REDUCED ORDER MODELING FOR VIRTUAL BUILDING COMMISSIONING

by

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The following individuals read and discussed the thesis submitted by student Sean Rosin, and they evaluated his presentation and response to questions during the final oral examination. They found that the student passed the final oral examination.

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DEDICATION

I want to dedicate this research to everyone who helped me achieve this accomplishment.

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ABSTRACT

Model order reduction can help reduce the time and monetary constraints associated with building commissioning and significantly decrease overall building energy consumption through virtual commissioning. This research aimed to determine the effectiveness of using reduced order models to simulate the overall building energy consumption, and to estimate the energy savings from control-based commissioning recommendations.

A case study building was modeled using a ‘Lumped RC’ thermal model with three thermal resistances and capacitances (3R3C) for the building interior and a 2R1C model describing the building foundation. Due to energy consumption being dependent on building systems, this model was coupled with a simplified HVAC model to translate indoor zone temperature predictions into total annual energy consumption. The coupled reduced order model (ROM) model was compared to an identical model constructed in EnergyPlus, and it was determined that a reduced order model was capable of predicting annual energy consumption.

The case study building lacked thermostat setbacks during periods the building was unoccupied, and the ROM was used to predict the energy savings associated with updating the controller. It was found that approximately 104,000 kWh of potential energy savings could be realized if the thermostat had properly programed temperature setbacks during times the building is unoccupied.

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LIST OF ABBREVIATIONS

AHU	Air Handler Unit
AMY	Actual Meteorological Data
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BACnet	Building Automation Controls Networking Protocol
BCVTB	Building Control Virtual Test Bed
BESTEST	Building Energy Simulation Test
COBE	College of Business and Economics
CVRMSE	Coefficient of Variation of the Root Mean Square Error
EMS	Energy Management System
HVAC	Heating Ventilation and Air Conditioning
IDL	Integrated Design Lab (University of Idaho)
MASP	Mixed Air Set Point Temperature
MPC	Model Predictive Control
NMBE	Normalized Mean Bia Error
ODE	Ordinary Differential Equation
ROM	Reduced Order Model
TOT	Terminal Outlet Temperature
VAV	Variable Air Volume
VFD	Variable Frequency Drive

CHAPTER ONE: INTRODUCTION

The Scope of the Study

Buildings are one of the primary users of electricity throughout the developed parts of the world. According to the U.S. Energy Information Administration, it is estimated that in 2015 about 40% of all U.S. energy consumption was through residential and commercial buildings, which is approximately 39 quadrillion British thermal units [1]. Building energy consumption has risen steadily over the last three decades at an average rate of 1.9% per annum for all North American countries. Factors leading to the continuous rise of building energy consumption are population growth, thermal comfort enhancement, and an increase in time spent in buildings [1]. In 2003, HVAC accounted for approximately 53% in the U.S., 42% in Spain, and 62% of the United Kingdom out of the total energy consumed per building [2]. Space conditioning accounts for approximately half of all energy consumed per building throughout all developed nations, which equates to approximately 20% of the total energy consumption annually [2], with the other 20% originating from plug loads, lighting, and other internal processes.

Commercial and residential space conditioning is highly dependent on the building's automating and control systems (BACs), which monitors all parameters that affect the performance of the space conditioning equipment. The BACs usually are programmed during the commissioning of the building, and then they are left unmonitored, leading to potential problems with the energy management system's (EMS) control logic, which can go unnoticed for long periods of time leading to exorbitant and unnecessary energy costs. It was found that non-optimal controls can account for an additional 15%-30% of equipment degradation and malfunction [3]. While new buildings increasingly rely on automated controls for their EMS, commissioning these controls in new buildings and verification of current sequences in existing buildings is a time intensive process, runs the risk of suboptimal occupant comfort, and exposes the building owners to unnecessary liability and higher energy costs during the time period before commissioning is accomplished.

