UNIVERSITEIT VAN PRETORIA
UNIVERSITY OF PRETORIA
YUNIBESITHI YA PRETORIA

OPTIMAL CONTROL OF A HEATING VENTILATION AND AIR-CONDITIONING SYSTEM WITH ICE-STORAGE VESSEL IN A COMMERCIAL BUILDING

by

Juan-Dirk Voigt

Submitted in partial fulfilment of the requirements for the degree

Master of Engineering (Electrical Engineering)

in the

Department of Electrical, Electronic and Computer Engineering
Faculty of Engineering, Built Environment and Information Technology

UNIVERSITY OF PRETORIA

December 2015

SUMMARY

OPTIMAL CONTROL OF A HEATING, VENTILATION AND AIR-CONDITIONING SYSTEM WITH ICE-STORAGE VESSEL IN A COMMERCIAL BUILDING

by

Juan-Dirk Voigt

Supervisor:	Prof X Xia
Co-Supervisor:	Prof Q Cheng
Department:	Electrical, Electronic and Computer Engineering
University:	University of Pretoria
Degree:	Master of Engineering (Electric Engineering)
Keywords:	Ice Storage, Optimal Control, Thermal Energy Storage, Commercial Buildings, HVAC

Current research indicates that optimal control methods can be used to control heating ventilation and air-conditioning (HVAC) systems with ice-storage tanks installed in commercial buildings that employ a time-of-use billing structure. An optimal control model for the cooling cycle of the HVAC system with an ice-storage tank for a commercial building is proposed. This nonlinear optimisation problem can be solved by existing software packages. The results are compared with the existing conventional control strategy employed on the HVAC system under consideration. It is shown that the optimal control of the ice storage can realise a greater energy and demand cost saving compared to the conventional control strategy for different billing periods. The research will present a comparative benchmark for other thermal energy storage systems installed in commercial buildings.

UNIVERSITEIT VAN PRETORIA
UNIVERSITY OF PRETORIA
YUNIBESITHI YA PRETORIA

OPSOMMING

OPTIMALE BEHEER VAN ’N VERHITTING-, VENTILASIE- EN LUGVERSORGINGSTELSEL MET ’N YSSTOORHOUER IN ’N KOMMERSIËLE GEBOU

deur

JUAN-DIRK VOIGT

Studieleier:	Prof X Xia
Mede-Studieleier:	Prof Q Cheng
Departement:	Elektriese, Elektroniese en Rekenaaringenieurswese
Universiteit:	Universiteit van Pretoria
Graad:	Magister in Ingenieurswese (Elektriese Ingenieurswese)

Sleutelwoorde: Ysstoor, Optimale Beheer, Termiese Energiestoor, Kommersiële Gebou

Huidige navorsing wys dat optimale beheermetodes aangewend kan word vir die beheer van ’n verhitting-, ventilasie- en lugversorgingstelsel met ysstoor, wat geïnstalleer is in ’n kommersiële gebou en waar die energieverbruik gemeet word volgens ’n tyd-van-verbruikelektrisiteittariefstruktuur. ’n Optimale beheermodel vir die verkoelingslassiklus van die lugversorgingstelsel met ysstoor in ’n kommersiële gebou word voorgestel. Die nie-liniêre optimaliseringprobleem kan deur middel van bestaande sagteware opgelos word. Die resultate van die model word vergelyk met die huidige konvensionele beheerstrategie wat toegepas word op die geëvalueerde lugversorgingstelsel. Daar word aangetoon dat optimale beheer groter aanvraag- en energiekostebesparing sal bewerkstellig in vergelyking met die konvensionele beheerstelsel vir die verskillende faktuur periodes. Die navorsing sal ’n vergelykende maatstaf bied vir soortgelyke stelsels met ysstore wat geïnstalleer is in kommersiële geboue en die verwagte kostebesparings.

LIST OF ABBREVIATIONS

COP	Coefficient of performance
CTMM	City of Tshwane Metropolitan Municipality
DOE	Department of Energy
DSM	Demand side management
EER	Energy efficiency ratio
FCU	Fan coil units
GBCSA	Green Building Council of South Africa
HVAC	Heating, ventilation and air-conditioning
HWC	Hot water cylinder
IDM	Integrated demand management
IP	Integer programming
kWh	Kilowatt hour, a billing unit for electrical energy
MPC	Model predictive control
MYPD	Multi-year price determination
TES	Thermal energy storage
TOU	Time-of-use
SANRAL	South African National Roads Agency Limited
SANS	South African National Standard

LIST OF FIGURES

LIST OF TABLES

TABLE OF CONTENTS

CHAPTER 1 INTRODUCTION

1.1 PROBLEM STATEMENT

1.1.1 Context of the Problem

South Africa is a developing country where the national electricity utility, Eskom, supplies 94% of the country's electrical energy, with generation from local municipalities and industry contributing to the balance [1].

In January 2008, South Africa experienced its first load shedding as an emergency measure to control energy demand and ensure that energy blackouts are averted. Load shedding was necessitated by the generation versus demand reserve margin hitting a record low of 5.6% at the start of 2008 [2].

The load shedding of 2008 was the start of a new era for South Africa’s electrical energy sector and economy as a whole. The main reason for the required load shedding is that the “. . . rise in electricity demand over the last two years has outstripped the new capacity we have brought on stream. The resultant tight supply situation makes the overall system vulnerable to any incident affecting the availability of energy. In this situation, we have to curtail the unplanned outages and the only way we can do this immediately is reduce demand and thus ensure a better reserve margin” [3].

The new era of the South African electricity industry is characterised on the consumer side by greater awareness of electricity usage and on the utility side by a major drive to ensure that the load shedding of 2008 is not repeated.

It is shown in Figure 1.1 that since the load shedding of 2008 electricity price increases have been extensive [4]. There were two main reasons for the sharp increase in electricity prices. The first was that Eskom had to embark on a major programme to bring new generation online, through construction of new power stations and rejuvenating old mothballed power stations. The second was that wasteful usage of electrical energy by consumers needed to be curbed.

Consumers are guided to introduce energy and demand-saving measures such that high electrical energy costs and future increases are mitigated.

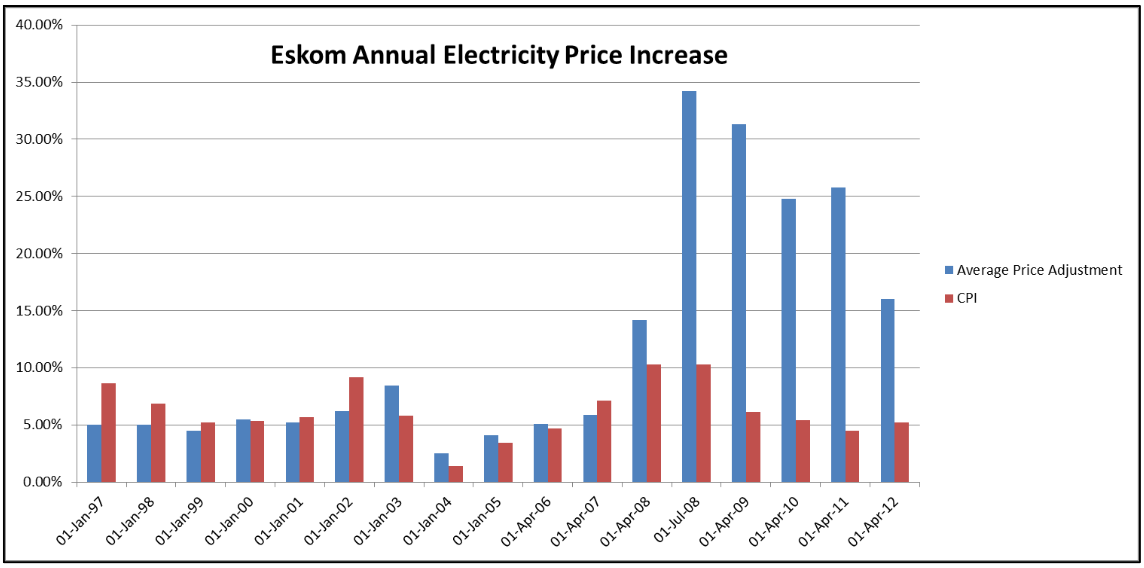


Figure 1.1. Eskom's historic annual electricity price increase

It is expected that South Africa's reserve margin will remain constrained unless energy users lower their consumption. As indicated in Figure 1.2, commercial buildings consume approximately 7% of the energy generated [5]. The commercial building sector can therefore make a significant contribution to reducing South Africa's electrical energy demand.

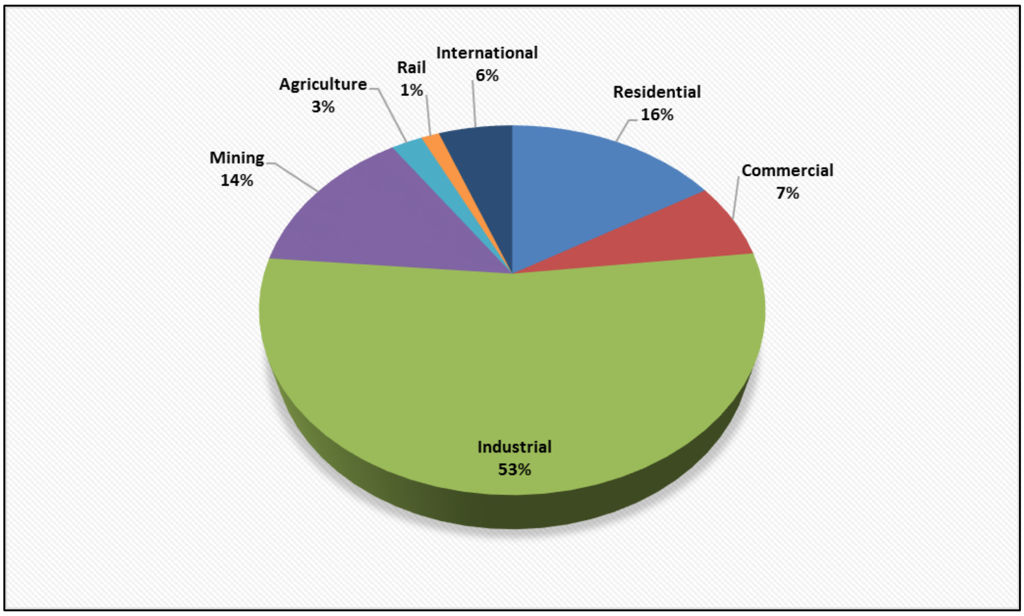


Figure 1.2. South African electricity consumption 2014

The South African government and more specifically the Department of Energy (DOE) has facilitated the introduction of a national standard for energy-efficient buildings, referred to as SANS 204. The DOE is aiming to achieve a total reduction of energy use in the commercial building sector of 15% by 2015, compared to the base year of 2010 [6].

The Green Building Council of South Africa (GBCSA) was established in 2007. As a full member of the World Green Building Council, the GBCSA is a non-profit company with the aim "… to promote, encourage and facilitate green building in the South African property and construction industry through market-based solutions focusing on; advocacy and promotion, rating tools, education and training and resources" [7].

Developers and owners of buildings are therefore motivated to construct buildings that are more energy-efficient. Energy-efficient or "green" buildings will be constructed at a premium initial capital cost, but will aim to have lower cost of ownership over the full life cycle of the building. It is therefore imperative that all energy-efficiency and demand-reducing equipment is functioning as closely as possible to optimal to ensure a maximum cost saving over the life of the building or shorter pay-back on the initial investment [7].

One such technology currently included in "green" buildings is the thermal energy storage (TES) system. In the case of the South African National Roads Agency Limited's (SANRAL) new corporate head office, design engineers selected an ice-storage tank as TES medium. The ice storage is used to shift the cooling load of the building from the peak daytime periods when energy and demand-related costs are high, to the off-peak evening periods when the costs are reduced [8].

The local electrical supply authority for Pretoria is the City of Tshwane Metropolitan Municipality (CTMM). The electrical supply to the SANRAL building is metered and billed on a time-of-use (TOU) energy tariff with demand charges. It is therefore imperative for the building owner to ensure that the least amount of energy is used in the high cost peak day time periods and maximum demand is restricted.

It is for these reasons that SANRAL decided to invest in an energyefficient building that will achieve a green building rating of four stars. This decision ensures that SANRAL is viewed as a company that is sensitive to its impact on the environment and in addition will save energy costs over the life of the building [9].

1.1.2 Research Gap

The current TES system installed in the SANRAL building relies on pre-specified, rule based conventional control strategies for each season. The conventional control strategies aim to reduce electricity costs by limiting the maximum demand and using cheaper off-peak energy tariffs.

In the summer months the control strategy employed does not maximize the potential energy cost saving that can be realised by the TES. The priority of the control strategy is to firstly utilize the chillers to cool the building up to a pre-specified discharge level, where after any additional energy requirement is met by the TES. The limitation of the conventional control strategy is that the potential energy usage, and thus the potential cost saving of the cooling cycle of the HVAC system, of the TES is not maximize by ensuring that the usage of the TES has priority during the peak time-of-use periods.

Due to the limitations of the conventional control strategies, optimal control theory was chosen as the preferred method of optimisation of the SANRAL HVAC system model. Optimal control reduces processing time and simplifies the optimization of very complex, linear or non-linear equations, thus allowing for easier analysis and implementation.

This research aims to investigate to what extent the conventional control strategy is minimising the electricity account of the building. Through rigorous modelling and optimisation of the heating, ventilation and air-conditioning (HVAC) system, the conventional control strategy can be compared to the cost minimal optimal control model results.

1.2 RESEARCH OBJECTIVE AND QUESTIONS

The research question is determining to what extent the existing conventional control strategy (chiller assist) employed on the HVAC system with ice-storage vessel installed in the SANRAL building is minimising the energy and demand cost. Research has shown that optimal control models with a cost minimal objective, realise a greater total energy and demand saving compared to conventional control strategies [10]. Conventional control strategies can, however, realise a cost saving that is as close as 3% to the energy cost realised with an optimal control model, with other control strategies falling short by 30% [11, 12].

The research questions are:

- How close to optimal is the current conventional control strategy?
- What is the maximum percentage cost saving that can be realised through optimal control?
- Will the optimal control model improve or reduce the energy efficiency of the system?
- What effect will the optimal control strategy have on the load factor of the electrical supply to the chillers?

1.3 HYPOTHESIS AND APPROACH

When analysing the design philosophy and conventional control strategy currently employed on the HVAC system in the SANRAL building, it is clear that the full potential of the ice-storage vessel is not maximised in terms of energy and demand cost saving. With daily mean temperatures for Pretoria above 18 °C from September to April [13], the HVAC system will mostly be operating in cooling mode.

In the summer period the control system is set to prioritise the use of the chillers for the cooling load supply. The HVAC control system is programmed to supply the building cooling load with up to 50% of the chiller maximum rated capacity. The ice storage will assist the chillers to supply the building cooling load for energy requirement in excess of 50% of the rated chiller capacity [8].

The sizing (maximum rated supply) of the chillers and thus the 50% supply capacity was based on the maximum design day cooling load plus some safety margin. It is assumed that the coincidence of the actual cooling demand of the building reaching the design day demand will be very low and can therefore be considered the exception. The ice-storage tank, after assisting with the supply of the design day cooling load, will be fully charged at night. For days when the cooling load is below the design day parameters, the ice storage will be underutilised.

Optimal control of the HVAC system will ensure that the ice storage is used more effectively to realise a greater energy and demand cost saving for a billing period.

The research procedure will include the following:

- Formulating a graphic representation of the existing system and indicating key energy components (gathering of information).

- Ring-fencing the exact portion of the system to be evaluated and making assumptions, e.g. for what energy cycle the system will be simulated, over what period (24 hours, month, year) and what season.

- Obtaining the billing information and TOU cost structure for the applicable utility.
- Identifying the electrical energy-consuming equipment and defining the total power equation.
- Defining the new optimal control model, consisting of energy balance equations, constraints and variables for the SANRAL HVAC system.
- Using the existing building cooling load data to find the expected billing cost (energy + demand) for the conventional control strategies.
- Optimising the optimal control model with the load data and obtaining the minimal cost for the billing period.
- Comparing the optimal control model minimal cost with the conventional control cost for energy and demand over the same defined period.
- Analysing the results and making recommendations for future research.

1.4 RESEARCH GOAL

The goal of the research is to minimise the energy and demand cost for different billing periods through a cost minimal objective optimal control model and optimisation of the cooling cycle for the HVAC system.

1.5 RESEARCH CONTRIBUTION

Buildings are unique and HVAC systems are custom-designed for the specific application and load requirement. What makes buildings unique is their specific application and geographic location. The geographic location not only dictates the expected climate, but in terms of energy consumption and cost, the utility rate structure and tariff.

This research, through the results of the simulations and the comparisons, will provide a benchmark for expected improved energy and demand cost minimisation through optimal control for HVAC systems with similar building cooling load and climatic conditions.

1.6 OVERVIEW OF STUDY

This chapter gave an introduction of the research and described the broader context of the research, indicating the specific research gap that this study will address. In addition, the research goals, questions, hypothesis, methodology and research were presented.

Chapter 2 presents the literature study related to the major topics covered by the research. The major topics include demand side management (DSM) techniques, TES technologies, conventional control strategies employed on TES systems, optimal control applications in general, optimal control of TES systems and research into the operation of the HVAC system for the building under consideration. The chapter concludes with the motivation for optimal control of this specific HVAC system with TES.

Chapter 3 presents the rigorous mathematical modelling of the HVAC system components and the formulation of the optimisation problems. The objective function, decision variables, input variables, state variables and constraints are clearly identified.

Chapter 4 includes the details of coding the optimisation algorithm in Matlab and presents the optimal control solution. In Chapter 5 simulation of the HVAC system without TES, TES with convention control and the optimal control monthly billing cost results are presented. The results of the three strategies are compared and discussed in Chapter 6 and concluding remarks are included in Chapter 7.

CHAPTER 2 LITERATURE STUDY

2.1 CHAPTER OVERVIEW

The chapter presents the literature survey of the study, which includes DSM in the South African context, TES technologies, conventional control strategies, optimal control in general and optimal control of HVAC systems with TES.

2.2 DEMAND SIDE MANAGEMENT

Electrical utilities need to ensure that the capacity utilisation of their power system infrastructure, from generation to load, is optimised. The capacity utilisation can be expressed as the average energy consumption divided by the maximum consumption or demand over a predefined period, which is usually a billing period of a month. This ratio is known as the system load factor and utilities aim to achieve a load factor of unity through the application of DSM [14]. In South Africa DSM refers to peak demand reduction through different DSM techniques. Internationally DSM refers to a much broader category of demand control and manipulation.

DSM can be defined as “the planning and implementation of those electric utility activities designed to influence customer uses of electricity in ways that will produce desired changes in the utility’s load shape” [15]. Consumer daily load patterns are industry or costumer group specific and utilities apply different DSM techniques, depending on the type of load pattern.

2.2.1 Demand Side Management Techniques

Different DSM techniques or load-shape objectives are available. Three common load control methods are used simultaneously as DSM strategies in commercial buildings [15], namely:

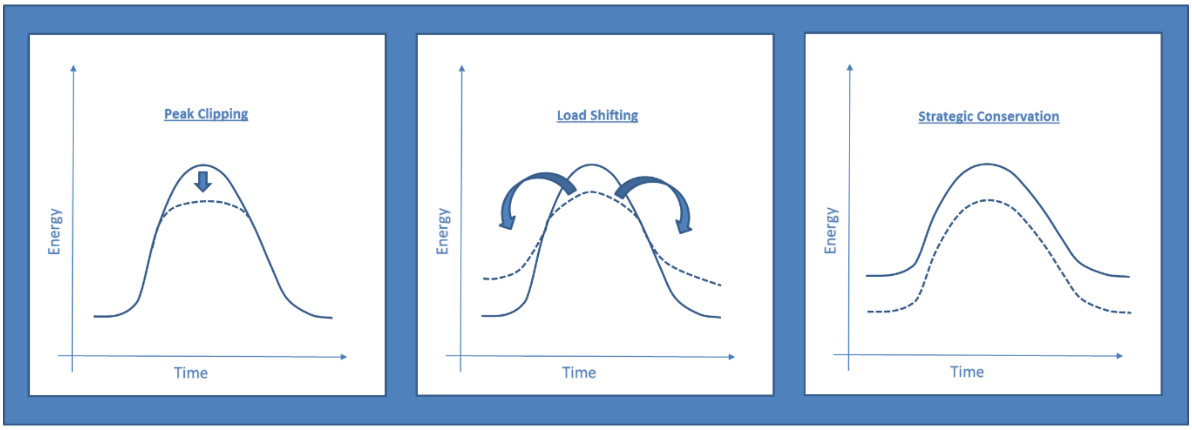


Figure 2.1. Demand side management techniques

2.2.1.1 Peak Clipping

This involves switching off loads in predetermined periods of high consumption for short instants to ensure that the costumer’s load does not exceed a targeted demand level. As indicated in Figure 2.1, for peak clipping the area between the solid and dashed line is reduced, indicating a decrease in total energy used in a specific time instant [15].

