Energy-based Control in HVAC Systems in Commercial Buildings to Optimize the Building Energy Performance

Sandali Rukshila Walgama

(198081T)

Degree of Master of Philosophy

Department of Electrical Engineering

University of Moratuwa
Sri Lanka

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DECLARATION

I declare that this is my own work and this thesis does not incorporate without acknowledgement any material previously submitted for a Degree or Diploma in any other University or institute of higher learning and to the best of my knowledge and belief it does not contain any material previously published or written by another person except where the acknowledgement is made in the text.

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Prof. Sisil Kumarawadu

Prof. Chandima Dedduwa Pathirana

Abstract

The severity of the emerging environmental problems and the criticality of a future energy crisis with associated socio-economic quandaries are becoming strengthen along with the inevitable growth of energy demand due to the ever-increasing population and urbanization. Inspired by this, many state-of-the-art and efficient engineering advancements under demand side management are being conceptualized, developed, and examined via worldwide research for several years.

As one of the largest global energy consumers, buildings, and their Heating, Ventilating and Air Conditioning (HVAC) systems acquire substantial attention regarding optimal and efficient energy management. Nevertheless, longer computational time, complicated implementation, deficient occupants' thermal comfort and productivity, and limited practicability and feasibility of most of the approaches that have been proposed, have demotivated the industry applications even though they offer consequential designing and operating advancements to the system. Therefore, the expected building energy performance can be further extended by unraveling these encountered disputes with novel energy saving strategies.

This research concentrates on energy optimal operation of HVAC systems by addressing secondary chilled water pumping system, temperature setpoint management, cooling water system, and temperature controllers in Air Handling Unit (AHU). Three different innovative energy saving strategies to determine the optimal number of chilled water pumps to be operated with their optimal speed, optimal zone temperature setpoint schedule, and optimal cooling water flow rate with a setpoint of the chilled water supply temperature have been proposed along with the consideration of system constraints, safety, occupant thermal comfort, and satisfaction. In addition, a novel temperature controller that can be utilized in AHU has been introduced and the performance in comparison with available controllers has been studied. Simulation results obtained via the case studies authenticate the effectiveness of the introduced approaches and encourage the functioning of these strategies in real engineering systems due to the inherent simplicity, robustness, and less computational complexity.

Keywords- **Chilled Water System, Energy Optimization, HVAC, Parallel Pumps, Variable Speed, Dijkstra, Energy Management, Optimal Setpoint, Productivity, Thermal Comfort, Cooling Water System, AHU, Elliot Function, Temperature Control**

DEDICATION

To my family,

for always loving and supporting me...

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Sandali Rukshila Walgama

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LIST OF ABBREVIATIONS

AHU	Air Handling Unit
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BAS	Building Automation Systems
BEP	Best Efficiency Point
BMS	Building Management Systems
CAV	Constant Air Volume
COP	Coefficient of Performance
CW	Cooling Water (Condenser Water)
FCM	Fuzzy Cognitive Map
FCU	Fan Coil Unit
FL	Fuzzy Logic
GA	Genetic Algorithm
HSS	Hydronic System Solutions
HVAC	Heating, Ventilating and Air Conditioning
ISO	International Organization for Standardization
MDP	Markov Decision Process
MIMO	Multiple-Input and Multiple-Output
MINLP	Mixed Integer Non-Linear Programming
MPC	Model Predictive Control
NN	Neural Network
OPC	Open Platform Communications
PD	Proportional Derivative
PID	Proportion Integration Differentiation

PMV	Predicted Mean Vote
PPD	Predicted Percentage Dissatisfied
PSO	Particle Swarm Optimization
RP	Relative Productivity
RTP	Real Time Pricing
SQL	Structured Query Language
TCP	Transmission Control Protocol
U.S.	United States
VAV	Variable Air Volume
VFD	Variable Frequency Drives

Chapter 1

INTRODUCTION

Due to the exponentially growing population and continued urbanization, energy demand is escalating day by day. This is causing many environmental problems such as air pollution, water pollution, thermal pollution, global warming, and climate change. With the unrestricted energy consumption, these impacts getting more severe than that of the current situation. On the other hand, we will have to face an energy crisis soon because of the limited natural resources. Therefore, how can we contribute to avoiding these unfortunate situations?

From the consumer perspective, demand-side energy management is the most effective solution where we can actively engage. This includes multiple actions designed for the management and optimization of a site's energy consumption. Aiming on diminishing these problematic situations, numerous researches are

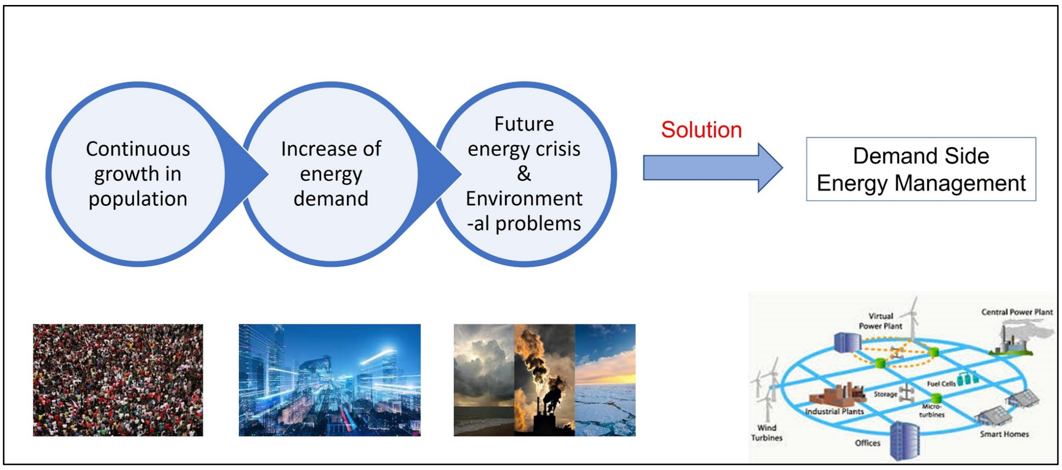


Figure 1.1: Significance of Demand Side Energy Management

being performed under the umbrella of demand side management.

1.1 Background

If the global energy consumption statistics are studied, the significant contribution from the commercial sector and its increasing tendency can be observed.

The buildings in the United States (U.S.) reported for 41% of the primary energy usage in 2010 [1] and are responsible for 40% of the greenhouse gas emissions [2, 3]. According to the recent energy reviews, the commercial sector consumes 18% of the total U.S. energy as depicted in Figure 1.2. Based on the growing inclination of this consumption as illustrated in Figure 1.3, we can expect a large share of upcoming energy usage from the building sector. Here, offices, schools, hotels, malls, hospitals, stores, places of worship, restaurants, warehouses, and public assembly are included in the commercial sector.

Heating, Ventilating and Air Conditioning (HVAC) systems, elevators, lighting loads, and other appliances are the main energy consumers in commercial

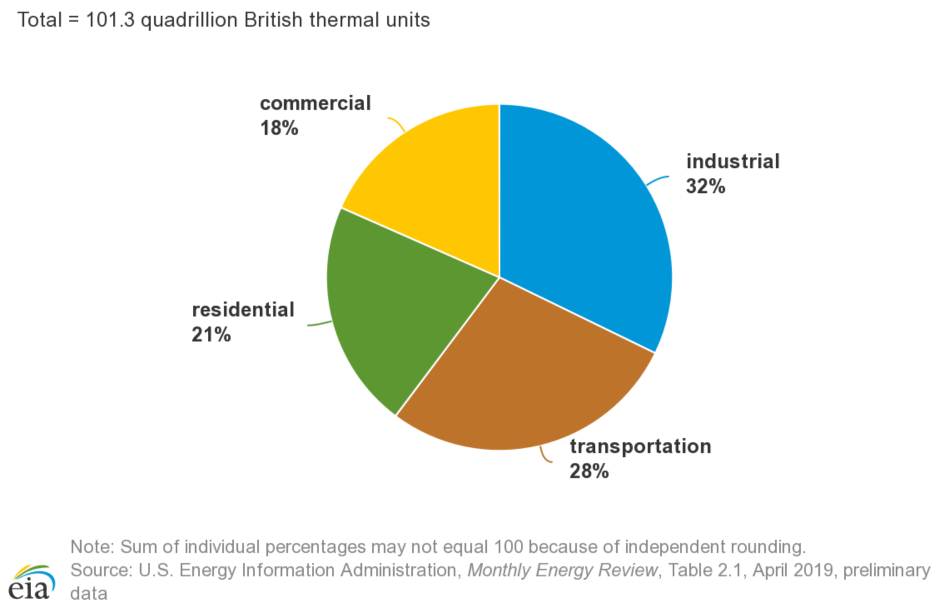


Figure 1.2: Share of Total U.S. Energy Consumption by End-use Sectors, 2018 [4]

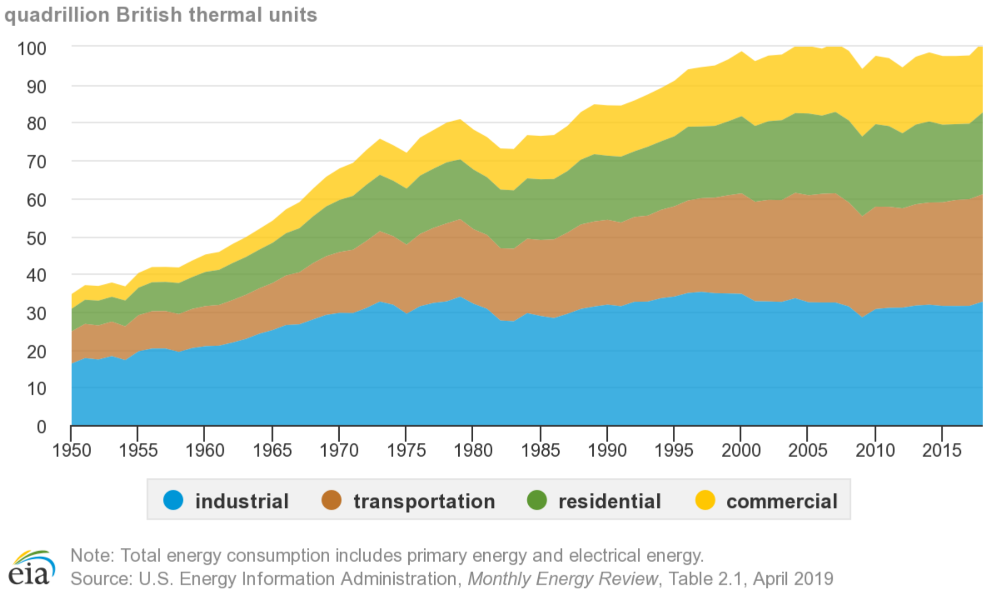


Figure 1.3: U.S. Total Energy Consumption by End-use Sector, 1950 - 2018 [4]

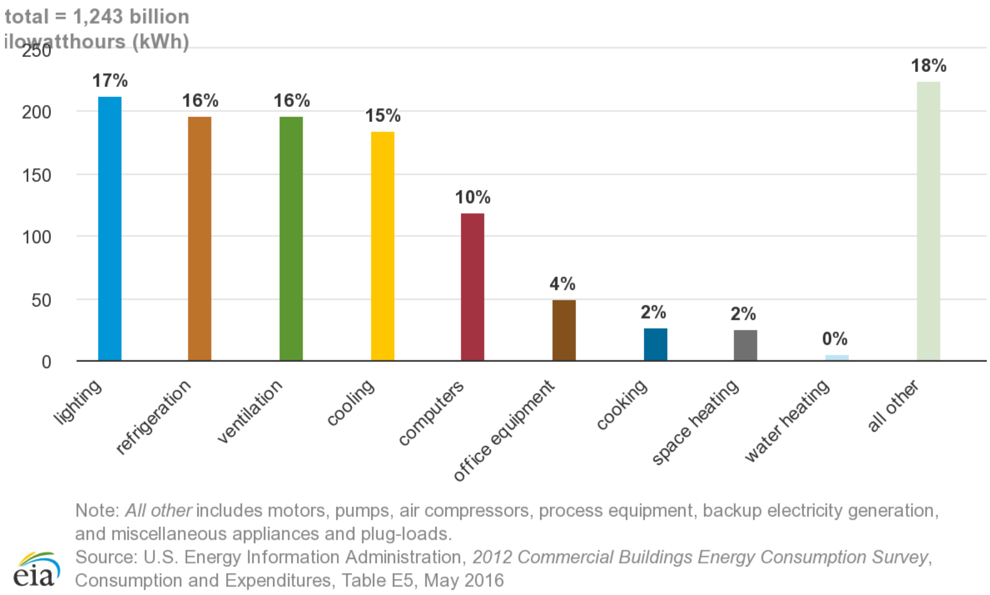


Figure 1.4: Electricity Use in U.S. Commercial Buildings by Major End Uses, 2012 [4]

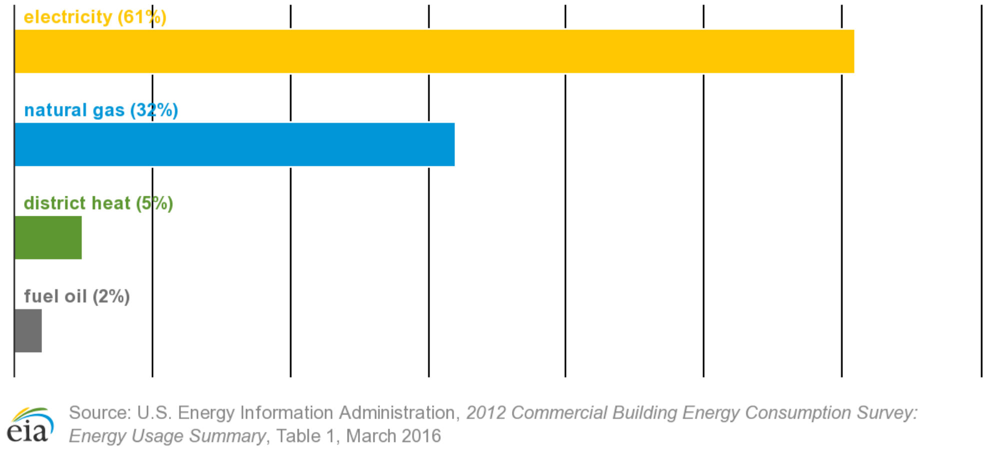


Figure 1.5: Share of Major Energy Sources Used in U.S. Commercial Buildings, 2012 [4]

buildings. Among them, HVAC systems consume more than any other energy user in buildings. Studies have proved that it constitutes over 50% of the building energy consumed in the U.S. [5]. This is also presented in Figure 1.4.

According to Figure 1.5, electricity is the major energy source in commercial buildings while as much as half of it is allocated for HVAC systems.

Therefore, energy optimal strategies for HVAC systems in commercial buildings are joining hands for achieving the ultimate goal of demand side management along with financial and environmental benefits.

1.2 Problem Statement

Regarding the energy optimal pump operation in complex air conditioning systems, much of the literature has focused on either speed control [6, 7] or sequence control [8, 9] of pumps. Though optimal deployment plan and pump speed optimization both together is of utmost importance to guarantee maximum energy saving, existing literature and related work on considering both together are still inadequate. The reduced complexity of the control algorithms is also a major concern to make the best match for real engineering applications. Furthermore, a lack of attention given to the actual problems and possible threats such as cavitation of the serial-parallel systems in the optimization procedure [10, 11] can be noticed.

A compromise between energy consumption and occupant comfort often appears in the building sector. The influence of adjusting the setpoints and widening the dead band on either energy use [12–14] or thermal comfort [15] has been examined through previous research efforts. Nevertheless, there is a dearth of research studying the synchronous consequences of optimal temperature setpoints on HVAC energy consumption, the cost for energy consumption, occupants' thermal comfort, and productivity [16, 17]. The requisite of substantial computational resources for model development and optimization is the main disadvantage of complex control strategies such as Model Predictive Control (MPC), although they exhibit good performance [18]. Therefore, the use of simpler control algorithms is more economical and ensures ease of implementation.

Because of the negative impacts on the performance of other system components caused by local optimizations, their results have been limited to the literature rather than being beneficial to actual Cooling Water (CW) systems. For instance, when the frequencies of the CW pump and cooling tower fan are varied, the return temperature of CW will be affected, and hence this will influence the chiller performance. Therefore, simplified models with low complexity algorithms-based holistic approaches are required. In addition, robustness, reliability, easiness to execute in existent engineering applications, and capability of assuring dynamic cooling loads while resulting in minimal energy consumption should be accomplished by a prospective control scheme.

The superior performance of the temperature controllers in the Air Handling Unit (AHU) assures the occupants' comfortability as well as healthiness. Besides that, unless the flow of heating or cooling agent is properly regulated, there would be a massive energy wasting and that also would append dispensable outlays for re-heating or re-cooling. Research studies on other alternatives to overcome existing performance issues such as hunting operation, the possibility of undesired operation conditions, and longer execution time, will highly contribute to enhancing the HVAC and building energy performance.

1.3 Research Objectives

The main objective of this research is to propose novel energy conserving strategies for HVAC systems in smart commercial buildings to enhance building energy performance. This will be achieved by accomplishing the specific objectives as follows.

- To identify the energy saving capability of parallel chilled water pumps and to suggest a novel pump control method such that energy usage is reduced.

This research aims to develop a new low complex algorithm to operate and control HVAC parallel chilled water pumps by selecting the optimal number of pumps to be operated and their operating speed that consume minimal total power to meet the variable chilled water flow requirement along with a safe and energy-efficient operation.

- To introduce an innovative method of optimizing zone temperature setpoints to obtain energy and cost benefits while assuring occupants' thermal comfort.

In this research, a novel approach based on the Dijkstra method is developed for optimally scheduling the temperature setpoints in order to achieve cost and energy savings while maintaining comfort indices and productivity in the acceptable range.

- To propose a novel method to minimize the consumed energy by CW system.

The optimizing method that is presented in this research, considers outdoor ambient and wet bulb temperature variations and decides the optimal hourly CW flow rate such that hourly cooling load requirement is fulfilled whilst causing minimal power consumption of the cooling system.

- To identify efficient temperature controllers which can be used in AHUs.

As one of the outcomes of this research, a modified Elliot function based controller which can be applied for regulating supply air temperature by adjusting the 2-way control valve opening of the heat exchanger coil in AHU is introduced. Furthermore, this study analyses its performance for heating mode operation by comparing it with Proportional Derivative (PD), Fuzzy Logic (FL), and Sigmoid function-based controllers.

1.4 Thesis Overview

This thesis consists of seven chapters, and they are briefly described below.

Chapter 1: Introduces the thesis. It discusses the background and the problem statement that inspired this research. Then, the specific objectives addressed by this research work to achieve the main research objective are stated.

Chapter 2: Provides a literature review on target systems and past research efforts that are focused on solving similar problems.

Chapter 3: Presents the proposed strategy to curtail the power usage of HVAC secondary chilled water pumps. Through the simulation case studies, its effectiveness is validated in the latter part of the chapter.

Chapter 4: Describes the proposed temperature setpoint management approach that offers energy and cost savings while assuring occupants' comfort and productivity. The later part of the chapter evaluates the results obtained from the simulation studies.

Chapter 5: Illustrates the suggested energy conserving method for HVAC cooling water systems. The later part of the chapter reveals the performance results with the help of simulation studies.

Chapter 6: Carries the performance analysis of the introduced temperature controller in comparison with other temperature controllers that are used in AHUs.

Chapter 7: Concludes the thesis by discussing the integration of the proposed strategies with the Building Management Systems (BMS), possible limitations, and recommendations for future developments.

Chapter 2

LITERATURE REVIEW

2.1 Heating, Ventilating and Air Conditioning Systems

2.1.1 Overview

HVAC systems are utilized in buildings to create an indoor environment comfortable for the occupants. Therefore, this system is responsible for total control of temperature, humidity, air movement in the occupied space, filtration of airborne particles, and supply of outside air for ventilation. In order to achieve this, seven main processes collaborate in the system as represented in Table 2.1.

However, depending on the objective criteria or the situation, the requirements of the processes may be varied and that is an important factor to be considered when determining optimal strategies to enhance building energy performance.

2.1.2 Central Plants

Chilled water makes used as a cooling medium for this type of HVAC system. These structures contain systems with chillers and cooling towers for heat rejection. Normally, a boiler is used to generate the heating and that heat is distributed via hot water or steam piping [20]. This set of equipment can fulfill the heating and cooling needs of any type of large building. Although there are both pros and cons as tabulated in Table 2.2, commercial buildings also often

Table 2.1: Main Processes in HVAC Systems [19]

Process Name	**Process**	**Purpose**
Heating	Adding thermal energy (heat) to the conditioned space	Raising or maintaining the temperature of the space
Cooling	Removing thermal energy (heat) from the conditioned space	Lowering or maintaining the temperature of the space
Humidifying	Adding water vapor (moisture) to the air in the conditioned space	Raising or maintaining the moisture content of the air
Dehumidifying	Removing water vapor (moisture) from the air in the conditioned space	Lowering or maintaining the moisture content of the air
Cleaning	Removing particulates (e.g. dust) and biological contaminants (e.g. insects) from the air delivered to the conditioned space	Improving or maintaining the air quality
Ventilating	Exchanging air between the outdoors and the conditioned space	Diluting the gaseous contaminants in the air and improving or maintaining air quality, composition & freshness
Air Movement	Circulating and mixing air through conditioned spaces in the building	Achieving the proper ventilation and facilitating the thermal energy transfer

prefer these systems among others [19, 21].

Table 2.2: Advantages and Disadvantages of a Central Plant [19, 21]

Advantages	**Disadvantages**
- Require less maintenance - Housed in a separate building which reduces noise and safety issues - Equipment usually has a longer life - Equipment can be monitored relatively easily - Easier to maintain a high standard of operation and maintenance - Can be extended to provide heating and cooling to many buildings - Heat recovery is financially worth while - Reduced starting in-rush current - Reduced power cost under part load conditions	- Initial cost is higher - Require installation floor area and space through the buildings for distribution pipes - Low seasonal efficiency - Higher replacement costs

2.1.3 Main Components

Figure 2.1 illustrates a schematic diagram of a central HVAC system with a water cooled chiller and the other main components.

The air comes from the space itself or outside by way of a return air system, is conditioned and supplied towards the space to be conditioned by the section called AHU. According to the selected control strategy, dampers are controlled to mix the air. When the temperature of the outdoor air is less than that of the return air, allowing a large percentage of outdoor air to mix with a small percentage of circulating return air is more economical. In the opposite situations also, even a small percentage of outdoor air has to be mixed in order to remove indoor contaminants such as carbon dioxide and continuously maintain fresh air ventilation. After the filtration, this mixed air is conditioned to the preferred temperature.

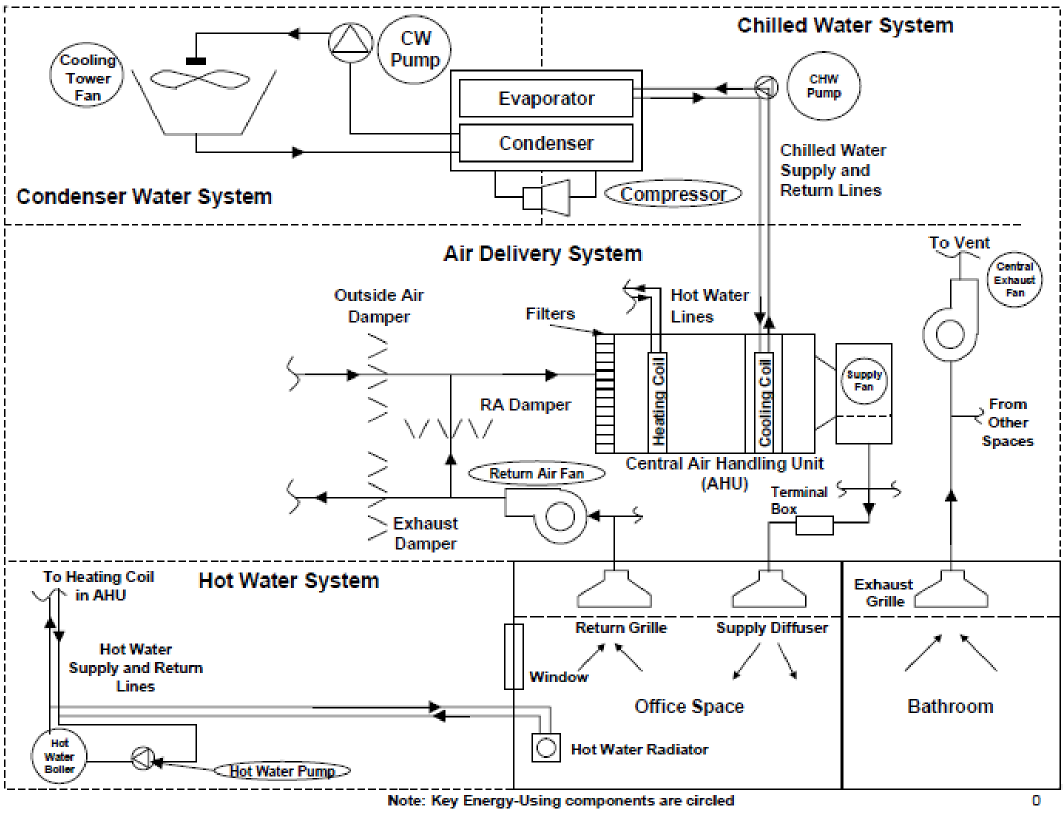


Figure 2.1: Schematic of a Central HVAC System with a Water Cooled Chiller [20]

Heat exchanger coils are useful for air heating and cooling. Steam or hot water and chilled water are supplied to the coils as a heat exchange medium for heating and cooling, respectively. When a Variable Air Volume (VAV) system is available, delivered airflow to the conditioned space is controlled by a terminal box that has a modulating valve. Finally, the air is distributed to the space via a diffuser, which mixes the supply air and the room air. If there is a reheat coil in the terminal box it can provide additional heat when the space requires not so much cooling as given by the supply air even at the minimum air quantity setting of the terminal box. It may comprise of a fan also. In the case of Constant Air Volume (CAV) systems, air delivery rates are not reduced and reheat coils are used to control the supplied cooling. However, in many applications, these systems are not allowed by energy codes. The leaving air goes out from the conditioned space by way of either the return system or the exhaust system.

