Design of Natural Ventilation Optimal Control System to Achieve Desirable Room Temperature and to Tolerate Uncertain Variants

by

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ABSTRACT

Natural ventilation models are becoming increasingly popular with increasing environmental awareness and the adverse effect of mechanical ventilation systems on the environment. Though simple in their working principle, natural ventilation models are challenging to control in meeting the thermal comfort demands of the occupants in changing weather conditions. This requires robust environment control and precise climatic parameters prediction for satisfactory performance. This study aims to design and assess the performance of optimal controllers for natural ventilation. Three optimal controllers, Linear Quadratic Regulator (LQR), Linear Quadratic Integral (LQI), and Linear Quadratic Gaussian (LQG), have been developed and assessed for their performance for thermal comfort for a naturally ventilated modeled with its window as the control input. In addition to these three controllers, another more efficient optimal predictive controller known as Model Predictive Controller is also designed and applied to the natural ventilation system. The three controllers are also compared with a Model Predictive Controller (MPC). Considering natural ventilation as a natural process with slower changing dynamics and subjected to various external and internal system uncertainties, the MPC is shown to outperform the other optimal controllers in terms of robust temperature prediction and control, efficient disturbance rejection, and uncertainty handling. Another challenge addressed by the study is the development of a simplified mathematical model that can be utilized for building a robust real-time controller. With an efficient runtime of 0.08 sec per optimization, the MPC shows robust temperature control by providing thermal comfort to users. This is achieved with a Predicted Mean Vote (PMV) index of 0.0013, which shows the controller performs effectively to keep the thermal comfort of the occupants to an acceptable level.

PREFACE

I, Muhammad Ahmad Hameed, declare that all this thesis/research work is original and done by me. The simulation and results are generated based on my research work. No part of this research has been previously published.

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First of all, I want to express my sincere gratitude to my supervisor, Prof. Dr. Zhan Shu, for his wise counsel, ongoing encouragement, and tolerance during my MSc studies. My academic research and daily life have both benefited greatly from his vast knowledge and wealth of experience. I would like to convey my appreciation to all the instructors who taught me various subjects throughout my study duration. Finally, I want to thank my parents, my wife, and my friends. Without their incredible support, especially during the COVID pandemic, I would not have been able to finish my studies and my thesis.

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CHAPTER 1: INTRODUCTION

1.1 Background and Motivation

Since mechanical ventilation, cooling, and heating systems became a standard by the latter half of the twentieth century, little development has been made toward improving natural ventilation models. However, the need for natural ventilation systems has resurfaced due to the recent surge in environmental impact and carbon footprint of mechanical systems. This has led to resuming research towards the development of natural ventilation models that are practicable and efficient enough to offer their maximum potential. Natural ventilation provides sufficient cooling to replace a mechanical ventilation system in places with feasible outside temperature conditions or in the cooler months of the year. This comes with the advantages of increased air quality due to fresh air and lowers CO_2 level, which is further associated with increased occupant productivity [1] and reduced energy consumption. Whereas a typical mechanical system consumes about 50% - 60% of a building's total energy consumption [2], the energy consumed by a natural cooling and ventilation system is half of this percentage [3]. Further, natural ventilation has a role in reducing CO_2 emissions associated with cooling processes. These reasons make natural ventilation a potential choice by the research and design community to play a crucial role in designing zero-energy buildings.

Many governments, as well as businesses and organizations around the world, have started to work towards an eco-friendly ventilation system, working tirelessly towards a zero-carbon emission environment with a well-organized infrastructure [4]. [5] has given a comprehensive contemporary building design literature suggesting how modern infrastructure is shifting again towards increased use of natural ventilation to maintain building temperatures. This will help achieve a goal towards the single aim of reducing global warming by reducing the number of harmful gases produced in the free air.

The design of natural ventilation system embodies two approaches. One is based on building code requirements for which four to five percent of the occupied floor area has to serve as an operable vent area. This standard requirement has been followed for decades and has provided sufficient ventilation. However, this approach ignores the issue of whether and when these vents are open or not, how much ventilation these vents can provide, and their ability to provide homogenous ventilation under variable climates. A recent approach in line with green and sustainable buildings suggests operable openings (such as windows or doors), as the critical control factor to maintain indoor air quality and thermal comfort in a form of maintained temperature. These openings can be designed in the form of windows, shafts, or clerestories while considering engineering design (building design, control techniques, etc.) and climatic and geographical requirements of the system.

Despite all the advantages and demands of natural ventilation, there still exist crucial challenges to the applicability of such a system that alone would meet the cooling requirements, specifically of commercial buildings. The two driving forces for natural ventilation, wind, and buoyancy, are highly fluctuated variables in intensity and quantity throughout the year. For effectiveness and practicality, natural ventilation and cooling systems are required to be designed for worst-case scenarios with increased turbulence, wind speed, and pressure fluctuations. Also, the primary control input for these systems is ventilation channel operation (a window, a louver, a duct, and a fan), which is a function of occupant behavior. Occupant behavior is highly unpredictable owing to the geographical, cultural, and social dynamics of the places the system is modeled for [6]. These challenges make a natural ventilation system a complex model to control.

Though various control techniques have evolved over the past few decades for all kinds of natural systems' active controls, choosing a control strategy for a natural ventilation system is still an open question. Starting from simple PID control [7], ventilation models are now being controlled using all classical and modern control techniques, including adaptive control [8], optimal control [9], rule-based control [10], predictive control [11], and hybrid control schemes incorporating combinations of these controllers. Owing to its highly uncertain dynamics, natural ventilation control becomes a challenging problem when dealing with conventional rule-based heuristic approaches. Accurate prediction requires the implementation of optimal and predictive control strategies for such systems. Where a variety of optimal control exists for ventilation control of mechanical HVAC systems, their implementation on a slow dynamics system such as natural ventilation needs to be explored further to deduce their efficacy for such uncertain, slow dynamics models. Slow dynamic system challenges are that they only rely on the natural environment without using any mechanical system and the process of ventilation takes a lot of time as fresh air only enters or leaves a building naturally from building opening areas, so the controllers are to be designed in a way that they can be applied to the best effective part of natural systems with which environment in a form of thermal comfort can be controlled most effectively. Also unlike mechanical systems where optimal controllers are designed for the specific mechanical system where each mechanical system has their own data-driven models and controllers are specific for that airflow model as air can be exchanged mechanically without depending upon building physical properties, slow-driven systems cannot only rely on data-driven models of ventilation but it also greatly depends on building physical models which is also a challenge as their air has to be exchanged naturally based on building properties.

This research aims to study, design, and assess optimal and predictive controllers for the slow dynamics system of natural ventilation. For this, the study is dedicated development of a simple yet efficient system model design that incorporates all crucial physical, atmospheric, and control variables of a naturally ventilated system. This includes building a model, model linearization through real-time control implementation of initial conditions and constraints, and identification of system inputs, states, outputs, and disturbances. Further, all three optimal and predictive controllers are designed and simulated in the state-of-the-art MATLAB interface for their performance assessment and occupant thermal comfort criteria achievement. The purpose is to analyze the controllers' performance and their suitability and efficacy in maintaining desired conditions for a highly uncertain and slow-responding natural system. The suitable controller choice is crucial in increasing system efficiency, reducing energy costs, and improving the building environment while affecting the environment to its minimum.

1.2 Thesis Objectives

This study is dedicated to the design, testing, and performance analysis of optimal and predictive controllers for a naturally ventilated system model. While the focus is on developing the optimal controllers, obtaining a sufficient system model to account for the system's crucial dynamics and associated uncertainties is also a challenge that this study addresses. This thesis, therefore, addresses the following objectives:

1. Identification of appropriate building model(Cross ventilated, Stack, ventilated or Single sided ventilation building model) to be used for natural ventilation.

2. Developing a sufficient mathematical model (white box, black box, or grey box model) to account for the system's dynamics, along with identifying and approximating primary and secondary parameters of heat exchange using the heat balance method in the building. This is about identifying the factors related to heat exchange in a form of system inputs, system outputs, and uncertainties in the building which impact the internal temperature.
3. Development of optimal and predictive controllers separately, (LQR, LQI, LQG, and MPC) for the designed system and applying on system input (windows) to obtain thermal comfort inside the room by handling uncertainties.
4. Implementation and evaluation of these control strategies for accurate response prediction and robust control performance in a form of thermal comfort by maintaining internal temperature.

1.3 Thesis Layout

The thesis is structured as follows:

Chapter 2 elaborates on the literature available related to natural ventilation and control of naturally ventilated systems. It discusses the basic concept of natural ventilation and the developed types of natural ventilation models. The design factors that play a crucial role in deriving the natural ventilation needs and subsequently determining the type of ventilation model are discussed in comprehension. It further extends to a review of natural ventilation system modeling and the criteria that need to be considered while designing a natural ventilation system, like building environment quality and the relation of control systems' effectiveness with the occupants using that building. Control design techniques of such systems adopted in literature are also discussed in detail. This review sets the basis of the system modeling approach, controller selection, and design process.

Chapter 3 is designated to the mathematical modeling of the system. The chapter briefly discusses the critical factors of indoor and outdoor environments. Choices for both building's geometrical modeling and the system's thermal modeling are discussed. The chapter further elaborates on the thermal modeling approaches available in the literature. With a review of the building, thermal modeling, and design options, the system model of a single-zone single-sided ventilation system is developed. The chapter presents the state space model of the system's dynamics to be used for controller design, and the steps of developing the linearized thermal model are illustrated in detail. After building and thermal modeling, the thermal comfort model is discussed and derived that is to be used for controller performance assessment, and that can be used to represent occupant feedback and satisfaction.

Chapter 4 is focused on controller design criteria and controller design for modeled natural ventilation systems. Control schemes of the optimal and predictive controller are discussed in detail to build the basis of the control designs. For pure optimal control design, three controllers based on LQR, LQI, and LQG are modeled and focused on handling the system's uncertain dynamics. To account for accurate prediction, a Model-based Predictive Controller is designed.

Chapter 5 illustrates the simulation environment, results, and analysis of the results obtained for all the control techniques designed in chapter 4. The four controllers are tested and analyzed in terms of uncertainty handling, outside temperature prediction, and better temperature control performance.

Chapter 6 concludes the study by discussing the achievements and limitations of the research carried out and projecting the future dimensions of the work. The chapter consists of concluding thoughts on extensive data modeling for the outdoor temperature and heat gains due to internal load and solar radiation. When applied to a real-world problem, additional notes are given on disturbance and constraint modeling for robust control.

CHAPTER 2: LITERATURE REVIEW

Ventilation systems are used worldwide in efforts to maximize energy efficiency for the heating, ventilation, and cooling of buildings. Their wide usage now constitutes 50% of global power usage. This means that the lion's share of building energy consumption is attributed to heating, ventilation, and cooling systems (HVAC). Around 30 to 50% of the electricity supplied to the domestic sector is consumed for ventilation purposes [12]. There are mainly three types of ventilation systems based on distinct working principles: natural, mechanical, and hybrid systems, which is a combination of both [13]. Natural systems that were traditionally employed are driven by natural forces such as wind pressure or temperature difference to ventilate the room. Therefore, the influencing factors in these systems are climate, human behavior, and the design of the building. Natural ventilation has the capability to reduce 20 to 25% of electricity consumption [14]. Hence designers create purposeful openings such as doors, windows, and exhaust chimneys to create a draft between fresh air and room air. Stack ventilation is a naturally driven system based on the temperature difference between the inside and outside air [15]. Hot air rises, whereas cold air descends. Taking advantage of this principle, exhaust openings are installed on the building roof and floor so that fresh air can enter from the ground floor and exit from the roof. Another type, cross ventilation, is a wind-driven ventilation system governed by wind behavior and speed [16].

In mechanical ventilation, the air is circulated by mechanical devices such as fans, air conditioners, and inverters. These systems can optimize the air flow rate and direction as per the requirement, thereby providing more control to the user. Mechanical ventilation systems can be operated for positive and negative pressure depending on the weather and need. For example, in winter, the room is operated under positive pressure so that air flows outside the room. In summer, the room is under negative pressure to facilitate the flow of air inwards. Individually natural and mechanical ventilation modes are inefficient or costly to operate. Therefore, the integration of these modes for an optimum hybrid solution is more desirable by the users for profitability [17].

Myriad researchers have explored the natural, mechanical, and combined modes of ventilation in their research work [18], highlighting the potential of saving electrical energy in Germany, Italy, and Turkey for various opening types and apertures in naturally ventilated buildings. Another study manifests the advantages of using cross ventilation and stack effect to maintain the air quality of a classroom in Spain [19]. The effect of airflow types such as laminar and turbulent flow along with eddy currents is investigated by [20] in their study. The study reveals that a high Reynolds number with a turbulent flow type provides good natural ventilation. An extensive study was conducted by [21] to understand the airflow of a naturally ventilated historical building in Italy. The variables under consideration in the study are airflow patterns, air distribution, temperature difference, and velocity of air [22]. It also examined the mechanism of natural ventilation in their work by considering multiple cases like single-sided ventilation with an opening on the windward and leeward walls and cross ventilation systems [23]. Studies were also made on the effect of stack size on ventilation in a four-room flat in Singapore [24]. It was also shown that the opening size, shape, aperture, and position could be used to acquire required ventilation rates for day and night modes. Another study [25] explains the natural limitations and solutions of ventilating tall buildings. A review by [26] studies the factors affecting the ventilation of thirty buildings in Japan. The study highlights that design of the building is a critical factor in naturally ventilated systems. Hence as early as the design phase, the user must consider the ventilation mechanism of the building structure.

For mechanical ventilation systems, Z.lei and L.junjie have studied about maintaining the quality of air as well as temperature in residential buildings [27]. However, in this study, the efficiency of these systems is calculated to be almost 50%. In high-rise buildings, factors involved in designing effective mechanical ventilation systems are efficient fans, increased volumetric flow rate, and optimum control process [28] [29]. Furthermore, it was analyzed the effect of heat recovery systems on the efficiency of mechanically ventilated apartments [30]. In their paper, they discuss the advancements in mechanical HVAC systems, including pumps, air conditioners, and energy storage units. For variable temperature and volume loads, [31] researched various modes and evaluated associated energy and cost implications for commercial and domestic buildings. To reduce greenhouse gas emissions, [32] analyzed the performance of a heat pump operated by solar and wind energy, reducing almost 30% of yearly carbon emissions. The heat pump was operable in both cooling and heating mode.

Even though mechanical ventilation is more reliable and comfortable, it consumes more energy and produces noise and air pollution. Consequently, numerous researchers have opted for hybrid systems that employ the best of both models. They are cost-efficient and simultaneously provide more control and flexibility to the user. In this regard, [33] examined hybrid systems in 46 apartments of China's five distinct climate zones for the summer and winter seasons. Another study compared the effect of various types of ventilation modes on the concentration of particles inside residential buildings. The study concluded that mechanical systems could more easily control the amount of these particles [34]. To explore the potentiality of hybrid cooling and heating systems, [35] modeled an integrated energy-saving system for residential areas to provide thermal comfort without depending entirely on powered temperature control. Factors that impact the mixed mode HVAC system for maintaining the air temperature of the building are studied by [36]. In Canada, environmental factors have pushed researchers to consider hybrid modes integrating natural ventilation with mechanically driven systems. In light of this, [37] conducted a feasibility study on mixed-mode systems for commercial buildings.

2.1 Natural ventilation models

Most researchers still favor natural ventilation due to its high potential for providing thermal comfort and good air quality, which has become a drawback in closed ventilation models. Natural ventilation consumes the least amount of energy among other models through increased airspeed during the day and a high ventilation rate at night. ASHRAE standard 62 states that whenever natural ventilation has been opted for, sufficient ventilation should be provided but does not specify a sufficient amount of ventilation in quantitative terms [38]. Nevertheless, the following requirements must be fulfilled in such systems.

1. Naturally ventilated rooms must always be open to and within 8m of window or roof opening.
2. At the minimum, 4% of the net area shall serve as an opening for ventilation.
3. Occupants must have access to means for opening vents in the space occupied.

The standard also mentions the outdoor air quality requirements, which are relevant for the case of natural ventilation systems. Natural ventilation systems can be regulated using several design parameters. The factors that control the efficiency and effectiveness of ventilation of buildings and rooms only naturally include:

- Current velocity of wind and direction
- Orientation of the building (sun direction)
- State or condition of the building
- Size and location of the window on walls etc.
- Functioning of window (open/shut or percentage of opening)
- Outdoor temperature and weather conditions
- Humidity or moisture content of the external environment

Hence, the design of a building must take into account the given factors for effective ventilation rates. The proceeding paragraphs discuss the general parameters that affect the selection and design of natural ventilation modes (all-natural, hybrid, night-cooling, etc.).

2.1.1 Indoor Air Quality and Occupant comfort:

Based on human body heat balance requirements, ASHRAE standard 55-2004 [39] specifies the conditions that need to be maintained for which at least 80% of the occupants of that environment find the thermal state of the space acceptable. The standard elaborates on the factors affecting the choice of ventilation, including environmental (building temperature, humidity, radiation, external airspeed, etc.) and physical (occupant activity, clothing, etc.). In the case of natural ventilation, air from the outside environment is introduced into the room. The outside air may be contaminated with pollutants and microparticles in polluted cities, causing devastating health issues and breathing problems. Hence maintaining the quality of air inside the room is of immense importance. According to a survey conducted by World Health Organization (WHO), 4.2 million people lose their lives because of medical issues resulting from inhaling poor-quality air [40]. In order to improve the quality of indoor air for comfort, filters are usually employed in mechanical systems. However, according to statistics obtained from Europe, mechanical systems with air filters are responsible for the production of 30% of carbon dioxide [41]. Therefore, the air quality index of a room can be improved by opening windows for cross ventilation. The air inside the room is constantly replaced by fresh air from outside, hence maintaining the desired quality and preventing the accumulation of CO2 in the room. Ricardo et al. assessed the indoor air quality of educational buildings and discussed strategies to improve it in naturally ventilated systems. The results concluded that CO2 concentration could be controlled without affecting the comfort level of the occupants [42]. Tariq et al. also conducted a study to evaluate indoor air quality and carbon dioxide concentration in a naturally ventilated room. The authors suggested studying naturally ventilated rooms with filters in a future study owing to the significant presence of unfiltered ventilation micropollutants during natural ventilation [40].