2.2.1.2 Load Shifting

The aim of TOU tariffs is to encourage consumers to use electricity in low-demand periods so that a smoother load curve is obtained, targeting a load factor closer to unity. Load shifting by definition therefore is not a reduction in energy consumption, but rather a shift in time of energy utilisation [15].

2.2.1.3 Energy Conservation

Conservation of energy entails the more efficient use of energy, leading to a total reduction or lowering of the demand curve, as illustrated in Figure 2.1 [15].

2.2.2 Demand Side Management in South Africa

Eskom has over the last couple of years invested heavily in DSM initiatives. Since 2005, Eskom has been able to realise an accumulated demand saving of 3587 MW, which is the equivalent of a single new coal-fired power station [16].

To realise this demand saving, different initiatives and incentives have been implemented through Eskom's Integrated Demand Management (IDM) division. One of the technologies highlighted by the IDM division is TES for building HVAC systems [17].

2.2.3 Local Utility Energy Tariffs

As the monopoly electricity generator, Eskom resells energy to the local supply authorities in South Africa. The building under consideration is located in the CTMM, which is the licensed electrical energy distributor [4, 18]. CTMM purchases electrical energy from Eskom at multiple in-feed substations and is billed on the Eskom Megaflex tariff for local authorities at all bulk metering points [4].

CTMM has multiple electrical tariff structures for different types of costumers. The SANRAL building has medium-voltage, 11 kV bulk electrical supply. In addition, the customer is able to shift load between different demand periods. The published electricity tariffs therefore allow the SANRAL building to be metered on an "11 kV Supply Scale: Time-of-Use" electricity tariff [19].

The defined daily TOU periods throughout the year, as published by CTMM, are according to the current Eskom Megaflex tariff and indicated in Table 2.1, which excludes the application of public holidays [19].

Table 2.1. Defined daily TOU periods

Period	Weekdays	Saturdays	Sundays
Peak	07:00 - 10:00 18:00 - 20:00	NONE	NONE

Standard	06:00 - 07:00 10:00 - 18:00 20:00 - 22:00	07:00 – 12:00 18:00 – 20:00	NONE
Off-peak	22:00 – 06:00	12:00 – 18:00 20:00 – 07:00	00:00 – 24:00

2.3 THERMAL ENERGY STORAGE TECHNOLOGY

The TES technology under consideration for this study is known as a latent cool storage system. This type of thermal storage technology makes use of the latent heat of fusion of a phase-change material, e.g. water to ice. Making use of the latent heat properties of a material allows for high energy densities and relatively compact storage systems [20].

Current distinct technology categories, within the latent cool storage group, include internal-melt ice-on-coil, external-melt ice-on-coil, encapsulated, ice harvester, ice slurry and unitary technologies. The categories of storage technologies can be subdivided into two main groups, referred to as direct and indirect ice production [21].

2.3.1 Direct Ice Production

The first group of technologies, which allows ice to form directly on the evaporator surface of the refrigeration machine or chillers, is known as direct ice production technologies [22].

2.3.1.1 Ice-on-coil, External Melt

Also known as directcontact cooling, ice-on-coil external melt technology allows coolant water to flow through the ice-storage tank, to and from the cooling load. The warm coolant water, returning from the cooling load, is in direct contact with the ice [23].

The ice-storage tank consists of tubes or pipes (coil) that are immersed in water; ice is formed on the outside of the tubes or pipes. The ice is made on the outside (externally) of the coil by circulating a secondary coolant at sub-zero temperatures through the inside of the coil [21].

2.3.1.2 Ice Slurry

The water particles in a water-glycol solution are frozen into slurry that can be pumped to a storage tank. Slurries can be pumped from the tank to heat exchangers or directly to cooling coils, resulting in high energy transport rates [24].

2.3.1.3 Ice Harvester

The ice harvester builds ice on a smooth cooling surface and then sheds layers of ice into a storage container. The ice build and thaw in the cooling surface is achieved by the cooling and heating cycles of a heat exchanger [21].

2.3.2 Indirect Ice Production

The second group, which consists of technologies that use a secondary fluid (brine solution) that can be circulated at sub-zero temperatures to produce ice on the surface of an external heat exchanger, is known as indirect ice production [22].

2.3.2.1 Ice-on-coil, Internal Melt

In contrast to the ice-on-coil, external melt technology, the internal melt type allows a glycol or brine coolant solution to flow through the ice-storage tank, to and from the cooling load. The warm coolant glycol or brine solution, returning from the cooling load, is in indirect contact with the ice [23].

The sub-zero secondary medium or refrigerant is circulated inside the fully water-immersed tubing or pipes, and is melted internally by circulating the same secondary coolant or refrigerant to the load [21].

2.3.2.2 Encapsulated Ice

This group of technologies consists of storage tanks that contain plastic containers filled with water or other phase- change material. The sub-zero glycol coolant is circulated through the storage tank, which freezes the water in the containers or thaws the ice in the containers when coolant is circulated to the cooling load [21].

2.4 CONVENTIONAL CONTROL OF THERMAL ENERGY STORAGE SYSTEMS

The control of HVAC systems with ice-storage tanks has been studied for many years, with conventional control strategies being proposed and defined. Early research has recognised the potential of control strategies to have an effect on the demand reduction of ice-storage systems [25].

2.4.1 Typical Modes of Operation of HVAC Systems with Ice Storage

TES systems operate in different modes, depending on the time of year (seasonal), time of day (day and night) and time of week (work day or weekend). The most common modes of operation used for cool storage applications are presented below.

2.4.1.1 Storage Charging Mode with No load

All the HVAC system cooling cycle equipment (e.g. valves, chillers and pumps) is configured or switched to restrict coolant flow through the ice-storage tank. The chillers will deliver coolant at sub-zero temperatures to allow for the freezing of the ice-storage medium (usually water) to undergo a phase change. By changing the phase, liquid to solid, the TES technology takes advantage of the latent heat properties of the medium, and thus ensures that the maximum amount of energy is stored in the available volume [21].

2.4.1.2 Storage Charging Mode while Meeting the Load

Some systems require that the HVAC system be able to charge the ice storage and provide coolant to the cooling load at the same time. This will necessitate a more complex system that has freeze protection in the cooling load cycle, compared to the storage-charging mode with no load [21].

2.4.1.3 Storage Discharge Mode

The chillers will be switched off in this mode and the secondary pumps will be used to cycle the cooling fluid through the ice-storage tank. The building cooling load requirement will thus be exclusively met by the discharging of the ice -storage [21].

2.4.1.4 Direct Chiller Mode with Storage Discharge

In this operational mode the cooling load is met by direct operation of the chillers (cooling plant) and discharging of the ice storage. When considering cool storage applications, three common conventional control strategies are used for this operational mode: chiller-priority, storage-priority and constant proportion control [21].

2.4.2 Conventional Control Strategies

2.4.2.1 Chiller Priority Control

Chiller priority control allows the cooling plant to supply the cooling load; thereafter the thermal storage will be used to supplement any additional cooling load required [21].

This is the simplest type of control strategy and is often used with an HVAC system where the cooling plant capacity is designed to supply only a portion of the design day cooling load, thus limiting the maximum demand of the chillers. The advantage is, however, that the cooling plant will operate at design set points, yielding high chiller efficiencies and a smooth demand curve. However, the disadvantage is the under-utilisation of the thermal storage to ensure maximum demand reduction [22].

2.4.2.2 Storage Priority Control

The control strategy aims to use as much of the stored TES in the peak periods as possible, to ensure the total depletion of the storage before the day period ends. The cooling plant will contribute to the cooling load requirement to avoid premature depletion of the ice storage [22].

Storage priority control can also be understood in terms of maximum discharge rate. The cooling load requirement is met by the thermal storage until the maximum discharge rate is achieved; thereafter the cooling plant will start to contribute to the cooling load. Control of the maximum discharge rate is essential to ensure that sufficient coolant supply to the cooling load is maintained and/or unexpected demand charges are not incurred [21]. In addition, it is expected that load forecasting may result in the optimisation of this control method [26].

2.4.2.3 Constant Proportion Control

To ensure that both the chillers and the ice storage contribute to meeting the load, the constant proportion control strategy allows both the chillers and the ice-storage to meet a portion of the cooling load under all conditions. The load can be divided equally or in response to changing load conditions [26].

This control strategy is easy to implement and allows for a greater demand reduction than the chiller priority control strategy [22].

2.4.3 Control Strategy of the SANRAL HVAC System

The conventional control strategy employed on the HVAC system with TES in the SANRAL building is divided into three separate seasonal control periods: summer, intermediate (spring and autumn) and winter. The modes of operation are predefined in terms of operational logic for certain periods of the day [8].

The electrical supply to the building is billed on a two-part electrical tariff structure. The two-part tariff consists of the sum of the total TOU energy cost and the maximum demand cost. The daily time instants of the TOU tariff structure are indicated in Table 2.1.

2.4.3.1 Winter Months – Control Logic

The winter months are May, June, July and August.

- **Ice Build**

  In the winter months ice is made at night from 22:00 to 06:00.

- **Ice Storage for Cooling, Chillers for Heating**

  When the building is under diverse load conditions in the winter months, the chillers are operated in heat pump mode and only provide heating to the building. The ice storage is used exclusively to provide for low cooling load requirements.

2.4.3.2 Spring/Autumn Months – Control Logic

The intermediate months are March, April, September and October.

- **Ice-build**

  In the intermediate months ice is made at night from 18:00 to 07:00 in the morning or until fully charged.

- **Chiller Cooling and Heating with Ice Storage Assistance**

  In the intermediate months one chiller is dedicated to the heating load requirement and one to the cooling load requirement. In the event that the chiller dedicated to the cooling load is insufficient to meet the cooling load, the ice storage will be used to supplement the chiller's capacity.

2.4.3.3 Summer Periods – Control Logic

The summer months are January, February, November and December.

The conventional control strategy used for the summer months is called chiller assist and can be explained as follows:

- **Ice-Build**

  In the summer months ice is made at night from 18:00 to 07:00 in the morning or until fully charged.

- **Melt Ice to Assist Chillers in Peak Demand**

  When the two chillers that are operating in parallel are unable to supply the building cooling load, the ice is melted to assist the chillers to meet the peak demand.

- **Chiller Cooling**

  For normal operation in the summer daytime periods, the chillers supply the building cooling load.

2.5 OPTIMAL CONTROL IN GENERAL

The use of optimal control has been studied for a wide variety of energy problems with the specific objective to minimise the monthly billing cost. It has been shown that the use of an optimal control tool for energy problems has resulted in lower and more precise billing cost reductions compared to standard optimisation techniques [27].

The optimal control of a single hot water cylinder (HWC) or geyser was studied. The objective of the study was to use optimal control theory to create an optimal control model to find the best switching time of the HWC under real life conditions. The study therefore considered not only billing cost minimisation but also comfort levels and losses when the HWC was idle. Applying the control model to a case study indicated that the optimal control method provided a better comfort level and lower electricity cost compared to a method without optimal control [27].

Optimal control theory has also been studied as closed-loop feedback control of a system that is billed on both TOU and a maximum demand cost structure. The case study is a municipal water purification plant with the aim of minimising the billing cost through load shifting of the water-pumping scheme of the plant. Model predictive control (MPC) with integer programming (IP) was used to simulate the closed-loop system. The research concluded that the closed-loop MPC optimisation gave a total billing cost saving of 5.75%. It was noted that open-loop control optimisation provided similar TOU energy cost savings to the closed-loop control MPC optimisation; however, the open-loop integer programming optimisation realised greater maximum demand cost savings [28].

Optimal control theory is well suited to optimise industrial processes through minimising billed energy cost while ensuring specified constraints are met. The study on the optimal hoist scheduling and control programme for rock winders, found in deep-level mines, is one example. The research indicated that a near-optimal hoist schedule can be achieved while ensuring reduced energy cost on a TOU energy tariff, under unstable and unpredictable operating conditions. An MPC algorithm containing an adapted branch and bound methodology solver was applied to the optimisation model of the system. The research indicated that a possible reduction of 30% can be achieved on billed energy cost [29].

2.6 OPTIMAL CONTROL OF THERMAL ENERGY STORAGE SYSTEMS

Case studies have indicated that, to achieve a sustainable optimal operation of a TES system, robust control systems are required [30]. The objective of the TES control system should be to take advantage of daily variations in climatic and operating conditions. This will ensure that the equipment is utilised effectively and the capital investment is returned through maximum cost savings.

Various studies compared the effectiveness of convention control strategies with optimal control models for different types of TES systems. This section presents multiple optimal control studies with different objectives applied to various types of systems. The results obtained from the objectives when applying an optimal control strategy are compared with the results from the various conventional control strategies.

2.6.1 Energy or Demand Cost Optimisation

Optimal control models for minimising either the energy or demand costs were studied and the results were compared with conventional control strategies for an ice-on-pipe system. It was found that for the minimisation of the energy cost component, considering only the design day parameters, there is no difference in the energy cost saving when comparing the chiller-priority, storage-priority and optimal control. This is true for the demand cost simulation as well [10].

For off-design days where the cooling loads are lower than the design day maximum, it was found that the different control strategies achieve different energy/demand cost savings, depending on the ratio of peak and off-peak TOU energy charges. With the peak/off-peak energy charge ratio greater than 1.4, load-limiting control was found to be near optimal for both energy and demand costs [10].

2.6.2 Rule-based Control

Multiple HVAC systems with internal-melt ice-storage tanks were investigated and daily and monthly optimisations were solved for these systems. A near-optimal, rule-based controller was developed through the results that were obtained from solving the different systems. It was found that rule-based control strategies can on average be within 3% of the optimal costs [31].

2.6.3 Minimising Energy and Demand Cost

Operating cost minimisation, considering both energy and demand costs over a billing period, was realised through a simulation environment that determines the optimal control strategy through dynamic programming. The simulation tool comprises a modular cooling plant that consists of different chiller and ice-storage technology types, to compare the performance of the conventional control strategies and optimal control [32].

The research also indicated that when the cost incentive to shift load is high and the energy penalty for making ice is low, deploying conventional control strategies to the HVAC system with ice storage will provide cost savings of up to 20%. However, as soon as the utility incentive for load shifting is reduced or the use of the ice storage is associated with a significant increase in total energy consumption, only optimal control will be able to yield energy cost savings [32].

2.6.4 Predictive Optimal Control

The potential of predictive optimal control for TES systems was investigated to determine the effectiveness of an optimal controller when there is uncertainty about future weather conditions and expected building cooling load variables. Various predictive models were analysed in terms of the performance in forecasting cooling load data, which included perfect prediction, a din model, a random walk model, a harmonic model and an autoregressive neural network model. The results indicated that the complex network model and the simple din model came very close to realising the cost savings of the perfect prediction model [33].

The research also included the simulation and comparison of the predictive optimal controller with conventional control strategies. It is important to note that the storage priority-control was indicated to be near-optimal for long planning windows and actually superior to optimal control for short planning windows. In the presence of complex (real-time pricing) rate structures that change hourly, predictive optimal control is superior to conventional control strategies. The research confirmed that knowledge of the utility electricity tariff structures is essential to minimise the energy and demand cost of the HVAC system with TES [33].

2.6.5 Comfort-based Control

Research has also focused on alternative control strategies where the main objective is not to meet the building cooling load, but rather to use occupancy comfort levels as the constraint. The constraint for comfort is taken as the predictive mean vote to define thermal neutrality between predefined temperature values. The optimal cooling load is obtained by varying room temperatures and humidity levels, which ensures minimising of the energy cost [34].

2.6.6 Optimal Systems and Strategies

Optimal design of the thermal storage technology on a system level has been studied. A design tool has been developed for the optimal selection of the ice-storage tank, considering charging/discharging rates and sizing. The study includes the evaluation of a real system and its impact on the monthly cost and energy consumption of the selected building. The input data are obtained through a building simulation model. It was concluded that the ice storage can have a high impact on energy consumption, demand and energy cost with an expected total cost saving of 28% for the building evaluated [35].

2.7 LIMITATIONS OF PREVIOUS STUDIES

None of the previous studies compared an optimal control strategy with conventional control strategies for a HVAC system with ice storage, installed in a commercial building in South Africa. As concluded by previous studies, the utility billing rates are crucial to establish what the potential cost saving can be when comparing different control strategies. Utility rate structures are unique and the possible cost savings that can be realised by the optimal control of the HVAC system installed in the SANRAL building needs to be investigated.

2.8 MOTIVATION FOR OPTIMAL CONTROL

Optimal control theory was chosen as the preferred method of optimisation of the SANRAL HVAC system model for the following reasons, when compared with “classical” control methods presented in previous studies:

- Optimal control models can easily be extended for multiple input and output systems.
- Because of the iterative process of classic control techniques a definite optimal solution cannot be obtained, but only a sub-optimal solution through trial and error.
- Optimal control allows for the evaluation and solution of linear and non-linear equations, including linear and non-linear constraints.
- Optimal control reduces processing time and simplifies the optimisation of very complex problems, thus allowing for easier implementation.

Evaluating the original design philosophy and system parameters of the HVAC system installed at the SANRAL corporate head office, the motivations below in support of optimal control in realising a greater total cost saving compared to current control strategy were found.

2.8.1 Original Design Philosophy

From Figure 2.2 it is clear that the expected daily temperatures are well below the design day temperature. With the current control strategy the chillers will always operate in the daytime and the ice storage will only assist the chillers when they have reached their design limits. For most days the ice storage will only use a fraction of the off-peak energy that is used at night to make the ice at a lower TOU tariff and the chillers will operate at the most expensive time of the day.

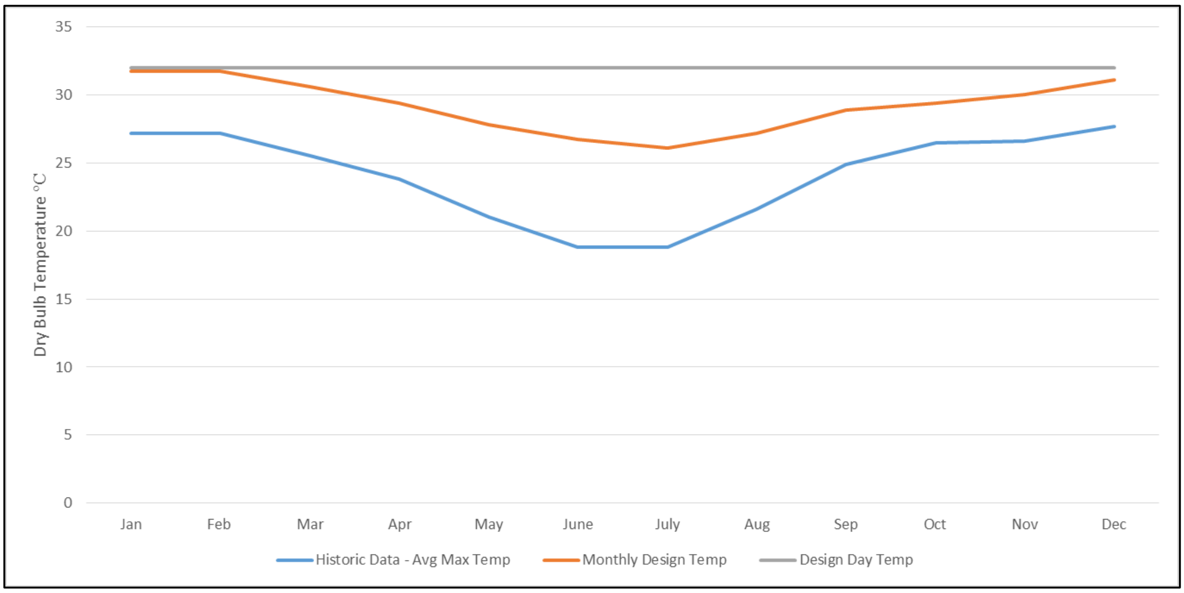


Figure 2.2. HVAC system design - temperature comparison chart

2.8.2 Maximum Storage Capacity

As included in Addendum A, the simulated design day values indicate that the total ice capacity discharged that is required to meet the balance of the building cooling load is 1324 kW.

