrated into main air-conditioning units. Their disadvantages include the additional requirement for duct space and higher installation and operation costs [95].

### 3.1.2 Air-water systems

In air-water systems both air and water are distributed to terminal units installed in the zones to be conditioned throughout a building. The air and water are cooled or heated in central HVAC system and from there are distributed to air-conditioned spaces. Air-water systems are typically categorized in three types, such as induction system with two or four pipe systems, fan-coil systems, and radiant panels. In such systems, it is possible to provide simultaneous cooling and control on individual zone in an economic
way through thermostats. However, the overall operation and control are complicated due to handling both primary air and secondary water. In general, such systems are limited to perimeter zones. Initial cost of these systems could be high compared to all air systems [95].

3.1.3 Water systems

In water systems, water is used in the thermal distribution system to heat and/or cool a space. Direct heat transfer process occurs between water and the indoor air. These systems can satisfy indoor cooling and heating loads except ventilation loads. The most common terminal units that are being used with chilled-water systems include fan coils, chilled ceilings and chilled floors. Such systems can also be used with systems using natural convection in combination with cooling coils such as, silent cooling and chilled beams. In water systems, the thermal distribution system required significantly less space compared to the all-air systems. Individual room control and simultaneous cooling and heating are also possible. However, these systems require higher maintenance compared to all-air systems [95].

3.1.4 Refrigerant-based systems (Unitary systems)

In such systems, cooling process takes place at a very short distance from the delivery terminals. These systems are mostly window or wall mounted and factory assembled having varying capacity and type. Examples of such systems are window, split, and packaged air conditioners. Advantages associated with such system are, simple and inexpensive room control and lower initial cost compared to central systems. But these systems are less flexible in terms of air flow rate, condenser and evaporator sizes. Power consumption per kW could be higher compared to central systems [95].

3.2 HVAC system equipment

HVAC system equipment can be categorized into the primary and secondary HVAC system equipment. The primary equipment is sometimes referred to as plants and the secondary as system. The primary equipment converts fuel and electricity and deliver heating and cooling through secondary equipment to the conditioned space. Examples of primary equipment and components include chillers, compressors, boilers, furnaces, cooling towers, thermal storage tanks, etc. Secondary HVAC system components can be further classified as, air-handling equipment, and air/ liquid distribution components between HVAC system and the building. Both types of components involve distribution and heat and mass transfer components. The distribution components are pumps, fans, dampers, valves, ducts, and pipes [28]. The distinction between primary and secondary HVAC components is shown in Figure 3.2.

Primary HVAC system equipment and components are main source of overall system energy consumption. It is observed that chillers account for most of the electricity use in
HVAC system configuration design depends on various factors, such as building heating and cooling load demands, type of primary and secondary system equipment and components, available energy resources, operation strategy, and climate conditions. In a particular HVAC configuration, various energy sources and processes govern determination of the type, size and number of HVAC equipment components to satisfy building load demand. Therefore, the selection of a particular primary or secondary equipment and component out of many existing options for a specific application is a very complex task. There are numerous types of chillers, compressors, boilers, furnaces, and cooling towers for the same design ranges that can be used along with several distribution components. An overview of available equipment and components of a generic HVAC system configuration for different requirements is presented here.

Figure 3.2: Primary and secondary HVAC Equipment and components
3.2.1 HVAC primary equipment

3.2.1.1 Cooling equipment and components

Cooling is a key factor, especially during summer, to achieve desired temperature of the conditioned space for occupant comfort. Desired indoor temperature varies depending on the specified application. But normally the summer comfort zone temperature varies between 22 °C and 26 °C [98].

Several alternatives of cooling equipment are being employed for cooling purposes. In centralized HVAC systems, various types of chiller are the key elements for satisfaction of required cooling load. Centralized HVAC systems are more complex than unitary systems and involve many subsystems and components. In building air conditioning, chillers are used to produce chilled water which is supplied to air handling components for achieving desired thermal condition of the supply air. Chillers can be categorized in two main types, such as mechanical chillers and sorption chillers. In some cases, the HVAC system may be combination of both types. Mechanical chillers include reciprocating chillers, centrifugal chillers, scroll chillers, and screw chillers. These chillers are available in different ranges in terms of the cooling capacity depending on a specific compressor type. Available capacities range of reciprocating chillers and screw chillers are from about 7 to 1600 kW and 100 to 4400 kW, respectively. However, the centrifugal packages are available for higher range from about 280 kW to over 14 MW as compared to former types [23]. The sorption chillers are divided into two general categories: absorption chillers and adsorption chillers. Absorption chillers use a solid sorbent and are mostly used in various applications. Different types of absorption chiller are available in market depending on the type of the refrigerant-absorbent combination, number of generators and type of energy source. Based on different refrigerant-absorbent combinations, the absorption chillers include NH3-H2O, H2O-LiBr, and H2O-LiCl. Based on number of generators absorption chillers are normally defines as single effect, double effect, and triple effect chillers. Various energy sources can be used to drive absorption chillers, such as, electric driven, gas fired, waste heat driven, and bio-gas driven chillers. Available range of absorption chillers is from about 50 kW to 5000 kW [99].

For decentralized air conditioning system, unitary air conditioners are mostly used, especially in residential buildings. In solar air conditioning, absorption chillers are also frequently employed along with desiccant cooling systems. The solar collector is the key component to convert solar energy to thermal energy that drives a solar air conditioning system. Three types of solar collectors are common for solar cooling, such as flat-plate, vacuum tube, and parabolic trough. Flat-plate collectors are the most common collector type that is being used. They represent about 90% of the market of covered solar collectors, excluding China. Vacuum tube collectors are more efficient as
compared to flat-plate but expansive [95]. Figure 3.3 shows few examples of available equipment options for cooling of centralized, decentralized, and solar systems [100].

3.2.1.2 Heating equipment and components

HVAC systems have year-round control that ensures comfort conditions of specified space during both summer and winter. Comfort zone changes a bit for the winter season to provide desired temperature through heating. Comfort temperature for winter is about between 21 °C and 24 °C [98]. Decision about central or localized heating equipment depends on the application. Usually, a central, fuel-fired is more desirable for heating large facilities. In small facilities, electric heating is a viable option and is often economical.

Heat pumps, boilers, and furnaces are the major heating equipment used in central heating plants. Heat pumps can be classified in two types, such as vapor compression heat pumps and absorption heat pumps. Vapor compression heat pumps include air-air and air-water heat pumps. Absorption heat pumps can be driven with different energy sources, like bio-gas, waste-heat, gas, and even with solar thermal. Geothermal heat pumps are also used for heating as well as for cooling.

In conventional system, boilers and furnaces are the key elements for fulfilling building heating demands. In general, boilers vary in type depending on different factors. For example, boilers may be grouped based on working pressure and temperature (Low-pressure, High-pressure), fuel used (Oil, Gas, Coal), material of construction (Cast iron, copper, Stainless steel), type of draft (natural, mechanical), and whether they are condensing or non-condensing. Boilers may also be classified according to shape, size, and application. Similarly, furnaces can also be categorized according to heat sources (Natural gas, Oil), combustion system (Natural draft, Forced draft), mounting (Vertical), and air flow (Up-flow, Down-flow) [23].

Solar heating is also emerged as an evident technology. Solar thermal energy is applied through solar water heating and solar air heating. Flat-plate, vacuum tube, and parabolic trough collectors are normally used for water heating. For solar air heating, flat-plate collectors are more common. Various possibilities for heating purposes are shown in Figure 3.4 [100].
Figure 3.3: Cooling equipment for conventional and innovative systems
3.2.2 HVAC secondary equipment

3.2.2.1 Air-Handling equipment and components

Packaged air-handling equipment is commercially available in many sizes, capacities, and configurations using any desired method of cooling, heating, humidification, filtration, and ventilation. Air-handling equipment is usually custom-designed for large systems (over 25 m\(^3\)/s) and fabricated to suit a particular application. A vast array of choices is available in terms of numerous component types of cooling and heating coils,
humidifiers, dehumidifiers, and filtration [23]. Humidifiers are generally classified as either residential or industrial. In centralized air systems, designed equipment differs from that for space humidification, while some units are applicable to both. Air washers and direct evaporative coolers can be used as humidifiers. Residential humidifiers include pan, wetted, and atomizing humidifiers. In industrial and commercial class of humidifiers for central air systems consists of heated pan, direct steam injection, electrically heated, and atomizing humidifiers. Types of Dehumidifiers found in air system involve mechanical dehumidifier, desiccant and enthalpy wheels. For ventilation, unit ventilators are commonly used. They consist of fan, motor, heating element, damper, filter, and control instrumentation. Ventilation may be mechanical or natural depending on building thermal requirements. Air conditioning has no meanings without efficient filtration which provides clean air to the condition spaces. Different types of filters are available in market depending on the required level of air purity. Few types of filters are: fiber glass, polyester, electrostatic, ultra-low-penetration air (ULPA) and high efficiency particulate air (HEPA) filters. Various component choices for humidification, dehumidification, ventilation, and filtration are shown in Figure 3.5 [100].

It can be concluded from Figures 3.3, 3.4, and 3.5 that a vast array of primary and secondary equipment and components choices is available. All these options would play their part to define wide range of HVAC system configurations. Selection of most appropriate equipment and components would ensure optimal performance of an overall HVAC system. HVAC system designer is responsible for considering various systems and recommending few alternatives that will meet the building load demand. It requires evaluation of several equipment and components to define optimal system. Therefore, in the initial design stage, the task of suitable equipment selection is complex in terms of the time and effort.
3.3 HVAC system configuration analysis

The overall HVAC system configuration is combination of the primary and secondary equipment and components. Various configurations of primary and secondary systems can be designed depending on the building load demands and the climate conditions. In the primary HVAC systems, the configurations are normally decided by the number, type, size and arrangement of chillers, cooling towers, and boilers or furnaces.

3.3.1 Primary system configurations

Chiller is a key component that decides the system configuration of the primary system. Many design options of chilled water system are implemented to achieve desired chilled...
water temperature. However, each option depends on flow, required temperature, system configuration and operation strategy. System configuration is an important aspect in terms of overall system performance. Here only few chilled water system configurations are discussed.

Multiple chiller systems are more common than single chiller systems. Typically, the most usual system configuration consists of two chillers, either same or different type. Multiple chiller systems can often operate with one chiller since the system load can vary over a wide spectrum. If the system is properly designed then energy required to operate a second chiller can be conserved. Chillers can be arranged in parallel or in series. In parallel arrangement, chillers can operate either with a single chilled water pump or with separate dedicated chiller pumps. In the former case, water flows in both chiller continually, whether the chiller is operating or not as shown in Figure 3.6. In this arrangement, supply chilled water temperature can be disrupted when only one chiller is operating. This may result in insufficient dehumidification capabilities or the incapability to satisfy specific loads. In such a situation, the running chiller can be reset to produce the desired supply temperature depending on the low temperature limit of the chiller [99].

![Figure 3.6: Parallel chillers with single, common chiller pump](image)

The second option operates parallel chillers with separate, dedicated chilled water pumps as shown in Figure 3.7. In such arrangement, a chilled water pump pair can be cycled together. This solves the problem associated with previous case but results in significant decrease of system water flow with one operating pump.
In series arrangement, chillers are normally piped together with a single, common chilled water pump as shown in Figure 3.8. The arrangement would solve the problems discussed in case of parallel chillers. In such condition, the flow rate through each chiller is the entire system flow, double the individual flow rate of the two parallel chillers. Chillers in series normally work very well in low flow system, where the temperature difference is above 9 °C resulting in less pressure loss [99].

The main reason of the problems with parallel chiller control is the fixed relationship between chiller and system flow rate. The problem can be addressed by decoupling the chiller system from the distribution system. Figure 3.9 shows the basic decoupled system. The method is also referred to as primary-secondary pumping arrangement. In this arrangement, separate pumps are dedicated to production (chillers) and distribution systems. The chilled water pumps overcome only the chiller side pressure drop while the supply distribution pumps overcome the distribution system pressure drops. Bypass line is unrestricted to hydraulically decouple the production and distribution pumps. Otherwise there is possibility that the system might operate in a series coupled pumping arrangement [99].
3.3.2 Secondary system configurations

The secondary systems are basically air conditioning and distribution systems including air-handling equipment, and air/liquid distribution systems for appropriate interaction between primary system and the building. Different types and arrangements of air handling equipment and distribution systems normally define the configuration of the secondary HVAC systems. ASHRAE defined various configurations of secondary systems [101]. Appropriate air conditioning and distribution through a secondary system can be achieved only by efficient design of the duct systems. Therefore, the secondary HVAC systems can be described by the type of duct system, such as single, double, and multi duct systems. After specifying duct structure, the secondary HVAC systems are further classified based on the air flow control strategy, such as constant air volume (CAV) or variable air volume (VAV) systems.

In single-duct systems, either heated or cooled air is delivered by the same duct. Single duct can be further classified based on the air flow control strategy, such as constant or variable volume systems. A typical single duct, single zone constant volume system configuration is shown in Figure 3.10
The dual-duct arrangement conditions all the air in the central equipment and distributes it to the conditioned space through two parallel ducts. One duct is carrying cold air and the other warm air. Representation of the dual-duct single-fan constant air volume system is shown in Figure 3.11. Other secondary system configurations can be found in ASHRAE handbook [101].

3.3.3 Alternative and innovative system configurations

Energy consumption for air conditioning purposes has increased remarkably during last decade, especially in the developed countries as discussed in the first chapter. In 1996 about 11,000 GWh of primary energy was consumed by small room air conditioners in Europe only. The value is expected to increase four times to about 44,000 GWh by 2020 [95]. Increased thermal loads, living standard and occupant comfort demands, and
architectural characteristics in terms of an increasing ratio of transparent to opaque areas are the core causes for higher energy demand.

Energy demands can be reduced by the implementation of energy conservation approaches. Few steps are taken towards the use of cheap cooling sources. For example, the use of heat sinks such as the outdoor air for night cooling, evaporative cooling, radiative cooling, ground cooling using earth-to-air heat exchangers etc. But these passive cooling techniques cannot fulfill the cooling demands of all building types due to their limited cooling capacity. Therefore, researchers are trying to explore alternative and innovative technologies in terms of their lower operational energy requirements and environmental benefits. Two technologies that have gained increased attention in the last decade include desiccant and solar thermal air conditioning systems.

3.3.3.1 Desiccant cooling system configurations

Air conditioning loads can be categorized into two types, the sensible and the latent loads. An air conditioning system has to properly control thermal load to maintain a desired comfort conditions. The traditional HVAC systems cool air down below its dew point in order to remove the latent heat through condensation. As the dehumidified air would be at the lower temperature than desired, therefore the air would be heated to maintain desired conditions. However, during the process the system energy requirements would increase in two aspects, 1) the energy required bringing the air from the supply temperature down to the dew point temperature, and 2) the energy needed to reheat the air from that temperature up to the desired supply temperature. The aggregate of the two energy aspects increases significantly when the sensible heat ratio of the conditioned space is low [102].

The desiccant cooling systems can be either a perfective supplement to the tradition HVAC systems to avoid the additional requirements, or an alternative for assuring more energy efficient air conditioning. In addition, such system can be operated by free energy resources like solar energy and waste heat. The desiccant cooling systems use sorption air dehumidification, whether with the help of solid desiccant material or liquid desiccants. These systems can be either combined with the traditional system or used alone to produce condition air directly. The components of such system are generally installed in an air-handling unit and are operated according to the operation mode that is decided by outdoor air conditions. Such systems are based on the physical principle of evaporative and desiccant cooling. Direct and/or indirect evaporative cooling processes are used to achieve desired air conditions.

Several configurations of desiccant cooling systems are being designed with either solid or liquid desiccant material for different climate conditions. Typical desiccant cooling system comprises of desiccant wheel, enthalpy wheel, evaporative cooler, heater, supply and return fans, and filters. Few configurations with solid desiccant are presented here.
However, the choice of the components is strongly dependent on climate and building load requirements.

Desiccant systems can be mainly classified as commercial desiccant systems, all-desiccant systems, and hybrid desiccant systems. A typical desiccant system designed for a commercial building includes desiccant wheel for humidity control and a conventional vapor compression cooling system for temperature control. In all-desiccant systems, desiccant wheel is used with a rotary heat exchanger to form a complete air conditioning system. Air is dried by the desiccant wheel and then cooled by the heat exchanger. Such system configuration are useful when large amount of fresh air is needed and exhaust air can be evaporatively cooled and used for post cooling the air leaving the desiccant wheel. The hybrid desiccant systems include a desiccant component along with conventional cooling or heating coils. In addition, all-desiccant system configuration can be further divided into systems operating either with or without exhaust recovery, as shown in Figure 3.11. The exhaust air is used in Figure 3.12 to cool the process air after desiccant wheel, while in Figure 3.13 uses outside air for post cooling [103].

![Figure 3.12: Desiccant system configurations with exhaust recovery](image)

![Figure 3.13: Desiccant system configurations without exhaust recovery](image)
Desiccant cooling systems also utilize evaporative cooling for efficient energy performance. The evaporative cooling process uses the evaporation of the liquid water to cool an air stream. The heat of evaporation required to transform liquid into vapor is taken by the air and water itself. However, it adds slightly to the purchase cost of the equipment but saves on downstream cooling capacity in the rest of the HVAC system. Evaporative cooling can be used for pre-cooling and post-cooling. Major advantage to evaporative cooling is its very low operating and maintenance cost compared to the conventional cooling systems. The only cost is of water and additional air flow resistance of the components. Both direct and indirect evaporation processes are used for the purpose. In direct evaporation, the air directly comes into contact with the water. The interaction results in cool but humidified air. Whereas indirect evaporative cooling involves the heat exchange with another air stream and the water contents of primary air remain constant. The desiccant system configurations with direct and indirect evaporative post-cooling are shown in Figure 3.14 and 3.15, respectively [103].

![Figure 3.14: Desiccant system configuration with direct evaporative post-cooling](image1)

![Figure 3.15: Desiccant system configuration with indirect evaporative post-cooling](image2)
The two evaporation techniques can also be combined in a process called as combined evaporative cooling. Evaporative cooling is also used in the process side to achieve desired conditions as shown in Figure 3.16. It is worthwhile to use direct evaporative cooling in the supply side in moderate to low humidity climates. The direct evaporative cooler causes further reduction of process air temperature but also increases its humidity.

Figure 3.16: Desiccant system configuration with indirect evaporative pre and post-cooling

A hybrid and all-desiccant system with evaporative cooling is shown in Figure 3.17. Chilled water from the conventional HVAC system is used through cooling coils to further cool the air after heat recovery wheel. Similarly, in such systems, heating coils can also be used to regenerate the desiccant wheel instead of conventional heater.

Figure 3.17: Hybrid desiccant system configuration with indirect evaporative post-cooling
3.3.3.2 Solar air conditioning system configurations

Solar air conditioning systems are mostly based on solar heat-driven chillers or solar desiccant and evaporative cooling systems. The chillers produce chilled water and therefore can be used for all-air, air-water, and only water systems. A solar heat driven chiller system can be coupled with air conditioning system by providing chilled water to the cooling coil in an air-handling unit or to the water-based system. The desiccant cooling systems use desiccant materials which improve the direct evaporative cooling potential of the system under certain conditions. Solar desiccant cooling systems can be used in all-air or air-water system for primary air treatment.

Solar system configurations based on desiccant cooling can be categorized into three classes, 1) solar-thermally autonomous desiccant cooling system with solar air collector integrated as well as ambient-air designs, 2) solar-assisted desiccant cooling system with a solar liquid collector, heat storage unit and back-up heat source, 3) solar-assisted desiccant cooling system with a solar air collector (or liquid-based collector) and back-up chiller [95].

The first class systems can provide condition air to the building as soon as the sun shines. Such systems can be used as evaporative cooler only, without a sorption wheel. The solar air collector field is the system heat source, therefore heat cannot be stored. Such systems can be either operated as an integrated design or ambient-air design. An example of integrated design system is shown in Figure 3.18.

![Solar-thermally autonomous desiccant cooling system with solar air collector](image)

**Figure 3.18: Solar-thermally autonomous desiccant cooling system with solar air collector**
The second class is a very common design for a solar-assisted air conditioning system using desiccant cooling technique. In such systems, solar heat is supplied either to the heat storage tank or directly to the load. If available solar is not adequate then the back-up heat source is used as shown in Figure 3.19.

![Figure 3.19: Solar-assisted desiccant cooling system with liquid solar collector, storage tank, and back-up heat source](image)

The third category of solar desiccant cooling systems normally uses vapor compression chillers as the back-up. Such systems have two further types based on chiller operation. In the first type, the chiller is used as heat pump between the supply and the return air. It operates by lowering the temperature of the supply air and delivering the heat of condensation to the regeneration air. Therefore, this type uses a direct evaporator and direct condenser without additional water circuits. The main benefit of such system is high heat recovery since the heat pump ensures cooling of supply air and heating of regeneration air. The second type uses ambient air cooled chiller with two cooling coils integrated into the supply side as shown in Figure 3.20. In such systems desiccant wheel is intended to carry out all of the dehumidification and the chiller will then cover the part of sensible load that is not covered by evaporative cooling [95].
Figure 3.20: Solar-assisted desiccant cooling system with liquid solar collector, storage tank, and back-up chiller

3.4 Conclusions

The chapter reviewed the several configurations of conventional and innovative HVAC systems. In general, HVAC system equipment and components are categorized into primary and secondary system equipment. The primary equipment and components are the major source of energy consumption. They include chillers, boilers, furnaces, cooling towers, etc. The secondary HVAC equipment and components consist of air-handling units, liquid distribution systems, and air distribution systems. The components include humidifiers, cooling and heating coils, fans, etc. Numerous types of primary and secondary equipment and components exist with wide range of capacities.

The HVAC system designer not only has to decide the suitable type of the component for a certain outcome but also the number as well the arrangement of the components to find the best system configuration. Decision of an appropriate overall HVAC system configuration involves both the primary and secondary systems configuration. Chillers
of the primary systems and air-handling units of the secondary systems are the key elements to define the overall system configuration. In view of rising energy demands in HVAC sector, various alternative and innovative systems gained increased consideration due to their low energy requirements and environmental implications. Such systems include desiccant and solar air conditioning systems. Various configurations of such systems are established to achieve the desired comfort requirements in the different climate conditions.

Therefore it can be concluded from the numerous existing HVAC system configurations that the selection of a suitable system configurations is really complicated at the initial design stage. The evaluation of all available options requires considerable amount of time and efforts. Therefore, it is important to propose an effective and efficient approach in terms of automated system selection. In the following chapter, a research methodology is described that could help the HVAC system designers for the automatic selection of optimal HVAC system configuration.
Chapter 04

4 RESEARCH METHODOLOGY

The task of evaluating various alternatives of HVAC systems can be effectively accomplished using suitable model-based simulation and optimization techniques. Model-based system analyses are extensively used, especially in building sector to reduce energy requirements and to fix various aspects in the early design stages. In the modern era, building modeling and simulation programs require further improvements to meet new difficulties to model the real systems and to enhance the reliability and flexibility of such approaches. Therefore, several studies highlighted the new requirements for building modeling and simulation programs and also pointed out the limitations of existing programs. In the current study, an equation-based object-oriented approach built on modern concepts of computer science and software engineering is applied in view of its enhanced modeling and simulation capabilities and the complexity of the research problem.

4.1 Why Equation-based object-oriented modeling and simulation approach

Implementation of model-based simulation requires experts form different fields to develop and integrate models of various domains to analyze their interaction or to redesign components to increase system-level effectiveness or controllability. In addition, models in such practices are also required to be used beyond time-domain simulation, for example in combination with frequency domain analysis to investigate stability and design dynamic behavior or with optimization algorithms to improve the design or operation.

Application of modeling and simulation programs for such cases indicates the need of additional features for modeling and simulation tools. For instance, the invention of alternative and innovative systems requires the building modeling and simulation programs to allow an analyst to quickly add new component models and use the models within rapid prototyping processes. Furthermore, HVAC systems involve components from multidisciplinary fields. Therefore simulation programs are needed to integrate different models developed by different domain experts. In addition, the dynamic behavior of components and their interaction within a system for control designs and analysis is another requirement that simulation programs need to handle [104]. Building simulation programs should be able to present any HVAC and controls configuration that can be built in reality. Such programs are also needed to contain data for HVAC system components, their system configuration and the specifications of control algorithms. Therefore the development of building simulation programs is required to improve in two aspects. First, the existing simulation programs need to be improved so that they can support the efficient design of large scale buildings. Secondly, a modeling
and simulation framework needs to be developed to upsurge the innovation of new HVAC components, systems and control algorithms [105].

The framework needs to meet the typically functional requirements of such applications. For example, models could promptly be implemented and used by tool users. In addition, simulation program should be able to share models among users and to model continuous time dynamics, discrete time and state events. The subsystem models could be used in isolation from the overall system model for validation, for more detailed analysis, for model reduction, or for use in operation. Furthermore, simulation models could also be used in conjunction with non-linear programming algorithms that require the cost function such as energy use. It would help to efficiently solve optimal control problems that may involve hundreds of independent parameters to define the control function [106].

The software architecture of modeling and simulation environment should include several characteristic features to fulfill such functional requirements. They include, object-oriented modeling to assist code reuse, and an equation-based language to ensure more physical modeling. Moreover, another required feature is support for interfacing computational models with real experimental facilities at the component and whole building level. Additional support for hierarchical model composition would help to manage the complexity of large systems. In addition, model connectivity framework is also required to assemble models in a more real way. Application of symbolic algebra tools to reduce the dimensionality of coupled equations and numerical solvers to solve stiff differential equations are the key additional features of the desired modeling and simulation program [105].

4.1.1 Limitations of traditional building simulation programs

Traditional building simulation programs mean that are written using an imperative language, such as FORTRAN, C and C++. Few examples of such programs are DOE2 [107], ESP-r [108], and EnergyPlus [109]. Traditional programs have in general not been designed to meet the above requirements. The program designer writes causal, sorted variable assignments and implements in a sources code. In such programs, component models mostly integrate their own numerical solver and program flow logic to simulate the physical behavior that is coupled with equations. Thus, the program code is hard to maintain and difficult to add new models. Furthermore, such programs result in nested solvers which lead to considerable numerical noise in the simulation outcome and can cause complications to use optimization programs.

In traditional programs, the HVAC and control components arrangement is determined by composition rules for developing HVAC system from component models. The composition rules are not conceived according to which the real HVAC components are connected to form a system. Rather, the composition rules were defined to achieve an efficient numerical solution in the program architecture. The program architecture
distributes solvers individual components and subsystems and that does not exploit the symbolic manipulations. In view of the limitations of traditional programs and features highlighted before, it can be concluded that it is better to apply a new approach based on modern advances in system modeling and simulation, such as equation-based object-oriented modeling technique [105].

4.1.2 Characteristics of the equation-based object-oriented physical-modeling and simulation

The key characteristics of equation-based object-oriented modeling and simulation programs include efficient numerical solution, management of complexity, simulation of dynamic effects, and the use of models in combination with optimization algorithms. Efficient numerical solution depends on the computational time that describes the time needed to execute a program. In general, it is product of the number of instructions of the program, times the average clock cycles needed to process an instruction, and time the seconds that elapse per clock cycle. An application developed can influence the first two aspects, while the third term is related with processor clock [105]. Selection of efficient symbolic and numerical algorithms and software structure could help to reduce the program instructions to be executed. Equation-based languages use symbolic manipulation methods such as partitioning, tearing, and inline integration for the system of equations [110] [111] [112]. However, traditional building simulation programs use imperative model formulation that does not permit the use of such symbolic manipulations. The number of cycles required by a program to process an instruction can be reduced by using algorithms taking the advantages of parallel hardware. The software structure of traditional simulation programs lack such design capabilities due to hundreds of thousands of lines of code.

In order to handle complex building system models, the equation-based languages use object-oriented modeling paradigm [113]. It helps for composing system models hierarchically to encapsulate subsystem models, for object-inheritance to reuse existing basic models, and for object-instantiation to use and parameterize an object in a simulation model. In addition, such programs also has connectivity framework to assemble system models as connected in a real system. The connections carry physical quantities such as mass flow rate, species concentration, heat, pressure and temperature. Model-based simulation analyses require dynamic effects, particularly for feedback control loops. By averaging system performance over a fixed time step could lead to inaccurate predictions. Furthermore, in equation-based object-oriented approach models can also be used in concurrence with optimization algorithms. It is generally believed that at least for the final iterates, the simulations need to be done with high accuracy of the numerical solvers to ensure negligible numerical noise in the cost function. However, in traditional building simulation programs it seems difficult due to several solvers spread throughout their code [114] [21]. Such tools do not allow to use tight solver tolerance or to adaptively adjust them during the simulation to reduce the
computation time [115]. All the aspects indicated that equation-based object-oriented modeling permits analyzing problems that are beyond the capabilities of traditional programs. However, despite of many challenges to be overcome, it is understood that equation-based object-oriented modeling allows the building simulation field to evolve towards the next generation of tools.

In several studies, equation-based and traditional modeling languages are compared. A study reported that the developing a simulation model in the equation-based modeling language NMF was about three times faster than it was in the BRIS simulation program [116]. However the computation time in BRIS was three times faster. Similarly, the computation time of IDA ICE was compared with EnergyPlus [117]. IDA ICE required about half the computation time compared to EnergyPlus for a three zone building. Moreover, SPARK and HVACSIM+ were compared for a variable air volume flow system that serves six thermal zones. It was concluded that SPARK computes about 15 to 20 times faster than HVACSIM+ [118]. It was also observed in the literature review that TRNSYS as modeling and simulation tool was extensively used in numerous studies.

In view of the desired characteristics, the development of a suitable computational environment involves the expertise from various research disciplines, such as computer science for language design and code generation, mathematics for symbolic and numerical methods, and engineering for developing modeling libraries. Such approach is followed by the Modelica consortium to develop equation-based object-oriented modeling language known as Modelica. The development of Modelica in the building simulation community provides an opportunity to revive the efforts of establishing more modular and flexible modeling and simulation techniques for building energy and control systems. A comparison between Modelica as an equation-based object-oriented modeling environment and TRNSYS as a procedural modeling environment was presented. It was concluded that the model development time in Modelica was five to ten times faster than TRNSYS. The shorter development time also manifests itself in a four times smaller code size. However, the computation time of Modelica model was three to four time longer than TRNSYS [119]. Still, I was believed that the longer time is not an inherent feature of equation-based simulation environments [117] [118]. Therefore, in view of various advantages associated with equation-based object-oriented modeling and simulation programs, Modelica language was adopted in the current study.

4.2 Overview of modeling, simulation and optimization tools

4.2.1 Modelica- an equation-based object-oriented modeling language

Modelica language was introduced in 1997. It was developed through an international co-operative effort to define an equation-based object-oriented language for modeling and simulation of generic physical models. The model of each physical component is
defined by differential, difference, and algebraic equations. The Modelica view on object-orientation is different compared to tradition object-oriented languages such as C++ and Java since the Modelica language emphasizes structured mathematical modeling. Object-orientation is regarded as a structuring concept that is used to handle the complexity of large system descriptions. A Modelica model is primarily a declarative mathematical description to simplify further analysis. Dynamic system properties are expressed in a declarative way through equations [120] [121] [122].