2.1.2 User Behavior

Apart from environmental factors, the natural ventilation model depends highly on window operation control, with windows being the primary control inputs for the mode of natural ventilation. Windows can be operated both manually by the occupants and can be controlled automatically. Manual window control solely depends on the building occupant, and hence occupant behavior is one of the dictating factors in natural ventilation models. Occupant behavior depends on the clothing pattern of that geography, occupancy patterns of the building (residential, commercial, office), and even the sociodynamics factors involving the hierarchy and social etiquette of the occupants [43]. The study conducted by [44] also reiterates the significance of occupant behavior in model accuracy and performance prediction.

2.2 Thermal Comfort Criteria:

ASHRAE standard 55-2010 defines thermal comfort to be a measure of satisfaction of a human with the thermal conditions of the environment of the space he is present in [39]. This implies that the heat felt by a person in a building must not represent the temperature of the air in the building directly. Instead, it will be the heat felt by the human with its clothing, body heat, metabolic rate, etc. These physiological factors, combined with the building's environmental factors, can be measured, and an index can be calculated to represent the thermal comfort felt by the occupant. Various scales have been proposed for the thermal comfort index, including Predicted Mean Vote (PVM) [45], two-node and multi-node models [46], and UCB model [47] to be one of them. For this study, the classical PVM model will be used to measure the thermal comfort index achieved for desired temperature range.

2.2.1 Evaluation method based on thermal comfort

For natural ventilation systems, the evaluation of the efficiency of the model differs from that of HVACs. In these cases, three adaptation models are generally used, as stated below

- Physiological adaptation model: The physiological behavior of occupants under the maintained temperature and conditions
- Behavioral adaptation model: The behavior of people in response to the conditions. For example, wearing more clothes to make themselves comfortable.
- Psychological adaptation model: People alter their sensations and perceptions as per their expectations; hence these results need also be studied.

2.2.2 Climate Parameters

A more technical definition of the parameters influencing the natural ventilation models is given by Belleri et al., which include factors such as envelope conductivity, internal heat gains, and wind pressure coefficient [44]. Outside temperature also affects the ventilation system performance to a great extent. For areas with higher peak temperatures, natural ventilation only becomes inefficient in maintaining the indoor temperature because the outdoor temperature is more than the set-point cooling temperature of the building during the daytime. For such cases, night-time cooling is the suggested mode of natural ventilation (reference). A general rule of thumb is to prefer night cooling for an average minimum diurnal temperature swing of 10º C-12º C. Night-time ventilation is a widely adopted model in various residential and commercial buildings in colder regions of Europe [48] and North America. [49] Divided the climate into five categories and determined each group's ventilation requirement. They are given below

- Hot temperatures: Natural ventilation is not possible
- Warm temperatures: Natural ventilation with high flow rates
- Comfortable temperatures: Moderate ventilation needed
- Humid Climate: Natural ventilation is not workable
- Cool, humid climate: minimal ventilation is required

2.3 Natural Ventilation System Modeling

Various studies have suggested a variety of system modeling approaches for naturally ventilated systems depending upon different ways of understanding and describing the system environment and respective control devices. There exists a variety of building modeling and simulation tools for physically modeling the system. System models obtained through these modeling tools are usually nonlinear and are relatively challenging to be used for real-time control computations.

Another approach to system modeling is through system identification, in which a system model is obtained using real-time data. Almahdi et al. have demonstrated the effectiveness of a MIMO HVAC system modeled through system identification [50]. The system model is used to simulate a state feedback controller for temperature and CO2 regulation of an HVAC system in a multi-zone building using hot water and energy usage as system inputs. Though system identification provides a simple estimate of the system's physics, the model accuracy depends significantly on the input data [51]. Generation of an extensive data set that captures all of the system excitation and response patterns is a cumbersome and complex task and may compromise the model's accuracy in case of incomplete or inaccurate input data.

A standard method is to empirically model the system to predict the numerical values of performance indicators – usually the airflow rate, temperature, humidity, and CO2 concentration. These statistic models provide a reasonable basis for the control theory and are found to give satisfactory results. This model has usually termed a state space model: a discrete time-based representation of the physical system. State space modeling is a matured technique adopted significantly for SISO (single input, single output) and MIMO (multiple input, multiple outputs) analysis. It also well accommodates system uncertainties and mode-switching dynamics of the system. A medium to high-fidelity linearized state space model is computationally less costly and efficient enough to provide a basis for accurate control predictions for an uncertain system. Depending upon the system under consideration and the required mode of control, the state space can be derived as linear time-invariant (LTI), linear time-variant (LTV) [52], non-linear [53], and bilinear.

A bilinear representation of the system is a model that is linear in states and inputs but is not linear in both. Being higher in order naturally, the bilinear models are required to reduce to low-order models for controls problem. Various natural ventilated systems are modeled as bilinear mathematical systems and are found to have been controlled through classical and modern control techniques. Significant literature exists on the control of bilinear systems using Linear Matrix Inequalities (LMI) [54], Hamilton Jacobi Bellman Isaac Inequalities [55] [56], Riccati equations

[57], and Bilinear Matrix Inequalities [58]. Various mechanical, natural, and hybrid ventilation systems are also found to be modeled as bilinear systems. One such model is proposed by Parisio et al., in which a mechanical ventilation system is modeled for CO2 concentration balance [59], and an LTI system is derived for control design. A similar method is explored by Kalman and Borelli for variable air volume HVAC systems with the model derived from a first-order energy balance equation keeping mass flow rate and supply temperature as control inputs [60].

2.4 Challenges of Naturally Ventilated Systems

Naturally driven models have numerous advantages in terms of cost, energy efficiency, environmental compatibility, and access to daylight and fresh air; it has several disadvantages, which are also challenges. Hence, if a user decides to ventilate their building through the wind-driven model, he may encounter the following problems.

- Naturally, ventilation depends on atmospheric conditions, which tend to change or vary drastically with seasons and climatic patterns. This makes natural systems very unreliable and unregulated. In periods of high humidity or low wind speed, the room temperature and conditions can get unbearable.
- Unlike mechanical systems, these models cannot control the entry of microorganisms, micropollutants, smell and odor, and tiny dust particles with the help of filtration devices.
- They can have limited application or efficiency due to building orientation or placement of the window. So, a poorly designed building cannot maximize ventilation rates with natural mode only. In comparison, mechanically operated ventilation systems can work anywhere.
- The direction of airflow can be controlled so that the temperature in the room is uniformly maintained.

2.5 Control Design for a Naturally Ventilated System

The history of ventilation systems control design is as mature as the classical control design practice. The simplest form of control adopted for such systems is rule-based control, which works on the basic “if-then” logic [61]. The system operates on 0, 1 logic guided by conditional if statements (for example, window open or closed based on set indoor temperature value, humidity level, CO2 concentration, etc.). Rule-based heuristic control finds its implementation in various mixed-mode ventilation systems for which the windows are operated for natural ventilation keeping various climatic conditions as control criteria. Keeping inside temperature, outside temperature, outwind velocity, and direction as control criteria, Eftekhari and Marjanovic have developed a rule-based fuzzy controller to set the position of the louver for a naturally ventilated building [62]. Breech and authors have proposed a rule-based control of a night ventilation model keeping inside temperature and humidity, temperature difference from inside to outside, maximum inside outside temperatures of the previous day, daily rainfall, and instantaneous wind velocity as control criteria [63].

Like all other fields, ventilation system control has also been revolutionized with PID control since it was introduced in the control industry in the late 19^{th} century. PIC control finds its implementation in a wide variety of heating, cooling, and ventilation systems ranging from simple proportional control to tailored PI [64] or a complete PID control [65] as per systems' modulation and control requirements. Where one aspect of all forms of system models remains common in terms of accounting for system uncertainties, various control spaces exist to test the effectiveness of the designed controllers. Although short and famous for the control of mechanical ventilation systems, these control techniques can offer limited control and autonomy when used for the control of natural ventilation systems. Another PID control strategy for maintaining indoor thermal comfort subjected to unpredictable occupant behavior and climatic conditions is to incorporate multivariable linear regression that can be used as a basis for the control design of personalized systems [66]. Dalia et al. have developed a personalized adaptive control model for naturally ventilated office space systems for both steady and transient state conditions keeping occupant activity level as a basis [67]. The study employs a PID controller in conjunction with regression models to set a target temperature-based occupant set flow rate. While testing for an indoor temperature range of 25-33 ºC and relative humidity of 55-80%, the controller demonstrates acceptable performance for maintaining occupant thermal comfort. However, where it shows better performance compared to simple PIC control, this approach has computation limitations and has a significant dependency on data accuracy.

On one side, where natural ventilation is a desired mode of operation for designers owing to its significant health and environmental benefits, it has not been a preferred mode of ventilation by the control system designers. The primary reason is the system's uncertainty. For a naturally ventilated model, the control action may be triggered by a change in indoor climate (temperature variation, CO2 concentration, relative humidity index, etc.), outdoor conditions (outdoor temperature, humidity, or airflow), and system occupancy preferences (desired temperature, etc.). A usual control actuator of such systems is the window which depends upon occupant behavior. This window control by occupants results in a highly dynamic system in which significant changes in these control triggers are projected. This puts the constraint on the controller for predictive assessment of the performance indicators and subsequent control actions for optimal performance to achieve desired results. Two potentially operative control techniques for such systems are optimal and predictive control. Both of the control techniques are reviewed for their implementation and effectiveness in the following paragraphs.

2.5.1 Optimal Control

Annotated literature suggests the advancement to be marked by the application of optimal control for temperature control of ventilation systems in all modes. Yahiaoui et al. have tested the optimal LQG (Linear Quadratic Gaussian) representation of a dynamic regulator by designing an optimal full-state feedback controller. By comparing the system with a Kalman Filter as a state estimator, the study suggests the optimal control theory as an effective mathematical tool for optimal controller design for temperature regulation systems [68]. When compared to traditional on/off control, optimal control is suggested that a robust, efficient, and energy-saving solution through accurate control of ventilation opening is achieved. This active control of natural ventilation models has allowed extending its application in commercial buildings, unlike its usual implementation of residential buildings only [69]. Optimal design usually relies on simulation-based design. Further, the technique requires that the original higher model should be reduced to lower order models to seek an optimal control solution [70].

Predictive control techniques are found to best suit the control of passive ventilation models, including natural and mixed-mode ventilation. Among predictive control strategies, Model-based Predictive Control (MPC) has shown significant potential to become a standard solution for intelligent control of HVAC systems. Various theoretical, simulation, and experimental models have been proposed to test the efficacy of such controllers. Kalman and Borelli have proposed an MPC approach for a closed HVAC system. The system, modeled as a bilinear optimization problem, is controlled based on a sequential quadratic programming algorithm [60]. The study aims to operate a mechanical ventilation system in what the authors called an economizer mode in which the system operates entirely on an external air supply for cooling instead of recirculating all or fraction of air through the air handling unit. These results in reduced coil energy required for cooling. Based on the energy balance equation, the system is modeled as a first-order thermal model with mass flow rate and temperature of the supply air as control inputs. Where the proposed approach shows promising results for heuristic control of HVAC systems, it does not account for system uncertainties and requires extensive experimentation for it to be implemented on real systems. Also, the controller performance is limited by the solution of the sequential programming algorithm constrained to the local solution, which is a limitation towards its application for robust MPC control of the system. Parisio et al. have developed functional simulation scenarios of MPC for HVAC systems.

Building on a tailored control-oriented model of the subject building, the authors propose a stochastic MPC approach to address system uncertainties in terms of external weather and building occupancy [71]. This is achieved by developing linearized models of CO2 concentration and temperature in the control environment. Based on the encouraging results obtained by stochastic and randomized model predictive controller [59], the study is further extended to the implementation of the proposed controller on a real testbed [72]. Stochastic optimization yet needs further research as the literature shows a somewhat conservative performance for ventilation control as compared to deterministic optimal control. One such conclusion comes from a study conducted by Hu and Karava that investigates the performance of a predictive controller for a control-oriented mixed-mode ventilation model of a multi-zone building [52]. The results obtained over a five-month simulation study showed that heuristic approaches have the advantage of saving more energy over simple predictive control. Based on these results, the study suggests the exploration of MPC strategies for the control of ventilation models that involve uncertainty in various forms, such as external temperature change and occupant behavior. Prasad et al. demonstrate the effectiveness of predictive control on a set of natural ventilation set parameters. Built on optimal control design, the research models objective cost function for the optimum selection of a ventilation configuration [73].

CHAPTER 3: SYSTEM MODELING

System modeling is crucial for controller design as all forms of controllers ranging from classical to modern, optimal, and predictive controllers, require appropriate system models for accurate response prediction and control optimization. This implies that the system should be sufficiently modeled mathematically to cover the system's dynamics. It should also be computationally efficient for real-time optimization and control computations. This requires the identification of the input, output, and control parameters of the system. This chapter focuses on the mathematical modeling of the system for this purpose.

3.1 Natural Ventilation Design Parameters

The deriving factor of natural ventilation system is the pressure difference that moves the fresh air through the designated space. This pressure difference can be caused by either wind-driven natural ventilation or buoyancy-driven, also called stack-driven natural ventilation. This fulfills the demand of maintaining mainly two output parameters: thermal comfort(internal temperature) and indoor air quality. Thermal comfort implies the maintaining of temperature between maximum and minimum allowable temperature limits for the system. Thermal comfort criteria defined by ASHRAE standard 55 also take into account other factors such as humidity level and airspeed [39]. On the other hand, indoor air quality is maintained by keeping the level of contamination in the air to the set level for acceptable air quality and maintaining a set ventilation rate as required or desired for good breathing levels of occupants. This study aims to focus on attaining the required thermal comfort while considering temperature as a reference criterion.

3.2 Building Model

There is a staggering number of design models proposed to drive natural ventilation systems. When combined with mechanical systems to form hybrid strategies, they become more complex. However, in summary, they can be categorized into basically three fundamental types of natural systems, namely, wind-driven cross ventilation, buoyancy-driven stack ventilation, and single-sided ventilation which are briefly discussed in the following paragraphs and illustrated in Figure1.

Cross ventilation system operates on the principle of cross-ventilation via ventilation openings on two opposing sides of a building or room. However, the opening must be designed to remove pollutants and heat effectively. The model requires pressure difference between inlet and outlet openings.

Stack ventilation works on density differences between cold outside air and warm indoor air. The cool air replaces the warm air in the room. This model requires a long chimney or exhaust to generate sufficient buoyancy forces to generate this type of flow. Moreover, a small opening can also generate this pressure difference and cause the movement of air.

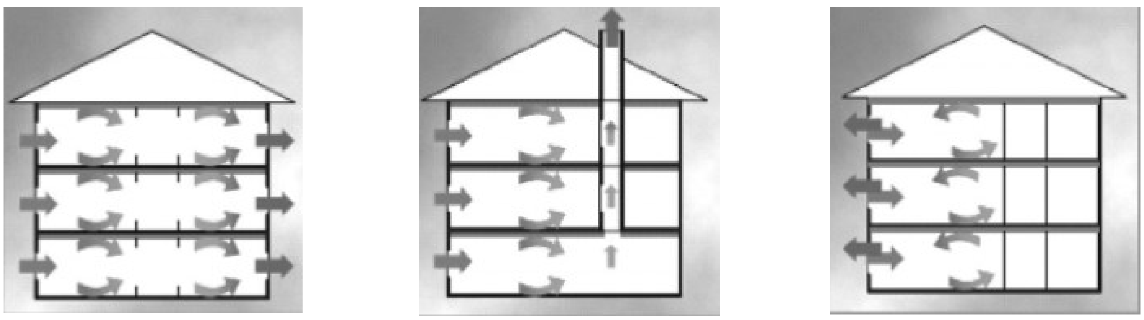

Figure 1 Building model for natural ventilation: a) cross ventilation, b) stack ventilation, c) single-sided ventilation

Single-sided ventilation is typically used for single units for local ventilation based on buoyancy effects, wind pressures, and turbulence. Therefore, these systems are incredibly variable to operate and never desired for highly accurate thermal controls. However, in the case of single rooms, office spaces, and personal rooms, this model is very attractive.

Out of these dwelling ventilation types, my thesis focuses on single-sided ventilation, as this type is primarily used in commercial and residential buildings and homes. A systematic design methodology is followed while designing the model.