The maximum energy storage capacity of the ice tank is 1693 kW. The surplus capacity that will not be used even at maximum design day cooling load is 369 kW.

2.8.3 Chiller Capacity Utilisation

The chillers are controlled not to operate above 50% of rated design capacity. Removing this constraint imposed on the chillers will allow for greater energy consumption in lower cost periods to build ice for use in high-cost TOU periods.

CHAPTER 3 MODELLING OF HVAC SYSTEM WITH ICE STORE

3.1 CHAPTER OVERVIEW

Chapter 3 presents the mathematic model of the HVAC system to derive the objective function, variables and constraints of the HVAC system.

3.2 THE SANRAL HVAC SYSTEM WITH TES

The HVAC system of the SANRAL building is a four-pipe fan coil unit system, with a central parallel heating and cooling system, consisting of two chillers and a single ice-storage tank. Figure 3.1. shows a schematic representation of the HVAC system.

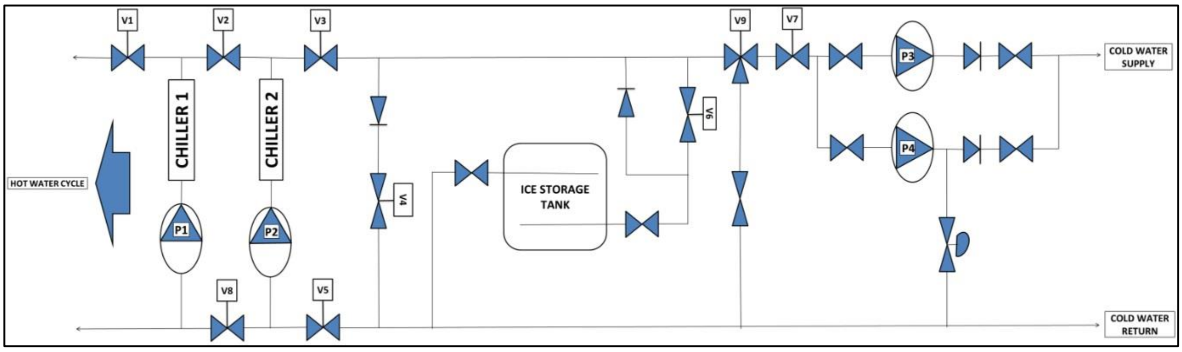


Figure 3.1. Schematic representation of the SANRAL HVAC system with TES

3.2.1 Current/Original Design Philosophy

The single largest contributor to the SANRAL building peak demand is the building's HVAC system. The total calculated design day chiller cooling load for a peak design day is 2382.9 kWh. A design day consists of 11 hours of cooling load from 07:00 to 18:00. The maximum chiller cooling capacity for the peak design day hour is 238.3 kW. The cooling load data were generated by Carrier's hourly analysis program (HAP) software package [36].

The hourly design day data are included in Addendum A. The table as presented in Addendum A, presents the hourly temperatures for the average hottest day in South Africa, which is the 29 December. From the historical temperature data the HAP Carrier program calculates the expected building cooling load for each hourly interval. The hourly energy data is then used to specify the required chiller capacity to meet the building cooling load and for this specific case the sizing of the ice storage vessel.

The system is designed to supply either chilled water to the cooling load at 6 ˚C or sub-zero coolant at -6 ˚C to produce ice in the TES tank. The HVAC system under consideration was not designed to allow for the charging of the ice storage and supply to the cooling load simultaneously [8].

In the chilled water mode (supply coolant at 6 ˚C) the maximum cooling capacity of a single chiller is 96.3 kW and with two chillers in parallel this allows for a maximum cooling capacity in chilled-water mode of 192.6 kW. In ice-build mode the chillers operated at a reduced capacity of 71 kW to produce coolant at -6 ˚C for a combined chiller capacity of 142 kW. The chiller data sheets are included in Addendum B and Addendum C respectively.

The design philosophy for the HVAC system with ice-storage tank at the SANRAL building was to flatten the demand curve by load shifting. The very simple control action that was applied to the system was to limit the chiller cooling output (capacity) to less than 50% of the rated chiller capacity. In chilled-water mode, the maximum capacity of both chillers operating together is 96.3 kW and in ice-build mode 71 kW.

The design day parameters that were used for the sizing of the ice-storage tank are included in Addendum D. The Cristopia STL-AC.00–28 ice-storage tank that was installed in the building has a maximum storage capacity of 1693 kWh.

The chillers are operated as primary cooling energy source for every day of the summer period until it reaches the 50% capacity limit of 96.3 kW. When additional cooling capacity is required the ice storage assists to supply the additional energy requirement.

3.2.2 Ice-Storage Vessel

The ice-storage vessel installed in the SANRAL building is a Cristopia STL.-AC.00-28 model. As stated, the ice-storage vessel is able to store 1693 kWh of energy and has a volume of 28 m^3. The tank consists of water-filled balls with a 28% glycol mixture passing through the balls acting as secondary coolant fluid.

3.2.3 Chillers

The HVAC system installed in the SANRAL building consists of two AQUACIAT2 400V chillers operating in parallel configuration (Chiller 1 and Chiller 2), as indicated in Figure 3.1. Both chillers are heat pump type chillers manufactured by CIAT, able to provide hot as well as cold water.

The chillers consist of a screw-type compressor and are able to sustain a coolant flow rate of 4.9 l/s. When the chiller is operated in normal cooling mode, not making ice but directly cooling the building, it has a direct capacity of 192.6 kW to supply coolant at an output temperature of 6 °C. The chiller specification and parameters for chilled-water operation are included in Addendum B.

When the chiller is used to build ice, it has a charge capacity of 142 kW to supply coolant at -6 °C. The chiller specification and parameters for the ice-making operation are included in Addendum C.

3.2.4 Pumps

The HVAC system consists of two primary pumps, one per chiller unit, and a secondary pump with an additional secondary (bypass) pump for maintenance purposes. The operation of the pumps is dependent on the specific strategy that is selected at the time. For the ice-build strategy only the two primary pumps operate. The combined power of the pumps is 3.5 kW.

The chiller-assist strategy requires both the primary and secondary pumps to run. The secondary pump is rated at 3.5 kW.

3.2.5 Air-handling Units

The fan coil units (FCU) are located in each office and are activated by an individual motion sensor per office. The combined average energy consumption of the FCU motors is 18.42 kW per operational hour.

3.3 HVAC SYSTEM MODEL PARAMETERS

The mathematic model presented will only include the chilled-water (cooling load) cycle of the HVAC system for each month of the year.

3.3.1 Chiller Units

The chillers and ice storage in combination can be operated in four distinct modes for the cooling cycle of the HVAC system for the building considered.

Table 3.1. HVAC cooling cycle modes

Mode	Cooling Load	Energy Source	Ice Store State	Chiller Supply Temp
1.	Building	Chillers Only	Passive	6 ˚C
2.	Building	Chillers + Ice Store	Discharging	6 ˚C
3.	Building	Ice Store Only	Discharging	6 ˚C
4.	Ice Store	Chillers	Charging	-6 ˚C

The chillers will not always operate at design capacity and performance will vary with the load and ambient conditions at different time instants. The proposed optimal model incorporates the part-load performance of the chillers, because of variation in cooling load and the coefficient of performance (COP) due to changes in the ambient conditions.

The combined power consumption of the two chiller units is expressed in (3-1).

$$P_k = P_k^{C1} + P_k^{C2} \tag{3-1}$$

where:

P_k: The total electrical power input to the chillers at time instant *k*;

P_k^{C1}: The electrical power input to chiller 1 at time instant *k*;

P_k^{C2}: The electrical power input to chiller 2 at time instant *k*.

To simplify the simulation model, it is assumed that the chilled-water flow will be equally divided through both chiller paths at the same flow rate. The two chillers are therefore modelled as a single chiller with total power represented by P_k.

The total electrical power input to the chillers is presented in (3-2).

$$P_k = \frac{Q_k}{COP_k} \tag{3-2}$$

where:

Q_k: The total thermal cooling output requirement to be supplied by the chillers;

COP_k: COP, considering both ambient conditions and part-load performance of the chillers.

The COP is described as the ratio of the rate of removal of heat to the rate of energy input to the chiller [37].

The energy balance equation presented in (3-3) indicates that the total cooling output on the chillers is the sum of the building cooling load Q_k^L and the charging/discharging rate u_k of the ice-storage vessel [38].

$$Q_k = Q_k^L + u_k \tag{3-3}$$

Table 3.2. Cooling modes and chiller load equation

Mode	Cooling Load	$Q_k =$	Ice-store State	Supply Temp
1.	Building	Q_k^L	Passive	6 ˚C
2.	Building	$Q_k^L + u_k$	Discharging	6 ˚C
3.	Ice Store	u_k	Charging	-6 ˚C

As described and assumed in [38], the variation of the actual chiller performance expressed as the COP, COP_k^{true} , owing to the changes in the environmental conditions and most notably ambient temperature, is proportional to the COP of the Carnot refrigeration cycle:

$$COP_k^{Carnot} = \frac{T_k^u}{T_k^u - T_k^l} \tag{3-4}$$

where,

COP_k^{Carnot}: COP of the Carnot refrigeration cycle at time instant *k*;

T_k^u: Upper process temperature of the chiller (condenser temperature) at time instant *k*;

T_k^l: Lower process temperature of the chiller (evaporator temperature) at time instant *k*.

The chillers under consideration will only be evaluated for the cooling load cycle and therefore the upper temperature limit will be the condensing temperature. The CIAT chillers installed in the SANRAL building are fitted with air-cooled condensers [39]. The upper temperature is assumed to be the dry-bulb temperature plus a condenser approach [38]:

$$T_k^u = T_k^{DB} + \Delta T^{air} \tag{3-5}$$

where,

T_k^{DB}: Dry-bulb outdoor air temperature at time instant *k*;

ΔT^{air}: Condenser approach, which is constant for the chillers under consideration.

The lower air temperature is the evaporator temperature, which is constant when the chillers are operated in either ice-making or chilled-water modes [38]:

$$T_k^l = \begin{cases} T^{evap,ice}, \text{if } u_k > 0 \text{ (charging)} \\ T^{evap,chw}, \text{if } u_k \leq 0 \text{ (discharging)} \end{cases} \tag{3-6}$$

Based on the assumption presented in (3-4), the actual COP (COP_k^{true}) at every time instant k is proportional to the COP of the Carnot refrigeration cycle:

$$COP_k^{true} \propto COP^{Carnot} \tag{3-7}$$

Therefore, an estimate of the actual $COP_k^{true,2}$ at any upper process temperature ($T_k^{u,2}$) and lower process temperature ($T_k^{l,2}$), denoted by $COP_k^{true,2\prime}$ can be obtained from the actual COP ($COP^{true,1}$) as provided by the chiller manufacturer for operating temperatures $T^{u,1}$ and $T^{l,1}$ as per:

$$COP_k^{true,2\prime} = \frac{COP_k^{Carnot,2}}{COP^{Carnot,1}} COP^{true,1} \tag{3-8}$$

Thus,

$$COP_k^{true,2\prime} = \frac{\frac{T_k^{u,2}}{T_k^{u,2} - T_k^{l,2}}}{\frac{T^{u,1}}{T^{u,1} - T^{l,1}}} COP^{true,1} \tag{3-9}$$

The validity of the above assumptions has been tested and it was found that $COP_k^{true,2\prime}$ differs from $COP_k^{true,2}$ within a range of ±10% for a hermetical reciprocating liquid chiller with air-cooled condenser as per the CIAT chillers installed in the SANRAL building [38, 39].

The part-load performance of the chillers is also incorporated into the optimal model equations to express the energy penalty at varying cooling load conditions. The part-load ratio is expressed as:

$$PLR_k = \frac{Q_k}{CCAP_k} \tag{3-10}$$

where:

PLR_k: Part-load ratio at time instant k;

Q_k: Total thermal cooling output requirement to be supplied by the chillers at time instant k;

$CCAP_k$: Full-load capacity of the chiller at time instant k.

The full-load chiller capacity at time instant k is adjusted for the outside ambient dry bulb air temperature, as presented in the linear equation (3-11).

$$CCAP_k = CCAP^n\{1 + \delta[T^{ref} - T_k^{DB}]\} \tag{3-11}$$

where:

$CCAP^n$: Nominal/rated chiller capacity at reference temperature T^{ref};

δ: Slope of the linear equation;

T^{ref}: Reference temperature for the nominal/rated chiller capacity.

T_k^{DB}: Dry-bulb outside air temperature at every instant k.

The nominal chiller capacity is as per the manufacturer's product specification sheets included in Addendum B and Addendum C, where:

$$CCAP^n = \begin{cases} 142\ kW, if\ u_k > 0\ (charging) \\ 192.6\ kW, if\ u_k \leq 0\ (discharging) \end{cases} \tag{3-12}$$

and the corresponding reference temperature for the nominal chiller capacity is:

$$T^{ref} = \begin{cases} 25\ °\mathrm{C}\,,\ \text{if}\ u_k > 0\ (\text{charging}) \\ 35\ °\mathrm{C}\,,\ \text{if}\ u_k \leq 0\ (\text{discharging}) \end{cases} \qquad (3\text{-}13)$$

The slope of the linear equation is $\delta = 0.005$ and T_k^{DB} is the outdoor air dry-bulb temperature in degrees Celsius at time instant k [33].

The chiller performance under part-load conditions was supplied by the chiller manufacturer, CIAT, and included in Addendum E. Only the part-load performance of the chiller when supplying chilled water at 6 °C was supplied. The data were plotted and the linear equation was obtained for the COP under different part-load conditions.

It should be noted that the chiller manufacturer, CIAT, provided the chiller performance data as per the Eurovent certification standards. The Eurovent certification refers to the energy efficiency ratio (EER) as the ratio of chiller cooling capacity (in kW) to the electrical input of the chiller (in kW) [40]. For the heating capacity of the chiller Eurovent refers to COP. In South Africa it is industry standard practice to refer to both the cooling and heating cycle in terms of COP, as a unitless ratio of the electrical energy input to the chiller divided by the thermal energy output.

The performance curves of various chillers have been simulated and studied. The performance curve of a chiller at a higher temperature differential (ΔT), have the same curve shape when compared with lower temperature differential performance curves, but at a lower efficiency or COP [41]. Since CIAT did not provide performance data of the chiller for producing coolant at sub-zero temperatures, the equation to obtain the COP when the chillers are making ice and supplying coolant at -6 °C was assumed to have the same gradient as the part-load performance when producing chilled water at 6 °C; however, at a reduced COP.

The manufacturer's data included in Addendum C were used as reference. The chiller COP is 2.64 when producing chilled water at -6 °C at a fully rated nominal capacity.

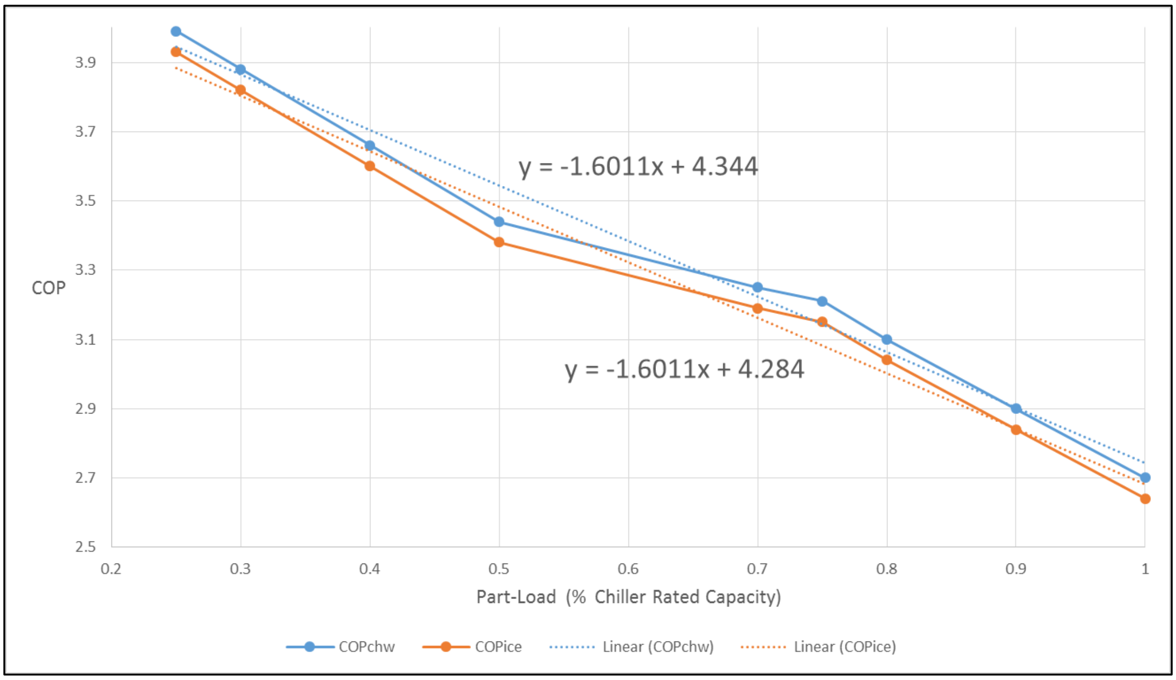


Figure 3.2. Chiller part-load performance vs. COP

Thus, the COP of the chillers under part-load conditions is expressed in (3-14).

$$COP_k^{PLR} = \begin{cases} -1.6011PLR_k + 4.284, \text{if } u_k > 0 \text{ (charging)} \\ -1.6011PLR_k + 4.344 \text{ , if } u_k \leq 0 \text{ (discharging)} \end{cases} \quad (3\text{-}14)$$

where

COP_k^{PLR}: Chiller COP under part-load conditions.

The COP considering both the ambient temperature conditions and the part-load performance of the chillers is presented in equation (3-16) when producing ice and (3-17) when producing chilled water. It is assumed that the actual COP for the chiller at every time instant *k* is the COP adjusted for part-load performance, thus

$$COP^{true,1} = COP_k^{PLR} \quad (3\text{-}15)$$

$$COP_k^{ice} = \frac{\frac{T_k^{u,2}}{T_k^{u,2}-T_k^{l,2}}}{\frac{T^{u,1}}{T^{u,1}-T^{l,1}}} COP_k^{PLR} \tag{3-16}$$

and,

$$COP_j^{chw} = \frac{\frac{T_j^{u,2}}{T_j^{u,2}-T_j^{l,2}}}{\frac{T^{u,1}}{T^{u,1}-T^{l,1}}} COP_j^{PLR} \tag{3-17}$$

The total electrical power consumption of the chillers from (3-2), considering both part-load performance and ambient conditions, is:

$$P_k = \begin{cases} \frac{Q_k}{COP_k^{ice}} & \text{if } u_k > 0 \text{ (charging)} \\ \frac{Q_k}{COP_k^{chw}} & \text{if } u_k \leq 0 \text{ (discharging)} \end{cases} \tag{3-18}$$

and from (3-3),

$$P_k = \begin{cases} \frac{Q_k^L + u_k}{COP_k^{ice}} & \text{if } u_k > 0 \text{ (charging)} \\ \frac{Q_k^L + u_k}{COP_k^{chw}} & \text{if } u_k \leq 0 \text{ (discharging)} \end{cases} \tag{3-19}$$

3.3.2 Ice-storage Vessel

The state of charge of the ice-storage tank is modelled as per formula (3-20) at each time instant *k* [38],

$$x_{k+1} = x_k + u_k \frac{1}{SCAP} \tag{3-20}$$

where:

x_k:	State-of-charge in hour *k*;
u_k:	Rate of charge or discharge;
SCAP:	Storage tank capacity [kW].

The state of charge of the ice-storage tank is subject to the per unit boundary constraints expressed in (3-21):

$$x^{min} \leq x_k \leq x^{max} \tag{3-21}$$

where $x^{min} = 0$ and $x^{max} = 1$.