The main features of the language include a-causal declarative modeling, code transparency, encapsulation and modularity, inheritance, multi-physical modeling, and reusability. A specific physical component model is based on a set of equations that define how the modeled object behaves, without explicitly defining that how the equations are solved [24]. The traditional modeling techniques are based on a causal approach. Causal modeling enforces severe constraints to the structure of each module, and how different models are connected together due to input and output nature of their interfaces. Such restrictions are not associated with a-causal, object-oriented approach. A-causal modeling is a declarative modeling style meaning modeling based on equations instead of assignment statements. Equations do not specify which variables are inputs and which are outputs. A-causal modeling allows greater flexibility and reusability for both base and derived models. The idea of declarative programming is motivated by mathematics where it is common to state or declare what holds, rather than giving a detailed stepwise algorithm on how to achieve the desired goal as in case of procedural languages. It would relieve the programmer from the concern of keeping track of such details [106]. Furthermore, the code becomes more concise and easier to change without causing errors. Modularity is one of the major motives to adopt object-oriented programs. Modelica supports equation-based a-causal connections; it means that connections are realized as equations. Models are connected together through such a-causal interfaces called ports. Fluid ports carry pressure, flow rate, and specific enthalpy and heat ports are carrying temperatures and heat fluxes. Any two components with compatible ports can be connected together irrespective of their internal details [24].

Modelica model libraries [123] are in a hierarchical structure in which more detailed models are developed from the basic models by adding specific variables, equations, or even models. In addition, it is also possible to model with replaceable fluid models. It is achieved by separating the equations of component model from the fluid model equations. Modelica also offer multi-domain modeling capability, that model components corresponding to physical objects from several different domains such as e.g. electrical, mechanical, thermodynamic, hydraulic, and control applications can be described and connected. In view of a-causal modeling, encapsulations, and inheritance, Modelica offers strong incentives towards reuse of models [124].
4.2.2 Dymola- Dynamic Modeling Laboratory

Modelica is a language for model representation. However, models written in the Modelica language cannot be executed directly. In order to create executable code, it needs to be parsed to a programmable language that can be compiled. Several commercial and freely a simulation environments to translate a Modelica model into an executable source are available that support Modelica textual and graphical modeling. For example Dymola, MathModelica, SimulationX, MapleSim, OpenModelica, Mosilab, and SimScape are included [125].

Dymola is the first Modelica environment and probably the best as it fully supports Modelica physical modeling features [126]. It is a commercial program and currently the most advanced among all of the commercially available environments for physical system modeling and simulation [110]. Moreover, it is used by several large industries like Toyota, Ford, United Technologies, Caterpillar, ABB, Alstom, TetraPak, etc. [28]. Dymola with Modelica is a high-level but general-purpose OO modeling tool and well proven in many complex applications. It is also very proficient and helpful in the modeling phase [127]. Dymola has a powerful graphic editor for composing different models. Dymola is based on the use of Modelica models stored on files. However, Dymola can also import other data and graphics files. Modelica models are composed of linear and non-linear equations. The dimensionality of such equations is needed to be reduced to achieve efficient numerical solution. Therefore, Dymola accomplishes the task through symbolic manipulations. Thus, a key feature of Dymola is the very sophisticated index reduction by the modified Pantelides algorithm. Moreover, for solving differential algebraic equations, modified DASSL algorithms are used. Dymola works through two models, one model computes the initial values and other perform the time integration. Implementation of Dymola also accomplishes automatic differentiation and finally generates and compiles C/C++ code for simulation. A time integrator can be selected before simulation from a library that consists of fourteen integration algorithms. Implicit integration algorithms with variable step size are also included which are quite helpful for stiff building energy and control systems [128].

However, the stages of translation and execution of a Modelica model are sketched in Figure 4.1 for better understanding of Modelica and Dymola interaction. The process starts with parsing the Modelica source code along with its conversion into an internal representation, normally an abstract syntax tree. Afterwards, such representation is analyzed, type checking is completed, classes are inherited and expanded, modifications and instantiations are performed, connect statements are converted to equation, etc. Such analysis and translation result in a flat set of equations, variables and function definitions. After flattening the Modelica model, all equations are topologically sorted according to the data flow dependences between the equations. However, equations if the model includes differential algebraic equations then sorting would also involves manipulation of the equations to transform the coefficient matrix into block lower
triangular form, known as BLT transformation. Subsequently, most of the equations are eliminated through an optimizer module including algebraic simplification algorithms, symbolic index reduction methods, except minimal set that eventually will be solved numerically. Finally, after generating C code, the drastically reduced set of remaining equations is solved with a numeric equation solver [128] [129].

![Figure 4.1: Phases of translation and execution of a Modelica model](image)

### 4.2.3 GenOpt- Generic Optimization Program

Numerous simulation tools are extensively being practiced in building energy system design. However, the full potential of simulation is generally not realized. Mostly the system designers guess different values of system input parameters and re-run the simulation many times to get the optimal system performance. Such way is inefficient and labor intensive. Meanwhile, the situation gets worse if the number of parameters exceeds two or three due to non-linear interactions of the parameters. However, to achieve better system design with less effort few approaches allow automatic and multi-dimensional optimization of a simulation model. GenOpt is one of such programs that automatically determine optimal parameter settings. It is completely written in Java to make it platform independent [130].

GenOpt is based on mathematical programming for determination the optimal values of system parameters through a computational procedure. Less effort is required for optimization through GenOpt as compared to preparing the simulation input. Furthermore, GenOpt can be used with any simulation program that has textual input and output, for example SPARK, DOE-2, EnergyPlus, BLAST, TRNSYS, Dymola or any user written code. Such coupling does not require any code modification except minor changes in the configuration file. The objective function, such as annual energy consumption, peak electric demand, equipment efficiency, or predicted percentage of dissatisfied people (PPD) is evaluated by a simulation program that is iteratively called by GenOpt [131].

GenOpt execution process automatically creates input files for the simulation environment. Such files are generated through input templates for the specific simulation tool. Then after launching the simulation tool, the GenOpt reads the value of object function from its result file. Furthermore, GenOpt evaluate the possible errors
occurred during simulation then defines a new combination of input parameters for the next iteration. The procedure recurrence will carry on until a minimum value of object function is achieved. Detailed information about GenOpt is available in its user manual [132].

4.3 HVAC system configurations modeling and simulation

In the current study, an automated simulation-based optimization approach is presented for automatic selection of optimal HVAC system configuration. An optimal configuration enables the system to provide high quality indoor conditions with minimum cost and environmental impact. Equation-based object-oriented (EOO) modeling and simulation approach is implemented through Modelica/Dymola and then coupling between Modelica/Dymola and GenOpt ensures the automatic selection of optimal HVAC system selection.

The HVAC system model consists of primary and secondary components. In building system simulation, ‘component model’ refers to the computer model of elementary part of HVAC systems. In Modelica, each component model is represented graphically by an icon that comprises a set of different equations. Moreover, each component model symbolizes a physical device of HVAC system, such as chiller, boiler, cooling tower, etc. with physical interface ports to properly connect them with other component models. Thus, various components are connected together through such interface ports to model an overall HVAC system.

The overall HVAC system model comprises different configurations and is used to evaluate their energy performance. The model is based on the design specifications that define the set of components and their connections. In addition, each component model of the developed model is accurately sized to meet the required building load demands. Then the possible configurations of the model are varied according to a specified criterion, to simulate their performance in terms of energy consumption.

4.3.1 Available component models in Modelica

In Modelica, various component models are defined in different libraries. Such libraries are based on multi-domain physics including models for control, thermal, electrical, and mechanical systems. In addition, models for fluid systems and different media are also part of Modelica libraries [133] [134]. Models from different libraries can be used within the same system model due to standardized model interfaces.

In current work, Modelica Standard Library (MSL) [135] and Buildings library developed by LBL [136] [106] are used to develop various configurations of HVAC systems. Beside existing libraries, several institutes, such as Austrian Institute of Technology (AIT), are also developing component models to fill the gap in terms of missing component models for building energy and control systems. The key available
HVAC component models from different free libraries to develop conventional and innovative systems are mentioned in Table 4.1.

**Table 4.1: HVAC system component models available for the current study**

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Component Model</th>
<th>Model Icon</th>
<th>Library Path</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td><strong>A. Primary HVAC Component Models</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Centrifugal chiller</td>
<td><img src="image1" alt="Icon" /></td>
<td>AIT</td>
</tr>
<tr>
<td>2</td>
<td>Carnot Chiller</td>
<td><img src="image2" alt="Icon" /></td>
<td>Buildings.Fluid.Chillers</td>
</tr>
<tr>
<td>3</td>
<td>Boiler/Furnace</td>
<td><img src="image3" alt="Icon" /></td>
<td>Buildings.Fluid.Boilers</td>
</tr>
<tr>
<td>5</td>
<td>Cooling Towers</td>
<td><img src="image5" alt="Icon" /></td>
<td>Buildings.Fluid.HeatExchangers.CoolingTowers</td>
</tr>
<tr>
<td></td>
<td>i. York Tower</td>
<td><img src="image6" alt="Icon" /></td>
<td></td>
</tr>
<tr>
<td></td>
<td>ii. Fixed Approach Towers</td>
<td><img src="image7" alt="Icon" /></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Storage Tanks</td>
<td><img src="image8" alt="Icon" /></td>
<td>Buildings.Fluid.Storage</td>
</tr>
<tr>
<td></td>
<td><strong>B. Secondary HVAC Component Models</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Desiccant and Enthalpy wheel</td>
<td><img src="image10" alt="Icon" /></td>
<td>AIT</td>
</tr>
<tr>
<td></td>
<td>Component</td>
<td>Category</td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>--------------------</td>
<td>-----------------------------------</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Pumps/Fans</td>
<td>Buildings.Fluid.Movers</td>
<td></td>
</tr>
</tbody>
</table>

The options of available component models mentioned in Table 4.1 are not as comprehensive as required to model all exiting real HVAC system configurations. The limited availability of component models was the main reason to focus on only few HVAC system configurations in the present work. However, lots of new libraries are being developed and expanded for building energy systems in last few years to bridge the gap [137] [138] [139] [140] [141].

In the present work, the focus of study is on the HVAC systems. However, HVAC system configuration analyses strongly depend on building load demands and climate conditions. Therefore, building load profile and climate conditions are considered as input variables of HVAC system models and appropriately integrated during modeling and simulation process.

**4.3.2 Inputs of HVAC system model**

**4.3.2.1 Building load profile**

For evaluation and optimization of HVAC systems, both design (peak) load and a cooling/heating load profile are significant. The building load profile defines the load variation over time and the design load describes the overall installed system capacity including chillers, pumps, piping and cooling towers. Furthermore, load profile is required especially for primary HVAC systems to design the system to stage efficiently.
Staging includes the design decisions like unloading strategy of the chillers, application of variable frequency drive on the chillers, cooling towers, and pumps, and the relative sizes of each piece of equipment. The building load profile is affected by various factors, such as the climate, hours of building or facility operation, and base loads like computer rooms [142].

The estimation of peak load demands depends on the weather conditions, building envelope, internal heat gains, and ventilation requirements [98]. For peak load calculations, load diversity is also occurs due the fact that each of the envelope, occupancy, and lighting will not peak at the same time in all spaces simultaneously. Moreover, the calculated peak load normally differs from the actual load. For example, local design conditions can be different from the weather station data, weather conditions can vary, changes may occur in the operation and maintenance of the system, and internal load can also vary over time. Several techniques are used to determine the peak load demands, such as calculations/simulations, site measurements, and rules of thumb. ASHARE handbook defines the accepted methods and procedures for load calculations [98]. Additionally, various computer tools are also developed based on one or other of such methods to calculate building load profiles, for example Trane’s Trace 700 [143], Carrier’s HAP [144], EnergyPlus [109], VisualDOE [145], etc.

4.3.2.2 Weather data profile

Weather data is a key element, both for accurate building load calculations and HVAC system analyses. Performance of primary as well as secondary HVAC systems strongly depends outside air conditions in terms of dry bulb and wet bulb temperatures, humidity, and solar radiation. The users of energy simulation programs have a wide range of weather data sources. The range varies from locally recorded data to preselected ‘typical’ years data. Several organizations, such as ASHRAE, NREL, WAATTSUN, and CEC have developed new typical weather data sets including WYEC2, TMY2, CWEC, and CTZ2, respectively [146]. Such data sets cover a year of hourly data synthesized to represent long-term statistical trends and patterns in weather data for a longer period of record. A study showed a maximum 5% difference between typical weather data sets (TMY, IWEC, and TMY2) and averaging the results of 30 years for the ten USA climates [147].

An accurate integration of the building load and weather profiles with the system model ensures the efficient analyses of HVAC system configurations. Therefore, in the current study, building load profile and weather data are properly connected with the overall HVAC system model.

4.3.3 Methods for the development of HVAC system configurations model

The main objective of the current study is to achieve automatic selection of optimal HVAC system configuration at the initial design stage. The task involves evaluation of
several HVAC system configurations. Therefore it is essential to develop a physical system model capable of varying and simulating different configurations of HVAC system. In addition, the model should also be able to vary the key design parameters of component models to achieve simultaneous optimization at both system configuration and design levels. Therefore, the complexity of such model is evident. After a comprehensive review of Dymola/Modelica, two possibilities are found to develop such system model.

4.3.3.1 First Method: Conditional declaration of component models

Overall HVAC system model consists of different sub-component models. Each sub-component model represents a certain physical model of a HVAC system, such as chiller, boiler, cooling tower, pump, etc. Overall model is developed through precise connections among such sub-component models through connectors from their respective interface ports.

In the first method, sub-component models are conditionally declared in the overall system model. The conditional declaration is based on the building load demands and/or weather conditions. However, the overall model should remain balance in terms of equal number of equations and unknown variables during the whole process otherwise the model would end up with an error. In Dymola/Modelica, the method is applied through a Modelica function and a package, known as ‘readRealParameter’ and ‘ExternalData’, respectively. The former implements any function that read the value of a parameter from an external file. The function opens the file and reads the lines of the file and then assigns a specific value to the parameter. Meanwhile, the package enables Modelica reading and writing data from such external text files.

For model development, the function has to be imported in the overall HVAC system model from Modelica standard library. Moreover, in the overall model, different characters are needed to be defined as parameter along with the name of the external file in which such characters are defined and assigned appropriate values. Such characters can be the HVAC system configuration parameters and design parameters. The external file is needed to have same configuration and design parameters with suitable assigned values according to the desired requirements for the component model variation. Subsequently, for conditional declaration of sub-component models, such configuration parameters are required to be used correctly with each sub-component model with ‘if’ statement, followed by an appropriate logical condition. Then, the declaration of a specific sub-component model is decide after comparing the value of specified configuration parameter in the model with the value mentioned in the external file. If the value in the external file is true according to the logical condition defined in the model, then that component model would be considered as a part of overall model otherwise it would be automatically removed along with its connections during the model simulation. Likewise, the same approach can be applied for system design parameters. However, the design parameters do not require any logical statement. The
value of a specific design parameter can be used directly in the sub-component model from the external file during simulation.

At this stage, the whole process can be explained by a simple example. However, a real HVAC system modeling is discussed in the Chapter 6. Let’s suppose an overall model consists of three sub-component models (CM), such as CM1, CM2, and CM3 connected with inlet and outlet system boundaries in parallel arrangement as shown in Figure 4.2. Now, it is required to conditionally declare these component models sequentially. The method can be described through following steps.

Step 1: Properly couple all three sub-component models CM1, CM2, and CM3 with the system boundaries through their connectors.

Step 2: Import function ‘readRealParameter’ in the overall system model.

Step 3: Define a system configuration parameter and component design parameter e.g. C and D, respectively. For simplicity, the D is same in all component models.

Step 4: Prepare an external text file e.g. Data.txt with suitable values of system configuration and design parameters.

Step 5: Define C and D as real parameters in the overall model along with the name of external file, Data.txt. Afterwards, the system configuration parameter, C is used with three sub-component models with ‘if’ statement with logical condition. Let’s assume a simple logical condition for each component model, such as for first component model “if C>=1”, for second component model is “if C>=2” and for third “if C>=3”. Now, during the model translation, the specified function will read the value of C from Data.txt file. If the value of C in the Data.txt file is 1 then the first component model will be simulated, if C=2 then both first and second component models will be simulated. Similarly if C=3, then all three component models will be considered during model simulation. However, the value of system design parameter would be the same for all three component models as specified in the Data.txt file. Although, different values of same design parameter or different design parameters of the same component(s) can also be applied through such method.
4.3.3.2 Second Method: Redeclaration/ Replaceable component models

In the second method, the overall HVAC system model can be developed through redeclaration of sub-components to evaluate different configurations. Modelica supports the redeclaration of the component models. However, such component models are needed to be marked as ‘replaceable’. For the present work, implementation of the method can be described in two phases, such as component model development phase and implementation phase.

In the development phase, a partial base model is developed to provide a common platform to all other component models of that family i.e. family of heat exchangers. Actually, the base model consists of different connecting ports through which other family component models can be connected. In addition, various conditional component control inputs can also be included in the base model. Thus, base model provides a common interface for coupling different component models with other models. Afterwards, the main component models can be developed by extending the base model in the each respective model. Thus base model acts as a ‘base class’ and the component
models built on it act as ‘sub class’. The implementation phase of the method needs a particular declaration during the development of overall system configuration model.

For implementation, each component model is needed to be declared as ‘replaceable’ and ‘constrained’ by its base model to change all possible component models. Furthermore, a special Modelica annotation is also required, as ‘ChoicesAllMatching = true’. The annotation ensures to change all possible component models built on specific base model through ‘change class’ option to vary various possible configurations. However, when the component models are changed, the default values of component design parameters remain unchanged. But any modification in the default values would be removed during the change of component model. Therefore, during selection of new component model, the system model developer has to again modify the design parameters of the respective component model. It means that the design parameters cannot be changed automatically. In addition, the key drawback of the method is that the current version of Dymola/Modelica lacks the feature to allow the automatic change the class of component models from an external file. The method is depicted in Figure 4.3 by considering two component models, CM1 and CM2.

Figure 4.3: Second method: redeclaration/ replaceable component models

### 4.3.4 Array of component models

HVAC system configuration is not only depends on the type of a certain component, but also on the number of components used in the system. For example, a system operating with one chiller to meet the building load demands can also be operated with three chillers of lower cooling capacity for the same load requirements. Consequently, Modelica supports to use an array of such component models to develop a more
compact overall HVAC system configuration model. The array structure of a component model enables to put only single component model in the overall system model. However, the single component model can be simulated as multiple component models to get the desired impact during variation of different configurations.

In Modelica, the development and implementation of the array structure for a component model can be described through an example. Let’s suppose a component model (CM) is required to be modified as an array of \( n \) sizes. First, the CM name is assigned with respect to \( n \), such as CM[n]. The value of \( n \) can be controlled in two ways. First way is to declare it as an ‘Integer’ model parameter and assign the desired value for that it is required to be simulated. The more precise way to get automatic control is to declare \( n \) as an external parameter from an external file as explained in the first method of HVAC system modeling approaches. Such approach helps to develop an automated overall HVAC system model. However, the array of components cannot only be constructed from ‘Real’ variables, while the Modelica function for external parameters only valid for ‘Real’ parameters. Therefore, an additional Modelica function is also required to be defined in the CM to convert external ‘Real’ parameter into an ‘Integer’ parameter.

The implementation of the array component model involves two aspects, one about component design parameters and the other about its connections. The value of design parameters can be kept either same or different throughout the component array. For the same value in each component model, a Modelica keyword ‘fill’ is used with such parameters, followed by the desired value of that design parameter. For example, if it is written as ‘fill (10, n)’ with certain design parameter then the value 10 will be assigned to that parameter throughout the array. Furthermore, the array of component model requires multiple connections during the coupling among all component models in the array and also with other component models. Such connections can easily be applied by defining a ‘for’ loop to the model connectors depending on the serial or parallel arrangement of component models in the array as shown in the Figure 4.4 [100].

![Figure 4.4: Array of a component model](image)
4.3.5 HVAC system sizing

The aspects of HVAC system configuration and sizing are strongly related with each other. The individual component models and the system of the developed overall HVAC system model must be accurately sized to fulfill the specified buildings load demands. However, the key difference between the configuration and the sizing issues is that the configuration of HVAC system is not predefined. Therefore, the aspect of system configuration is mainly focused on the decisions related to the type, the number, and the arrangement of primary and secondary system components. Thus, after the selection of a suitable configuration, the sizing variables associated the particular configuration are identified. Meanwhile, the optimal configuration design is evaluated by the overall system performance that means solving the sizing problem is also required for a certain system configuration. As a result, in the current study, the applied approach for the optimization of HVAC system configurations can vary the both system configuration and system design parameters.

In the present work, two strategies are proposed to link the configuration problem with the sizing problem during development of overall HVAC system model. In Modelica, the design parameters of any component model can be varied to achieve different component sizes. However, the way in which the component design parameters could be varied, decides the strategy for handling HVAC system configuration and sizing problem. The first strategy involves developing an overall HVAC system model in which the same component model is multiple applied based on the different sizes available in the market for a particular application. Finally, after simulating all different sizes of a component model, the optimal component size can be determined. For example, if a component model (CM) is available in three different sizes, then the same CM would be applied three times in the overall system model with three different sizes. All three options would be simulated for a particular application and finally the CM with optimal performance shows the optimal component size and would be selected accordingly. In the second strategy, only a single CM can be used in the overall system model. However, the three options of component sizes can be implemented through developing a record of all three sizing possibilities. Afterwards, the sizing options can iteratively be coupled to the CM during simulation to determine the optimal component size. The two strategies are shown in Figure 4.5 (a) and (b).
4.3.6 Simulation of HVAC system model

The process of developing an overall HVAC system model is concerned with the implementation of an appropriate method to automatically vary different system configurations. However, the efficient model analysis can only be achieved through accurate simulation of the developed model. The simulation process deals with performing experiments on the model to predict its behavior under the real conditions.

In the current study, HVAC system modeling and simulation processes are executed through Dymola/Modelica. Dymola has powerful experimental, plotting and animation features. It has two modes, one is modeling mode for the development of system model, and the other is simulation mode to make experiments on the developed model. The simulation mode has a simulation setup, plot and animation windows, and variable browser. The simulation setup mainly defines three groups, such as the simulation setup...
interval, the output interval and the integration groups. However, additional aspects of
the simulation translation, output, debug, compiler, and real-time are also included. The
integration group specifies ‘algorithm’ to solve the differential equations, and
‘tolerance’ to define the desired accuracy. In addition, ‘fixed integrator step’ is also
defined for the fixed step integrator, such as Euler. However, sixteen different
algorithms are available in the list for the user selection [128]. The default integration
algorithm in Dymola is DASSL (Differential/Algebraic System Solver) which is a
variable-step, variable-order code that uses order buildup during the startup phase.
Currently, it is one of the most successful simulation codes in the market. DASSL as a
stiff-system solver is selected because Dymola was designed to be used in large and
complex system modeling. Moreover, most engineering users do not know whether
their models are stiff or not. Meanwhile, non-stiff models can be solved with a stiff-

The complexity of evaluating several configurations of HVAC system requires effective
interaction between modeling and simulation modes of Dymola/Modelica. In the
present work, the default solver DASSL is used for simulation. However, the two
modeling methods discussed before also influence the simulation process. For example,
if the HVAC system is modeled based on the first method of conditional declaration of
component models, then an external file is also needed to be properly created. Any
change in the external file with respect to the system configuration and/or design
parameters would require re-translation of the model before simulation to take into
account the modification impact [100]. All aspects of HVAC system modeling, sizing,
and simulation defined in the current study are combined in Figure 4.6 to give an
overview of whole modeling and simulation process.

4.4 Optimization of HVAC system configurations

In general, optimization is concerned with the minimization or maximization of the
objective function(s) depending on what someone is looking for. However, normally
optimization problems are defined as ‘minimization’ and if a criterion is subjected to
‘maximization’, then the negation of the objective function is minimized. In
optimization problems, the building simulation programs are increasingly being used to
evaluate the objective function. The current study is based on the ‘minimization’ of the
objective function in which the Dymola/Modelica overall HVAC system model is
coupled with optimization program, GenOpt by using an appropriate optimization
algorithm. Such an interaction provides automatic selection of optimal configuration.
However, it requires suitable selection of optimization algorithm along with
optimization variables in terms of system design and configuration parameters.
4.4.1 Optimization algorithms

A wide range of optimization algorithms is used in various HVAC optimization problems. However, decision about the selection of a suitable algorithm is vital for optimization of HVAC configurations. Generally, optimization algorithms can be categorized into two basic classes: deterministic and probabilistic algorithms. In deterministic algorithm, after each execution step, at most one way exits to proceed. The algorithm is terminated if no way exists to proceed. Such algorithms do not contain instructions that use random numbers to decide what to do or how to modify data. In addition, deterministic algorithms are most often used if a clear relation exists between the characteristics of the possible solutions and their utility for a given problem. However, if the relation between a solution and its ‘fitness’ is not so evident or too complicated, or the dimensionality of search space is very high, then it becomes
difficult to solve a problem deterministically. Therefore, probabilistic algorithms are used to handle such problem. Most of the optimization research activities in various fields are implementing probabilistic algorithms. Mainly, the probabilistic algorithms are based on the Monte Carlo approaches that include numerous algorithms, such as Hill Climbing, Simulated Annealing, Evolutionary algorithms, etc. [149].

Generally, building simulation programs comprise code features, such as adaptive integration meshes, iterative solvers, and if-then-else logic. The feature can cause the failure of optimization algorithm that requires smoothness of the objective function. However, if such simulation programs are used in conjunction with the optimization algorithms then tolerances of the adaptive solvers must be tight. Though, it is observed that in many building simulation programs the solver tolerances are so coarse that such algorithms can indeed fail far from minimum. Therefore, probabilistic optimization algorithms that do not require smoothness are frequently used to solve building optimization problems with small number of simulations [21].

Though, it is also to remember that no general optimization algorithm performs the best for all applications and in addition, no optimization algorithm can guarantee finding the global minimum if local minima exits. Furthermore, the selection of a suitable algorithm for a particular application depends on various factors, like structure of the objective function, availability of analytic first and second derivatives, number of independent parameters, and problem constraints.

GenOpt is used in the current study for the optimization of HVAC system configurations. GenOpt optimization library consists of local and global one-dimensional and multi-dimensional algorithms, as well as algorithms for parametric runs. Several optimization algorithms can be applied in GenOpt that include Generalized Pattern Search algorithms (GPS), Particle Swarm Optimization algorithms (PSO), Discrete Armijo Gradient algorithm, etc. Moreover, a hybrid global optimization algorithm can also be used in which PSO performs the global optimization and Hooke-Jeeves achieves the local optimization. Beside existing algorithms, GenOpt superclass ‘optimizer’ can be extended to apply users defined optimization algorithms [130]. While the selection of a suitable optimization is based on the form of optimization variables involved in the study.

4.4.2 Optimization variables and constraints

In the literature, the optimization variables are also called the problem variables or decision variables. In the present work, the optimization variables consist of system configuration variables that define the type and the number of component model, and system design variables for the design optimization of HVAC system configurations. The configuration variables are discrete and defined to vary the all possible alternatives of overall HVAC system model during simulations. In overall HVAC system model, such variables are assigned to the component models that are conditionally introduced
to evaluate different system configurations. Such variable would define the type as well as the number of specified component model. For example, a HVAC system model with two chillers and three cooling towers. Therefore, distinct configuration variables are required to be used for each type of component model.

The optimization design variables are mostly continuous and vary the design parameters of different component models to achieve optimal system performance. For example, mass flow rate and temperature difference across condenser of a chiller, and fan speed of a cooling tower, etc. Such variables are also defined separately for each individual component model.

However, both categories of optimization variables are needed to be properly constrained with respect to their feasible variation range. The feasible variation range depends on specified building load demands that would decide the feasible type and the number of components to be varied along with system and component design variables. Such constraints also ensure that each configuration of the HVAC system model is properly sized to fulfill the specified building load demand.

4.4.3 The objective function

The objective function can be any performance factor of the HVAC system configuration, such as total power consumption, annual energy consumption, or coefficient of performance. However, in the current research, the objective of the optimization problem is to minimize either the estimated total system power consumption (kW) or the annual energy consumption (kWh) of each HVAC system configuration. The estimated annual energy consumption is the integration of a selection of typical design conditions and their occurrence in a typical year.

The objective function is clearly defined during the development of overall HVAC system model along with the result file to store the calculated values of the objective function. Obviously, the equation to calculate the objective function changes during the variation of component models for each HVAC system configuration. Therefore, it is ensured that the variables of the equation are properly constrained to calculate the correct value of the objective function for each HVAC system configuration. In Modelica, any design parameter or variable of the component models can easily be used to develop the equation for calculation of an objective function. It only requires to accurately defining the component model’s name followed by the name of the parameter or variable of that respective model. Furthermore, the model configuration parameters are also included in the equation of the objective function to evaluate different quantities of specific component models. Finally, the constraints of the optimization variables ensure the accurate computation of the objection function for each system configuration. For example, the equation 4.1 calculate the objective function ‘f’ where CM1.Ptot and CM2.Ptot as variable of component models would compute the total power consumption (kW) of the component model 1 and 2,
respectively, while \( N_{CM1} \) and \( N_{CM2} \) define the quantity of respective component model of a particular system configuration.

\[
f = N_{CM1} \times CM1.Ptot + N_{CM2} \times CM2.Ptot
\]

4.4.4 GenOpt optimization setup

A simulation input file is required to be written after the development of an overall HVAC system model in Dymola/Modelica. The simulation input file consists of the configuration and design parameters that are defined in the overall HVAC system model for the conditional declaration of various components models. Afterwards, the task of setting up the optimization problem commences. The task is comprised of five steps to properly execute the optimization process [132].

4.4.4.1 Specification of independent parameters

The independent parameters are actually the optimization variables that are defined based on the simulation input file. However, in order to specify them as independent parameters, the numerical value such variables is replaced with a variable and then enclosed in percentage signs. After specifying the independent parameters, the new file is denoted as ‘simulation input template file’ in GenOpt. For example, if ‘A’ is an optimization variable in the simulation input file then it will be written as ‘%A%’ in the simulation input template file.

4.4.4.2 Specification of optimization algorithm

Various optimization algorithms are available in the GenOpt library. The algorithm is defined in the ‘optimization command file’. In that file, the optimization settings are specified that describes the number of iteration and the values of the fixed parameters associated with the particular algorithm. In addition to the optimization algorithm, the optimization variables and their variation range is also defined in the ‘parameter’ section of the command file. The section specifies the minimum value, the initial value, and the maximum value of each variable. Moreover, it also declares a step size that is used by the optimization algorithm.