New building control commissioning typically takes one full year of building operation so all weather conditions and operational modes can be experienced, and often takes two full years before the system is operating nearest to its optimal potential. In addition, commission is a time-consuming process leaving large gaps in the verification process while waiting for some modes of operation to occur. Older buildings can also suffer from poor controls that are out of tune with the current building occupancy patterns or not up to date with current control techniques. The building commissioning process encompasses a wide scope, starting with design development and ending at least one year after the building is occupied [4]. In current practice, suboptimum and incorrect control programming can take months or years to detect, if they are at all. When controls issues arise, they can also be difficult to reproduce and take weeks or months to rectify [5]. Operational issues can also go undetected, especially if they do not directly affect human comfort. One way to ensure that the controls are functioning properly is through whole building simulation. Simulating the thermal performance of the building generates a baseline for energy consumption which can be compared and tested with different control techniques to find optimal operational modes and identify underlying control errors that may go unnoticed.

Simulation-based commissioning holds potential as a way to reduce or avoid the hazards associated with traditional commissioning approaches. The research previously carried out at the University of Idaho's Integrated Design Lab (IDL) [6] used energy simulations as a tool to virtually commission buildings. In this research, a hardware clone of the building controller was connected to an EnergyPlus model of the College of Business and Economics (COBE) building in Moscow, ID. This was accomplished through enabling communications between the controller and the model using the Buildings Control Virtual Test Bed (BCVTB). BCVTB is "middle-ware" which translates the outputs from the EnergyPlus simulation to either a voltage or digital input that the building controller can understand [7]. The variables that were chosen for the study included outdoor air temperature, outdoor air damper position, mixed air temperature, and return air temperature. The Alerton controller required inputs from other equipment and feedbacks from each of the thermal zones, which was not practical to model in EnergyPlus due to computational limitations. These inputs were bypassed by adjusting the logic to allow the controller to continue to function without each individual feedback loop. This method of simulation-based commissioning is less time intensive than traditional approaches, but developing an accurate energy model also takes time and knowledge that most practitioners do not possess. Although the cost of commissioning a building is prohibitive for many owners, the research demonstrated that virtually commissioning a building is a viable alternative. The current phase of the research, as described in this thesis, explores ways to reduce the time and monetary expenditures of virtual commission still further.

The current research aims to simplify the modeling process to allow practitioners a means of virtually commissioning a building without the steep learning curve associated with modeling in EnergyPlus. This approach reduces the modeling time, allows for innovative control strategies to be investigated quickly, and can be used by practitioners to quickly diagnose an operational or control issues. There are still limitations with reduced order energy modeling that need to be addressed before the methods of virtual commissioning can be fully utilized.

The COBE building was also chosen for the ROM virtual commissioning research so the new method's results could be compared against the calibrated baseline EnergyPlus model from the previous research. The COBE Building was chosen for the previous research due to the building's controller communicating through a standard building automation and control network protocol: BACnet. This communication protocol was essential for the research so that the energy model could interact with the controllers in a standard way.

The COBE building's ROM was composed of two sets of differential equations: one equation with three thermal resistances and three thermal capacitances (3R3C) to describe the dynamics of the buildings, and one equation with two thermal resistances and one thermal capacitance (2R1C) to describe the dynamics of the foundation. The thermal parameters of both models were determined through optimizing the ROM's zone temperature with the optimization baseline being the zone temperature as simulated by EnergyPlus.. This only predicted the internal zone temperature of the COBE building which does not have a direct correlation to overall energy consumption. To use this model as a virtual commissioning tool, the model need to accurately predict overall energy consumption, which was done through modeling the heating, ventilation and air-conditioning system (HVAC) at the COBE building in a separate model that was coupled with the ROM.

The HVAC system needed to be an accurate representation of the actual system, so each individual component was modeled. This included the terminal reheat box, the air handler heating and cooling coils, the economizer, as well as the supply and return air fans. The ROM was coupled with the simplified HVAC model to predict the total annual energy consumption of the building which then could be used to determine potential energy saving measures from control based virtual commissioning.

CHAPTER TWO: BACKGROUND

When designing a building, the heating and cooling loads need to be calculated to ensure the HVAC system is sized accordingly and can meet the cumulative demands of the spaces. If these loads are miscalculated, the HVAC will not be able to correctly condition the building causing the unit to consume excess energy and cause unneeded wear on the mechanical equipment. In its simplest form, the energy the HVAC system must remove can be described as:

$$Q_{BL} = Q_{Gain} - Q_{Loss} \tag{2.1}$$

The amount of energy that needs to be removed from the space is the difference between the energy gains and losses (Q_{BL} denotes the building load with positive indicating heat entering the space). Thermal zones gain and loose energy through numerous methods and each one needs to be meticulously accounted for if the HVAC system is going to operate optimally. Figure 1 shows a representative space with typical heat gains for most residential and commercial buildings.