3.4 OBJECTIVE FUNCTION

The objective of the optimal control model is to minimise the monthly billing cost consisting of the energy and demand cost over a predefined period for the energy consumption of the chillers, as presented in (3-22),

$$J = \sum_{k=1}^{N} P_k e_k + Max_{1 \leq k \leq N}(P_k d_k) \tag{3-22}$$

where,

J: Total billed cost for energy and demand over a billing period;

N: Total number of time instants in a chosen time period;

P_k: Electric power input (kW) to the chillers for each time instant *k;*

e_k: Cost per unit of electrical energy (kWh) at the instant *k*;

d_k: Maximum demand (MD) cost (R/kW) over the time period.

3.5 SYSTEM VARIABLES

3.5.1 Decision/Control Variables

The control variable is the rate of charging or discharging of the ice-storage tank represented by the vector ***u*** for *k = 0, …, N-1* and expressed in kW,

$$\boldsymbol{u} = [u_0, u_1, u_2, \ldots, u_k, \ldots, u_{N-1}]^T$$

The rate of charge/discharge is constrained as per (3-23):

$$u_k^{min} \leq u_k \leq u_k^{max} \tag{3-23}$$

The maximum discharging rate (ice melting) is represented by the negative value of u_k^{min} at time *k*, and the positive vale u_k^{max} is the maximum charge rate (ice build) at time *k* [38].

It is important to note that the SANRAL HVAC system with TES is not designed to allow for the simultaneous charging of the ice storage and provision of the building cooling load. The system can either build ice or supply the building cooling load at each time instant *k* [8].

To ensure that the optimisation model does not "overcharge" or "undercharge" the ice-storage tank, equations (3-24) and (3-25) represent the constraints on the maximum and minimum discharging rates respectively [38].

$$0 \leq u_k^{max} \leq SCAP \tag{3-24}$$

As indicated by (3-24), the maximum amount of thermal energy that can be extracted (ice build) by the chillers from the ice-storage tank should be less than the total storage capacity of the ice-storage tank ($SCAP$) at every time instant *k*. Because the rate of ice-storage charge is positive, u_k^{max} is always greater than zero and if no ice is produced the rate of charge is equal to zero.

$$-SCAP \leq u_k^{min} \leq 0 \tag{3-25}$$

Equation (3-25) indicates the constraint on the rate of discharging of the ice-storage tank. The maximum rate of discharge, u_k^{min}, is a negative value that should be greater than the negative value of the total storage capacity of the ice-storage tank ($SCAP$) at every time instant *k*. Because the rate of ice-storage discharge is negative, u_k^{min} is always less than zero and if no ice is produced the rate of charge is equal to zero.

3.5.2 State Variables

The system state variable is the amount of ice-store inventory at every time instant k.

The state of charge of the ice-storage tank is represented by the vector $\boldsymbol{x}$ and expressed as a per-unit value.

$$\boldsymbol{x} = [x_0, x_1, x_2, \dots, x_k, \dots, x_{N-1}]^T$$

It is assumed that the ice-storage tank is fully charged when the simulation is initiated, therefore $x_0 = 1$ and the vector $\boldsymbol{x}$ is thus constrained as per (3-26):

$$0 \leq x_k \leq 1 \tag{3-26}$$

3.5.3 Input Variables

The input variables of the system are presented and discussed under the following items:

3.5.3.1 Building Cooling Load

The building cooling load is derived from the data that were used for the design of the HVAC system, as obtained from the commercial design simulation software HAP Carrier for the building under consideration. The building cooling load will be expressed as the vector $\boldsymbol{Q^L}$ with unit kW, for every time instant k.

$$\boldsymbol{Q^L} = [Q_0^L, Q_1^L, Q_2^L, \dots, Q_k^L, \dots, Q_{N-1}^L]^T$$

3.5.3.2 Outdoor Dry-bulb Air Temperature

The dry-bulb air temperature outside the building is contained as part of the data set in the HAP Carrier software for the geographic location of the building under consideration.

The dry-bulb outside air temperature is expressed as the vector $\boldsymbol{T^{DB}}$, at every instant k.

$$\boldsymbol{T^{DB}} = [T_0^{DB}, T_1^{DB}, T_2^{DB}, \dots, T_k^{DB}, \dots, T_{N-1}^{DB}]^T$$

3.5.3.3 Energy Cost Rate

The energy cost rates as applied by the local electricity supply authority (CTMM) for the different TOU periods are expressed as the vector $\boldsymbol{e}$ with unit R/kWh, for every time instant k and for each 24-hour time period, where:

$$\boldsymbol{e} = \begin{cases} 0.7475, & k \in (7,8,9,18,19) \\ 0.4590, & k \in (6,10,11,12,13,14,15,16,17,20,21) \\ 0.3215, & k \in (22,23,0,1,2,3,4,5) \end{cases}$$

3.5.3.4 Demand Cost Rate

The maximum demand cost rate is expressed as the vector $\boldsymbol{d}$ with unit R/kW for specified 24 hour time instants, k:

$$\boldsymbol{d} = \begin{cases} 124.00, & k \in (6,7,8,9,10,11,12,13,14,15,16,17,18,19,20,21) \\ 0, & k \in (22,23,0,1,2,3,4,5) \end{cases}$$

3.6 TIME INSTANT

The building cooling load data produced by the HAP Carrier software for the building under consideration are for hourly instants.

The simulation horizon will be one month and the number of discrete time instants (N) will be dependent on the specific month considered for simulation.

3.7 SYSTEM CONSTRAINTS

3.7.1 Equality Constraints

The energy balance equation of the HVAC system with ice storage is presented in (3-3).

$$Q_k = Q_k^L + u_k$$

The maximum energy output of the chiller is the adjusted chiller capacity for each time instant.

$$Q_k \leq CCAP_k \tag{3-27}$$

From (3-3) and (3-27) the total energy requirement of the system is therefore constrained by (3-28).

$$0 \leq Q_k^L + u_k \leq CCAP_k \tag{3-28}$$

3.7.2 Inequality Constraints

The decision variable $\boldsymbol{u}$ and the state variable $\boldsymbol{x}$ are both unequally constrained.

3.7.2.1 Ice-storage Charge/Discharge Constraint

The decision variable $\boldsymbol{u}$ should be constrained to ensure that the correct amount of ice is either charged or discharged at each time instant to obtain the optimal ice-storage utilisation to minimise the objective function. From (3-23):

$$u_k^{min} \leq u_k \leq u_k^{max}$$

The maximum amount of ice that can be discharged from the ice-storage tank is represented by the negative value u_k^{min} at instant k, as presented in (3-25). The maximum charging rate at each instant k is represented by the positive value u_k^{max}, as presented in (3-24).

3.7.2.2 Ice-storage State of Charge Constraint

The state of charge variable $\boldsymbol{x}$ is constrained to ensure that the ice-storage tank is not overcharged or undercharged, as obtained from (3-20) and (3-21), and expressed in (3-29):

$$0 \leq x_k + u_k \frac{1}{SCAP} \leq 1 \tag{3-29}$$

3.8 SYSTEM CONSTANTS

The system constants are listed in Table 3.3 below.

Table 3.3. Simulation model constants

Constant	Value	Unit	Description
Δt	1	Hours	Time instant length
SCAP	1693	kWh	Ice-storage tank capacity
$CCAP^{n,chw}$	192.4	kW	Nominal chiller capacity for chilled water (6 °C)
$CCAP^{n,ice}$	142	kW	Nominal chiller capacity for ice build (-6 °C)
$T^{ref,chw}$	35	°C	Chiller reference outdoor condensing temperature for chilled water
$T^{ref,ice}$	25	°C	Chiller reference outdoor condensing temperature for ice build
ΔT_{air}	3	°C	Condenser approach
$T_{evap,ice}$	-6	°C	Evaporator temperature – ice mode
$T_{evap,chw}$	6	°C	Evaporator temperature – chilled water mode
N	Hours in Month	hours	Maximum time instants
δ	0.005		Slope of linear equation

3.9 ASSUMPTIONS

The following list summarises the assumptions of the mathematical model of the HVAC system:

- Only the cooling load cycle of the HVAC system is modelled.
- The two chillers are modelled as a single chiller.
- The initial state of the ice store is fully charged, therefore $x_0 = 1$. The optimal controller will discharge the ice storage at the end of each simulation period (month), to meet the objective to minimise the billing. This is known as the turnpike property.

Future research will have to include MPC control. Studies have indicated that MPC control will allow for the final state of the ice storage to converge to the initial state, independent from the initial state [42, 43].

- The ice-storage rate of charge/discharge is not dependent on the ice-storage state of charge.
- Studies have indicated that the efficiency of an encapsulated ice storage tank is greater than 98 % and even as high as 99.7 % [44-46] . System losses are thus assumed to be negligible.
- The length of a time instant for the study is 1 hour.
- The electrical energy of the secondary supply pumps and FCUs is not included in the simulation.
- The building cooling load and expected outdoor ambient dry-bulb temperature is theoretical and obtained from the HAP Carrier simulation software for the building under consideration.
- Each month of the year is calculated individually.
- The time instant for the MD calculation is according to the simulation data instants of a single hour.

CHAPTER 4 OPTIMISATION MODEL ALGORITHM

4.1 CHAPTER OVERVIEW

The nonlinear optimisation model, algorithm and results are presented in Chapter 4.

4.2 OPTIMISATION EQUATION

For the system under consideration, the energy balance equation (3-3) can only have three distinct forms, as indicated in Table 3.2. The HVAC system can only build ice when there is no building cooling load requirement as per Mode 1 in Table 4.1. Mode 2 indicates that the building cooling load is supplied by the chillers only and Mode 3 indicates the ice storage assisting the chillers in meeting the building cooling load requirement.

Table 4.1. Energy balance equation and operational scenarios

Mode	$Q_k =$	Q_k^L	u_k	**Ice Store State**
1.	u_k	$= 0$	> 0	Charging
2.	Q_k^L	> 0	$= 0$	Idle
3.	$Q_k^L + u_k$	> 0	< 0	Discharging

The optimisation models are simplified by introducing the binary variable M to distinguish between charging (ice build) and discharging (ice melt)/idle modes. In practice the optimal controller will be implemented as a discrete time controller.

The building cooling load Q_k^L is a known input variable to the system and Table 4.1 indicates that it is either equal to zero or greater than zero. The decision variable u_k is either greater than zero (charging) when Q_k^L is equal to zero or u_k is less than (discharging) or equal to zero when Q_k^L is greater than zero.

Thus the building cooling load, Q_k^L, is a known variable for each time instant that is used to set the binary variable M for each time instant. The system is either in ice-storage charge mode or ice-storage discharge mode. When the cooling load is greater than zero, the system equations will be set to represent the chillers producing chilled water at 6 °C and the binary variable *M* is set to be equal to 1. When the cooling load is zero, all equations will be set to represent the chillers producing chilled water at -6 °C to produce ice in the ice-storage tank and *M* will equal 0, as indicated (4-1):

$$M = \begin{cases} 0, \text{if } Q_k^L = 0 \text{ (charging)} \\ 1, \text{if } Q_k^L > 0 \text{ (discharging/idle)} \end{cases} \tag{4-1}$$

The following equations present the system in a summarised way to define the objective function equation from Chapter 3.

The total thermal energy produced by the chillers is expressed in (3-3):

$$Q_k = Q_k^L + u_k$$

The part-load ratio of the chiller is expressed in (4-2):

$$PLR_k = \frac{Q_k}{CCAP_k} \tag{4-2}$$

Combining (3-3) and (4-2) yields (4-3):

$$PLR_k = \frac{Q_k^L + u_k}{CCAP_k} \tag{4-3}$$

The chiller capacity at time instant *k* is presented in (3-11):

$$CCAP_k = CCAP^n\{1 + \delta[T^{ref} - T_k^{DB}]\}$$

with the nominal capacity of the chiller as defined in (3-12) and expressed in terms of Q_k^L:

$$CCAP^n = \begin{cases} 142\ kW, \text{if } Q_k^L = 0 \text{ (charging)} \\ 192.6\ kW, \text{if } Q_k^L > 0 \text{ (discharging)} \end{cases} \tag{4-4}$$

and the reference temperature is obtained from the manufacturer's data as indicated in (3-13) and expressed in terms of Q_k^L:

$$T^{ref} = \begin{cases} 25\ °\text{C}, \text{if } Q_k^L = 0 \text{ (charging)} \\ 35\ °\text{C}, \text{if } Q_k^L > 0 \text{ (discharging)} \end{cases} \tag{4-5}$$

Combining (3-11), (4-4) and (4-5):

$$CCAP_k = 142 \cdot \{1 + 0.005 \cdot [25 - T_k^{DB}]\} \cdot (1 - M) + 192 \cdot \{1 + 0.005 \cdot [35 - T_k^{DB}]\} \cdot M \tag{4-6}$$

and from (4-3) one obtains (4-7):

$$PLR_k = \frac{Q_k^L + u_k}{142 \cdot \{1 + 0.005 \cdot [25 - T_k^{DB}]\} \cdot (1 - M) + 192 \cdot \{1 + 0.005 \cdot [35 - T_k^{DB}]\} \cdot M} \tag{4-7}$$

The COP adjusted for part-load performance of the chillers as presented in (3-14) is rewritten as a single equation and is represented in (4-8):

$$COP_k^{PLR} = (-1.6011 PLR_k + 4.284) \cdot (1 - M) + (-1.6011 PLR_k + 4.344) \cdot M \tag{4-8}$$

The Carnot temperature ratio equation from (3-9) is simplified by introducing the temperature variable T' as presented in (4-9);

$$T' = \frac{\frac{T_k^{u,2}}{T_k^{u,2} - T_k^{l,2}}}{\frac{T^{u,1}}{T^{u,1} - T^{l,1}}} \tag{4-9}$$

When charging (ice build) the ice-storage tank, from (3-16) and (4-9), T'^{ice} is defined as:

$$T'^{ice} = \frac{\frac{T_k^{DB} + 3}{T_k^{DB} + 3 - (-6)}}{\frac{25}{25 - (-6)}} \tag{4-10}$$

For discharging of the ice-storage tank, from (3-17) and (4-9), T'^{chw} is defined as:

$$T'^{chw} = \frac{\frac{T_k^{DB} + 3}{T_k^{DB} + 3 - (6)}}{\frac{5}{35 - (6)}} \tag{4-11}$$

The COP considering both part-load performance and ambient temperature is presented in (4-12):

$$COP_k = T'^{ice}.COP_k^{PLR}.(1 - M) + T'^{chw}.COP_k^{PLR}.M \tag{4-12}$$

The input electrical power to the chillers is expressed in (4-13), and expanded in (4-14).

$$P_k = \frac{Q_k^L + u_k}{COP_k} \tag{4-13}$$

$$P_k = \frac{Q_k^L + u_k}{T'^{ice}.COP_k^{PLR}.(1 - M) + T'^{chw}.COP_k^{PLR}.M} \tag{4-14}$$

The objective function is thus, as presented in (3-22), expressed as:

$$J = \sum_{k=1}^{N} P_k e_k + Max_{1\leq k\leq N}(P_k d_k) \tag{4-15}$$

4.3 OPTIMISATION ALGORITHM

Because of the complexity of the optimisation problem, a modelling system was used to convert the optimisation equations to standard form. The selected optimisation tool is the Yalmip open source modelling system that uses various solvers to reduce complex optimisation problems to standard form [47].

The logical block flow diagram for the optimisation algorithm is included in Addendum L. The main sections of the algorithm are summarised in Figure 4.1. The M-File code for the optimisation algorithm is included in Addendum K.

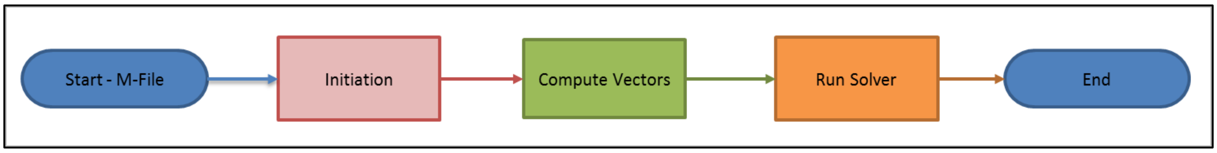


Figure 4.1. Main sections of optimisation algorithm

4.4 INITIATION OF ALGORITHM

4.4.1 Known Data/Input Variables

The input data to the algorithm are the simulated building cooling load data (Q_k^L) and the corresponding outside ambient dry-build air temperature (T_k^{DB}) for each corresponding time instant as obtained from the HAP Carrier simulation software for the building under consideration.

The TOU electrical energy cost (e_k) and the maximum demand cost (d_k) for the metered electrical energy tariff are also included in the Excel spreadsheet for each corresponding time instant.

The data for each month of the year are located in separate spreadsheets. All input data for each time instant is called into the Matlab M-File with the "*xlsread*" command from the Excel spreadsheet file.

4.4.2 Set Constants

The constants declared in the algorithm are listed in Table 3.3.

4.4.3 Declare Variables

4.4.3.1 State Variables

The state variables for the chiller capacity ($CCAP_k$), upper (T^u) and lower process (T^l) temperature of the chiller and the Carnot temperature ratio (T') for the performance of the chiller are declared as row vectors of zeros as initial values of length *N*.

4.4.3.2 Decision Variables

The main decision variable is the state of charge/discharge of the ice-storage tank, represented by vector $\boldsymbol{u}$. The state of charge/discharge is defined as a row vector using Yalmip *sdpvar*.

To obtain the maximum demand value for each month, a scalar *sdpvar* decision variable *lamda* is declared to represent the maximum demand cost after each time instant. The decision variable *lamda* is then used within a constraint function to limit the maximum value of the maximum demand for each time instant with *lamda* included in the objective function. This technique of minimising the maximum demand cost as part of the objective function by defining it as a variable within a constraint function is presented in [28].

4.4.3.3 Constraints

The constraint vectors are declared as empty matrixes initially and defined in Yalmip form, where:

- ***ct1*** – Defines the constraint on the ice-storage state of charge to ensure that the ice storage is not overcharged or undercharged. The constraint function is formulated from (3-29). Thus,

$$\boldsymbol{ct1} = \boldsymbol{ct1} + \left(0 \leq x_k + u_k \frac{1}{SCAP} \leq 1\right) \tag{4-16}$$

- ***ct2*** – Defines the maximum boundary constraints on the rate of charge ***u***. For time instants when no building cooling load exists, vector ***u*** can only be positive, indicating charging of the ice store as per (4-17). When building cooling load is present, the ice storage cannot be charged and ***u*** can thus only be a negative value, indicating ice-storage discharge as per (4-18).

$$\boldsymbol{ct2} = \boldsymbol{ct2} + (0 \leq u_k \leq SCAP) \tag{4-17}$$

$$\boldsymbol{ct2} = \boldsymbol{ct2} + (-SCAP \leq u_k \leq 0) \tag{4-18}$$

- ***ct3*** – Defines the energy balance equation for each time instant. The load on the chiller is the sum of the cooling load requirement and charge/discharge rate as per (3-3). The maximum cooling capacity that can be produced by the chillers is the nominal chiller capacity adjusted for ambient air temperature, as indicated in (4-6). The minimum value is when there is no building cooling load present and the ice storage is not being charged, therefore the chillers are switched off.

$$\boldsymbol{ct3} = \boldsymbol{ct3} + (0 \leq Q_L + u_k \leq CCAP_k) \tag{4-19}$$

- ***ct4*** – Defines the maximum demand cost equation by combining (4-14) and the second term of (4-15). As described in 4.4.3.2, the decision variable *lamda* is defined within the constraint as the maximum value. Part of the objective will be to minimise the maximum demand cost, *lamda*¸ through optimal selection of the decision variable ***u***.

When $Q_L = 0$, thus ice build or charging mode, ***ct4*** is represented in (4-20).

$$\boldsymbol{ct4} = \boldsymbol{ct4} + \left(\frac{Q_L + u_k}{T'^{ice} * \left(-1.6011 * \frac{Q_L + u_k}{CCAP_k} + 4.282\right)} * d_k \leq lamda\right) \tag{4-20}$$

When $Q_L > 0$, thus ice melt or discharging mode, ***ct4*** is represented in (4-21).

$$\boldsymbol{ct4} = \boldsymbol{ct4} + \left(\frac{Q_L + u_k}{T'^{chw} * \left(-1.6011 * \frac{Q_L + u_k}{CCAP_k} + 4.344\right)} * d_k \leq lamda\right) \tag{4-21}$$

- ***ct5*** – Defines the energy cost constraint as per the first term of (4-15).