To meet the cooling needs of all the AHUs in the building, required chilled water is provided by the chilled water system. Chilled water is circulated across the evaporator area of the chiller and the building by chilled water pumps. There may be primary and secondary chilled water pumps also in the system. The primary pumps establish persistent chilled water flow across the chiller, meanwhile, the secondary pumps supply only upon the demand of the building. The chiller can be found as a packaged vapor compression cooling system that takes in heat from the incoming chilled water and gives out heat either to condenser water or to ambient air relying on the water-cooled or air-cooled type of chiller. In a water-cooled chiller as displayed in Figure 2.1, the condenser water flows through the chiller's condenser and the cooling tower with the help of another water pump. At the cooling tower, heat is released to nature as a result of direct heat exchange between the condenser water and cooling air. In an air-cooled chiller, the air is moved across the condenser coil by condenser fans.

2.2 Chilled Water Pumping Systems and Energy Saving

Pumping systems in the central air-conditioning systems expend higher than 20% of the overall power utilized by building HVAC system [8]. This fact indicates the significance of introducing energy optimal strategies for these pumping systems.

In central cooling systems, the implementation of a group of pumps rather single pump is beneficial with regard to enhancing flexibility and system efficiency [8, 22]. Also, concerning the ability to avert production loss during maintenance and to compensate flow requirements, small capacity pumps connected in parallel are more advantageous than a single large unit pump [23]. Through the comparative analysis done by S. Shiels et al. [24], the pros and cons offered by running a single pump and dual pumps have been presented. Hydraulic characteristics, reliability, operating costs, capital investment, and safety are the factors taken into account in this study.

A constant speed that occurs efficient pump performance is used commonly, even though this wastes much energy with the varying loads. To overcome this, staging and de-staging of subsequent pumps are done according to the flow demand, as a traditional pump control method. Nevertheless, once a pump turns on, it reaches the rated value ignoring the pump efficiency [25]. Inspired by the affinity laws, pumps can be made energy optimal by improving variable speed capability with the introduction of Variable Frequency Drives (VFD). However, this flexibility can be led to poor system efficiency when the pump operates distant from its Best Efficiency Point (BEP) [26]. Also, the maximum reliability of a pump can be reached at BEP, and deviating the flow rate from that point causes to decrease in the reliability [27]. Avoiding harmful operations and hence extending the pump lifetime can be convinced by controlling the pump operation in the domain of maximum efficiency [23]. As per W. Burke [28], functioning the pumps in the range of $\pm 20\%$ of their BEPs very rarely occurs operation issues. To circumvent radial thrust, operating each pump with a flow rate between 50% and 110% of the flow at BEP is beneficial [29]. E.G. Hansen [30] has discovered that the dissimilar operating speeds for chilled water pumps that are configured in parallel are useless.

Therefore, deciding an energy optimal deployment plan for the HVAC chilled water pumps along with lower power consumption and maintenance costs, higher system efficiency, and satisfying performance with safety is stimulating.

An adaptive and derivative method based online control strategy has been proposed by S. Wang et al. [6] for the optimization of condenser cooling water pump speeds. L. Szychta [7] has decided the speeds at optimal conditions by considering the correlation between the pump speed and efficiency dynamics. The efficiency of the cascade pumping system at each operating condition has been maintained at the maximum with the help of an optimum speed reference predictions-based control algorithm in the research work done by S. Umashankar et al. [23]. With a focus on determining the optimal control frequency for chilled water pumps and cooling towers, a load-based method for speed control has been developed by F. Yu et al. [31]. K. Ma et al. [32] have manipulated the frequency of chilled water pumps using a Proportion Integration Differentiation (PID) strategy with the differential temperature setpoint and deviations. Moreover, intelligent algorithms such as evolutionary algorithm [33], reinforcement learning algorithm [34] and Genetic Algorithm (GA) [35] have been utilized to acquire improved pump performance in the literature. Nevertheless, the importance of optimized pump deployment has been disregarded in these studies.

According to the intersection of flow and power lines at different pump frequencies, W. Lu et al. [9] have derived a method of obtaining the optimal number of operating pumps. Z. Ma et al. [8] have introduced a pump sequence optimization approach to lessen the power consumption and maintenance cost of the pumps with the use of pressure drop models for different water networks. However, further energy savings have been restricted since the optimization of pump speed has been discounted. As an intelligent method, GA has been used in the literature [36,37] to optimally schedule the pump station in a water supply system. Due to the requirement of operations such as chromosome encoding, selection, crossover, and mutation, the implementation of this complicated algorithm is more difficult.

L. Tillack et al. [38] have utilized wire-to-water efficiency of the pumping system and the pressure differential transmitters at critical loops for sequencing the pumps and speed control purposes, respectively. Nevertheless, system costs and maintenance may be increased due to the use of transmitters. Based on an analytical model with the characteristic of bypass-loop adjustment, L. Xuefeng et al. [39] have proposed a hybrid control strategy to combine the single pump variable frequency and pump quantity controls. Results illustrate its suitability for chilled water systems with high differential pressure or a low flow ratio. For a pump system comprising different types of pumps connected in parallel, the number of pumps and their speed have been optimized using dynamic programming [40]. However, dealing with many pumps makes this method complicated. The extreme value analysis method has been applied to obtain the optimized number and speed proportions of parallel variable frequency hydraulic pumps in the research work carried out by Z. Tianyi et al. [41]. The complexity of the method limits the actual applicability even though key requirements have been addressed. With an aim to minimize the power usage of parallel pump systems while improving the reliability of pumps, Z. Lai et al. [27] have adopted a Particle Swarm Optimization (PSO) algorithm to determine the optimal number of pumps and speed ratios. This method has shown superior performance through the results compared to the conventional PID speed regulation method and other intelligent algorithms. However, for a particular system, the parameters of the PSO algorithm should be carefully selected and the penalty coefficients are chosen by several trial and error.

According to the literature, either pump speed control [6,7] or pumps sequence control [8,9] of complex air conditioning systems has been mostly concentrated. The lack of studies on contributing both factors to harness optimal energy savings and possible practical issues due to the complexity of the control algorithms also can be emphasized. Another observable downside of some studies [10,11] is the insufficient attention given to the actual problems and possible threats such as cavitation and overheating that can be emerged in serial-parallel systems, when defining the problem constraints.

This motivated to direct the research focus on developing a novel low complex algorithm that can operate and control HVAC parallel chilled water pumps by determining the optimal number of pumps to be operated and their optimal speed such that reduction in energy usage, safety operation, higher efficiency, and variable flow requirements are well achieved.

2.3 Indoor Temperature Setpoint Management, Thermal Comfort and Energy Saving

Even though accomplishing energy efficiency and saving targets are major concerns in the HVAC systems nowadays, their responsibility on sustaining occupants' thermal comfort should not be disrupted at any time. This yields an inevitable complexity in the optimization problem with multiple constraints.

A thermal environment is a creature of different personal and environmental factors such as clothing, metabolic heat production, air temperature, mean radiant temperature, humidity, and relative airspeed. How far that satisfies the occupants' minds can be simply termed as thermal comfort [42]. With the thermal discomfort, expected performance cannot be achieved due to the productivity loss and negative impacts on physical and mental health. To assess the level of occupants' thermal comfort, different indices such as Predicted Mean Vote (PMV) [43], Predicted Percentage Dissatisfied (PPD) [44], and productivity have been introduced with the developments on Fanger's model [45]. This implies the necessity of integrating occupant comfort, health, and safety concerns in the energy optimal operation of HVAC systems.

Since the users' demand is altered with the thermal comfort, that is crucial in the automatic controls of HVAC systems [46]. For different temperature setpoints, the resulting actual indoor temperature and indoor thermal comfort are also different. At the same time, when the temperature setpoint is adjusted, the required cooling load and hence the power consumption of the HVAC system are also varied. In that case, temperature setpoint adjustment is a very important factor to maintain the balance between occupants' thermal comfort and HVAC energy usage as presented in Figure 2.2. Therefore, an effective and efficient strategy to optimize the temperature setpoints is particularly capable of enhancing the operational and energy performance of any building.

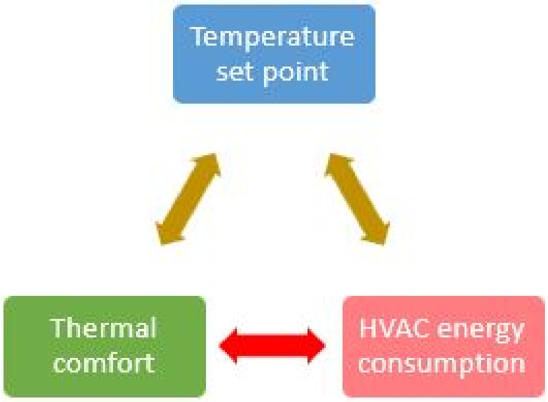


Figure 2.2: Importance of Optimal Temperature Setpoints

A similar study has been committed by J. Dejvises et al. [47] on an air conditioning system with the consideration of factors such as airflow, indoor temperature, thermal insulation of the building, and outdoor temperature using MATLAB® simulations. Nevertheless, energy saving has been attained by turning off the air conditioner within small intervals. S. Papadopoulos et al. [48] proposed a GA multi-objective optimization framework to obtain optimal temperature setpoint settings of the HVAC systems in commercial buildings. Although thermal comfort, energy consumption, and productivity have been addressed in the optimization problem, building operation costs have not been included and the granularity of decision variables is small since it offers annual setpoints. X. Zhang et al. [49] introduced a real-time decentralized temperature control strategy in order to accord energy saving and user comfort. Here, the steady-state optimization problem has been solved using a convex relaxation approach with an approximation of the thermal dynamic building systems model. Although the control strategy has been developed under approximations, the model's accuracy can be further increased. A Fan Coil Unit (FCU) temperature control has been proposed by C.M. Lin et al. [50] to conserve more energy in chilled water systems whilst improving the HVAC system's operating efficiency. When estimating the comfortable indoor temperature, this study has only considered the indoor thermal comfort and outdoor temperature.

F. Jazizadeh et al. [51] implemented an FL controls based thermal comfort prediction method with an objective to decrease the daily mean flow of the HVAC system. A non-linear MPC method has been examined in the applications of optimizing the low-load operation efficiency and sustaining indoor comfort without allowing the HVAC system to be operated at the high-load during the research carried out by M. Castilla et al. [52]. Another two MPC systems have been derived in the literature [53, 54] to obtain the ideal solution that assures indoor thermal comfort. Moreover, the authors were able to develop a chilled water system control concept by combining MPC models and dynamic thermal sensation. J.F. Nicol et al. [55] have discovered an adaptive thermal comfort model and fine-tuned the room temperature with the use of an FL system. These systems are capable of estimating indoor comfort and predicting energy consumption. However, real indoor heat demand has not been addressed in the above studies.

D. Nikovski et al. [56] have presented a procedure using model based control for building air conditioning systems to minimize the cost of energy whilst preserving occupant comfort. Here, building thermal dynamics have been decreased to a Markov Decision Process (MDP) and a series of temperature set-values during a selected horizon can be obtained as the result. Nonetheless, the influence of model inaccuracy and ambiguity in some input data such as outdoor temperature has not been taken into account. By altering the indoor temperature constraint and reducing the operating time, N.K. Kim et al. [57] have introduced a novel energy saving approach for HVAC systems. However, the involvement of the optimization happens only during the forecast peak time. N. Kampelis et al. [58] have utilized models of building energy, energy cost, and optimization together with a GA-based approach towards HVAC optimal temperature setpoints. Even though its aptitude for operational optimization of HVAC systems including day-ahead real-time pricing demand response programs has been evaluated, the significance of the incurred computational cost of the suggested method is an issue. M. Wani et al. [59] have analyzed the suitability of on-off control and economizer control strategies for HVAC energy optimization in terms of their performance and outcomes using EnergyPlus^{TM} building simulations. However, optimal temperature setpoints have not been incorporated into this research work.

The cooling setpoint temperature of the HVAC system in office buildings has been optimized with the aid of a derivation method in order to create a comfy indoor environment by J. Han et al. [15]. This study has considered the dynamics in thermal characteristics and weather conditions. However, this strategy provides daily setpoint temperatures and energy conservation has not been focused on. J. Park et al. [60] have innovated a thermal comfort-based controller for bringing down the energy usage for cooling purposes. Nevertheless, further detailed analysis and experimentation of the sensor configuration are required for the actual applications. By integrating multiple real-time condition input variables, S. Krinidis et al. [61] applied a multi-criteria deterministic algorithm to minimize the energy consumption of the HVAC system and to maximize the building occupants' satisfaction with the comfort level. This approach requires a multi-sensorial network and uses normalized values to examine thermal comfort.

As per the reviewed literature, a significant number of research works have studied the impact of setpoint adjustments and dead band widening on either thermal comfort [15] or energy use [12–14]. Moreover, the lack of research studies on the simultaneous outcomes of temperature setpoint optimization on energy consumption of the HVAC system, the cost of energy, thermal comfort, and productivity [16, 17] can also be emphasized. Complex control strategies such as MPC [18] are not much popular in this area since they entail considerable computational resources for modeling and optimization. Therefore, the simplicity of the control algorithm assures the ease of implementation and makes the approach more economical.

To overcome the stated problems, the suitability of the Dijkstra technique is evaluated in this research. Along with guaranteed acceptable PMV, PPD and Relative Productivity (RP) [62], the proposed method intended to reduce both energy consumption and cost of energy of the HVAC system. This is achieved by optimally scheduling the day ahead temperature setpoints considered by an AHU.

2.4 Cooling Water System and Energy Saving

CW systems are accountable for nearly 20% of the HVAC system's energy consumption [63]. This stimulates the involvement of energy saving methodologies in these systems since it offers both financial and environmental benefits.

As a subsystem in an HVAC system, the CW system is comprised of chillers, cooling towers with fans, and CW pumps. At the condenser, cooling water takes in heat from the refrigerant and dissipates that heat at the cooling tower through the air. This heat exchanging process continuously happens to accomplish the variable cooling demand of the building. Therefore, a successful control strategy should reflect the connectivity between system components.

Most of the literature has concentrated on highly accurate modeling of CW system components and heat transfer processes. $\epsilon - NTU$ [64] and Merkel [65] models of cooling towers and Allen – Hamilton [66] and GN [67] models of chillers provide evidence for that. However, due to the complexity together with inconvenience in calculations and matching different components, these models are not much popular in real optimization applications.

A combination of Neural Network (NN) with particle swarm algorithm has been applied in [68] for the simulation and energy optimization of a chiller. The outcomes of that research emphasize the greater performance in power saving compared to the methods of linear regression and equal loading distribution. For cooling tower fans, the authors in [69] have employed capacity control methods to improve the energy efficiency and studied its influence on the parameters of approach and range. Even though X. Wei et al. [70] aimed to reduce water and energy usage of the system while meeting the building's cooling requirement, their study has been mainly centered on the designing of a cooling tower. The proposed optimization strategy has been contrasted with typical capacity control methods.

The research work carried out by L. Lu et al. [71] illustrated the applicability of the zone temperature method instead of the setpoint method for the control of cooling tower fans. Without accelerating the on-off switching, that method has offered energy savings in the case studies. A. Mohiuddin et al. [72] have proposed an optimal design algorithm that integrates Mixed Integer Non-Linear Programming (MINLP) and the rigorous Poppe model to trim down the yearly expenses of mechanical draft counter flow wet-cooling towers. In order to prevent negative impacts on local performances, global optimization of the system is important. However, the lack of consideration for the performance of the whole system can be observed in the above studies.

With a holistic view, controlling the speeds of the cooling tower fan and CW pump concurrently according to the load conditions and the weather has been identified as a better strategy in the literature [31, 73]. This is also confirmed in [74] since the presented MINLP model provided global optimality solely during the holistic system studies. A superstructure description based MINLP formulation has been applied to achieve global system optimization by the authors in [75]. To minimize the annual cost of both heat exchanger and CW systems, F. Liu et al. [76] have introduced another integrated optimizing method based on an MINLP model. Considering a multi-plant CW system, an MINLP model with a novel pump network has been utilized to reduce yearly costs in the research done by J. Ma et al. [77].

The network cost saving has been addressed in [78] by introducing an innovative two-step sequential methodology with hydraulic and thermodynamic models for pump and cooling networks. A robust optimal methodology with sequential Monte Carlo simulation has been applied to diminish the CW system annual cost by Q. Cheng et al. [79]. Furthermore, the influence of different operation conditions on the minimum cost has been examined through case studies. G.Y. Jin et al. [80] scrutinized the characteristics of the cooling tower and chiller with an aim to develop an algorithm for the realization of energy and operational cost reduction in the water loop. Optimal points for water discharging flow rate, fan speed, and valve locations at heat exchanger with minimized operation cost have been obtained via an integrated model in [81].

In a water-cooling plant, the total power consumption of the chiller and the cooling tower has been reduced with the use of a dither-based extremum-seeking control [82]. The authors in [83] introduced and investigated the correlation among frequencies of cooling tower fans and condenser water pumps to meet optimum energy usage by taking outdoor wet bulb temperature and cooling demand of an office building into account. Another research carried out by F. Yu et al. [84] has used a load-based speed control method to enhance energy and cost savings. Although substantial design and operation advances for the CW system have been offered through these studies, complicated implementation, longer computational time, and limited practicability are downsides that can create practical issues.

As a result of the analysis of air and water flow characteristics with the consumed energy by the CW system, Zhao et al. [85] have developed optimal water flow control method that outperforms the constant speed and constant temperature difference methods with lesser complexity. Nevertheless, the supply temperature of chilled water that affects the energy efficiency of the chiller has been kept at the same value throughout the study.

As per discussed literature, holistic approaches with simplified models to address energy and cost aspects are required to better suit real engineering applications. Moreover, low complexity, robustness, reliability, easiness to implement, and capability of maintaining cooling demand with a secure operation are expected to be featured in the optimization algorithm.

Inspired by this, developing an optimal strategy to enhance the energy saving in the CW system while addressing the encountered issues was also aimed during the presenting research. Here, optimal hourly CW flow rates will be determined based on cooling demand, outdoor wet bulb temperature, and ambient temperature variations of the considered building.

2.5 Temperature Controllers in Air Handling Unit

HVAC systems maintain a comfortable indoor environment over the thermal dynamics such as varying outdoor temperature and occupancy. The supply air temperature provided by AHU must be regulated so as to elude possible health and comfortability issues. Generally, as depicted in Figure 2.3 heating and cooling coils in AHU will be operated following the heating or cooling mode of operation. In the systems where variable secondary flow features are enabled, the required air temperature will be achieved by adjusting the flow of the thermal agent through the coils. This is accomplished by temperature controller based valve operation control. The non-regulated flow of thermal agents causes a huge energy wastage and unexpected costs for re-heating or re-cooling.

Literature provides interesting information on several temperature controllers which are proposed to function upon various types of control methods. Despite the simplicity and less expensiveness, applying on-off controllers in HVAC systems has been restricted due to hunting operation, the possibility of undesired operation conditions, and insufficient accuracy [86, 87]. With an aim to deduct the impact of disturbances while improving the interior comfortability, Ziegler Nichols method based PID controllers were employed for HVAC system models in the researches executed by G. Geng et al. [88] and B. Tashtoush et al. [89]. Moreover, M. Xu et al. [90] have combined a PID controller with a generalized predictive control method. However, imperfect gain selection of the PID controller can result in stability problems for the whole system [87].

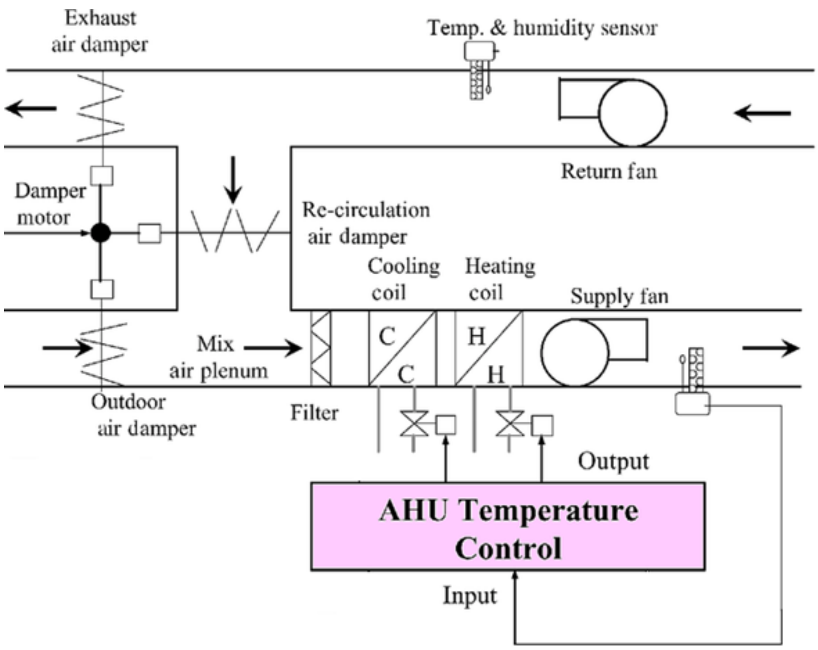


Figure 2.3: Schematic Diagram of the AHU Temperature Control

The involvement of mathematical models in deriving optimal control solutions for building heating systems also can be observed when going through the literature [91, 92]. A linear quadratic Gaussian controller has been applied on a linearized direct expansion air conditioner model in order to achieve temperature control with variable airflow [93]. Nevertheless, the necessity of an accurate system model and inherent complexity can be identified as the main drawbacks of these control approaches. Even though the robust method is trustworthy in setpoint tracking and disturbance rejection [94], this has not been improved for extreme weather changes. Adaptive control is another potential approach considering faster reactions and excellent stability [95], but then again proper system models are needed.

According to C. Guo et al. [96], the utilization of a NN controller in an AHU is capable of both improving the temperature setpoint tracking and avoiding the interference caused by the chilled water flow. Though it is fitting for non-mathematical models while showing a higher capacity in predicting, its practical usage has been limited by the requirement of a large set of data and proper training.

With the purpose of controlling the outlet fluid temperature of the heat exchanger in a chemical plant, N. Srivastava et al. [97] have developed an FL controller and the obtained results witness its superior performance compared to the traditional PID controller. For an efficient HVAC system, FL controllers are also more beneficial than the other typical controllers in terms of similarity to human reasoning, rapid operation, and high precision. Another effort to adjust the supply air temperature via controlling the chilled water flow rate and the temperature has succeeded by developing a dynamic heat transfer model and an FL controller [98]. Nonetheless, this is a very complex strategy and consumes more time for designing [99].

The Fuzzy Cognitive Map (FCM) method, introduced by F. Behrooz et al. [87] is an advanced and intelligent technique for HVAC system control. This can be implemented for nonlinear, Multiple-Input and Multiple-Output (MIMO), and complex systems. Although results can be obtained with small errors, settling time is significant. X. Yan et al. [100] have functioned a Smith predictive controller to realize temperature regulation during the winter seasons and it gives reduced time lag and faster response where the disturbances do not appear. While assuring the minimal temperature difference between two rooms and well-maintained desired temperatures, AHU outlet air temperature has been controlled using a feedback control system [101]. The main shortcoming of this strategy is the higher cost for required sensors.