The process is schematically explained in Figure 2. A usual process of a system design starts with main system requirements that are dictated by parameters like the indoor and outdoor climate of the region, number and types of occupants using the system/dwelling, equipment or load to be used in the region of interest that is the room, etc. Based on these requirements, a description of the overall process of heating/ventilation is made as, how the system will be affected by temperature change due to heat coming from different sources as discussed in stage one. The layout of the building is considered which includes, what kind of dwelling style will be, like either cross ventilation, stack ventilation, or single-sided ventilation as each has different heat transfer criteria, followed by the thermal

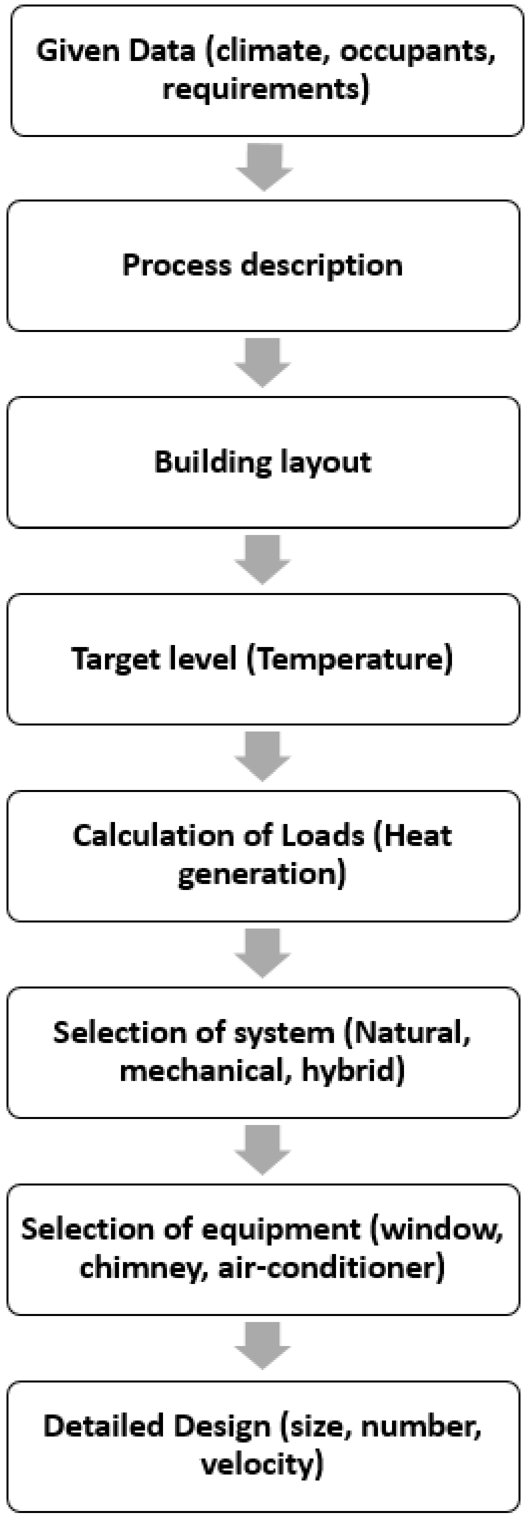

Figure 2 Design process of ventilation systems

comfort requirements to be met in the building as the temperature has to be controlled as an outcome of the system after applying control and an estimated required energy loading is calculated. Once the preliminary calculations are done, the system selection of design components is performed. Here next step is to identify the mode of heating and ventilation that most suits the thermal comfort and energy requirements of the building. It can be a purely mechanical or natural ventilation system or a combination of both as currently, I have selected a natural system. Based on the mode of ventilation selected, a system control mechanism along with control actuators can be selected which is followed by further control system design details. In this, study the control mechanism is a window in the building which is being controlled for thermal comfort to maintain temperature. The window will open and will keep on adjusting the position according to internal temperature and by that controlled window that is by continuously getting its position change, the air exchange rate continuously keeps on changing, and outside temperature impact also changes which results in keeping internal temperature maintained. So, this is generally the overall procedure for designing a ventilation system.

3.3 Thermal model

3.3.1Thermal Modeling Approaches

The thermal model of buildings in steady state conditions uses a static behavior approach, where all inputs can be controlled. In cases where the inputs and outputs of internal and external environments vary, dynamic models are employed. Thermal models are commonly divided into three categories, [74] namely, white, grey, and black box models. These approaches have been excessively used in the past to describe thermal models of residential areas, energy demands, prediction of heat gain, and decreased energy consumption of systems. When developing thermal control models for human comfort, dynamic approaches are used.

White, black, and grey box are illustrated and explained for dynamic models in Figure 3, where the white box purely depends on physical understanding and deterministic equations of a system, whereas the black box is based entirely on collected data and does not have any previous knowledge of system and it is an observed or recorded data-driven system. The grey box lies in the middle of both approaches meaning it has some physical knowledge available based on deterministic equations of the designed system and some parameters of the system need some observed or recorded data. [75].

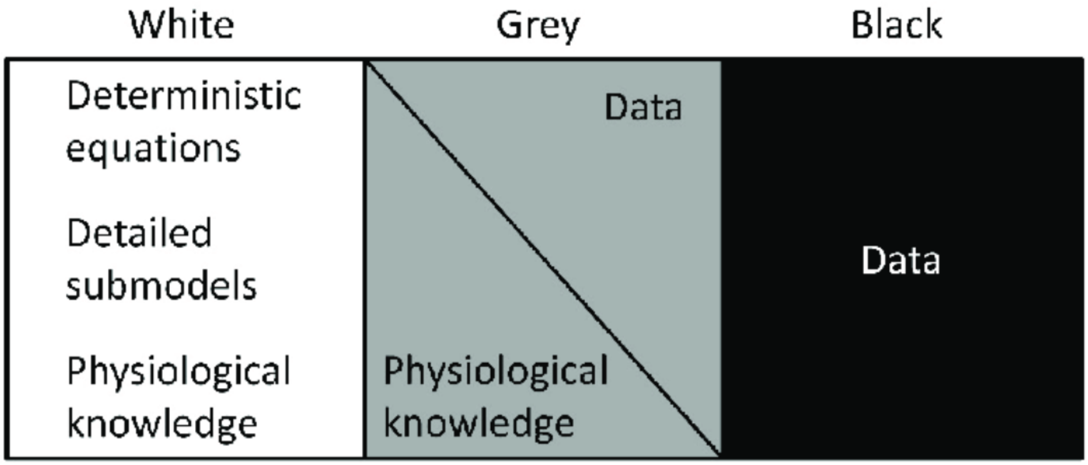

Figure 3 Three common modeling approaches for developing thermal dynamics model

In literature, all three models have been frequently employed for various building configurations as per application requirements. A comprehensive comparison and analysis of such black box and grey box models is performed by [76]. In this study, the authors have developed black box models of various ventilation systems based on different modeling schemes, including transfer functions, neural networks, and state space, and have compared them with preceding grey box models of the same systems. Where black models give ease of obtaining specialized models for data prediction [75], the amount of data required to develop these models is enormous, making the development of such models a challenging design task. This implies that a black box model does not provide flexibility to users. A white box model, on the other hand, is closest to a real-time case. Hence the model can be tweaked to the finest detail in order to measure the changes in the model. It does not make assumptions but rather models these parameters. For example, instead of assuming the heat transfer coefficient, it models the air flow rate within a thermal model. In such cases, grey box models are generally preferred. Hence a user can make changes to the structure and material parameters and determine its impact on the thermal model or temperature, and for that reason, my study is utilizing the grey box model.

3.4 Thermal Dynamics Modeling Approaches

The model adopted in this research for modeling transient thermal dynamics is an energy balance model based on conservation energy respectively. It uses physiological knowledge of building parameters as well as data for uncertainties in the natural ventilation system and temperatures.

3.4.1 Energy Transfer

Dissipation of energy is in the form of heat and is driven by temperature difference. Therefore, for an energy balance model, the heat transfer in a single zone building can be in three standard forms of conduction, convection, and radiation explained in the following paragraphs.

Conduction:

Described by Fourier's Law, it is the heat transfer through solid surfaces, such as through building envelope and surfaces (inside to outside and vice versa), between equipment and surfaces. For example, in a window, heat is conducted through the panes of the window.

The Fourier law of conduction states that the steady-state rate of transfer of heat between two solid objects, represented by, q_{cond} is directly proportional to the area of the surfaces, 'A', and the temperature gradient between them and inversely related to the distance between the solid objects.

$$q_{cond} = -UA\frac{\partial T}{\partial x} = \frac{UA(T_1 - T_2)}{\Delta x} \qquad (3.1)$$

In the above equation, $'T_1'$ and $'T_2'$ are the temperatures (in Kelvin) of the two surfaces and $'\Delta x'$ is the distance (m) between them. $'A'$ represents the area of the surface(m^2). $'U'$ is the constant of thermal conductivity and is introduced with an SI unit of W/m.K.

Convection:

Convection occurs between a fluid and a solid. The fluid can be a liquid or gas. The law of convection is described by Newton's Law of Cooling. In a building, convection occurs through the opening (indoor to outdoor and vice versa) between the equipment and the environment. For example, for a window, the inner surface of the glass pane is exposed to hot air inside the room, whereas the outer surface is in contact with cold atmospheric air.

Using Newton's law of cooling, convective energy, q_{conv} , for natural ventilation system is mathematically defined as:

$$q_{conv} = hA\left(T_{surface} - T_{fluid}\right) = \frac{T_{surface} - T_{fluid}}{R_{conv}} \qquad (3.2)$$

Where, $'T_{surface}'$ and $'T_{fluid}'$ are the temperatures of the surface and the fluid interacting with it, respectively. In this study, the fluid is air inside the building. $'A'$ is the surface area in meters and $'h'$ is the convection coefficient (W/m^2.K) , also called the proportionality constant . $'R_{conv}'$ is the thermal resistance for convection in ohms. In some cases, the convection coefficient is dependent on temperature and thus varies. The mechanism of convection is actually conduction through the thermal boundary layer, which is very thin in comparison with the object.

Radiation:

Radiation is the heat transfer between two radiating surfaces in the form of electromagnetic energy. Stefan Boltzmann Law governs the rules of radiation. The energy is exchanged through photons that are either emitted by a solid surface or impinge on a surface to get absorbed. Heat exchange between two distant bodies is done through a radiation mechanism. In building models, it will be heat exchanged through walls, roofs, windows, equipment, and occupants.

Stefan-Boltzmann and Kirchhoff's radiation laws guide the mechanism of radiation conducted through electromagnetic waves or photons requiring no medium to travel as opposed to convection and conduction. The law states that energy emitted by a radiating body, q_{rad}, with surface area $'A'$ (in meters m), $'\sigma'$ the Stefan-Boltzmann constant with value of $'5.67 \times 10^{-8} W/m^2K^4'$and absolute temperature T(in Kelvin) is:

$$q_{rad} = \sigma A T^4 \text{ (Watts)} \qquad (3.3)$$

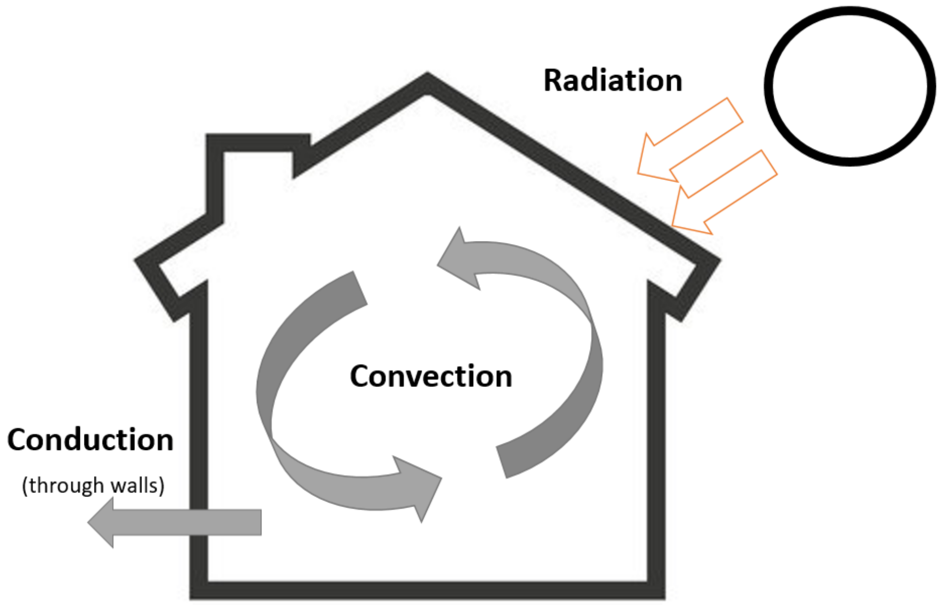

Figure 4 Modes of heat transfer in a general residence

All three mechanisms of heat transfer that can occur in a building, which are conductive heat transfer, convective heat transfer, and radiation heat transfer as discussed in the previous paragraph are illustrated in Figure 4. This shows that the conductive heat transfer is being done from the solid surface of walls (roof and floor also included), the convective heat transfer is occurring between the solid surface of walls (roof and floor included) and the fluid that is the air inside the building or outside the building and radiation heat transfer is due to sun energy.

Some of the standard terms that will be repeatedly used in the energy balance and modeling of thermal control of naturally ventilated zone are given below.

Specific heat (c_p): It is the energy absorbed by a substance to raise the unit temperature of a unit mass. It usually changes with temperature. However, the given value of the specific heat of air is 1.005 KJ/Kg.K.

Heat energy (q): The energy of the system which is transferred from one body to another. It is given in J.

3.4.2 Thermal Modeling Using Energy Balance

The general energy balance equation for a single-sided ventilation zone shown in Figure 5 that predicts the indoor air temperature variation with time can be written as:

$$\dot{q_t} = \dot{q_{en}} + \dot{q_{in}} + \dot{q_r} + \dot{q_{nv}} \qquad (3.4)$$

The equation illustrates the total change in the energy of the single-sided ventilation zone as a combination of the change in the energy of the building envelop that is $'q_{en}'$, internal heat gain from occupancy that is $'q_{in}'$, heat gain from solar radiation q_r, and heat loss from natural ventilation $'q_{nv}'$. Figure 5 is the illustration of a single-sided ventilation building/system with all modes of energy in a form of heat entering or leaving the building. Each mode is defined below.

The heat changes from the building envelop, $\dot{q_{en}}$, includes heat gain or loss through walls, floor, roof, and closed windows, and can be summarized in the form of eq 3.5 as:

$$\dot{q_{en}} = \sum_{i=1}^{n} UA(T_e - T_i) \qquad (3.5)$$

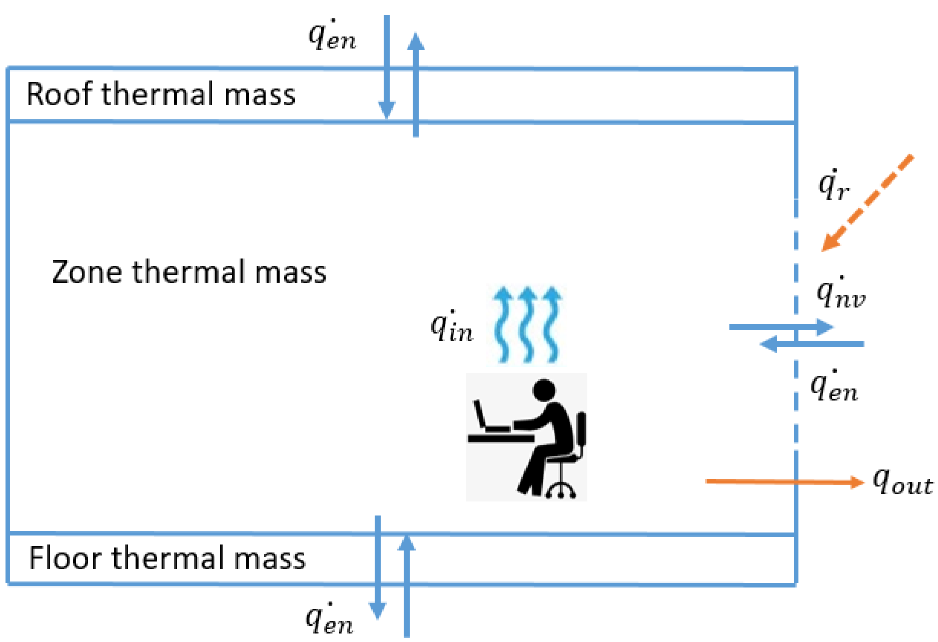

Figure 5 Single zone building model for heat transfer rate and heat change for natural ventilation

For the above mathematical system equation (3.5), $'UA'$ is the mean conductance of the surfaces of walls, roofs, floors, and windows with certain surface areas with i representing the number of surfaces. $'T_e'$ and $'T_i'$ represent the temperatures(Kelvin) of external and internal surfaces of the above-mentioned surfaces respectively.

Similarly, heat change through natural ventilation, $\dot{q_{nv}}$, can be described as:

$$\dot{q_{nv}} = \rho c_p \dot{v_{nv}}(T_e - T_i) \qquad (3.6)$$

Equation (3.6), illustrates that heat change through natural ventilation depends upon the difference between the outside environment temperature $'T_e'$ and internal room air temperature $'T_i'$ in degree Kelvin. Airflow rate $'\dot{v_{nv}}'$ is from outside the building to inside the zone/building space, that is the air flow from outside to inside when the window is open and no airflow when the window is closed. Air density $'\rho'$ (kg/m^3) and specific heat capacity c_p (J/Kg.K) also, affect the amount of heat that can be absorbed in the air at a certain temperature. The airflow rate depends significantly on the window dynamics and position and can be modeled as follows:

$$\dot{v_{nv}} = C_w A_w u \qquad (3.7)$$

This implies that the airflow rate across the window is a function of the opening area (m) $'A_w'$ of window and window opening effectiveness coefficient $'C_w'$ that can be controlled to achieve desired temperatures as per users' convenience or through an automated system. Another important factor that needs to be considered while developing a control law is local wind velocity $'u'$ (m/s) which depends upon outdoor climatic conditions and cannot be controlled once the window is opened.

Keeping the heat gain through solar radiation $\dot{q_r}$ and internal heat gain $\dot{q_{in}}$ as system disturbances, the energy balance equation for the selected natural vent model can be written as:

$$mc_p \frac{dT}{dt} = \rho c_p \dot{v_{nv}}(T_e - T_i) + UA(T_e - T_i) + \dot{q_{in}} + \dot{q_r} + hA\,(T_e - T_i) \qquad (3.8)$$

The first term on the right-hand side of eq 3.8 is the heat transfer through the window when air flows into or outside the system provided that the window is open otherwise no airflow from the window. The second term includes the heat transfer through walls, windows, roofs, and other surfaces through conductance. The third term represents the internal heat gain of the building, that is, the combined effect of heat transfer between equipment and environment and occupants and environment; the fourth term is heat transfer through radiation, and the last term is convective heat energy with T_e being the temperature of surfaces and T_i as room air temperature.

3.5 Linearized Thermal Model

The system and all the physical phenomena relevant to it, when modeled, result in one or a set of coupled partial differential equations. The system presented by eq 3.8 is bilinear in the form in which the state variable temperature is also a function of the system input, which is the heat loss through natural ventilation. However, for modeling an accurate real-time control, the dynamic model should be simplified in the form of a reduced-order model. This can be done by linearizing the system at operable points called trim points or steady state points. One of the standard effective methods used for developing such reduced-ordered models are through Resistor-Capacitor (RC) Networks, also called lumped thermal parameter model. A detailed understanding of the linearization process is discussed in the proceeding paragraphs.