4.4.4.3 Specification of simulation program

The specifications of simulation program i.e. Dymola are defined in the ‘simulation configuration file’, i.e. DymolaWinXP.cfg. The specifications include the information that how GenOpt would start the Dymola. In addition, a message is also defined in the file that would be used in the Dymola log file if an error stops the simulation process. Furthermore, the configuration files for various other simulation programs, such as DOE-2, EnergyPlus, TRNSYS, etc. are also contained in the GenOpt installation. The user does not need to modify available configuration file.
4.4.4.4 Defining the objective function

GenOpt reads the value of the objective function from the results text file of Dymola. However, it is also required to accurately define the objective function in terms of its name e.g. \( P_{total} \), and the location in ‘initialization file’ of GenOpt. In addition, the Dymola result file is also defined in the initialization file as an output file.

4.4.4.5 File location

After defining the all necessary simulation and optimization parameters in their respective files, then files location is defined so that GenOpt can properly trace them. The all files are located in the initialization file of GenOpt. Finally, the optimization can be started after complete the task of file location.

4.4.4.6 Coupling between Dymola and GenOpt

One of the key objectives of the present study is to achieve the automated selection of the optimal HVAC system configuration. Therefore, proper coupling between Dymola and GenOpt is essential. Thus, Dymola scripting feature is used in the study.

The goal of scripting in Dymola is to fully automate the simulation. The script ability makes it possible to load model libraries, set parameters, set start values, simulate and plot variables. In addition, it can be considered as a way of storing a successful sequence of interactive events. Therefore, it is fairly supportive for an overall HVAC system model in which different component models are being varied during simulation.

Dymola support easy handling of scripting, both with functions and script files. However, the same functionality can be obtained with both ways. Therefore, Modelica script file ( .mos, from Modelica scripts) is used in the current research work. Modelica script file provides the convenient way to ‘pack’ a number of actions to simplify their use. Moreover, the script file can be handled in both modeling and simulation modes of Dymola. For the current work, a script file is created ‘HVACModel.mos’. The file specifies the Modelica file (HVACModel.mo) of the overall HVAC system model and also includes the specifications of simulation setup of the specified model along with its result file. The built-in functions ‘openModel’ and ‘simulateModel’ are used in the script file. The script file can be run in five different ways. In one option the script file can also be executed before simulation to set the desired values of parameters. It requires running the Dymola a number of times in ‘batch mode’ with a batch file that specifies the script file. The approach is used in the current study that because the different configurations are decided based on the conditional declaration of component models depending on the configuration and design parameters. Thus, the execution of script file opens the model and defines the HVAC system configuration based on configuration parameters and finally simulates the model. For each simulation a new result file is created and Dymola is terminated after the last simulation.
It is described in the GenOpt setup that the specifications of Dymola are defined in the configuration file to start the simulation. Therefore, the path of Dymola batch file is specified in the configuration file. In addition, the other files are also defined in the initialization file, including the external text file with the configuration and design parameters for the conditional declaration of component models, HVACModel.mo file, and HVACModel.mos file. Finally, it can be concluded that during the coupling GenOpt execute Dymola batch file, which in turn calls the script file invoking Dymola model as shown in Figure 4.7 [100].

**Figure 4.7: Coupling between Dymola and GenOpt**

GenOpt starts Dymola after writing the simulation input file based on the optimization variables which are determined by the selected optimization algorithm. After the completion of simulation, GenOpt check the Dymola log and output files. If no error is recorded in the log file, then GenOpt reads the value of objective function the Dymola result file. Afterwards, a new set of optimization variables is selected by the optimization algorithm that defines new HVAC system configuration for the next simulation run. The whole process continues iteratively until all the possible configurations are evaluated and minimum value of the objective function is reached. The overview of complete interaction between Dymola and GenOpt is shown in Figure 4.8.

4.4.4.7 Summary

In the research methodology, all aspects of the implemented research approach are described. In view of the complexity of the optimization problem of HVAC system configurations in terms of automated configuration selection, the limitations of the conventional modeling languages are defined. However, the capabilities of equation-based object-oriented modeling and simulation environments, such as a-causal and declarative modeling, encapsulation and modularity, inheritance, reusability, and multi-domain are the key factors to select a such modeling and simulation approach in the present research work. Therefore, Dymola/Modelica, as an equation-based object-oriented (EOO) modeling and simulation platform is used. A brief overview of the Dymola/Modelica is also presented. However, the capabilities of the approach and the
program are not entirely explored and exploited in the field of building energy systems. Consequently, the current study also provides the opportunity to enhance the awareness and effectiveness of such modeling and simulation approach for the analyses of HVAC systems.

![Diagram of Automated simulation-based optimization process]

**Figure 4.8: Automated simulation-based optimization**

The evaluation of various HVAC system configurations requires developing such physical model that is capable of varying different system configurations and design parameters. Therefore, two methods are described in view of Dymola/Modelica modeling capabilities to develop such HVAC system model. However, the first method involves the conditional declaration of component models from an external file, therefore can be used to automatically vary the HVAC system configuration. The second method varies the system configuration through changing the class of a specific component model, but Dymola/Modelica still lacks the feature to automatically change the class of component model.

The optimization of HVAC system configurations involves suitable selection of the optimization algorithm, the optimization variables, and the objective function. GenOpt
is used in the present work for the optimization. However, it requires to appropriate coupling between GenOpt and Dymola/Modelica to achieve the automated simulation-based optimization. The coupling is based on creating the different files of GenOpt and Dymola/Modelica. Though, the research methodology of simulation-based optimization is explained generally in the current chapter but it is implemented on the real HVAC systems described in the upcoming chapters.
Chapter 05

5 DEVELOPMENT OF DESICCANT WHEEL MODEL

HVAC system researchers are focusing on alternative cooling techniques due to high energy costs and the environment impacts of conventional cooling systems. Such alternative cooling systems primarily involve air humidification/dehumidification along with evaporative cooling based on the low-grade and renewable energy resources. In such desiccant evaporative cooling (DEC) systems, the desiccant wheel is the key component that strongly affects the overall system performance in terms of capacity, size, and cost. Therefore, in the current chapter, a detailed desiccant wheel model is developed and validated in Dymola/Modelica under real transient conditions.

5.1 Basic Concepts

The desiccant wheel can also be defined as a dehumidification wheel or rotary dehumidifier. In general, a desiccant wheel comprises of a large number of air flow channels. The channel walls are created by matrix material as a supporting material. The main matrix materials used are paper, synthetic fiber, aluminium or plastic. The matrix material is then layered with desiccant materials, also known as adsorbents. Several adsorbents are being used for fabricating desiccant wheels including silica gel, activated alumina, lithium chloride, synthetic polymers, and natural and synthetic zeolites (molecular sieves) [150] [151] [98].

- Desiccant Cycle

All desiccant wheels work in same manner by absorbing moisture from the surrounding air due to the pressure difference between water vapors and the desiccant surface. When the vapor pressure of surrounding air is higher than that of desiccant surface, moisture is adsorbed by the adsorbent, otherwise, the adsorbent releases moisture. The water vapor pressure at the desiccant surface increases with the adsorption of moisture. Consequently, at some point, the water vapor pressure at the surface becomes equal to the surrounding air i.e. the equilibrium state. When this happens further moisture cannot be adsorbed. However, in order to operate the system continuously, the absorbed water vapor must be released to make the adsorbent dry enough to adsorb moisture again in the next cycle. The continuous operation is ensured by changing the vapor pressure at the adsorbent or in the air by some external force. In general, heat is an external force that increases the temperature of the adsorbent to increase its surface vapor pressure as compared to the surrounding air. Under these conditions, moisture leaves the desiccant and the process is described as ‘reactivation’ or ‘regeneration’. However, the regeneration of the adsorbent causes too high a vapor pressure to adsorb moisture that condition. Therefore, cooling of the desiccant is required to reduce its vapor pressure so that it can adsorb moisture again. The desiccant cycle is shown in Figure [3] [98]. Normally the wheel cross-section is divided into two zones, one for process air, which
is dehumidified and heated, and other for regeneration air, which removes moisture from adsorbent. The ratio of process and regeneration zones varies from 50/50 to 75/25 by different manufacturers depending on wheel application [152] [150].

![Figure 5.1: Typical desiccant cycle [98]](image)

### 5.2 Adsorption Characteristics

The Adsorption of moisture on the surface of the desiccant is quite a complicated phenomenon that involves vander-waals and electrostatic forces between adsorbate (moisture) and adsorbent. The process differs for each adsorbent. However, in general it is characterized by some physical properties of adsorbents such as surface area, volume of capillaries, and the range of capillary diameters [153] [98]. For higher adsorption capacity, large specific surface area is desirable. Though, the large internal surface area increases number of small sized pores between adsorption surfaces in a limited volume. In addition, the intensity of accessibility of adsorbate to the adsorbent surface is determined by the size of the micropore. Thus, the pore size distribution of a micropore is a key characteristic of the adsorption process [154] [155].

#### 5.2.1 Adsorption Equilibrium and relations

Adsorption equilibrium is the critical characteristic of an adsorbent as it defines the adsorption capacity that determines the amount of adsorbate taken up by the unit mass of adsorbent under a range of operating conditions. It depends on the operating conditions in terms of adsorbate concentration and temperature. In addition, adsorption capacity also indicates the amount of adsorbent required to fill the volume of the adsorption device [156].

Adsorption equilibrium can be expressed in three ways, (1) when the adsorbent temperature is kept constant the adsorbate pressure varies, the change in amount of
adsorbate against the pressure is described as an adsorption isotherm, (2) and when pressure is kept constant called an adsorption isobar, and (3) if the amount of adsorbate is kept constant then it is an adsorption isostere. However, generally, an adsorption isotherm is used to define the effects of adsorption [157]. The adsorption isotherms depend on the type of adsorbent material and different types of adsorption isotherms for various materials are studied in several research works [158] [159]. A large variation is observed for such isotherms because manufacturers use different approaches to optimize the adsorption material for different applications [98]. Under adsorption operation, the International Union of Pure and Applied Chemistry (IUPAC) have recognized six different types of equilibrium isotherms as shown in Figure 2 [157] [160]. Type I isotherms are of the classical Langmuir adsorption form and represents the characteristics of micro-porous adsorbents. The adsorbents of this type include several types of charcoal and silica gel. The type II isotherms represent macro-porous adsorbents, non-porous materials and a few compacted powders. The prediction of an adsorption/desorption process of these types of isotherms are based on the B.E.T theory developed by Brunauer, Emmett and Teller. Graphitized carbon and compact powders of silica are the examples of adsorbents of type II adsorption isotherms. The type III isotherms also characterize various adsorbents that include polymers, silica aerogels or graphitized carbon. However, Figure 5.2 shows how the type I forms a concave downward curve and is considered as ‘favorable’ when compared to a type III that forms a concave upward curve and is ‘unfavorable’ due to large deviation from the Langmuir model. While the remaining, type IV and V form inflection curves.

![Figure 5.2: Classification of adsorption isotherms accepted by IUPAC [161]](image-url)
Moreover, the adsorption isotherms are also presented through various mathematical relationships. Few forms are based on the simplified physical phenomenon of adsorption and desorption. While some relations are empirical relating to the experimental data in simple equations based on a few empirical parameters. Different experimental techniques are used to determine the adsorption characteristics of various materials, such as nitrogen and water adsorption experiments [162]. An example of the simplest type of model is the ‘Langmuir Isotherm’ as defined by Equation 5.1 in which localized adsorption proceeds on an energetically uniform surface without any interaction between adsorbed molecules [163].

\[ W = C \frac{P}{1+C \cdot P} \]  \hspace{1cm} (5.1)

In addition, Henry law is also used for linear approximation to drive the adsorption isotherms. However, the method is valid for only low pressure values. Thus the linear function can be defined by Equation 5.2 [157].

\[ P = \text{constant} \cdot W \] \hspace{1cm} (5.2)

Sometimes however, the whole isotherm is not linear. Therefore, the Henry law can be applied to part of the isotherm. The Langmuir adsorption isotherm or BET theory is used for the non-linear adsorption isotherms.

The empirical relationship of the adsorption isotherm frequently employed is the ‘Freundlich’ model shown through Equation 5.3. The empirical constants \( C_F \) and \( n_F \) of the model describe the adsorption capacity and efficiency of the process respectively [164]. The constants are different for different adsorbents.

\[ W = C_F \cdot P^{\frac{1}{n_F}} \] \hspace{1cm} (5.3)

### 5.2.2 Real and ideal desiccant wheel behavior

The outlet conditions of a desiccant wheel are influenced by several factors, such as the adsorption characteristics of the adsorbent, the wheel rotation speed, the air mass flow rates, the fraction wheel area for adsorption and desorption. The ideal outlet conditions correspond to the maximum dehumidification that can be identified through adsorption equilibrium curves in terms of adsorption isotherms. The maximum value for ideal dehumidification in terms of moisture content and temperature is imposed by the inlet conditions of process air. The inlet conditions of regeneration air dictate the minimum ideal value [165] [166]. Typically the maximum ideal outlet conditions are used as a reference during the optimization of desiccant wheel performance. In addition, the ‘characteristic potentials’ F1 and F2 are also used as psychrometric variables to identify the ideal desiccant wheel behavior. The F1 and F2 lines are close to the specific enthalpy and relative humidity lines of psychrometric chart, respectively. An example of real and ideal conditions is shown in Figure 5.3.
Figure 5.3: Ideal and actual desiccant wheel behavior
5.2.3 Potential theory of adsorption

Various factors affect the adsorption capacity of an adsorbent material. However, the key factors are the relative humidity and temperature of the surrounding air. Therefore, the relationship is determined to show the dependence of relative humidity and temperature on adsorption capacity of a desiccant wheel. Such a relationship indicates the performance of a desiccant wheel. The relationship was first presented by Polanyi [167] and further developed by Dubinin [168]. In the theory, the adsorption potential was related to relative humidity and temperature through equation 5.4. The adsorption capacity was assumed to be a function of adsorption potential as given in Equation 5.5.

\[ A = RT \ln \left( \frac{P_s}{P_v} \right) \]  \hspace{1cm} 5.4

\[ W = f (A) \]  \hspace{1cm} 5.5

Where \( P_v \) and \( P_s \) are the actual vapor pressure and saturation pressure at that temperature. Thus \( P_v/P_s \) represents the relative humidity.

Penetration theory also helps to analyze the simultaneous heat and mass transport phenomenon under transient conditions during which the interaction times and distances are short. In such a situation, for example in adsorption process, the heat and mass co-efficient can be determined by penetration theory. The generalized form of heat and mass co-efficient is presented by Equations 5.6 and 5.7, respectively [169] [159].

\[ \alpha = 2 \sqrt{\frac{\lambda c \rho}{\pi \tau}} \]  \hspace{1cm} 5.6

\[ \gamma = 2 \sqrt{\frac{D_{sf}}{\pi \tau}} \]  \hspace{1cm} 5.7

Here, \( \lambda \), \( c \), \( \rho \), \( \tau \) and \( D_{sf} \) are thermal conductivity, heat capacity and density, time, and surface diffusion co-efficient, respectively.

5.3 Operating modes of desiccant wheels

Desiccant wheels are widely used in different air conditioning applications. The wheels are used for air dehumidification and enthalpy recovery, depending on system requirements and ambient conditions. Currently, manufacturers are producing various desiccant wheels that can work in both modes throughout the year.

In the dehumidification mode, the process air is dehumidified and heated, while the hot regeneration air dries the desiccant wheel, becomes humidified and cools down. The wheel rotates slowly in dehumidification mode. The wheel schematics and processes are shown on a psychrometric chart in Figure 5.3a and b, respectively.
In the enthalpy mode, the desiccant wheel rotates between the supply and room exhaust air. The wheel rotates at higher speed compared to dehumidification mode. The enthalpy mode can be further classified into two sub-modes, such as cooling and heating modes depending on the inlet conditions of supply and return air with respect to temperature and relative humidity. In cooling enthalpy mode, the process air is dehumidified and cold down in summer, while in heating enthalpy mode the process air is humidified and heated in winter by the room return air. The wheel schematics and the both enthalpy modes on psychrometric chart are shown in Figure 5.4 and 5.5, respectively.
Figure 5.5: Schematics and psychrometric chart of cooling and heating modes of enthalpy wheel

5.4 Desiccant wheel model development

Model-based performance analysis of different types of desiccant wheels is presented in several studies [170] [171] [172] [173]. Such models are developed in a wide range from simplified complex to complex models based on single or hybrid desiccant materials, respectively. An overview of various desiccant wheel models is presented in a study [174]. However, despite many theoretical and experimented analyses it cannot be said that the design and operation of desiccant wheels are completely understood. The
desiccant wheel design varies from manufacturer to manufacturer and largely depends on their experience. Such theoretical and numerical approaches seek to solve differential equations or to interpret specific parameters derived from heat and mass governing equations. However, this could result in large errors under specific operating conditions whilst predicting good performance for a small range of other conditions. Thus an acceptable, easy, and effective method was presented to determine the performance of desiccant wheels in real operation [175].

5.4.1 Mathematical Model

The desiccant wheel is a rotating cylinder of length L and diameter D at constant speed and driven by an electric motor. It provides a large contact surface area for desiccant-air interaction through numerous small channels in which the adsorbent adheres to the thin walls of the matrix material. The channels are structured in different shapes, such as rectangular, triangular, and sinusoidal [165]. The wheel is divided into two sections: the adsorption section, in which process air is heated and dehumidified, and a desorption section in which hot regeneration air is used to dry the desiccant wheel at the same time cooling and humidifying. The process and regeneration air streams are in a counter flow arrangement. The schematics of the desiccant wheel and channel section are shown in Figure 5.6.

![Figure 5.6: Schematic of desiccant wheel](image)

The physical model is based on the following assumptions:

1. No chemical reaction and leakage of fluid takes place in the channels.
2. The air flow on the process and regeneration side is one-dimensional.
3. The inlet conditions of air in terms of temperature, humidity, and velocity of each flow are uniform.
4. The thermal properties of desiccant material remain uniform and constant.
5. The concentration and porosity of sorption through channels are negligible.
The mathematical model is based on a set of algebraic equations, in which the enthalpy and humidity changes simultaneously to determine the outlet temperature and absolute humidity ratio of the process and regeneration air with varying rotation speed [175]. Additional correlations are used to determine some specific properties of moist air based on transient operating conditions. For example, Equation 5.8 is used to calculate the relative humidity of the process and regeneration air based on the inlet and outlet absolute humidity and temperature [98].

$$
\varphi = \frac{P_w}{P_{ws}} 
$$

(5.8)

$P_w$ the partial pressure of water vapor in moist air, and $P_{ws}$ the pressure of saturated pure water are calculated by Equation 5.8a and 5.8b, respectively.

$$
P_w = \omega P_{atm}/(0.62189 + \omega) 
$$

(5.8a)

$$
P_{ws} = \exp(P_{sat})/1000 
$$

(5.8b)

Where the $P_{sat}$ is determined by the Equation 5.8c for the temperature range between 0°C to 200°C [98].

$$
P_{sat} = \frac{A1}{T} + A2 + A3T + A4T^2 + A5T^3 + A6\ln(T) 
$$

(5.8c)

Here $T = ^{\circ}C+273.15$, $A1 = -5.8002206 \times 10^3$, $A2= 1.3914993 \times 10^0$, $A3= -4.8640239 \times 10^2$

$A4= 4.1764768 \times 10^5$, $A5= -1.4452093 \times 10^8$, and $A6= 6.5459673 \times 10^6$.

The enthalpy of moist air is calculated by the Equation 5.9 [98].

$$
h = 1.006T + \omega(2501 + 1.86T) 
$$

(5.9)

In a desiccant wheel, the amount of moisture adsorbed is characterized by adsorption isotherms, i.e. the amount of adsorbate on the unit quantity of adsorbent as function of pressure or concentration at the constant temperature. In the current model, Freundlich Equation 5.10 is used to determine the equilibrium amount of moisture adsorbed $q_{eq}$ as a function of relative humidity.

$$
q_{eq} = C_F(\varphi)^{1/n_F} 
$$

(5.10)

Here, $C_F$ and $n_F$ are Freundlich constants and are related to adsorption capacity and adsorption efficiency, respectively.

The isosteric heat of adsorption is calculated from Clausius-Clapeyron relation and the surface diffusivity is determined according to penetration theory. The surface diffusivity is correlated with temperature by the Arrhenius Equation 5.11.
The average temperature of process inlet air and regeneration air is used to estimate the surface diffusivity.

The outlet conditions of the process and regeneration air involve the heat and mass transfer between two air streams depending on the respective inlet conditions along with the adsorbent material and structural characteristics of desiccant the wheel. The heat transfer from the regeneration air to the process air depends on the heat capacity of the desiccant wheel and the adsorbate water. Therefore, the amount of sensible heat exchange between the hot regeneration air and the process air through the rotating wheel, per unit time, is calculated by Equation 5.12.

\[
H = AL\rho_{dw}N\{(C_{pc} + q_{eqreg}C_{pw})T_{reg} - [C_{pc} + (q_{eqreg} + q)C_{pw}]T_{sin}\} 
\]  

(5.12)

Moreover, the Equation 5.13 can also be used to determine the sensible heat transfer through a rotating wheel [176].

\[
H = \frac{\rho_{dw}C_{pc}(T_{reg} - T_{sin})NL}{\beta_s\rho_d U_s} 
\]  

(5.13)

For mass transfer, it is assumed that surface diffusion is the dominant mass transfer mechanism. The adsorption time of moisture diffusion to determine the optimal rotation speed is related to the state in which the penetration theory is applicable. Therefore, the amount of water vapor adsorbed by the adsorbent material at the completion of the adsorption period is calculated by Equation 5.14.

\[
q = \left(\frac{2}{\pi}\right)\sqrt{D_{sf}^*}\sqrt{t_{ad}}A_{ws}q_{eqsin} 
\]  

(5.14)

Here the adsorption time \( t_s = 3600\beta_s / N \) sec.

Thus, the volume of water transfer from process air to the regeneration air through the desiccant wheel per hour is calculated by Equation 5.15.

\[
Q = qAL\rho_{dr}N = \left(\frac{2}{\pi}\right)\sqrt{D_{sf}^*}\sqrt{3600\beta_sN\rho_{ad}A_{ws}q_{eqsin}}AL\rho_{dw} 
\]  

(5.15)

The outlet conditions of process and regeneration air in terms of absolute humidity and specific enthalpy with respect to heat and mass transfer process are calculated by Equations 5.16, 5.17, 5.18, 5.19, respectively.

\[
\omega_{sout} = \omega_{si} - Q/m_s
\]  

(5.16)

\[
h_{sout} = h_{si} + H/m_s
\]  

(5.17)
The above equations are used for the dehumidification mode of a desiccant wheel in which the process air is dehumidified and heated whilst the regeneration air is humidified and cooled. However, a desiccant wheel from different manufacturers can also be used as an enthalpy wheel under high rotation speeds. Therefore, the enthalpy aspect can also be considered based on a specific low value of regeneration temperature that will cause no regeneration in the wheel. In such a situation, the outlet conditions of the process air are calculated from the following Equations 5.20 and 5.21.

\[
T_{\text{out}} = T_{\text{in}} + \epsilon_{\text{sn}} (T_{\text{reg}} - T_{\text{in}}) 
\]

\[
\omega_{\text{out}} = \omega_{\text{in}} + \epsilon_{\text{lt}} (\omega_{\text{reg}} - \omega_{\text{in}})
\]

The cooling and heating modes of the enthalpy wheel are decided based on the inlet operating conditions.

5.4.2 Optimal rotation speed

The rotation speed, N of the desiccant wheel is the most important operating parameter that can significantly improve wheel performance. The optimal rotation N_{opt} speed attains the maximum moisture removal to minimum absolute humidity ratio at the process air outlet [175] [176] [177]. Thus, the rotation speed should be low enough for complete regeneration or fast cooling of the desiccant wheel but also high enough to keep the adsorbent far from the equilibrium state. This gives us the figure N_{opt}. However, N_{opt} is influenced by various operating parameters, such as face velocity, inlet temperature and humidity of the process and regeneration air. In different research activities, the optimal rotation speed is predicted by a numerical solution of differential equations governing the underlying processes [174] [178] [150]. However, in the current study, N_{opt} is determined by the validated proposed method [175] based on the ideal and actual outlet conditions described in section 5.2.2. The line of constant relative humidity is considered as an isostere. Therefore, the relative humidity at the outlet of process air is considered equal to the relative humidity of the inlet and regeneration air through Equation 5.22 to predict the optimal rotation speed and outlet conditions.

\[ \varphi (\omega_{\text{sopt}}, h_{\text{sopt}}) = \varphi (\omega_{\text{rin}}, h_{\text{rin}}) \]

Finally, the Equations 5.23, 5.24, 5.25 are used to calculate the optimal values of rotation speed, and absolute humidity and specific enthalpy at the outlet of process air, respectively.
\[ N_{\text{opt}} = \left( \frac{Q_{\text{opt}}}{\pi} \right) \sqrt{D_{s} \sqrt{3600 \beta_s \sqrt{N_{\text{opt}}}} A_{\text{ws}} q_{\text{es}} A L \rho_{\text{dw}}} \] ^{2} \quad (5.23)

\[ \omega_{\text{opt}} = \omega_{\text{in}} - \frac{Q_{\text{opt}}}{m_{s}} \] \quad (5.24)

\[ h_{\text{opt}} = h_{\text{in}} + \frac{H_{\text{opt}}}{m_{s}} \] \quad (5.25)

Here \( Q_{\text{opt}} \) and \( H_{\text{opt}} \) are the optimal values at \( N_{\text{opt}} \).

However, the mathematical model of the proposed method is able to predict the outlet conditions only when the rotation speed is either equal to or below the optimal value. Therefore, Equations 5.26 and 5.27 introduce two correction factors in the model, one for temperature, \( K_{t} \), and another for absolute humidity, \( K_{w} \), to estimate the outlet conditions of process air when rotation speed is higher than optimal.

\[ T_{\text{crsout}} = T_{\text{opt}} + T_{\text{sin}} K_{t} (N_{\text{in}} - N_{\text{opt}}) \] \quad (5.26)

\[ W_{\text{crsout}} = W_{\text{opt}} + X_{\text{sin}} K_{w} (N_{\text{in}} - N_{\text{opt}}) \] \quad (5.27)

### 5.4.3 Pressure drop

The total pressure drop across the desiccant wheel is determined by Equation 5.28 as the sum of distributed and local pressure drops [150]. The distributed and local pressure drops are calculated by Equations 5.29 and 5.30, respectively.

\[ dP_{\text{total}} = dP_{\text{d}} + dP_{\text{l}} \] \quad (5.28)

\[ dP_{\text{d}} = 4f \left( \frac{L}{D_{\text{eq}}} \right) \left( \frac{1}{2} \right) \rho_{a} u^{2} \] \quad (5.29)

The following correlation is used to calculate the equivalent diameter of the sinusoidal duct.

\[ D_{\text{eq}} = a \left[ 1.0542 - 0.4670 \left( \frac{a}{b} \right) - 0.1180 \left( \frac{a}{b} \right)^{2} + 0.1794 \left( \frac{a}{b} \right)^{3} - 0.0436 \left( \frac{a}{b} \right)^{4} \right] \]

\[ dP_{\text{l}} = k \left( \frac{1}{2} \right) \rho_{a} u^{2} \] \quad (5.30)

Finally, Equations 5.31 and 5.32 are used for process air outlet temperature and absolute humidity, respectively, to account for a possible time delayed response of the monitoring sensors.

\[ \frac{dT_{\text{sout}}}{dt} = \left( \frac{GT_{\text{sin}} - T_{\text{sout}}}{\tau_{t}} \right) \] \quad (5.31)
Recent advancements in the field of physical modeling offer opportunities to use an equation-based object-oriented (EOO) modeling and simulation approach for building energy systems [179] [100]. Therefore, in the present work, the mathematical model of desiccant wheel is implemented through the Modelica language [56] with simulations being performed in Dymola [129]. Appropriate control strategies are applied through Modelica features to estimate the wheel performance in the dehumidification and enthalpy modes under a wide range of design and operating conditions as presented in Figure 5.7.

The operational mode of a wheel is determined based on a particular minimum value of regeneration temperature with respect to adsorbent. The aspect is implemented in the model as an input, $T_{reg\_min}$. The actual input regeneration temperature is compared with the $T_{reg\_min}$ through the mode decision maker component, ‘MDMaker’. If the regeneration temperature is lower than the defined value of $T_{reg\_min}$ then the model would be in the enthalpy mode, otherwise it will operate in the dehumidification mode. The outlet conditions of either dehumidification or enthalpy mode are controlled by the ‘ModeSwitch’. ‘ModeSwitch1’ is for the temperature selection and ‘ModeSwitch2’ is for the absolute humidity based on the respective operating mode. Furthermore, the pipe model from the LBL buildings library is used to determine the pressure drop in the process and regeneration sides across the desiccant wheel. Additionally, a delay component model is also used at the simulated outlet values to encounter the delay of sensors in the real monitoring system.

In the dehumidification mode, two approaches can be implemented for the rotation speed through the conditional input, $N_{in}$. The first approach is related to the wheel operation when rotation speed is not defined by the model user i.e. $N_{in}$ is inactive in the list of model parameters. In such a situation, the model automatically determines the $N_{opt}$ and all outlet conditions of the process and regeneration air would be at the optimal values. In the second approach, the user can implement the desired rotation speed by making $N_{in}$ active in the parameter list to determine the actual performance of the desiccant under the particular operating conditions based on user defined $N_{in}$. Despite activating $N_{in}$, the model always determines the $N_{opt}$ and then the model control strategy compares $N_{in}$ with $N_{opt}$. If these rotation speeds are equal then the actual outlet conditions would also equal the optimal, otherwise the difference in the actual and the optimal conditions would depict performance of a desiccant wheel under user’s defined $N_{in}$.
Additionally, in the dehumidification mode, an appropriate control strategy is also applied when the relative humidity at both inlets of the wheel are equal under real transient operations. This situation causes some problems during the model numerical translation. Therefore, for such real operational situations and based on the typical sensor accuracy for relative humidity measurements, a 5% difference in the relative humidity at the both wheel inlets is considered as the threshold through the ‘DehumSwitch’. Thus, if the difference is greater than 5%, actual equations are considered for the determination of outlet conditions in terms of the temperature and absolute humidity ratio, otherwise the average of inlet values would be the outlet condition. Consequently, all the control strategies of the developed desiccant wheel model are based on the real scenarios which are essential to analyze the wheel performance under real transient operating conditions. The graphical representation of the desiccant wheel model is shown in Figure 5.8.

![Flow chart of desiccant wheel model control strategies](image)

**Figure 5.7: Flow chart of desiccant wheel model control strategies**

### 5.5 Model parametric performance analysis

The performance of the desiccant wheel is affected by several variables [98]. The key factors include rotation speed along with the inlet air temperature, moisture content, and
face velocity for both process and regeneration air. The effects of varying operating parameters are published in different studies [176] [180] [150]. In the current work, the results of the developed desiccant wheel model are in good agreement with the published data.

For the performance analysis, the thermodynamic and geometric properties of the desiccant wheel used in the simulation process are defined in Table 5.1. The variation ranges of operating parameters considered in the analysis are listed in Table 5.2.