Simple, accurate, efficient, less expensive, and practical novel temperature controllers are always warmly welcomed by the building industry intending to achieve expected energy goals. During the latter part of this research, the potential of utilizing a modified Elliot function based temperature controller to regulate the temperature of AHU supply air through adjustments on the valve operation and hence the thermal agent flow towards the heat exchanger coil is examined with some of the available.

Chapter 3

NOVEL ENERGY SAVING STRATEGY FOR HVAC SECONDARY CHILLED WATER PUMPS

This chapter describes a novel strategy to enhance the energy savings from the HVAC parallel chilled water pumps considering the required water flow variations caused due to the changes in the cooling load of the building. The power usage minimization problem of HVAC water pumps is solved using a step-wise approach which reduces the complexity, to determine the optimal number of pumps to be operated and their optimal operating speed so that the flow requirement and other operational constraints are satisfied. The mathematical approach, methodology, and simulation results are presented in the chapter's subsections.

3.1 Mathematical Modeling of the Chilled Water Pumping System

Variable flow configuration is popular in chilled water systems since that can avoid significant energy wasting caused by the constant flow through cooling coils in AHUs when the building's cooling load varies with some factors such as occupancy changes, outdoor air temperature variations, and other thermal loads, etc. Therefore, a primary constant, secondary variable chilled water plant with 2-way control valves and VFD on secondary pumps is considered for simulation studies as illustrated in Figure 3.1.

A temperature control system is useful to determine the required chilled water

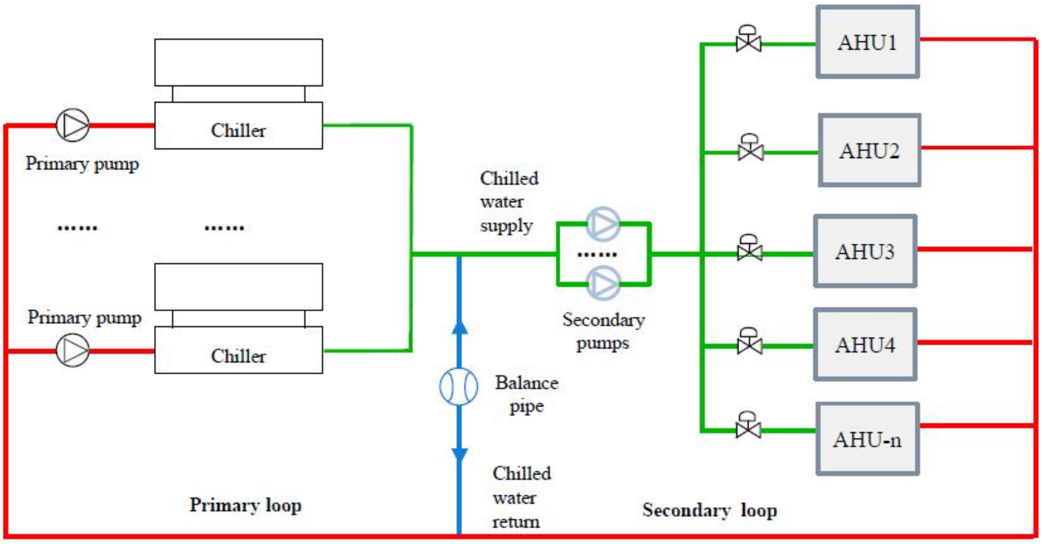


Figure 3.1: Schematic Diagram of the Considered Chilled Water System [102]

flow through a cooling coil. Figure 3.2 illustrates a diagrammatic of a temperature control system that can produce variable flow signals in order to maintain the temperature at a constant in the presence of system input variations. Therefore, the expected chilled water flow across the secondary pump system is the addition of the required flow through each cooling coil in the building. Other than that, the loop differential pressure variation also can be utilized to determine the required chilled water flow since the differential pressure represents the changes in coil loads. The secondary chilled water pumping system has the responsibility to compensate the demanded chilled water flow with minimum power usage while the constant flow primary pumps are maintaining the primary side pressure. That can be accomplished by the proposed method in this research.

Head, power consumption, and efficiency for a typical pump, are varied with the flow rate through the pump. Therefore, these parameters can be modeled using mathematical equations. At the constant speed, the head can be expressed as Equation (3.1) which is a quadratic function of the flow rate, and this gives the pump curve [29].

$$H = aQ^2 + bQ + c \tag{3.1}$$

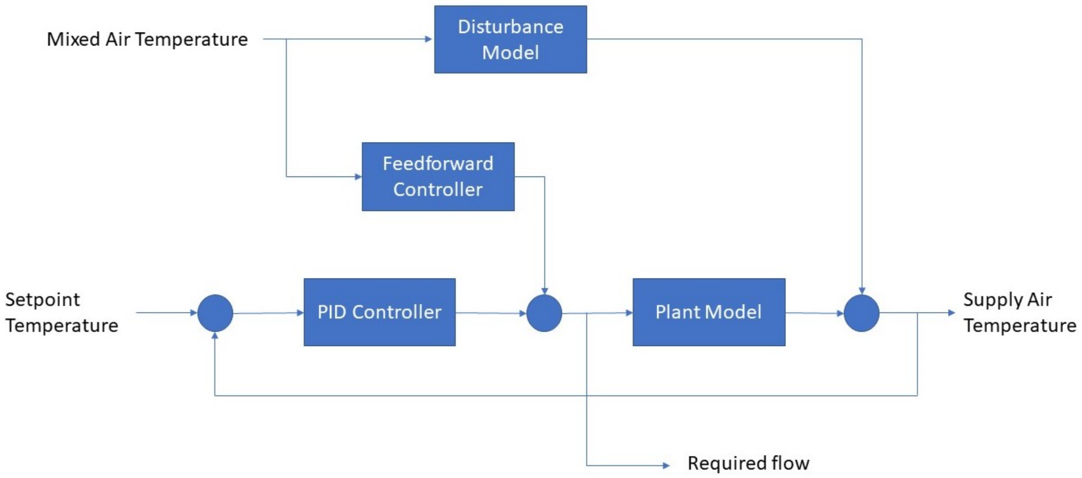


Figure 3.2: Proposed Control System for Controlling Room Temperature

Here, H is the pump head (m) at the rated frequency, Q is the chilled water flow rate through the pump (m^3/s) at the rated frequency, and coefficients, a, b and c can be found based on the data provided by the manufacture.

Similarly, power consumption also can be presented as a quadratic function of the flow rate which gives the power curve of the pump as given in Equation (3.2) [29].

$$P = a'Q^2 + b'Q + c' \tag{3.2}$$

In Equation (3.2), P is the power consumption (kW) of the pump running at rated frequency. a', b' and c' are constants of the performance and can be calculated similar to the a, b and c.

The manipulation of multiple pumps in parallel permits an extended range of flow rates than would be attainable with a single pump and the pump curve for the group of pumps is obtained by combining individual pump curves as shown in Figure 3.3. Therefore, when the identical pumps are connected parallel in the system, for a given value of head, flow can be increased by the times the number of pumps. When the parallel pump system comprises dissimilar pumps, a significant contribution cannot be expected from the small pump and its dead head may be exceeded by the system requirements. If the lowest head of the larger pump exceeds the smaller pump's dead head, the small pump is unable to contribute to increasing the flow while the large pump is operating. Therefore, similar pumps are mostly preferred in HAVC parallel pump systems.

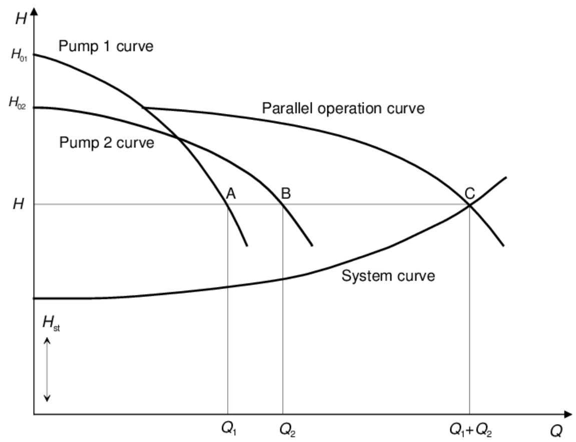


Figure 3.3: Pump Curve for Parallel Operation

The operating flexibility of the system can be further extended by varying the operating frequency of the pumps via VFD. To obtain performance models for a pump at variable frequencies, the above-developed models for pump head and power consumption can be modified using the affinity laws. Flow rate, head, and power at variable frequencies can be derived from the affinity laws as presented in Equations (3.3), (3.4) and (3.5).

$$Q_f = \left(\frac{f}{f_{rated}}\right) Q \tag{3.3}$$

$$H_f = \left(\frac{f}{f_{rated}}\right)^2 H \tag{3.4}$$

$$P_f = \left(\frac{f}{f_{rated}}\right)^3 P \tag{3.5}$$

Where, f is the variable frequency and f_{rated} is the rated frequency in Hz, Q_f

is the flow rate through the pump (m^3/s) at the frequency f, H_f is the pump head (m) at the frequency f and P_f is the power consumption (kW) for the pump running at the frequency f. It is observed from Equation (3.5) that the power usage can be significantly reduced by lowering the operating frequency.

Substituting for Q and H in Equation (3.1) from Equations (3.3) and (3.4),

$$H_f = aQ_f^2 + b\left(\frac{f}{f_{rated}}\right)Q_f + c\left(\frac{f}{f_{rated}}\right)^2 \tag{3.6}$$

Using Equation (3.6), the required frequency for the variable flow rate and head can be calculated.

In the same way, Equation (3.2) also can be modified by substituting parameters from the Equations (3.3) and (3.5). Then, power consumption at the variable frequency can be found using Equation (3.7).

$$P_f = a'\left(\frac{f}{f_{rated}}\right)Q_f^2 + b'\left(\frac{f}{f_{rated}}\right)^2 Q_f + c'\left(\frac{f}{f_{rated}}\right)^3 \tag{3.7}$$

The efficiency of the pump system, η_s is expressed by Equation (3.8). Q_s is the flow rate, H_s is the pump head, P_s is the power consumption, ρ is the density of the water, and g is the acceleration due to gravity.

$$\eta_s = \frac{Q_s H_s \rho g}{P_s} \tag{3.8}$$

When the power consumption is minimized, efficiency will be increased. Hence, reducing the power consumption of the pumping system is the aim of proposing this novel strategy.

The flow signal which is received from the temperature control system is only the required total flow rate to be flown through the cooling coils. However, the chiller system should deliver chilled water to create more head than the value giving the required flow to compensate the frictional losses as well. The Darcy Weisbach equation which is given in Equation (3.9) is used in this algorithm to model the frictional losses.

$$h_f = f_f \frac{L}{D}\frac{V^2}{2g} \quad \& \quad V = \frac{Q}{A} \tag{3.9}$$

h_f is the frictional loss, L, D and A are the length, diameter, and cross-sectional area of pipes, respectively and Q is the flow rate. All the parameters are in SI units. Assuming pipes have a circular cross-section,

$$h_f = f_f \frac{L}{D}\frac{Q^2}{2g\left(\frac{\pi D^2}{4}\right)^2} \tag{3.10}$$

Friction factor f_f in Equation (3.10) can be calculated using Equation (3.11) which is for the Swamee Jain approximation for a full-flowing circular pipe.

$$f_f = \frac{0.25}{\left(\log_{10}\left(\frac{\epsilon}{3.7D} + \frac{5.74}{Re^{0.9}}\right)\right)^2} \tag{3.11}$$

Here, ϵ is the roughness of the pipe and Re is the Reynolds number which can be expressed as in Equation (3.12). ν is the kinematic viscosity of water.

$$Re = \frac{VD}{\nu} \tag{3.12}$$

Now, if the system head is H_{sys}, total required head from the pumps; H_{total} equals to,

$$H_{total} = H_{sys} + h_f \tag{3.13}$$

After the required flow and head are determined accurately, the optimal operating point of the pump(s) can be decided according to the minimal power consumption criteria.

3.2 Methodology

The target of this work is to introduce a low complexity algorithm, which will reduce the power usage of HVAC parallel chilled water pumps by giving the optimal number of pumps to be operated and their optimal speed while satisfying the system flow requirement and other performance constraints. The optimization problem is formulated after the mathematical approach and then, the algorithm is developed in MATLAB®. Finally, the effectiveness can be discussed and compared through the simulation results.

3.2.1 Definition of the Constraints

Let the required flow as Q_{total}, head as H_{total}, the number of available pumps as N, the selected number of operating pumps as n and their operating frequency as f. Therefore, n is limited by N and f should be in the allowable frequency range. The recommended minimum frequency is 20 Hz since frequencies below this threshold will not save much energy and can cause problems in motors, chiller minimum flows, etc [11]. Also, the efficiency of VFD decreases rapidly when the speed ratio ($\mathrm{f/f_{rated}}$) of the drives is less than 0.5 [27].

$$0 \leq n \leq N$$
$$f_{min} \leq f \leq f_{max}$$

The flow rate through each operating pump should be within the range of 50% and 110% of the flow at BEP to circumvent radial thrust [29]. It is rare to have operational problems when the pumps are operated in the range of ±20% of their BEPs [28]. By combining these two facts,

$$\left(\frac{f}{f_{rated}}\right)0.8Q_{@BEP} \leq \frac{Q_{total}}{n} \leq \left(\frac{f}{f_{rated}}\right)1.1Q_{@BEP}$$

Also, selected pumps with optimal frequency should be able to provide the required total head.

$$H_{total} = a\left(\frac{Q_{total}}{n}\right)^2 + b\left(\frac{f}{f_{rated}}\right)\left(\frac{Q_{total}}{n}\right) + c\left(\frac{f}{f_{rated}}\right)^2$$

3.2.2 Problem Statement

Total power consumption of the secondary chilled water pumps; P_{total} equals the accumulated power consumption of each operating pump.

$$P_{total} = \sum_{i=1}^{n} P_i \tag{3.14}$$

If all pumps connected in parallel are identical and each pump consumes P_p power,

$$P_{total} = nP_p \tag{3.15}$$

Using Equations (3.7) and (3.15), the minimization problem can be defined as,

$$min\left(n\left[a'\left(\frac{f}{f_{rated}}\right)\left(\frac{Q_{total}}{n}\right)^2 + b'\left(\frac{f}{f_{rated}}\right)^2\left(\frac{Q_{total}}{n}\right) + c'\left(\frac{f}{f_{rated}}\right)^3\right]\right)$$

Subject to,

$$f_{min} \leq f \leq f_{max}$$

$$0 \leq n \leq N$$
$$\left(\frac{f}{f_{rated}}\right) 0.8Q_{@BEP} \leq \frac{Q_{total}}{n} \leq \left(\frac{f}{f_{rated}}\right) 1.1Q_{@BEP}$$
$$H_{total} = a\left(\frac{Q_{total}}{n}\right)^2 + b\left(\frac{f}{f_{rated}}\right)\left(\frac{Q_{total}}{n}\right) + c\left(\frac{f}{f_{rated}}\right)^2$$

3.2.3 Proposed Algorithm

The algorithm which is described by the flow chart in Figure 3.4, is developed and validated using MATLAB® simulation in order to find the combination of n and f, which gives the minimum power consumption for a given flow rate signal. First, the required total head is calculated, and the total flow will be determined. Then, the algorithm checks whether Q_{total} and H_{total} are greater than or equal to the minimum system requirements for the assurance of better operating conditions. Next, water flow and head through each pump are found by varying the number of pumps. The optimal frequency and total power consumption will be calculated if the flow value and obtained optimal frequency were within the high-efficiency region and the acceptable range respectively. Finally, the optimal speed and number of pumps with minimum power consumption are selected from the set and these values will be provided as the output. This step-wise approach is capable of reducing the complexity of solving the formulated minimization problem.

3.3 Simulation Results

3.3.1 Case Study I: Validating Parallel and Variable Speed Pumps

A commercial building with seven floors in Bethesda, U.S. is considered for the simulation. A schematic of the cooling system for this building has been designed in an energy efficient manner by using Taco Hydronic System Solutions (HSS), one of the commonly used HVAC solution software [103]. Based on that, the

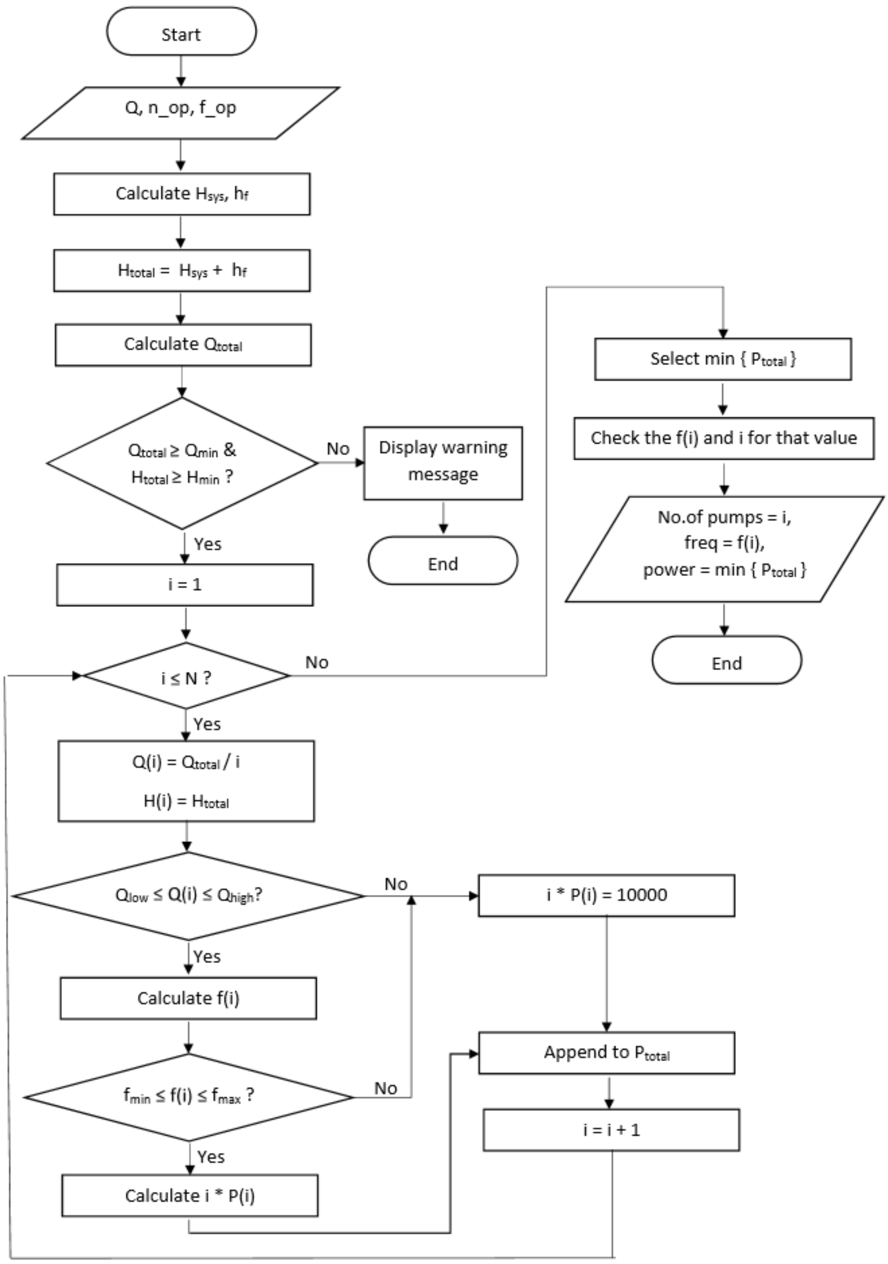


Figure 3.4: Flow Chart for the Proposed Algorithm

considered total design flow is 67.87 L/s, and the design head is 47.244 m for this study. In order to allow a better comparison, three types of pump arrangements, namely, a single pump, dual pumps connected in parallel, and three pumps connected in parallel were considered. Table 3.1 provides the pumps selected by the above software for three different arrangements with the same design flow and head.

For comparison, five cases including the three aforementioned pumping configurations were utilized. They are a single pump with constant frequency, two pumps connected in parallel with constant frequency, two pumps connected in parallel with the proposed algorithm, three pumps connected in parallel with constant frequency, and three pumps connected in parallel with the proposed algorithm. According to the number of pumps in parallel, pump curves for selected pumps were modeled in each scenario. However, the same inputs were given for all cases to have the following results.

As presented in Figure 3.5, twelve 1-hour time slots with different flow requirements were considered. In real time applications, these flow signals can be taken from the temperature controller as proposed in Section 3.1. Also, ν = 2.945x10^{-7}

Table 3.1: Selecting Pumps

Pump arrangement	Design flow for each pump (L/s)	Design head for each pump (m)	Selected pump type	Impeller diameter (m)	Speed (RPM)	Efficiency (%)
Single pump	67.8661	47.244	Taco FI4075	0.1811	3500	85.2
Two pumps in parallel	33.9331	47.244	Taco FI4013	0.3110	1760	77.3
Three pumps in parallel	22.6220	47.244	Taco FI3013	0.3136	1760	70.0

m^2 s^{-1} , ϵ = 0.003048 m, g = 9.81456 m s^{-2}, L = 628.65 m and D = 0.2032 m were used to obtain the results below.

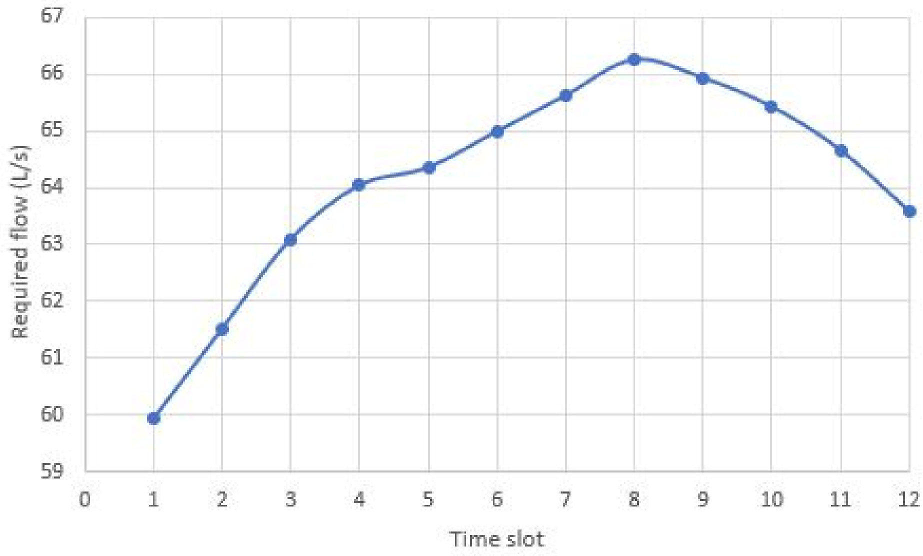


Figure 3.5: Required Flow Variation for 12 Time Slots

Figure 3.6 shows frequency variations obtained in the two cases that are with the proposed algorithm.

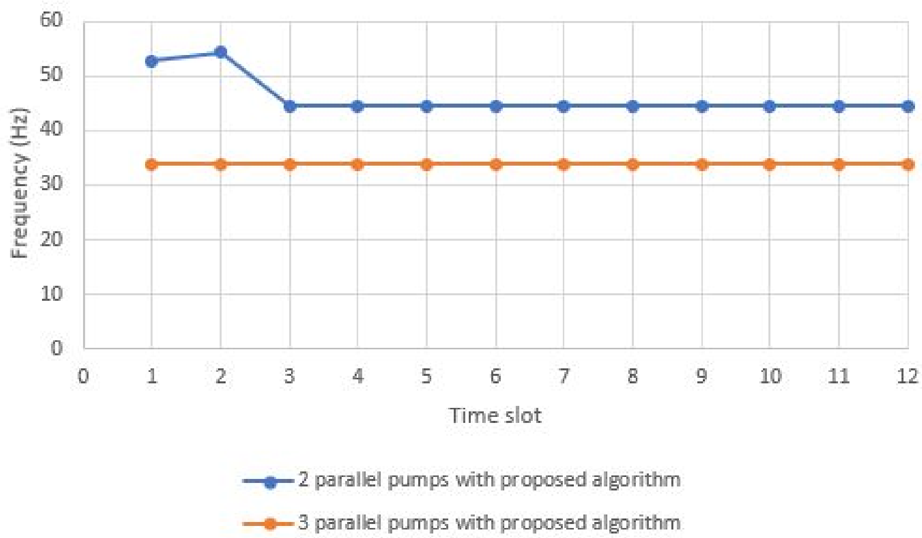


Figure 3.6: Frequency Variations with Proposed Algorithm

Figure 3.7 depicts the energy consumption at each 1-hour time slot for five different scenarios.