3.5.1 Heat Transfer through Building Envelop

An equivalent thermal circuit of heat transfer through a plan wall is shown in Figure 6. In this figure, there is a wall of natural ventilation building with $'T_e'$, as external environment temperature, $'T_i'$, as internal room temperature and $'T_{s1}', 'T_{s2}'$, as walls surface temperatures. The flow of charges in a solid is known as electrical conduction, and thus, heat transfer is also known as thermal conduction. Therefore, a reduction in heat conduction is thermal resistance, just as electrical resistance is for electrical conduction.

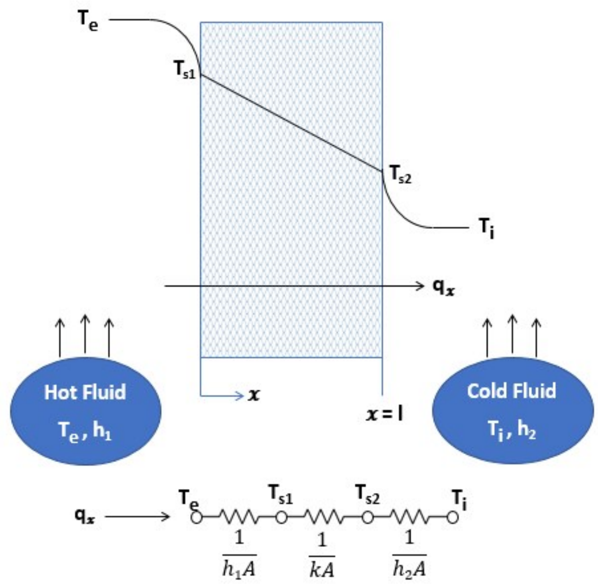

Figure 6 Heat Transfer through a Plane Wall and Equivalent Thermal Circuit

As resistance can be modeled as a ratio of driving potential to the corresponding transfer rate, thus the following equation of heat flow becomes:

$$Heat\ flow = q = \frac{KA}{l}\,(T_e - T_i) \qquad (3.9)$$

Considering the above equation, heat flows through any surface depends on the thermal conductivity K of wall material (W/m.K), area of the wall A (m^2), wall thickness l (m), internal and external Temperatures $T_{i,e}$ (K) and temperatures of outside surface 1 and inside surface 2, $T_{s1,s2}$. Thus, the thermal resistance for heat conduction becomes:

$$R_{tcond} = \frac{T_{s1} - T_{s2}}{q_x} = \frac{l}{kA} \qquad (3.10)$$

Similarly, Ohm's law provides for electrical resistance as well;

$$R_e = \frac{E_{s1} - E_{s2}}{I} = \frac{l}{\sigma A} \qquad (3.11)$$

With $'E_{s1} - E_{s2}'$ is the difference of electrical fields of the two sides of the surface and σ and I are the electrical conductance and current respectively.

Thermal resistance for thermal transfer can also be associated with convection through a surface as per newton's law of cooling, illustrated mathematically below:

$$Heat\ flow\ (q) = \ hA\ (T_s\ - T_\infty) \qquad (3.12)$$

Where $'h'$ (W/m^2.K)is the convection heat transfer coefficient, T_s and T_∞ are the temperatures (Kelvin) of the surface and air($T_{\infty 1,} T_{\infty 2}$ *is outside and inside air temperature*) and $E_{s1,s2}$ is for electrical conduction in surface 1 and surface 2. Thus, the thermal resistance for convection becomes:

$$R_{tconv} = \frac{T_s - T_\infty}{q} = \frac{1}{hA} \qquad (3.13)$$

The pictorial and circuit representations as in Figure 6 of thermal activities help to explain natural phenomena much more effectively. Thus, Figure 6 shows the equivalent thermal circuit for a plane wall with surface heat convection conditions. Separate consideration of each element in the circuit helps to determine the final results of thermal transference. The element q_x is observed to be constant throughout, and therefore, the overall relation becomes;

$$q_x = \frac{T_{\infty 1} - T_{s1}}{1/h_1 A} = \frac{T_{s1} - T_{s2}}{l/kA} = \frac{T_{s2} - T_{\infty 2}}{1/h_2 A} \qquad (3.14)$$

Therefore, the overall heat transfer rate may also be expressed as,

$$q_x = \frac{T_{\infty 1} - T_{\infty 2}}{R_{tot}} \qquad (3.15)$$

Since conduction and convection resistance are in series, they may be summed up as follows;

$$R_{tot} = \frac{1}{h_1 A} + \frac{l}{kA} + \frac{1}{h_2 A} \qquad (3.16)$$

Thermal Potential

Thermal resistances can be defined for various modes of heat transfer, such as conduction and convection, only when the steady state conditions are met. Thus, an identical circuit diagram can be developed to study the thermal behavior in such a condition. These elements and conditions are also equivalent to the electrical circuit, similar to what was discussed above for thermal resistances. It is observed that the identical circuit that is mentioned above is known as thermal potential in steady-state conditions that are equivalent to electrical circuit elements. Furthermore, under steady-state conditions, the heat transfer rate is constant while it changes over time in transient heat transfer.

Thermal Capacitance

We need to introduce the theory of thermal capacitance as well if we need to study the behavior of transient heat in a building model. There is a variation in the temperature of a body during internal thermal transference, and thus, thermal capacitance is needed to be studied, which is stated as the capacity of a body to store heat. Typically, it is expressed in units of (J/Kg.C) or (J/Kg.K) that are also interchangeable. If the material of the wall is a homogenous material with sufficient physical properties, then the thermal mass of the wall becomes;

$$Thermal\, mass\ =\ m\,.\,c_{pw} \qquad (3.17)$$

Where m is the mass of material used in the wall and c_{pw} is specific heat capacity of the wall(KJ/Kg.K). However, for a wall/surface that is made up of different materials, the sum of their specific heat capacitances should be used in the calculations. For the current study, the walls, floor, roof, and window are assumed to be made of homogenous materials individually.

The system is linearized in state space form using thermal network representation. The thermal model of the designed building model shown in Figure 7 is based on the heat balance equation. In the thermal network representation Figure 7, each temperature zone that is $T_z(room/zone), T_r(roof)\ and\ T_f(floor)$ is considered a node. Each node is a combination of thermal capacitance (thermal mass), with its state represented by the temperature of the node. All nodes are connected to each other through thermal resistances.

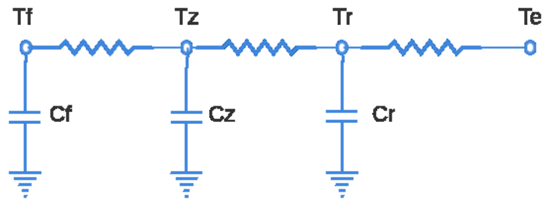

Figure 7 Thermal network representation of the single zone building model with walls and floor insulation

The values of capacitance and resistance are a function of the building's physical properties and geometric parameters. The three capacitances C_z, C_r, C_f depend on the lumped masses and the specific heat capacities of the room air, roof, and floor, respectively. For each node, the lumped thermal mass can be estimated based on the mode of heat transfer through it.

For conduction within the roof and floor, the resistance will be $R_{con} = \frac{l}{2KA}$, where $l\ (m), A\ (m^2)$, and $K\ (W/m.K)$are the length, area, and conduction heat transfer coefficient of the surfaces.

For convection between the surfaces and room air, the resistance can be calculated using the, $R_{conv} = \frac{1}{hA}$, where $h\ (w/m^2.\mathrm{K})$ and $A(m^2)$ are the convection heat transfer coefficient and area of the surface, respectively.

Hence, the overall thermal resistance between two nodes can be computed as follows:

$$R_{ij} = R_{cond} + R_{conv} = \frac{l}{2KA} + \frac{1}{hA} \qquad (3.18)$$

Where, R_{ij} represents the resistance between two divisions'i' and 'j.' These two zones could be any two divisions, such as the room which we have labeled zone, the roof, or the floor. Furthermore, C_z, C_r, C_f represent zone air capacitance, roof capacitance, and floor capacitance, respectively while T_z, T_r, T_f is the zone air/room temperature, roof temperature, and floor temperature, respectively.

In order to simplify the complicated calculations involved in the determination of the temperature of a room, assumptions are made. The heat transfer equations discussed in the preceding section will now be utilized to determine the heat balance of the room. The assumptions made in the process are as follows.

1) It is assumed that the heat transfer in the room is a steady state process which means that temperature at all points of a system remains constant and unchanged with time. As the heat flow changes and the temperature of the room fluctuates from the reference, it shifts into a transient state. However, it eventually attains thermal equilibrium and again shifts to a steady state.
2) For the energy balance, it is assumed that a homogenous temperature is maintained in the entire room. Therefore, the temperature at any point in the room is taken as a constant value. This means that conduction and convection occur at steady-state conditions.
3) The specific heat c_p of air within the room is also taken as a constant value. At room temperature, the value of c_p is normally 1.005 KJ/kg K, so in this work, the value of c_p at 300K is assumed to be 1.007 KJ/kg K.
4) The pressure in the room and of the outside environment is taken to be atmospheric pressure, that is, 1atm. This facilitates the exchange of air through convection between the room and the external environment at isobaric conditions.
5) Heat gain due to solar radiation is taken as a bulk value because of limited, exact data. In reality, the position of the sun with respect to each wall affects the heating and temperature of the walls.

It is supposed that the temperature of the wall does not change with the thickness of the wall. In reality, the outer surface of the wall will be at a higher temperature than the inner surface. Using the above-mentioned assumptions for model linearization and reduction, assuming negligible thermal heat transfer through convection within the zone, the linear ODE of the building has been linked to corresponding RC nodes of the solid surfaces.

A full-scale model of the system will consist of temperatures of all the nodes involved in the building model, including nodes internal to the roof, wall, and floor. Despite being simpler in the lumped parameter form, this model still suffers from computation complexities due to the extended state-space dimension. For the building design chosen, to avoid computation costs, the three nodes will be the room zone, roof, and floor of the building from where the heat exchange takes place.

Given, q_{in}, and q_r are the system disturbances, the state space model for the temperature of each node can be developed as follows:

$$C_z \frac{dT_z}{dt} = \frac{T_r - T_z}{R_{rz}} + \frac{T_f - T_z}{R_{fz}} + \dot{q_{nv}} + \dot{q_{in}} + \dot{q_r} \tag{3.19}$$

$$C_r \frac{dT_r}{dt} = \frac{T_e - T_r}{R_{er}} + \frac{T_z - T_r}{R_{rz}} \tag{3.20}$$

$$C_f \frac{dT_f}{dt} = \frac{T_z - T_f}{R_{fz}} \tag{3.21}$$

Above equations (3.19),(3.20), and (3.21) give the linearized state space representation of the system in which the state variables zone temperature T_z, roof temperature T_r, and floor temperature T_f are the function of the building's geometrical parameters, climatic parameters, and heat gains through internal loads, solar radiation, and natural ventilation. $\dot{q_{nv}}$ is the natural ventilation heat exchange due to window opening so this is the control input. $\dot{q_{in}}$ and $\dot{q_r}$ are uncertainties inside the system and are considered as fixed values. T_e is the external temperature that directly impacts roof temperature and also affects the zone when the window opens as air exchanges from outside to inside. R_{rz} is the thermal resistance between the roof and the zone, R_{fz} represents thermal resistance between floor and zone and R_{er} is that of roof and external air as this is the only surface that can transfer heat keeping zone walls and windows as insulated surfaces. For controller implementation, the state space can be further transformed into the form:

$$\frac{dT}{dt} = AT(t) + B_1[q_{nv} + q_{in} + q_r] + B_2T_e(t) \tag{3.22}$$

here, A is the state vectors coefficient matrix given as:

$$A = \begin{bmatrix} \left(\frac{-1}{C_zR_z} - \frac{1}{C_zR_{fz}}\right) & \frac{1}{R_{rz}C_z} & \frac{1}{R_{fz}C_z} \\ \frac{1}{R_{rz}C_r} & \left(\frac{1}{R_{rz}C_r} - \frac{1}{R_{er}C_r}\right) & 0 \\ \frac{1}{R_{zf}C_f} & 0 & -\frac{1}{R_{zf}C_f} \end{bmatrix} \tag{3.23}$$

While state vector T and input matrices $B_1{}'$ and $B_2{}'$ are defined as:

$$T = \begin{bmatrix} T_z \\ T_r \\ T_f \end{bmatrix} \qquad B_1{}' = [(\frac{1}{C_z} + \frac{1}{C_z} + \frac{1}{C_z}) \quad 0 \quad 0] \qquad B_2{}' = [0 \quad \frac{1}{R_{er}C_r} \quad 0] \tag{3.24}$$

Vector u1, with $B_1{}'$ is input 1 of the system and is modeled as a function of the window continuously changing position (controlled by window opening or closing) and the heat gain or loss through equipment, occupants, and radiation. Vector u2 with $B_2{}'$ that is $T_e(t)$ that is the outside temperature is modeled as a disturbance and is input 2 of the system (uncontrollable). T_e only affects the whole system as when the window is open then external temperature affects the internal room and also continuously affects the roof with which roof temperature keeps on varying. The controllable input is only the 1st vector u1 with B_1 where the window will be controlled. Once the window will open, the controlled input will be in a form of a continuously changing opening position of the window with which the air exchange rate will change and will result in thermal comfort. Hence overall, the $T(t)$ vector is a state vector that has a zone, roof, and floor temperatures and out of these the controlled state will be zone temperature as the zone is the internal room temperature, the input of the system is the window, which after applying controls will continuously change window position and will behave as a continuous controlled input which will result in maintaining internal temperature. The internal heat load and radiation energy is uncertainty which acts as a disturbance. Outside temperature T_e is outside temperature acting as a process disturbance that cannot be controlled only its impact on internal room temperature can be altered by having a controlled window (u1).

The output matric Y becomes

$$Y = CT(t) - 273.15 \tag{3.25}$$

With Temperature (T) as the state element,

$$C = [1 \quad 0 \quad 0] \tag{3.26}$$

3.6 Thermal Comfort

A standard method used widely to evaluate human thermal comfort is the Predicted Mean Vote (PMV) index. The PMV index, proposed by [77], is a number calculated through the energy balance of the human body with its surrounding environment under given climatic conditions [78] such that:

$$PMV = (0.33e^{-0.03met} + 0.028)q_{body} \qquad (3.27)$$

PMV is a function of the human body's metabolic rate $'met'\,(\,calories)$, and the heat of the human body q_{body} that is the difference between heat produced and lost by the human body and can be estimated as:

$$q_{body} = met - q_{sen} - q_{ev} - q_{res} - q_{work} \qquad (3.28)$$

The terms q_{sen}, q_{ev}, q_{res}, and q_{work} denote the sensible heat, evaporation heat, respiration heat, and heat loss during work done by the human body respectively.

All of these terms are modeled in the equations below based on various climatic and body parameters defined by ASHRAE standard 55 [39] and AASHRE standard 62 [38].

$$q_{sen} = 39.6 \times 10^{-9} f_{clo}(T_{clo}^4 - T_{rm}^4) + f_{clo}h_{conv}(T_{clo} - T_{air}) \qquad (3.29)$$

The factors that contribute to sensible body heat include the clothing factor f_{clo}, clothing temperature T_{clo}, mean radiant temperature T_{mr}, convection heat transfer coefficient h_{conv}, and the temperature of the surrounding air T_{air}.

The clothing temperature T_{clo}, as a function of skin temperature and clothing insolation Ins_{clo} can be determined as:

$$T_{clo} = T_{skin} - R_{clo}[f_{clo}h_{conv}(T_{clo} - T_{air})] - Ins_{clo}[39.6 \times 10^{-9} f_{clo}(T_{clo}^4 - T_{mr}^4)] \qquad (3.30)$$

Equation 3.30 derives a relationship between clothing temperature T_{clo} with skin temperature T_{skin}, clothing resistance to heat R_{clo}, clothing factor f_{clo}, zone air temperature T_{air}, convection heat transfer coefficient h_{conv}, clothing insolation Ins_{clo}, and mean radiant temperature T_{mr}.

Evaporation heat of the body depends upon Vapor pressure p_{vap} and is modeled as a function of metabolic rate and heat loss during work as:

$$q_{ev} = 0.42(met - q_{work} - 58.15) + 3 \times 10^{-3}[5733 - 6.99(met - q_{work}) - p_{vap}] \quad (3.31)$$

Respiration heat is also a function of metabolic rate and vapor pressure as well as ambient temperature T_a can be written as eq 3.32.

$$q_{res} = 0.0014met(307.15 - T_a) + 1.72 \times 10^{-5}met(5867 - p_{vap}) \quad (3.32)$$

The heat gain through radiation is nonlinear in nature, along with the vapor pressure as it depends upon varying climate conditions. An approximate linearization of this heat gain/loss can be modeled as:

$$q_r = 39.6 \times 10^{-9} f_{clo}(T_{clo} + T_{mr})(T_{clo} - T_{mr})(T_{clo}^2 + T_{mr}^2) \quad (3.33)$$

Reorganizing the equation, the radiation heat gain comes out to be:

$$q_r = h_r f_{clo}(T_{clo} - T_{mr}) \quad (3.34)$$

Where, h_r is the heat transfer coefficient for radiation.