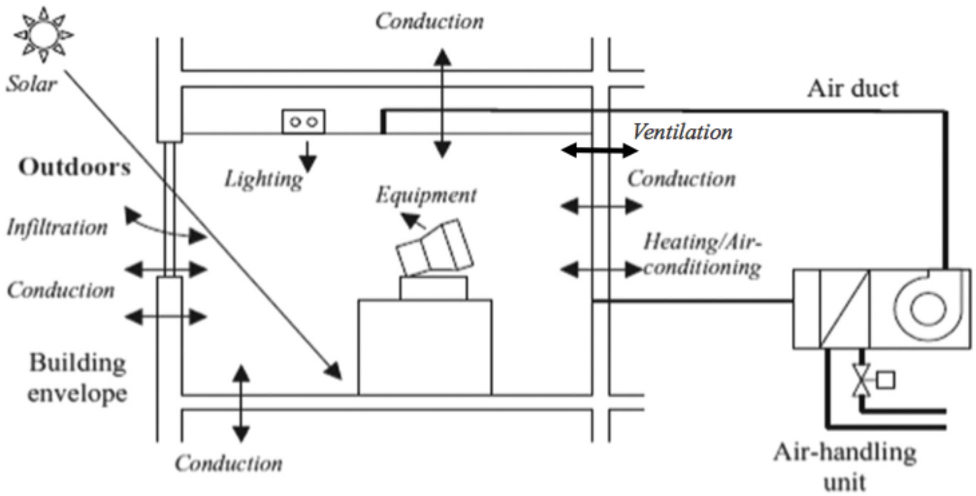

Figure 1. Processes of Energy Transfer in a Conditioned Space [8]

Zone heat gains include radiation from lighting, convective gains or losses from infiltration, ventilation, solar gains, and conductive heat transfer from the exterior and adjacent zones to name a few. Each heat transfer mode may include both latent and sensible gains, a table labeling each building element with its respective heat transfer mode can be seen below in Table 1.

Table 1. Building Elements and their Modes of Heat Transfer [8]

Heat and Mass Transfer Process	*Building Elements*
Conduction	External wall, roof, ceiling and floor slabs and internal partition wall, doors, skylights
Conduction heat transfer and solar radiation transmission	Window glazing
Conduction and/or radiation heat transfer and moisture dissipation	Occupants, lights, and other equipment
Convection heat and mass transfer	Infiltration from outside and adjoining spaces, ventilation from supply air

Internal equipment discharges a known amount of energy into the space and the total amount of energy dispersed into the space from equipment can be determined if the quantity and rating of the lights, computers, etc. in the building are known. This is true with solar gains as well, by knowing the orientation, glazing material, frame type, etc. the amount of radiation emitted into the space can be determined, but when the building has multiple zones, these tasks becomes extensive and unfeasible without simulation tools.

There are several different modeling approaches used in whole building simulations: black-box, grey-box and white-box modeling. Black-box modeling is a data driven modeling approach which uses time series data to statistically fit a model to determine building parameters. Black-box modeling does not provide any information about the behavioral mechanism of the building [9], it solely is a statistical representation of building data correlations. Black box models only focus on finding relationships between the model's inputs and outputs [10], which is useful for predicting building performance given a specific outdoor condition, but not especially for virtual commissioning.

Another modeling approach is white-box modeling, which are modeling approaches developed through physics and first principles [10]. One of the best known white box modeling techniques for whole building energy simulation is EnergyPlus. EnergyPlus was developed for engineers and architects to model energy and water usage of buildings, but this process is exhaustive and can take several months to accurately complete. All the building's geometry, building construction, zoning characteristics, and HVAC controls and layouts must be defined properly for Energy Plus to accurately predict energy consumption of the building. Equipment and material degradation cannot be accounted for as the building ages, so once the building has been virtually constructed the model must undergo calibration using actual building energy data. Due to this lengthy process, many building owners tend to veer away from whole building energy modeling, and as a result many energy savings opportunities go undetected. One approach that has been underutilized is grey box modeling.