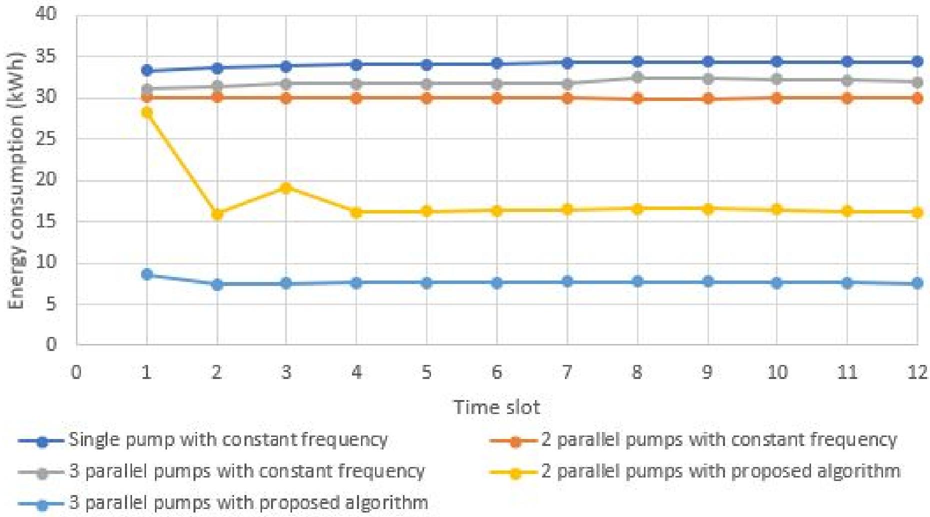


Figure 3.7: Energy Consumption of the Pumps at Each Time Slot

From the above results, we can contrast that pumping systems with the proposed algorithm causes a significant reduction in energy consumption at each different flow signal.

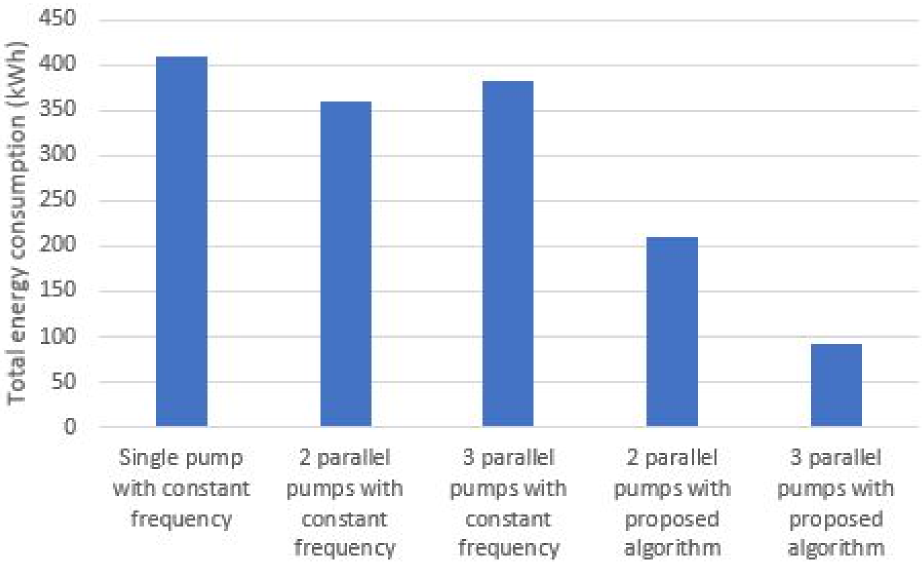


Figure 3.8: Total Energy Consumption Comparison for Different Strategies

When the accumulated energy consumption for the above considered 12 operating hours which is presented in Figure 3.8 is taken into account, a single pump with constant frequency has consumed energy the most, and three pumps connected in parallel with the proposed algorithm has consumed the least. By using two pumps configured in parallel, 11.93% of energy can be saved while using three pumps in parallel gives a 6.60% energy saving compared to that of using a large single pump. The energy savings by using the proposed algorithm are 41.62% and 75.91% for two and three parallel pumps, respectively. When we utilize three parallel pumps with the proposed algorithm instead of a large single pump, nearly 77.50% of energy consumption can be reduced.

Therefore, the energy consumption of pumps that are connected in parallel is less than operating a single large pump. Furthermore, higher energy saving targets can be achieved by using optimal speed and number of operating pumps as proposed in the algorithm.

3.3.2 Case Study II: Comparing the Proposed Method with Conventional Methods

The purpose of this study is to investigate the performance of the proposed method through the simulation results of an actual system with and without the developed algorithm. A commercial building in Colombo, Sri Lanka was considered for this study, and the model parameters were modified accordingly. This system consists of five parallel Paco KPV 80153 secondary chilled water pumps and they are operated based on a conventional cascading method. When the flow requirement comes to certain defined levels, pumps will be staged or de-staged and then the frequency will be determined. For the comparison, operational data at every 5 minutes of a weekday is utilized. Figure 3.9 depicts the assumed water flow variation during the considered period.

The number of operating pumps and their frequency determined by the proposed algorithm and the actual system are presented in Figure 3.10 and Figure 3.11 respectively. When the flow requirement is higher, the difference in the results given by the two methods can be observed. Although there are five pumps, due to low cooling demand in the considered season, the number of operating

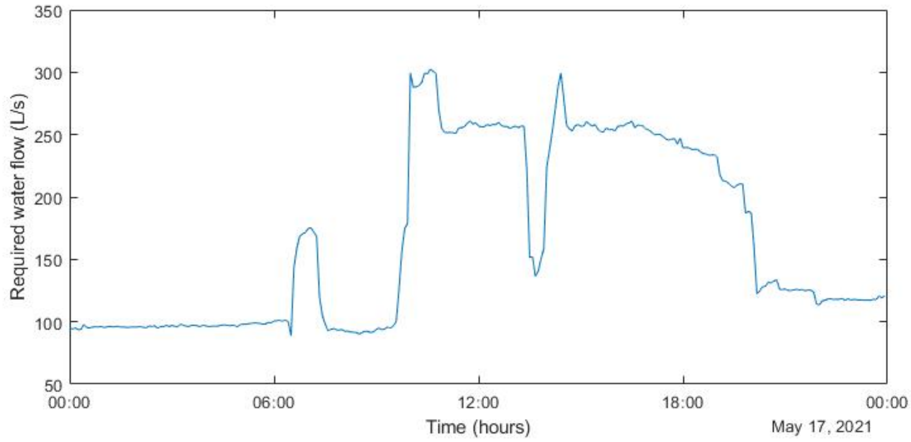


Figure 3.9: Required Water Flow Variation

pumps has not exceeded three.

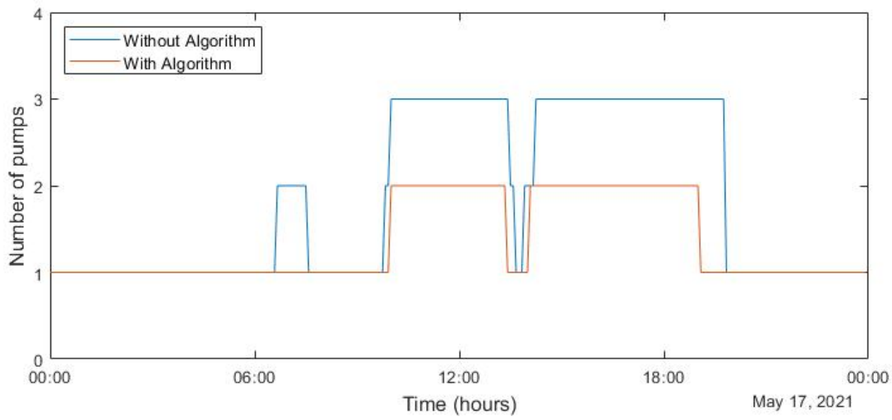


Figure 3.10: Number of Operating Pumps with and without the Proposed Algorithm

This system has been designed to maintain the frequency between 30 - 50 Hz using VFD. Figure 3.11 depicts that the proposed method has not violated this constraint at any time. Also, a lot of frequency dynamics can be observed in the actual system.

By using these results, the power consumption of the pumps when applying and not applying the proposed strategy can be calculated. Through Figure 3.12, the energy saving capability of the proposed algorithm can be emphasized.

As per Figure 3.13, total energy consumption at the end of the day has been

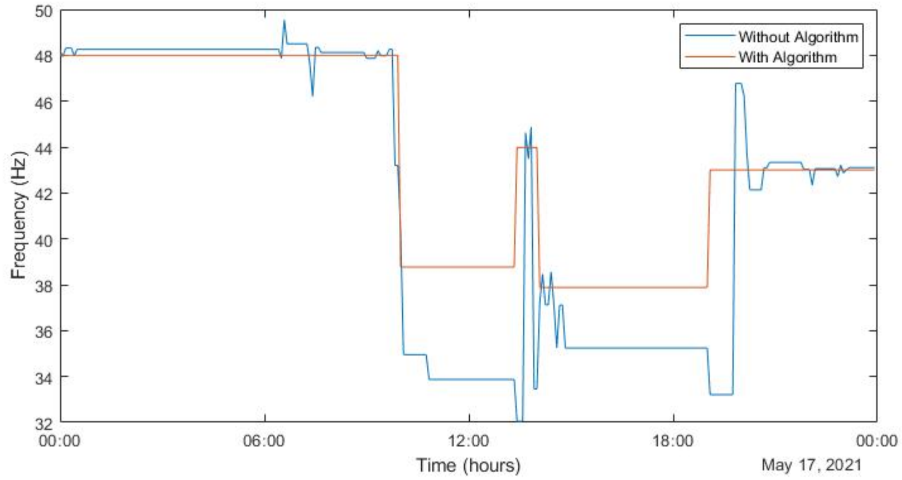


Figure 3.11: Frequency Variation of the Pumps with and without the Proposed Algorithm

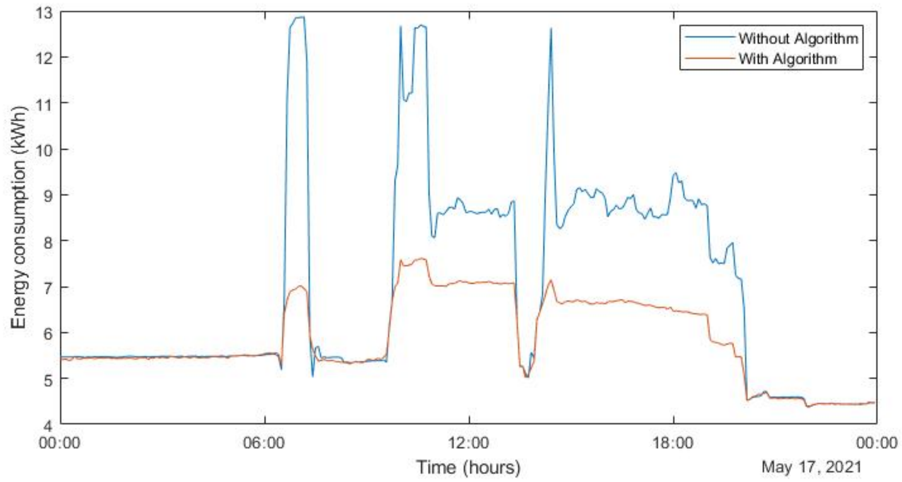


Figure 3.12: Energy Consumption of the Pumps with and without the Proposed Algorithm

reduced by the introduced approach in the comparison with the existing system. The difference in the total energy consumption between the two scenarios is 303.96 kWh and this is a 15.26% of reduction in consumed energy which can be realized by the proposed method.

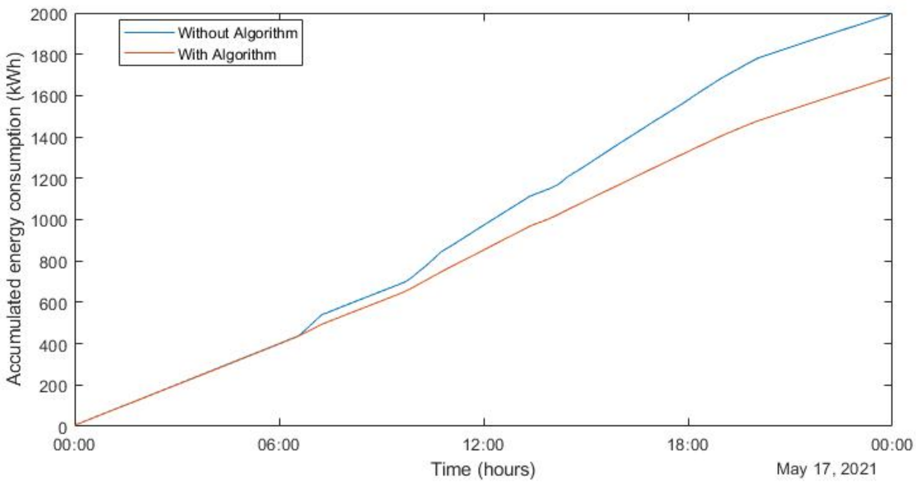


Figure 3.13: Accumulated Energy Consumption of the Pumps with and without the Proposed Algorithm

3.3.3 Case Study III: Comparing the Proposed Method with Intelligent Methods

Here, the study carried out in [27] for the application of PSO in the operation optimization of a parallel pump system was considered due to its validated better performance compared to other intelligent methods. All the system model parameters were updated and instead of frequency limitations, pump speed ratio (f/f_{rated}) has been constrained in the proposed algorithm as provided in the same literature. The respective system consists of two VFD driven centrifugal pumps and results were obtained for ten, hourly-time slots with different water flow requirements. The considered flow variation is shown in Figure 3.14.

As depicted from Figure 3.15, the optimal number of pumps determined by both algorithms is the same for most time slots. For only two flow values, decisions show a difference.

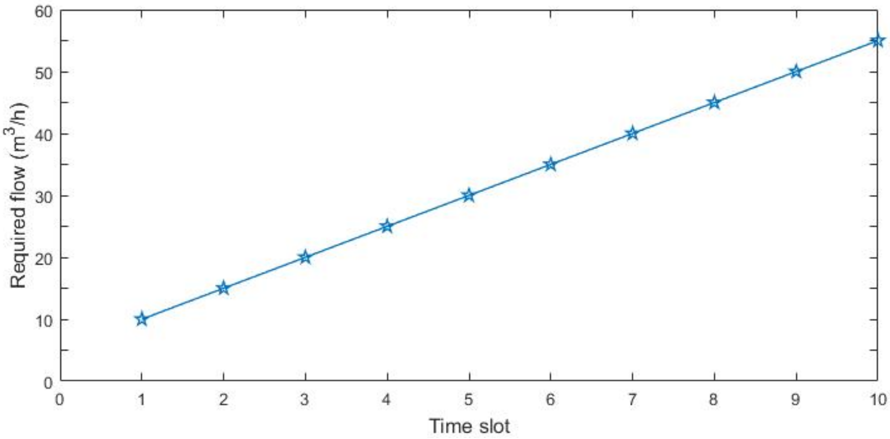


Figure 3.14: Required Flow Variation for 10 Time Slots

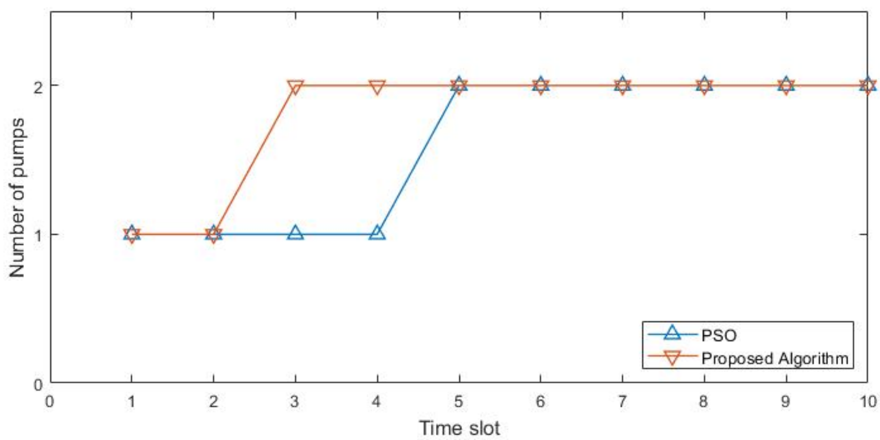


Figure 3.15: Optimal Number of Operating Pumps Determined by the proposed and PSO Methods

In the considered system, the speed ratio should be maintained between 0.5 and 1. Figure 3.16 illustrates that the proposed method has not violated this constraint at any time. Also, the suggested algorithm has offered similar results as given by PSO, except for two-time slots.

Figure 3.17 presents the total power consumption of the pumps at each time slot according to the optimal values of pump number and speed ratio decided by the two methods. As per Figure 3.18, total energy consumption at the end of the considered period has decreased by 1.78% in the introduced approach compared to the PSO method.

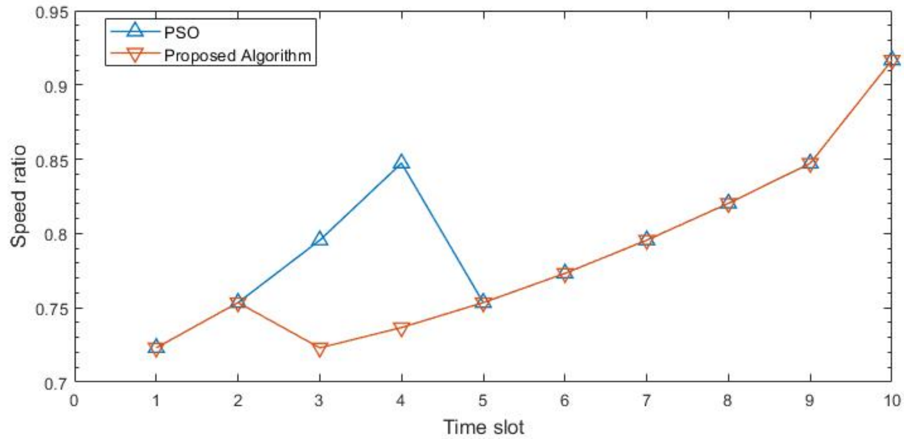


Figure 3.16: Optimal Speed Ratio of Operating Pumps Determined by the proposed and PSO Methods

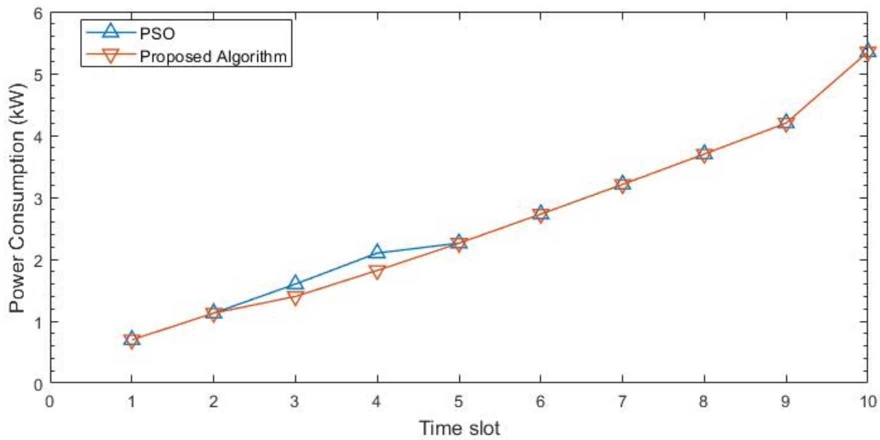


Figure 3.17: Power Consumption of the Pumps at Each Time Slot

The execution time of the PSO method is about 3 s on a 2.3 GHz Intel CPU for each optimization case [27]. However, this time is about 0.006 s for the proposed method in a similar facility.

Therefore, these results prove the better performance of the proposed method in comparison with the PSO intelligent method.

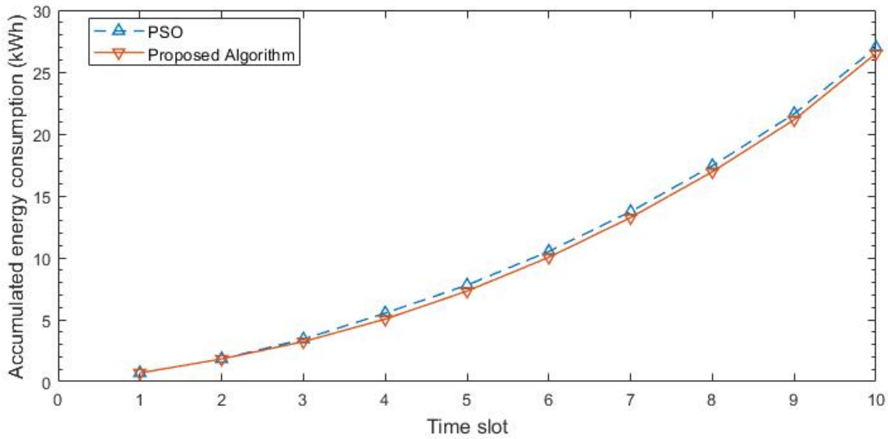


Figure 3.18: Accumulated Energy Consumption of the Pumps at Each Time Slot

Chapter 4

ZONE TEMPERATURE MANAGEMENT STRATEGY FOR OPTIMAL ENERGY AND COST SAVING WITH GUARANTEED THERMAL COMFORT

This chapter discusses an innovative approach to optimally schedule the temperature setpoint of AHU such that the energy usage and cost are reduced while assuring the occupants' comfort in an acceptable range. The proposed optimizing algorithm has been developed based on the Dijkstra technique and this ensures low computational complexity of the proposed method. Meantime, the comfort level has been addressed in terms of the PMV, PPD, and RP indices. The following subsections provide the methodology, proposed method, and simulation results in detail.

4.1 Methodology

The HVAC system in a building is comprised of a number of AHUs while each unit is serving one or a few zones. To regulate the indoor temperature of the serving area, a temperature setpoint is considered by each AHU. By using a Dijkstra method based novel algorithm, we suggest scheduling temperature setpoints of the day ahead of operation in an optimal manner such that the cost of energy consumption is minimized without compromising the users' thermal comfort and productivity.

4.1.1 Mathematical Background

Let $T = \{t_1, t_2, t_3, ..., t_N\}$ be the set of N scheduling time slots with t_n denoting the n^{th} time slot and every time slot having the same time duration d. A set of predicted outdoor temperatures at each considered time slot can be defined as, $O = \{b_1, b_2, b_3, ..., b_N\}$ where, b_n denotes the predicted outdoor temperature in the time slot t_n. Also, a variable a_{in} is defined as the indoor temperature in the time slot t_n.

If S_n is the setpoint temperature in the time slot t_n, then

$$S_n = f(a_{in}, b_n) \tag{4.1}$$

where f is a function of the indoor and outdoor temperatures.

If P_n is the power consumption during the time slot t_n, due to the setpoint temperature S_n

$$P_n = g(S_n) \tag{4.2}$$

where g is a function of the setpoint temperature.

Let C_n be the per-unit cost of electricity at time slot t_n as per the Real Time Pricing (RTP) scheme. Then, the total cost of electricity during the considered period can be calculated as

$$
\begin{aligned}
C_{total} &= \sum_{n=1}^{N} dP_n C_n \\
&= \sum_{n=1}^{N} dC_n g(S_n) \\
&= \sum_{n=1}^{N} dC_n g[f(a_{in}, b_n)]
\end{aligned} \tag{4.3}
$$

4.1.2 Definition of the Constraints

Occupants' thermal comfort cannot be compromised and hence that must be incorporated into the optimization problem in the form of constraints. To ensure a sufficient comfort level, acceptable limits of three thermal comfort indices, namely, PMV, PPD, and RP become the problem constraints.

PMV for a specific environment can be calculated using the equation provided by the International Organization for Standardization (ISO) [42]. Although zero PMV condition is the best, according to the standard of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) [104], PMV values between +0.5 and -0.5 are acceptable as the sensation is neutral in this region. Beyond [-0.5, +0.5] it is cooler and warmer, respectively. This is formulated as the first constraint in the problem.

$$
-0.5 \leq PMV_n \leq +0.5, \quad \forall n
$$

Equation (4.4) was used in this work to determine PPD index [44].

$$
PPD(\%) = 100 - 95\ e^{(-0.03353 PMV^4 - 0.2179 PMV^2)} \tag{4.4}
$$

As per the ISO standard, PPD values greater than 10% are not recommended.