Using the above equation, the sensible heat gain can be simplified as follows:

$$q_{sen} = f_{clo} h_r (T_{clo} - T_{mr}) + f_{clo} h_{conv}(T_{clo} - T_{air}) \quad (3.35)$$

Typical room temperatures cover the temperature range between 293.15 K and 303.15 K. For this range; the vapor pressure can be calculated as:

$$p_{vap} = \varphi_z p_z / (\varphi_z + 0.622) \quad (3.36)$$

With this, the vapor pressure is the approximated value of $1.598 \times 10^5 \varphi_z$ for which φ_z is the humidity of the zone. This reduces the evaporation and respiration heat as:

$$q_{ev} = 0.42(met - q_{work} - 58.15) + 3 \times 10^{-3}[5733 - 6.99(met - q_{work}) - 1.598 \times 10^5 \varphi_z] \quad (3.37)$$

$$q_{res} = 0.0014met(307.15 - T_{air}) + 1.72 \times 10^{-5}met(5867 - 1.598 \times 10^5 \varphi_z) \quad (3.38)$$

To make the PMV equation linear, the coefficients like metabolic rate, clothing factor, room pressure, and external work are assumed constant values recommended for a room temperature of 25 ^{0}C. Considering the assumptions, the linear PMV equation takes the form:

$$PMV = [0.683h_{conv} + 0.005h_r + 0.0602)T_{air} + 0.678h_r T_{mr} + 35.71h_{conv} + 35.71h_r + 418.9)\varphi_z + 7.308 - 207.6h_r - 207.6h_{conv}]/(h_{conv} + h_r + 11.73) \qquad (3.39)$$

For a range of air velocity variation, the heat transfer coefficient for convection for the coefficient of convection heat transfer h_{conv} can be modeled as:

$$h_{conv} = \begin{cases} 2.38(T_{clo} - T_{air})^{0.25} & for\ 2.38(T_{clo} - T_{air})^{0.25} > 12.1v_{air}^{0.5} \\ 12.1v_{air}^{0.5} & for\ 2.38(T_{clo} - T_{air})^{0.25} < 12.1v_{air}^{0.5} \end{cases} \qquad (3.40)$$

This model is still non-linear and requires regressive calculations of the above-mentioned equations, which makes this model computationally expensive. For real-time control applications, a similar model, for a representative air velocity value of 0.136 m/s (calculated as per ASHRAE standard), the PMV equation can be simplified as:

$$PMV = 0.153T_z + 0.142T_{mr} + 35.71\varphi_z - 88.34 \qquad (3.41)$$

This equation will be used for the PMV estimation for the zone temperature, humidity, and mean radiant temperature for determining the thermal comfort index achieved after the control application.

CHAPTER 4: CONTROL DESIGN

Unlike a controllable energy source in mechanical systems, the driving parameter for a natural ventilation system is the pressure difference across the building, and that exists when the window is open. When the window is closed, there is no air exchange. This variable pressure difference makes the controller design of such systems a challenging task. Wind pressure depends significantly on wind intensity and direction, both of which are subjected to significant fluctuations even on minor time spans like a day or an hour. In a typical urban atmosphere, wind pressure fluctuations vary up to 40% with a 10%-20% change in wind intensity; and can be more when combined with a change in wind direction patterns [79]. Also, where natural ventilation is the preferred choice of most users due to more control over window operation – and hence the building environment – unexpected user behavior makes the system more uncertain. So, the following optimal controls have been designed in order to provide thermal comfort by controlling the windows in the system with uncertainties.

4.1 LQR Control:

Linear Quadratic Regulator (LQR) finds the optimal control input for a linearized system with the aim of minimizing the quadratic cost function. The control signal is the function of system states. At the same time, the cost function is also a quadratic function of system states and inputs.

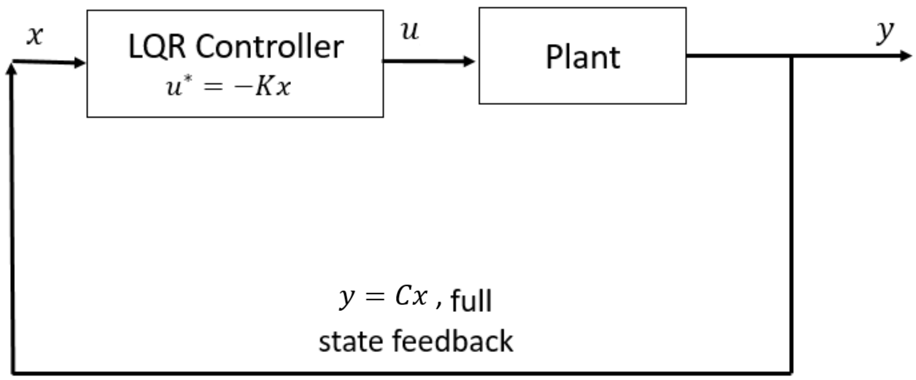

Figure 8 Control architecture of Linear Quadratic Regulator (LQR)

For LQR implementation, the system modeled in chapter 3 can be shown in the form of state space as a linear time-invariant system:

$$\dot{x} = Ax + B_1 u_1 + B_2 u_2 \qquad (4.1)$$

Where,

$$A = \begin{bmatrix} \left(\frac{-1}{C_z R_z} - \frac{1}{C_z R_{fz}}\right) & \frac{1}{R_{rz} C_z} & \frac{1}{R_{fz} C_z} \\ \frac{1}{R_{rz} C_r} & \left(\frac{1}{R_{rz} C_r} - \frac{1}{R_{er} C_r}\right) & 0 \\ \frac{1}{R_{zf} C_f} & 0 & -\frac{1}{R_{zf} C_f} \end{bmatrix},$$

The state matrix $x = [T_z \quad T_r \quad T_f]^T$ consists of three state variables: zone temperature, roof temperature, and floor temperature, respectively.

Input vector u_1 (window opening position due to natural ventilation which affects the air flow rate)= $[q_{nv} \ + \ q_{in} \ + \ q_r]$ that is augmented by load vector $B_1{}' = \ [(\frac{1}{C_z} + \frac{1}{C_z} + \frac{1}{C_z}) \quad 0 \quad 0]$. The load vector, $B_2{}' = \left[0 \quad \frac{1}{R_{er} C_r} \quad 0\right]$, is augmented with the second input vector u_2 i.e., outside temperature 'T_e' affects zone temperature but only u1 is controllable and control is only applied to that u1 as the only window that can be controlled. The window position will keep on changing by having some fixed amount of radiation heat and internal heat as uncertainty and q_{nv} will be controlled depending upon the window position. As the indoor temperature that is zone temperature will reach the upper limit of exceeding the temperature limit, the window will switch to open and then continuously will keep on changing opening position due to which air exchange rate will change by that internal temperature will come back to required internal temperature.

The controller is designed to minimize the quadratic cost function given as:

$$J(u) = \int_{t_0}^{\infty} (x^T Q x + u^T R u) dt \qquad (4.2)$$

Where,

Q and R are the weightage matrices assigned to the states and inputs, respectively.

By varying these weightage matrices, the controller computes the optimum control input as:

$$u^* = -Kx^* \qquad (4.3)$$

The gain matrix K in eq 4.3 is computed recursively by using the following algebraic Riccati equation:

$$K = -R^{-1}B^T P \qquad (4.4)$$

$$0 = Q + A^T P + PA - PBR^{-1}B^T P \qquad (4.5)$$

The existence of a steady-state solution to the Riccati equation necessitates that the pair [A, B] can be controllable. Another condition is that the solution obtained is a unique solution which dictates that the pair [A, C] be detectable. The R should be positive definite and symmetric and Q should also be symmetric and semi-positive definite in order to drive optimal control law.

4.2 LQI Control:

Aimed at minimizing the quadratic cost function like LQR, LQI is designed by adding an Integral to the LQR control sequence. The added integral effect (as shown in Figure 9) is to increase the system performance by decreasing the steady-state error, hence, giving a more robust system response under varying system conditions.

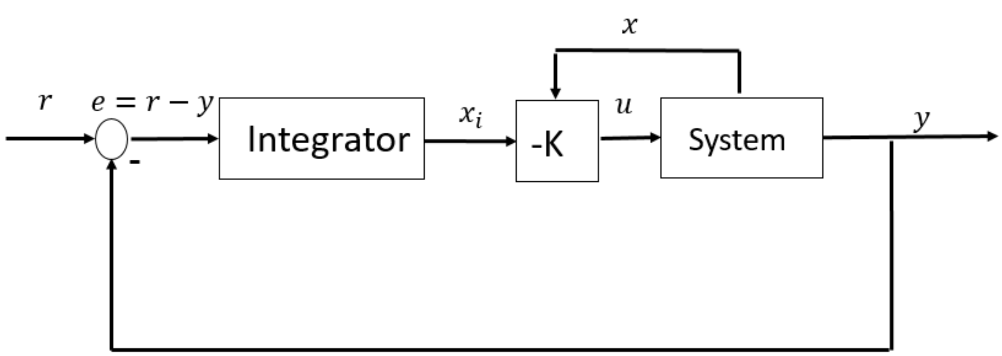

Figure 9 Linear Quadratic Integrator (LQI) control architecture

For the given state space of the form of $\dot{x} = Ax + Bu$, the LQI controller computes the state feedback of the form $u^* = -K(x, x_i)$ where x_i is the output of error gain of integral effect.

4.3 LQG Control

To make all the concerned states of the system observable, the LQR control law is augmented by a Kalman Filter that acts as a system observer. This Kalman Filter combined with LQR constitutes the LQG (Linear Quadratic Gaussian) controller. The LQG control, where it gives the optimum control through the tradeoff of best performance achieved and control effort required, considers the disturbances and noises occurred in the system for the overall system to take the form of:

$$\dot{x} = Ax + B_1 u_1 + B_2 u_2 + Gw \qquad (4.6)$$

$$y = Cx + v \qquad (4.7)$$

For the LQG design of temperature control of the selected system, w and v are modeled as white noise for ease of computation;

Where the process disturbance w is the outdoor temperature that is affecting the system. Measurement noise is represented as v

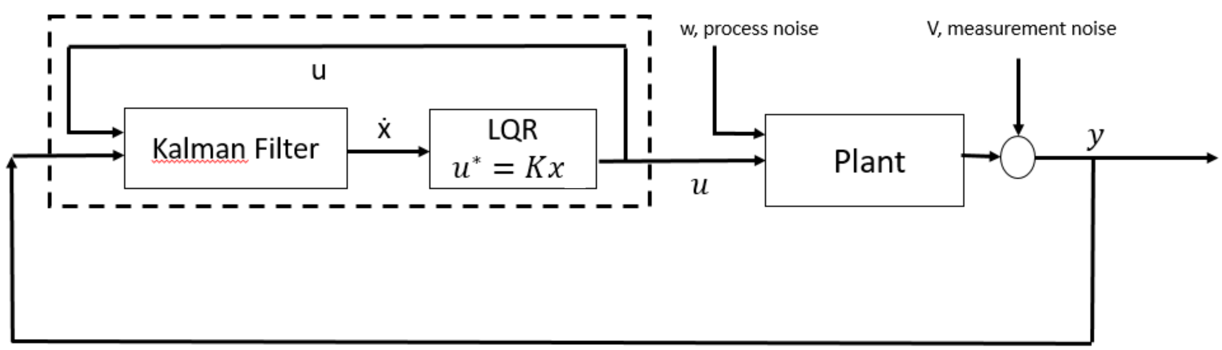

Figure 10 Control architecture of Linear Quadratic Gaussian (LQG) technique

Minimizing the same cost function as in LQR, the Kalman filter incorporated in the control law also gives the estimate of system states in the form of:

$$\dot{x^*} = Ax^* + Bu + L(y - Cx^*) \qquad (4.8)$$

Where L Is the gain of the Kalman estimator

For given control input u and the system output y, the Kalman gain is determined by solving the algebraic Riccati equation with the noise covariance data modeled as:

$$Q_n = E(ww^T), \ \ R_n = E(vv^T) \qquad (4.9)$$

4.4 Model Predictive Control

In this thesis, the MPC is deployed using MATLAB MPC toolbox that automatically converts continuous-time systems into discrete time modes internally. Model-based predictive control (MPC) is the optimal predictive controller with the added property of predicting future events and taking control action accordingly. At each sampling instant, the model predictions are used to perform two calculations: set point calculations and control calculations. The edge it has over the optimal controllers is the inclusion of system constraints. The inequality constraints on input and output can be included in any of or in both set point and control calculations. To save more energy, MPC is utilized to incorporate system constraints well for efficient control and effective energy management. This study utilizes these constraints in the form of upper and lower temperature limits of the zone temperature. Through these interval base calculations, the controller aims to develop a sequence of control actions to move the predicted response to the set point temperature in an optimal way;

$$M = \{u(k + i - 1), for\ i = 1,2, \ldots \ldots, M\} \qquad (4.10)$$

Where,

M is the set of control actions and is called the control horizon. MATLAB MPC toolbox converts automatically continuous-time systems to discrete-time systems as MPC does processing for discrete-time systems.

The set consists of control inputs u at each sampling time k and future inputs $M - 1$. These inputs are calculated such that the set P of predicted outputs is given as:

$$P = \{y^*(k + i), i = \ 1, 2, \ldots \ldots, P\} \qquad (4.11)$$

This set is denoted as the prediction horizon.

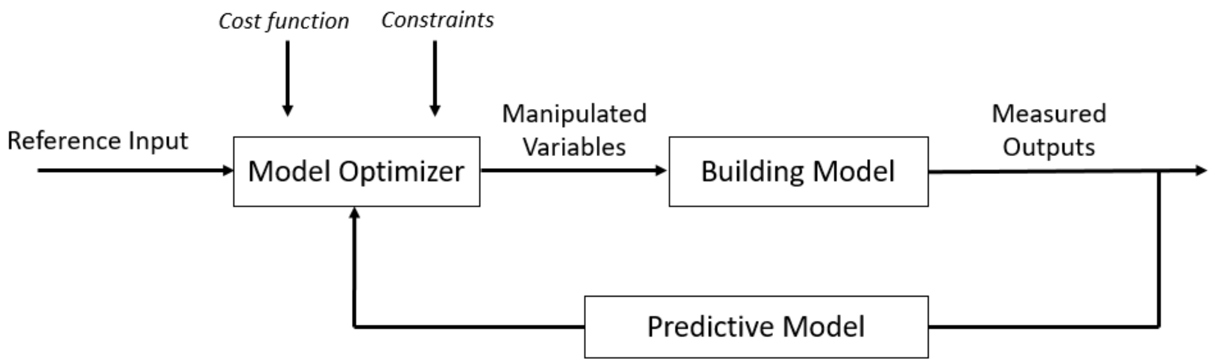

Figure 11 Model-based predictive Control Architecture

For this study, MATLAB MPC designer is utilized for developing receding horizon-constrained controllers. This implies that the controller, despite calculating the control sequences M at each sampling time, implements only the first control sequence.

For MPC design, the system is considered a continuous-time model like that in LQG, which the controller later discretizes for predicting using a sampling time t_s as per controller estimation and optimization calculation requirements. For a given state space, the controller transforms the plant input and output variables into a dimension form such as:

$$x_k(j+1) = Ax_k(j) + BS_iu_k(j) \qquad (4.12)$$

$$y_k = S_o^{-1}Cx_k(j) + S_o^{-1}DS_iu_k(j) \qquad (4.13)$$

Here, A, B, C, and D are the state space matrices and, S_i, and S_o, are the diagonal matrices of input and output scale factors, respectively. x_k is the state vector and u_k is the transformed input vector that includes manipulated variables and measured and unmeasured input disturbances

y_k is the dimensionless output vector

The controller is set to use the optimization problem using its standard quadratic program aimed at solving the cost function of the form:

$$J(zk) = J_y(zk) + J_u(zk) + J_{\Delta u}(zk) + J_\varepsilon(zk) \qquad (4.14)$$

Where zk is the quadratic program decision.

Starting with a fixed optimization horizon of length N, the controller starts by setting the system's current state as the initial state. Then the receding horizon control sequence follows as follows:

- At time instant k, setting the current state x_k as the initial state, the optimization problem is solved over the optimization horizon length N, while considering the system’s current and future constraints.
- Corresponding to the system states x_k, the control vector u_k is applied as a control signal
- States obtained at time instant $k + 1$ are computed
- The same steps are repeated from time instant $k + 1$ onwards

Overall, in this thesis, four controllers have been implemented independently, and separately and gain-tuned for varying temperatures on the developed model.

Figure 12 and Figure 13 below show the overall setup and control implementation. The zone is the room of a single-sided building where there is a window that works as a control input to the room. The room has, a fixed amount of internal heat gain and radiation energy as uncertainty, and the external temperature outside the building is acting as a process input in a form of noise. The window can either be opened or closed. Once the zone temperature is outside the limits then the window gets open and then the designed controller either LQR, LQI, LQG, and MPC is separately applied to control the window position as the window can change its position while the window is open. By changing its opening area continuously, it works as controlled input to the system, and by different window opening area positions while the window is open, the air exchange rate inside the room keeps on changing and the impact of outside temperature in the room keep on changing making zone temperature T_z to be maintained with reference temperature values. Once the internal temperature touches the required reference set temperature then the window closes that is 0 and u1 goes to 0 again. But when the window is 1 that is open then the window opening area u1 keeps on changing continuously with continuous values resulting and continually varying air flow rate.