Grey-box modeling is still built on the foundation of first principles, but in conjunction, it also uses parameter optimization with actual operational data [9]. For thermal systems, grey-box modeling uses sets of differential equations to model the dynamics of heat transfer and thermal storage. There are no limits to the order of the system, but as the complexity of the model increases so does the computational expense, and taken to the limit, the model approaches the complexity seen in white-box modeling. Each model order provides an additional differential equation, and each additional order equates to an additional set of dynamics that must be accounted for. The differential equations form coupled systems and their solutions contain the time constant of the building, which is related to the decay of thermal energy and it is composed of the effective thermal resistance and thermal capacitance of the building. These parameters are optimized using actual data from the physical structure to get the best fit. This modeling technique is less computationally expensive than white box modeling, in addition, developing an accurate simulation to predict the thermal performance of buildings is much quicker.

Reduced Order Thermal Models (Grey Box Models)

Low order models are primarily composed of two parameters, the thermal capacitance and the thermal resistance. The thermal capacitance is analogous to electrical capacitors, both store energy but instead of electrical energy, thermal capacitor stores thermal energy, the amount of which is indicated by the temperature of the thermal capacitance. This parameter is a function of known material properties defined as:

$$C = \rho C_p V \tag{2.2}$$

Where C is the thermal capacitance of a material [J/K], ρ is the density of the material [kg/m^3], C_p is the specific heat [J/kgK], and V is the volume [m^3]. The rate of energy storage in a system is described as:

$$Q_{stored} = C\frac{dT}{dt} \tag{2.3}$$

When a building has a large effective thermal capacitance, otherwise known as massive construction, the rate at which the building's temperature can change due to environmental and internal effects is low. The thermal capacitance is an important parameter to estimate the transient behavior of a building [11], but oftentimes is hard to calculate even when the material properties of a building are known. Another parameter used to describe thermal systems is the thermal resistance, which is the material's natural tendency to resist the flow of heat. There are several different forms of thermal resistance, all having units of [K/W] and all describing the resistance to heat transfer. It is known that the amount of heat transferred through a material is inversely proportional to the effective thermal resistance (R) as is shown in Equation 2.4.

$$\dot{Q} = \frac{T_1 - T_2}{R} \tag{2.4}$$

Where the amount of heat stored within a system is given by:

$$\dot{Q}_{Stored} = \dot{Q}_{in} - \dot{Q}_{out} + \dot{Q}_{gen} \tag{2.5}$$

Where $\dot{Q}_{in}$ and $\dot{Q}_{out}$ is the amount of heating flowing into and out of the system respectively, and $\dot{Q}_{gen}$ is the amount of heat generated within the system. It is assumed the heat losses of the system are negligible and the walls of the building have no internal heat generation. Combining Equation 2.2, 2.3, and 2.4 yields a first order differential equation describing how the transfer of heat is related to the change in energy storage and the thermal parameters of a material:

$$C\frac{dT}{dt} = \frac{T_1 - T_2}{R} \tag{2.6}$$

Equation 2.6 is the foundation of applying ROMs to describe the dynamics of buildings. This method can only be employed if a homogenous temperature distribution throughout each lump is assumed. A material can be broken into an infinite number of lumps, but each additional lump increases the system order which increase the computational complexity and increases the run time of the simulation.

Reduced order thermal models are commonly referred to as lumped RC models and are represented using thermal circuits which ware similar to electrical circuits, an example is shown below in Figure 2.

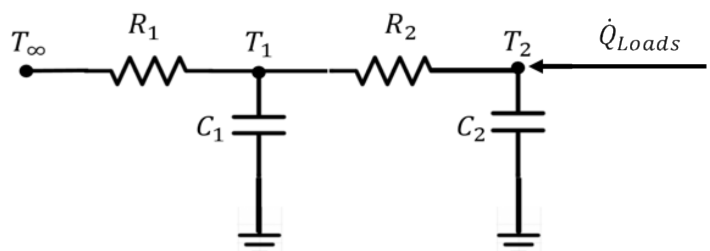

Figure 2. 2nd Order Lumped RC Thermal Network

ROMs are one of the most powerful methods to model dynamics systems due to their simplicity when compared to other approaches. There exists a minimum number of variables, i.e. states, that when known can completely describe the system [12]. These states are measurable and for thermal systems they are the temperatures of the effective heat capacitances. State variables can be described using vectors, and the linearized state space representation of the thermal circuit illustrated above is shown in Equation 2.7.