![Dymola graphical representation of the desiccant wheel model](image)

**Figure 5.8: Dymola graphical representation of the desiccant wheel model**

**Table 5.1: Thermodynamic properties and geometric parameters of the desiccant wheel used for the performance analysis**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Desiccant material</td>
<td>Silica Gel- Type A</td>
</tr>
<tr>
<td>Wheel diameter, D (mm)</td>
<td>320</td>
</tr>
<tr>
<td>Wheel length, L (mm)</td>
<td>200</td>
</tr>
<tr>
<td>Overall cross-sectional area, A (m²)</td>
<td>0.08</td>
</tr>
<tr>
<td>Parameters</td>
<td>Baseline values</td>
</tr>
<tr>
<td>---------------------------------------------------------------------------</td>
<td>-----------------</td>
</tr>
<tr>
<td>Temperature of ambient air, $T_{amb}$ ($^\circ$C)</td>
<td>30</td>
</tr>
<tr>
<td>Humidity ratio of ambient air, $\omega_{amb}$ (g/kg)</td>
<td>10</td>
</tr>
<tr>
<td>Inlet temperature of process air, $T_{sin}$ ($^\circ$C)</td>
<td>31</td>
</tr>
<tr>
<td>Inlet humidity ratio of process air, $\omega_{sin}$ (g/kg)</td>
<td>10.3</td>
</tr>
<tr>
<td>Flow rate of process air, $m$ (m/s)</td>
<td>2</td>
</tr>
<tr>
<td>Regeneration temperature, $T_{reg}$ ($^\circ$C)</td>
<td>80.5</td>
</tr>
<tr>
<td>Inlet humidity ratio of regeneration air, $\omega_{reg}$ (g/kg)</td>
<td>12.5</td>
</tr>
<tr>
<td>Rotation speed, $N$ (rph)</td>
<td>36</td>
</tr>
</tbody>
</table>

Table 5.2: Range of operating parameters for the performance analysis

5.5.1 Effect of process air inlet conditions

The key operating parameters related to the inlet conditions of the process air include the air velocity, temperature $T_{sin}$ and humidity ratio $\omega_{sin}$. The influence of these parameters on the outlet humidity along with moisture removal from the process air is shown in Figure 5.9. In addition, the inlet humidity ratio also affects the outlet temperature of the process air.

It can be observed from Figure 5.9A that the process air velocity significantly affects the outlet humidity ratio. This ratio increases with rising air velocity and moisture removal decrease. Thus, air that passes through the wheel the higher velocity is dried less deeply as compared to low air velocity. Therefore, if the process air is required to be dried more at the slower air velocity, then the size of the wheel must be increased. Similarly, the $\omega_{sin}$ of process air also influences the outlet humidity ratio as shown in Figure 5.9B. If air of a higher humidity is entering then the air leaving the wheel will also be more humid. Likewise, Figure 5.9C shows that the desiccant wheel has better dehumidification performance with the lower $T_{sin}$ as it causes higher moisture removal. In addition, the outlet temperature of process air is also affected by the $\omega_{sin}$ as shown in
Figure 5.9D. The outlet temperature rises if more moisture is removed i.e. due to higher inlet humidity. The increase in temperature is roughly proportional to the increase in moisture removal. The overall impact of the inlet humidity ratio of process air on the outlet conditions for inlet temperatures of 25 °C and 35 °C is presented in Figure 5.10.

Figure 5.9: Effect of face velocity, inlet humidity ratio, inlet temperature of process air on process outlet humidity ratio (A), (B), (C); effect of process inlet humidity ratio on process outlet temperature (D)

Figure 5.10: Effect of process inlet conditions on process air outlet conditions
5.5.2 Effect of regeneration air inlet conditions

On the regeneration side, the performance of a desiccant wheel is mainly affected by the regeneration temperature $T_{\text{reg}}$ and humidity ratio $\omega_{\text{reg}}$ as shown in Figure 5.11. Increasing $T_{\text{reg}}$ increases moisture removal and in turns a lower outlet humidity ratio of process air as shown in Figure 5.11A. However, higher $T_{\text{reg}}$ requires more energy when it rises above about 90 °C that is more significant than increase in moisture removal. Contrary to the former aspect, the more humid air at the inlet of regeneration air decreases the dehumidification of the process air due to reduced absolute moisture difference as shown in Figure 5.11B. Moreover, Figure 5.11C shows that the high $T_{\text{reg}}$ also results in high outlet temperature of process air. This is due to the rise in the heat of adsorption with more latent-to-sensible heat conversion.

5.5.3 Effect of rotation speed

The rotation speed $N$ of desiccant wheel significantly affects the wheel dehumidification performance as shown in Figure 5.12. Ideally the desiccant wheel will always operate at its optimal rotation speed $N_{\text{opt}}$ at which maximum moisture removal is achieved. If the wheel rotates at high speed then the process side of the wheel does not have enough time to remove the moisture. The Figure 5.12A, B shows that $N_{\text{opt}}$ increases with respect to $\omega_{\text{sin}}$. The higher inlet humidity ratio improves mass transfer capacity that reduces the time to reach adsorption equilibrium which causes high $N_{\text{opt}}$.

![Figure 5.11: Effect of inlet humidity ratio and regeneration temperature on process outlet humidity ratio (A) and (B), respectively; effect of regeneration temperature on outlet temperature (C)](image-url)
In addition, the low $T_{\text{sin}}$ is more favorable for high dehumidification at low rotation speeds compared to high $T_{\text{sin}}$ as shown in Figure 5.12C. At high $T_{\text{sin}}$, the wheel is required to rotate at high speed to avoid an equilibrium state of the adsorbent material. However, an increase in $N$ above $N_{\text{opt}}$ does not cause any further dehumidification as the equilibrium state had already been reached. Moreover, $N_{\text{opt}}$ also increases with rising $T_{\text{reg}}$ as shown in Figure 5.12D, E. $N_{\text{opt}}$ increases from 11 rph to 39 rph when $T_{\text{reg}}$ changes from 45$^\circ$C to 110$^\circ$C when all other parameters are kept constant. At high $T_{\text{reg}}$, it is easier to desorb the adsorbent. Thus, the wheel rotation speed is increased for rapid movement of the regeneration zone towards the process side.

Figure 5.12: Effect of process inlet humidity ratio on optimal rotation speed (A); effect of rotation speed on process outlet humidity ratio with process humidity ratio (B), and process inlet temperature (C) as parameter; effect of regeneration temperature on optimal rotation speed (D), and effect of rotation speed on process outlet humidity ratio with regeneration temperature as parameter, (E).
5.6 Model Calibration and Validation

Performance analysis of energy recovery devices is critical to predict the energy transfer in heating, ventilation and air-conditioning (HVAC) design applications. However, such analysis under the test conditions of ARI Standard 1060-2003 [152] has proven to be very expensive and prone to large uncertainties [181] [182]. In the ARI standard, two inlet conditions for summer and winter are used to certify the performance of air-to-air recovery devices [153]. Since the variations in the ambient air conditions are significant over the year, HVAC designers need to be concerned with the optimal design and system performance to minimize the system cost and energy requirements. Furthermore, the performance characterization of energy wheels is more difficult than air-to-air heat exchangers due to the coupled heat and mass transfer aspects associated with energy wheels and the need to keep steady state balances of air flow, water vapor flow and energy. In such a situation, it is challenging and time consuming to maintain small uncertainties for all the energy wheel parameters [181]. Several studies for performance analysis of energy exchangers are performed to determine the effectiveness of heat exchangers under steady state operating conditions [183] [170] [171]. However, the analysis of the desiccant wheel performance under transient operating conditions is seldom studied [181]. For example, a transient test method was proposed based on an experimental setup to determine the transient response of energy wheels [181] [182]. In addition, the effect of humidity and temperature sensors transient characteristics are also analyzed to measure the performance of energy wheels. It was concluded that the performance analysis of energy wheels based on the transient response is essential for the optimal performance in real life operation. Therefore, in the current research, the developed desiccant wheel is calibrated and validated under real transient operating conditions.

5.6.1 Model Calibration

The desiccant wheel model is calibrated for the identification of design parameters under the transient operating conditions. Such key design parameters are often not provided from the manufacturer. The considered transient operating conditions are based on the monitoring data of a commercial desiccant cooling system installed in the ‘ENERGYbase’ office building in Vienna, Austria [184].

5.6.1.1 Real system description

The ‘ENERGYbase’ was designed as a passive house building with respect to the architectural building conceptions. The conditioned area of the building is 7,025 m². The energy system design targets three aspects: (1) HVAC system with high level energy efficiency, (2) Increased use of renewable energy resources, (3) High inside comfort level. Consequently, the desiccant cooling system (DCS) of the building is solar thermally driven, composed of two identical units, each design for an 8,240 m³/h air flow rate. The DCS consists of a desiccant wheel, a heat wheel, and two spray
dehumidifiers’ with one on the supply and one on the return sides. The DCS normally operates for 12 hours a day from 7:00 to 19:00, 5 days a week. The whole building is equipped with over 500 sensors to continuously measure all relevant parameters of the building HVAC system and indoor air conditions. The DCS is monitored and controlled with two data monitoring tools: 1. DESIGO [185] and 2. JEVis [186]. The desiccant wheel is equipped with 9 sensors as shown in Figure 5.13. Four sensors measure temperature and the other four are the relative humidity sensors installed at the inlet and outlet of process and regeneration air. One sensor monitors the rotation speed of the wheel. The details of the desiccant wheel sensors in terms of their accuracy and operating range are available in the manufacturer user guide [187]. In addition, the other operating conditions, such as flow rates of process and regeneration air are also acquired from the buildings monitoring system. The desiccant wheel can operate in either dehumidification or enthalpy mode based on the operating conditions. For the dehumidification mode, the maximum rotation speed is 20 rph and in enthalpy mode wheel can be rotated up to 10 rph. The details provided by the wheel manufacturer only include basic properties such as adsorbent material (LiCl), wheel diameter (1680 mm) and length (450mm). Consequently, the developed model is calibrated for the identification of design parameters of the installed wheel. The wheel design parameters identified through calibration include the thermodynamic and structure properties, such as constants of adsorption isotherm, surface diffusivity, specific surface area, bulk wheel density, and density of adsorbent. The variation range of all design parameters used in the calibration phase is shown in Table 5.3. The variation range is based on the available data of different commercial wheels and the thermodynamic properties of LiCl [188].

![Figure 5.13: Desiccant wheel temperature, relative humidity and rotation speed sensors](image)

In the current work, the calibration is executed through optimization process by coupling the Dymola/Modelica with an optimization tool, GenOpt [132]. In such an approach, GenOpt used the defined set of design parameters of desiccant wheel to provide inputs to the Dymola simulation program. Meanwhile, Dymola provided text files with values of the target functions essential for optimization. The advantages of such approach are the capabilities of GenOpt to utilize multi-core processing units for
improved optimization performance. The Discrete Armijo Gradient optimization algorithm of generalized pattern search methods is applied with the typical values of the algorithm parameters [132]. The transient operating conditions from the monitoring data of the ‘ENERGYbase’ system are appropriately provided to the desiccant wheel model. The mean squared error (MSE) between the monitored and simulated outlet specific enthalpy of process air is considered as a calibration objective function. Figure 5.14 depicts the process flow chart of the desiccant wheel model calibration and validation.

Table 5.3: Variation range of calibration parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Initial values</th>
<th>Parametric variations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific area of wheel, $A_{sp}$ (m$^2$/m$^3$)</td>
<td>2500</td>
<td>1000-5000</td>
</tr>
<tr>
<td>Bulk density of wheel, $\rho_b$ (kg/ m$^3$)</td>
<td>200</td>
<td>150-1000</td>
</tr>
<tr>
<td>Density of adsorbent, $\rho_{aw}$ (kg/ m$^3$)</td>
<td>1150</td>
<td>700-2000</td>
</tr>
<tr>
<td>Diffusion co-efficient, $D_0$ (---)</td>
<td>$1.5 \times 10^{-7}$</td>
<td>$1 \times 10^{-9} - 6 \times 10^{-7}$</td>
</tr>
<tr>
<td>Activation energy, $E_{ai}$ (kJ/kg)</td>
<td>1800</td>
<td>1500-3500</td>
</tr>
<tr>
<td>Adsorption co-efficient, $C_F$ (---)</td>
<td>0.15</td>
<td>0.1-10</td>
</tr>
<tr>
<td>Adsorption co-efficient, $n_F$ (---)</td>
<td>1</td>
<td>1.5-10</td>
</tr>
<tr>
<td>Humidity correction factor, $K_w$ (---)</td>
<td>0.01</td>
<td>0-0.05</td>
</tr>
<tr>
<td>Temperature correction factor, $K_t$ (---)</td>
<td>0.01</td>
<td>-0.01-0.05</td>
</tr>
<tr>
<td>Sensible and latent effectiveness, $\epsilon_{sn}$, $\epsilon_{lt}$ (---)</td>
<td>0.6</td>
<td>0.5-0.9</td>
</tr>
<tr>
<td>Time delay of temperature sensors, $\tau_1$ (---)</td>
<td>1</td>
<td>1-2000</td>
</tr>
<tr>
<td>Time delay of humidity sensors, $\tau_2$ (---)</td>
<td>1</td>
<td>1-2000</td>
</tr>
</tbody>
</table>
5.6.1.2 Model calibration data

The actual desiccant wheel and the developed model can operate in both dehumidification and enthalpy mode depending on the outdoor conditions. Consequently, a transient data set for the calibration of the model is selected which allows the wheel to work in both modes. The transient data set consists of 12 hours with 1 minute interval of six days of three months is acquired from the monitoring system to calibrate a model under the wide range of operating conditions. The real building system performance is affected by various factors, such as equipment installation, deterioration with time, sensor errors, and monitoring and controlling bugs. Therefore, the calibration of the wheel model under real transient measurements is more reliable compared to the steady state test bed conditions. The real transient conditions represent the actual operation of individual components and the whole system. In addition, the real measurements also depict the actual performance of the monitoring and controlling
devices, such as different types of sensors including the measurement uncertainties. Thus, the current calibration and validation task of the wheel model is performed under the transient measurements.

For the model calibration, extensive simulations are performed through coupling Dymola with GenOpt for all selected days. Each calibration error is based on the transient data of each particular day. The calibration process is completed in three stages due to the sensitivity of the design parameters. In the first stage, the wheel design parameters are considered. The second stage is related with the sensor delay parameters. In the last stage, temperature and humidity correction coefficients are calibrated. The resultant values of all calibrated parameters are given in Table 5.4. In addition, the values of the desiccant wheel design parameters identified by the optimization algorithm during the calibration process are also given in Table 5.4. The calibration RMSE of each day along with the ambient conditions for 12 hour system operation is given in Table 5.5.

Table 5.4: The values of calibration parameters determined by the optimization algorithm

<table>
<thead>
<tr>
<th>Cal. Stage</th>
<th>Parameters</th>
<th>30th April</th>
<th>19th June</th>
<th>29th June</th>
<th>02nd July</th>
<th>06 July</th>
<th>09 July</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st stage</td>
<td>( A_{sp} )</td>
<td>3021.032</td>
<td>3134.877</td>
<td>3085.816</td>
<td>3075.673</td>
<td>3021.893</td>
<td>3021.145</td>
</tr>
<tr>
<td></td>
<td>( \rho_b )</td>
<td>245.966</td>
<td>241.496</td>
<td>244.359</td>
<td>245.020</td>
<td>248.122</td>
<td>247.938</td>
</tr>
<tr>
<td></td>
<td>( \rho_{dw} )</td>
<td>1201.010</td>
<td>1203.479</td>
<td>1202.257</td>
<td>1202.169</td>
<td>1200.668</td>
<td>1200.681</td>
</tr>
<tr>
<td></td>
<td>( D_0 )</td>
<td>2.303E-7</td>
<td>2.623E-7</td>
<td>2.494E-7</td>
<td>2.506E-7</td>
<td>2.323E-7</td>
<td>2.319E-7</td>
</tr>
<tr>
<td></td>
<td>( E_{st} )</td>
<td>1729.769</td>
<td>1729.439</td>
<td>1729.631</td>
<td>1729.670</td>
<td>1729.883</td>
<td>1729.875</td>
</tr>
<tr>
<td></td>
<td>( C_F )</td>
<td>0.226</td>
<td>0.312</td>
<td>0.283</td>
<td>0.281</td>
<td>0.247</td>
<td>0.245</td>
</tr>
<tr>
<td></td>
<td>( n_F )</td>
<td>1.506</td>
<td>1.506</td>
<td>1.503</td>
<td>1.502</td>
<td>1.501</td>
<td>1.501</td>
</tr>
<tr>
<td></td>
<td>( \epsilon_{sn} )</td>
<td>0.892</td>
<td>0.700</td>
<td>0.697</td>
<td>0.691</td>
<td>0.698</td>
<td>0.699</td>
</tr>
<tr>
<td></td>
<td>( \epsilon_{lt} )</td>
<td>0.600</td>
<td>0.698</td>
<td>0.692</td>
<td>0.698</td>
<td>0.691</td>
<td>0.694</td>
</tr>
<tr>
<td>2nd stage</td>
<td>( \tau_1 )</td>
<td>451.223</td>
<td>431.698</td>
<td>384.877</td>
<td>313.343</td>
<td>329.626</td>
<td>417.335</td>
</tr>
<tr>
<td></td>
<td>( \tau_2 )</td>
<td>3.962</td>
<td>122.273</td>
<td>38.040</td>
<td>27.152</td>
<td>50.193</td>
<td>63.030</td>
</tr>
<tr>
<td></td>
<td>( T_{reg_min} )</td>
<td>312.841</td>
<td>312.720</td>
<td>311.966</td>
<td>312.815</td>
<td>311.836</td>
<td>312.708</td>
</tr>
<tr>
<td>3rd stage</td>
<td>( K_w )</td>
<td>0.0099</td>
<td>0.0099</td>
<td>0.0099</td>
<td>0.0099</td>
<td>0.0099</td>
<td>0.0099</td>
</tr>
<tr>
<td></td>
<td>( K_t )</td>
<td>-4.259E-4</td>
<td>-9.748E-4</td>
<td>-9.544E-4</td>
<td>-8.706E-4</td>
<td>-9.635E-4</td>
<td>-8.898E-4</td>
</tr>
</tbody>
</table>

Furthermore, the calibration results of an example day, i.e. 02\(^{th}\) July, 2012 in terms of monitored and simulated values of the process outlet specific enthalpy, temperature and humidity ratio are shown in Figure 5.15. The Figure also represents the transient conditions of inlet temperature and absolute humidity of process and regeneration air for the same day.
Table 5.5: Ambient conditions and calibration errors of different days

<table>
<thead>
<tr>
<th>Calibration Day</th>
<th>$T_{\text{avg}}$ ($^\circ$C)</th>
<th>$(T_{\text{min}}, T_{\text{max}})$</th>
<th>$\phi_{\text{avg}}$ ($\phi_{\text{min}}, \phi_{\text{max}}$)</th>
<th>$\omega_{\text{avg}}$ ($\omega_{\text{min}}, \omega_{\text{max}}$) (g/kg)</th>
<th>Calibration Error (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30th April</td>
<td>28.0 (15, 33.1)</td>
<td>32.0 (14.4, 94.5)</td>
<td>7.5 (4.1, 9.9)</td>
<td>2.8</td>
<td></td>
</tr>
<tr>
<td>19th June</td>
<td>31.8 (23.7, 35.3)</td>
<td>46.7 (33.9, 77.7)</td>
<td>13.6 (10.9, 14.1)</td>
<td>1.0</td>
<td></td>
</tr>
<tr>
<td>29th June</td>
<td>30.6 (21.2, 36.1)</td>
<td>53.7 (32.4, 100)</td>
<td>14.6 (11.1, 16.7)</td>
<td>1.2</td>
<td></td>
</tr>
<tr>
<td>02nd July</td>
<td>32.5 (21.6, 36.6)</td>
<td>47.1 (27.5, 100)</td>
<td>14.3 (10.1, 17.2)</td>
<td>1.3</td>
<td></td>
</tr>
<tr>
<td>06th July</td>
<td>31.6 (23.2, 36.7)</td>
<td>47.6 (29.8, 85.1)</td>
<td>13.7 (10.9, 17.5)</td>
<td>1.2</td>
<td></td>
</tr>
<tr>
<td>09th July</td>
<td>28.1 (21, 30.1)</td>
<td>62.2 (40.4, 100)</td>
<td>14.6 (10.1, 20.3)</td>
<td>0.8</td>
<td></td>
</tr>
</tbody>
</table>

Figure 5.15: Comparison between monitored and simulated results of process air outlet: specific enthalpy (A), temperature (B), and absolute humidity (C)

5.6.2 Model Validation

The calibrated desiccant wheel model is validated against the transient measurements of all other days except the calibration day by considering MSE between the monitored and simulated outlet specific enthalpy of process air as an objective function. The model is validated against the transient measurements of five different days except the calibration day. For example, the desiccant wheel design parameters identified during the calibration based on 30th April are used to validate the model to determine the
RMSE against the transient operating conditions of five other days. In addition, the mean percentage error (MPE) is also calculated based on the simulated and measured specific enthalpy at the process air outlet. The validation RMSE and MPE of all validation days inclusive of the respective calibration are given in Table 5.6. The resultant minimum average validation error is 1.9 on four days i.e. 29th June, 2nd, 6th, and 9th July, 2012 and the maximum error is 3.6 on 30th April, 2012. However, the minimum and maximum MPE are 0.2% and 4.6%. The predicted results and monitored values with the minimum and maximum validation errors of 2nd July and 30th April, respectively in terms of the outlet specific enthalpy, temperature and humidity ratio of process are shown in Figure 5.16.

The percent of measurement points within the 5% MPE over all 6 measurement days were calculated with respect to the selected calibration days, as follows: The parameters resulting from the calibration on 29th June will yield simulation results within 5% of the measured values 84% of the time over the 6 days taken into consideration. Thus on average 10 out of 12 hours per day have less than 5% MPE. Similarly the percentage with 5% MPE is 46%, 77%, 83%, 82%, and 82% for 30th April, 19th June, 02nd, 06th, and 09th July, respectively. Therefore, it can be concluded that the high percentage of the considered transient measurements have MPE of less than 5%. Consequently, the desiccant wheel model has improved accuracy to predict the specific enthalpy, temperature and humidity ratio of the outlet conditions under a wide range of transient operating conditions.

Table 5.6: Validation errors of different days

<table>
<thead>
<tr>
<th>Calibration Day</th>
<th>Validation Error, RMSE [kJ/kg] and MPE (%)</th>
<th>Average Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>30th April</td>
<td>2.8 (4.6) 4.3 (3.8) 4.1 (4.8) 3.9 (5.4) 3.4 (4.7) 3.0 (4.5)</td>
<td>3.6 (4.6)</td>
</tr>
<tr>
<td>19th June</td>
<td>4.6 (3.8) 1.0 (0.3) 1.7 (-0.7) 1.9 (-1.2) 2.3 (-2.4) 2.8 (-3.8)</td>
<td>2.4 (-0.7)</td>
</tr>
<tr>
<td>29th June</td>
<td>4.4 (4.2) 1.2 (1.0) 1.2 (0.1) 1.3 (-0.2) 1.6 (-1.4) 1.9 (-2.5)</td>
<td>1.9 (0.2)</td>
</tr>
<tr>
<td>02nd July</td>
<td>4.3 (4.5) 1.5 (1.4) 1.4 (0.7) 1.3 (0.4) 1.3 (-0.6) 1.5 (-1.8)</td>
<td>1.9 (0.8)</td>
</tr>
<tr>
<td>06th July</td>
<td>4.1 (4.7) 1.9 (1.9) 1.7 (1.4) 1.5 (1.2) 1.2 (0.2) 1.0 (-0.6)</td>
<td>1.9 (1.5)</td>
</tr>
<tr>
<td>09th July</td>
<td>4.0 (5.0) 2.2 (2.3) 1.8 (1.8) 1.6 (1.7) 1.1 (0.7) 0.8 (0.0)</td>
<td>1.9 (1.9)</td>
</tr>
</tbody>
</table>

5.6.3 Validation of optimal rotation speed

The concept of estimating the optimal rotation speed is also validated by comparing with the experimental results. The predictions of the N_{opt} are carried out under wide range of operating conditions at different regeneration temperatures as shown in Figure 5.17. The considered desiccant wheel design parameters are given in Table 5.1. The results showed that the predicted values of N_{opt} are in agreement with the experimental results [175] with an acceptable difference. Therefore, it can be concluded that the developed desiccant wheel model can also predict N_{opt} with good accuracy in most conditions.
5.7 Model calibration criteria

It was analyzed during the model calibration and validation that the errors are influenced by the ambient conditions as shown in Table 4. Therefore a criterion was defined for appropriate calibration and validation of the model in the different climate conditions. The effects of ambient conditions on the calibration and validation errors are shown in Figure 5.18. The results showed that both calibration and validation errors are less at the high ambient absolute humidity ratio. Therefore, it is better to perform the model calibration and validation at high outdoor absolute humidity.

![Comparison between the simulated and monitored results](image)

**Figure 5.16:** Comparison between the simulated and monitored results: process air outlet specific enthalpy, and temperature and humidity ratio with minimum validation error (A), (C), respectively, and with maximum validation error (B), (D).
5.8 Conclusions

In chapter 5, a desiccant wheel model is developed in an equation-based object-oriented environment, Dymola/Modelica for performance prediction in the real operation. The
model can estimates the actual and optimal operation under wide range of operating conditions with respect to actual and optimal wheel rotation speed. Additionally, the developed wheel model is also capable of predicting accurately the wheel operation in dehumidification and enthalpy modes.

The model is calibrated and validated under the transient measurements of six days of a commercial desiccant wheel installed in ENERGYbase. The mean squared error (MSE) between the monitored and simulated outlet specific enthalpy of process air is considered as a calibration and validation objective function. The resultant maximum and minimum validation RMSE and MPE between the simulated and measured data of the considered transient measurements are 3.6kJ/kg, 1.9kJ/kg, 4.6% and 0.2%, respectively. The calibration and validation results showed that the model is able to predict the performance of desiccant wheels with good accuracy. The effects of the inlet conditions of process and regeneration air are also analyzed through the model and the resultant trends are in agreement with the published data.
Chapter 06

6 OPTIMIZATION OF CHILLED WATER SYSTEM CONFIGURATIONS

The chilled water systems consist of primary HVAC system components, such as chillers, cooling towers and pumps and account for most of the energy use in HVAC systems. The optimization of chilled water at the initial design stage can significantly enhance the overall energy saving potential of HVAC systems. However, the optimization of a chilled water system is a multifaceted task with respect to system optimization either at the design or configuration level. The combination of both levels makes the process more complex. Therefore, in the current chapter, a novel simulation-based optimization approach described in chapter 4 is implemented and validated through a real chilled water system for the automated optimization of chilled water systems.

In addition, the study presents an incremental development of the methodology for a chilled water system design optimization. Initially, the system configuration parameters are varied with fixed design conditions to confirm the established best practice design criteria, followed by a comprehensive system design optimization. The implemented simulation-based optimization approach couples the Dymola/Modelica dynamic modeling and simulation program with GenOpt generic optimization program to find optimal system configuration. A dynamic system model is developed to vary and simulate different chilled water system configurations. Optimization of the chilled water system is achieved at both design and configuration level using five design variables. Two discrete variables are related to the system configuration, number of chillers and cooling towers. Also three continuous variables relating to system design are building load demand, temperature difference across the condenser, and cooling tower fan speed. The proposed approach is applied in three different strategies: (1) use fixed system design conditions to validate the methodology in comparison with the real system, (2) in addition, vary the number of cooling towers according to the cooling tower flow turndown ratio to validate the best practice design criteria, and (3) the overall system optimization approach varying the system design and configuration parameters.

6.1 Description of investigated chilled water system

The chilled water system considered in the current study serves a large office building in Southern California [66]. A schematic of the system is shown Figure 6.1. As Table 6.1 illustrates, the system contains three equally sized chillers rated at a cooling capacity of 2725 kW (775 tons) each. The system has five equally sized draw-through cross-flow cooling tower cells, designed for approximately 76 l/s (1200 gpm). Three equally sized condenser water pumps, designed for 111 l/s (1760 gpm) with 19 kW (25 hp) motors, are piped together in a headered arrangement so that each pump can serve any of the
chillers or tower cells. Similarly, three equally sized chilled water pumps, designed for 57 l/s (900 gpm) with 30 kW (40hp) are also headered. In the current study, the design conditions are based on the typical operating conditions specified by the Air Conditioning and Refrigeration Institute (ARI) standard 550/590-2003 [189] and Cooling Tower Institute’s (CTI) test conditions [190]. Design wet bulb temperature of 17°C [63°F] and dry bulb temperature of 26°C [78.8°F] are used for cooling tower model according to the ASHRAE standard 90.1-2004 climate data for San Francisco [191].

![Schematic of the referred chilled water system](image)

**Figure 6.1:** Schematic of the referred chilled water system

**Table 6.1:** Specifications of the investigated chilled water system

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total cooling capacity kW [tons]</td>
<td>8175 [2325]</td>
</tr>
<tr>
<td>(Three identical sets of chillers, pumps, and cooling towers)</td>
<td></td>
</tr>
<tr>
<td>Chillers:</td>
<td></td>
</tr>
<tr>
<td>Compressor type</td>
<td>Centrifugal</td>
</tr>
<tr>
<td>Nominal cooling capacity kW [tons]</td>
<td>2725 [775]</td>
</tr>
<tr>
<td>Nominal compressor power kW [tons]</td>
<td>446.4 [127]</td>
</tr>
<tr>
<td>Minimum cooling capacity kW [tons]</td>
<td>457 [130]</td>
</tr>
<tr>
<td>Design COP</td>
<td>6.1</td>
</tr>
<tr>
<td>Design chilled water supply/return temperature °C[°F]</td>
<td>6.7/12.2 [44/54]</td>
</tr>
<tr>
<td>Design chilled water flow rate l/s [gpm]</td>
<td>57 [900]</td>
</tr>
<tr>
<td>Design condenser water entering temperature °C[°F]</td>
<td>29.4 [85]</td>
</tr>
<tr>
<td>Design condenser water flow rate l/s [gpm]</td>
<td>111[1760]</td>
</tr>
<tr>
<td>Cooling towers:</td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>Draw-through</td>
</tr>
</tbody>
</table>
### Water flow rate l/s [gpm]

- 76 [1200]

### Fan motor power kW [hp]

- 18.65 [25]

### Design wet bulb temperature °C[°F]

- 17 [62.6]

### Design dry bulb temperature °C[°F]

- 26 [78.8]

### Design approach temperatures °C[°F]

- 8.3[15]

### Design range temperature °C[°F]

- 5.56 [10]

### Pumps:

- Rated power of each chilled water pump kW [hp]: 30 [40]
- Rated power of each condenser water pump kW [hp]: 19 [25]

### 6.2 Model development of the system

In current approach, the equation-based object-oriented (EOO) chilled water system model is developed in Dymola/Modelica. Component models for the investigated chilled water systems were used from the Modelica standard and LBL Buildings libraries.