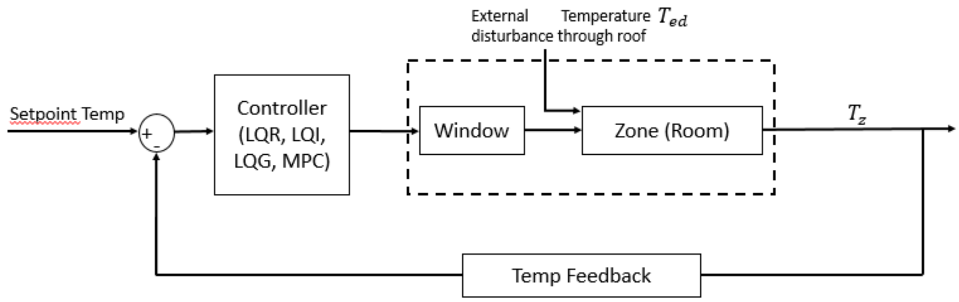

Figure 12: Overall system architecture for controller implementation

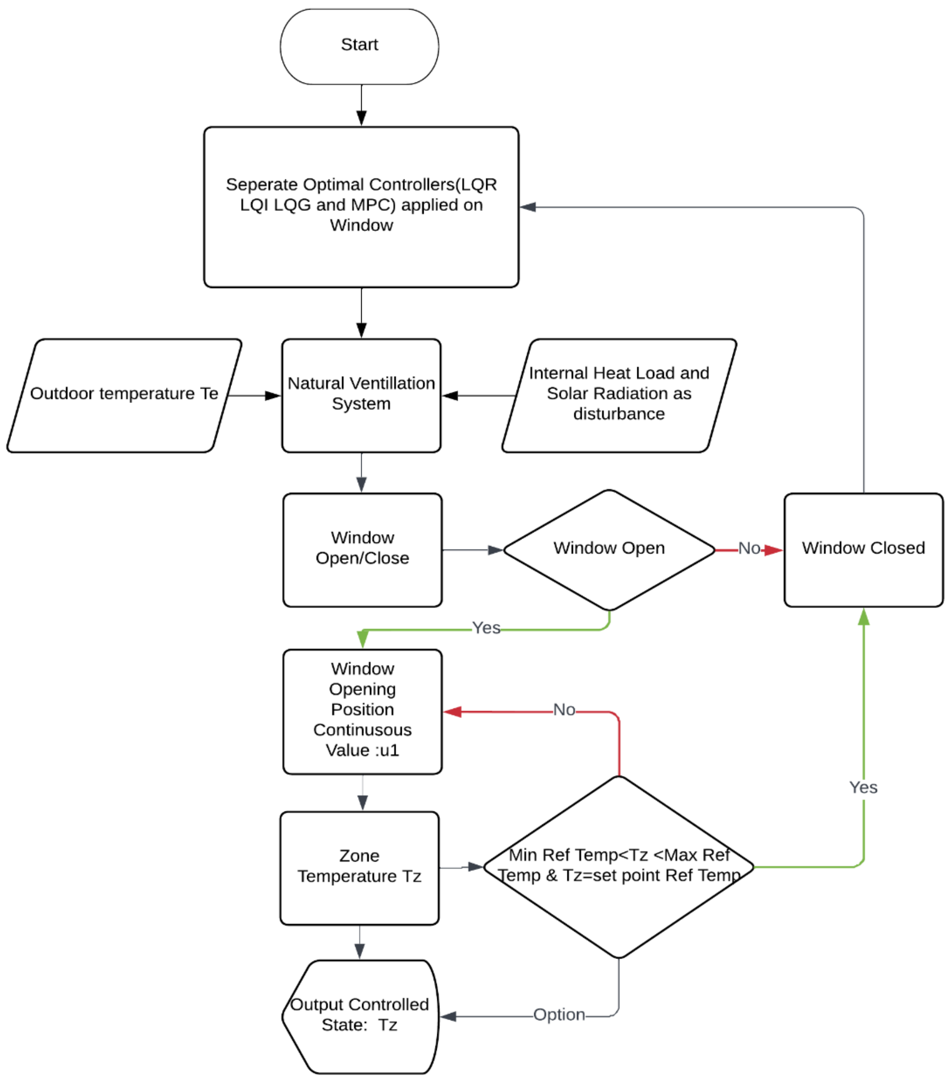

Figure 13: Overall Flow chart of System description with controlled Temperature State

CHAPTER 5: SIMULATION AND RESULTS

In this study, we use LQI, LQR, LQG, and the MPC framework of MATLAB/SIMULINK to obtain the optimal control solution of the system. The control architectures for temperature control using the three optimal controllers LQI, LQR, and LQG and the predictive controller MPC are discussed in this chapter in detail. The set point temperature or the desired zone temperature T_z is set to 25 ^{o}C, with the upper and lower limits being 28 ^{o}C and 20 ^{o}C, respectively. The window is set to open when the zone temperature exceeds the upper or lower-point temperature. Once the window gets open then the control is applied in the window position in q_{nv} where the window position then continuously keep on changing making the internal temperature follow a set point temperature. For simulations, the outside temperature is set to 10 ^{o}C initially then a sinusoidal noise is added to that, which makes the outside temperature a sinusoidal variation. The solar radiation heat gain, q_r is selected to be a fixed value of '50 Watts'. The internal heat gain q_{in} is given a value of "500 Watts" considering the latent and sensible heat of three to four occupants and three sets of PCs and related electrical equipment of an office.

5.1 LQR Implementation:

As mentioned in chapter 4, the designed LQR controller aims to regulate the temperature of its set value. Figure 14 shows the MATLAB implementation of the LQR control architecture.

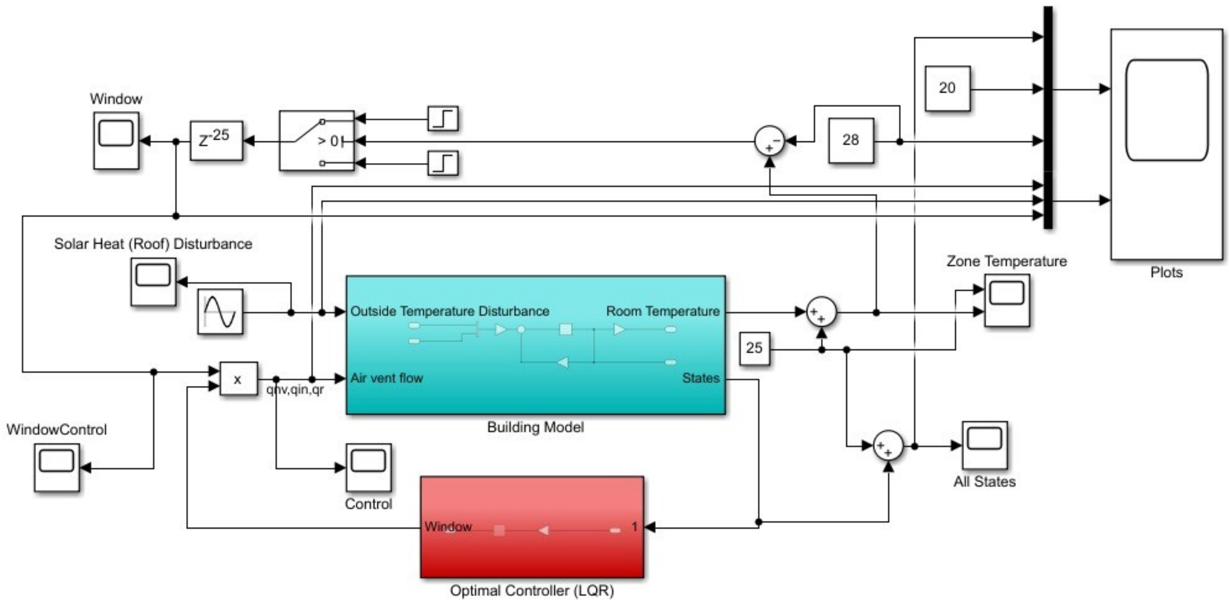

Figure 14 MATLAB implementation of LQR for regulating desired zone temperature

For the selected weights of $Q = \begin{bmatrix} 5 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$ and$R = [1]$, Figure 17 shows the step response of the designed controller for a reference zone temperature of 25 ºC.

For the selected values of Q and R, the new continuous-time state-space of the system is obtained as:

$$A_{lqr} = \begin{bmatrix} -0.2488 & 0.1188 & 0.13 \\ 0.128 & 0.2642 & 0 \\ 0.1488 & 0 & -0.1488 \end{bmatrix} \qquad (5.1)$$

$$B_{lqr} = \begin{bmatrix} -1.539 & 0 \\ 0 & 0.1362 \\ 0 & 0 \end{bmatrix} \qquad (5.2)$$

Column 1 of the computed input matrix B_{lqr} consists of input 1 u_1 and column 2 represents input 2 u_2. The two input columns show the contribution of ventilation flow rate and outside temperature disturbance into the system.

And

$$C_{lqr} = [1 \quad 0 \quad 0] \qquad (5.3)$$

The C matrix shows that in the first state, the zone temperature is controlled by the controller.

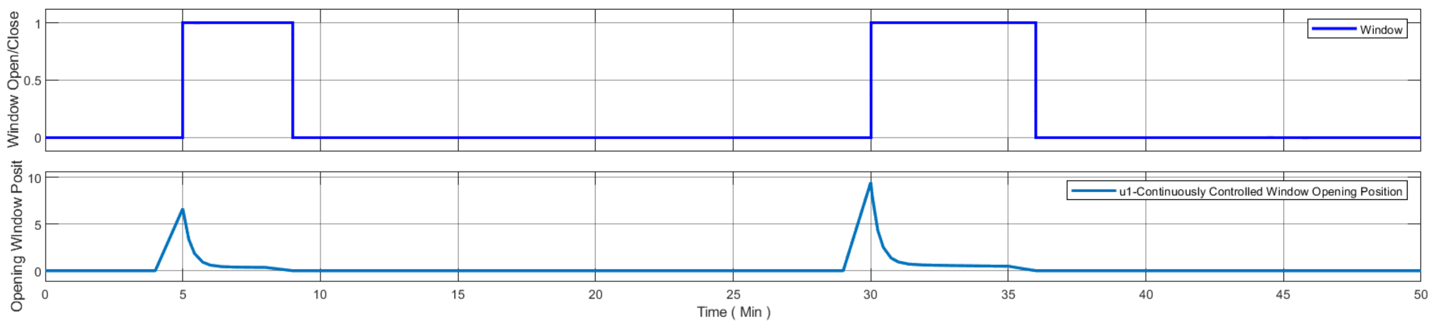

Figure 15: Window Position controlled input by LQR for set point temperature regulation

As shown in Figure 15 above, the continuous window position which is controlled input to the system that is u1 is modeled with the window area coefficient of either 0 or 1. This implies that the window is either closed or open allowing a certain airflow into the zone depending upon the window opening position. As once the window will be open that is 1 then control will be applied to that window opening position and with different continuously changing window positions the air flow rate will alter as less window opening area will result in less airflow as compared to a large opening area hence resulting in maintaining the internal temperature of room/zone. The system gets other noise input in the form of outside temperature that directly affects the roof temperature that changes zone temperature consequently. This disturbance is modeled as a sinusoidal input, as shown in Figure 16 below, considering a sinusoidal variation of outside temperature under an average day effect for a period of 60 min (1 hour). As it is modeled as a random disturbance subjected to changes of varying intensity, the values and subsequent effects on the roof and zone temperature keep on changing. This is done to test the controller under different temperature variations that compensate for the lack of real-time climate data required for simulations.

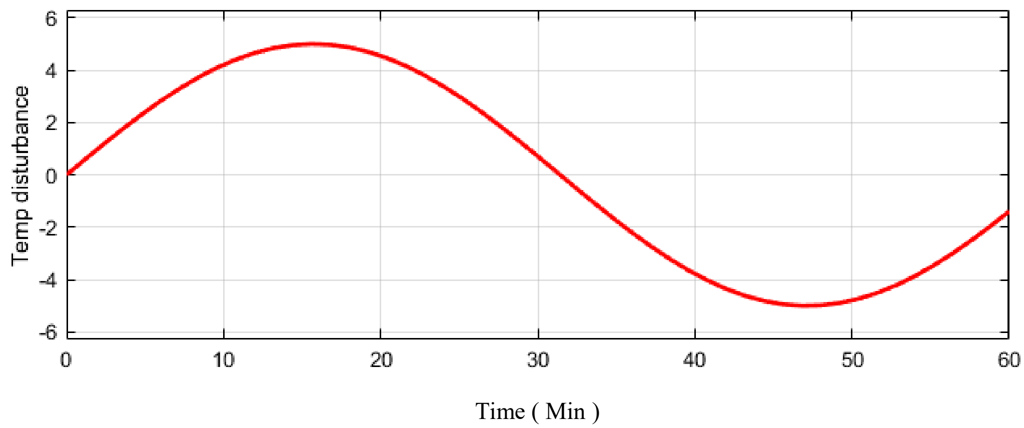

Figure 16: outside temperature variation modeled as varying sinusoidal input under the effect of disturbances

LQR controller gain $K_{lqr} = [-2.0983 \quad -0.1645 \quad -0.2795]$ consists of 1 row, i.e., ventilation flow rate; and three columns representing states zone temperature, roof temperature, and floor temperature, respectively.

The designed controller results in the optimal control of zone temperature while regulating the system states, as shown in Figure 17 below.

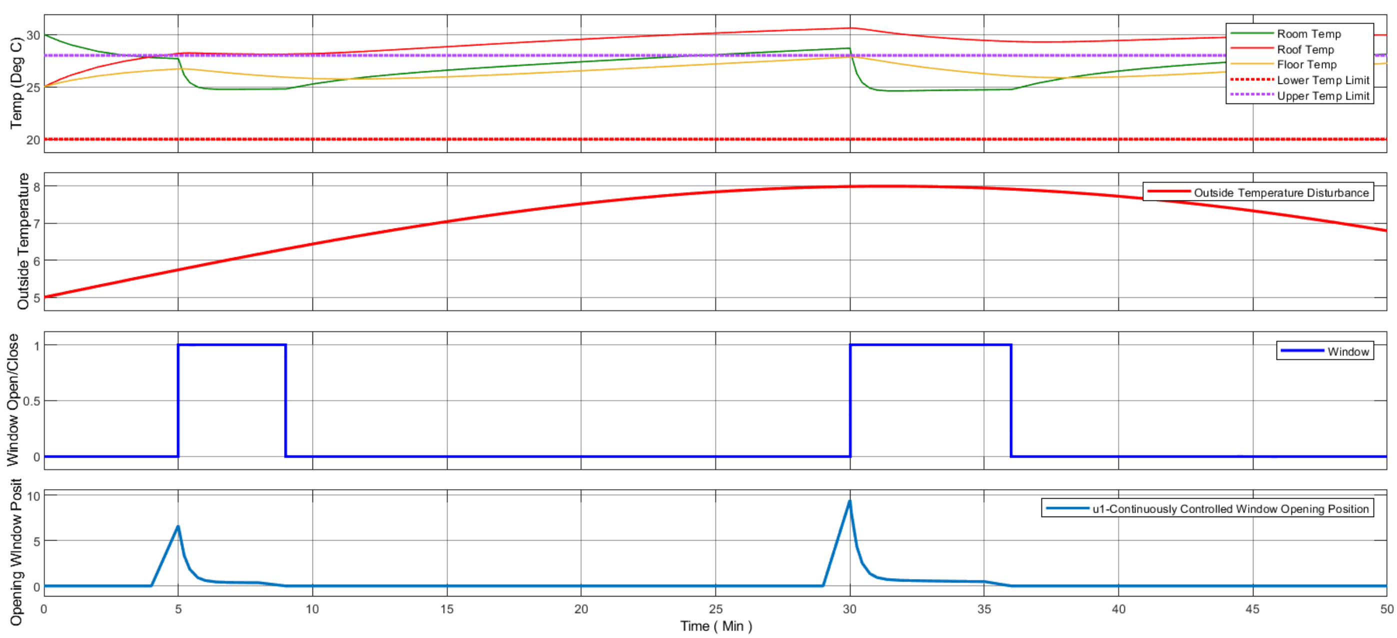

Figure 17: Step response of LQR for desired set point temperature of 25 °C for system's state regulation

Figure 17 shows LQR controller performance. The room temperature initially set at 30 ºC is higher than the set point temperature of 25 ºC. At the start, the LQR tries to regulate the temperature to a set point, but the outside temperature keeps affecting the system's roof, and due to this zone temperature starts to increase again. The controller input is applied to the window when the zone temperature exceeds the upper limit of 28 ºC, which results in the window opening and a step input of ventilation flow due to the window opening position. As soon as the window opens, the air start flowing from outside to inside, and with a continuously controlled window opening position meaning continuously changing of window opening position the amount of air-flown inside changes, the zone temperature starts to decrease and moves towards a set point that is 25 ºC. As the zone temperature starts moving towards the set point temperature, the opening position of the window also keeps on changing, the opening area keeps on reducing accordingly until the set point is reached and the window opening area turns out to be 0 and hence window closes at that point. Along with this, the floor temperature also starts declining. Once the thermal comfort point is achieved, that is, zone temperature reaches the set point temperature, the window is closed with no air exchange from outside to inside.

5.2 LQI Implementation

The purpose of LQI design is to achieve the desired temperature. Providing the desired zone temperature as a reference, the input to the system (i.e., continuously window opening position $\dot{q_{nv}}$, along with heat gain due to solar radiation $\dot{q_r}$ and internal heat gain $\dot{q_{in}}$ as constant input disturbances), can be increased or decreased to achieve the desired performance.