When $Q_L = 0$, thus ice build or charging mode, ***ct5*** is represented in (4-22).

$$\boldsymbol{ct5} = \boldsymbol{ct5} + \left(\frac{Q_L + u_k}{COP_k^{true} * \left(-1.6011 * \frac{Q_L + u_k}{CCAP_k} + 4.282\right)} * e_k\right) \tag{4-22}$$

When $Q_L > 0$, thus ice melt or discharging mode, ***ct5*** is represented in (4-23).

$$\boldsymbol{ct5} = \boldsymbol{ct5} + \left(\frac{Q_L + u_k}{COP_k^{true} * \left(-1.6011 * \frac{Q_L + u_k}{CCAP_k} + 4.344\right)} * e_k\right) \tag{4-23}$$

4.4.4 Initial Conditions

It is assumed that at the start of each month the ice-storage vessel is fully charged and the total energy cost at the start of each month is equal to zero, thus;

$$x_0 = 1 \text{ and } \boldsymbol{ct5} = 0.$$

4.5 VECTOR CALCULATION

4.5.1 FOR-Loop/IF Statement

A FOR-loop is introduced to the algorithm to calculate the different vectors for each time instant of the month, from 1 to the maximum number of time instants N.

Embedded within the FOR-loop is an IF statement, which allows for the two operational scenarios applicable to the system under investigation, namely:

- When there is no building cooling load requirement for the time instant, thus $Q_k^L = 0$, all equations are set to allow for ice-storage charging (ice-build) and $M = 0$.

- Else, if there is building cooling load present for the time instant, all equations are set to storage discharge equations and $M = 1$.

4.5.2 Objective Function and Constraints

The objective of the Yalmip solver is to minimise the objective function, *objfun*, which is equal to the total billing energy cost, which is the sum of the energy cost vector, ***ct5*** and the maximum demand scalar *lamda*.

The minimisation of the objective function is subject to the constraint vector ***F***, which consists of the sum of the constraints ***ct1***, ***ct2***, ***ct3*** and ***ct4*** as defined in 4.4.3.3.

4.5.3 Equation Analysis and Type

To select a particular solver algorithm for the optimal control model presented, it should be known whether the objective function and constraint equations are linear or non-linear equations.

From (4-15), the objective function is expressed as:

$$J = \sum_{k=1}^{N} P_k e_k + Max_{1\le k\le N}(P_k d_k)$$

Where, the electrical energy input to the chillers, P_k, is presented in (4-14):

$$P_k = \frac{Q_k^L + u_k}{T'^{ice}.COP_k^{PLR}.(1-M) + T'^{chw}.COP_k^{PLR}.M}$$

Replacing (4-7) and (4-8) within (4-14), the objective function is expanded to indicate that the decision variable u_k is not to the first degree and therefore non-linear.

For $M = 1$, (4-14) equates to:

$$P_k = \frac{Q_k^L + u_k}{T'^{chw}(-1.6011(\frac{Q_k^L+u_k}{CCAP_k}) + 4.344)} \tag{4-24}$$

and for $M = 0$, (4-14) equates to:

$$P_k = \frac{Q_k^L + u_k}{T'^{ice}(-1.6011(\frac{Q_k^L+u_k}{CCAP_k}) + 4.284)} \tag{4-25}$$

The same argument holds true for constraint equations $\boldsymbol{ct4}$ and $\boldsymbol{ct5}$, as represented in (4-20), (4-21), (4-22) and (4-23).

The optimal control model therefore consists of a non-linear objective function and non-linear constraints. The selected solver should therefore be able to solve a non-linear optimal control model.

4.5.4 Idle Modes

Table 4.3 indicates the conditions under which the chillers will be in idle mode. The control model equations are evaluated to ensure that all equations hold true and do not equate to an indeterminate form, should an idle mode state occur in any time instant.

Table 4.2. Chiller idle modes

Mode	$P_k =$	Q_k^L	u_k	Ice Store State
1.	0	$= 0$	$= 0$	Idle
2.	0	$= u_k$	$= Q_k^L$	Discharging

Evaluating equation (4-24) and (4-25), it can be concluded that the objective function equation (4-15) and the constraint functions (4-20), (4-21), (4-22) and (4-23) will be indeterminate if either $CCAP_k$ or the denominator of (4-24) and (4-25) is equal to zero.

From (3-11) it is concluded that $CCAP_k$ can only be equal to a real number and the denominator in (4-24) and (4-25) can therefore not equate to zero.

4.6 SOLVER OPTIONS

An interior point optimizer, IPOPT, was used as the chosen solver in the Yalmip toolbox to solve the optimisation model. IPOPT solves smooth, twice differentiable, nonlinear programs [48].

4.7 OPTIMAL CONTROL SOLUTION

The figures below present the results of the optimal scheduling of the ice storage to minimise the billing cost at the end of each month through the use of the Yalmip toolbox with IPOPT solver in Matlab.

4.7.1 Solution – Summer day

The optimal control simulation and scheduling of the ice storage and chiller to meet the load for a typical 24-hour summer day is presented in Figure 4.2.

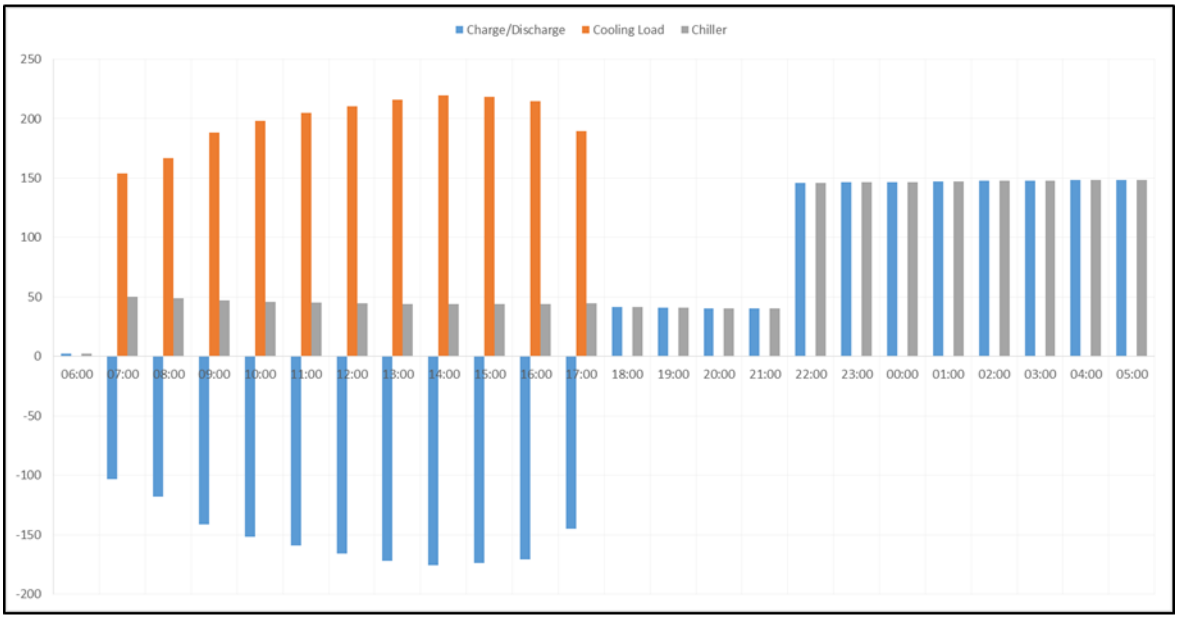


Figure 4.2. Optimal control – summer day

It can be concluded that the control strategy aims to use as much energy contained in the ice-storage tank during the peak and standard billing cost periods as possible and then replenish the ice storage during the off-peak hours. The optimal control strategy places a high priority on the ice-storage tank to meet the building cooling load requirement in the higher cost time instants.

4.7.2 Solution – Winter day

In the winter months the optimal control strategy only uses the ice storage to meet the daily cooling load. The chiller only operates in the off-peak periods at night to replenish the ice-storage tank, as indicated in Figure 4.3.

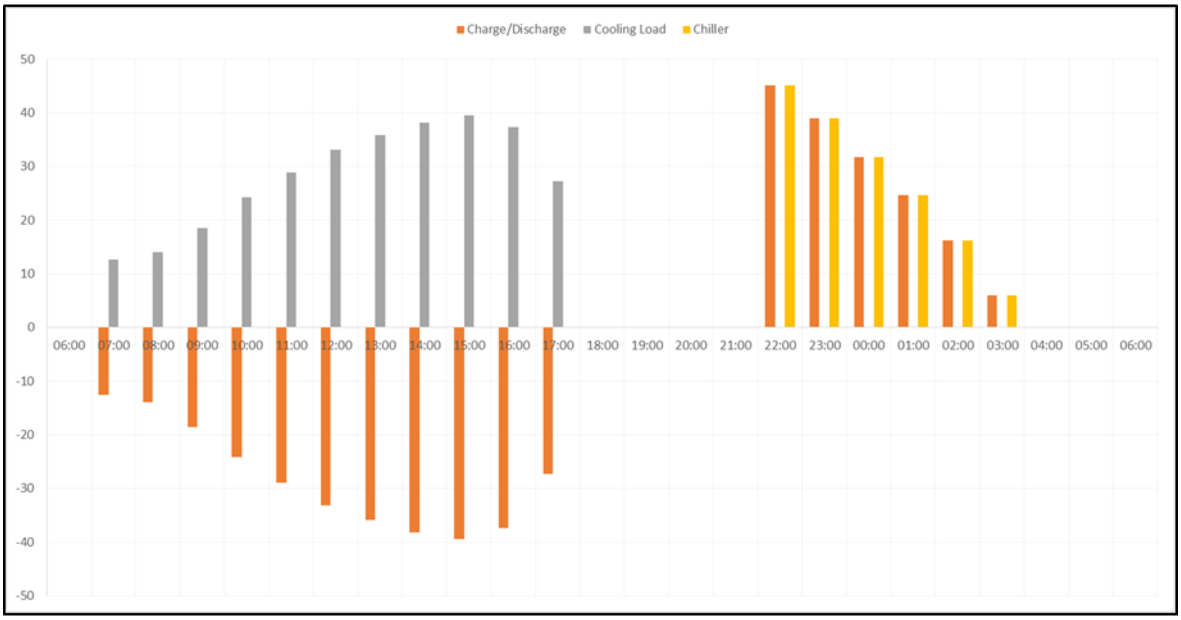


Figure 4.3. Optimal control – winter day

CHAPTER 5 SIMULATION RESULTS AND COMPARISON

5.1 CHAPTER OVERVIEW

Chapter 5 presents the simulation experiments of the HVAC system, considering the different control strategies. Three control strategies are presented and the results are compared. The first set of results obtained is for a system without a TES. The second set of simulation results is for the conventional control system currently deployed on the HVAC system with TES and the third is the optimal control simulation presented in Chapter 4. All simulations were programmed in Matlab and executed for each month of the year, considering only the building cooling load cycle.

5.2 SIMULATION EXPERIMENTS

The input variables to the system are the building cooling load, the dry-bulb ambient air temperature and the energy and demand cost for each time instant. The input variable data were obtained from the HAP Carrier simulation software that was used for the design of the HVAC system. The hourly instant data are available for each month of the year, as included in Addendum F.

The hourly instant input data are contained in a separate Excel spreadsheet for each month. To ensure that exactly the same input data are used, each algorithm calls the same spreadsheet for the applicable month.

5.3 HVAC SYSTEM WITH NO TES

The first simulation that was performed was intended to determine the benchmark monthly billing cost when the HVAC system is not equipped with an ice-storage tank. The algorithm was coded as a Matlab M-File for each individual month [49]. The example code for the month of January is included in Addendum G.

For the simulation of the HVAC system without an ice-storage tank, it is noted that the maximum chiller capacity is less than the maximum design day cooling load. This holds true only for a few time instants and the majority of cooling loads are below the chiller maximum design capacity. This simulation is therefore only for a theoretical benchmark of the expected billing cost for each month if the HVAC system was designed without an ice-storage tank.

The total billing cost is the sum of each time instant energy cost and the maximum demand cost for the month. The results are graphically represented in Figure 5.1, including the total billing cost.

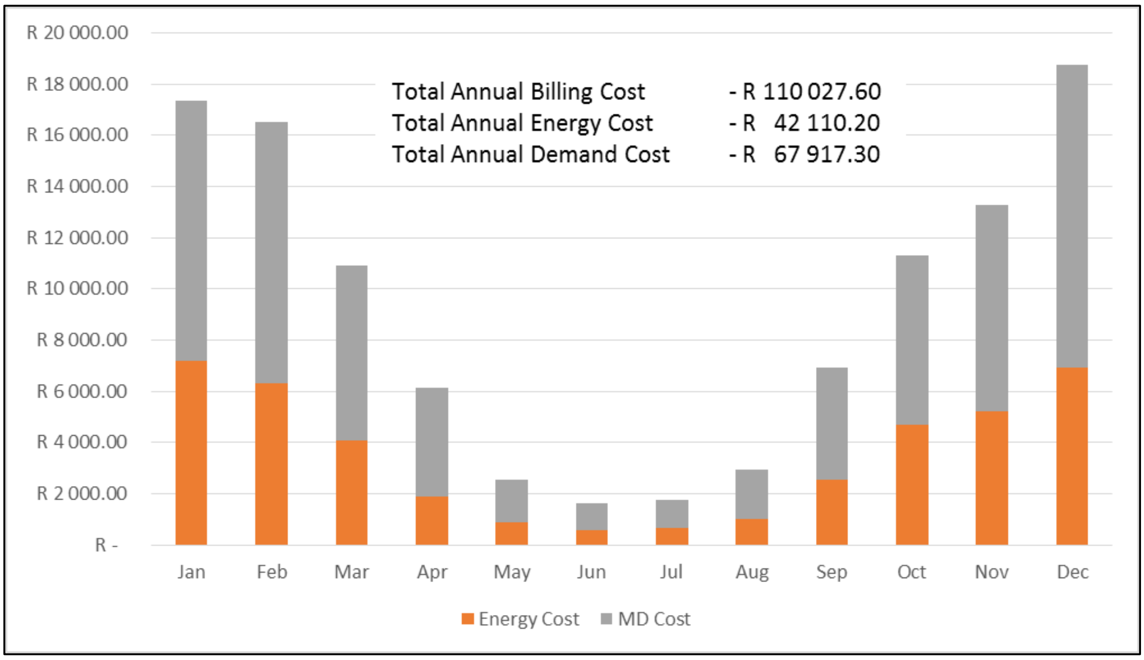


Figure 5.1. Monthly billing cost – no thermal energy storage

Evaluating the energy and demand cost as a percentage of the total cost indicates that 38% of the cost is energy cost and 62% is maximum demand cost.

The total cost for the electrical energy consumed by the chillers to supply the building cooling load is R 110 027.60 for the year under consideration.

5.4 HVAC SYSTEM WITH TES - CONVENTIONAL CONTROL

The conventional control algorithm was written to represent the same decision-making objectives as the current conventional control strategy employed in the HVAC system with ice-storage tank. A control algorithm was defined for each of the conventional control strategies employed for the different seasons: summer, intermediate and winter.

As presented in Chapter 2, each of the seasons has predefined actions depending on the month of the year.

It is once again emphasised that only the building cooling load cycle is modelled and evaluated in this study.

5.4.1 Summer Months

The summer months include the months from November to February. In the summer months, both chillers supply the building cooling load until they reach 50% of their rated capacity; thereafter the ice storage assists the chillers to supply the additional cooling load requirement.

At the start of each month, it is assumed that the ice-storage tank is fully charged. The initial value of the state-of-charge variable x_0 is equal to one. The ice storage can only be charged when there is no cooling load requirement. The conventional control strategy will aim to keep the ice-storage tank fully charged and will charge the ice storage as long as the state-of-charge variable x_k is less than one (not fully charged) and there is no cooling load present.

It should be noted that even though the chillers are capped at 50% of their supply capacity when meeting the cooling load, when building ice this restriction is not in place and the chillers will run at 100% of their rated capacity.

The algorithm for conventional control of the summer months is included in Addendum H.

5.4.2 Intermediate Months

The intermediate months include the autumn and spring months of March, April, September and October. In the intermediate months, one chiller is used to supply the heating load requirement and the other chiller 50% of the cooling load requirement. Any additional cooling load capacity, over and above what the single chiller can provide, is supplied by the ice storage.

At the start of each month, it is assumed that the ice-storage tank is fully charged. The initial value of the state-of-charge variable x_0 is equal to one. The ice storage can only be charged when there is no cooling load requirement. The conventional control strategy will aim to keep the ice-storage tank fully charged and will charge the ice storage as long as the state-of-charge variable x_k is less than one (not fully charged), and there is no cooling load present.

It should be noted that even though the single chiller is capped at 50% of its supply capacity when meeting the cooling load, when building ice this restriction is not in place and the chiller will be allowed to operate at 100% of its rated capacity.

The algorithm for conventional control of the intermediate months is included in Addendum I.

5.4.3 Winter Months

The winter months include the months of May, June, July and August. In the winter months, both chillers are used to supply the heating load requirement of the building. The cooling load requirement is fully met by the ice storage only.

At the start of each month, it is assumed that the ice-storage tank is fully charged. The initial value of the state-of-charge variable x_0 is equal to one. The ice storage is only allowed to charge from 22:00 to 06:00. The conventional control strategy will aim to keep the ice-storage tank fully charged and will charge the ice storage as long as the state-of-charge variable x_k is less than one (not fully charged).

The algorithm for conventional control of the winter months is included in Addendum J.

5.4.4 Conventional Control Results

The simulation results are included in Table 5.1 and graphically represented in Figure 5.2.

Table 5.1. Monthly billing cost – conventional control

	Total Cost	**Energy Cost**	**MD Cost**
Jan	R 9 021.30	R 5 754.00	R 3 267.30
Feb	R 8 703.00	R 5 143.20	R 3 559.80
Mar	R 8 114.00	R 4 333.80	R 3 780.20
Apr	R 6 390.50	R 2 736.20	R 3 654.30
May	R 1 270.00	R 1 270.00	R -
Jun	R 918.47	R 918.47	R -
Jul	R 1 098.20	R 1 098.20	R -
Aug	R 1 421.70	R 1 421.70	R -
Sep	R 7 008.20	R 3 347.40	R 3 660.80
Oct	R 8 824.00	R 4 980.60	R 3 843.40
Nov	R 8 250.20	R 4 773.60	R 3476.60
Dec	R 8 665.00	R 5 382.50	R 3 282.40
Total	R 69 684.57	R 41 159.67	R 28 524.80

As indicated in Table 5.1, the total annual billing cost for the system with TES and conventional control strategy is R 69 684.57, which is an effective reduction of 37% in total billing cost compared with the system without an ice-storage tank.