#### 6.2.1 Description of component models

The chilled water system consists of chillers, cooling towers, and chilled water and condenser water pumps. The components model used in the chilled water system model are briefly described here.

- **Chiller component model**

  The Centrifugal chiller component model, as shown in Figure 6.2 is developed by AIT (Austrian Institute of Technology) and is based on TRNYSY centrifugal chiller model TYPE 68 [37]. The chiller component model was operational in three modes: (1) shutdown mode, if the applied $Q_{load}$ is less than the minimum load, $Q_{min}$, specified in the model, (2) normal mode, if applied load is more than the specified minimum and less than the specified maximum load, and (3) overload mode, if $Q_{load}$ is more than the specified maximum load, $Q_{max}$. The modes enable the chiller model to properly react during the simulation to the applied load. In the shutdown mode, the temperatures of the water leaving the condenser and evaporator become equal to the corresponding temperatures and total power consumption, condenser heat transfer rate, and COP are set to zero. In the overload mode, $Q_{load}$ is set to $Q_{max}$ and is the new water temperature of the evaporating water and is used to determine the COP. As such, the modes are needed for the simulation control, rather than a real system, in order to prevent the chiller model from coming up with unrealistic chilled water supply temperatures to meet the cooling demand. In the current study, the chiller is always operated in the normal mode. Therefore, the number of chillers satisfying the normal operating mode requirements was varied. The Equation 6.1 is used to calculate the chiller’s power requirements.
\[
\frac{P_{\text{req}}}{P_{\text{des}}} = \left[ 0.140 + 0.544 \left( \frac{Q_{\text{load}}}{Q_{\text{des}}} \right) + 0.316 \left( \frac{Q_{\text{load}}}{Q_{\text{des}}} \right)^2 \right] \times \left[ 1 + 0.012 \left( ECWT - DECWT \right) - 0.015 \left( LEWT - DLEWT \right) \right]
\]  

(6.1)

Where \( P_{\text{req}} \) and \( P_{\text{des}} \) are the actual and design power requirements, \( Q_{\text{load}} \) and \( Q_{\text{des}} \) are the actual and design loads, \( ECWT \) and \( DECWT \) are the actual and design entering condenser water temperatures, and \( LEWT \) and \( DLEWT \) are actual and design leaving evaporative water temperatures [192]. The temperatures are in degrees Fahrenheit.

- Cooling tower component model

The cooling tower component model is developed by LBL [136] and used from the ‘Buildings’ library. The model represents a steady state cooling tower with variable speed fan using a ‘York’ cooling tower performance curve to compute the approach temperature. The ambient conditions are applied through an input port in which the ambient dry or wet bulb temperature can be applied as shown in Figure 6.3. However, another component model is used to calculate the wet bulb temperature. In addition, the cooling tower operates in free convection mode when the fan control variable is zero. The fan power consumption is proportional to the power of the control variable cubed.

![Figure 6.2: Centrifugal chiller component model](image)

![Figure 6.3: Cooling tower component model](image)

- The chilled water and condenser water pump component model

The prescribed mass flow rate pump component model from ‘Buildings’ library is used for both chilled and condenser water flows. The efficiency of the device is computed based on the efficiency curves that take as an argument the actual volume flow rate divided by the maximum possible volume flow rate. The Figure 6.4 is a graphical presentation of the pump component model.
6.2.2 Model development

The design parameters of all components model are specified according to the specifications of the investigated chilled water system defined in Table 6.1. Each component model is properly connected to another component model in parallel arrangement based on the inlet and outlet fluid ports. Additionally, the input ports at the interface of each component model are also assigned appropriate values with respect to the system design.

Centrifugal chillers were operated with fixed chilled water supply and return temperatures at 6.7°C [44°F] and 12.5°C [54°F], respectively. Chiller sequencing allowed the running chillers to operate at the same part load conditions. The demand loads, $Q_{load}$, were applied from an external text file with an increment of 35.16 kW [10tons] up to the peak load demand. Each applied load determined the evaporator mass flow rate assuming a fixed design temperature difference of 5.55°C [10°F] across the evaporator. Thus load-based mass flow rates were applied through the prescribed flow chilled water pumps on the evaporator side of the chillers. Similarly, the condenser water flow rate was determined to satisfy the required amount of heat rejection by the condenser. Such heat rejection was computed from the chiller model based on the applied load, $Q_{load}$. The condenser water flow rates were calculated by keeping the fixed design temperature difference across the condenser the same according to the calculated heat rejection rate for the first and second strategies, while the third strategy additionally varied the temperature differences across the condenser. The number of cooling towers was determined based on the calculated condenser water flow rate. The cooling towers operated with fixed design conditions in terms of design range and approach temperatures given in Table 1. In the first strategy, the number of cooling towers varied according to the real considered system, but in the remaining strategies cooling towers were varied according to the flow turndown ratio up to one-third of the design [193]. According to such criterion, the maximum number of cooling towers in the current study could be 25 provided there was an appropriate temperature difference across the condenser at higher loads. Such a high number of cooling towers is modeled as an array of parallel components with identical design parameters as described in chapter 4.

Consequently, the Dymola/Modelica model was capable of varying the number of chillers and cooling towers as well as the temperature difference across the condenser,
the cooling tower fan input signal, and part load value from an external text file, in order to find optimal chilled water system configuration setup. Figure 6.5 depicts a graphical representation of the overall chilled water system.

Figure 6.5: Graphical representation of the chilled water system model in Dymola

6.3 Objective Function

The objective function is defined in the current study as the total system power consumption for system design optimization. However the initial cost of each system configuration is also an important aspect for system economics. For the optimal configuration design, the objective function is minimal while satisfying the building load demand. Equation 6.2 computes the objective function within the Dymola/Modelica model for various system configurations considering the power consumption of a specific number of chillers (chiller.PTot), chilled water pumps (CHPump.PElc), condenser water pumps (CTPump.PElc), and cooling towers (CoolingTower.PFan) against the fixed load demand.

\[ P_{total} = CHPump.PElc + CTPump.PElc + CH \times Chiller.P_{Tot} + CT \times CoolingTower.PFan \] (6.2)

Where CH is the number of chillers and CT is the number of cooling towers.
6.3.1 Economics of system configuration

The initial cost of each configuration involves the cost of chillers, cooling towers, and cost of piping and labor/fitting/valve required for efficient hydronic design. However, the cost of such equipment varies country to country and region to region, therefore the costs are estimated based on the data published by selected manufacturers [194] and cost studies related to such equipment [142] [193] [195] [196]. The initial and installation costs of water-cooled centrifugal chillers and cooling towers are estimated based on their unit capacity i.e. cost per ton. However, the piping cost including fitting, and valves of the chilled water system are estimated from Pipe Size Optimization tool [142] [197]. The tool calculates the initial cost of piping based on the flow rate for specified piping segments. In addition, it also accounts for the number and types of various valves and fittings normally used in chilled water systems. The costs of different valves and fittings are also provided based on the respective pipe size. In the current study, the number and type of valves and fittings are decided from the piping schematic of the reference system [66]. The initial and installation costs of water-cooled centrifugal chillers and cooling towers along with the piping/fitting/valve cost are summarized in the Table 6.2. The estimated value of 25% of contractor markup is considered in the calculations [193].

Table 6.2: Summary of initial costs

<table>
<thead>
<tr>
<th>Description</th>
<th>Cost/ton</th>
<th>Total unit cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description</td>
<td>€ [$]</td>
<td>€ [$]</td>
</tr>
<tr>
<td>Water-cooled centrifugal chiller (Nominal 775 tons each)</td>
<td>147 [180]</td>
<td>113925 [139500]</td>
</tr>
<tr>
<td>Chiller installation cost</td>
<td>37 [45]</td>
<td>28370 [34875]</td>
</tr>
<tr>
<td>Cooling Tower (Nominal 400 tons each)</td>
<td>106 [130]</td>
<td>42400 [52000]</td>
</tr>
<tr>
<td>Piping/Fitting/Valve</td>
<td>---</td>
<td>13015 [16000]</td>
</tr>
<tr>
<td>a chilled water side</td>
<td>---</td>
<td>19523 [24000]</td>
</tr>
<tr>
<td>b condenser water side</td>
<td>---</td>
<td>4393 [5400]</td>
</tr>
<tr>
<td>c adding another chiller in the system</td>
<td>---</td>
<td>2440 [3000]</td>
</tr>
<tr>
<td>d adding another cooling tower in the system</td>
<td>---</td>
<td>273591 [335469]</td>
</tr>
<tr>
<td>Contractor Markup</td>
<td>25%</td>
<td></td>
</tr>
<tr>
<td>Estimated total baseline cost for 1 chiller and 1 cooling tower</td>
<td>273591 [335469]</td>
<td></td>
</tr>
<tr>
<td>Estimated total cost for adding each chiller</td>
<td>183360 [224719]</td>
<td></td>
</tr>
<tr>
<td>Estimated total cost for adding each cooling tower</td>
<td>58100 [71250]</td>
<td></td>
</tr>
</tbody>
</table>

As the current study involves different configurations of the chilled water system varying number of chillers and cooling towers, the final estimated initial cost of each configuration is calculated from Equation 6.3.
\[ C_{total} = CH \times CP_{chiller} \times (C_{chiller} + C_{inst.CH}) + CT \times CP_{tower} \times (C_{tower} + C_{inst.CT}) + C_{piping/fitting/valve} \]  

Whereas the piping/fitting/valve cost is calculated by Equation 6.4.

\[ C_{piping/fitting/valve} = C_{CHW side} + C_{CW side} + CH \times C_{add.chiller} + CT \times C_{add.tower} \]  

Where

\[ CH, CT = \text{number of chillers and cooling towers} \]
\[ CP_{chiller}, CP_{tower} = \text{Nominal capacity of chiller and cooling tower, tons} \]
\[ C_{chiller}, C_{tower} = \text{unit initial cost of chiller and cooling tower, cost/ton} \]
\[ C_{inst.CH}, C_{inst.CT} = \text{unit installation cost of chiller and cooling tower, cost/ton} \]
\[ C_{piping/fitting/valve} = \text{total cost of piping/fitting/valve} \]
\[ C_{CHW side}, C_{CW side} = \text{cost of piping/fitting/valve for the chilled water and condenser water side} \]
\[ C_{add.chiller}, C_{add.tower} = \text{cost of piping/fitting/valve for adding chiller and cooling tower in the system} \]

The focus of the current work is development and validation of a methodology for automated selection of optimal configuration design. Therefore, total system power consumption is considered as an objective function. However, the initial cost of each system configuration also impacts the ultimate system selection. Therefore the initial cost estimation of each configuration is also provided for economic considerations.

### 6.4 Optimization Algorithm

As previously mentioned, overall optimization of the chilled water system involves five design variables with the total system power consumption as an objective function. Two discrete design variables are the number of cooling towers CT, and the number of chillers CH. Three continuous design variables are building load demand Q_{load}, temperature difference across the condenser \( \Delta T \), and cooling tower fan speed F. Table 6.3 shows boundaries of these variables.

**Table 6.3: Design variables and boundaries**

<table>
<thead>
<tr>
<th>Tower Fan Speed, F (%)</th>
<th>Temp. difference condenser side, ( \Delta T ) (°C) [°F]</th>
<th>No. of chiller, CH</th>
<th>No. of cooling towers, CT</th>
<th>Building load Q_{load} (kW) [Tons]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum 0.3</td>
<td>3 [5.4]</td>
<td>1</td>
<td>3</td>
<td>1055 [300]</td>
</tr>
<tr>
<td>Maximum 1</td>
<td>15 [27]</td>
<td>3</td>
<td>18</td>
<td>7032 [2000]</td>
</tr>
<tr>
<td>Step 0.01</td>
<td>0.01</td>
<td>1</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Initial 1</td>
<td>3</td>
<td>1</td>
<td>3</td>
<td></td>
</tr>
</tbody>
</table>

GenOpt optimization algorithms are categorized with respect to the problems they are meant to solve. Thus, various algorithms are available for continues, discrete, and mix
of continuous and discrete independent variables. For problems including both continuous and discrete independent variables, such as the current optimization, GenOpt contains Particle Swarm Optimization (PSO) algorithms, and a hybrid Generalized Pattern Search Particle Swarm Optimization with Constriction Coefficient Hooke-Jeeves (GPSPSOCCHJ) algorithm.

The hybrid global optimization algorithm GPSPSOCCHJ consists of a stochastic population-based constriction coefficient PSOCC algorithm and a direct search Hooke-Jeeves (HJ) algorithm. The key advantage of this algorithm is that the global PSO search increases the possibility of getting close to the global minimum rather than achieving only a local minimum, while the HJ algorithm then refines the search locally [21]. The GPSPSOCCHJ algorithm parameters chosen in the current study are summarized in Table 6.4 [130].

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Neighborhood topology</td>
<td>von-Neumann</td>
</tr>
<tr>
<td>Neighborhood size</td>
<td>5</td>
</tr>
<tr>
<td>Number of particles</td>
<td>20</td>
</tr>
<tr>
<td>Number of generations</td>
<td>5</td>
</tr>
<tr>
<td>Seed</td>
<td>1</td>
</tr>
<tr>
<td>Cognitive acceleration</td>
<td>2.8</td>
</tr>
<tr>
<td>Social acceleration</td>
<td>1.3</td>
</tr>
<tr>
<td>Max velocity gain continuous</td>
<td>0.5</td>
</tr>
<tr>
<td>Max velocity discrete</td>
<td>4</td>
</tr>
<tr>
<td>Constriction gain</td>
<td>0.5</td>
</tr>
<tr>
<td>Mesh size divider</td>
<td>2</td>
</tr>
<tr>
<td>Initial mesh size exponent</td>
<td>0</td>
</tr>
<tr>
<td>Mesh size exponent increment</td>
<td>1</td>
</tr>
<tr>
<td>Number of step reductions</td>
<td>4</td>
</tr>
</tbody>
</table>

6.5 Simulation-based Optimization

After the development of the chilled water system model in Dymola/Modelica, the simulation model is coupled with GenOpt to find the optimal values of design parameters which ultimately decide the optimal system configuration. Figure 6.6 shows the implementation of the simulation-based optimization approach. The optimization process repeats iteratively until a minimum of the objective function is found [130].
6.6 Results and Discussion

A wide range of design and configuration factors influence the chilled water system performance. At a system design level, the selection of suitable design conditions plays a crucial role, e.g. nominal water flow and temperature difference across the evaporator and condenser, cooling tower approach and wet bulb temperature. At a system configuration level the decision about the number, type, and size of chillers, pumps, and cooling towers extensively affect the energy performance of chilled water systems.

6.6.1 Baseline System Configurations: 1st Strategy

In the first strategy, the number of chiller(s) was varied between 1 and 3, while the number of cooling towers was varied between 3 and 5 to find an optimal system configuration of the considered chilled water system. The temperature difference across the condenser was fixed at 4.45°C [8°F] according to the ARI standard design conditions with a fixed design chilled water supply, return temperature and full tower fan speed.
The chiller component model was operational in three modes: (1) shutdown mode, if the applied $Q_{\text{load}}$ is less than the minimum load, $Q_{\text{min}}$, specified in the model, (2) normal mode, if applied load is more than the specified minimum and less than the specified maximum load, and (3) overload mode, if $Q_{\text{load}}$ is more than the specified maximum load, $Q_{\text{max}}$. The modes enable the chiller model to properly react during the simulation subject to the applied load. In the shutdown mode, the leaving condenser and evaporator water temperatures become equal to the corresponding entering temperatures and total power consumption, condenser heat transfer rate, and COP are set to zero. In the overload mode, $Q_{\text{load}}$ is set to $Q_{\text{max}}$ and the new leaving evaporator water temperature and COP are determined. As such, the modes are needed for the simulation control, rather than a real system, in order to prevent the chiller model from coming up with unrealistic chilled water supply temperatures to meet the cooling demand.

In the current study, the chiller is always operated in the normal mode. Therefore, the number of chillers was varied satisfying the normal operating mode requirements. Figure 6.7 shows the total system power consumption and initial cost of different configurations at critical loads where system configuration changes to yield energy benefit. The power consumption of each configuration shown in Figure 3 is based on fixed design conditions throughout the load variation. In such cases the variable loads correspond to the part load operation of the system due to variable building occupancy.

In Figure 6.7(A), the optimal configuration requiring minimum total power consumption at $Q_{\text{load}}$ of 1054.8 kW [300 tons] has 1 chiller (CH) and 3 cooling towers (CT). Similarly, the optimal configuration at 1582.2 kW [450 tons] is 1CH-4CT, at 2461.2 kW [700 tons] is 1CH-5CT, at 3516 kW [1000 tons] is 2CH-5CT, and at 5274 kW [1500 tons] is 3CH-5CT as shown in Figures 6.7(B), 6.7(C), 6.7(D), and 6.7(E), respectively. The results show that adding cooling towers is beneficial for two reasons: (1) reduction of the entering condenser water temperature reduces chiller power consumption due to a decrease in chiller lift, and (2) the decrease in pressure drop across the condenser water pump reduces the pump power consumption.

The power consumption is also affected by the number of chillers, depending upon the chiller coefficient of performance (COP) at part load conditions. For example, consider the case when 2461.2 kW [700 tons] $Q_{\text{load}}$ is applied on chiller(s) with nominal load of 2725 kW [775 tons] and five cooling towers. If the system operates with one chiller, the applied load is approximately 90% of the nominal load, yielding a chiller COP of 7.68. The chiller COP reduces to 7.57 and 6.83 when the system operates with two and three chillers at 45% and 30% of the nominal load, respectively. Figure 6.8 shows the power consumption of each system configuration at the component level.

Figure 6.9 summarizes the optimal configuration options minimizing the total system power consumption with respect to the number of chillers and cooling towers at the full range of load demand. The optimal configurations shown in Figures 6.7 and 6.9 are in agreement with the real system design [66] and thus validate the proposed methodology.
Moreover, due to the initial configuration costs shown in the Figure 6.7, savings in total power consumption are more significant at higher than at lower demand loads per unit initial cost increment.

Figure 6.7: Total system power consumption and initial cost of configurations when (A) $Q_{load}$ is 1055kW, (B) $Q_{load}$ is 1582kW, (C) $Q_{load}$ is 2637kW, (D) $Q_{load}$ is 3516kW, (E) $Q_{load}$ is 5274kW, and (F) $Q_{load}$ is 7032kW
Figure 6.8: Power consumption at component level of configurations when (A) $Q_{\text{load}}$ is 1055kW, (B) $Q_{\text{load}}$ is 1582kW, (C) $Q_{\text{load}}$ is 2637kW, (D) $Q_{\text{load}}$ is 3516kW, (E) $Q_{\text{load}}$ is 5274kW, and (F) $Q_{\text{load}}$ is 7032kW
6.6.2 Modified System Configurations: 2nd Strategy

More cooling towers within the flow limit, i.e. flow turndown ratio of one-third of the design flow [193], are generally beneficial as lower fan speed and power are required. The original chilled water system design used five cooling towers with 76 kg/s design flow. In the second strategy, the number of cooling towers within the flow constraint is based on the temperature difference across the condenser. Such temperature difference determines the required water flow rate through the condenser and cooling towers to satisfy the condenser heat rejection for specific $Q_{load}$. Equations 2 and 3 were used to find the lower and upper limits for the number of cooling towers, respectively, satisfying the flow constraints.

Upper limit:

$$CT \leq \frac{C_1}{C_2} \left(\frac{Q_{load}}{\Delta T}\right) \cdot \frac{1}{m_l}$$

(2)

Lower limit:

$$CT \geq \frac{C_1}{C_2} \left(\frac{Q_{load}}{\Delta T}\right) \cdot \frac{1}{m_l}$$

(3)

Where CT is the number of cooling towers, $C_1$ is the coefficient between the applied load and condenser heat rejection, $C_2$ is the specific heat of water, $Q_{load}$ is the applied load, $\Delta T$ is the temperature difference across the condenser, $m_l$ is the cooling tower design water flow rate, i.e. 76 kg/s, and $m_l$ is limiting water flow rate, i.e. 25.3 kg/s.
Integer output of the above equations was used to determine the number of cooling towers. For example, if the $Q_{\text{load}}$ is 2637 kW [750 tons], the temperature difference across the condenser could vary between 3°C [5.4°F] and 9.5°C [17.1°F] within the cooling tower flow design limits. At 3°C [5.4°F], mass flow rate to satisfy the condenser heat rejection was 236 kg/s [3741 gpm], that mean the chilled water system could be operated with a maximum of 9 cooling towers within flow limit. Thus, 3 to 9 cooling towers were evaluated to find the impact on system performance at this specific load. Figure 6.10 is an example that shows the possible numbers of cooling towers for different values of temperature difference across the condenser along with the corresponding water mass flow rates at 2637 kW [750 tons]. Similarly, the feasible number of cooling towers is determined for all load values.

![Figure 6.10: Number of cooling towers within flow limit at $Q_{\text{load}}$ of 2637 kW [750 tons]](image)

Compared to the real system design operating with a maximum of five cooling towers, the investigated chilled water system was operated with more cooling towers in accordance to the aforementioned relationships 4 and 5. The upper and lower limits of the number of cooling towers depend on the temperature difference, $\Delta T$ across the condenser for each load value. In the second strategy, the feasible number of cooling towers operating at full fan speed is presented in Table 6.5 with fixed $\Delta T$ of 4.45°C [8°F] at various $Q_{\text{load}}$ values. Number of cooling towers with varying $\Delta T$ across the condenser is considered in the third strategy. The results of the optimization show that operating the chilled water system with more cooling towers is beneficial compared to the referent design system. Even with full fan speed, increasing the number of cooling towers will decrease the total system power consumption. Figure 6.11 shows significant decrease in the total system power consumption due to the introduction of additional cooling towers at higher loads. Although such effect diminishes as the number of cooling towers rises, the second strategy confirms the best design practice: maximize the efficiency by running as many cooling towers as possible within the flow limit [66] [193]. At the same time, the costs linearly depend upon the number of used equipment.
components and a break-even point can be determined based on the economic analyses evaluating the cost-benefit of various design configurations from studying the yearly operating profile. However, such analyses fall outside the scope of the current study.

Table 6.5: Range of configuration parameters used for optimization at various loads with constant condenser side temperature difference (4.45°C [8 °F]) at full fan speed (100%).

<table>
<thead>
<tr>
<th>Load, Q_{load} (kW [tons])</th>
<th>No. of Chiller, CH</th>
<th>No. of Cooling Towers, CT</th>
</tr>
</thead>
<tbody>
<tr>
<td>7032 [2000]</td>
<td>3</td>
<td>6-17</td>
</tr>
<tr>
<td>5274 [1500]</td>
<td>2-3</td>
<td>4-12</td>
</tr>
<tr>
<td>3516 [1000]</td>
<td>2-3</td>
<td>3-8</td>
</tr>
<tr>
<td>2461 [700]</td>
<td>1-3</td>
<td>2-6</td>
</tr>
<tr>
<td>1582 [450]</td>
<td>1-2</td>
<td>2-4</td>
</tr>
<tr>
<td>1055 [300]</td>
<td>1-2</td>
<td>2-3</td>
</tr>
</tbody>
</table>

6.6.3 Overall Chilled Water System Optimization: 3rd Strategy

Optimal chilled water systems can be selected at the design stage with respect to system design and configuration parameters. For the design optimization, it is significant to vary condenser water flow rate and the temperature difference across the condenser, and cooling tower fan speed for specific load demand. Optimization of system configuration in terms of the number of chillers and cooling towers also extensively affect the energy performance of the whole system. For the overall system optimization, simulation-based optimization strategy was implemented by coupling Dymola/Modelica with GenOpt and using a hybrid PSOGPSCCHJ optimization algorithm.

Figure 6.12 shows a sample optimization search made for one specific load demand. The difference between the two algorithms, PSO stochastic population-based and HJ direct search, forming the hybrid PSOGPSCCHJ algorithm can be identified from the iterations shown in Figure 6.12. The iterations involve the computation of an objective function with respect to varying parameters related to system design and configuration. The execution time for an optimization using a single generation as PSO input parameter at a specific load demand was about 1.5 hours on a 3.2 GHz PC with 1 GB RAM as compared to about 10 minutes on a 3.16GHz Core 2 PC with 4GB RAM. Increasing the number of PSO generations from 1 to 5 increased the execution time threefold.

Peaks of P_{total} in both, PSO and GPS Hooke-Jeeves regions of the hybrid algorithm depict the penalty value because of unfeasible number of cooling towers during the variation of ΔT across the condenser. Thus, for the optimal chilled water system design, the key concern is selection of the design temperature difference across the condenser. One way to ensure optimal design could be through the investigation of chiller lift, i.e. the difference between chilled water supply and condenser water return temperatures, as a proxy for the difference between condenser and evaporator refrigerant pressures.
Lower lift would imply lower chiller power consumption but increased consumption of other components. However, the current strategy finds the optimal condenser supply-return temperature difference which can be considered as a proxy for the chiller lift in the case of fixed chilled water supply temperature.

Figure 6.11: Total system power consumption of configurations when (A) $Q_{\text{load}}$ is 1055kW, (B) $Q_{\text{load}}$ is 1582kW, (C) $Q_{\text{load}}$ is 2637kW, (D) $Q_{\text{load}}$ is 3516kW, (E) $Q_{\text{load}}$ is 5274kW, and (F) $Q_{\text{load}}$ is 7032kW
Figure 6.12: Iteration runs for the minimization of $P_{\text{total}}$ at $Q_{\text{load}}$ of 3516 kW [1000 tons]

The feasible range of the possible system optimization parameters against various load demands is shown in Table 6.6. Figure 6.13 shows a subset of the possible alternatives evaluated by the optimization algorithm deciding the selection of parameters that cause minimization of the objective function. The presented ranges of $P_{\text{total}}$ are based on the variation of $\Delta T$ across the condenser and variation of cooling tower fan speeds. The estimated initial costs of each evaluated configuration are also presented. As the overall optimization resulted in the selection of maximum five cooling towers, the results indicate the benefit of varying the water temperature difference across the condenser and cooling tower fan speeds instead of using constant design values. In addition, such variation resulted in lower $P_{\text{total}}$ and smaller number of cooling towers than the other two previously considered strategies. Therefore, such a design is also more appropriate from an economic perspective as it would decrease the initial cost of the overall system. Moreover, Figure 6.14 shows the total power consumption at the component level based on the third optimization strategy.

Table 6.6 The range of system parameters used for optimization at various loads

<table>
<thead>
<tr>
<th>Load, $Q_{\text{load}}$ (kW [tons])</th>
<th>No. of chillers, CH</th>
<th>No. of cooling towers, CT</th>
<th>Temperature difference across condenser, $\Delta T$ ($^\circ\text{C} [^\circ\text{F}]$)</th>
<th>Fan speed, F (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7032 [2000]</td>
<td>3</td>
<td>3-25</td>
<td>3-25 [5.4-45]</td>
<td>30-100</td>
</tr>
<tr>
<td>5274 [1500]</td>
<td>2-3</td>
<td>3-18</td>
<td>3-15 [5.4-27]</td>
<td>30-100</td>
</tr>
<tr>
<td>3516 [1000]</td>
<td>2-3</td>
<td>3-12</td>
<td>3-12 [5.4-21.6]</td>
<td>30-100</td>
</tr>
<tr>
<td>2461 [700]</td>
<td>1-3</td>
<td>3-8</td>
<td>3-8 [5.4-14.4]</td>
<td>30-100</td>
</tr>
<tr>
<td>1582 [450]</td>
<td>1-2</td>
<td>3-6</td>
<td>3-5 [5.4-9]</td>
<td>30-100</td>
</tr>
<tr>
<td>1055 [300]</td>
<td>1-2</td>
<td>2-4</td>
<td>3-4 [5.4-7.2]</td>
<td>30-100</td>
</tr>
</tbody>
</table>

Table 6.7 shows the optimal combination of system parameters to achieve the minimum total power consumption. For example, at the low load demand of 1054.8 kW [300
tons], the chilled water system optimally operates with one chiller and three cooling towers maintaining the optimal design ΔT of 4°C [7.2°F] across the condenser, while the optimal cooling tower fan speed is 47.94%. The system has a minimum total power consumption of 171.56 kW. All other feasible combinations in terms of the number of chillers, cooling towers, temperature differences across the condenser, and CT fan speeds will result in higher total system power consumptions. Similarly, at the peak design load of 7032 kW [2000 tons], the optimal system design configuration yields a total power consumption of 1557.3 kW comprising of three chillers and five cooling towers operating with optimal design ΔT of 16.94°C [30.5°F] at fan speed of 85.3%. The optimal configurations comprise of maximum five cooling towers despite of the fact that the number of cooling towers within the flow limit can be up to 25. Consequently, in the case of varying condenser water temperature difference, water mass flow rate and cooling tower fan speed, may not be beneficial to run as many cooling towers as possible within the flow constraints. The results shown in Table 6.7 are in agreement with the real system configurations. The optimal fan speeds below 90% also confirm the best design practice, as suggested by Mark Hydeman, P.E., Taylor Engineering: running the cooling tower fans below 90% speed would achieve higher energy efficiency. Due to the cubic relation between the fan power and airflow rate, the cooling tower energy consumption greatly increases with the top 10% fan speed, achieving a very small drop in condenser water supply temperature.
Figure 6.13: Total system power consumption of configurations at different $Q_{\text{load}}$ (A) $Q_{\text{load}}$ is 1055kW, (B) $Q_{\text{load}}$ is 1582kW, (C) $Q_{\text{load}}$ is 2637kW, (D) $Q_{\text{load}}$ is 3516kW, (E1) $Q_{\text{load}}$ is 5274kW with 2 chillers, (E2) $Q_{\text{load}}$ is 5274kW with 3 chillers and (F) $Q_{\text{load}}$ is 7032kW
Table 6.7: Optimal values of system parameters and objective function at various loads

<table>
<thead>
<tr>
<th>Load, Q_{\text{load}} (kW[tons])</th>
<th>No. of chiller, CH</th>
<th>No. of cooling towers, CT</th>
<th>Temperature difference across condenser, ΔT (°C [°F])</th>
<th>Fan speed, F (%)</th>
<th>Optimal P_{\text{total}} (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7032 [2000]</td>
<td>3</td>
<td>5</td>
<td>16.94 [30.5]</td>
<td>85.31</td>
<td>1557.3</td>
</tr>
<tr>
<td>5274 [1500]</td>
<td>3</td>
<td>5</td>
<td>12.68 [22.8]</td>
<td>75.44</td>
<td>993.1</td>
</tr>
<tr>
<td>3516 [1000]</td>
<td>2</td>
<td>5</td>
<td>8.45 [15.2]</td>
<td>62.8</td>
<td>582.7</td>
</tr>
<tr>
<td>2461 [700]</td>
<td>2</td>
<td>4</td>
<td>7.63 [13.7]</td>
<td>59.75</td>
<td>394.9</td>
</tr>
<tr>
<td>1582 [450]</td>
<td>1</td>
<td>4</td>
<td>4.88 [8.8]</td>
<td>49.3</td>
<td>240.7</td>
</tr>
<tr>
<td>1055 [300]</td>
<td>1</td>
<td>3</td>
<td>4 [7.2]</td>
<td>47.9</td>
<td>171.6</td>
</tr>
</tbody>
</table>

Figure 6.14: Power consumption at component level of optimal configurations

6.7 Summary of the Chilled Water System Analyses

A methodology is proposed for design optimization of the chilled water system at initial design stage by applying three strategies: (1) baseline system comprising of a maximum of five cooling towers with fixed design temperature difference across the condenser at full fan speed, (2) modified system with increased number of cooling towers according to the flow turndown limit and fixed design temperature difference across the condenser at full fan speed, and (3) modified system with varying both system design and configuration parameters. While the first two strategies confirm the established best
design practice and verify the simulation models, the third strategy defines a systematic approach for overall design optimization of chilled water systems. In addition, the strategies consider initial costs of equipment, including chillers, cooling towers and piping/fittings/valves, having in mind that additional chillers and cooling towers typically decrease the annual energy costs and payback period [195] [198].