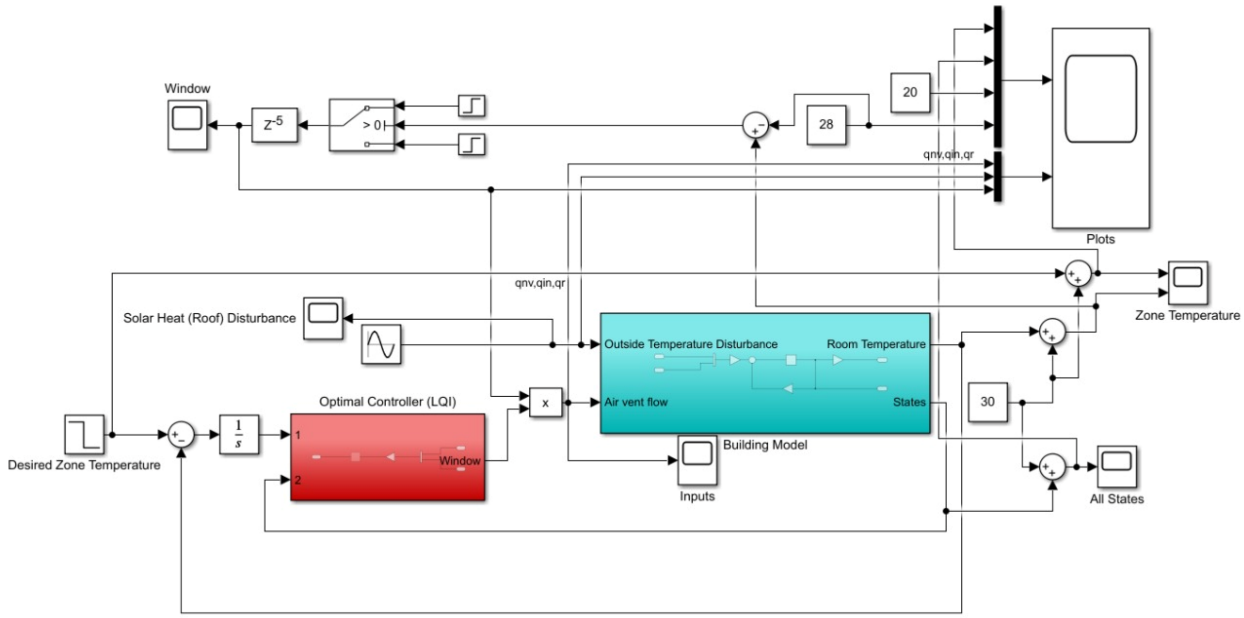

Figure 18 MATLAB LQI control design for zone temperature control through natural ventilation by controlling window

LQI controller gain matrix $K_{lqi} = [-1.8309 \quad -0.0979 \quad -0.1127 \quad 2.2361]$ consists of one row representing ventilation airflow and four columns are representing three states of zone, roof, and floor temperature as in LQR, with an additional fourth state as an error state. The step response of the system shown in Figure 19 shows the controller tracks the desired set point temperature optimally. This is achieved using the weightage matric Q as:

$$Q_{lqi} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{5.2}$$

And $R = [8]$

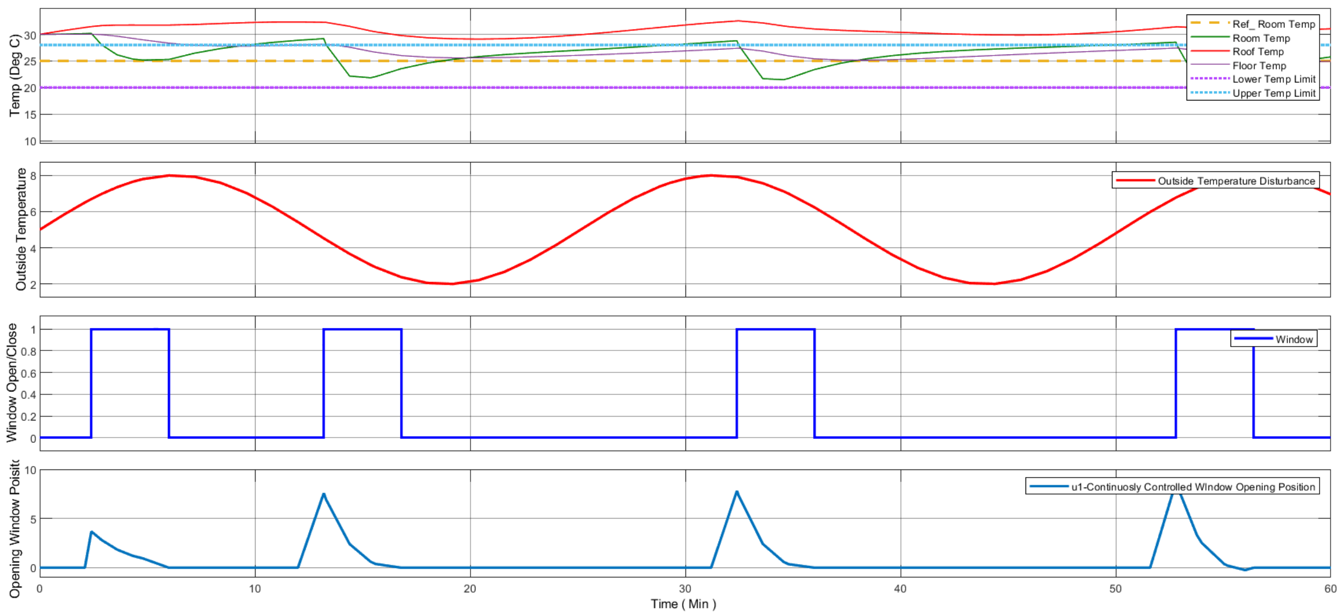

Figure 19: Step response of LQI for desired set point temperature of 25 ºC with outside temperature as disturbances

Usually, the LQI system does suffer from a lack of disturbance rejection, as shown in Figure 19. This is mainly due to the inherent nature of LQI of having stability constraints on the system performance imposed by the instability of the controller's inner loops. This results in increased control effort that shows in the form of frequent window opening and more window opening area required (as shown in Figure 20) and higher fluctuations in zone temperature.

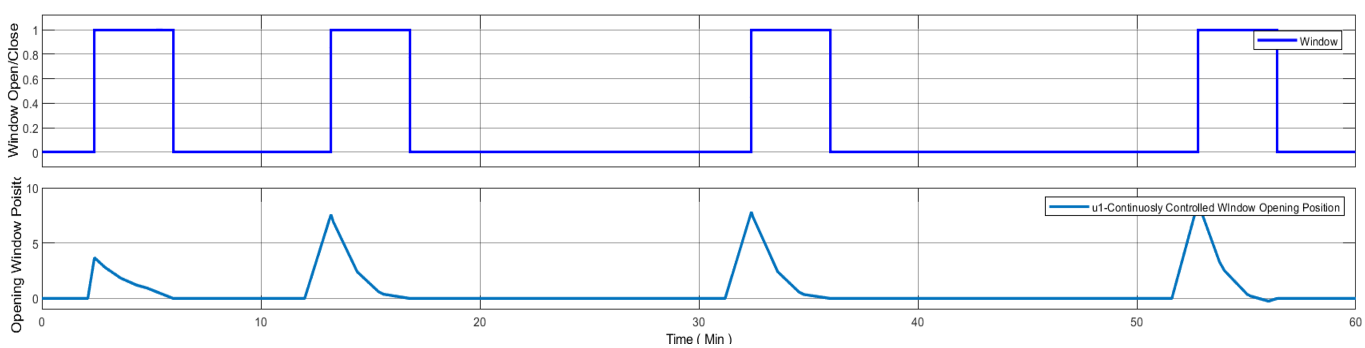

Figure 20: Window position controlled by LQI

Nonetheless, the controller gives optimum performance for set point temperature tracking while keeping the zone temperature in a maximum temperature bound of 28 ºC and a minimum limit of 20 ºC. For a randomly modeled sinusoidal varying outside temperature with disturbance, the controller gives satisfactory performance for zone temperature tracking. However, keeping the temperature disturbances within temperature bounds remains a challenge.

5.3 LQG Implementation

The final choice among optimal controllers for the study is the LQG controller to remove the unwanted disturbances which cannot be measured through the sensor or in the case where the sensor is not available. MATLAB implementation of the modeled LQG controller is shown in Figure 21 below. So, we estimate the disturbance by using a sinusoidal noise signal (as shown in Figure 22) and estimate system states using the Kalman filter. The estimated states of the system come out to be:

$$A_{lqg} = \begin{bmatrix} -0.2639 & 0.1188 & 0.13 \\ 0.1086 & -0.2642 & 0 \\ 0.1368 & 0 & -0.1488 \end{bmatrix}$$

$$B_{lqg} = \begin{bmatrix} -1.539 & 0.01505 \\ 0 & 0.01938 \\ 0 & 0.01196 \end{bmatrix}$$

$$C = \begin{bmatrix} 1 & 0 & 0 \\ 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{5.3}$$

As a result of the accurate rejection of input disturbances in the roof and floor temperature due to disturbances in outside temperature, the D matrix of the estimated states is zero.

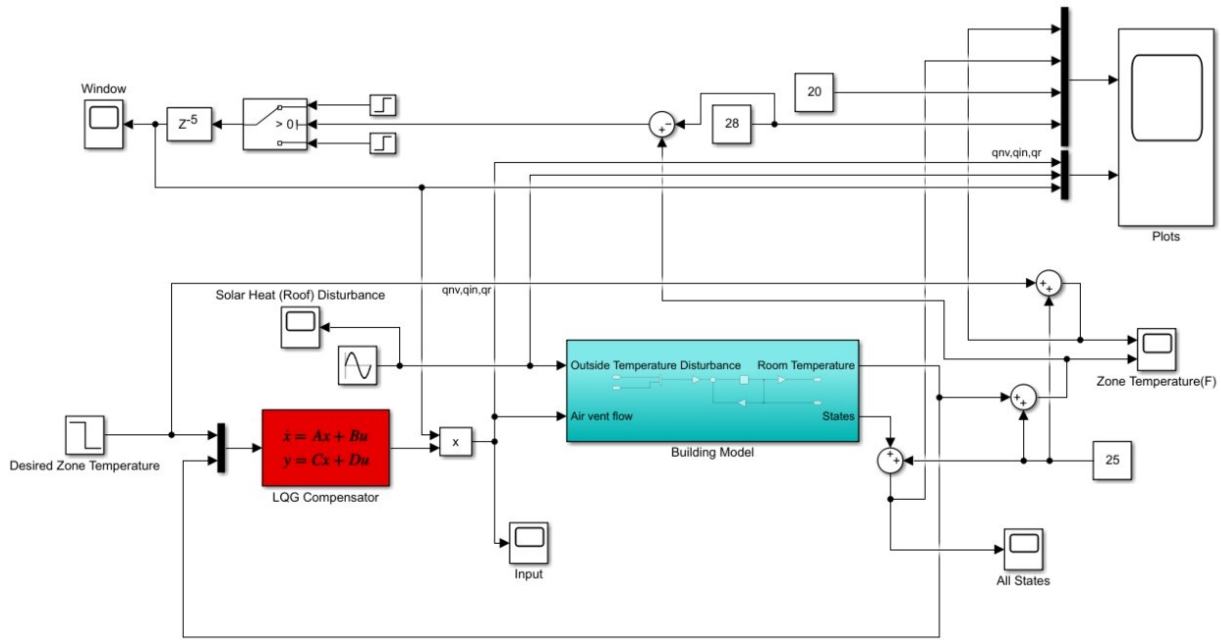

Figure 21 MATLAB LQG Control architecture designed for zone temperature control

In our case, we have added sinusoidal noise as input disturbance that causes an increase or decrease of roof temperature directly and eventually causes an increase or decrease in room temp and floor temp as well.

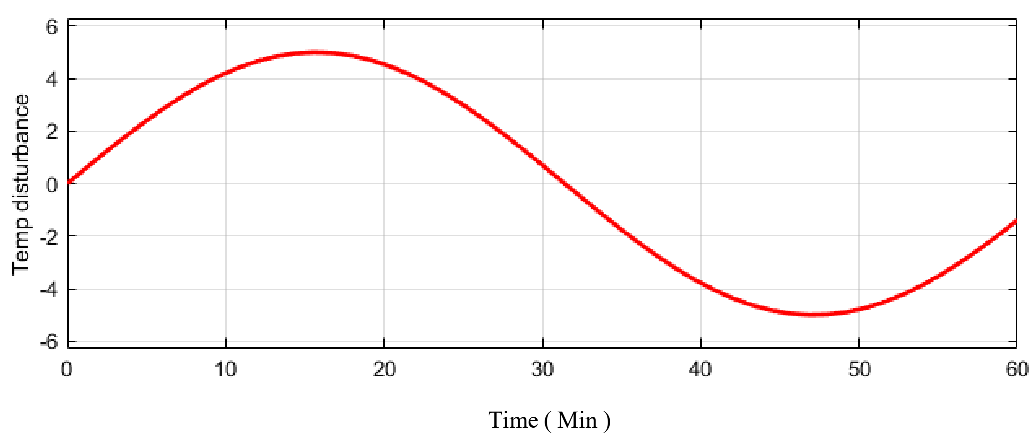

Figure 22: Outside temperature disturbance modeled as sinusoidal noise signal for LQG design

Figure 23 shows the system response with a varying sinusoidal input.

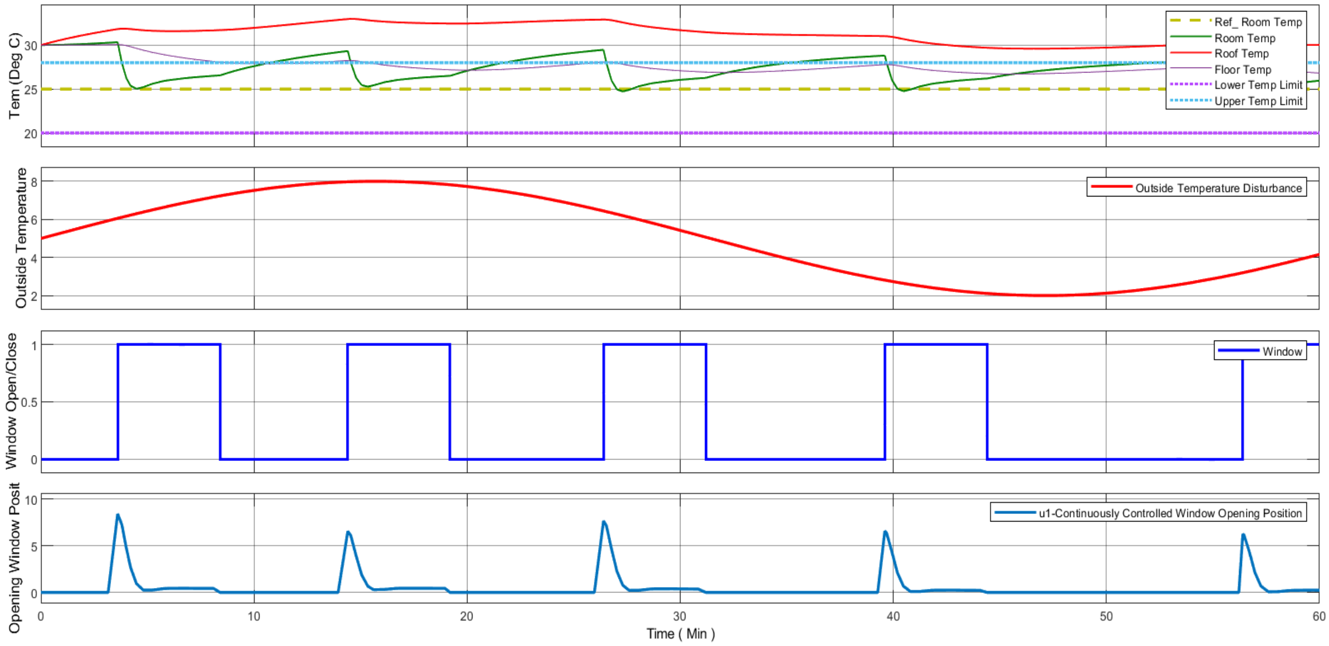

Figure 23: System response with LQG implemented for set point zone temperature of 25° C with minimum and maximum temperature bounds for sinusoidal input

This is achieved by the window control input shown in Figure 24 below.

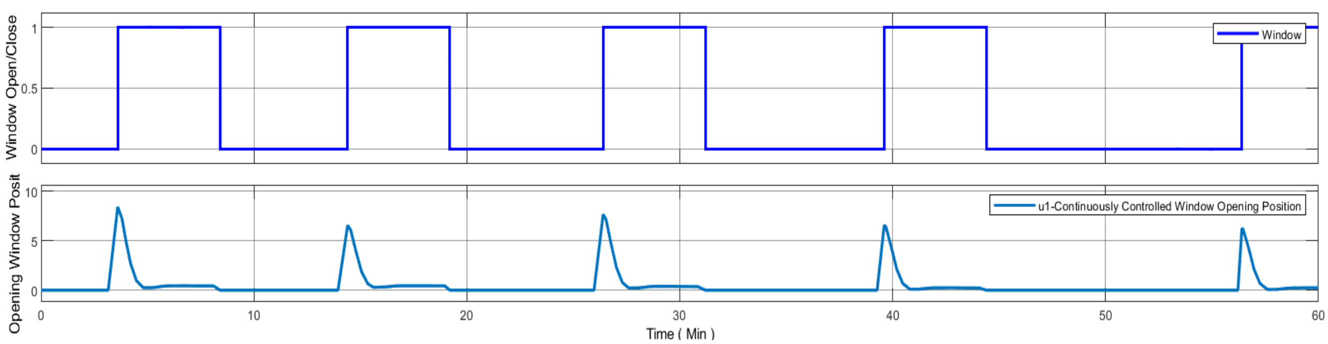

Figure 24: Window control pattern for LQG

The system response shown in Figure 24 illustrates the optimal control performance of the LQG. The controller is shown to keep the zone temperature in desired temperature range after two input pulses and a slightly increased control effort in a form of more opening area of the window for some time and then reducing the window opening area position as the zone temperature started coming near the set temperature.

5.4 MPC Implementation

Model predictive control is an advanced optimal control version from LQI and LQG. As discussed, it not only deals with disturbances but also caters to inputs and outputs constraints. Figure 25 below illustrates the control architecture designed for zone temperature control through natural ventilation.