$$\dot{T} = \boldsymbol{A}\vec{T} + \boldsymbol{B}\vec{U} \tag{2.7}$$

Where $\vec{T}$ is a vector of the temperatures of the effective heat capacitances and $\vec{U}$ is a vector of all system inputs. **A** and **B** are coefficient matrices containing the thermal parameters describing the relationship between the system inputs and the desired outputs. Equation 2.7 can be expanded and expressed in matrix form for the second order model shown in Figure 2.

$$\begin{bmatrix} \dot{T}_1 \\ \dot{T}_2 \end{bmatrix} = \begin{bmatrix} -\frac{1}{C_1}\left(\frac{1}{R_1} + \frac{1}{R_2}\right) & \frac{1}{R_2 C_1} \\ \frac{1}{C_2 R_2} & -\frac{1}{C_2 R_2} \end{bmatrix} \begin{bmatrix} T_1 \\ T_2 \end{bmatrix} + \begin{bmatrix} \frac{1}{C_1 R_1} & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} T_\infty \\ \dot{Q}_{Loads} \end{bmatrix} \tag{2.8}$$

Where:

Table 2. **2R2C Variable Definitions**

Variable	***Description***	***Units***
T_∞	The outside ambient temperature	[°C]
T_1	The wall temperature	[°C]
T_{zone}	The zone temperature	[°C]
R_1	Effective thermal resistance between T_∞ and T_1	[°C/W]
R_2	Effective thermal resistance between T_1 and the zone temperature	[°C/W]
C_1	Effective thermal capacitance of the wall	[J/°C]
C_2	Effective thermal capacitance of the zone	[J/°C]
$\dot{Q}_{gains}$	The heat load of the system (solar, internal, infiltration, HVAC)	[W]

The system inputs are the ambient temperature, solar and internal heat gains, which are all applied at zonal node. A simplified diagram illustrating the locations of the Rs and Cs is shown in Figure 3.

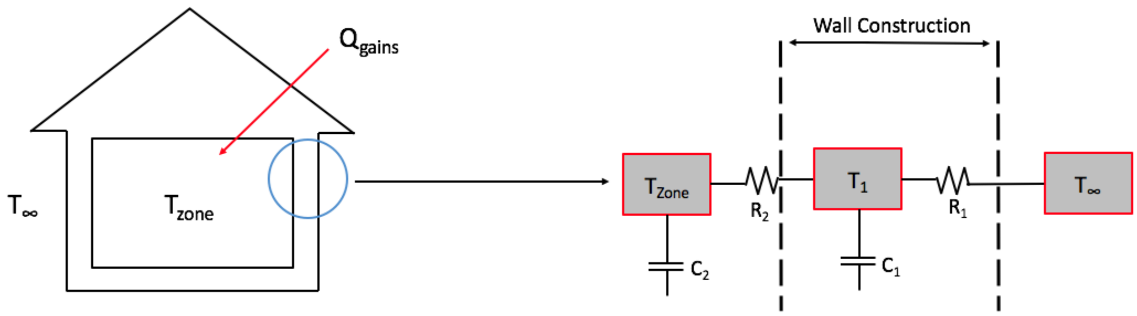

Figure 3. 2R2C Thermal Parameter Diagram

C_2 is the thermal capacitance of the zone, which includes the air and all the interior mass, i.e. furniture, carpet, etc. *C_1* is the effective thermal capacitance of the building constructions, the location of *C_1* is arbitrary and the only known information about its location is that it falls somewhere in between the building's wall construction. This is due to the wall partitioning happening at the nodes where the temperature is uniform throughout, and not in any symmetrical manner. *R_1* is the effective thermal resistance in between the ambient temperature and *T_1*, and *R_2* is the effective thermal resistance in between T_1 and the center of the zone. This model structure was utilized in a simplified case study to investigate the effects of the thermal parameters before modeling the COBE Building. The model used with the COBE building is more intricate than the 2R2C model shown above due to the complexity of the building's dynamics.