Optimal total system power consumptions and energy use in kW/ton from the considered approaches are shown in Table 6.8. The values of energy use (kW/ton) are quite in agreement with the typical values for the chilled water systems [195]. It can be observed that the lowest total power consumption, \( P_{\text{total}} \), and energy use is achieved for the modified system with varying temperature difference across the condenser and varying fan speed. Also, the performance of chilled water systems strongly depends on the appropriate selection of mass flow rates and temperature differences across the condenser. The simulation-based optimization approach enhances the decision making about optimal variation in these parameters resulting in the minimum total system power consumption. Significant power savings between 17% and 43.5% could be achieved by operating chilled water systems at the optimal design and configuration parameters compared to the baseline case.

Table 6.8: Optimal values of \( P_{\text{total}} \) and energy use of all strategies (minimum values highlighted)

<table>
<thead>
<tr>
<th>( Q_{\text{load}} ) (kW [tons])</th>
<th>1st Strategy</th>
<th>2nd Strategy</th>
<th>3rd Strategy</th>
<th>Percentage of power saving (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( P_{\text{total}} ) (kW)</td>
<td>kW/ton</td>
<td>( P_{\text{total}} ) (kW)</td>
<td>kW/ton</td>
</tr>
<tr>
<td>7032 [2000]</td>
<td>2757.3</td>
<td>1.37</td>
<td>1808.4</td>
<td>0.9</td>
</tr>
<tr>
<td>5274 [1500]</td>
<td>1498.8</td>
<td>0.99</td>
<td>1178.8</td>
<td>0.78</td>
</tr>
<tr>
<td>3516 [1000]</td>
<td>752.7</td>
<td>0.75</td>
<td>706.9</td>
<td>0.71</td>
</tr>
<tr>
<td>2461 [700]</td>
<td>479.7</td>
<td>0.68</td>
<td>479.7</td>
<td>0.68</td>
</tr>
<tr>
<td>1582 [450]</td>
<td>297.8</td>
<td>0.66</td>
<td>297.8</td>
<td>0.66</td>
</tr>
<tr>
<td>1055 [300]</td>
<td>207.4</td>
<td>0.69</td>
<td>207.4</td>
<td>0.69</td>
</tr>
</tbody>
</table>

6.8 Potential Benefits and Limitations

In the current study, model of the chilled water system was developed based on EOO approach using open sources component libraries. Optimization of HVAC systems involved variation of component design parameters as well as configurations. Dymola/Modelica enabled control of all component parameter values from external text files for the purpose of design optimization. Serial and/or parallel arrangements, as well as the number of component models were also decided from external files. Multiple instances of single component models were used within the component arrays to vary the number of cooling towers. The aforementioned features helped to develop efficient and effective multi-domain system models for simulation and optimization. Automated optimization of the systems involving many design and configuration parameters offered the possibility to identify the most energy efficient design, a task which would
otherwise require intense efforts and time. The applied optimization algorithm used fewer generations compared to the literature recommendations [130], as no variation in system optimization parameters was observed after the initial two generations. Such a selection of optimization input parameters saved simulation time, while possibly hindering the achievement of globally optimal results. The methodology of coupling Dymola/Modelica and GenOpt proved as efficient in terms of efforts, time, and ease to vary the design and configuration parameters for selection of optimal system configurations. The main weakness of Dymola/Modelica application to building simulations is the lack of sufficient HVAC component models within the existing modeling libraries. However, research activities are underway to enhance the available libraries by developing additional component models [199] [200].

The current study assumes that the chilled water system component sizes are predefined and optimizes the number of components and operating conditions using fixed design parameters: chilled water supply and return temperatures, dry bulb and wet bulb temperatures. As such the proposed approach could be considered as part of the iterative design process, rather than an all-inclusive procedure.

### 6.9 Conclusions

In the current study, an incremental development of the methodology for chilled water system optimization is proposed. The equation-based object oriented modeling approach was used to model a real chilled water system located at a Symantec Corporation building in South California. The chilled water system model developed in Dymola/Modelica was capable of varying system design and configuration parameters at different load demands. Dymola/Modelica model was coupled with GenOpt optimization software. The simulation-based optimization used a hybrid PSOGPSCCHJ optimization algorithm to find the optimal system configuration. The current study analyzes several design parameters having significant impact on system performance. In addition, the estimated initial cost of each configuration is also provided for economic considerations. The chilled water system was analyzed considering three strategies, two of which involved constant temperature difference across the condenser at full fan speeds applied to the baseline and modified system configurations. The third strategy varied the temperature difference across the condenser as well as cooling tower fan speeds in addition to the system configuration parameters and proved to be the most energy efficient. Consequently, the optimal values of the considered parameters were provided together with the calculated power consumption. Operating chilled water systems at the optimal conditions could result in significant total system power savings amounting up to 43.5% for the considered cases. The implemented automated simulation-based optimization approach proved efficient in terms of model development and computational time to find optimal configurations. The methodology represents a step towards the design of software systems able to synthesize new and optimal HVAC
system configurations. Such development should help the design practitioners to select the optimal system configuration parameters at the initial stage.
Chapter 07

7 MODEL-BASED DEVELOPMENT AND VALIDATION OF DESICCANT COOLING SYSTEM: ENERGYBASE SYSTEM

The desiccant cooling systems provide new possibilities in air conditioning technology through the use of either solid or liquid sorption air dehumidification. Such systems can be reliable alternative compared to the conventional systems in terms of their environmental friendly operation. The focus of current research is to present an equation-based object-oriented modeling and optimization approach of HVAC system configurations. However, the component models available in various modeling libraries are ideal. Therefore, in the current chapter, the component models of desiccant cooling system are modified for real system performance analyses. Afterwards, the component models are experimentally calibrated and validated at both components as well as system level under the actual transient operating conditions of a real system.

7.1 Description of desiccant cooling system

The desiccant evaporative cooling system (DEC) of ENERGYbase building consists of a desiccant wheel, a heat wheel, and two direct humidifiers i.e. supply humidifier and return humidifier. The system is design by “Robatherm” air handling company. The whole HVAC system is equipped with around 500 sensors continuously measuring all significant parameters at the individual component level. Two data monitoring systems, Siemens DESIGO Insight and JEVIs are used for data recording from all sensors. The system schematic is shown in Figure 7.1. Technical data of main components of the installed DEC system is given in Table 7.1.

![Figure 7.1: Desiccant cooling system of ENERGYbase building](image-url)
Table 7.1: Component design parameters of ENERGYbase system

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>DEC component design parameter</th>
<th>Design value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td><strong>Desiccant wheel</strong></td>
<td>Model SECO 1770, Klingenburg</td>
</tr>
<tr>
<td></td>
<td>Air volume flow rate (m$^3$/h)</td>
<td>8240</td>
</tr>
<tr>
<td></td>
<td>Adsorbent</td>
<td>LiCl</td>
</tr>
<tr>
<td></td>
<td>Wheel diameter (m)</td>
<td>1.77</td>
</tr>
<tr>
<td></td>
<td>Wheel depth (m)</td>
<td>0.45</td>
</tr>
<tr>
<td></td>
<td>Pressure drop (Pa)</td>
<td>164</td>
</tr>
<tr>
<td></td>
<td>Wheel rotation speed (rph)</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Ambient air (C / %)</td>
<td>32 / 40</td>
</tr>
<tr>
<td></td>
<td>Supply air (C / %)</td>
<td>46.9 / 12</td>
</tr>
<tr>
<td></td>
<td>Regeneration air (C / %)</td>
<td>70 / 10</td>
</tr>
<tr>
<td></td>
<td>Exhaust air (C / %)</td>
<td>55.1 / 20</td>
</tr>
<tr>
<td>2</td>
<td><strong>Heat wheel</strong></td>
<td>ModelRRS-P-16-18-1770-400, Klingenburg</td>
</tr>
<tr>
<td></td>
<td>Air volume flow rate (m$^3$/h)</td>
<td>8240</td>
</tr>
<tr>
<td></td>
<td>Wheel diameter (m)</td>
<td>1.77</td>
</tr>
<tr>
<td></td>
<td>Wheel depth (m)</td>
<td>0.4</td>
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<tr>
<td></td>
<td>Wheel rotation speed (rpm)</td>
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<td></td>
<td>Pressure drop (Pa)</td>
<td>119</td>
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<tr>
<td></td>
<td>Effectiveness</td>
<td>0.8</td>
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<tr>
<td></td>
<td>Outdoor air (C / %)</td>
<td>45 / 14</td>
</tr>
<tr>
<td></td>
<td>Supply air (C / %)</td>
<td>31.24 / 29</td>
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<td></td>
<td>Regeneration air (C / %)</td>
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<td>Exhaust air (C / %)</td>
<td>41.62 / 40</td>
</tr>
<tr>
<td>3</td>
<td><strong>Supply and return spray humidifiers</strong></td>
<td>Robatherm</td>
</tr>
<tr>
<td></td>
<td>Air volume flow rate (m$^3$/h)</td>
<td>8240</td>
</tr>
<tr>
<td></td>
<td>Saturation efficiency (%)</td>
<td>90%</td>
</tr>
<tr>
<td></td>
<td>Dehumidification capacity (g/kg)</td>
<td>4.4</td>
</tr>
<tr>
<td></td>
<td>Water flow rate (m$^3$/h)</td>
<td>4.0</td>
</tr>
<tr>
<td></td>
<td>Pump power (kW)</td>
<td>0.8</td>
</tr>
<tr>
<td></td>
<td>Pressure drop (Pa)</td>
<td>235</td>
</tr>
<tr>
<td></td>
<td>Supply air (C / %)</td>
<td>32C / 33%</td>
</tr>
<tr>
<td></td>
<td>Exhaust air (C / %)</td>
<td>21.26C / 89%</td>
</tr>
<tr>
<td>4</td>
<td><strong>Supply and return fans</strong></td>
<td>RH 45C</td>
</tr>
<tr>
<td></td>
<td>Air volume flow rate (m$^3$/h)</td>
<td>8240</td>
</tr>
<tr>
<td></td>
<td>Pressure drop (Pa)</td>
<td>600</td>
</tr>
<tr>
<td></td>
<td>Power (kW)</td>
<td>4.3</td>
</tr>
</tbody>
</table>
7.2 Physical component models calibration and validation

The real time transient data from the monitoring systems is used for the calibration and validation of individual components and the whole system. The transient measurement data sets are based on the same six selected days already described in chapter 5.

7.2.1 Desiccant wheel model

The desiccant wheel is a key component of the DEC system but its model is not available in Modelica library. Therefore, a detailed model of desiccant wheel is developed in Modelica/Dymola environment. The development and validation of desiccant wheel model is described in chapter 5. The additional validation of desiccant wheel model is presented for LiCl as adsorbent. The effects of operating conditions at the inlet of process and regeneration air are shown in Figures 7.2 and 7.3, respectively. The overall trends of the results are in good agreement with the published data. However, the detail explanation of the effects remains same as in case of silica gel that is already provided in section 5.3.

Figure 7.2: Effects of process air inlet conditions
7.2.2 Humidifier component model (direct evaporative cooler)

The ENERGYbase desiccant cooling system consists of two humidifiers, one on the process side called as ‘supply humidifier’ and other on the return side termed as ‘return humidifier’. Both humidifiers have same physical design but operate under different conditions with different control strategies.

- Dymola Model

The component model for both humidifiers is used from the class of ‘Mass Exchangers’ available in LBL Buildings library [201]. Graphical presentation of model is shown in Figure 7.4. The quantity of water added to the air stream is used as a model parameter. In addition, air temperature and its absolute humidity are also specified as inputs. The evaporative cooling process is assumed isenthalpic. The model can act as humidifier or dehumidifier depending on the value of humidity input control signal, u. A positive value of u indicates humidification. The amount of moisture exchanged is determined using Equation 7.1. The value to the water temperature port is assigned conditionally either by user or default value is used.

\[ m_{\text{wat,flow}} = u \times m_{\text{wat,flow,nominal}} \]  

(7.1)
The monitoring data in terms of transient measurements is obtained from the Siemens ‘DESIGO’ monitoring system [185]. Each humidifier is equipped with four sensors, two at inlet and two at outlet with each pair measuring temperature and relative humidity at both points. Consequently, the data set of six selected days consists of air temperature and relative humidity along with the rotation speed of spray pump. The monitoring setup of both humidifiers is shown in Figure 7.5.

In the actual monitoring set-up water mass flow rate from the spray pump was not measured. The water mass flow rate depends on the inlet conditions of air in terms of its temperature, absolute humidity, and mass flow rate along with the rotational speed signal of spray pump. Multiple Linear Regression (MLR) [202] is used to determine an appropriate correlation between input variables and monitored humidification with respect to the absolute humidity difference across the humidifier. The function is defined based on the dependence of each input variable through MLR. MLR is applied through Excel in which the significance of each input variable is decided based on its P-value. The variable with P-value greater than 0.05 has less significance, thus not considered in the function. An example of such implementation based on transient data of 6th July is shown in Figure 7.6.
Figure 7.6: Implementation of Multiple Linear Regression (MLR) to determine the function dependence on; (A) absolute humidity at inlet, (B) temperature at inlet, (C) spray pump position, and (D) mass flow rate with respect to absolute humidity difference across humidifier.

The function determined based on the results of Figure 7.6 is defined in Equation 7.2.

\[ D_x = 0.335 - 0.365\omega_{in} + 0.0155S + 0.145T_{in} + 0.523m \]  

(7.2)

Similarly, such functions for all selected days are determined based on the transient measurement data of each respective day. Afterwards, these functions are implemented in Dymola for the calibration and validation of humidifier model. In general, the value of \( D_x \) varies from 0.0005 to 0.0035 kg/kg of dry air.

- Estimation of actual enthalpy

The LBL Buildings model of the humidifier is based on isenthalpic humidification process. However, the humidifier operation in real system deviates from the constant enthalpy line with a significant difference due to various heat losses as shown in Figure 7.7. Therefore, in order to account for deviations available humidifier model is modified.
A variable ‘$Q_{\text{offset}}$’ is introduced in the model to account the efficiency loss due to the difference between isenthalpic and real humidification process. The $Q_{\text{offset}}$ is defined in terms of a customized function that relates the enthalpy difference, $D_h$ to $Q_{\text{offset}}$ based on the monitoring data of a real humidifier under operation in ENERGYbase. However, the function’s dependence on various inlet variables is again determined by MLR. An example of implementation of MLR for supply humidifier based on the measurement data of 19th June is shown in Figure 7.8. Figure shows that all four inlet variables have influence on $D_h$. The significance determination criterion is P-value. The inlet variables having P-value greater than 0.05 are not consider in the function. The function is defined in Equation 7.3.

$$D_h = -1.542 - 0.701 \omega_{in} - 0.0635S + 0.285T_{in} + 2.43 \dot{m}$$  \hspace{1cm} (7.3)

The value of $D_h$ vary around 1 to 4 kJ/kg for the considered measurement data.
Figure 7.8: Implementation of Multiple Linear Regression (MLR) to determine the function dependence on; (A) absolute humidity at inlet, (B) temperature at inlet, (C) spray pump position, and (D) mass flow rate with respect to specific enthalpy difference across humidifier.

7.2.2.1 Calibration and validation of supply and return humidifiers

The supply and return humidifier component models are calibrated for the identification of water mass flow rate sprayed in the humidifier. The monitoring data of six days is used in terms of air dry bulb temperature, absolute humidity, and mass flow rate at the humidifier inlet. The calibration and validation setup used for humidifier model is shown in Figure 7.9.
The root mean squared error (RMSE) between the simulated and monitored specific enthalpy at the humidifier outlet is used as an objective function for model calibration and validation determined by Equation 7.4.

\[ f_{\text{RMSE}} = \sqrt{\frac{(h_{\text{osim}} - h_{\text{omon}})^2}{dt}} \]  

(7.4)

Calibration errors in terms of RMSE of both supply and return humidifiers for all days are presented in Tables 7.2 and 7.3, respectively. In addition to RMSE, calibration errors are also defined with respect to mean percentage error (MPE) based on the simulated and monitored values of specific enthalpy at outlet calculated by Equation 7.5.

\[ f_{\text{MPE}} = \frac{(h_{\text{osim}} - h_{\text{omon}})}{h_{\text{osim}}} \times 100 \]  

(7.5)

The results show that the simulated values are in good agreement with the monitored values having minimum RMSE and MPE of 0.367kJ/kg and 0.018% of supply humidifier, respectively. For return humidifier, minimum errors are 0.332kJ/kg and 0.02%. Negative values in Table 7.2 show under prediction of simulated values compared to monitored data.
Table 7.2: Calibration errors of selected days of supply humidifier model

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Calibration day</th>
<th>Calibration Error, RMSE (kJ/kg)</th>
<th>MPE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>19th June</td>
<td>0.367</td>
<td>-0.018</td>
</tr>
<tr>
<td>2</td>
<td>29th June</td>
<td>0.990</td>
<td>-0.069</td>
</tr>
<tr>
<td>3</td>
<td>02nd July</td>
<td>1.058</td>
<td>-0.023</td>
</tr>
<tr>
<td>4</td>
<td>06th July</td>
<td>0.700</td>
<td>-0.082</td>
</tr>
<tr>
<td>5</td>
<td>09th July</td>
<td>0.933</td>
<td>0.023</td>
</tr>
</tbody>
</table>

Table 7.3: Calibration errors of selected days of return humidifier model

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Calibration day</th>
<th>Calibration Error, RMSE (kJ/kg)</th>
<th>MPE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>30th April</td>
<td>0.332</td>
<td>0.1</td>
</tr>
<tr>
<td>2</td>
<td>19th June</td>
<td>0.752</td>
<td>0.06</td>
</tr>
<tr>
<td>3</td>
<td>29th June</td>
<td>0.523</td>
<td>0.05</td>
</tr>
<tr>
<td>4</td>
<td>02nd July</td>
<td>0.574</td>
<td>0.02</td>
</tr>
<tr>
<td>5</td>
<td>06th July</td>
<td>0.510</td>
<td>0.05</td>
</tr>
<tr>
<td>6</td>
<td>09th July</td>
<td>0.330</td>
<td>0.04</td>
</tr>
</tbody>
</table>

The component models of both humidifiers are validated after their calibration. The design values predicted by the calibrated model along with the established function of each calibrated day are used for model validation. Thus, the calibrated model is validated based on the transient measurement data of other days except the calibration day. The validation errors of supply humidifier component model with respect to RMSE and MPE are given in Tables 7.4 and 7.5, respectively. Similar results for the component model of return humidifier are presented in Tables 7.6 and 7.7, respectively. The resulted average minimum validation errors in terms of RMSE of supply and return humidifiers are 1.14kJ/kg and 0.6kJ/kg, respectively. Moreover minimum errors of supply and return humidifiers with respect to MPE are 0.88% and 0.43%, respectively. The minimum average value of absolute validation errors is highlighted in Tables.

Table 7.4: Validation errors of supply humidifier model in terms of RMSE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, RMSE (kJ/kg)</th>
<th>Average values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>19th June</td>
<td>29th June</td>
</tr>
<tr>
<td>19th June</td>
<td>---</td>
<td>1.432</td>
</tr>
<tr>
<td>29th June</td>
<td>0.521</td>
<td>---</td>
</tr>
<tr>
<td>02nd July</td>
<td>0.574</td>
<td>0.990</td>
</tr>
<tr>
<td>06th July</td>
<td>1.459</td>
<td>2.385</td>
</tr>
<tr>
<td>09th July</td>
<td>0.755</td>
<td>1.187</td>
</tr>
</tbody>
</table>
Moreover, the validation results with minimum values of average errors show that the simulated values are in good agreement with the monitored values. This comparison for supply and return humidifiers is shown in Figures 7.10 and 7.11, respectively. Simulated and monitored results of selected days with minimum errors in term of specific enthalpy, dry bulb temperature, and absolute humidity are presented in these figures.

Table 7.5: Validations errors of supply humidifier model in terms of MPE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, MPE (%)</th>
<th>Average of absolute values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>29&lt;sup&gt;th&lt;/sup&gt; June</td>
</tr>
<tr>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>---</td>
<td>-0.694</td>
</tr>
<tr>
<td>29&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>0.320</td>
<td>---</td>
</tr>
<tr>
<td>02&lt;sup&gt;nd&lt;/sup&gt; July</td>
<td>0.337</td>
<td>-0.260</td>
</tr>
<tr>
<td>06&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>-1.087</td>
<td>-1.493</td>
</tr>
<tr>
<td>09&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>-0.786</td>
<td>-1.493</td>
</tr>
</tbody>
</table>

Table 7.6: Validations errors of return humidifier model in terms of RMSE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, RMSE (kJ/kg)</th>
<th>Average values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30&lt;sup&gt;th&lt;/sup&gt; April</td>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
</tr>
<tr>
<td>30&lt;sup&gt;th&lt;/sup&gt; April</td>
<td>---</td>
<td>0.631</td>
</tr>
<tr>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>1.631</td>
<td>---</td>
</tr>
<tr>
<td>29&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>1.269</td>
<td>0.382</td>
</tr>
<tr>
<td>02&lt;sup&gt;nd&lt;/sup&gt; July</td>
<td>0.938</td>
<td>0.429</td>
</tr>
<tr>
<td>06&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>0.914</td>
<td>1.127</td>
</tr>
<tr>
<td>09&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>0.949</td>
<td>0.859</td>
</tr>
</tbody>
</table>

Table 7.7: Validations errors of return humidifier model in terms of MPE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, MPE (%)</th>
<th>Average of absolute values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30&lt;sup&gt;th&lt;/sup&gt; April</td>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
</tr>
<tr>
<td>30&lt;sup&gt;th&lt;/sup&gt; April</td>
<td>---</td>
<td>0.77</td>
</tr>
<tr>
<td>19&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>3.02</td>
<td>---</td>
</tr>
<tr>
<td>29&lt;sup&gt;th&lt;/sup&gt; June</td>
<td>2.49</td>
<td>0.36</td>
</tr>
<tr>
<td>02&lt;sup&gt;nd&lt;/sup&gt; July</td>
<td>1.81</td>
<td>0.46</td>
</tr>
<tr>
<td>06&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>-1.69</td>
<td>1.71</td>
</tr>
<tr>
<td>09&lt;sup&gt;th&lt;/sup&gt; July</td>
<td>-1.77</td>
<td>1.18</td>
</tr>
</tbody>
</table>

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Figure 7.10: Comparison between monitored and simulated results of supply humidifier model based on 2nd July (A), specific enthalpy (B), and temperature and absolute humidity at outlet (C).
Figure 7.11: Comparison between monitored and simulated results of return humidifier model based on 30th April (A), specific enthalpy (B), and temperature and absolute humidity at outlet (C).

7.2.2.2 Parametric analysis of humidifier component model

Typically, performance of direct humidifiers is described by the cooling efficiency defined by Equation 7.6. In addition, temperature drop and increase in absolute humidity are also indication of humidification capacity. Such performance factors of direct humidifier are influenced by various inlet operating parameters, like air dry bulb temperature, air relative humidity, air mass flow rate, water temperature, and water mass flow rate.

\[ \eta_c = \frac{(T_{dbin} - T_{dout})}{(T_{dbin} - T_{wbin})} \]  

(7.6)

In the current study, parametric performance analysis of humidifier is carried out through validated model. Effects of above mentioned parameters on cooling performance are evaluated. During analysis, each parameter was varied while keeping the other variables constant. The variation range of each parameter is defined in Table 7.8 based on the real system operation installed in ENERGYbase. Performance analysis of humidifier based on all parameters is presented in Figure 7.12.
Figure 7.12(A) shows the well-established facts about the operation of direct humidifiers [203] [204] [205]. High inlet air dry bulb temperature indicates dry air that favors the evaporation process, thus increases the cooling efficiency. However, higher air relative humidity decreases temperature drop and thus the cooling efficiency as shown in Figure 7.12(B). In addition, effects of air mass flow rate are also presented in 7.12(C). It is clear that lower mass flow rate causes greater temperature drop that increases cooling capacity. High water flow rate also enhances the possibility of more temperature and humidity drops as shown in Figure 7.12(D). Finally, effects of water temperature are also analyzed that has variation is commercial applications. High water temperature decreases the temperature difference between air and water that cause reduction in the evaporation process as shown in 7.12(E). Thus cooling capacity decreases. All trends verify that the humidifier component model predicts the accurately the performance of direct humidifiers.

Table 7.8: Range of operating parameters for the performance analysis

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Baseline Values</th>
<th>Parametric variations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature of ambient air, $T_{amb}$ (°C)</td>
<td>30</td>
<td>---</td>
</tr>
<tr>
<td>Humidity ratio of ambient air, $\omega_{amb}$ (g/kg)</td>
<td>10</td>
<td>---</td>
</tr>
<tr>
<td>Inlet temperature of air, $T_{sin}$ (°C)</td>
<td>30</td>
<td>25-40</td>
</tr>
<tr>
<td>Inlet relative humidity of air, $\phi_a$ (%)</td>
<td>50</td>
<td>35-65</td>
</tr>
<tr>
<td>Air mass flow rate, $m_a$ (kg/s)</td>
<td>2.1</td>
<td>1.7-3.0</td>
</tr>
<tr>
<td>Water mass flow rate, $m_w$ (kg/s)</td>
<td>2.5</td>
<td>1-2.7</td>
</tr>
<tr>
<td>Water temperature (°C)</td>
<td>20</td>
<td>18-30</td>
</tr>
</tbody>
</table>
Figure 7.12: Performance analyses of humidifier in terms of cooling efficiency, temperature drop and absolute humidity rise with respect to; air inlet temperature (A), air inlet relative humidity (B), air mass flow rate (C), water flow rate (D), and water temperature (E).

7.2.3 Heat wheel component model

Heat wheel is another key component of desiccant cooling system. It helps to reduce the temperature of process air heated during dehumidification process in the desiccant wheel.

- Dymola component model

Heat wheel component model is again taken from the ‘Heat Exchanger’ class of LBL Buildings library. It is model of heat exchanger with constant effectiveness. However, the existing library model assumes no heat loss to ambient compared to real system operation in which heat losses were observed. Therefore the model is modified with an additional heat loss variable $Q_{\text{loss}}$ as presented in Figure 7.13.
In the real installation, heat wheel is equipped with nine sensors similar to desiccant wheel as shown in Figure 7.14. Eight sensors, four each for temperature and relative humidity measurements are installed at the inlet and outlet of primary and secondary fluids. One sensor is installed for determination of rotation speed of heat wheel. Transient measurement data of six selected days with one minute time interval is used for the calibration and validation of heat wheel model.

Effectiveness of heat wheel is determined by the monitoring data through typical Equation 7.7 according to ARI standard 1060 [152].

\[
\varepsilon = \begin{cases} 
\frac{C_h (T_{ho} - T_{hi})}{C_c (T_{hi} - T_{ci})} & \text{if } C_c < C_h \\
\frac{C_h (T_{ho} - T_{hi})}{C_h (T_{hi} - T_{ci})} & \text{if } C_h < C_c 
\end{cases}
\]  

(7.7)

Here C is the heat capacity rate i.e. C = mC_p.
• Determination of heat loss

Heat loss, $Q_{\text{loss}}$ through the heat wheel is implemented in the Dymola model by a function defined in Equation 7.8 based on the measurement data of 29th June, 2012. The function is determined by the monitoring data. The function’s dependence on maximum temperature difference across heat wheel is determined through linear curve fitting as shown in Figure 7.15.

$$Q_{\text{loss}} = 0.357\Delta T_{\text{max}} - 1.658$$  \hspace{1cm} (7.8)

![Figure 7.15: Relationship between heat loss, $Q_{\text{loss}}$ and $\Delta T_{\text{max}}$ cross heat wheel for 29th June](image)

7.2.3.1 Calibration and validation of Heat wheel model

Calibration and validation of heat wheel model is performed considering the temperature difference between the simulated and monitoring values as an objective function of both process and regeneration sides. Dymola setup used for calibration and validation is shown in Figure 7.16. For heat wheel model, the objective function is defined in terms of RMSE of outlet temperature difference between simulated and monitored values of both fluid streams define by Equation 7.9. In addition, MPE is also determined by Equation 7.10.

$$f_{\text{RMSE}} = \sqrt{\langle (T_{\text{osim}} - T_{\text{omon}})^2 \rangle / dt}$$  \hspace{1cm} (7.9)

$$f_{\text{MPE}} = (T_{\text{osim}} - T_{\text{omon}}) / T_{\text{osim}} \times 100$$  \hspace{1cm} (7.10)
Figure 7.16: Graphical representation of calibration and validation setup of heat wheel model

Calibration errors in terms of RMSE and MPE of all selected days are given in Table 7.9. It is observed that resulted errors in both forms are quite small. Afterwards, calibrated model is validated based on the transient measurements of five other days except the calibration day. The resulted validation errors in form of RMSE and MPE are given in Tables 7.10 and 7.11, respectively. The errors show that the simulated values are in good agreement with the monitored values and the highlighted minimum average errors are 0.475 °C and 0.1% with respect of RMSE and MPE, respectively on 29th June, 2012. Additionally, comparison between simulated and monitored values with respect to outlet temperatures of process and regeneration sides is presented in Figure 7.12 based on the validation results of 29th June, 2012.