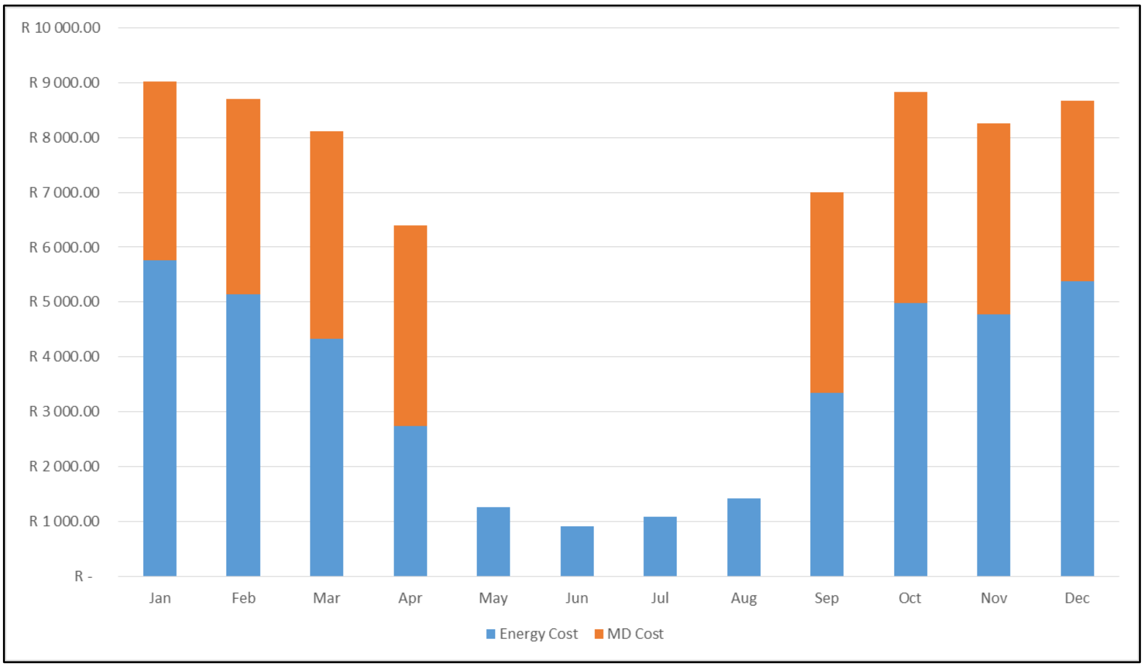


Figure 5.2. Monthly billing cost – conventional control

5.4.5 Data Evaluation

To test if the simulation algorithms are functioning correctly, all variables were checked to ensure that they were within limits. The most important variable is the state of charge of the ice-storage tank, which should be a value between zero and one. It was found that for all months this held true.

In addition, the results were assessed by evaluating the ratio between known cooling load data and the actual cooling energy produced by the chiller. This ratio will be referred to as the correlation factor. Secondly, the performance of the chiller for each month was assessed by calculating the average COP for each month, thus evaluating the ratio between the electrical energy input to the chillers and the total building cooling load energy requirement to be supplied by the chiller for each month.

5.4.5.1 Correlation Factor

As presented in Table 5.2, a variance exists between the input data cooling load and the calculated chiller output for most months. The variance is due to the fact that the initial condition of the model assumes a fully charged ice-storage tank at the start of each month and the algorithm aims to replenish the ice store as long as no cooling load is present and the state of charge is less than one.

The months that have 100% correlation have a perfect balance between the initial state of charge energy used during the month and energy replenished at the end of the month. The state-of-charge of the ice-storage tank for these months at the end of the month is equal to one.

Table 5.2. Correlation factor – conventional control

	Cooling Load (kWh)	Chiller Output (kWh)	Correlation Factor
Jan	43 843.00	43 523.00	99.27%
Feb	39 041.00	38 784.00	99.34%
Mar	29 071.00	28 630.00	98.48%
Apr	16 108.00	16 108.00	100.00%
May	8 853.00	8 853.00	100.00%
Jun	6 144.90	6 015.00	97.89%
Jul	7289.50	7 127.80	97.78%
Aug	9 816.80	9 816.80	100.00%
Sep	20 388.00	20 330.00	99.72%
Oct	33 086.00	32 624.00	98.60%
Nov	34 555.00	34 555.00	100.00%
Dec	41 275.00	40 519.00	98.17%

5.4.5.2 Energy Performance of the Chiller

The energy performance or the average COP for each month is presented in Table 5.3. The COP values were evaluated and found to be realistic for the chiller type used in the building under consideration.

Table 5.3. Coefficient of performance – conventional control

	Chiller Output (kWh)	Chiller Input (kWh)	COP
Jan	43 523.00	11 964.00	3.64
Feb	38 784.00	10 625.00	3.65
Mar	28 630.00	9 577.20	2.99
Apr	16 108.00	4 851.60	3.32
May	8 853.00	3 950.10	2.24
Jun	6 015.00	2 856.80	2.11
Jul	7 127.80	3 415.80	2.09
Aug	9 816.80	4 422.10	2.22
Sep	20 330.00	6 443.40	3.16
Oct	32 624.00	11 028.00	2.96
Nov	34 555.00	9 436.90	3.66
Dec	40 519.00	11 233.00	3.61

It can be seen from Table 5.3 that the COP is much lower in the winter months compared to the other months of the year. The decrease in the COP in the winter months is as expected and is due to the following reasons:

- In the winter months the chillers only produce ice at -6°C, which contributes to a very high temperature difference compared to the other months where the chillers also produce chilled water at 6°C.

- The chillers always run at full capacity (part-load-ratio equal to one) when producing ice, thus the COP_{plr} is always at the lowest value, as indicated in Figure 3.2.

5.5 OPTIMAL CONTROL SIMULATION

The optimal control simulation as presented in Chapter 4 is calculated for each month of the year. The results are then compared with the simulation results of the system assuming no TES is installed and the conventional control strategy.

The optimisation algorithm is included in Addendum K.

5.5.1 Optimisation Simulation Results

The simulation results for the optimal control simulation are included in Table 5.4 and graphically represented in Figure 5.3.

Table 5.4. Monthly billing cost – optimal control

	Total Cost	Energy Cost	MD Cost
Jan	R 6 399.54	R 5079.54	R 1 320.00
Feb	R 5 777.42	R 4 472.83	R 1 304.59
Mar	R 3 459.58	R 3 169.65	R 289.93
Apr	R 1 447.00	R 1 447.00	R -
May	R 728.18	R 728.18	R 0.00
Jun	R 478.26	R 478.26	R 0.00
Jul	R 608.11	R 608.11	R -
Aug	R 833.89	R 833.89	R -
Sep	R 2 004.70	R 2004.70	R 0.00
Oct	R 4 010.17	R 3719.23	R 290.94
Nov	R 4 678.18	R 3 943.46	R 734.72

Dec	R 6 474.00	R 4 600.89	R 1 873.11
Total	R 36 899.04	R 31 085.75	R 5 813.30

As indicated in Table 5.4, the total electrical energy cost for the optimal control strategy is R 36 899.80, which is an effective reduction of 47% in total annual energy cost compared with the system with TES and conventional control strategy.

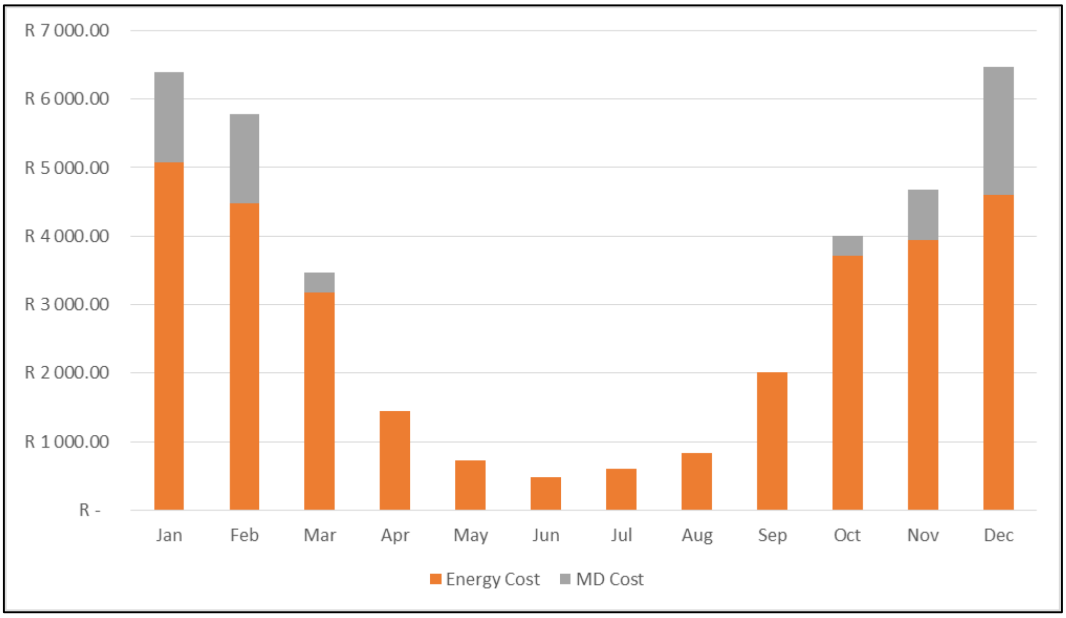


Figure 5.3. Monthly billing cost – optimal control

5.5.2 Data Evaluation

To test if the optimisation algorithm is functioning correctly, all variables were checked to ensure that they were within specified constraints. The state of the ice-storage charge was evaluated and found to be greater or equal to zero and less than or equal to one. The optimal control mode therefore did not overcharge or undercharge the ice-storage tank. This was true for all months.

As for the conventional control strategy, the results were assessed by evaluating the ratio between known cooling load data and the actual cooling energy produced by the chiller (energy correlation factor). The performance of the chiller for each month was assessed by calculating the average COP for each month, thus evaluating the ratio between the electrical energy input and the total cooling load on the chiller for each month.

5.5.2.1 Correlation Factor

As presented in Table 5.5, the optimal control algorithm produces a 100% correlation between the cooling load energy required by the system and the energy produced by the chillers when considering the energy available at the start of each month. The initial state of the ice-storage tank is being fully charged, therefore there is 1693 kW of stored energy available at the start of each month.

The 100% correlation concludes that the optimal control algorithm uses all energy available in the ice-storage tank to meet the cooling load and totally depletes the ice store at the end of each month.

Table 5.5. Correlation factor – optimal control

	Cooling Load (kWh)	Chiller Output (kWh)	Initial State Energy (kWh)	Correlation Factor
Jan	43 843.00	42 149.99992	1 693	100.00%
Feb	39 040.80	37 347.79996	1 693	100.00%
Mar	29 071.20	27 378.19994	1 693	100.00%
Apr	16 108.30	144 15.30001	1 693	100.00%
May	8 853.00	7 159.999995	1 693	100.00%
Jun	6 144.90	4 451.899986	1 693	100.00%
Jul	7 289.50	5 596.499985	1 693	100.00%
Aug	9 816.80	8 123.799992	1 693	100.00%
Sep	20 387.60	18 694.6	1 693	100.00%
Oct	33 085.60	31 392.59994	1 693	100.00%
Nov	34 555.00	32 861.99994	1 693	100.00%

Dec	41 274.50	39 581.50	1 693	100.00%

5.5.2.2 Energy Performance of the Chiller

The energy performance or the average COP for each month is presented in Table 5.6. The COP values were evaluated and found to be realistic for the chiller type used in the building under consideration.

Table 5.6. Coefficient-of-performance – optimal control

	Chiller Output (kWh)	Chiller Input (kWh)	COP
Jan	42 150.00	13 363.01	3.15
Feb	37 347.80	11 794.06	3.17
Mar	27 378.20	9 326.74	2.94
Apr	14 415.30	4 500.79	3.20
May	7 160.00	2 264.95	3.16
Jun	4 451.90	1 487.60	2.99
Jul	5 596.50	1 891.47	2.96
Aug	8 123.80	2 593.76	3.13
Sep	18 694.60	6 235.45	3.00
Oct	31 392.60	10 931.27	2.87
Nov	32 862.00	10 917.51	3.01
Dec	39 581.50	11 940.43	3.31

CHAPTER 6 DISCUSSION

6.1 CHAPTER OVERVIEW

In this chapter the results from the simulations are compared and discussed.

6.2 BILLING COST COMPARISON

The comparison of the total monthly billing costs is presented in Figure 6.1.

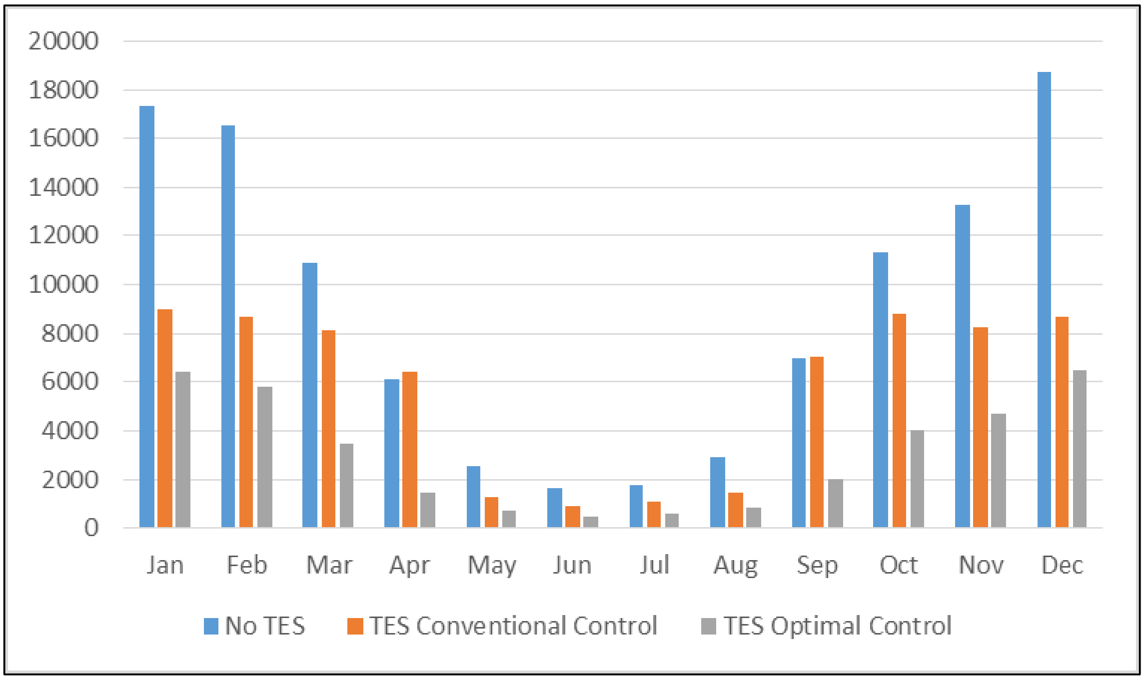


Figure 6.1. Total monthly billing cost comparison

The total cost per annum and the percentage difference are indicated in Figure 6.2. The following conclusions can be made from Figure 6.2:

- The TES drastically reduces the demand-related cost compared with a system that does not have an ice-storage tank.

- The optimal control strategy offers an additional cost saving, compared to conventional control, owing to optimised switching and utilisation of the ice storage.

- The reduction in total cost saving is mainly due to the reduction in demand cost as indicated in Figure 6.2, with the energy cost making a minor contribution.

- The optimal control strategy has a greater saving in energy cost than the conventional control strategy.

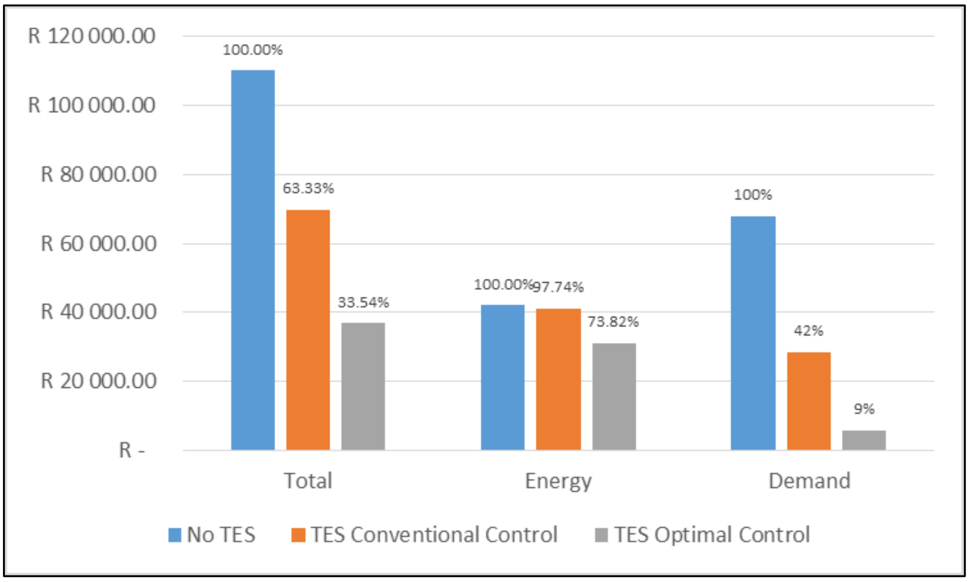


Figure 6.2. Comparison of total annual cost

6.3 ENERGY CONSUMPTION COMPARISON

The total electrical energy consumption of the chillers was compared for the different control strategies, to ascertain which of the systems were more energy-efficient. The total electrical energy consumption in kW for each month of the year is presented in Figure 6.3.

It was found that the system without an ice-storage tank was most energy-efficient, followed by optimal control of the HVAC system with TES. The result was as expected, because the efficiency of the chillers is lower when producing ice than when supplying the cooling load directly.

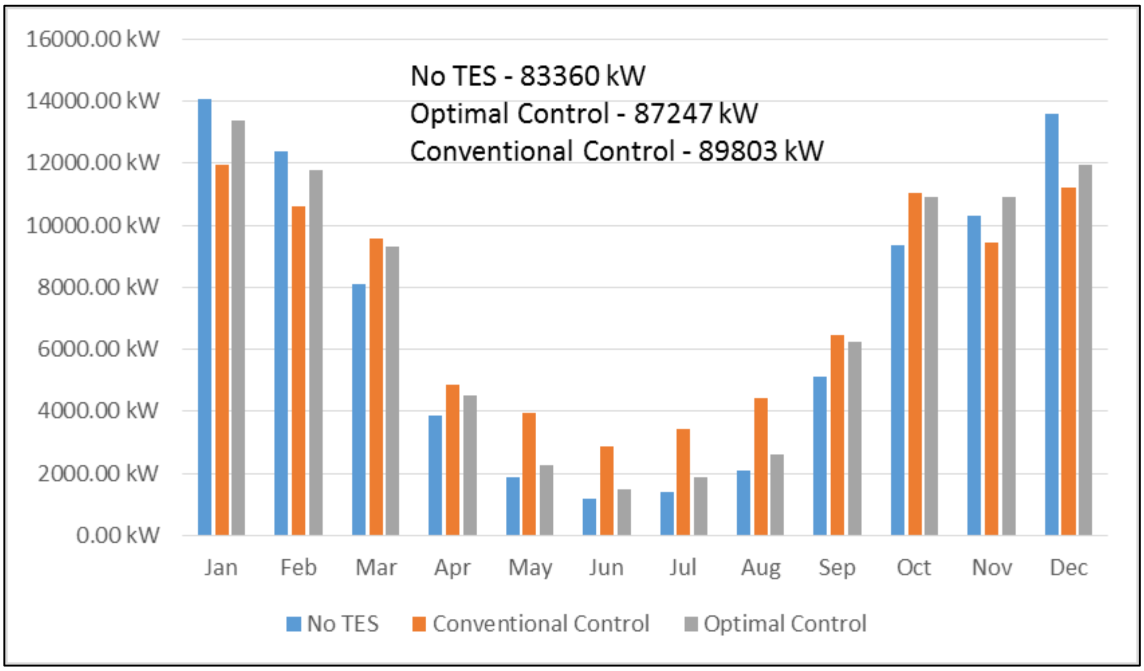


Figure 6.3. Total energy consumption

Evaluating Figure 6.3, the following conclusions can be made:

- Using an ice-storage tank for demand cost reduction through load shifting incurs an energy penalty.

- The optimal control of the HVAC system under consideration is more energy-efficient compared to the conventional control strategy.

6.4 RECOMMENDATIONS

The research indicated that an optimal control strategy will realise considerable cost savings annually compared to the current conventional control strategy employed. To this end, it is recommended that future research focus on the requirements to implement an optimal controller on the HVAC system with TES installed at the SANRAL corporate head office.

Future research may include the following aspects:

- Confirming the results obtained from the simulation algorithms, as part of this research, using actual on-site historical cooling load and ambient condition data.

- Expanding the optimisation algorithm to predict future cooling load and ambient conditions by using closed-loop optimal control, e.g. MPC and comparing the results to similar studies employing MPC.

- Financial analysis of the cost of implementation of the optimal predictive controller and the rate of return due to the saving in annual billing cost over the full life cycle of the plant.

CHAPTER 7 CONCLUSION

The research investigated optimal control of the cooling cycle of an existing HVAC system with an encapsulated ice-storage tank as TES. The TES was originally included as part of the HVAC system design to allow for demand reduction through load shifting of the cooling load requirement to the off-peak evening periods. These measures also contributed to the building under consideration achieving a four-star green building rating based on the GBCSA measurement standards for newly designed energy-efficient buildings.