Prior Research in Reduced Order Building Modeling

There has been extensive research using ROMs to describe the thermal response of buildings, not for virtually commissioning, but for other areas such as model predictive control (MPC), day ahead scheduling and cost minimization. Buildings are currently controlled reactively, meaning they adjust continuously for weather conditions and demands [13]. But that is expected to change with MPC and day ahead predictions gaining traction due to integration of renewable energy and thermal energy storage.

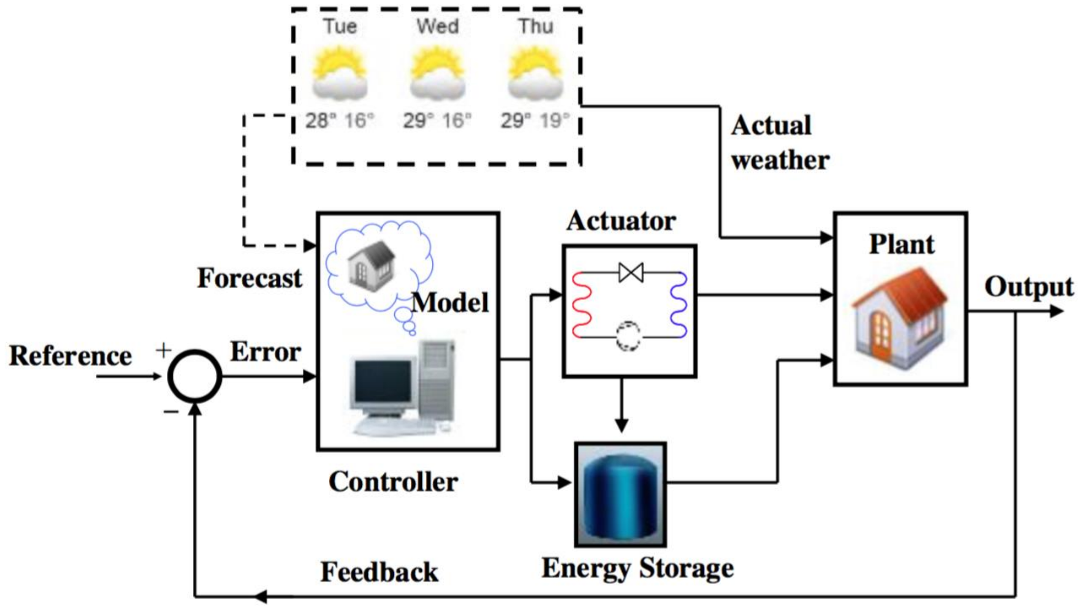

Figure 4. Diagram Illustrating Model Predictive Control Information Flow [13]

MPC allows the controller to make decisions based on weather forecasts and other critical parameters to determine the most cost-effective method of controlling integrated building systems such as energy storage. The weather forecast is fed into the simplified model, which calculates the loads of the building based on the system inputs. The controller then determines how to optimally heat or cool the building based in the upcoming weather conditions.

Additionally, MPC has been used to minimize peak demands during summer when the grid is vulnerable to blackouts. During peak periods, the controller aims to minimize energy consumption through altering the cooling set point of the building. The simplified model predicts the building’s internal temperature and evaluates thermal comfort through a discomfort tolerance index which informs the model how uncomfortable the occupants of the buildings are [14]. The data flow used in both MPC instances are the same and can be seen above in Figure 4. Unlike other areas of application (i.e. Aerospace) where the controls must be correct all the time, buildings are more resilient to control faults [13], making virtually commissioning a promising avenue to verify the buildings are operating at their peak performance.

When buildings are not correctly commissioned they may consume excessive amounts of energy and are not likely to operate as designed. This may lead to tenant health or comfort issues, as well as shortened equipment lifespans [15]. Many building owners are hesitant to commission a building's HVAC system if the space is being conditioned adequately and occupant complaints are minimal. The cost of commissioning a building is prohibitive for many owners, but it was shown by the IDL Boise [6], that virtually commissioning is a viable solution to the time and cost barriers of traditional building commissioning. There are still limitations with energy modeling that need to be addressed before the methods of virtual commissioning become utilized, and by using ROMs to simulate the building thermal performance, the cost and time expenditures of commissioning can be minimized even further.