This becomes the second constraint in the optimization problem.

$$PPD_n \leq 10\%, \quad \forall n$$

The third constraint is related to RP [62] and can be expressed as

$$RP(\%) = -0.0351PMV^3 - 0.5294PMV^2 - 0.215PMV + 99.865 \tag{4.5}$$

In order to guarantee higher performance, RP is kept greater than 99.6% [62].

$$RP_n \geq 99.6\%, \quad \forall n$$

4.1.3 Problem Statement

The objective of this study is to minimize the sum of the cost of electricity throughout the inscribing period in such a way no contravention happens for any constraint defined above. Now, the optimization problem can be presented as

$$min\left(\sum_{n=1}^{N} dC_n g[f(a_{in}, b_n)]\right) \tag{4.6}$$

subject to the following constraints

$$\begin{gathered} -0.5 \leq PMV_n \leq +0.5 \\ PPD_n \leq 10\% \\ RP_n \geq 99.6\%. \end{gathered}$$

4.2 Proposed Strategy

For solving the formulated problem with reduced complexity, a novel approach as in Figure 4.1 is introduced in this section. The number of time slots to be considered in the operating time interval, real time electricity prices, and outdoor temperature at each time slot are taken as the input to the algorithm, which has been developed in MATLAB®. Three sub-models are also included and one of them gives the interrelation between indoor temperature, outdoor temperature, and supply air temperature. The other two provide the characteristics of setpoint temperature – supply air temperature and power consumption – setpoint temperature. These models can be obtained either from a building simulation model or by regression analysis on actual data of the considered building. Airspeed, relative humidity, metabolic rate, and clothing level of the occupants in the zone are assumed to be kept constant during the operating period. In conjunction with those parameters, PMV indexes for different indoor temperatures are found with the help of CBE Thermal Comfort Tool [105]. Further, Equations (4.4) and

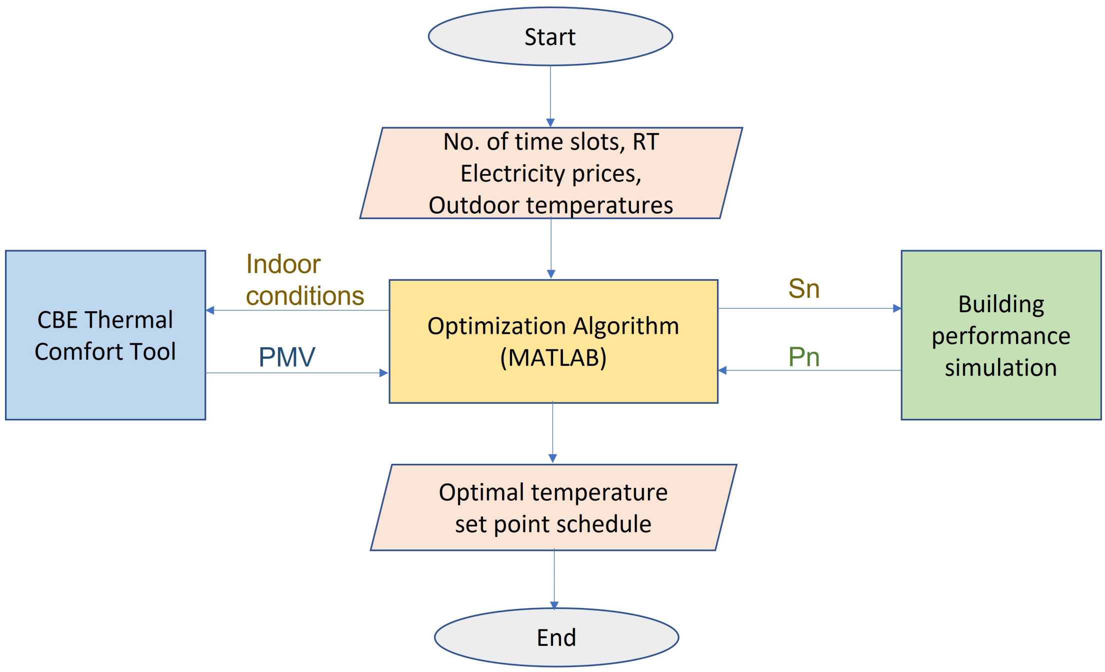


Figure 4.1: Flow Chart for the Proposed Strategy

(4.5) are used for calculating PPD and RP indices.

After taking allowable indoor temperatures a_i, belongs to $I = \{a_1, a_2, a_3, ..., a_M\}$ such that PMV, PPD, and RP constraints are satisfied, all the combinations of indoor temperatures with each outdoor temperature are arranged in the optimization algorithm. Then, the required supply air temperature ($ts_{i,n}$) for each situation will be calculated. Using that, required setpoint temperatures will be determined. At this stage, all possible setpoint temperatures are inserted into a matrix (T_s). By calculating power ($p_{i,n}$) and energy consumption ($e_{i,n}$) for each value, energy consumption matrix (E) can be obtained. Afterward, the Dijkstra technique is utilized in this algorithm to find the minimum cost path [106]. Here, $D_{X \to Y}$ is the distance between X and Y points in the matrix, and let S denote the starting point. In the situation where multiple paths offer minimum cost, the path with the highest average RP will be selected as the final solution. This will give the optimal temperature setpoint schedule for the considered period.

Optimization Algorithm

1. Initialize $d, N, M \geq 0, TD_i = 0, C, I, O \neq \phi,\ T_s, E, D_i, Path_i, I^*, RP_{i^*}, ARP = \phi$

2. for n=1 to N,

    for i=1 to M,

        $ts_{i,n} = f(a_i, b_n)$

        $T_s = T_s \cup \{ts_{i,n}\}$

    end for

  end for

3. for n=1 to N,

    for i=1 to M,

        $p_{i,n} = g(ts_{i,n}),\ \forall\ ts_{i,n} \in T_s$

        $e_{i,n} = dp_{i,n}$

        $E = E \cup \{e_{i,n}\}$

    end for

  end for

4. for i=1 to M,

    Calculate : $D_{S \to e_{i,1}} = e_{i,1}c_1$

    Update : $TD_i = D_{S \to e_{i,1}}, Path_i = Path_i \cup \{e_{i,1}\}$

  end for

5. for n=1 to N,

    for i=1 to M,

        for x=1 to M,

            Calculate : $d_x = D_{e_{i,n} \to e_{x,n+1}}$

            $D_i = D_i \cup \{d_x\}$

        end for

        Select x^* with $min\ (D_i)$

        Update : $TD_i = TD_i + d_{x^*}, Path_i = Path_i \cup \{e_{x^*,n+1}\}$

    end for

end for

6. Select i^* with $min\ (TD_i)$

7. Update : $I^* = I^* \cup \{i^*\}$

8. if $length(I^*) = 1$

$Path_{i^*}$ is the optimal path $(Path_{opt})$

else if $length(I^*) > 1$

Calculate : $rp_{i,n} = h(ts_{i,n}),\ \forall\ e_{i,n} \in Path_{i^*},\ \forall\ i^* \in I^*$

Update : $RP_{i^*} = RP_{i^*} \cup \{rp_{i,n}\}$

Calculate : $ARP_{i^*} = Avg(RP_{i^*}),\ \forall\ i^*$

Update : $ARP = ARP \cup \{ARP_{i^*}\}$

Select i^{**} with $max\ (ARP)$

$Path_{i^{**}}$ is the optimal path $(Path_{opt})$

end if

9. Assign : $e_{i,n} \to ts_{i,n},\ \forall\ e_{i,n} \in Path_{opt}$

10. $Path_{opt}$ is the optimal temperature setpoint schedule

---

4.3 Case Study

For evaluating purposes, a simulation study was carried out for the proposed strategy. The operation of AHU unit 5 in block 1 of a university building in Singapore [107] was considered and the collected data in 2018 was used for the study. In the considered AHU, the return air exhausted from the conditioned area is mixed with fresh air and flows through the air filter as presented in Figure 4.2. When this mixed air passes across the cooling coil, supplied chilled water absorbs heat from the air and returns to the chiller. Then the cooled air is delivered to the conditioned space via ducts.

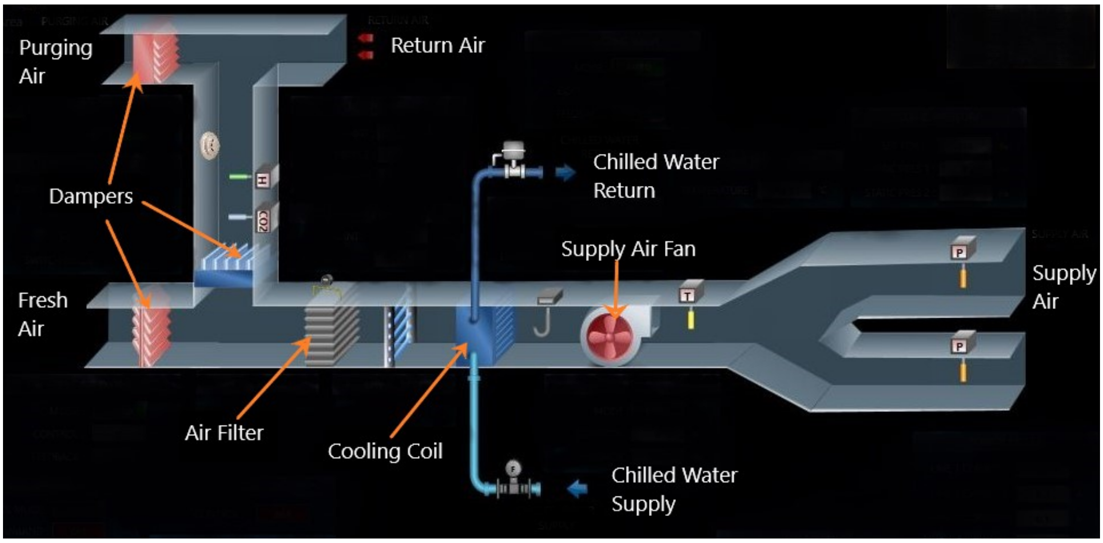


Figure 4.2: Schematic Diagram of the AHU

Through the regression analysis, three sub-models which were described in the previous section were obtained.

4.3.1 Room Temperature, Outdoor Temperature, and Supply Air Temperature

Outdoor temperatures for Changi airport [108] were used since it is very close to the university premises. The relation between room temperature (T_{room}), outdoor temperature ($T_{outdoor}$), and supply air temperature (T_{SA}) is graphically shown in Figure 4.3 and all the temperatures are in ^{0}C. Equation (4.7) expresses the relationship obtained with $R^2 = 99.89\%$ through the regression analysis.

$$T_{room} = 0.007817\ T_{outdoor} + 0.557098\ T_{SA} + 13.97529 \tag{4.7}$$

4.3.2 Supply Air Temperature and Temperature Setpoint

In the actual system, off coil temperature setpoint had been used and therefore that becomes the optimizing variable of the study. When the temperature

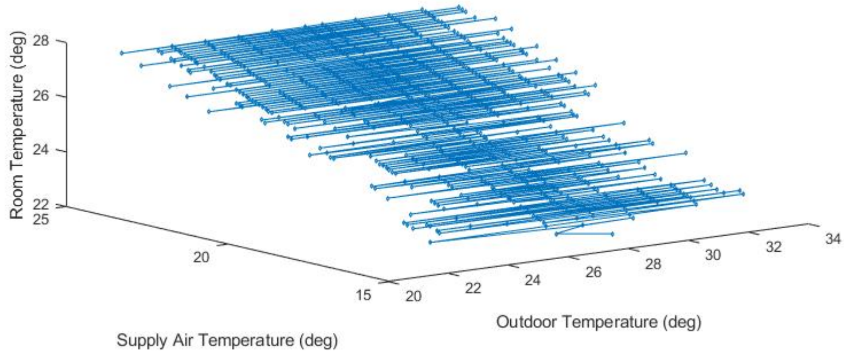


Figure 4.3: Relationship between Room Temperature, Outdoor Temperature, and Supply Air Temperature

setpoint varies, the temperature of the supply air will change as given by Figure 4.4 and Equation (4.8) expresses the relationship between them. Here, T_{sp} denotes the setpoint temperature in ^{0}C. According to the regression analysis, $R^2 = 96.11\%$.

$$T_{SA} = 0.8147\ T_{sp} + 4.1587 \tag{4.8}$$

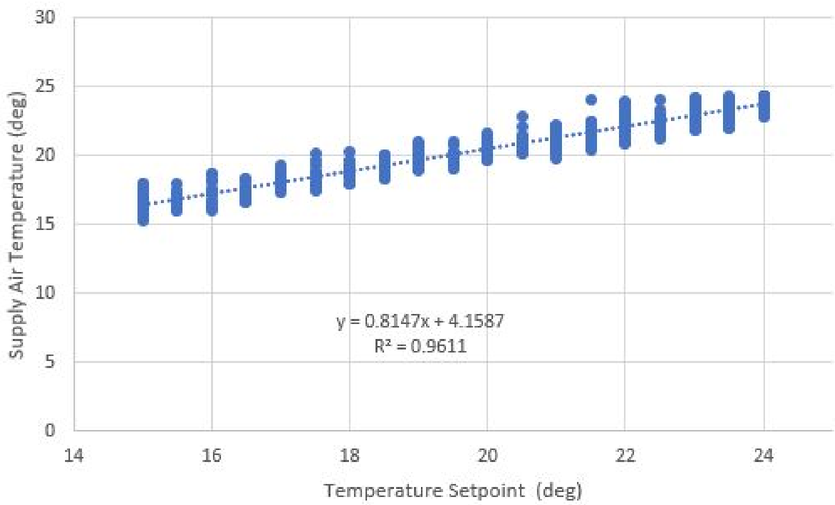


Figure 4.4: Relationship between Supply Air Temperature and Setpoint Temperature

4.3.3 Power and Temperature Setpoint

According to the actual power consumption and temperature setpoint data provided in Figure 4.5, relationship Equation (4.9) was found with $R^2 = 97.47\%$. P is the power consumption in kW.

$$P = -0.09756\ T_{sp} + 3.34837 \tag{4.9}$$

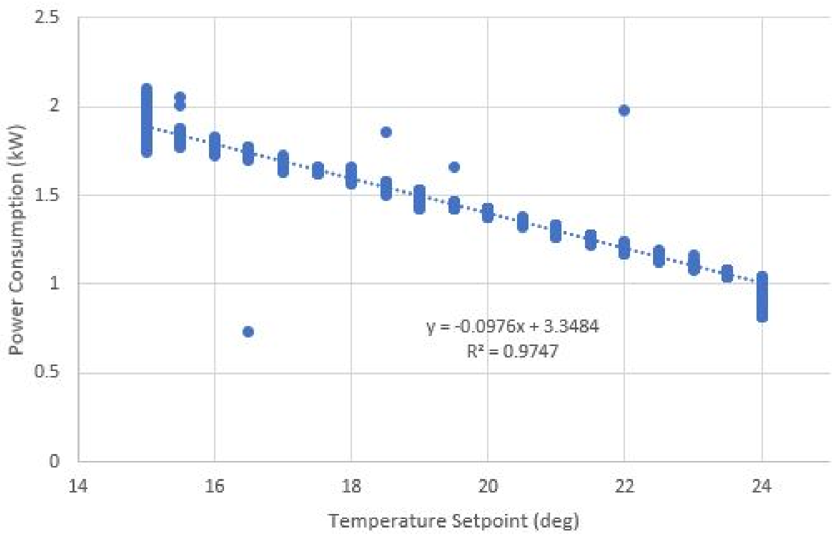


Figure 4.5: Relationship between Temperature Setpoint and Power Consumption

4.4 Simulation Results and Comparison

The introduced algorithm was developed in MATLAB® and tested in simulations for the zone that is served by the aforementioned AHU unit. Airspeed of 0.1 m/s and relative humidity of 60% were maintained in the considered area. It was assumed that the chilled water enters the cooling coil at 7.7 ^{0}C and leaves at 12.7 ^{0}C. The metabolic rate and clothing level of the occupants were assumed to be 1.1 met and 0.61 clo, respectively since their activities are similar to typing and they are wearing trousers with long sleeve shirts. Results have been obtained for the month of January 2018, to enable a fair comparison. Although the normal working hours of the respective zone lie between 0800h to 1900h on weekdays, the 14th day can be seen as a special working day since the working hours are different. However, for the simulation, all working hours during the month were taken into account. The duration of a time slot is 30 min allowing to determine the optimal temperature setpoint at every half-hour interval. That means at the starting of each time slot, the temperature setpoint can be changed and retained until the time slot ends.

The outdoor temperature during each time slot throughout the month was collected from the weather data [108] and is shown in Figure 4.6. These dynamics can be used for effective and efficient HVAC operations. In this study, temperature setpoints will be optimized to achieve both thermal comfort and low cost.

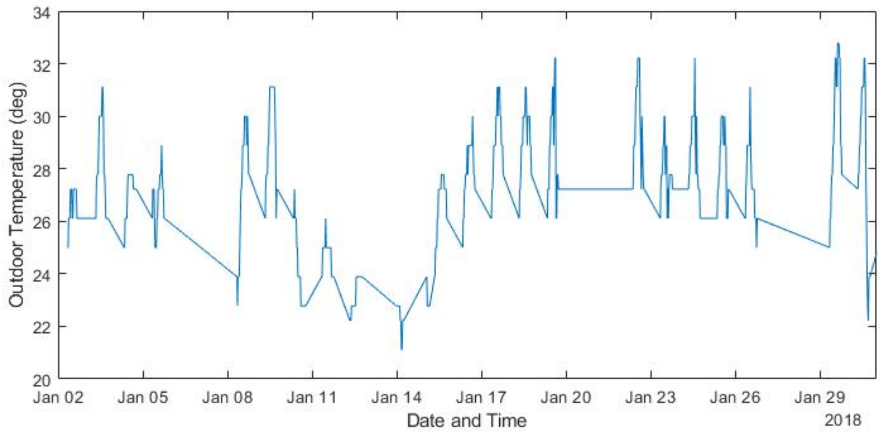


Figure 4.6: Outdoor Temperature at Each Time Slot

Figure 4.7 depicts the optimal temperature setpoints for each time slot determined by the proposed algorithm and the actual setpoints used in the system. These actual setpoints were adjusted using a method that considers the outdoor temperature dynamics only. It is observed from the data that the actual setpoints have been moved between 15 – 25 ^{0}C. Lower setpoints often cause higher energy consumption and hence increased cost. Since both low and high temperatures may affect thermal comfort, it needs to be very careful when moving to those values. The variation of optimized setpoints has been narrowed around 21 ^{0}C to enable a reduction in energy consumption and cost.

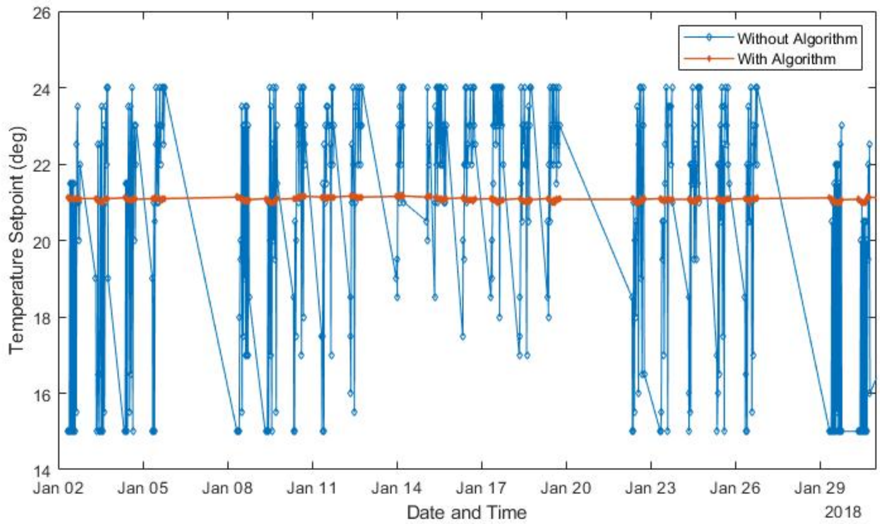


Figure 4.7: Selected Temperature Setpoints at Each Time Slot with and without the Algorithm

Power consumption by AHU as a subsystem varies with the setpoint as seen in Figure 4.8. In the system without the proposed strategy, higher power values have been reached during some time slots. However, the proposed algorithm has allowed reduced power values along the simulation time.

Since each power value is consumed for 30 min, this data can be converted to energy as presented in Figure 4.9.

For understanding the energy savings from the suggested method, consideration of accumulated energy consumption over the total period would be beneficial. By observing Figure 4.10, total energy consumption occurred after the simulation horizon was 325.74 kWh and that was 338.75 kWh in the existing case. Therefore, a 3.84% of reduction can be achieved if the novel strategy had been implemented in the system.

One of the principal objectives of this study is to reduce the cost of electrical energy and that can be evaluated from the cost analysis. Real-time prices for the

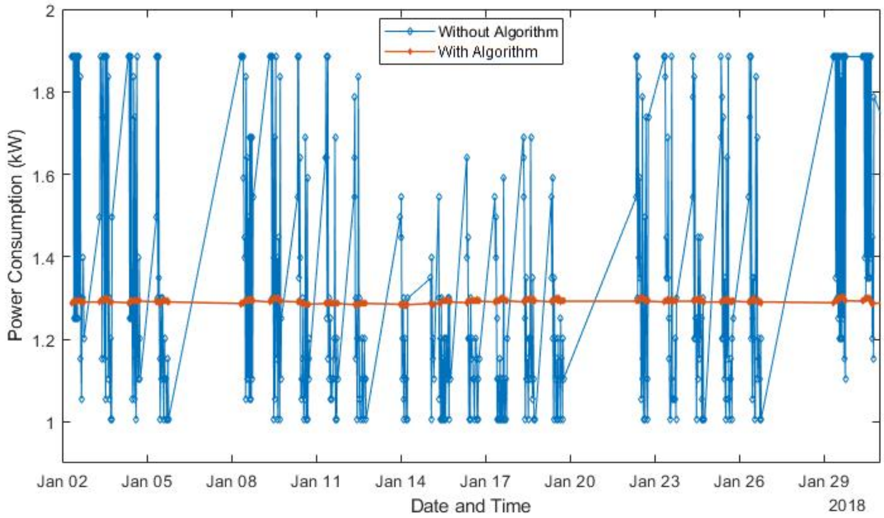


Figure 4.8: Power Consumption at Each Time Slot with and without the Algorithm

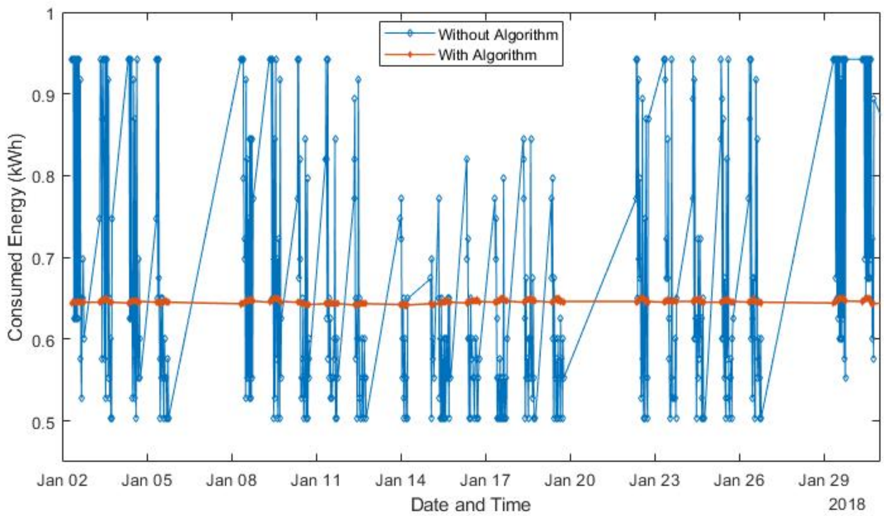


Figure 4.9: Energy Usage at Each Time Slot with and without the Algorithm

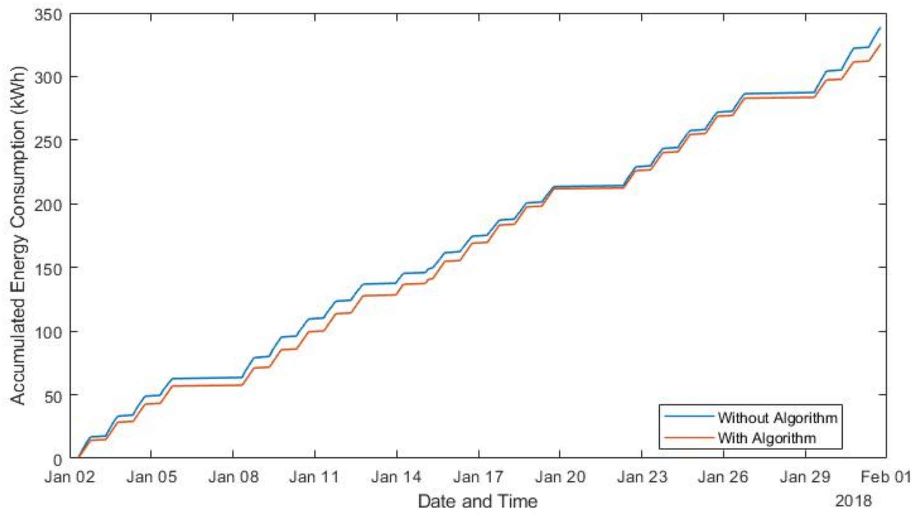


Figure 4.10: Accumulated Energy Usage at Each Time Slot with and without the Algorithm

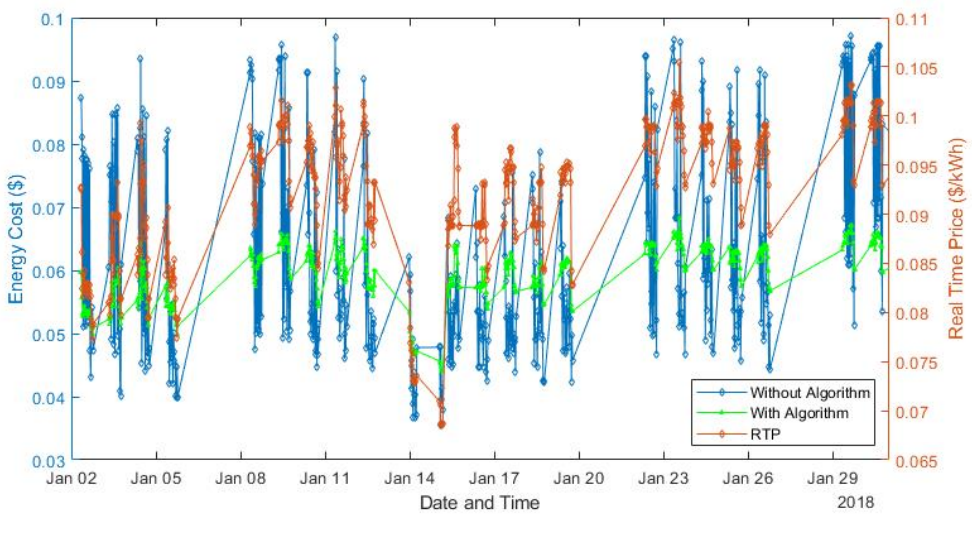


Figure 4.11: RTP of Electricity and Cost for Electricity at Each Time Slot with and without the Algorithm

simulated time slots were obtained from [109] and they are presented in Figure 4.11. Also, that includes how the cost of electricity was varying in actual and optimized situations.