Table 7.9: Calibration errors of heat wheel model

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Calibration day</th>
<th>Calibration Error, RMSE (°C)</th>
<th>MPE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Pro-side</td>
<td>Reg-side</td>
</tr>
<tr>
<td>1</td>
<td>30th April</td>
<td>0.045</td>
<td>0.811</td>
</tr>
<tr>
<td>2</td>
<td>19th June</td>
<td>0.032</td>
<td>0.636</td>
</tr>
<tr>
<td>3</td>
<td>29th June</td>
<td>0.454</td>
<td>0.587</td>
</tr>
<tr>
<td>4</td>
<td>02nd July</td>
<td>0.835</td>
<td>0.656</td>
</tr>
<tr>
<td>5</td>
<td>06th July</td>
<td>0.277</td>
<td>0.491</td>
</tr>
<tr>
<td>6</td>
<td>09th July</td>
<td>0.077</td>
<td>0.479</td>
</tr>
</tbody>
</table>
Table 7.10: Validations errors of heat wheel model in terms of RMSE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, RMSE (°C)</th>
<th>Average values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30th April</td>
<td>19th June</td>
</tr>
<tr>
<td>30th April</td>
<td>Pro-side</td>
<td>---</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>---</td>
</tr>
<tr>
<td>19th June</td>
<td>Pro-side</td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>0.709</td>
</tr>
<tr>
<td>29th June</td>
<td>Pro-side</td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>1.114</td>
</tr>
<tr>
<td>02nd July</td>
<td>Pro-side</td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>1.738</td>
</tr>
<tr>
<td>06th July</td>
<td>Pro-side</td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>1.697</td>
</tr>
<tr>
<td>09th July</td>
<td>Pro-side</td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>0.894</td>
</tr>
</tbody>
</table>

Table 7.11: Validations errors of heat wheel model in terms of MPE

<table>
<thead>
<tr>
<th></th>
<th>Validation Error, MPE (%)</th>
<th>Average of absolute values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30th April</td>
<td>19th June</td>
</tr>
<tr>
<td>30th April</td>
<td>Pro-side</td>
<td>---</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>---</td>
</tr>
<tr>
<td>19th June</td>
<td>Pro-side</td>
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</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>-1.16</td>
</tr>
<tr>
<td>29th June</td>
<td>Pro-side</td>
<td>0.071</td>
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<tr>
<td></td>
<td>Reg-side</td>
<td>-3.45</td>
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<tr>
<td>02nd July</td>
<td>Pro-side</td>
<td>0.071</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>-6.49</td>
</tr>
<tr>
<td>06th July</td>
<td>Pro-side</td>
<td>0.071</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>-6.26</td>
</tr>
<tr>
<td>09th July</td>
<td>Pro-side</td>
<td>0.071</td>
</tr>
<tr>
<td></td>
<td>Reg-side</td>
<td>-2.31</td>
</tr>
</tbody>
</table>
Figure 7.17: Comparison between monitored and simulated results of heat wheel model based on 29th June (A), specific enthalpy (B), and temperature and absolute humidity at outlet (C).

7.3 Calibration and validation of desiccant evaporative cooling system

Following the calibration and validation of individual modified component models, validation is performed at the system level. Monitoring setup of whole system equipped with several sensors is shown Figure 7.18.

Figure 7.18: Monitoring setup of desiccant cooling system
At system level, a system model is developed with validated models as shown in Figure 7.19. In system model, functions of different individual components are also implemented based on the resulted minimum validation error. The regeneration temperature is maintained at 70 °C in view of maximum allowable temperature for LiCl adsorbent. For system model calibration and validation, transient measurement data of four days is used due to few operational problems at system level on 30th April and 06th July, 2012. Two objective functions are used for system model calibration and validation. First is the root mean squared error (RMSE) between the monitored and simulated outlet specific enthalpy of supply air. In addition, calibration errors based on specific enthalpy are also presented in terms of MPE. Calibration errors in both forms of specific enthalpy are given in Table 7.12. Secondly, RMSE between the monitored and simulated cooling capacity provided by the system is considered an objective function through Equation 7.11. Calibration errors with respect to second objective function are given in Table 7.13.

\[
f_{ob} = \frac{\sqrt{(CC_{sim} - CC_{mon})^2}}{dt} \tag{7.11}
\]

Here CC is cooling capacity of the system determined by Equation 7.12.

\[
CC = m_s (h_{amb} - h_{supply}) \tag{7.12}
\]

Calibration errors are slightly high in Tables 7.12 and 7.13. The fact is that the calibration errors at individual component model level are compiled at the system level. Although, simulated values in table 7.13 based on MPE are underestimated compared to monitored data with minimum absolute value of 7.79%.

Figure 7.19: Graphical representation of calibration and validation setup of desiccant evaporative cooling system model
Table 7.12: Calibration errors based on outlet specific enthalpy

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Calibration day</th>
<th>Calibration Error, RMSE (kJ/kg)</th>
<th>MPE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>19th June</td>
<td>1.015</td>
<td>1.49</td>
</tr>
<tr>
<td>2</td>
<td>29th June</td>
<td>1.386</td>
<td>1.69</td>
</tr>
<tr>
<td>3</td>
<td>02nd July</td>
<td>2.627</td>
<td>3.20</td>
</tr>
<tr>
<td>4</td>
<td>09th July</td>
<td>0.883</td>
<td>1.15</td>
</tr>
</tbody>
</table>

Table 7.13: Calibration errors based on system cooling capacity

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Calibration day</th>
<th>Calibration Error, RMSE (kW)</th>
<th>MPE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>19th June</td>
<td>2.629</td>
<td>-14.36</td>
</tr>
<tr>
<td>2</td>
<td>29th June</td>
<td>3.180</td>
<td>-14.46</td>
</tr>
<tr>
<td>3</td>
<td>02nd July</td>
<td>3.787</td>
<td>-18.56</td>
</tr>
<tr>
<td>4</td>
<td>09th July</td>
<td>1.949</td>
<td>-7.79</td>
</tr>
</tbody>
</table>

For model validation, three days transient measurement data is used excluding the calibration day. Again, two objective functions, outlet specific enthalpy and cooling capacity are used objective functions in terms of RMSE and MPE. Tables 7.14 and 7.15 present validation errors of system model considering specific outlet enthalpy as objective function with respect of RMSE and MPE, respectively. Resulted average minimum RMSE and MPE are 0.965kJ/kg and 1.53%, respectively. Therefore, it is concluded that simulated values are in good agreement with monitored results. However, validation errors in cooling capacity form are slightly high with average minimum values of 2.872 kW and 6.76% but well within acceptable range.

Table 7.14: Validation errors with specific outlet enthalpy based on RMSE

<table>
<thead>
<tr>
<th>Validation Errors, RMSE (kJ/kg)</th>
<th>Average values</th>
</tr>
</thead>
<tbody>
<tr>
<td>19th June</td>
<td>29th June</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>2.53</td>
<td>---</td>
</tr>
<tr>
<td>3.14</td>
<td>3.25</td>
</tr>
<tr>
<td>2.71</td>
<td>1.82</td>
</tr>
</tbody>
</table>

Table 7.15: Validation errors with specific outlet enthalpy based on MPE

<table>
<thead>
<tr>
<th>Validation Errors, MPE (%)</th>
<th>Average of absolute values</th>
</tr>
</thead>
<tbody>
<tr>
<td>19th June</td>
<td>29th June</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>2.53</td>
<td>---</td>
</tr>
<tr>
<td>3.14</td>
<td>3.25</td>
</tr>
<tr>
<td>2.71</td>
<td>1.82</td>
</tr>
</tbody>
</table>
Finally, system model validation results of few exemplary days are presented in Figures 7.20 and 7.21. Figure 7.20 shows the simulated and monitored values in terms of outlet specific enthalpy, temperature and absolute humidity. In addition, simulated and monitored results based on cooling capacity are also presented in Figure 7.21. Moreover, simulated and monitored cooling energy delivered by the whole system is also shown. All validation results confirm that the developed model of desiccant cooling system is capable to predict the real system performance with good accuracy.
Figure 7.20: Comparison between monitored and simulated results of system model based on 19th June (A), specific enthalpy (B), and temperature and absolute humidity at outlet (C).
Figure 7.21: Comparison between monitored and simulated cooling capacity of system model based on 2nd July based on 19th June calibration (A), specific enthalpy (B), and cooling energy delivered (C).

7.4 Conclusions

In the current chapter, calibration and validation of desiccant cooling system model based on the ENERGYbase system design is performed at both component and system levels. The real transient operating conditions are used. The existing component models in LBL Buildings library can only be used for the ideal component operation. However, the actual performance of such components is significantly differs from the ideal operation in reality. Therefore, humidifier and heat wheel component models are modified to match the performance in real operation. Thus, additional variables are added in the models that are determined by developing several functions based on the real transient measurements obtained from monitoring system of ENERGYbase.

Afterwards, the modified component models are calibrated and validated with the six different days transient operating conditions. Calibration and validation errors are presented in two categories, RMSE and MPE. The resulted errors of all component models are almost less than 1% showing that simulated values are well in agreement with the monitored results. Following the calibration and validation at the component level, desiccant cooling system model is developed with the validated models. Then
system model is validated with respect to outlet specific enthalpy and system cooling capacity as objective functions. The resulted RMSE and MPE are slight higher than at component level but well within acceptable range.
Performance of desiccant evaporative cooling (DEC) systems is strongly influenced by the climate conditions. Ambient air dry bulb and wet bulb temperatures along with absolute humidity are key climate factors affecting the performance of DEC. Thus, DEC systems performance varies widely in different climate zones. In current chapter, five different climate zones are selected to analysis the performance of DEC system using equation-based object-oriented (EOO) modeling and simulation approach. The proposed approach can also be used for multifunctional system modelling and simulation analysis. In addition, optimal configurations of DEC system are also configured for specific climate zones. Therefore, DEC system models are developed capable of varying system configurations and climate conditions. ENERGYbase system design is used in the current study for performance analysis.

8.1 Desiccant cooling system configurations

In general, configurations of desiccant cooling systems are categorized through several characteristics as detailed in chapter 3. In order to improve the performance of DEC systems staged regeneration and multi-stage desiccant cooling were proposed. However in hot and humid climates, the cooling capacity of DEC systems would be limited. Therefore, DC system are usually incorporated with other air conditioning technologies to constitute hybrid system, such as hybrid desiccant system with VAC system, absorption chiller, adsorption chiller, and chilled-ceiling [206]. However, current study is focused on DC system configurations based on operating cycles.

8.1.1 Ventilation cycle system configuration

Ventilation cycle is the first rotary desiccant cooling cycle introduced by Pennington in 1955. Figure 8.1 shows the schematics of cycle along with heat and mass transfer processes on psychrometric chart. Ambient air enters in the system as process air at point 1. During the way through desiccant wheel, process air is dehumidified and air temperature increases due to heat of adsorption. Thus, air is dry and warm at point 2. Then dry and hot air is sensibly cooled in a heat wheel from state point 2-3. Afterwards, process air is further cooled and humidified through supply direct humidifier in order to achieve desired conditions of supply air. On the regeneration air side, state 5 shows the state of return air from space that is cooled and humidified by the return direct humidifier. Then cold and humid return air is pre-heated through heat wheel by the heat exchanged from the hot process air. The hot air is then further heated through the heat source from point 7-8 to achieve the desired regeneration temperature depending on the
nature of adsorbent material. Finally, desiccant wheel is regenerated by hot air that would be exhausted to atmosphere at point 9 [206].

A modified form of ventilation cycle is also proposed for the situations in which the building exhaust cannot be incorporated for co-processing. In such systems, ambient air is used for regeneration as shown in Figure 8.2. However, the thermal performance of such cycle is lower than standard cycle in terms of specific cooling capacity and coefficient of performance (COP). The key reason is high temperature and humidity ratio of ambient air compared to return air.

![Figure 8.1: Schematic and psychrometric representation of standard ventilation cycle](image-url)
8.1.2 Recirculation cycle system configuration

Recirculation desiccant air cooling cycle is a variant of standard ventilation cycle to increase the cooling capacity of the system. In such a cycle, return air is reused as process air and ambient air is used for regeneration as elaborated in Figure 8.3. The COP of recirculation cycle is commonly not higher than 0.8 due to relatively low temperature and humidity ratio. The key drawback of such cycle is lack of fresh air provided by the system because it employs 100% recirculation [206]. However, ventilation air can be easily added to the return air according to requirement. The system is termed as ventilated-recirculation system as depicted in Figure 8.4 [207]. The amount of ventilation air for commercial and institutional buildings is approximately 10-40% of outdoor air [98].
Figure 8.3: Schematic and psychrometric representation of standard recirculation cycle
8.1.3 Dunkle cycle system configuration

Dunkle cycle defines another configuration of descant cooling systems that combines the advantages of ventilation and recirculation cycles. An additional heat exchanger is integrated in the system to take the advantages of relative low temperature of supply air and large cooling capacity associated with ventilation and recirculation cycles, respectively. The schematic of dunkle cycle is shown in Figure 8.5. Dunkle cycle also has limited use due to lack of fresh air [206]. Though, ventilated air can be introduced to make it more practical, called ventilated-dunkle cycle [207] as shown in Figure 8.6.
Figure 8.5: Schematic and psychrometric representation of standard dunkle cycle
8.2 Model development of desiccant cooling system

In the current chapter, desiccant cooling system model is developed based on the second methodology described in chapter 4. The approach is termed as “Redeclaration/Replaceable component models” consists of component development and implementation phases [128]. The approach is executed on key component models of existing ENERGYbase solar desiccant cooling system including desiccant wheel, heat wheel, humidifier, fan/ventilator, and solar collector. Moreover, all component models contain an additional empty model based on the “frictionless pipe model” that does not change fluid properties at all and used as a conduit. The empty model is used for the system performance analyses when either the component model is dysfunctional or not required in the system operation. In current approach, the component models are used from LBL Buildings library [136] except desiccant wheel and solar thermal collector component models.
8.2.1 Desiccant wheel component models

In the overall model development, the base model provides a platform consisting of fluid ports for both process and regeneration air. Two desiccant wheel models are incorporated through current approach by extending base model. First is a simplified model, developed in Dymola/Modelica environment is based on heat and mass transfer relations proposed in a study [208]. The other model is more detailed with respect to the real-time control strategies described in chapter 5. An additional empty model is also introduced as “no desiccant wheel” with only fluid ports and without any heat and mass transfer. The empty desiccant wheel model is used to analyze the system performance without desiccant wheel. Figure 8.7 shows the model options for desiccant wheel.

At the implementation stage, desiccant wheel models package are declared “replaceable” and constrained by the base model, in order to ensure that all component models are in the change class list as elaborated in Figure 8.7.

Figure 8.7: Component models of desiccant wheel

8.2.2 Heat exchanger component models

Heat exchanger component models are used for simulating heat wheel. Like desiccant wheel model, the base model comprises of fluid ports for primary and secondary air streams. Four different types of heat exchanger component models are used including ideal heat exchanger with Constant-Effectiveness, Modified heat exchanger with constant effectiveness, Dry-Effectiveness-NTU, and Constant-Effectiveness-Mass [136]. All component models are extended from the base model. Additionally, an empty heat exchanger model is also included to analyze system performance without heat wheel. To implement all component models at the system level, the models are declared as “replaceable”, again constrained-by from the base model. During model development, five options can be used from the change-class list as shown in Figure 8.8.
8.2.3 Humidifier component models

The base model of humidifier component models consists of humidifier control signal and conditional water temperature ports in addition to fluid inlet and outlet ports. Three possibilities are included in change-class list including ideal humidifier \[136\], modified humidifier, and an empty model as shown in Figure 8.9. The empty humidifier component model does not account for heat and mass transfer. It is used to determine system performance without humidifier. Again, all component models are declared “replaceable” and constrained by the based model.
8.2.4 Heat source component models

For heat source component model, three model options are used comprising of electric heater, boiler, and constant temperature heater. The base model for all three options consists of fluid inlet and outlet ports with control signal that determines the fluid outlet temperature based on heat flow rate. The three model change possibilities are available in the change-class list after declaring them “replaceable” and constrained by the base model as elaborated in Figure 8.10.

![Figure 8.10: Component models of heat source](image)

8.2.5 Solar thermal collector and energy storage component models

The ENERGYbase desiccant cooling system is based on solar thermal system for regeneration of desiccant wheel. Therefore, the developed overall system model also includes solar thermal collectors. Two solar collector component models, simplified and enhanced developed by Austrian Institute of Technology (AIT) are considered [179]. The base model for solar collectors includes inputs of global radiation, wind speed, and ambient temperature. The change-class option is used to select any of the three models after declaring them “replaceable” and constrained by the base model as depicted in Figure 8.11. The empty model possibility determines the system performance without any solar collector. Moreover, two thermal energy storage component models are also considered for the proposed approach as shown in Figure 8.12. The first model is Stratified component model and the other is StratifiedEnhanced component model [136]. An empty model is also defined to analyze the system performance without a storage device.
8.2.6 Fan/ventilator component models

In general, two fans/ventilators are used in a desiccant cooling system, supply and return fans. However, different fan component models are available in LBL Buildings library [179]. Fan component models are ideally controlled by normalized speed, speed $N_{rpm}$, head $dp$, or mass flow rate. Through proposed approach, all fan component models are included as shown in Figure 8.13.
In several industries, especially in chemical industries model of analysis of dysfunctional systems (MADS) is mostly considered for safety and risk assessment [209]. However, such approach is not commonly used in air conditioning industry. Therefore, various features of multifunctional modeling are being incorporated in Dymola/Modelica environment [210] [211].

HAVC system consists of a multi-domain of systems and sub-systems, with all interconnected functions that must be properly orchestrated. Initial design stage of air conditioning systems, model development and simulations for performance assessment is crucial tasks. The primary objective of designing a good HVAC system is achieving the required level of comfort with minimized environmental impacts. Intend of system modeling and simulation is to track and foresee the impact of system and sub-system on one another during these operations while ensuring occupants comfort and health. Typically, an air conditioning system within its life span, experience several operation implications in terms of routine full or partial shutdown and start-up to conduct inspection, maintenance along with dismantle activities. Therefore, dysfunction air conditioning systems can cause severe implications, especially in health sectors. Consequently, the multifunctional modeling and simulations can be very helpful for overall system, human and climate safety.

The proposed methodology “Redeclaration/ Replaceable component models” presents multifunctional approach to simulate variety of components of HVAC systems. It is also a way forward to analyze system performance in both, functional mode related to design aspects, and dysfunctional mode focusing on safety characteristics. For dysfunctional mode, the empty/no component model option is used for each component of desiccant cooling system. Additionally, such option can be used to evaluate different system configurations. For example, desiccant cooling system with or without supply

**Figure 8.13: Component models of fan**

**8.3 Model development of multifunctional desiccant cooling system**

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humidifier, return humidifier, solar thermal system, or heater etc. Through proposed approach, any available component model can be selected by changing the class of respective component, such as supply and return humidifier, and heater are not considered as shown in Figure 8.14. Consequently, simulations of such system model predict the performance when the components are out of order or not desired during the system operation.

Figure 8.14: Multi-functional system model

8.4 Climate zones

Desiccant evaporative cooling systems involve evaporation process that is significantly influenced by the ambient conditions. Therefore, performance of whole system is dependent on the climate conditions in which the system operates. The current study is focused on system performance in different climate zones.

The climate zones are classified in several ways in various studies. In general, Köppen [212] and ASHARE [191] climate zone classifications are widely used. Köppen climate classification is based on the native vegetation concept. The boundaries of climate zone are decided with respect to vegetation distribution. This classification system divides world climates in five groups, such as A, B, C, D, and E. Each major group is further categorized in different types and subtypes, i.e., Af, Am, Aw etc. However, in current study only three major groups with different subtypes are used to have five climate zones briefly described as follow:
- BWh: Group B represents dry (arid and semi-arid) climates. The subtypes are decided based on the threshold values with respect to the annual precipitation received. The threshold value in millimeter is defined as:

\[ \lambda_{thres} = T_{avg\text{annual}} \times 20C + a \text{ or } b \text{ or } c \]

\( a.280 \text{ if precipitation received is } 70\% \text{ or more of the total precipitation} \)

\( b.140 \text{ if } 30 - 70\% \text{ if total precipitation} \)

\( c.0 \text{ if less than } 30\% \text{ if total precipitation} \)

The subtype BW characterizes desert climate where the annual precipitation is less than 50% of threshold. The subtype BS represents steppe climate with annual precipitation of 50-100% of threshold value. The third letter i.e. h or k indicates the temperature in which h specifies low latitude climate with average annual temperature above 18°C. Whereas, k represents middle latitude climate in which average annual temperature is below 18°C. Few examples of BWh climate include Isfahan (Iran), Mendoza (Argentina), Turpan, Xinjiang (China), Karachi (Pakistan) etc.

- Cwa, Cfa and Csa: Group C signifies mild temperate /mesothermal climates. Such climates have an average temperature above 10°C during summer, i.e. April to September in north hemisphere and during winter, average temperature is between -3°C to 18°C. In Group C, the second letter e.g. w, s, or f represents the precipitation profile, indicating dry winters, dry summers, and significant precipitation in all seasons. For dry winter condition, the driest winter month has average precipitation of less than 1/10th of wettest summer month precipitation indicating that driest winter month have less than 30mm average precipitation. However, the driest summer have less then 30mm average precipitation and less than 1/3rd of wettest winter month precipitation. Whereas, f represents the conditions in which neither dry winter nor dry summer are satisfied. The third letter “a” of considered climates indicates the average temperature of hottest month above 22°C. Few examples of Cfa and Cwa also known as humid subtropical climates include Huston, Texas (United States), Shanghai (China), etc.; and Sao Paulo (Brazil), Lahore (Pakistan), etc., respectively. Adelaide (Australia), Sanremo (Italy), Porto (Portugal) etc. have climate classification Csa that is termed as dry-summer subtropical climates.

- Dfb: Group D is representation of continental/Micro-thermal climates. Such climates have average temperature above 10°C in warmest months and below -3°C in the coldest season. The second and third letters are indicating the same subtype as of group C. The type Dfb represents warm summer continental climates, such as Vienna (Austria), Moscow (Russia), etc.

The climate zones classification is also established by ASHARE 169-2006 standard. The climate zones are assigned different numbers from 1 to 7 based on cooling degree.
days (CDD) and hot degree days (HDD) at specific dry bulb temperatures. However, in current study, only first five climates are considered for performance analysis of a desiccant cooling system in five different cities. The climate zone 1 is further categorized in two parts, 1A and 1B. The 1A represents very hot-humid climate and 1B indicates dry climate both having conditions of 5000 < CDD10 °C. The climate zone is for hot-humid (2A) and dry (2B) with 3500 < CDD10°C ≤ 5000. The climate zone 3 is further divided in 3 types, 3A signifies warm-humid and 3B for dry with 2500 < CDD10°C < 3500, and 3C represents warm-marine with CDD10°C ≤ 2500 AND HDD18°C ≤ 2000. Similarly, 4A climate zone is mixed-humid and 4B indicates dry climate, both with CDD10°C ≤ 2500 AND HDD18°C ≤ 3000 and 4B represents mixed-marine with SI Units 2000 < HDD18°C ≤ 3000. Whereas, climate zones 5A shows cold-humid, 5B specifies dry, and 5C indicates marine climates, all with 3000 < HDD18°C ≤ 4000.

The objective of mentioning both methods is to indicate that the selected five cities have different climate conditions in both approaches. Table 8.1 shows selected cities and their classification in both methods along with their respective cooling design days. Additionally, the selected cities are highlighted on world map according to Köppen climate classification in Figure 8.15. The climate conditions in terms of dry-bulb temperature, relative humidity, wind speed, and global horizontal radiation of the selected cities according to 12 hours (7AM to 7PM) cooling design day data are shown in Figure 8.16 [213].

Table 8.1: Climate zones and cooling design days of selected cities.

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Location (country)</th>
<th>Latitude and longitude</th>
<th>ASHRAE climate classification</th>
<th>Köppen climate classification</th>
<th>Cooling design day</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Karachi (Pakistan)</td>
<td>24.85° N and 67.02° E</td>
<td>1</td>
<td>BWh</td>
<td>21st June</td>
</tr>
<tr>
<td>2</td>
<td>Sao Paulo (Brazil)</td>
<td>23.5° S and 46.62° N</td>
<td>2</td>
<td>Cwa</td>
<td>21st February</td>
</tr>
<tr>
<td>3</td>
<td>Shanghai (China)</td>
<td>31.2° N and 121.5° E</td>
<td>3</td>
<td>Cfa</td>
<td>21st July</td>
</tr>
<tr>
<td>4</td>
<td>Adelaide (Australia)</td>
<td>34.93° S and 138.58° E</td>
<td>4</td>
<td>Csa</td>
<td>21st February</td>
</tr>
<tr>
<td>5</td>
<td>Vienna (Austria)</td>
<td>48.208° N and 16.37° E</td>
<td>5</td>
<td>Dfb</td>
<td>21st July</td>
</tr>
</tbody>
</table>
Figure 8.15: Selected cities according to Köppen climate classification

Figure 8.16: Climate conditions of selected cities with respect to cooling design day; dry-bulb temperature (A), relative humidity (B), global horizontal radiation (C), and wind velocity (D)
8.5 Description of desiccant cooling system

The ENERGYbase is an office building built in Vienna, Austria. The building is designed as a passive house in which renewable energy technologies are used to fulfill HVAC and lighting energy requirements. Both air and water based heating and cooling systems are designed to achieve the desired comfort conditions. The air handling system is only considered to control space humidity and to supply fresh air to the offices. Thus, air conditioning system covers only the latent loads of the offices. Moreover, air conditioning in summer is provided by utilizing solar energy during regeneration process of desiccant cooling system. The desiccant cooling system of ENERGYbase is comprised of two identical units of each 8,240 m$^3$/h design flow rate. However, single unit is considered in the current study. In the cooling mode, the supply air conditions are maintained with moderately cool temperature of 22°C and relative humidity below the maximum values of 80%. Similar to other desiccant cooling systems, during cooling mode, several operation modes are activated in a specific order according to the load requirements. The modes include ventilation, heat recovery, return air adiabatic cooling, solar heat regeneration, supply air adiabatic cooling, and auxiliary heater regeneration modes. The technical design aspects of the desiccant cooling system are already defined in Table 7.1. Moreover, the conditioned air is supplied to the offices through displacement distribution system [214][215].

8.6 Model development of desiccant cooling system configurations

Configuration models of ENERGYbase desiccant cooling system are developed in Dymola/Modelica. The modified validated individual system component models are used to develop overall system model. The system configuration models are developed using the “component model replaceable/redeclaration” approach described in section 8.4. In such approach, several component model options can be employed to analysis the system performance.

8.6.1 Ventilation cycle model development

In ventilation cycle, ambient air is used as process air and return air from the building is employed for system regeneration. The flow rate of both air streams are kept constant at same design value of 2.5 kg/s (8,240 m$^3$/h). The component model ‘Weather_Data’ contains all relevant ambient weather conditions based on respective design day of all five climates in terms of dry bulb temperature T$_{amb}$, relative humidity RH$_{amb}$, wind speed v$_{amb}$, and global radiation G$_{amb}$. The data is used from the weather data file of EnergyPlus. The room’s return air conditions are kept constant at temperature, T$_{ret}$ 26.7°C and RH$_{ret}$ 50% relative humidity. Likewise, desiccant wheel regeneration temperature, T$_{reg}$ is also kept constant at 70 °C through constant temperature heater component model. Figure 8.17 shows the Dymola representation of ventilation cycle model.
8.6.2 Recirculation and ventilated-recirculation cycle model development

In recirculation cycle model, return air is used as process air. While, ambient air is employed for system regeneration as shown in Figure 8.18. However, standard recirculation cycle lacks the fresh air supply that is undesirable for most practical applications. Therefore, fresh outside air is introduced to develop a ventilated-recirculation cycle model as depicted in Figure 8.19. A specified amount of fresh air (10-40%) with increment of 5% is considered in the model. A mixing box component model is used to mix fresh air with the return air keeping overall flow rate constant.
Figure 8.18: Dymola representation of standard recirculation cycle model

Figure 8.19: Dymola representation of ventilated-recirculation cycle model
8.6.3 Dunkle and ventilated-dunkle cycle model development

An additional heat exchanger is incorporated in dunkle cycle model as shown in Figure 8.20. The heat exchanger helps to enhance the system cooling capacity by utilizing ambient conditions. The standard dunkle cycle also faces same demerit of lacking fresh air like standard recirculation cycle. Therefore, standard cycle model is modified to induct specified amount of fresh air, termed ventilated-dunkle cycle model as shown in Figure 8.21.

Figure 8.20: Dymola representation of standard dunkle cycle model
ENERGYbase desiccant cooling system uses solar thermal energy to achieve required regeneration temperature 70 °C of desiccant wheel. Therefore, a solar thermal system model is developed using validated solar collector component model developed by AIT [179] as presented in Figure 8.22. The technical specifications of whole solar thermal system and solar collector are given in Table 8.2. The solar collector component model has three input ports of ambient temperature $T_{amb}$, wind velocity $Wind_{vel}$, and global radiation $Global_{rad}$. The ambient air conditions are stored in ‘Weather_Data’ component model. Two pump component models are used. Solar pump component model circulates water between the heat exchanger and solar collector, while primary pump provides water flow between the two heat exchangers. One heat exchanger is used with solar collector and other between solar system and desiccant wheel.
Table 8.2: Technical data of solar thermal system

<table>
<thead>
<tr>
<th>Design parameter</th>
<th>Design value</th>
</tr>
</thead>
<tbody>
<tr>
<td>System</td>
<td></td>
</tr>
<tr>
<td>Solar collector field (m$^2$)</td>
<td>274</td>
</tr>
<tr>
<td>Heat storage capacity (m$^3$)</td>
<td>15</td>
</tr>
<tr>
<td>Primary pump flow rate (kg/s)</td>
<td>0.83</td>
</tr>
<tr>
<td>Solar pump flow rate (kg/s)</td>
<td>1.25</td>
</tr>
<tr>
<td>Solar collector</td>
<td></td>
</tr>
<tr>
<td>First coefficient of the efficiency equation, $a_0$ ($\text{m}^2\text{K}$)</td>
<td>0.5018</td>
</tr>
<tr>
<td>Coefficient measured in normal ventilation conditions, $a_1$ _wind ($\text{W/m}^2\text{K}$)</td>
<td>3.486</td>
</tr>
<tr>
<td>Coefficient measured no wind, $a_1$ _nowind ($\text{W/m}^2\text{K}$)</td>
<td>3.1689</td>
</tr>
<tr>
<td>Third coefficient of the efficiency equation, $a_0$ ($\text{W/m}^2\text{K}$)</td>
<td>3.105e-5</td>
</tr>
<tr>
<td>Conductive coefficient (W/K)</td>
<td>21.255</td>
</tr>
<tr>
<td>Overall capacitance of material (kJ/K)</td>
<td>650</td>
</tr>
</tbody>
</table>

8.7 Performance analysis of desiccant cooling system configurations

In current study, performance analysis of ENERGYbase desiccant cooling system is performed in five different climate zones. Three different system configurations are analyzed, i.e. ventilation cycle, recirculation cycle, and dunkle cycle system configuration. The performance analysis presents the feasibility investigation to decide suitability of the referred desiccant cooling system in selected climates. The whole analysis is based on the climate conditions with respect to respective cooling days of each climate zone. The climate conditions are obtained from the weather data file of EnergyPlus software for 12 hours (7am -7pm) with 1 hour time interval. Moreover, three performance parameters are used for overall system analysis that includes system
cooling capacity (kW), cooling energy delivered (kWh), and COP through Equations 8.2, 8.3, and 8.4, respectively [95]. Two control strategies are implemented at the system level.

\[
CC = \dot{m} (h_{amb} - h_{supply}) \tag{8.2}
\]

\[
COP = (h_{amb} - h_{supply}) / (h_{heaterout} - h_{heaterin}) \tag{8.3}
\]

\[
En = \int_{t_0}^{t_f} CC \tag{8.4}
\]

### 8.7.1 Fixed-return and fixed-humidifier control

In the first strategy, the conditions of return air are kept constant at temperature 26.7°C and relative humidity 50% according to ARI rating of room air. Additionally, the control of supply and return humidifiers is also fixed with respect to their control inputs based on the dehumidification capacity and heat loss. The functions used for two control inputs are fixed for all climates based on the minimum validated error results as described in chapter 7. The value of pump position of supply and return humidifiers is also kept constant at 50% and 100%, respectively based on the resulted values from the ENERGYbase monitoring system. Finally, the first strategy results in different conditions of supply air.