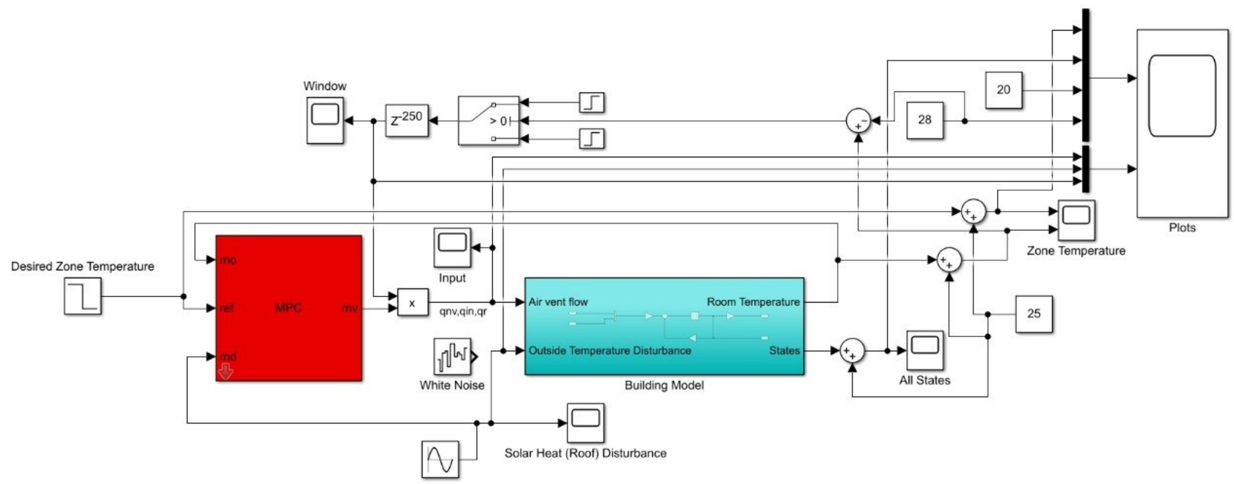

Figure 25 MATLAB MPC Control design architecture for zone temperature control with window opening area as input and outside temperature disturbances

Using MPC designer in MATLAB, the control law is gain-tuned for Prediction horizon $P = 100$ and control horizon $M = 2$. With the controller adopted sampling time of $t_s = 0.08\, sec$, the controller setup is shown in Figure 26 below.

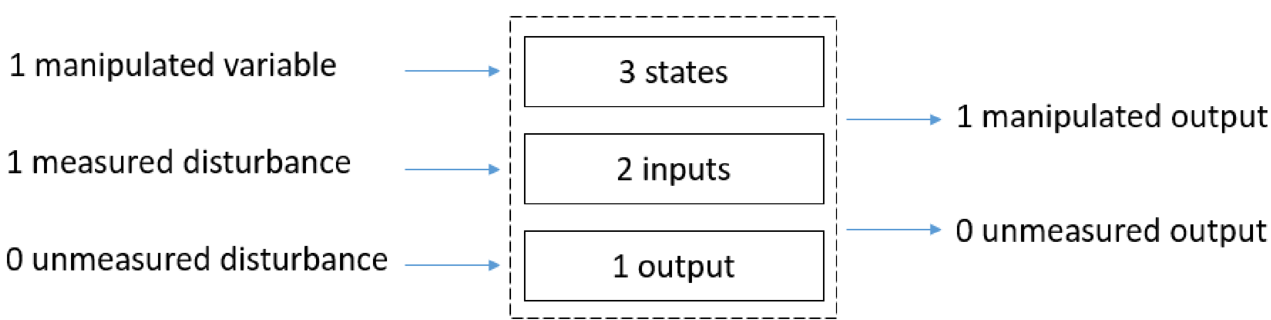

Figure 26 MPC input-output and state identification

When used for the temperature control model, it estimates the disturbance and removes its effect on the system efficiently. It tracks the reference command and deals with constraints, such as if we need to limit our room temperature from 20 to 28, as shown in Figure 27. The calculated state matrices for MPC are:

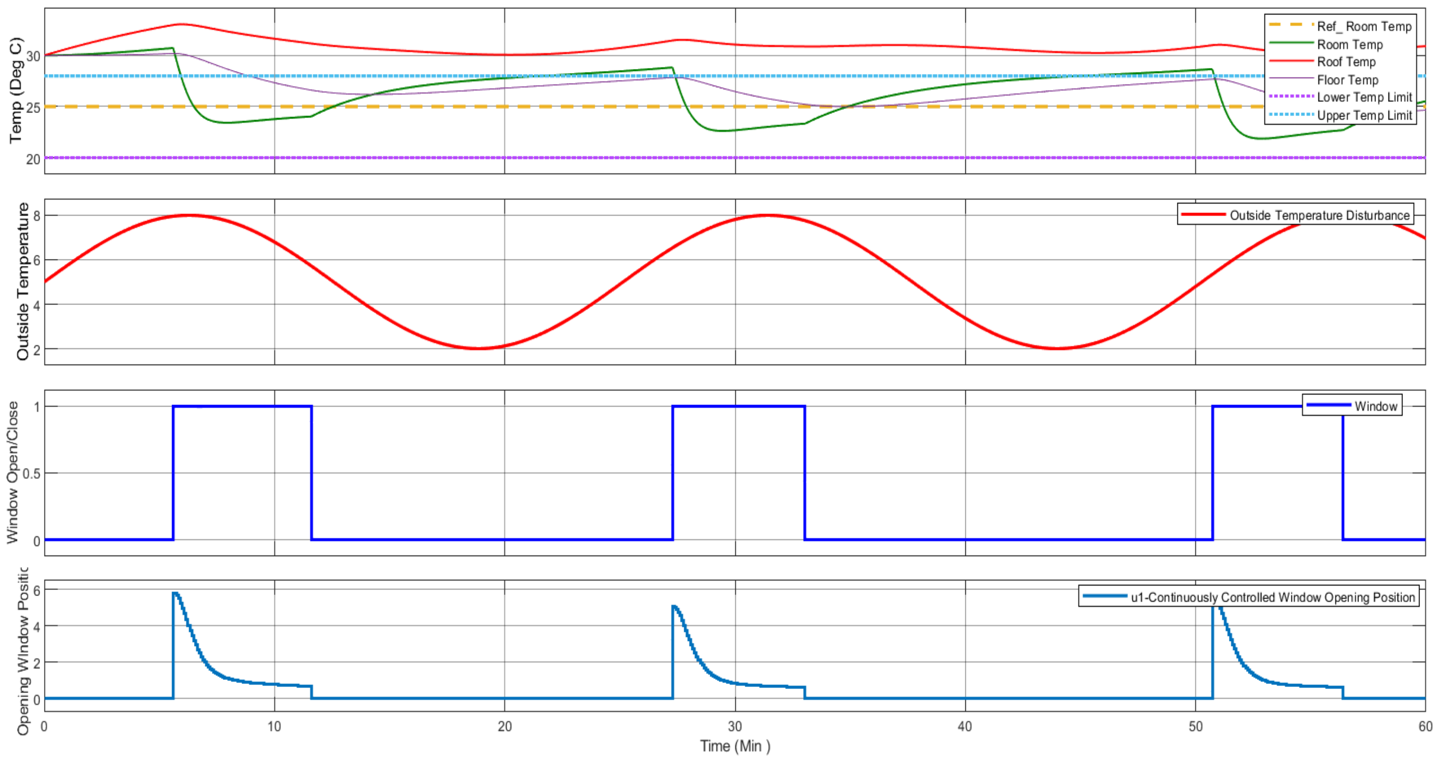

Figure 27 Step response of MPC control for zone temperature control with upper and lower temperature limits as constraints and window opening area as input with disturbance in outside temperature

Even for the sinusoidal input, the controller shows accurate temperature control and regulation (Figure 27) with a controlled window opening position and continuously changing opening area of the window. Also, the disturbances are well regulated in the form of regulated roof temperature, which projects efficient controller performance under varying input of airflow prone to disturbances and uncertainties in outside temperature, solar radiation, and internal heat gain. This requires less window operation with less opening areas, leading to more energy savings (Figure 28).

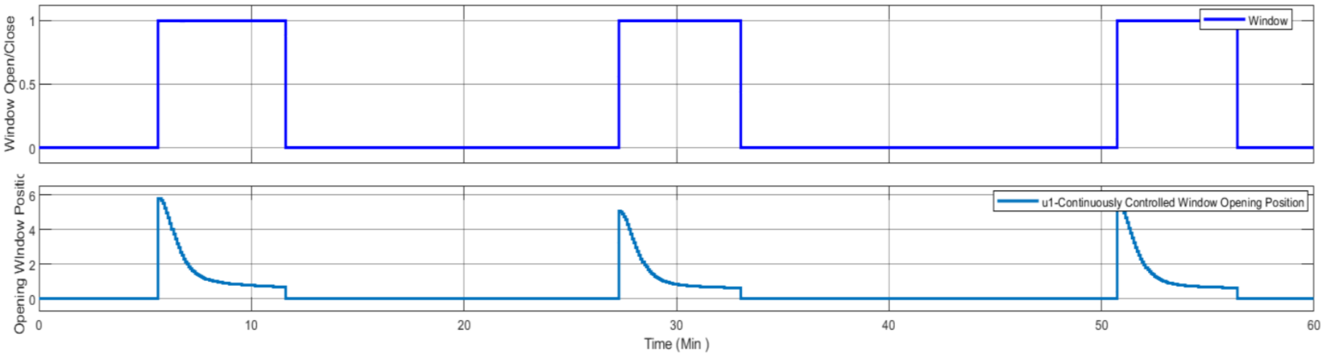

Figure 28: Window opening and respective window opening area with MPC as a controller for temperature control

5.5 Discussion and Analysis

Starting with LQR as a basic controller for temperature control, it shows that the LQR can be used as an efficient temperature regulator. Though good for regulation, reducing steady-state error remains a challenge for systems subjected to uncertainly changing dynamics with LQR alone.

To make the controller track the set point temperature, LQI is implemented in the system. However, to track the set point zone temperature and maintain the temperature within temperature limits, LQI demands more control power than LQR, as LQI finds an optimal state-feedback control law for the tracking loop. It stabilizes the augmented states and minimizes the cost function associated with step reference inputs and step disturbances (as shown in figure 19) since it is a static feedback controller based upon augmented states, i.e., plant state plus integral of tracking errors. LQI control's achievable dynamic performance is mainly constrained by open-loop instability. Although solutions exist to solve this problem through careful selection of diagonal weightage matrix Q, it compromises controller robustness and system performance. This leads to the adoption of an efficient control scheme for clear disturbance rejection and uncertainty handling.

LQG, on the other hand, shows the best optimum control results for the designed building model with better disturbance rejection and efficient temperature control. However, the natural ventilation system is a slow system with the system's dynamics changes happening at slower rates like usual natural processes, and to keep the zone temperature control LQG requires more control effort in a sense of more window opening with a large window opening area. The system subjected to similar disturbances, when controlled by MPC, requires lesser control effort compared to LQG, as shown in the results.

Though all four controllers show optimum performance, the difference comes in three forms, reference temperature tracking, disturbance rejection, and control effort. For a set time interval, LQR responds to the system's changing dynamics very slowly with an increased control effort required for temperature regulation, as shown in Figure 17 in the form of a large opening area of the window on each following interval. On the other hand, LQI shows a robust response towards system dynamics as the temperature is brought to a set point temperature in less time. However, the output which is zone temperature keeps oscillating about the set point temperature. The reason for this is increasing roof temperature under the effect of outside temperature as a disturbance. As a result of which, the zone temperature and floor temperature also keep fluctuating about the set point. Consequently, the control effort that is window opening area position keeps increasing with increasing time which shows in the form of the frequent window opening and increased ventilation flow required due to the large window opening area (Figure 19) for the same window opening at each passing interval.

The system, when controlled by LQG, shows efficient temperature control with optimum control effort for the same interval of time in the form of a less window opening area and lesser ventilation flow due to less opening required for temperature control. Also, the increase in the opening area of the window required for the next opening interval is relatively higher than that required in the previous opening interval, yet it is satisfactory and allowable as compared to LQI.

For almost similar window opening intervals, MPC shows lesser control effort than LQG. This is mainly due to the prediction property of the controller of the following temperature value. This allows the controller to minimize the ventilation energy required for the next time interval.

As per calculation of eq 3.39, using MPC with zone temperature ranging from 20 ºC as the minimum and 28 ºC as the maximum value, a mean radiance temperature of 25.7 ºC, with a relative humidity of 50%, a PMV of 0.0013 is achieved which lies in the range of $-0.5 < PMV < +0.5$ for general thermal comfort shows the MPC performed better when compared to all other controllers.

CHAPTER 6: CONCLUSION AND FUTURE WORK

The natural ventilation system is a slow system, with the system's dynamic changes happening at slower rates. This thesis has been dedicated to the control design and analysis of temperature control for a naturally ventilated system. The system studied is modeled as a single zone with one window as the system's control actuator with airflow rate through the open window as the control input. The control approach is to maintain system temperature within the desired temperature range of 28 ^{o}C to 20 ^{o}C, which is a preferred temperature range by building occupants generally. The selected controllers show optimum control performance under modeled building parameters, climatic parameters, and internal and external environmental disturbances.

As usual natural processes and keeping the zone temperature control LQG requires more control effort. The system subjected to similar disturbances, when controlled by MPC, requires lesser control effort compared to LQG, as shown in the results

6.1 Challenges and Limitations

One of the significant challenges for the study was the data modeling for the system model. MPC requires an accurate mathematical model to represent system dynamics for accurate optimization and precise prediction. This also puts constraints on the overall system's computation requirements. An efficient control design requires an accurate weather data model for accurate temperature predictions and temperature control. Also, there is an effect in thermal transference terminology known as the flywheel effect. This is when the thermal mass provides inertia against the temperature changes in a body. For example, when there is a considerable variation in temperature in outside air, a specific portion of the house acts as ample thermal mass storage. It absorbs heat and then releases it when the outside temperature cools down. This is, however, different from the insulation ability of a material, which only allows the material to stay hot or cold or just retains the body heat during external temperature variation by lowering its own thermal conductivity.

In terms of controls, due to the unavailability of real-time data, the controllers are applied for a short time interval. Therefore, only step responses of the system are assessed and analyzed. Although the system can be scaled and simulated for a longer time, the response remains the same and hence can be assessed with short time intervals as well. Further, this overlooks various system uncertainties' and controllers' responses to them as the system ought to show similar responses with varying times if system conditions are not varied over that time period. This also puts constraints on disturbance modeling. For this study, the disturbances are modeled as sinusoidal outdoor temperature T_e and fixed internal heat gain with fixed radiation heat, and it was challenging to assess the effect of disturbance of temperature variation for such a short interval of time.

6.2 Future work

1. Control can be applied to the system with real-time data. The data needs to be modeled in the form of outside temperature variation under the effect of changing weather conditions during day/night time, weekly, or monthly basis. Also, the current model keeps the system disturbance's internal heat gain and the effect of solar radiation on the system constant. For a real-time setup, these will also vary depending upon the number of occupants, type of equipment and frequency of its use, lighting conditions during different times of the day and night, and changing weather's effect on solar radiation.

2. Window position is modeled with the window to be either fully open or fully closed for simplicity of system modeling and control design for robust response analysis. A window can be modeled with varying opening patterns like half open or a percentage of area in the case of sliding windows and at a certain angle in the case of slanted windows.

3. Another improvement can be the incorporation of the temperature range. For systems with passive thermal storage like that of natural ventilation, the controllers are shown to provide the best optimum control when provided with a temperature range to allow the set point to float in the set upper and lower bound.

4. Although the designed controllers show a satisfactory response to the system modeled, the results can be refined and validated further with more accurate data modeling. This data is in the form of weather data sets, disturbance models, and constraint models.

5. For the current study, MPC is modeled and applied to the linearized system for accurate system modeling and performance assessment. But MPC is also a robust and efficient nonlinear controller owing to its accurate prediction and precisely satisfactory response and control of systems with slower dynamics

6. The controllers can be tested for system models with additional climatic parameters such as humidity and Carbon di Oxide control for increased thermal comfort in the sense of indoor air quality. Similar controllers can be defined for indoor air quality control by using a mass balance model. Mass transfer can account for the accumulation of carbon dioxide, moisture content or water vapors, and other pollutants. So, we can track air quality with the help of mass balance. Mass is transferred in two ways, advection, and diffusion. In advection, fluid crosses a boundary in bulk flow, and energy is transferred in this manner. In the second type of mass transfer, molecular diffusion is caused by concentration differences resulting in energy loss through phenomena like evaporation.

   The general equation of mass balance is written as follows

$$Accumulation = Input + Generation - Output - consumption$$

   Mathematically,

$$\frac{dm_{cv}}{dt} = m_{in} + r_{gen} - m_{out} - r_{cons}$$

   If there are no chemical reactions involved in the considered system, the equation reduces to

$$\frac{dm_{cv}}{dt} = m_{in} - m_{out}$$

   Here, m_{in} is the mass flow of air entering the control volume, m_{out} is the mass flow of air exiting the control volume, r_{gen} is the generated mass within the control volume, and r_{cons} is the mass consumed within the system.

   So, using this system model and by adding more uncertainties to this system, we can further provide natural ventilation and thermal comfort to the occupants by utilizing similar kinds of optimal controls.

7. The system can be upgraded to a hybrid model with day and night cooling control schemes, provided comprehensive weather data is available.

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APPENDIX A

Table 1 – Building envelop properties

Envelope	**Window**	**Floor (W/meter sq. kelvin)**	**Roof**	**Zone**
Layers	Single	Single	Single	Single
Length (m)	1	6	6	6
Width (m)	1	6	6	6
Height	1	-	-	3
Thickness (m)	0.001	0.15	0.15	-
U value (W/m^{2}K)	-	880	935	100z
Solar transmittance cumulative (gain)	0.498			

Table 2 Control Parameters

Parameter	**Symbol**
Set point temperature	T_z
Roof temperature	T_r
Floor Temp	T_f
Outside room temperature	T_o

Table 3 Modeling parameters for thermal comfort modeling (ASHRAE 55/62)

Parameter	**Symbol**	**Value**
Internal heat gains Equipment People (latent + sensible heat)	q_{in}	500 Watts
Solar Radiation Heat	q_r	50 Watts
Airflow rate	$\dot{u}$	0.2 m/s for MET 1 – 2 and clo of 0-1.5 (from AASHRE 62) For the UK 0.4 – 1 from CIBSE guide A

Table 4 System Inputs, States, and Outputs for system modeling and control design

Inputs	**States**	**Output**
Outside air temperature(sinusoidal)	Zone Temperate	Indoor air temperature (zone temperature)
Internal heat gain (fixed)	Floor temperature	Thermal comfort index (PMV)
Heat gain from a window (Controllable)	Roof temperature	
Radiation heat gain (fixed)		