The scheduling of the storage charge or discharge is currently controlled by means of a conventional control strategy. The conventional control strategy predefines modes of operation for the different seasons of the year and operating hours of the day. The current conventional control strategy can be classified as a chiller priority control strategy.

To reduce energy demand, the chillers are only allowed to operate up to 50% of their rated capacity in summer months, whereafter the ice storage allows for any additional energy requirements for the day. In the autumn months one chiller is dedicated to the heating load and the second in combination with the ice storage for the building cooling load requirement. In the winter months both chillers are dedicated to supply the heating load requirement and the ice storage is exclusively used to meet the winter cooling load requirement. All strategies aim to replenish the ice storage fully in the evening hours.

Through a detailed analysis of the cooling load cycle of the HVAC system, a mathematical model is presented with the use of a binary variable *M* to denote different operational scenarios. The mathematical model is used to compile simulation algorithms for three different control strategies. To obtain a comparative benchmark for the system, the first algorithm assumed that the entire building cooling load is directly supplied by the chillers and the HVAC system does not include a TES. The second algorithm consisted of three parts, each representing the conventional control strategy currently employed on the HVAC system with TES. Lastly, an optimal control algorithm was written using the Yalmip toolbox in Matlab with the IPOPT as the nonlinear optimisation solver.

The objective of the optimal control strategy is to minimise the total billing cost per month. The billing cost for each month has two parts. The first part is the energy consumption, which is measured on a TOU tariff structure, and the second is the maximum demand cost. The chosen decision variable is the charge/discharge rate of the ice-storage tank. The optimal control algorithm is constrained by the state of the ice-storage capacity, the chiller capacity and the energy balance equation.

The input data are the simulated design cooling load data for the building under consideration as produced by the HAP Carrier HVAC system design software package. To ensure an accurate comparison of the results obtained for each of the simulation algorithms, exactly the same input data were used. Each month had a separate Excel spreadsheet containing the input data for the applicable month, which was subsequently called within the different algorithms from Matlab when running the simulations. The input data consist of the building cooling load, outdoor dry-bulb temperature, energy cost and demand cost per hourly instant.

Comparing the results of the simulations leads to the conclusion that the optimal control strategy realises the lowest annual billing cost. The optimal control strategy realises an additional cost saving of 47% compared to the conventional control strategy currently employed on the HVAC system with TES for the building under consideration.

To replace the conventional control strategy with an optimal control model, it is recommended that further research should obtain historic building cooling load and ambient temperature data to confirm the theoretical results presented in this research. In addition, to implement the optimal control strategy in the future, the input variables will have to be accurately predicted. Future research should include closed-loop, MPC theory to accurately incorporate the expected building cooling load and ambient conditions based on historical data and to handle a degree of possible disturbances.

REFERENCES

[1] M. Maleka *et al*., "South African energy synopsis 2010," The South African Department of Energy, Tech. Rep. 978-1-920435-4, 2010.

[2] G. Heffner *et al*., "Minding the gap: World Bank's assistance to power shortage mitigation in the developing world," *Energy*, vol. 35, pp. 1584-1491, 2010.

[3] T. Mbeki, "State of the nation address of the president of south africa," The Presidency, Government of South Africa, 8 February 2008.

[4] Eskom, *Tariffs & Charges Booklet 2012/2013*. www.eskom.co.za: 2012.

[5] Eskom, *Integrated Report 2014*. integratedreport.eskom.co.za: 2014.

[6] Eskom, *Mind Your Own Business: Energy Efficiency in Commercial Properties.* www.eskom.co.za/dsm: 2010.

[7] (2013, June 12). *About* [Online]. Available: https://www.gbcsa.org.za/.

[8] P. C. Pelser and J.B. Theron, "Project Specification for the HVAC system for the new SANRAL corporate head office," Plantech Associates, Project A-1240, 2010.

[9] Creamer Media, "Green building groundswell sees 50 Green Star SA certified buildings in SA," *Engineering News*, Feb, 2014.

[10] J. E. Braun, "Comparison of chiller-priority storage-priority, and optimal control of an ice-storage system," *ASHRE Trans.*, pp. 893-902, 1992.

[11] P. Rupanagunta *et al.*, "Scheduling of cool storage using non-linear programming techniques," *IEEE Trans. Power Syst.*, vol. 10, pp. 1279-1285, 1995.

[12] J. E. Braun, "A near-optimal control strategy for cool storage systems with dynamic electric rates (RP-1252)," *HVAC and R Research*, vol. 13, pp. 557-580, 2007.

[13] SA Weather Service. (2011, June). *Climate data for Pretoria* [Online]. Available: www.weathersa.co.za.

[14] J. Zhang, "Energy optimization lecture notes 2," University of Pretoria, *ENO732*, 2014.

[15] C. W. Gellings, "The concept of demand-side management for electric utilities," *Proceedings of the IEEE*, vol. 73, pp. 1468-1470, 1985.

[16] Creamer Media. (2013, Jul 10). *Company announcement: Eskom reports sound financial results for the fourth consecutive year* [Online]. Available: http://www.engineeringnews.co.za

[17] Eskom. (2013, July 29). *HVAC (heating, ventilation and air conditioning) – Thermal energy storage* [Online]. Available: http://www.eskom.co.za

[18] National Energy Regulator of South Africa. (2014, August 21). *Electricity licenses registered entities* [Online]. Available: http://www.nersa.org.za/

[19] City of Tshwane Metropolitan Municipality, "Electricity tariffs part I_2013-14 NERSA approved," Jun. 2013.

[20] E. Oró *et al.*, "Review on phase change materials (PCMs) for cold thermal energy storage applications," *Appl. Energy*, vol. 99, pp. 513-533, 2012.

[21] *HVAC systems and equipment*, ASHRAE, 2012.

[22] G. P. Henze, "An overview of optimal control for central cooling plants with ice thermal energy storage," *Journal of Solar Energy Engineering, Transactions of the ASME*, vol. 125, pp. 302-309, 2003.

[23] G. Henderson. (2009, Oct 14). *Ice Thermal Storage Systems* [Online]. Available: http://www.ashraebistate.org

[24] P. C. Pelser, "Modelling and evaluation of an energy efficient heating ventilation and air conditioning (HVAC) system in an office building," M.Eng. thesis, Dept. Mech. Eng., Univ. Johannesburg, Auckland Park, Gauteng, 2012.

[25] L. K. Rawlings, "Ice storage system optimization and control strategies," *ASHRAE Transactions*, pp. 12-23, 1985.

[26] *HVAC applications*, ASHRAE, 2011.

[27] J. Zhang and X. Xia, "Best switching time of hot water cylinder-switched optimal control approach," in *AFRICON*, 2007 © IEEE. doi: 10.1109/AFRCON.2007.4401505

[28] A. J. van Staden *et al.*, "A model predictive control strategy for load shifting in a water pumping scheme with maximum demand charges," *Appl. Energy*, vol. 88, pp. 4785-4794, June, 2011.

[29] W. Badenhorst *et al.*, "Optimal hoist scheduling of a deep level mine twin rock winder system for demand side management," *Electr. Power Syst. Res.*, vol. 81, pp. 1088-1095, 2011.

[30] H. Akbari and O. Sezgen, "Performance evaluation of thermal energy storage systems," *Energy Build.*, vol. 22, pp. 15-24, 1995.

[31] K. H. Drees and J. E. Braun, "Development and evaluation of a rule-based control strategy for ice storage systems," *HVAC and R Research*, vol. 2, pp. 312-336, 1996.

[32] G. P. Henze *et al.*, "Development of a predictive optimal controller for thermal energy storage systems," *HVAC and R Research*, vol. 3, pp. 233-264, 1997.

[33] G. P. Henze *et al.*, "Development of a predictive optimal controller for thermal energy storage systems," *HVAC and R Research*, vol. 3, pp. 233-264, 1997.

[34] P. Simmonds, "Comparison of energy consumption for storage priority and chiller priority for ice-based thermal storage systems," in *SYMPOSIA*, © 1994 ASHRAE Transactions.

[35] N. Nassif *et al.*, "Optimal design and operation of ice thermal storage system for a typical chilled water plant," in *Proc. 13th Int. Conf. Enhanced Building Operations*, Montreal, Quebec, 2013, pp. 248-255.

[36] Carries. (2014, August 21). *Hourly analysis program* [Online]. Available: http://www.commercial.carrier.com

[37] *Energy standard for buildings except low-rise residential buildings,* ANSI/ASHRAE/IES Standard 90.1-2010.

[38] G. P. Henze and M. Krarti, "Ice storage system controls for the reduction of operating cost and energy use," *Journal of Solar Energy Engineering, Transactions of the ASME*, vol. 120, pp. 275-281, 1998.

[39] CIAT, "*AQUACIAT2 R410A Product Brochure*," 2010.

[40] Eurovent. (2014, Aug 20). *Liquid chilling packages and heat pumps* [Online]. Available: http://www.eurovent-certification.com/

[41] K. M. Powell *et al.*, "Optimal chiller loading in a district cooling system with thermal energy storage," *Energy*, vol. 50, pp. 445-453, 2013.

[42] J. Zhang and X. Xia, "A model predictive control approach to the periodic implementation of the solutions of the optimal dynamic resource allocation problem," *Automatica*, vol. 47, pp. 358-362, 2011.

[43] X. Xia *et al.*, "An application of model predictive control to the dynamic economic dispatch of power generation," *Control Eng. Pract.*, vol. 19, pp. 638-648, 2011.

[44] D. MacPhee *et al.*, "Numerical simulation and exergetic performance assessment of charging process in encapsulated ice thermal energy storage system," *Energy,* vol. 41, pp. 491-498, 2012.

[45] D. MacPhee and I. Dincer, "Performance assessment of some ice TES systems," *Int. J. Therm. Sci.,* vol. 48, pp. 2288-2299, 2009.

[46] M. Rosen *et al.*, "Thermodynamic Performance of Ice Thermal Energy Storage Systems," *Asme,* vol. 122, pp. 205-211, Sept, 2000.

[47] J. Löfberg, "YALMIP: A toolbox for modeling and optimization in MATLAB," in *Proceedings of the CACSD Conference,* Taipei, Taiwan, 2004.

[48] Y. Kawajir *et al.*, "Introduction to IPOPT: A tutorial for downloading, installing and using IPOPT." MIT, Tech. Rep. 2422, Nov. 2013.

[49] Mathworks. (2014, Nov 25). *MATLAB and Simulink for technical computing* [Online]. Available: http://www.mathworks.com/.

ADDENDUM A – DESIGN DAY DATA

				A	B	C	D	E	F	G	H	I	J
				CHILLER PLANT OPERATION							PUMP OPERATION		
				HAP v4.5 Simulation Results		Chiller Assist Operation as per HVAC Plant Schedule (07:00-18:00) 11 hours				Ice Build Operation (18:00-07:00) 13 hours	HAP v4.5 Simulation Results		Ice Build Operation (18:00-07:00) 13 hours
						Chiller Plant		Ice Plant (Melting of Ice)					
Month	Day	Hour	Dry-Bulb Temp	Chiller Output (Plant Load) *Building Cooling Load*	Chiller Input *Under Partial Load Conditions - derated COP of ice build not accounted for*	Chiller Output *Limited to 96.3 kW thereafter building load is met by melting ice*	Chiller Input *Under Partial Load Conditions*	Ice capacity discharged to meet balance of building cooling load	Energy input required from Ice Build Operation to deliver discharge capacity	Chiller Input *Actual Chiller Plant input at derated COP of 2.54 needed to recharge storage vessel to full capacity of 1549 kWh*	Primary Chilled Water Pump	Secondary Chilled Water Pump	Primary Chilled Water Pump
			(−C)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)	(kW)
Dec	29	0	17.8	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	1	17.1	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	2	16.5	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	3	15.9	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	4	15.4	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	5	15.1	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	6	15.2	0	0	0	0	0	0	40.0848	0	0	3.5
Dec	29	7	16.1	172.4	51.7	96.3	25.5	76.1	29.9606299		3.5	3.5	0
Dec	29	8	17.8	184.9	56.2	96.3	25.5	88.6	34.8818898		3.5	3.5	0
Dec	29	9	20.1	205.4	64.2	96.3	25.5	109.1	42.9527559		3.5	3.5	0
Dec	29	10	22.3	215.5	68.5	96.3	25.5	119.2	46.9291339		3.5	3.5	0
Dec	29	11	24.2	223.5	71.9	96.3	25.5	127.2	50.0787402		3.5	3.5	0
Dec	29	12	25.8	231.1	75.6	96.3	25.5	134.8	53.0708661		3.5	3.5	0
Dec	29	13	26.8	236.5	78.5	96.3	25.5	140.2	55.1968504		3.5	3.5	0
Dec	29	14	27.6	238.3	79.4	96.3	25.5	142	55.9055118		3.5	3.5	0
Dec	29	15	27.7	238	79.2	96.3	25.5	141.7	55.7874016		3.5	3.5	0
Dec	29	16	27.2	232	76.1	96.3	25.5	135.7	53.4251969		3.5	3.5	0
Dec	29	17	26.1	205.3	64.1	96.3	25.5	109	42.9133858		3.5	3.5	0
Dec	29	18	24.6	0	0			0	0	40.0848	0	0	3.5
Dec	29	19	22.9	0	0		0	0	0	40.0848	0	0	3.5
Dec	29	20	21.4	0	0		0	0	0	40.0848	0	0	3.5
Dec	29	21	20.3	0	0		0	0	0	40.0848	0	0	3.5
Dec	29	22	19.4	0	0		0	0	0	40.0848	0	0	3.5
Dec	29	23	18.6	0	0		0	0	0	40.0848	0	0	3.5
				2382.9	**765.4**	**1059.3**	**282.7**	**1323.6**	**521.1**	**521.1**	**38.5**	**38.5**	**45.5**

ADDENDUM B – CHILLER SPECIFICATION (CHILLED WATER)

Reversible water chiller with built-in hydraulic equipment
AQUACIAT2 400V ILDC R410A

Attractive and very quiet packaged outdoor unit. Includes a built-in hydraulic module (Sizes 180 to 300: buffer tank delivered separately with connection hose) for easy installation on site.

Complete control and management by a microprocessor-based electronic unit.

See technical manual No. CAT

Refrigerant fluid / kg : R410A / 24
Number of refrigerant circuits : **1**
Cooling capacity control : 100-63-37 0 %
Starting mode : in cascade

EUROVENT

COOLING OPERATION

Cooling capacity	**: 96.3 kW**
EER / EER without pump	**: 2.62 / 2.70**
ESEER	**: 3.91**
Fluid	: MEG 30%
Inlet/outlet temperature	: 11.0 ℃ / 6.0 ℃
Flow rate	**: 4.83 l/s**
System available pressure	: 91.6 kPa
Connection diameter	: G 2" 1/2
Air inlet temperature	: 35.0 ℃
Fans rotation speed	: 750 rpm
Air flow	: 8.06 m3/s
Number of fans	: 2
Unitary motor power	: 1.20 kW

HEATING OPERATION

Heating capacity	**: 97.4 kW**
COP / COP without pump	**: 2.73 / 2.81**
Fluid	: MEG 30%
Inlet/outlet temperature	: 39.9 ℃ / 45.0 ℃
Flow rate	: 4.83 l/s
System available pressure	: 91.6 kPa
Connection diameter	: G 2" 1/2
WB air inlet temperature	: 5.0 ℃
Fan rotation speed	: 750 rpm
Air flow	: 8.06 m3/s
Number of fans	: 2
Unitary motor power	: 1.20 kW
Total electrical consumption	: 35.7 kW

Total absorbed power : 36.8 kW
Electrical supply : Three-phase, 400 V, 50 Hz
Intensity for selection of the electric cable : 81.3 A
Starting current : 303 A
Starting current with SOFT START option : 191 A

ADDENDUM C – CHILLER SPECIFICATION (ICE BUILD)

Reversible water chiller with built-in hydraulic equipment
AQUACIAT2 400V ILDC R410A

Packaged, quiet and reversible unit well adapted for heating and cooling of various warehouses.
Control by electronic microprocessor module.
As per technical brochure N°CAT

EUROVENT CERTIFIED PERFORMANCE
www.eurovent-certification.com

Refrigerant/kg	:	R410A / 24
Number of refrigerant circuits	:	1
Capacity control	:	100-63-37 0 %
Starting mode	:	in cascade

COOLING OPERATION			HEATING OPERATION		
Cooling capacity	**:**	**71.0 kW**	**Heating capacity**	**:**	**101.9 kW**
EER / EER without pump	**:**	**2.54 / 2.64**	**COP / COP without pump**	**:**	**2.75 / 2.84**
ESEER	**:**	**3.84**	Fluid	:	MEG 30%
Fluid	:	MEG 30%	Inlet/outlet temperature	:	39.8 ℃ / 45.0 ℃
Inlet/outlet temperature	:	-2.2 ℃ / -6.0 ℃	Flow rate	:	4.9 l/s
Flow rate	**:**	**4.9 l/s**	System available pressure	:	80.5 kPa
System available pressure	:	80.5 kPa	Connection diameter	:	G 2" 1/2
Connection diameter	:	G 2" 1/2			
Air inlet temperature	**:**	**25.0 ℃**	**Air inlet temperature**	**:**	**5.0 ℃**
Fan rotation speed	:	900 rpm	Fan rotation speed	:	900 rpm
Air flow	:	11.7 m3/s	Air flow	:	11.7 m3/s
Number of fans	:	2	Number of fans	:	2
Unitary motor power	:	1.70 kW	Unitary motor power	:	1.70 kW
Total power input	**:**	**28.0 kW**	**Total power input**	**:**	**37.0 kW**

Electrical supply : Three-phase, 400 V, 50 Hz
Intensity for selection of the electric cable : 91.6 A
Starting current : 302.0 A
Starting current with SOFT START option : 194.0 A

ADDENDUM D – ICE-STORAGE TANK SPECIFICATION

Selection

Base chiller

Base capacity :	0 kW	Base outlet temperature :	6.0 °C
Base flow rate :	0 m3/h		

Brine chillers

Number of chillers :	1	Total direct capacity :	198 kW

Chiller 1

Chiller Type :	Screw	Flow rate :	37 m3/h
Direct Capacity :	198 kW	Outlet Temperature :	6.0 °C
Charge Capacity :	142 kW	Charge Temperature :	-5.7 °C
Charge set-point :	-7.7 °C	Direct set-point :	5.0 °C

STL

STL - AC.00 - 28

Stored Energy :	1549 kWh	Maximum STL capacity :	1693 kWh
Nodules :	AC.00		
Fusion temperature :	0.0 °C	Volume :	28 m3
Tank 1			
Type :	Vertical	Volume :	28 m3
Diameter :	3.00 m	Service Pressure :	4.0 bar
Length :	5.06 m	Weight :	31624 kg

Additional specifications

Percentage of glycol :	*28%*
Pressurized Tanks - Total useful expansion volume (5%):	*1.40 m3*
Atmospheric Tanks - Total useful expansion volume (1%):	*0.28 m3*

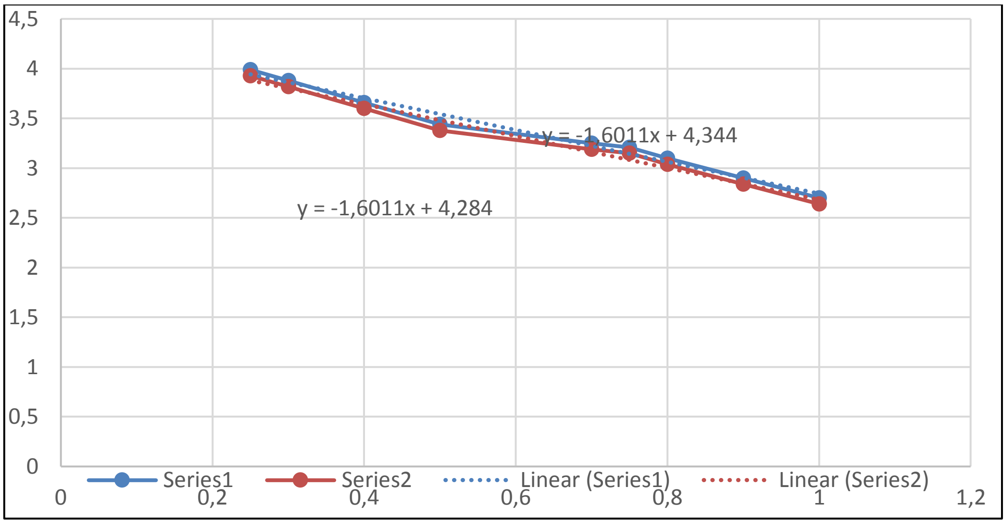
ADDENDUM E – CHILLER PART-LOAD PERFORMANCE

Eurovent conditions

Appareil	LD	400 V
EER charge partielle normalisé	100%	2.99
	75%	3.55
	50%	3.81
	25%	4.42
ESEER = 0,03*EER100+0,33*EER75+0,41*EER50+0,23*EER25		**3.84**

Design conditions

	400 V
100%	**2.7**
90%	2.90
80%	3.10
75%	**3.21**
70%	3.25
60%	3.35
50%	**3.44**
40%	3.66
30%	3.88
25%	**3.99**



ADDENDUM F – INPUT DATA

Simulated building cooling load, ambient dry-bulb outside temperature and electrical energy tariffs are included on the attached CD.