Further, the accumulated cost over the simulation period can be considered in order to identify cost benefits from the introduced approach and that is depicted in Figure 4.12. All the time, the proposed algorithm has offered the minimized cost, and it is a 4.22% reduction compared to the total cost where the proposed method has not been utilized.

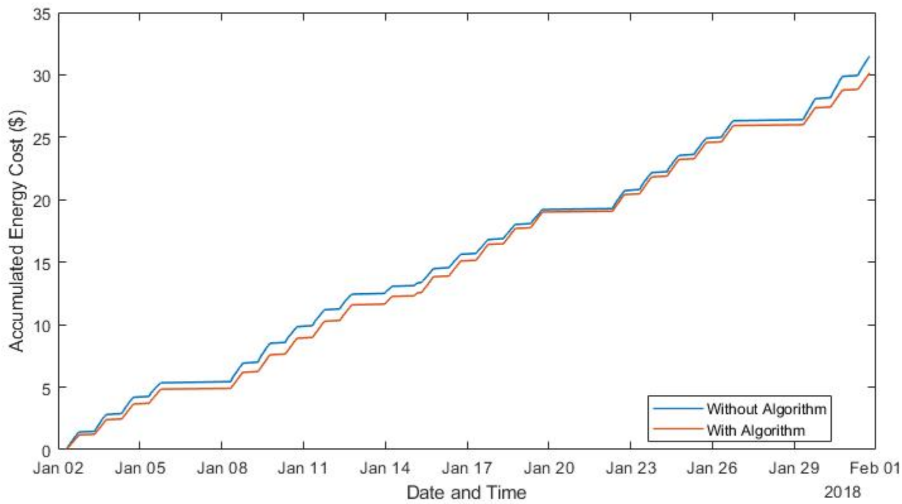


Figure 4.12: Accumulated Cost for Electricity at Each Time Slot with and without the Algorithm

Even though the proposed strategy shows better performance over the existing system concerning the consumed energy and cost, its impact on the indoor temperature, thermal comfort, and productivity also has greater importance. Figure 4.13, Figure 4.14, Figure 4.15 and Figure 4.16 illustrate the indoor temperature, PMV, PPD and RP variations, respectively.

The considered system has caused huge temperature variations in the indoor environment compared to the system with the proposed method.

The impact of the indoor temperature dynamics on thermal comfort is reflected well in the PMV index. Although values beyond the range [-0.5 0.5] are not

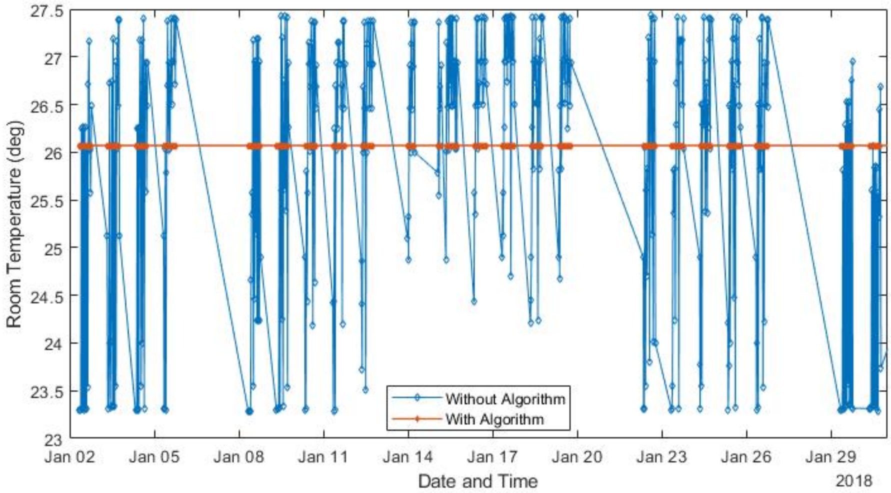


Figure 4.13: Indoor Temperature at Each Time Slot with and without the Algorithm

comfortable and acceptable by the standards, Figure 4.14 depicts that the actual system has caused PMV out of range most of the time. Nevertheless, PMV limitations and hence the comfort level has not been violated any time by the suggested algorithm in its quest to achieve its key objective of minimizing energy usage and energy cost.

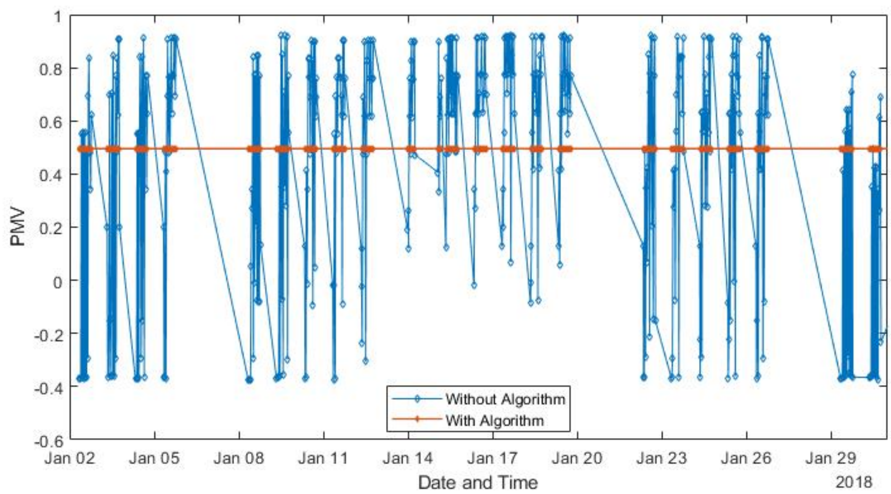


Figure 4.14: PMV at Each Time Slot with and without the Algorithm

When considering the PPD index, the proposed algorithm remained at 10% while the existing system has gone to higher values during many time slots reaching as far as 24% at times as illustrated in Figure 4.15. Thus, maintaining significant comfort and occupants' satisfaction can be assured through the strategy presented here.

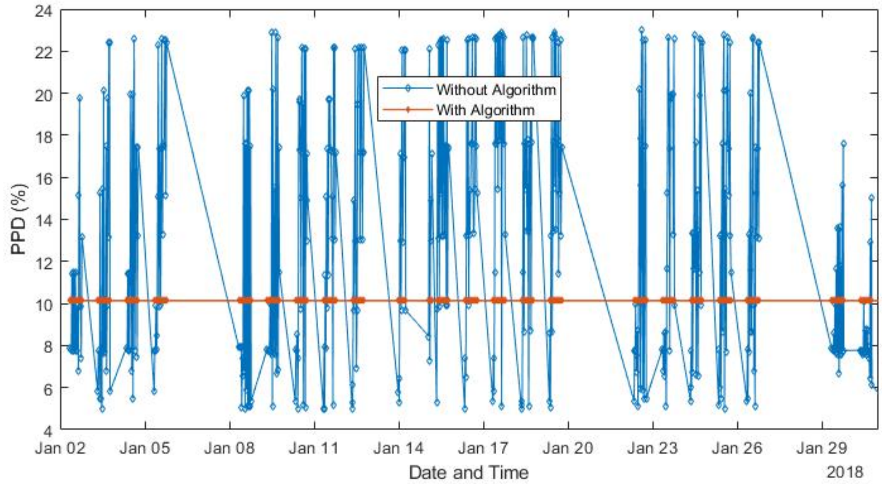


Figure 4.15: PPD at Each Time Slot with and without the Algorithm

RP during each time slot is provided by Figure 4.16. If the average RP throughout this month is taken into account, the actual system performed 99.56% while the introduced method was showing 99.62%. Hence, the results have shown a 0.06% improvement in productivity with the proposed algorithm.

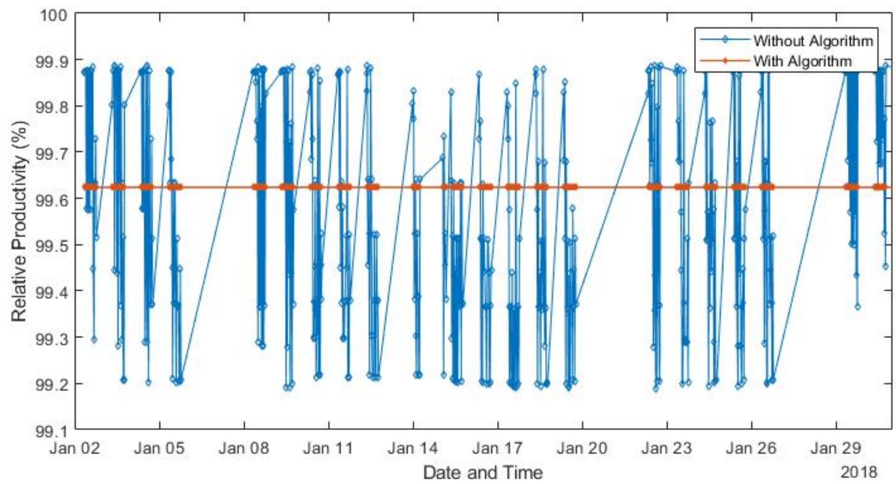


Figure 4.16: RP at Each Time Slot with and without the Algorithm

Benefits produced by the suggested method in terms of total energy, total

Table 4.1: Summary of Total Energy, Total Cost of Energy and Average RP for the Month

	Existing System	Proposed Algorithm	Difference
Total Energy (kWh)	338.75	325.74	13.01
Total Cost of energy ($)	31.49	30.16	1.33
Average RP (%)	99.56	99.62	0.06

energy cost, and average RP are summarized in Table 4.1.

Chapter 5

NOVEL ENERGY SAVING SCHEME FOR HVAC COOLING WATER SYSTEM

This chapter explains a new approach to decreasing the energy consumption of CW systems in central HVAC systems. This has been achieved by utilizing the outdoor and indoor environmental status of the served building and simplified system model components in the proposed scheme. For real-time control, this method is useful because of the offered simplicity, robustness, and efficiency. The methodology and simulation results are further described in the subsections of this chapter.

5.1 Methodology

Ahead of formulating the optimization problem, semi-empirical models for the key units of the HVAC CW system were derived from the integrated point of view. Next, the prospective algorithm was developed in MATLAB® and simulations were executed for a model of an office building. Lastly, consumed energy reduction potential was investigated via the comparison with recently introduced methods.

5.1.1 System Model

A variable flow CW system including a chiller, a cooling tower, and a pump as appeared in Figure 5.1 was contemplated during this work. Reducing the

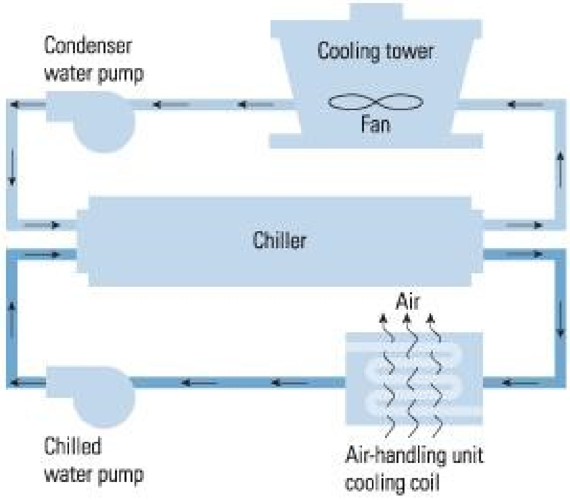


Figure 5.1: Cooling Water System

complexity, having a theoretical basis, and improving the general applicability were the main causes of utilizing semi-empirical models for these units.

In view of the association between power, cooling capacity, condenser, and evaporator, the input power of the chiller can be stated as in Equation (5.1).

$$P_c = (Q_e + q_e)\frac{T_{ci}}{T_{eo}} + (f_{HX} - 1)Q_e - q_c \tag{5.1}$$

Where P_c is the input power of chiller (kW), Q_e is the cooling capacity of chiller (kW), q_e is the heat loss of evaporator (kW), T_{ci} is the inlet temperature of condenser (^{0}C), T_{eo} is the outlet temperature of evaporator (^{0}C), f_{HX} is the dimensionless term defined by J. M. Gordon [67] and q_c is the heat loss of condenser (kW).

If q_e, f_{HX} and q_c were placed constant, Equation (5.1) can be simplified in to Equation (5.2).

$$P_c = (Q_e + A)\frac{T_{ci}}{T_{eo}} + (B - 1)Q_e - C \tag{5.2}$$

A, B, and C coefficients can be obtained through fitting historical operational data of the system. Moreover, Equation (5.2) can be corrected as given in Equation (5.3) by applying the outlet temperature model updating method (OT method) [85] that offers higher fitting results.

$$P_c = (Q_e + A)\frac{T_{co}}{T_{eo}} + (B - 1)Q_e - C \tag{5.3}$$

Equation (5.4) is used to calculate the power consumption of the CW pump, P_p (kW). Here, ρ is the density of water (kg/m^3), S is the resistance of the pipe network of the CW system (m^3/s)^2, η is the efficiency of the water pump (%), ΔH is the height difference between the water distributor and the liquid level of the water tank (m) and Q is the real flow rate of the pump (m^3/h) [85].

$$P_p = \frac{\rho S}{\eta}Q^3 + \frac{\rho \Delta H}{\eta}Q \tag{5.4}$$

The efficiency of the pump is assumed to remain constant as the water flow does not alter considerably with the frequency variations practically. Therefore, Equation (5.5) can be obtained with k_1 and k_2 parameters that can be determined using actual data.

$$P_p = k_1 Q^3 + k_2 Q \tag{5.5}$$

The connection between pump speed and flow across the pump can be presented by the affinity law as stated in Equation (5.6).

$$\frac{Q}{Q_r} = \frac{N}{N_r} \tag{5.6}$$

Where, Q_r is the rated flow rate of the pump (m^3/h), N and N_r are real and rated speeds (rpm) of the pump, respectively.

Power consumption of the cooling tower fan, P_f (kW) in respect of relative rotation speed of the fan, v_f (%) is stated in Equation (5.7) [85].

$$P_f = a + bv_f + cv_f^2 + dv_f^3 \tag{5.7}$$

Using Equation (5.8), the outlet temperature of the cooling tower can be computed when neglecting the loss of heat transfer [85].

$$T_{ci} = T_{co} - \frac{c_3 G_w^{c_2 - 1}}{\left(1 + c_1 \left(\frac{G_w}{G_a}\right)^{c_2}\right)} (T_{co} - T_{wb}) \tag{5.8}$$

T_{wb} is the wet-bulb temperature of inlet air (^{0}C), G_w and G_a are mass flow rates of water and air (kg/s) respectively. a, b, c, d, c_1, c_2, and c_3 coefficients can be found by fitting real data.

5.1.2 Problem Formulation

This research work aims to minimize the total power consumption of the HVAC CW system whilst assuring all system limitations. Though the speeds of the fan and pump are taken as the ideal control variables, cooling tower airflow, and CW flow can be used as regulating variables. Due to the decreasing behavior of the system's power consumption against the increasing cooling tower air flow [85], the fan is operated at the full speed throughout all time. Besides, the water flow rate along the system, the temperature of the water flow across the chiller, and the speed of the pump should be controlled within acceptable limits. Therefore, these define the constraints for the minimization problem.

Let the total power consumption of the CW system during the considered hour is P_{tot} (kW) and that is presented by Equation (5.9).

$$P_{tot} = P_c + P_p + P_f \tag{5.9}$$

Now, minimization problem can be stated as,

$$\begin{gathered} Min\ P_{tot} = P_c + P_p + P_f \\ \text{Subject to,} \\ T_{eo,min} \leq T_{eo} \leq T_{eo,max} \\ G_{w,min} \leq G_w \leq G_{w,max} \\ \upsilon_{p,min} \leq \upsilon_p \leq \upsilon_{p,max} \end{gathered}$$

Where υ_p is the relative rotational speed of the pump (%).

5.1.3 Algorithm Development

To search for the optimal solution for the CW flow rate which minimizes the power consumption of the CW system, a novel algorithm is introduced in this section and that can be explained with the use of a flow chart provided in Figure 5.2. Instead of referring to other variables to change the water flow, the proposed strategy straightly finds the optimal flow, avoiding the approach from getting complex.

In the optimizing algorithm, cooling load demand, ambient temperature, and outdoor wet-bulb temperature are considered as the input variables. Following the ambient temperature, T_{eo} will be altered based on the defined constraints. Particularly, this adjustment is advantageous for the system's energy saving in times with low temperatures. After, T_{co}, the power consumption of each component, and the total power will be computed appropriately by assigning dissimilar water flow rates. Until the minimum power value is received, this process will be continued. In the end, the algorithm will output optimal setpoints for chilled water supply temperature and cooling water flow rate that are corresponding

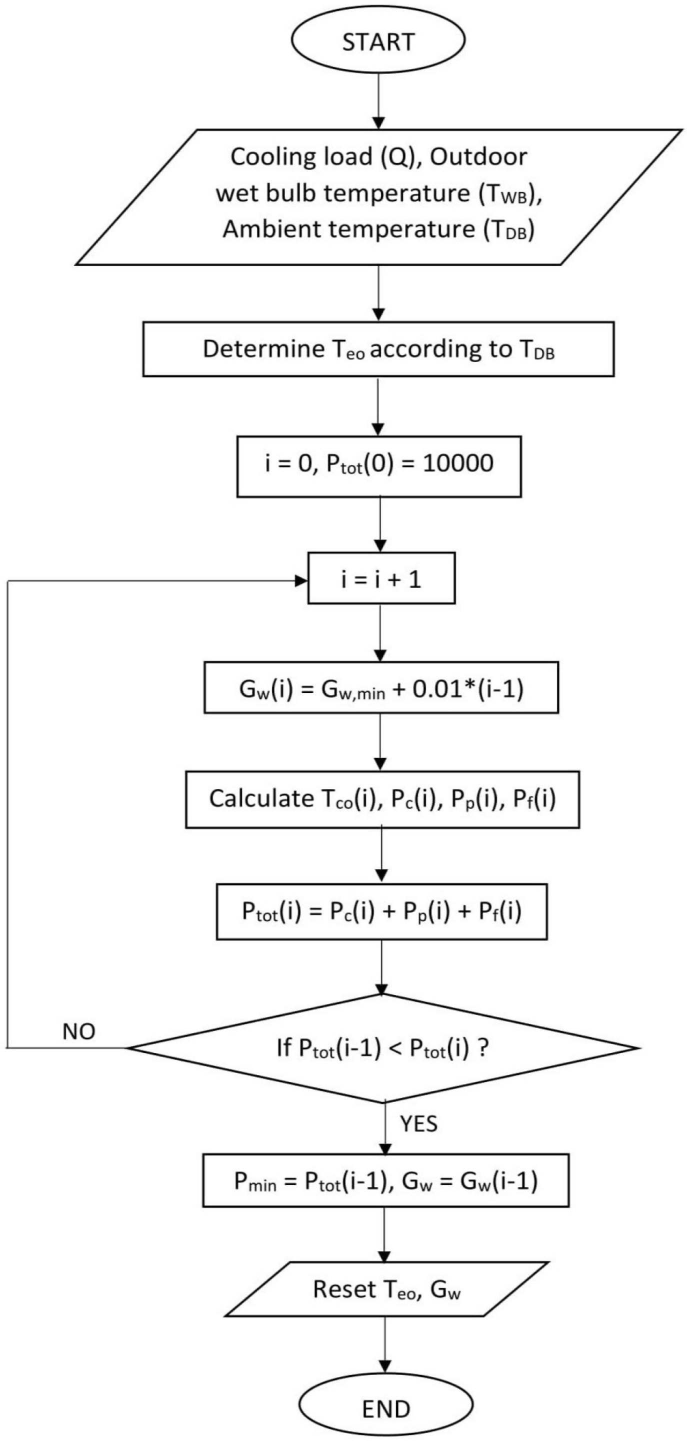


Figure 5.2: Flow Chart for the Proposed Algorithm

to the minimum total power consumption for the CW system with assured system limitations. In consideration of these values, the speed of the pump can be determined.

5.2 Simulation Results and Comparison

A simulation study was carried out by developing the suggested algorithm with the derived system model in MATLAB®. In order to allow an independent comparison of the outcomes, the same parameters utilized by the authors in [85] were applied for simulation. Hourly outdoor conditions and cooling loads were acquired by simulating a model of an office building in TRNSYS16. Hourly cooling load and outdoor temperatures for the simulated building are depicted in Figure 5.3 and Figure 5.4, respectively.

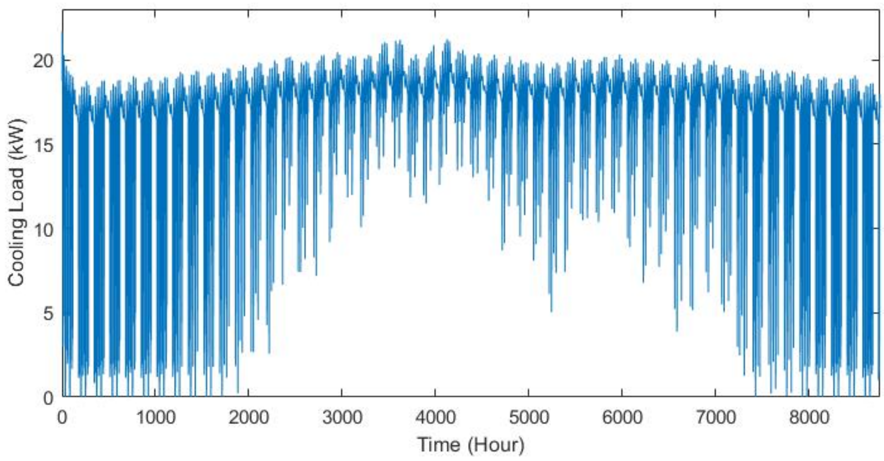


Figure 5.3: Hourly Cooling Load Variation through the Year

Due to the proven superior performance of the water flow control method introduced in [85], over to the conventional constant speed and constant temperature difference control methods, a model developed based on that method has been executed to compare with the suggested method via the simulation results. For the simulation, defined values of the parameters are, $A = 16.218$, $B = -1.067$, $C = -12.81$, $k_1 = 0.0073$, $k_2 = 0.033$, $c_1 = 1.504$, $c_2 = 1.198$, $c_3 = 1.278$, $a =$

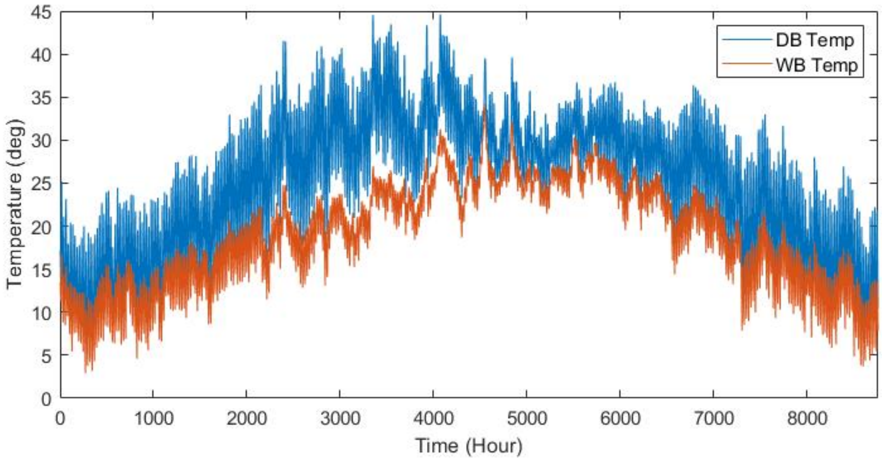


Figure 5.4: Outdoor Wet Bulb (WB) and Ambient Temperature (DB) Variation through the Year

1.789x10^{-4}, b = 2.818x10^{-4}, $c = -2.37$x10^{-5}, d = 1.83x10^{-7}, G_a = 4500 kg/s, $\rho = 997$ kg/m^3, specific heat capacity of the water (C_{pw}) = 4.2 kJ/kg^{0}C, $G_{w,min}$ = 2.76 m^3/h, $G_{w,max}$ = 12 m^3/h, $T_{eo,min}$ = 7 ^{0}C, $T_{eo,max}$ = 10 ^{0}C, $v_{p,min}$ = 50% and $v_{p,max}$ = 100%.