### 8.7.2 Fixed-return and fixed-supply-air control

The system control strategy is based on fixed return air conditions similar to first strategy. However the supply air conditions are control by the fixed supply air absolute humidity. The design value of supply air absolute humidity is determined from ENERGYbase design ventilation load. In addition, infiltration load is also considered. The design ambient absolute humidity for each climate is determined by the ASHRAE standard [98]. For example, the set point absolute humidity for Vienna is determined by the following equations.

The latent load is calculated by Equation 8.5.

\[
q_l = 3010 \dot{V} \Delta \omega \tag{8.5}
\]

While, Equation 8.6 is used to determine \( q_t \).

\[
q_t = n f_{occ} q_{lat} \tag{8.6}
\]

The number of people, \( n \) is decided by the occupancy density defined in ASHARE standard [216] based on 100 m² area, i.e. 5/100 m² for office space. Whereas, the total area of conditioned is around 3000 m² space for each unit of system. Therefore for ENERGYbase office building, the number of people is 150. The standard value of \( q_{lat} \) for office activities is 45 kW. Therefore, using the value of \( q_t \) at 100% occupancy i.e.
Infiltration load is also accounted to find out the set point value of absolute humidity. Typical values of infiltration rate are between 0.01-0.045 for passive buildings [217]. In current study, infiltration rate of 0.045 is considered assuming the worst passive building. The design ambient absolute humidity value of Vienna is 14.7 g/kg [98]. The calculated moisture added to the space by infiltration is 0.334 per kg of air with 2.5kg/s of air flow rate. Thus, the total absolute humidity difference between supply and indoor space including latent and infiltration loads is 1.402 g/kg. While the design indoor space value is maintained at 9.2 g/kg. Therefore, the supply air set point value of absolute humidity is 7.79 g/kg. Finally, moisture added per kg of air by infiltration is calculated for all five selected climate zones.

Figure 8.23 shows the control strategy of supply humidifier to achieve the set point values of absolute humidity for all climate zones. The absolute humidity sensor at the humidifier outlet measure initial value and then maintains the set point value through feedback and PI controller component models. Similarly, the return humidifier is also control such that the air conditions at its outlet are always at the saturation point, i.e. 100% relative humidity as elaborated in Figure 8.24.
8.7.3 Control strategy of solar thermal system for regeneration temperature

In the current study, two different approaches are implemented with respect to control of regeneration temperature to analyze the three configuration of desiccant cooling system. In the first approach, a constant temperature heater is used to achieve required regeneration temperature fixed at 70 °C. However, in the second methodology, a combination of prescribed flow heater and solar thermal system is used through a heat exchanger. The solar thermal system provides hot water based on respective climate zone. The hot water is used to heat up the return air through heat exchanger. Though, an additional heater is incorporated during the time when solar energy is not enough to achieve required regeneration temperature of 70°C. Similar to the humidifier control, the prescribed heater is control to ensure the required regeneration temperature as shown in Figure 8.25. The temperature sensor measures the initial temperature at the outlet of heater and then controls its heat flow to maintain 70°C temperature through feedback and PI controller.

Finally, Figure 8.26 shows an example of an overall model of solar assisted desiccant cooling system based on control strategies of supply and return humidifier, and solar thermal system.
8.7.4 Vienna climate zone

In Vienna climate zone, the performance of desiccant cooling system configurations is presented applying both control strategies with respect to cooling capacity, COP, and energy delivered by the each system configuration.

The first strategy resulted in a slightly higher cooling capacity of ventilation cycle during the mid-day compared to recirculation and dunkle cycles as shown in Figure 8.27(A). Lower values of dunkle cycle cooling capacity indicate that the additional heat exchanger in such climatic conditions does not enhance the system cooling capacity but
results in improved COP due to lesser input energy requirements for regeneration desiccant wheel as observed in Figure 8.27 (B). The recirculation cycle shows increased total delivered energy compared to other cycles as depicted in Figure 8.27 (C).

Figure 8.28 and 8.29 show the effects of variation in fresh air induction in recirculation and dunkle cycles from 5% to 40%. The cooling capacity and COP of ventilated recirculation and dunkle cycles increase with the increasing percentage of fresh air. The results show that fresh air of Vienna enhances the heat and mass transfer processes of regeneration air through return humidifier, heat wheel, and descant wheel that ultimately increases cooling capacity and COP of both cycles.

![Figure 8.27](image)

**Figure 8.27: 1st strategy for Vienna: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)**
Figure 8.28: 1st strategy for Vienna: Performance analysis of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.29: 1st strategy for Vienna: Performance analysis of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

The second control strategy uses fixed supply air conditions with respect to absolute humidity resulted in almost similar trends as seen using the first control strategy (Figure
The set point value of absolute humidity is 7.79 g/kg for Vienna. The overall system performance is high in terms of cooling capacity and COP compared to first strategy. The cycle performance comparison with respect to COP is also reflected in Figure 8.31. The Figure shows that the energy requirement of the dunkle cycle is the lowest compared to other resulting in higher COP (see Figure 8.30 B). It is due the fact that the energy requirement in dunkle cycle is almost fulfilled by the solar energy. The negative values seen in Figure 8.31 A show that the heater acts as a cooler when the regeneration temperature is above 70°C. In practice, a mixing valve is used to maintain the desired 70°C, the maximum allowable limit for LiCl. In addition, the trends of ventilated recirculation and dunkle cycles are also similar to first control strategy with better system performance as shown in Figure 8.32 and 8.33.

Figure 8.30: 2nd strategy for Vienna: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
Figure 8.31: Vienna: Energy input requirement of three cycles with respect to: heater and solar system (A), temperature at inlet heat exchanger of solar system (B)

Figure 8.32: 2nd strategy for Vienna: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
Figure 8.33: 2nd strategy for Vienna: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

8.7.5 Karachi climate zone

The ambient conditions of Karachi represent hot-humid climate that favor dunkle cycle in terms of high cooling capacity and cooling energy delivered by the system as shown in Figure 8.34. However, the ventilation cycle has high COP that indicates the less energy requirements to maintain 70°C. The performance of ventilated-recirculation and dunkle cycles decreases with the addition of more fresh air. It shows that the ambient conditions of Karachi do not enhance the heat and mass transfer processes in the regeneration side of desiccant cooling system as shown in Figure 8.35 and 8.36.

The implementation of 2nd control strategy with Karachi climate conditions shows that the set point value of absolute humidity 7.31 g/kg can be achieved without supply humidifier. However, skipping the supply humidifier causes high temperature of supply air outside the comfort zone. Therefore, an additional sensible cooler would be required to achieve comfort supply air temperature.
Figure 8.34: 1st strategy for Karachi: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.35: 1st strategy for Karachi: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
Figure 8.36: 1st strategy for Karachi: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

8.7.6 Sao Paulo climate zone

In the Sao Paulo climate zone, the first control strategy shows that the performance of standard recirculation cycle configuration in terms of cooling capacity is much better than the other two cycles as shown in Figure 8.37. Although, the low COP of recirculation cycle represents high input energy demands compared to other cycles. Moreover, the induction of fresh air in the ventilated-recirculation cycle decreases the system performance with low cooling capacity as shown in Figure 8.38. Whereas, the impact of fresh air is almost negligible for ventilated-dunkle cycle as presented in Figure 8.39.
Figure 8.37: 1st strategy for Sao Paulo: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.38: 1st strategy for Sao Paulo: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
The results of second control strategy with set point value of absolute humidity 7.60 g/kg showed that all three options have minor difference in their performance as shown in Figure 8.40. However, the dunkle cycle is more efficient in terms of high COP as it has less energy requirements for heater as shown in Figure 8.41. Additionally, Figure 8.42 and 8.43 show that the performance of ventilated recirculation and dunkle cycle is independent of amount of fresh air inducted in the system.
Figure 8.40: 2nd strategy for Sao Paulo: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.41: Sao Paulo: Energy input requirement of three cycles with respect to: heater and solar system (A), temperature at inlet heat exchanger of solar system (B)

Figure 8.42: 2nd strategy for Sao Paulo: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
Figure 8.43: 2nd strategy for Sao Paulo: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

8.7.7 Shanghai climate zone

Shanghai climate conditions represent humid subtropical weather. In such a climate, the performance of the both recirculation and dunkle cycles is similar and much better in terms of cooling capacity compared to the ventilation cycle as shown in Figure 8.44. However, the dunkle cycle is built with an additional heat exchanger that increases the initial system cost. In addition, the two cycles required more energy input compared to the ventilation cycle that has high COP. Figure 8.45 shows that the cooling capacity of ventilated recirculation cycle with 10% fresh air is higher than other amount of fresh air. The COP of the cycle with different amount of fresh air is almost same except with 5% fresh air. Though, the performance of ventilated dunkle cycle decreases with increased amount of fresh air as shown in Figure 8.46.

The implementation of second control strategy with set point value of absolute humidity 7.34 g/kg for Shanghai climate zone resulted that the supply humidifier is not feasible in such a humid climate. The supply humidifier increases the absolute humidity more than the desired value. However, an additional sensible cooler is required to achieve the required sensible load.
Figure 8.44: 1st strategy for Shanghai: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.45: 1st strategy for Shanghai: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
Figure 8.46: 1st strategy for Shanghai: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

8.7.8 Adelaide climate zone

The Adelaide city is representation of dry-summer subtropical climates. The performance of the ventilation cycle is slightly better than other two cycles according to the first control strategy as shown in Figure 8.47. Though, the additional heat exchanger enhances the COP of dunkle cycle. Moreover, the effects of different amount of fresh air are less significant in both ventilated recirculation and ventilated dunkle cycles as shown in Figure 8.48 and 8.49, respectively.
Figure 8.47: 1st strategy for Adelaide: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C).

Figure 8.48: 1st strategy for Adelaide: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C).
Figure 8.49: 1st strategy for Adelaide: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C).

The second control strategy is also implemented based on the fixed value of absolute humidity of 8.03 g/kg. The resulted system performance of recirculation cycle is almost similar to the ventilation cycle as shown in Figure 8.50. Though, the COP of these cycles is less than the dunkle cycle that shows the low input energy requirements with additional heat exchanger as shown in Figure 8.51. Moreover, the effects of varying amount of fresh air on the performance of ventilated recirculation cycle are negligible as shown in Figure 8.52. While the performance of ventilated dunkle cycle decreases with increasing amount fresh air as elaborated in Figure 8.53.
Figure 8.50: 2nd strategy for Adelaide: Performance comparison of three cycles with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C).

Figure 8.51: Adelaide: Energy input requirement of three cycles with respect to: heater and solar system (A), temperature at inlet heat exchanger of solar system (B).
Figure 8.52: 2nd strategy for Adelaide: Performance comparison of ventilated-recirculation cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)

Figure 8.53: 2nd strategy for Adelaide: Performance comparison of ventilated-dunkle cycle with respect to: cooling capacity (A), COP (B), and cooling energy delivered (C)
8.8 Summary of results

The performance analysis of three configurations of a desiccant cooling system in terms of the operational cycles is performed in five climate zones. Additionally, the standard recirculation and dunkle cycles are modified by the induction of fresh air to analyze more practical options. In the current study, two control strategies with respect to supply and return air streams are implemented for the performance analysis. The results of all three climates in which both strategies are feasible summarized in Table 8.3. In the table, average values of cooling capacity (CC) and COP are given, while the maximum values of cooling energy delivered (En) are provided. The arrow directions show the increasing (), decreasing ( ), or constant effects ( ). Table 8.4 provides the values of same performance parameters for Karachi and Shanghai based on the first control strategy. In such climate having more humid ambient conditions, the supply humidifier is not feasible. Instead, a sensible cooler or indirect humidifier can be used achieve desired set point value. However, remaining three climates have additional benefits of free cooling to reduce supply air temperature through supply humidifier.

Table 8.3: Performance summary of three climates

<table>
<thead>
<tr>
<th>Control strategy</th>
<th>System configuration</th>
<th>Climate zone</th>
<th>Vienna</th>
<th>Sao Paulo</th>
<th>Adelaide</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>CC</td>
<td>COP</td>
<td>En</td>
</tr>
<tr>
<td>1st</td>
<td>Ventilation</td>
<td></td>
<td>16.9</td>
<td>0.19</td>
<td>208.9</td>
</tr>
<tr>
<td></td>
<td>Recirculation</td>
<td></td>
<td>18.5</td>
<td>0.3</td>
<td>223.5</td>
</tr>
<tr>
<td></td>
<td>Dunkle</td>
<td></td>
<td>10.8</td>
<td>0.27</td>
<td>135.3</td>
</tr>
<tr>
<td></td>
<td>Ventilated-recirculation(5-40%)</td>
<td></td>
<td>19.2↑</td>
<td>0.29↑</td>
<td>232↑</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>23.9</td>
<td>0.33</td>
<td>287.3</td>
</tr>
<tr>
<td></td>
<td>Ventilated-dunkle(5-40%)</td>
<td></td>
<td>12.1↑</td>
<td>0.29↑</td>
<td>148↑</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>19.0↑</td>
<td>0.4</td>
<td>231.8</td>
</tr>
<tr>
<td>2nd</td>
<td>Ventilation</td>
<td></td>
<td>18.7</td>
<td>0.21</td>
<td>232.8</td>
</tr>
<tr>
<td></td>
<td>Recirculation</td>
<td></td>
<td>18.4</td>
<td>0.29</td>
<td>225.8</td>
</tr>
<tr>
<td></td>
<td>Dunkle</td>
<td></td>
<td>12.2</td>
<td>0.32</td>
<td>153.1</td>
</tr>
<tr>
<td></td>
<td>Ventilated-recirculation(5-40%)</td>
<td></td>
<td>16.5↑</td>
<td>0.3↑</td>
<td>234.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>20.7↑</td>
<td>0.3</td>
<td>-284</td>
</tr>
<tr>
<td></td>
<td>Ventilated-dunkle(5-40%)</td>
<td></td>
<td>8.4↑</td>
<td>0.25↑</td>
<td>148.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>16.1</td>
<td>0.36</td>
<td>-233</td>
</tr>
</tbody>
</table>
Table 8.4: Performance summary of two climates

<table>
<thead>
<tr>
<th>Control strategy</th>
<th>System configuration</th>
<th>Karachi</th>
<th>Climate zone</th>
<th>Shanghai</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>CC</td>
<td>COP</td>
<td>En</td>
</tr>
<tr>
<td>1st</td>
<td>Ventilation</td>
<td>71.5</td>
<td>2.46</td>
<td>857.8</td>
</tr>
<tr>
<td></td>
<td>Recirculation</td>
<td>89.7</td>
<td>1.44</td>
<td>1074.1</td>
</tr>
<tr>
<td></td>
<td>Dunkle</td>
<td>90.8</td>
<td>1.74</td>
<td>1089.2</td>
</tr>
<tr>
<td></td>
<td>Ventilated-recirculation (5-40%)</td>
<td>87.7</td>
<td>1.15</td>
<td>1050.3</td>
</tr>
<tr>
<td></td>
<td>Ventilated-dunkle (5-40%)</td>
<td>87.7</td>
<td>1.66</td>
<td>1058.9</td>
</tr>
</tbody>
</table>

8.9 Economic aspects of system configurations

The focus of current study is appropriate selection of desiccant cooling system configuration in a certain climate zone at the initial design stage. Therefore, only estimated initial cost is provided for economic considerations. The estimated initial cost of desiccant cooling system configurations includes the capital cost of individual components mentioned in Table 8.5 [218] [219]. The initial cost of the ventilation and recirculation cycles is same however the initial cost of the dunkle cycle is high due to the additional heat exchanger. Additionally, the duct arrangement of the heat exchanger also increases the overall configuration cost. However, the operating cost of the dunkle cycle is less due to efficient system operation.

Table 8.5: Estimated initial cost of desiccant cooling system configurations

<table>
<thead>
<tr>
<th>Component</th>
<th>Properties</th>
<th>Estimated initial cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Desiccant wheel</td>
<td>Adsorbent: LiCl</td>
<td>7,500</td>
</tr>
<tr>
<td></td>
<td>Substrate: Cellulose</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Diameter: 1.77 m, depth: 0.45m</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Pressure: 65 Pa</td>
<td></td>
</tr>
<tr>
<td>Solar collector</td>
<td>Type: Flat plate</td>
<td>70/m² x 274</td>
</tr>
<tr>
<td></td>
<td>Area: 274 m²</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hot water: 60-70 °C</td>
<td></td>
</tr>
<tr>
<td>Heat wheel</td>
<td>Type: Aluminum rotary</td>
<td>5,800</td>
</tr>
<tr>
<td></td>
<td>Diameter: 1.77 m, depth: 0.4m</td>
<td></td>
</tr>
<tr>
<td>Supply and return fans</td>
<td>Type: Spray humidifiers with 4 nozzles with Automatic cleaning process</td>
<td>1,150 x 2</td>
</tr>
<tr>
<td></td>
<td>Pressure: 235 Pa</td>
<td></td>
</tr>
<tr>
<td>Heat exchanger</td>
<td>Type: Plate-fin</td>
<td>1,100</td>
</tr>
</tbody>
</table>
8.10 Conclusion

In the current chapter, performance analysis of ENERGYbase desiccant cooling system is presented to select an appropriate configuration through a new modeling approach. The methodology ‘Replaceable/Redeclaration’ is proposed for system modeling in which various component models can be selected by changing the component class. Moreover, the approach incorporates empty component models for system multifunctional modeling and simulation of HVAC systems.

Three standard and two modified configurations of a desiccant cooling system based on the operating cycle are analyzed in five climate zones. The hourly climate data of each zone is selected with respect to design cooling day for 12 hour system operation. For each system configuration, two control strategies are implemented; fixed-return and fixed-humidifier control, and fixed-return and fixed-supply control. The system analysis is performed based on three parameters; cooling capacity, COP, and cooling energy delivered. In general, the dunkle cycle is the most appropriate configuration for Vienna, Sao Paulo, and Adelaide climate zones. While, the system performance is much better with ventilation cycle configuration in Karachi and Shanghai climates.
Chapter 09

9 CONCLUSIONS AND FUTURE WORK

In the current work, a systematic simulation-based optimization approach is proposed for the automated optimal selection of HVAC system configurations at the initial design stage. The particular conclusions for each chapter are provided in the closing remarks. However, in present chapter the key outcomes are compiled to give an overview of the research work. Additionally, directions of the future work are also highlighted.

9.1 Conclusions

The overall world’s energy demands are increased remarkably in last few decades and projected to ascend by almost 50% from 2009 to 2035. In the total energy consumption, the share of buildings is significant that accounted about 21% of total energy requirements at the world level. In the developed countries and regions, such as USA and EU the buildings energy demands are around 41% and 40% of the total energy consumption, respectively. While, in the underdeveloped countries like Pakistan, buildings share is about 44%. In the buildings energy demands, the key contribution is from HVAC systems that maintain the desired comfort conditions. Therefore, optimization of HVAC systems has great potential for energy savings in the buildings sector (Chapter 01).

Optimization of HVAC systems is performed in numerous research activities. A comprehensive literature review is performed in the study focusing on the various aspect of HVAC system optimization. The literature review resulted that the majority of studies are either related to the optimization of individual components or the operational control optimization at the system level while the system configurations analysis is rarely studied. Moreover, no systematic approach was found for the optimal selection of HVAC system configuration. Therefore, the current research work is focused to propose a systematic methodology for the optimal selection of HVAC system configurations (Chapter 02).

Selection of an optimal configuration of HVAC system is one of the crucial aspects at the initial design stage and significantly influences the overall building energy demands. Decision of HVAC system configurations comprises the type of system, the type and number of various components, and the operational control strategies along with the system economic and environmental characteristics. A comprehensive overview of several HVAC system components is presented that are being used to develop various configurations for different applications. Moreover, various categories of HVAC system configurations are identified including primary and secondary system configurations, and alternative and innovative system configurations. The primary system configurations consist of single and multiple chiller systems in series and parallel arrangement. While, the secondary system configurations represent...
the air distribution systems, such as single-zone single duct or dual duct system either with constant air volume or variable air volume configurations. The alternative and innovative system configurations utilize the renewable energy recourses, like various configurations of desiccant cooling systems and solar air conditioning systems. All such configurations highlighted the complexity of decision making to select an appropriate configuration for a particular load demand in a specific climate (Chapter 03).

In practice, HVAC system configuration and design selection assessment is dependent on the experience and skills of the system designer. Though, the experience and skills of an individual or a group are generally limited based on the expertise of the frequently designed systems. Therefore, a systematic simulation-based optimization approach is developed and validated for the optimal automated selection of HVAC system configuration.

The developed approach is based on the equation-based object-oriented (EOO) modeling and simulation environment, Dymola/Modelica. The EOO modeling and simulation is comprised of physical component models of several HVAC systems. Thus, the individual component models develop an overall HVAC system model through appropriate connections. Each component model represents a set of algebraic, differential, and event-triggered difference equations that are based on the thermodynamic principles. The characteristics of EOO modeling and simulation environment are described in detail. Similarly, the overview of Dymola/Modelica and optimization program GenOpt is also provided (Chapter 04).

In the current work, two model-based simulation and optimization methods are proposed to develop dynamic HVAC system models. The developed models are capable of varying different configurations by coupling Dymola/Modelica with GenOpt.

- **First system modeling method: Conditional declaration of component models**

  In first method, sub-component models of the overall system are conditionally declared. The conditional declarations are decided in view of the variation in the building load demands and climate conditions that act as model input. The method is implemented through Dymola/Modelica function and package, ‘readRealParameter’ and ‘ExternalData’, respectively. The function reads the values of real parameters from an external file used in the conditional declaration and assigns a specified value to each parameter. Then the specified value decides the certain inclusion or exclusion of a component model. The package enables Modelica reading and writing data from an external file. In the overall system model, the conditional parameters are properly incorporated with conditional ‘if’ statement with an appropriate logical condition. Finally, the method is elaborated through an example.
Second system modeling method: Redeclaration/Replaceable component models

The second method is composed of two phases. In the development phase, partial base models are developed for each component model that provides a platform for other types of component models from the same class. While in the implementation phase, each component model is declared as ‘replaceable’ and constrained by its respective base model. Afterwards, the ‘changeclass’ feature enables to select any type of the respective component model. Moreover, empty models are also developed with only connection ports but without any heat and mass transfer equations to analyze multifunctional system modeling. The method also has a limitation in the current version of Modelica language that the component model class cannot be changed from an external file, thus it does not allow automatic configuration change compared to the first method.

The present work is focused at both component and system level. At the component level, a desiccant wheel model is developed. The model based predictions of a desiccant wheel performance is presented using the transient measurements obtained from a real system. The model is based on a set of equations to simulate the optimal and measured transient performance as a function of measurable input variables related to the desiccant wheel material and structure. The model is adapted to analyse the influence of different working conditions on the desiccant wheel performance: rotation speeds, air velocity, inlet temperature, and inlet air humidity for both process and regeneration air. The model is capable of estimating the optimal rotation speed and pressure drop of the desiccant wheel. Moreover, the developed model can be applied in both, dehumidification and enthalpy modes. The model is validated in comparison to the published data and measurements from the real ENERGYbase building desiccant wheel installation. The specific enthalpy at the outlet of process air is considered performance parameter. The obtained results are in agreement with the published data, while the resulting maximum and minimum validation root mean square error (mean percentage error) between the simulated and measured transient performance is 3.6kJ/kg (4.6%) and 1.9kJ/kg (0.2%), respectively. The results are based on the transient measurements of six selected days having both operating modes. Additional measurements can be used to further enhance the range of operating conditions used for the validation of the model (Chapter 05).

The research work at the system level is performed with the both system modeling approaches. Two HVAC systems are considered for optimization and performance analysis of different configurations including a chilled water system and a desiccant system as a primary and secondary HVAC system, respectively.

The optimization of real chilled water system configurations installed in the USA is performed using the first system modeling approach of conditional declaration of component models. The optimization is achieved at the system design and
configuration levels. A dynamic system model is developed to vary and simulate five system design parameters and configurations. Two discrete variables are linked with the system configuration: number of chillers and number of cooling towers and three continuous variables are associated with the system design: building load demand, temperature difference across condenser, and cooling tower fan speed. The simulation-based optimization approach is implemented by coupling Dymola/Modelica with GenOpt in three different performance analysis strategies that represents an incremental methodology development. The first strategy is use fixed system design conditions to validate the approach with the real system, while the second strategy additionally vary the number of cooling towers according to the design flow turndown ratio to validate the best practice design criteria. In the third strategy, overall system optimization is performed simultaneously varying the system design and configuration parameters. However, the third strategy proved to be the most energy efficient and resulted with the 17.3-43.5% of energy savings. Additionally, the initial cost of each system configuration is also provided for economic consideration. The case study provides the proof of concept of the proposed methodology after the validity of all three strategies. However, the research work performed assumes that the chilled water system components sizes are predefined and optimizes the design and configuration parameters using constant chilled water supply and return temperatures along with dry bulb and wet bulb temperatures. In addition, measurement uncertainties related with the equipment selection are ignored as well as their potential impacts on the optimization results of the total power consumption (Chapter 06).

The performance of desiccant cooling system configurations is also analyzed. The desiccant system model is developed based on the real system design installed in the ENERGYbase building in Vienna. However, the available component models of desiccant cooling system such as humidifier and heat wheel are designed for the ideal performance analysis. Therefore, the component models are modified in terms of heat and mass losses normally occurred in the real system operation. Appropriate functions of each design parameter showing the depending of various input conditions are determined from the real system measurements through multiple linear regression (MLR) technique. Afterwards, each modified component model is calibrated and validated under the transient conditions obtained from ENERGYbase monitoring system. The resulted validation errors are nearly less than 1%. Then validated component models are used to develop the whole desiccant cooling system model. Then the calibration and validation is performed at the system level. The specific enthalpy of the supply air and system cooling capacity are used as a performance parameter. The resulted minimum validation errors of specific enthalpy in terms of RMSE and MPE are 0.965kJ/kg and 1.53%, respectively showing that the simulated values are in good agreement with the monitored results. The calibration and validation at the system and component level can be further improved considering the wide range of transient measurements of approximately six months (Chapter 07).
Afterwards, the validated model of desiccant cooling system is used for performance analysis of different desiccant cooling system configurations. The DEC system model is developed based on the second proposed modeling approach, Redeclaration/Replaceable component models to select an appropriate configuration. Moreover, empty component models are also incorporated in the overall system model to introduce multifunctional modeling concept of HVAC systems.

Five desiccant cooling system configuration including three standard and two modified configurations with respect to the operation cycles are evaluated in five different climate zones. The standard ventilation, recirculation, and dunkle cycles and modified ventilated recirculation and ventilated dunkle cycles are analyzed. The selected climate zones consist of Vienna, Karachi, Sao Paulo, Shanghai, and Adelaide. The cooling design day of each respective climate zone is considered and hourly ambient conditions are used from the weather data files of EnergyPlus software. It is concluded that the modified dunkle cycle configuration in general is suitable in all climate except Karachi and Shanghai climates that favor the ventilation cycle configuration. However, the additional heat exchanger in the dunkle increases the initial system cost. Therefore, estimated initial cost of each component is also provided for economic aspects. The current performance analysis is based on the design day data of each respective climate zone that can be further enhanced through yearly simulation using whole year climate data of each climate zone (Chapter 08).

Finally, the thesis research work established that the systematic simulation-based optimization approach proved quite efficient in terms of labor and time to handle the complex task of optimal selection of HVAC system configurations. Such development should help the design practitioners to evaluate various system alternatives and select the optimal under various building load demands and climate conditions at the initial design stage. Moreover, the methodology represents a step forward toward the design of software systems able to synthesize new and optimal system configurations.

9.2 Future work

In view of research work and conclusions, the following aspects provide the directions of future work for further enhancement of the proposed methodology.

The available HVAC system component models in Dymola/Modelica are not enough to evaluate all possible HVAC system configurations present in the market. More contribution is required for enhancing the existing Modelica component library. The additional development of component models, like various types of compressors, heat pumps, and absorption chillers could be valuable input. Moreover, generally the available component models also predict the ideal performance. Therefore, such models are also required to be modified to estimate the real system performance.
The limitation of second modeling approach need more research upgrading Modelica language so that the class of each component model could be changed automatically. Additionally, changing the component class also changes the parameter assigned values so that for each change the values are required to be defined again. Such language characteristic also hinders the possibility of automatic class change of each component. Therefore future advancements in Modelica language should address such aspects.

Optimization of chilled water system is based on the operating strategy that assume fixed chilled water supply and return temperature. However, these temperatures can be considered as optimization variables to determine the optimal values that would enhance the overall system optimization. Moreover, measurement uncertainties associated with the total power consumption and actual wet bulb temperature variations of actual climate can also be considered to further improve the resulted solutions. Additionally in future, yearly transient simulations of the chilled water system configurations can be helpful to decide an appropriate control strategy and system configuration for the whole year.

The transient data are obtained from ENERGYbase monitoring systems equipped with over 500 sensors for calibration and validation of desiccant cooling component and system models. The accuracy of results is strongly influenced by the sensor accurate measurements. Therefore, it is important to develop methods for measurement assessment in terms of measurement accuracy for such complex system in future.

The data used for calibration and validation of the desiccant cooling component and system models is based on the transient measurements of six selected day. The wide range of operating conditions in terms of yearly transient measurement can be considered for further enhancement of components and system models validity range and more accurate solutions.

The performance analysis of five desiccant evaporative cooling system configurations in five climate zones is achieved based on the hourly data of the design day of each respective climate. However, in future the performance study of each configuration considering whole year climate data through yearly simulations would provide more critical investigations.

In the current work, only initial costs of chilled water and desiccant cooling system configurations are provided. However, further cost investigation considering the life cycle cost of each system configuration could be worthy for overall economic aspects.

The proposed multifunctional system modelling approach needs to be further investigated in future. For example, the performance analysis of desiccant evaporative cooling with or without either supply of return humidifiers.
Finally, the developed methodology should be transformed into a software program for the evaluation and selection of various HVAC system configurations. The software program should consider the building load demand and climate conditions as model inputs. An open component models library need to be developed consist of all available HVAC component and control models. Moreover, initial, operating and maintenance costs of each component model can be included as model parameter to provide detailed cost analysis of each system configuration.
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