ADDENDUM G – ALGORITHM FOR SYSTEM WITH NO TES

```
 % Read External Data
Data = 'Jan.xlsx';% Data = 'Feb.xlsx';% Data = 'Mar.xlsx';% Data =
'Apr.xlsx';% Data = 'May.xlsx';% Data = 'Jun.xlsx';% Data =
'Jul.xlsx';% Data = 'Aug.xlsx';% Data = 'Sep.xlsx';% Data =
'Oct.xlsx';% Data = 'Nov.xlsx';% Data = 'Dec.xlsx';

% Create Variable Arrays
%%Input Variables
QL = xlsread(Data,'E2:E739'); %Building Cooling Load
Tdb = xlsread(Data,'F2:F739'); %Dry-Bulb Temp
k = xlsread(Data,'D2:D739'); %Instant

e = xlsread(Data,'G2:G739'); %Energy Cost
d = xlsread(Data,'H2:H739'); %Demand Cost

%Set Constants
Dt = 1;
SCAP = 1693;
N = length(k);
CCAPice = 142;
CCAPchw = 192.6;
Trefice = 25;
Trefchw = 35;
Tevapice = -6;
Tevapchw = 6;
DT = 3;

%%Decision Variables
u = zeros(length(QL),1);

%%State Variables
x = zeros(length(k),1); % State of Ice Tand charge
CCAP = zeros(length(k),1); %Chiller capacity
Q = zeros(length(k),1); %Load on Chiller
P = zeros(length(k),1); %Chiller energy consumption
COP = zeros(length(k),1); %Coefficient of Performance
PLR = zeros(length(k),1); %Part Load Ratio
COPplr = zeros(length(k),1);
Tu = zeros(length(k),1); %Upper Temp
Tl = zeros(length(k),1); %Lower Temp
COPtrue = zeros(length(k),1);

%Cost Variables
```

```
E = zeros(length(k),1); %Total Energy Cost per Instant
D = zeros(length(k),1); %Total Demand Cost per Instant

%Initial Conditions
x(1,1) = 1;

for R = 1:N
    Q(R,1) = QL(R,1); %The chiller supplies cooling load
    CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefchw- Tdb(R,1)));
    PLR(R,1) = Q(R,1)/CCAP(R,1);
    COPplr(R,1) = -1.6011*PLR(R,1)+4.344;

    Tl(R,1) = Tevapchw;
    Tu(R,1) = Tdb(R,1) + DT;

    COPtrue(R,1) = (Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefchw/(Trefchw-
Tevapchw));
    COP(R,1) = COPtrue(R,1) * COPplr(R,1); %Coefficient of Performance

    %Total chiller power consumption
    P(R,1) = Q(R,1)/COP(R,1);

    %Cost equations
    E(R,1) = P(R,1) * e(R,1);
    D(R,1) = P(R,1) * d(R,1);
end

%Objective Function
J = sum(E)+max(D)
```

ADDENDUM H – CONVENTIONAL CONTROL ALGORITHM (SUMMER)

```
% Read External Data
Data = 'Jan.xlsx';
% Data = 'Feb.xlsx';
% Data = 'Nov.xlsx';
% Data = 'Dec.xlsx';

% Create Variable Arrays
%%Input Variables
QL = xlsread(Data,'E2:E739'); %Building Cooling Load
Tdb = xlsread(Data,'F2:F739'); %Dry-Bulb Temp
k = xlsread(Data,'D2:D739'); %Instant

e = xlsread(Data,'G2:G739'); %Energy Cost
d = xlsread(Data,'H2:H739'); %Demand Cost

%Set Constants
Dt = 1;
SCAP = 1693;
N = length(k);
CCAPice = 142;
CCAPchw = 192.6;
Trefice = 25;
Trefchw = 35;
Tevapice = -6;
Tevapchw = 6;
DT = 3;

%%Decision Variables
u = zeros(length(QL),1);

%%State Variables
x = zeros(length(k),1); % State of Ice Tank charge
CCAP = zeros(length(k),1); %Chiller capacity
Q = zeros(length(k),1); %Load on Chiller
P = zeros(length(k),1); %Chiller energy consumption
COP = zeros(length(k),1); %Coefficient of Performance
PLR = zeros(length(k),1); %Part Load Ratio
COPplr = zeros(length(k),1);
Tu = zeros(length(k),1); %Upper Temp
Tl = zeros(length(k),1); %Lower Temp
COPtrue = zeros(length(k),1);
```

```
%Cost Variables
E = zeros(length(k),1); %Total Energy Cost per Instant
D = zeros(length(k),1); %Total Demand Cost per Instant

%%Constraints
U1 = zeros(length(k),1);
U0 = zeros(length(k),1);
Umin = zeros(length(k),1);
Umax = zeros(length(k),1);

%Initial Conditions
x(1,1) = 1;

for R = 1:N
    % Conventional control strategy of summer months
    if QL(R,1)== 0; %
        if x(R,1) < 1;
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            U1(R,1) = SCAP*(1-x(R,1));
            Umax(R,1) = min((CCAP(R,1)-QL(R,1)),U1(R,1));
            u(R,1) = Umax(R,1)/2;
            Q(R,1) = u(R,1) - QL(R,1);

            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

        elseif x(R,1) >= 1;
            u(R,1) = 0;
            Q(R,1) = u(R,1) - QL(R,1);
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge
        end

    elseif QL(R,1) >= CCAPchw/2;
        Q(R,1) = CCAPchw/2;
        u(R,1)=  Q(R,1) - QL(R,1);
        CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefice- Tdb(R,1)));
        x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

    else 0 < QL(R,1) < CCAPchw/2;
        u(R,1) = 0;
        Q(R,1) = QL(R,1) + u(R,1);
        CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefice- Tdb(R,1)));
```

```
        x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge
    end;

    if u(R,1) >= 0;
        M = 1;
    elseif u(R,1) < 0;
        M = 0;
    end;
    %Coefficient-of-Performance
    PLR(R,1) = Q(R,1)/CCAP(R,1);
    COPplr(R,1) = (-1.6011*PLR(R,1)+4.282)*M + (-
1.6011*PLR(R,1)+4.344)*(1-M);
    Tl(R,1) = Tevapice*M + Tevapchw*(1-M);
    Tu(R,1) = Tdb(R,1) + DT;
    COPtrue(R,1) = ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefice/(Trefice-
Tevapice)))*M + ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefchw/(Trefchw-
Tevapchw)))*(1-M);
    COP(R,1) = COPtrue(R,1) * COPplr(R,1);

    P(R,1) = Q(R,1)/COP(R,1); %Total chiller power consumption

    %Cost equations
    E(R,1) = P(R,1) * e(R,1);
    D(R,1) = P(R,1) * d(R,1);
end

%Objective Function
J = sum(E)+max(D)
```

ADDENDUM I – CONVENTIONAL CONTROL ALGORITHM (INTERMEDIATE)

```
% Read External Data
% Data = 'Mar.xlsx';% Data = 'Apr.xlsx';% Data = 'Sep.xlsx';
Data = 'Oct.xlsx';

% Create Variable Arrays
%%Input Variables
QL = xlsread(Data,'E2:E739'); %Building Cooling Load
Tdb = xlsread(Data,'F2:F739'); %Dry-Bulb Temp
k = xlsread(Data,'D2:D739'); %Instant

e = xlsread(Data,'G2:G739'); %Energy Cost
d = xlsread(Data,'H2:H739'); %Demand Cost

%Set Constants
Dt = 1;
SCAP = 1693;
N = length(k);
CCAPice = 142/2; %only single chiller used for ice build in winter
CCAPchw = 192.6/2; %only single chiller used for cooling load in winter
Trefice = 25;
Trefchw = 35;
Tevapice = -6;
Tevapchw = 6;
DT = 3;

%%Decision Variables
u = zeros(length(QL),1);

%%State Variables
x = zeros(length(k),1); % State of Ice Tand charge
CCAP = zeros(length(k),1); %Chiller capacity
Q = zeros(length(k),1); %Load on Chiller
P = zeros(length(k),1); %Chiller energy consumption
COP = zeros(length(k),1); %Coefficient of Performance
PLR = zeros(length(k),1); %Part Load Ratio
COPplr = zeros(length(k),1);
Tu = zeros(length(k),1); %Upper Temp
Tl = zeros(length(k),1); %Lower Temp
COPtrue = zeros(length(k),1);

%Cost Variables
```

```
E = zeros(length(k),1); %Total Energy Cost per Instant
D = zeros(length(k),1); %Total Demand Cost per Instant

%%Constraints
U1 = zeros(length(k),1);
U0 = zeros(length(k),1);
Umin = zeros(length(k),1);
Umax = zeros(length(k),1);

%Initial Conditions
x(1,1) = 1;

for R = 1:N
    % Conventional control strategy.
    if QL(R,1)== 0; %
        if x(R,1) < 1;
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            U1(R,1) = SCAP*(1-x(R,1));
            Umax(R,1) = min((CCAP(R,1)-QL(R,1)),U1(R,1));
            u(R,1) = Umax(R,1);
            Q(R,1) = u(R,1) - QL(R,1);

            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

        elseif x(R,1) >= 1;
            u(R,1) = 0;
            Q(R,1) = u(R,1) - QL(R,1);
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge
        end

    elseif QL(R,1) >= CCAPchw/2; % 1/2 of single chiller capacity
        Q(R,1) = CCAPchw/2; % 1/2 of single chiller capacity
        u(R,1)=  Q(R,1) - QL(R,1);
        CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefchw- Tdb(R,1)));
        x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

    else
        u(R,1) = 0;
        Q(R,1) = QL(R,1) + u(R,1);
        CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefchw- Tdb(R,1)));
        x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge
```

```
    end;

    if u(R,1) >= 0;
        M = 1;
    elseif u(R,1) < 0;
        M = 0;
    end;
    %Coefficient-of-Performance
    PLR(R,1) = Q(R,1)/CCAP(R,1);
    COPplr(R,1) = (-1.6011*PLR(R,1)+4.282)*M + (-
1.6011*PLR(R,1)+4.344)*(1-M);
    Tl(R,1) = Tevapice*M + Tevapchw*(1-M);
    Tu(R,1) = Tdb(R,1) + DT;
    COPtrue(R,1) = ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefice/(Trefice-
Tevapice)))*M + ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefchw/(Trefchw-
Tevapchw)))*(1-M);
    COP(R,1) = COPtrue(R,1) * COPplr(R,1);

    P(R,1) = Q(R,1)/COP(R,1); %Total chiller power consumption

    %Cost equations
    E(R,1) = P(R,1) * e(R,1);
    D(R,1) = P(R,1) * d(R,1);
end

%Objective Function
J = sum(E)+max(D)
```

ADDENDUM J – CONVENTIONAL CONTROL ALGORITHM (WINTER)

```
% Read External Data
% Data = 'May.xlsx';% Data = 'Jun.xlsx';% Data = 'Jul.xlsx';
Data = 'Aug.xlsx';

% Create Variable Arrays
%%Input Variables
QL = xlsread(Data,'E2:E739'); %Building Cooling Load
Tdb = xlsread(Data,'F2:F739'); %Dry-Bulb Temp
k = xlsread(Data,'D2:D739'); %Instant
h = xlsread(Data,'C2:C739'); %Hour of day

e = xlsread(Data,'G2:G739'); %Energy Cost
d = xlsread(Data,'H2:H739'); %Demand Cost

%Set Constants
Dt = 1;
SCAP = 1693;
N = length(k);
CCAPice = 142/2; %only single chiller used for ice build in winter
CCAPchw = 192.6/2; %only single chiller used for cooling load in winter
Trefice = 25;
Trefchw = 35;
Tevapice = -6;
Tevapchw = 6;
DT = 3;

%%Decision Variables
u = zeros(length(QL),1);

%%State Variables
x = zeros(length(k),1); % State of Ice Tand charge
CCAP = zeros(length(k),1); %Chiller capacity
Q = zeros(length(k),1); %Load on Chiller
P = zeros(length(k),1); %Chiller energy consumption
COP = zeros(length(k),1); %Coefficient of Performance
PLR = zeros(length(k),1); %Part Load Ratio
COPplr = zeros(length(k),1);
Tu = zeros(length(k),1); %Upper Temp
Tl = zeros(length(k),1); %Lower Temp
COPtrue = zeros(length(k),1);
```

```
%Cost Variables
E = zeros(length(k),1); %Total Energy Cost per Instant
D = zeros(length(k),1); %Total Demand Cost per Instant

%%Constraints
U1 = zeros(length(k),1);
U0 = zeros(length(k),1);
Umin = zeros(length(k),1);
Umax = zeros(length(k),1);

%Initial Conditions
x(1,1) = 1;
A = [0 1 2 3 4 5 22 23]'; %Hourly instants to build ice

for R = 1:N
    % Conventional control strategy.
    if QL(R,1)== 0 && any(A == h(R,1)); %
        if x(R,1) < 1;
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            U1(R,1) = SCAP*(1-x(R,1));
            Umax(R,1) = min((CCAP(R,1)-QL(R,1)),U1(R,1));
            u(R,1) = Umax(R,1);
            Q(R,1) = u(R,1) - QL(R,1);

            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

        elseif x(R,1) >= 1;
            u(R,1) = 0;
            Q(R,1) = u(R,1) - QL(R,1);
            CCAP(R,1) = CCAPice*(1 + 0.005*(Trefice- Tdb(R,1)));
            x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge
        end

    else
        Q(R,1) = 0;
        u(R,1)=  Q(R,1) - QL(R,1);
        CCAP(R,1) = CCAPchw*(1 + 0.005*(Trefchw- Tdb(R,1)));
        x(R+1,1) = x(R,1) + u(R,1)*(Dt/SCAP); %Ice storage
charge/discharge

    end;

    if u(R,1) >= 0;
        M = 1;
```

```
    elseif u(R,1) < 0;
        M = 0;
    end;
    %Coefficient-of-Performance
    PLR(R,1) = Q(R,1)/CCAP(R,1);
    COPplr(R,1) = (-1.6011*PLR(R,1)+4.282)*M + (-
1.6011*PLR(R,1)+4.344)*(1-M);
    Tl(R,1) = Tevapice*M + Tevapchw*(1-M);
    Tu(R,1) = Tdb(R,1) + DT;
    COPtrue(R,1) = ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefice/(Trefice-
Tevapice)))*M + ((Tu(R,1)/(Tu(R,1)-Tl(R,1)))/(Trefchw/(Trefchw-
Tevapchw)))*(1-M);
    COP(R,1) = COPtrue(R,1) * COPplr(R,1);

    P(R,1) = Q(R,1)/COP(R,1); %Total chiller power consumption

    %Cost equations
    E(R,1) = P(R,1) * e(R,1);
    D(R,1) = P(R,1) * d(R,1);
end

%Objective Function
J = sum(E)+ max(D)
```

ADDENDUM K – OPTIMAL CONTROL ALGORITHM

```
% Read External Data from Excel file
Data = 'Jan.xlsx';

% Create Variable Vectors
%%Input Variables
QL = xlsread(Data,'E2:E739')'; %Building Cooling Load
Tdb = xlsread(Data,'F2:F739')'; %Dry-Bulb Temp
e = xlsread(Data,'G2:G739')'; %Energy Cost
d = xlsread(Data,'H2:H739')'; %Demand Cost

%Set Constants
Dt = 1;
SCAP = 1693;
CCAPice = 142;
CCAPchw = 192.6;
Trefice = 25;
Trefchw = 35;
Tevapice = -6;
Tevapchw = 6;
DT = 3;

N = length(QL);

%%State Variables
CCAP = zeros(1,N); %Chiller capacity
Tu = zeros(1,N); %Upper Temp
Tl = zeros(1,N); %Lower Temp
COPtrue = zeros(1,N); %Coefficient of Performance

%%Decision Variables - Yalmip sdpvar
u = sdpvar(1,N);
lamda = sdpvar(1,1);

%%Constraint Variables
ct1 = []; % Ice Storage State of Charge Constraint
ct2 = []; % Rate of Charge/Discharge bound constraint
ct3 = []; % Chiller capacity constraint
ct4 = []; % Maximum demand cost constraint
ct5 = []; % Energy cost Constraint

F = set([]); %Open Constraint set Object
```

```
%Initial Conditions
x0 = 1;  ct5 = 0;

for R = 1:N
    if QL(1,R) == 0 % Ice build cycle
        CCAP(1,R) = CCAPice *(1 + 0.005*(Trefice - Tdb(1,R)));
        Tl(1,R) = Tevapice;
        Tu(1,R) = Tdb(1,R) + DT;
        COPtrue(1,R) = ((Tu(1,R)/(Tu(1,R)-Tl(1,R)))/(Trefice/(Trefice-
Tevapice))) ;

        ct1 = ct1+( 0 <= x0 + sum(u(1,1:R))/SCAP <= 1 ); %State of ice-
storage charge constraint
        ct2 = ct2+( 0 <= u(1,R) <= SCAP ); %For ice-build vector u
positive
        ct3 = ct3+( 0 <= QL(1,R) + u(1,R) <= CCAP(1,R) );  %Chiller
capacity constraint
        ct4 = ct4+( (QL(1,R) + u(1,R))/(COPtrue(1,R) * (-
1.6011*(QL(1,R)+u(1,R))/CCAP(1,R)+4.282))*d(1,R) <= lamda); %Maximu
demand constraint
        ct5 = ct5+((QL(1,R) + u(1,R))/(COPtrue(1,R) * (-
1.6011*(QL(1,R)+u(1,R))/CCAP(1,R)+4.282)))*e(1,R); %Energy cost
constraint
    else % Chilled water cycle
        CCAP(1,R) = CCAPchw *(1 + 0.005*(Trefchw - Tdb(1,R)));
        Tl(1,R) = Tevapchw;
        Tu(1,R) = Tdb(1,R) + DT;
        COPtrue(1,R) = (Tu(1,R)/(Tu(1,R)-Tl(1,R)))/(Trefchw/(Trefchw-
Tevapchw));

        ct1 = ct1+( 0<=  x0 + sum(u(1,1:R))/SCAP <= 1 ); %State of ice-
storage charge constraint
        ct2 = ct2+( -SCAP <= u(1,R) <= 0 ); %For chilled water vector u
negative
        ct3 = ct3+( 0 <= QL(1,R) + u(1,R) <= CCAP(1,R) ); %Chiller
capacity constraint
        ct4 = ct4+((QL(1,R) + u(1,R))/(COPtrue(1,R) *(-
1.6011*(QL(1,R)+u(1,R))/CCAP(1,R)+4.344))*d(1,R) <= lamda); %Maximum
demand constraint
        ct5 = ct5+((QL(1,R) + u(1,R))/(COPtrue(1,R) *(-
1.6011*(QL(1,R)+u(1,R))/CCAP(1,R)+4.344)))*e(1,R); %Energy cost
constraint
    end;
end

Objfun = ct5 + lamda; %Objective function

F = ct1 + ct2 + ct3 + ct4; %Constraints function
```

```
ops =
sdpsettings('solver','IPOPT','verbose',1,'showprogress',1,'warning',1);
%Yalmip solver options

tic, solvesdp(F, Objfun, ops); toc
```

ADDENDUM L – ALGORITHM FLOW DIAGRAM