Figure 5.5 illustrates the variation of the optimal hourly CW flow throughout the year. A deviation between the results of the two methods can be noticed during the period with low ambient temperatures.

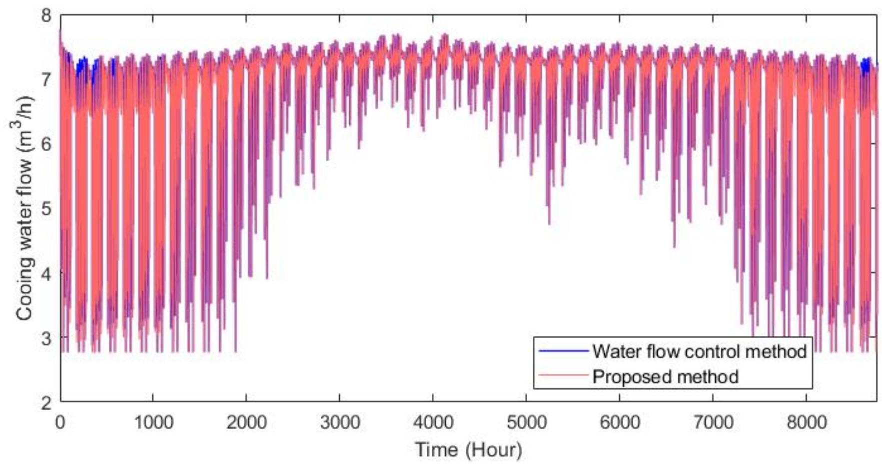


Figure 5.5: Optimal Hourly Water Flow through the Year for the Proposed and Water Flow Control Methods

The setpoint of the chilled water supply temperature has been kept at the same value throughout the year in the water flow control method. Nevertheless, as presented in Figure 5.6, the suggested method adjusts the chilled water supply temperature appropriately by taking the benefit of lower ambient temperatures.

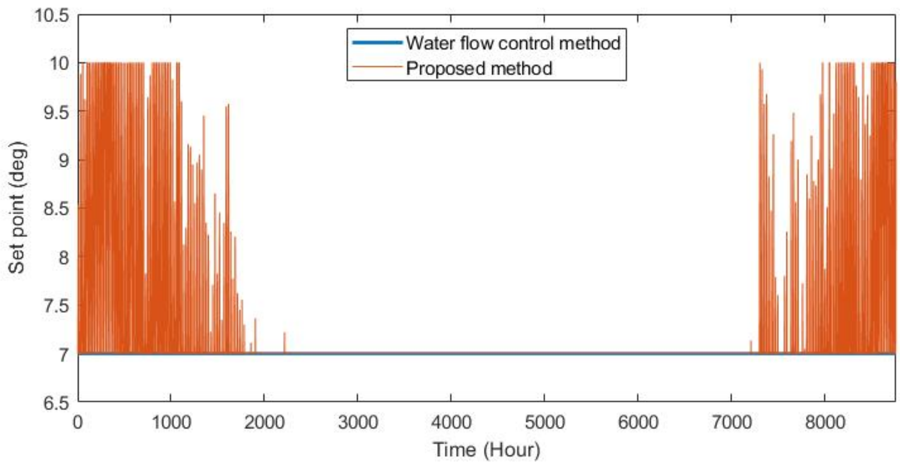


Figure 5.6: Optimal Hourly Chilled Water Supply Temperature through the Year for the Proposed and Water Flow Control Methods

As per the chiller setpoint and CW flow dynamics, power consumption is also varied. If the optimal setpoints were adjusted as resulted by the algorithm, hourly values for the minimum power consumption can be attained as depicted in Figure 5.7.

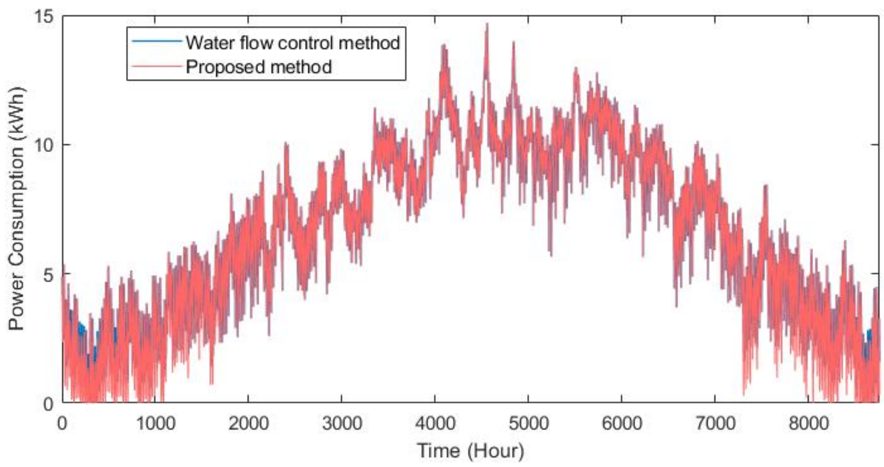


Figure 5.7: Hourly Power Consumption through the Year for the Proposed and Water Flow Control Methods

For the comparison of energy conserving potential offered by the two methods, accumulated power consumption over the year can be taken into account and that is shown in Figure 5.8.

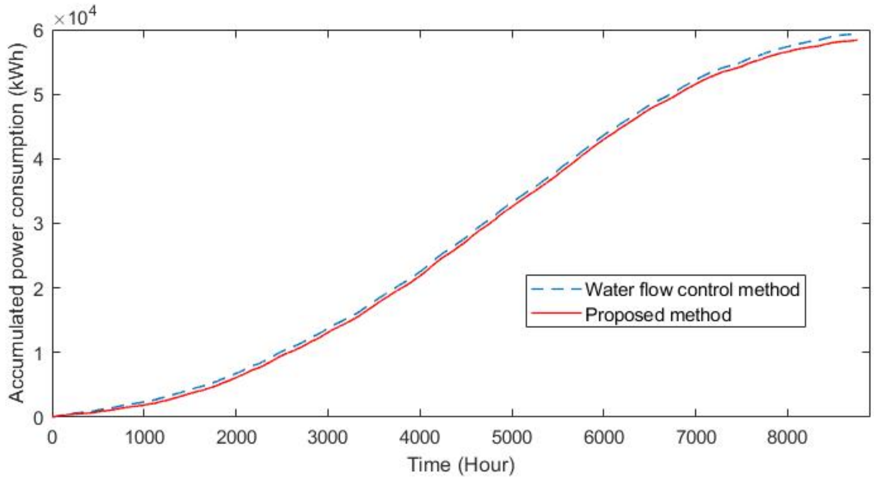


Figure 5.8: Accumulated Power Consumption through the Year for the Proposed and Water Flow Control Methods

In contrast to the water flow control method, 1078.235 kWh can be conserved from the described method. This is a 1.81% of reduction in energy consumption that is granted by the introduced strategy. If the annual Coefficient of Performance (COP) value for the CW system was considered, a 1.85% of improvement can be achieved with the proposed method. These outcomes evidence the greater performance of the presented strategy over the other typical methods.

Chapter 6

MODIFIED ELLIOT FUNCTION BASED TEMPERATURE CONTROLLER FOR AHU

This chapter presents the performance analysis of a novel temperature controller that is based on a modified Elliot function. In the AHU, supply air temperature is regulated with the help of valve opening adjustments of the heat exchanger coil and the proposed controller can be utilized for this control purpose. Its performance in the heating mode operation has been studied and compared with PD, FL, and Sigmoid function-based controllers. The system model, different control methods, and simulation results are discussed in the following subsections.

6.1 System Model

To deliver supply air with the desired temperature, a cooling or heating coil in the AHU will be operated corresponding to the cooling or heating mode of operation. Although the introducing method will work for both modes, the rest of the discussion will be based on the heating coil operation.

The heating coil is continuously supplied with steam at a higher temperature by the central boiler system. When the cold air passes across the heating coil, the heat of the steam is absorbed by the cold air via the coil surface. This results in a temperature increment in the air and a temperature decrement in the steam. The amount of transferred heat and the temperature dynamics in the air stream can be handled by altering the steam flow through the valve. This can be accomplished using a closed-loop or feedback control system that maintains the air temperature at a desired set value. A temperature sensor measures the present temperature of the output air and transmits it to the controller that calculates the deviation of that value from the set value. According to the calculated error, the controller then sends suitable signals toward the valve actuator system for taking necessary actions to achieve accurate steam flow control through the coil. This overall process is useful to preserve supply air temperature at the set value. The disturbance caused by temperature changes in the mixed air is also considered in the system model whilst assuming the constant temperature of the supply steam, constant airflow, and negligible heat storage capacity of insulating walls.

Mathematical models for heating coil, valve, actuator, and sensor are obtained using the available experimental data. Coil response to the steam flow gain = 50 ^{0}C/kg/s, time constant = 30 s, coil response to variation of mixed air temperature gain = 3 ^{0}C/^{0}C, time constant of control valve = 3 s and time constant of temperature sensor = 10 s are used to derive transfer functions and gains [97]. Figure 6.1 presents the simulation model that is developed in MATLAB® Simulink.

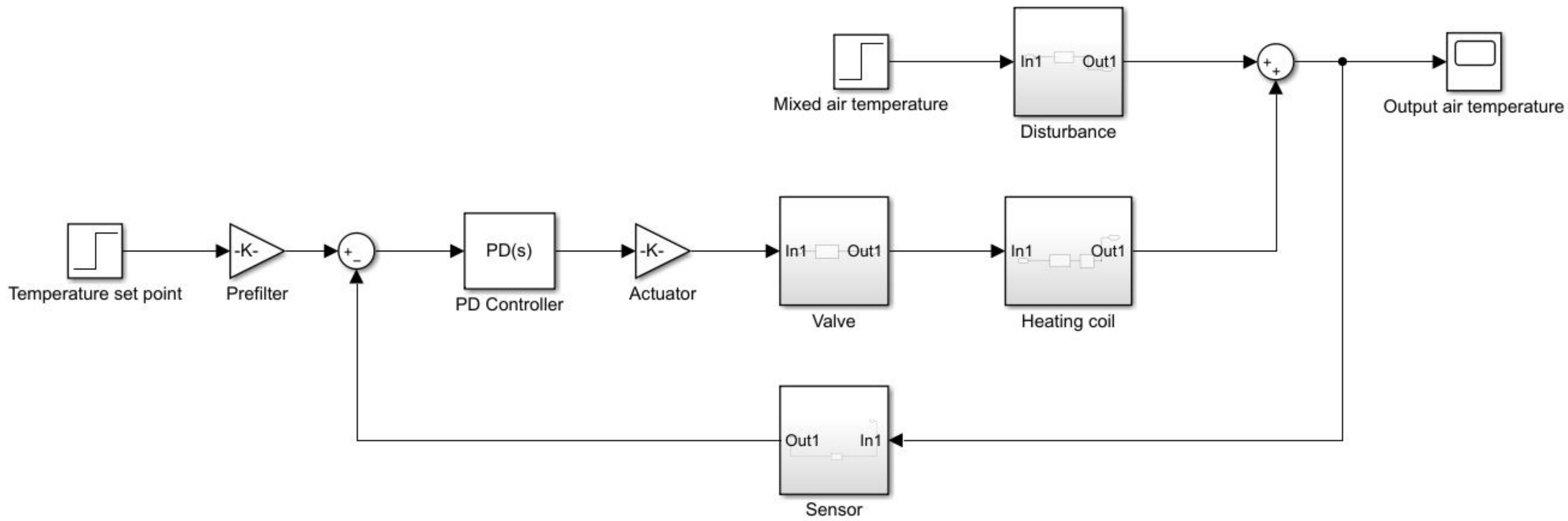


Figure 6.1: System Model with PD Controller

6.2 Different Control Methods

The suitability of the proposed controller is examined by comparing it with the performance of other three different controllers namely, PD, FL, and Sigmoid function based controllers in the same system.

6.2.1 PD Controller

Equation 6.1 gives the continuous time representation of PD controllers and the Laplace domain representation can be stated as in Equation (6.2) [97].

$$y(t) = K_p\Big(\ e(t) + T_d\frac{de(t)}{dt}\ \Big) \tag{6.1}$$

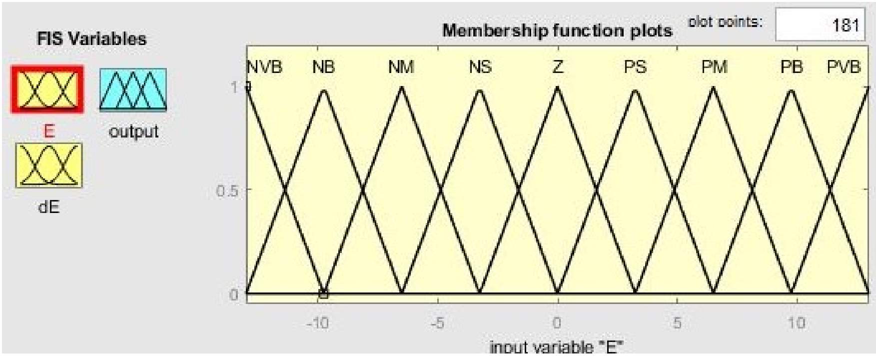
$$G_c(s) = \frac{Y(s)}{E(s)} = K_p\Big(1 + T_d s\Big) \tag{6.2}$$

As shown in Figure 6.1, a PD controller is connected to the MATLAB® Simulink system model. The tuned parameters of the controller are P = 0.45171, D = 0.001095 and N (filter coefficient) = 0.046562.

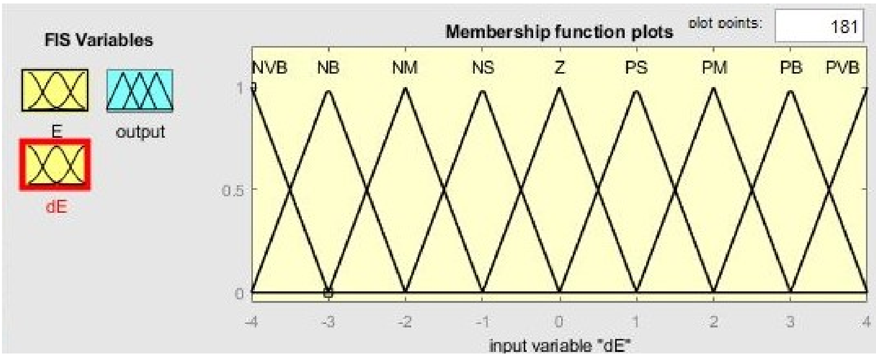
6.2.2 Fuzzy Logic Controller

An FL controller is designed in this study with two input variables and a single output variable. Error and derivative of error are the input variables while the output variable is for controller output. Next, the FIS editor in MATLAB® creates triangular membership functions as presented by Figure 6.2a, Figure 6.2b and Figure 6.2c with the ranges of [-13,13], [-4,4] and [-5,5] for error, derivative of error and output, respectively.

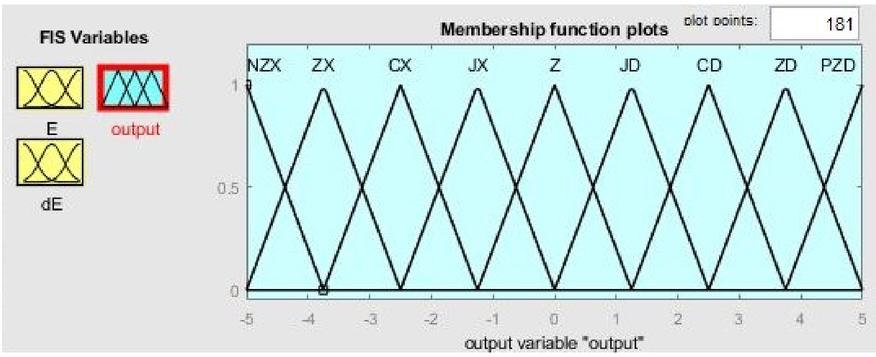
The base for the designed fuzzy rules is provided in Table 6.1 and linguistic



(a)



(b)



(c)

Figure 6.2: Membership Functions for (a) Error (b) Derivative of Error (c) Output

Table 6.1: Fuzzy Rules Designed for the Controller

dE,E	NVB	NB	NM	NS	Z	PS	PM	PB	PVB
NVB	NZX	NZX	NZX	NZX	NZX	ZX	CX	JX	Z
NB	NZX	NZX	NZX	NZX	ZX	CX	JX	Z	JD
NM	NZX	NZX	NZX	ZX	CX	JX	Z	JD	CD
NS	NZX	NZX	ZX	CX	JX	Z	JD	CD	ZD
Z	NZX	ZX	CX	JX	Z	JD	CD	ZD	PZD
PS	ZX	CX	JX	Z	JD	CD	ZD	PZD	PZD
PM	CX	JX	Z	JD	CD	ZD	PZD	PZD	PZD
PB	JX	Z	JD	CD	ZD	PZD	PZD	PZD	PZD
PVB	Z	JD	CD	ZD	PZD	PZD	PZD	PZD	PZD

variables for inputs are selected as {negative very big (NVB), negative big (NB), negative medium (NM), negative small (NS), zero (Z), positive small (PS), positive medium (PM), positive big (PB), positive very big (PVB)} while variables for output are selected as {minimum (NZX), little (ZX), very small (CX), small (JX), medium (Z), large (JD), very large (CD), more (ZD), maximum (PZD)}. This FL controller applies the Mamdani algorithm and works out the fuzzy with the max-min method with centroid defuzzification. A 3-D surface view of the variables is given in Figure 6.3 and Figure 6.4 shows how to link the FL controller with the system.

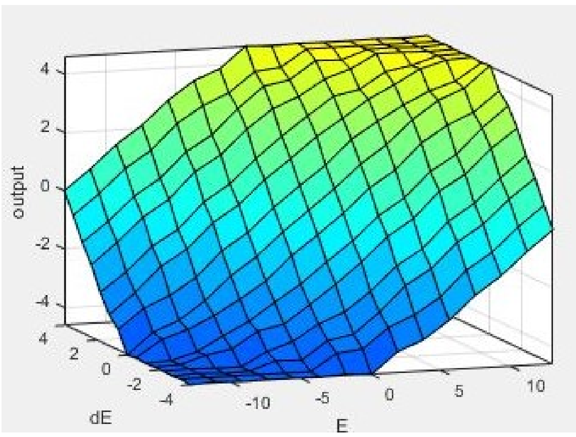


Figure 6.3: 3-D Surface of the Variables

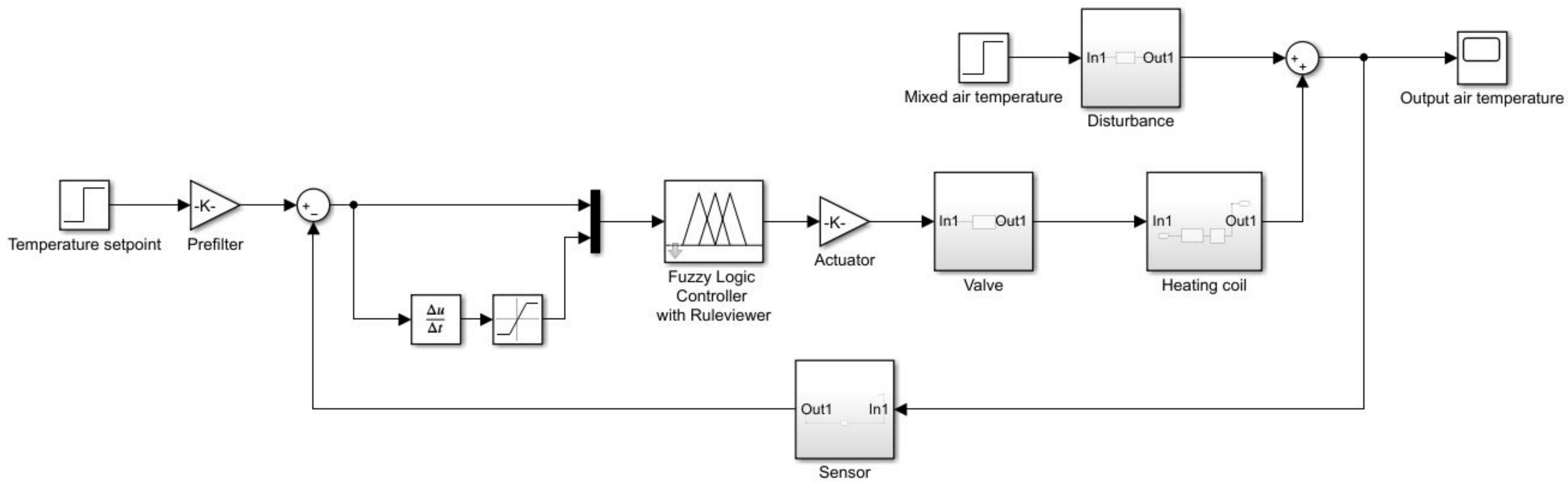


Figure 6.4: System Model with FL Controller

6.2.3 Sigmoid Function Based Controller

The output of any value that is input to the symmetric Sigmoid function will be within the range of [-1,1] [110]. This implies some similar characteristics between this type and a PD controller. Inspired by this, we analyzed its feasibility as a temperature controller. Equation (6.3) gives the simplified equation for the symmetric Sigmoid function and Figure 6.5 illustrates how it is applied in the system model.

$$f_s(x) = \frac{1 - e^{-2x}}{1 + e^{-2x}} \tag{6.3}$$

6.2.4 Proposed Controller

In this section, a modified Elliot function based controller is proposed for temperature control in AHU. According to the basic equation for the Elliot function that is provided in Equation (6.4), the output for any given value is limited to be between 0 and 1 [110].

$$f_e(x) = \frac{1}{2} + \frac{\frac{1}{2}x}{1 + |x|} \tag{6.4}$$

Equation (6.4) can be modified as stated in Equation (6.5) such that the output domain is widened to [-1,1].

$$f_{me}(x) = \frac{x}{1 + |x|} \tag{6.5}$$

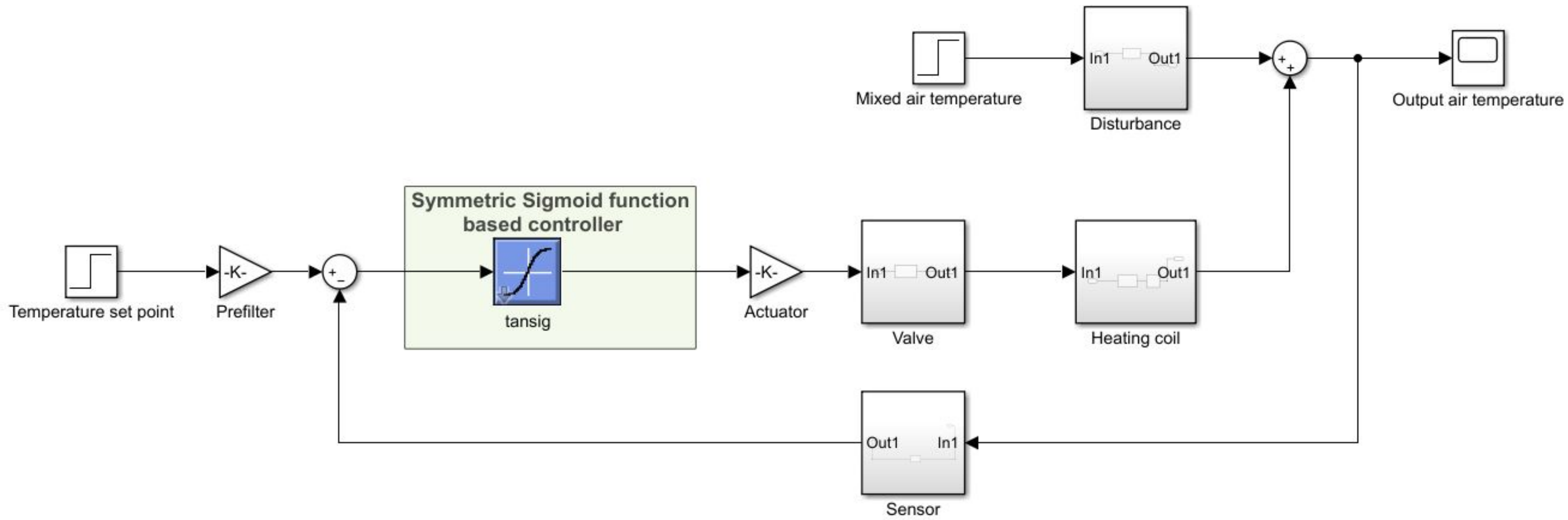


Figure 6.5: System Model with Symmetric Sigmoid Function Based Controller

Figure 6.6 depicts the characteristic curves for symmetric Sigmoid, Elliot, and modified Elliot functions. It is observed that the Elliot function has been improved symmetrically over the range performing very closer to the symmetric Sigmoid function as a result of the modification.

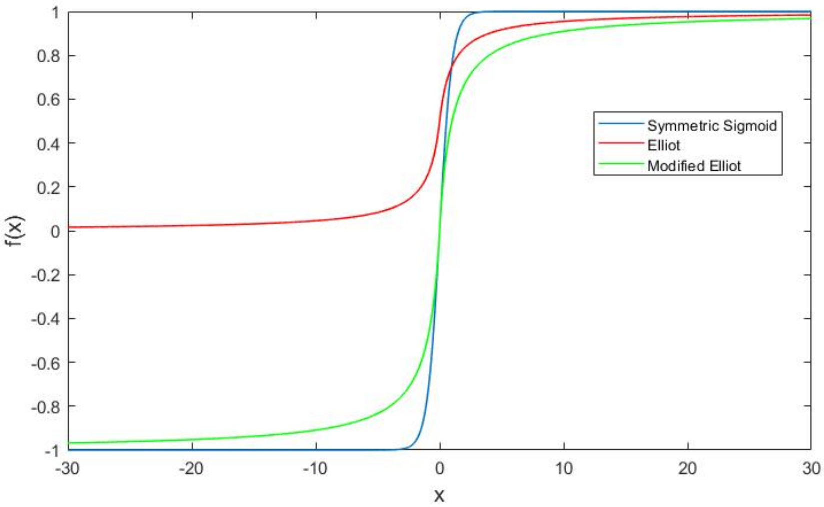


Figure 6.6: Characteristic Curves for Symmetric Sigmoid, Elliot, and Modified Elliot Functions

The modified function is utilized in this study to introduce a novel temperature controller and that is presented in Figure 6.7.

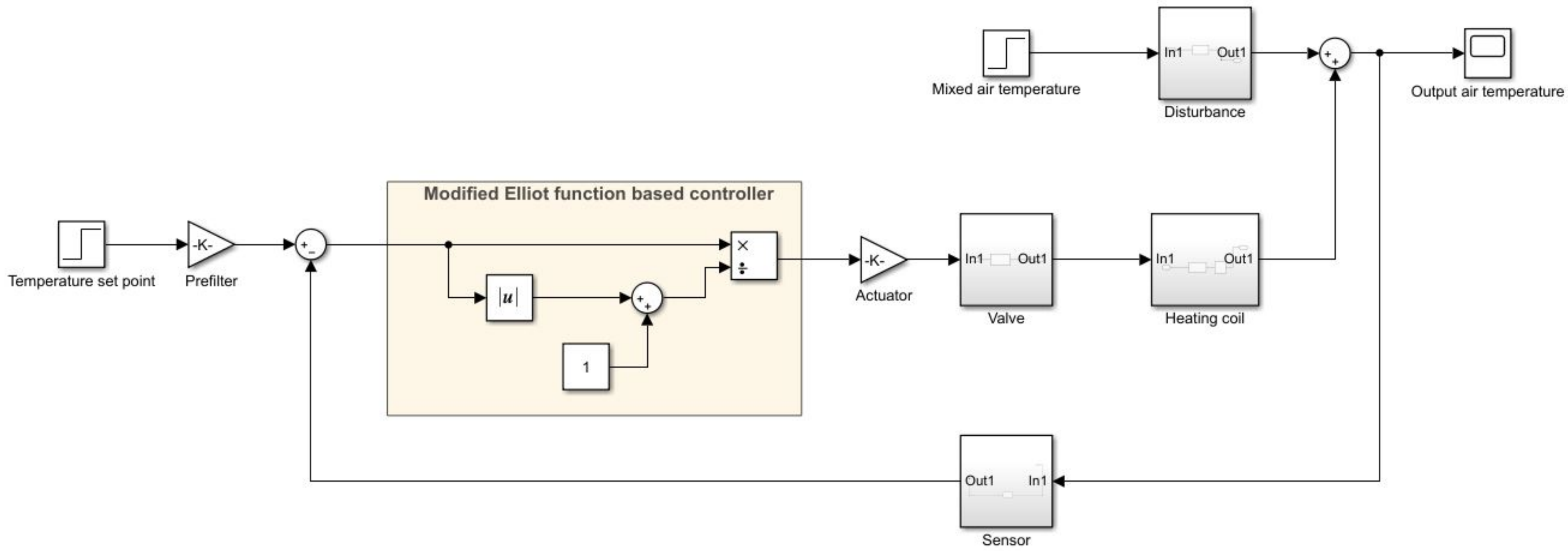
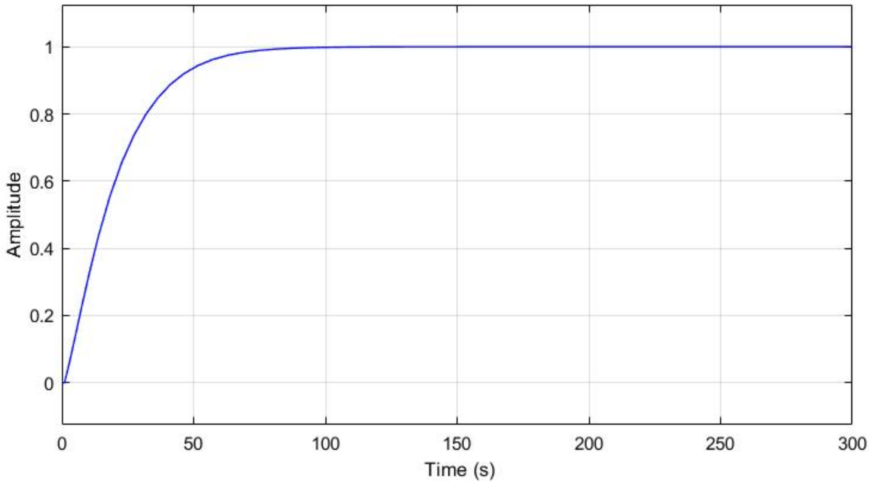


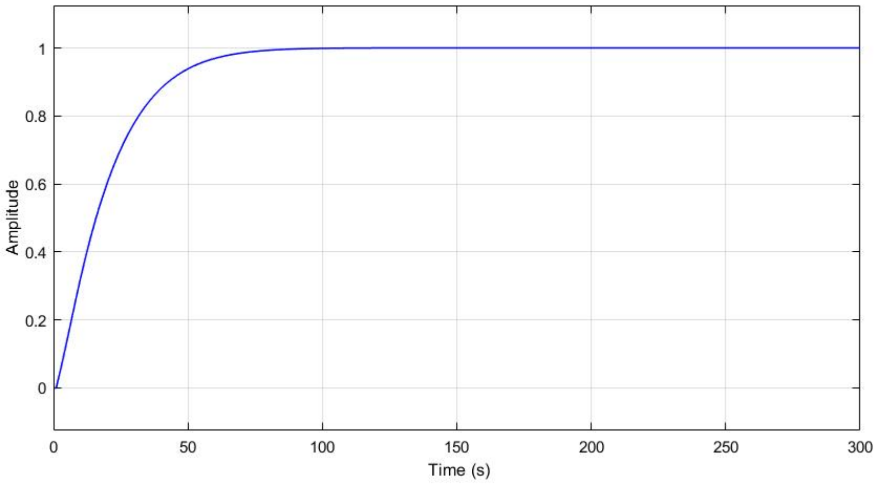
Figure 6.7: System Model with the Proposed Controller

6.3 Simulation Results and Comparison

Above four different temperature controllers were applied in the same MATLAB® Simulink system model. To analyze the performance of each controller, step input was introduced at both ends of setpoint temperature and mixed air temperature. Figure 6.8a, Figure 6.8b, Figure 6.8c, and Figure 6.8d illustrate the step response of PD, FL, symmetric Sigmoid function based, and modified Elliot function based temperature controllers, respectively.

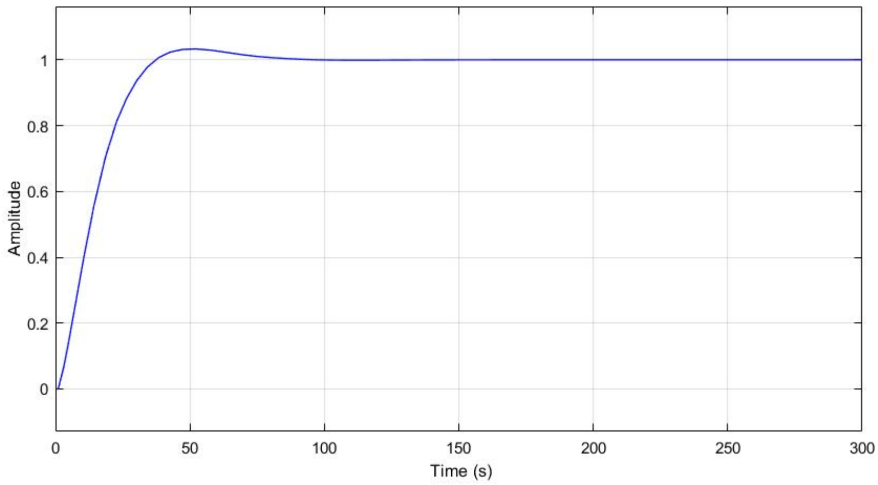


(a)



(b)

Figure 6.8: Step Response of (a) PD Controller (b) FL Controller (c) Symmetric Sigmoid Function Based Controller (d) Modified Elliot Function Based Controller



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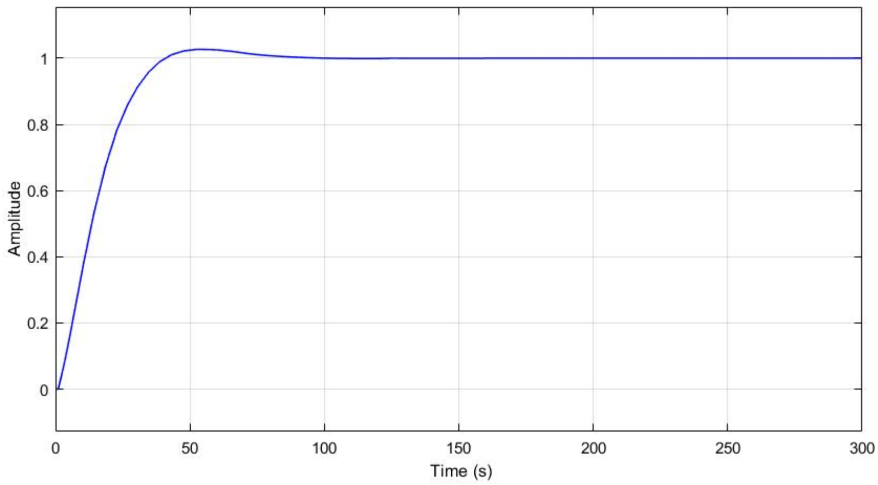

(d)

Figure 6.8: Step Response of (a) PD Controller (b) FL Controller (c) Symmetric Sigmoid Function Based Controller (d) Modified Elliot Function Based Controller (cont.)

Based on the results, the FL controller gives a smoother response than the other controllers. Even though the responses of PD and FL controllers are nearly the same, the PD controller introduces a small undershoot (1.134%) as well. However, the exhibited step responses of symmetric Sigmoid and proposed controllers are also closer to the FL controller.

Table 6.2 provides a comparison of the performance of each controller in terms of settling time, rise time, maximum overshoot, slew rate, and execution time. The settling time for the proposed controller is 4.05% smaller than the PD controller and 33.87% higher than the FL controller. Nevertheless, the rise time is 33.30% and 32.65% less than the PD and FL controllers, respectively. The least maximum overshoot (0.50%) is given by the PD controller while the symmetric Sigmoid function based controller records the highest value (3.65%). In the meantime, the maximum overshoot given by the proposed controller is 1.07% lower than the symmetric Sigmoid function based controller and 2.08% greater than the PD controller. Though performances of symmetric Sigmoid and modified Elliot functions based controllers are almost similar, the proposed controller offers a 4.8% reduced settling time. It also can be observed that the FL controller that offers better performance concerning settling time and overshoot, takes longer execution time in contrast to the other controllers. As per the results, the execution time for the proposed controller is remarkably small, and it is 20876 times smaller in comparison with the FL controller. Due to the manifested sufficient performance with faster execution, the suitability of the proposed temperature controller in the field of HVAC can be emphasized.

Table 6.2: Performance Comparison of Different Controllers

Parameters	**Controllers**			
	PD	**Fuzzy Logic**	**Symmetric Sigmoid Function Based**	**Modified Elliot Function Based**
Settling time (s)	175.167	125.544	176.541	168.068
Rise time (s)	38.385	38.014	23.498	25.603
Max.m overshoot (%)	0.500	0.505	3.646	2.577
Slew rate (/ks)	20.633	20.835	33.759	31.111
Execution time of the controller (s)	0.002	104.380	0.010	0.005

Chapter 7

CONCLUSIONS

Chapter 3 presented a novel algorithm to select the optimal number of parallel pumps to be operated and their optimal operating speeds with an aim to optimally decrease the power usage of HVAC chilled water pumps whilst essentially satisfying the inevitable operational constraints and variable flow requirement imposed by the varying thermal load conditions of the building. After formulating the power usage minimization problem, a step-wise approach was used to solve it with low complexity using an algorithm that is developed using MATLAB®. The simulation results showed that for large chilled water systems, parallel pumps are more beneficial compared to a large single pump. As per the case study in Subsection 3.3.1, the utilization of two and three parallel pumps offered 11.93% and 6.60% energy savings respectively compared to the large single pump. Also, when there are multiple parallel pumps, optimal scheduling with different operational speeds were proved to optimally minimize the power consumption. Outcomes of this case study verified the performance of the suggested method by giving 41.62% and 75.91% consumed energy reductions for two and three parallel pumps respectively. The case study in Subsection 3.3.2 resulted in a 15.26% decrement in energy consumption compared to the conventional method by applying the algorithm that was introduced. The case study in Subsection 3.3.3 provided a performance comparison of the proposed method with the PSO method. In the simulation studies, the suggested strategy offered a 1.78% of energy saving over to the PSO method with a comparatively small execution time. Therefore, a highly efficient, safe, and reliable pump operation can be assured with the proposed method. Owing to its simplicity, reduced computational complexity, less computational time, ease of implementation, and reliability, the proposed algorithm is applicable in real-world applications to optimize the energy performance in industrial and commercial smart buildings.

Minimizing the HVAC energy consumption in buildings without compromising on thermal comfort and productivity is a challenging research problem. To address this problem with reduced complexity, the Dijkstra technique based novel strategy has been proposed in Chapter 4. The research problem was defined to be obtaining an optimal temperature setpoint schedule, which reduces the energy consumption and the cost of electricity in a thermal zone while assuring occupants' thermal comfort and productivity at a higher level. The optimization algorithm that was developed in the MATLAB® environment takes the outdoor temperatures and RTP of electricity during the scheduling period as the inputs. With the help of building models and the CBE thermal comfort tool, the proposed method provides the optimal temperature setpoint schedule for a particular thermal zone. Computer simulation results obtained for a single AHU of a university building in Singapore yielded a 3.84% saving in the total energy consumption and a 4.22% reduction in the total cost of electricity with a 0.06% improvement in RP. In a commercial building, the number of AHUs can be as many as several hundred in total. Hence, energy and cost saving that can result in large commercial buildings is significant. Most importantly, this is achieved without compromising occupant comfort and productivity. Lesser computational complexity, ease of implementation, and higher reliability are additional advantages offered by the proposed novel strategy.

As a noteworthy contributor to the continuous growth in worldwide energy usage, HVAC CW systems also essentially need energy management strategies to diminish their energy consumption. Chapter 5 suggested a state-of-the-art approach to achieve this requirement using a CW flow and chilled water supply temperature optimization along with system constraints, cooling load dynamics, and varying outdoor conditions. Its better performance compared to other conventional methods as to power consumption and average system COP can be emphasized through the simulation results. In comparison with the water flow control method, the proposed strategy has been able to offer a 1078.235 kWh of reduction in annual total energy as well as a 1.85% of improvement in the average COP. The simplicity, lower computational time than the other advanced algorithms and numerical methods, reliability, robustness, and efficiency of this introduced algorithm attract real building applications that are minding to enhance buildings' performance.

In Chapter 6, a modified Elliot function-based temperature controller has been introduced for the heating mode operation of AHU. PD, FL, and symmetric Sigmoid function-based controllers with the developed model in MATLAB® Simulink were simulated to examine the performance in terms of settling time, rise time, maximum overshoot, and execution time. Results witness that the proposed Elliot function based controller performs well by reaching closer to the FL controller's performance with 20876 times lower execution time. Besides, the proposed controller is simple, easy to implement, more efficient, and less expensive. Therefore, this allows the HVAC systems to reduce energy usage along with an efficient operation in contrast with other controllers.

7.1 Integration with Building Management Systems

Earlier, buildings were very simple since they had few facilities and services. With the introduction of many services and increasing occupancy, buildings also have become more complex day by day and this results in higher energy consumption. In order to enhance a building's energy performance, the need for central monitoring and control systems can be emphasized. Building Automation Systems (BAS) or BMS is the latest solution that addresses those problems while offering many other advantages such as the efficient operation of building systems, improved life cycles of utilities, and a decrease in operating costs and energy consumption. Since BAS is a computer-based control system, it can be connected with almost all the systems and equipment established in a building to facilitate many services.

Attaining ultimate benefits from the HVAC energy saving strategies that are proposed in this research can be accomplished by the proper integration with BAS in the considered building. Generally, protocols such as Open Platform Communications (OPC) and Transmission Control Protocol (TCP) are utilized to link these control algorithms with BAS. Real-time data for the algorithm can be collected directly by accessing the supervisory controllers through relevant protocols. However, if the developed algorithm needs historical data for processes such as predictions, accessibility for databases will also be needed. Microsoft Structured Query Language (SQL) databases can be used in such cases. When embedding novel algorithms developed in MATLAB® to a controller, sometimes installing addons or converting to intermediate language may be required. Finally, the output of the developed control algorithms can be delivered to appropriate actuator systems by the controllers via the previously mentioned protocols.

The developed algorithm can be stored either in a physical device or in a cloud platform. Cloud platforms are more reliable than physical devices and easily accessible from any location. It will be an advantage for a client who has several buildings to optimize. However, it is more secure on physical devices because of the risk of data theft.

Figure 7.1 explains how the integration of these systems appears in the BAS architecture.

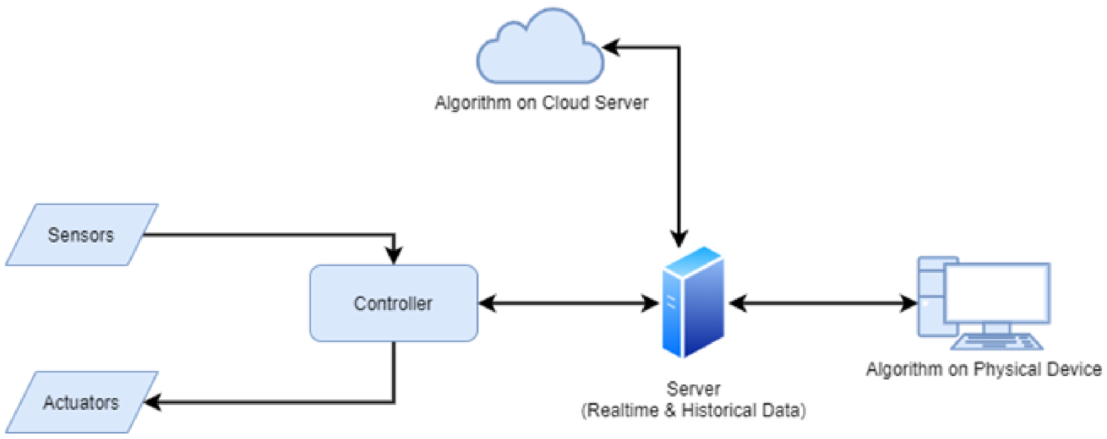

Figure 7.1: Integration of Developed Systems with the BAS

7.2 Limitations

Even though we observed the satisfactory performance of the introduced novel methodologies for HVAC systems in the simulation studies, identifying any limitations behind them is also useful for future developments.

For the energy saving algorithm suggested for secondary chilled water pumps, the required chilled water flow rate for a particular time slot is supposed to be calculated using a control signal output from the temperature control system. This may occur errors in the calculated values if there was any fault in the relevant temperature control system.

During the research study carried out for temperature setpoint management in AHUs, occupant characteristics such as metabolic heat production and clothing level are assumed to be kept constant since the considered zone does not have a varying occupancy. Therefore, when implementing this methodology for a building with highly dynamic occupancy, some modifications may be required to encounter these effects.

When moving to the CW system energy optimization, is achieved by deriving and utilizing semi-empirical system models with ignoring the loss of heat transfer at system components for the simplicity of simulation studies. In practical situations where heat transfer losses cannot be neglected, necessary corrections may need to be incorporated before applying.

Performance analysis of modified Elliot function based controller is studied only for the heating mode operation of AHUs. However, it is better to have a study on cooling mode operation as well to identify any potential issues. Apart from that, this controller also shows a small overshoot and further tuning can be considered to have more reductions.

7.3 Recommendations for Future Developments

For the identification of further possible improvements and to enhance the performance, the developed algorithm for secondary chilled water pumps can be combined with the temperature control system which creates a complete energy management and control system. Besides that, to overcome the possible errors affected by the temperature control system, another distinct method to directly calculate the flow requirement can be designed in future works.

Another interesting future direction may be to develop similar approaches to the one presented in Chapter 4, for energy optimization via chilled water and condenser water temperature resetting scheduling in the chiller and cooling tower plants of the building HVAC systems.

In addition, optimization of chiller setpoints considering both outdoor conditions and cooling load can be researched to enhance the CW system efficiency during future developments.

Furthermore, modifications required to improve the current performance and feasibility of the introduced controller that is based on a modified Elliot function in other applications also can be investigated to make popular this simple controller in the industry.

LIST OF PUBLICATIONS

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1. S. Walgama, S. Kumarawadu and C. D. Pathirana, "Performance Analysis of Modified Elliot Function Based Temperature Controller in the Heating Mode Operation of AHU," 2021 Moratuwa Engineering Research Conference (MERCon), 2021, pp. 89-94, doi: 10.1109/MERCon52712.2021.95257-70.

2. S. Walgama, S. Kumarawadu and C. D. Pathirana, "Indoor and Outdoor Conditions Utilized Energy Saving Scheme for HVAC Cooling Water Systems in Smart Commercial Buildings," 2021 IEEE Electrical Power and Energy Conference (EPEC), 2021, pp. 137-142, doi: 10.1109/EPEC52095.202-1.9621546.

3. S. Walgama, S. Kumarawadu and C. D. Pathirana, "Optimal Temperature Setpoint Scheduling Method for AHUs Using Dijkstra Method" (Submitted to the Journal of Energy Efficiency and pending review)

4. S. Walgama, S. Kumarawadu and C. D. Pathirana, "A Step-Wise Approach Based Optimal Energy Utilization Scheme for HVAC Secondary Chilled Water Pumps in Smart Commercial Buildings" (Submitted to the International Energy Journal and pending review)

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