There are several ways to represent the dynamics of a building, the first being the time constant and the thermal delay of the structure, which is related to the effective thermal resistance and capacitance [16] and it can be determined from actual building performance data. The time constant characterizes the rate at which the outdoor temperature influences the interior zone space [17], whereas the thermal delay is the time it takes the mean indoor temperature to change a specific degree under net thermal loads [17]. Antonopoulos et al. [11] found there to be a significant difference between the apparent and the effective thermal capacitance of buildings. Summing the thermal capacitances of all construction materials is a rough approximation because the elements store heat differently when they are distributed throughout, instead of lumped together in one volume. The best way to determine the effective thermal capacitance is to compare numerical solutions to actual building data and fit the parameters. The time constants are composed of the system's thermal parameters and can be determined with a set of linear differential equations that describes the building's thermal behavior, i.e. a ROM.

The simplest way to represent the construction of a building is with a 1R1C model, which only has a single thermal resistance and a single thermal capacitance. This representation of a building is overly simplistic because it 'lumps', or combines, the mass of the exterior wall and interior construction together [18]. This forces the temperatures of these two masses to be equal at all times by not designating them as two separate thermal capacitances. Additionally, most of the thermal capacitance of a building is contained in the wall of the structure, and there is a thermal barrier between the wall construction and the interior zone, which this model ignores. It has been found by Rabl [18] that if you add an additional resistance between the zone temperature and the external temperature it significantly reduces the peak instantaneous loads during warmups which improves the model's overall fidelity.

Bacher et al. [19] researched which models offer the best performance for the least complexity to fully describe the dynamics of the buildings. This process started with determining the simplest model that described all the information embedded in the data, a 1R1C thermal network. In the research, the model's order was incrementally increased to compare how the higher order model statistically compared to the previous version using a likelihood test, which compares the predicted results with the previous model to determine the likelihood that there exists a higher order model that statistically predicts the zone temperature more accurately. This method was applied to an experimental facility in Denmark where all the construction materials were known. It was determined that a 3R3C model described the building's dynamics adequately enough and a higher order model did not add sufficient fidelity to warrant the additional computational expense. The three thermal capacitances were associated with the heater, the interior space, and the construction of the building's envelope.

Gouda et al. [20] found that high order models could be tuned such that a low order model would produce the same results using formulas to calculate the thermal parameters of the wall. Gouda et al. recognized that dynamic models must satisfy two conditions: it must be computationally inexpensive, otherwise order reduction is impractical; further, the model must accurately describe the system. Through parameter tuning this research showed the best model structure is the compromise between simple and accurate. This ideology has almost entirely been lost due to the exponential growth in computational power and the progression in white-box building modeling software.

Models of different order can be coupled together to describe the dynamics of different building components, which is a method used to maintain a low model order and still capture all the response frequencies. In the early day of development this was not attainable due computational limitations, but with high performance simulation software, this is a viable option for achieving better model fidelity. Wang and Xu [15] used a 3R2C model for the roof and the external walls of the buildings. Additionally, they concluded that each wall needed to be modeled individually due to each having separate forcing functions associated with the changing position of the sun. A 2R2C model was used to describe dynamics associated with the internal air and mass. Similarly, Amara et al. [21] used a 3R4C model to predict the conduction through the walls, disregarding their orientation. The long wave radiation was modeled using a single resistance, and the hot water heat exchanger, in the HVAC system, was modeled using a 2R1C network. The results were compared with a reference solution using frequency analysis (Bode Plots), and both above studies relied heavily on computational power to optimize all the thermal parameters. Another method to increase ROM accuracy is using a separate set of parameters for different operational conditions. Fazenda et al. [22] modeled the building using separate contexts throughout the entire year. It was found that better results could be found if the year was discretized into sections based upon the different conditions that influence the zone temperature, with those being the occupancy and activity level, state of electrical equipment, the temperature of the adjacent zones, and outdoor environmental conditions. Instead of using a single model to describe the building throughout the year, the building would be modeled as a set of context-based scenarios with their own independent thermal parameters. Extensive research has been done in determining the most accurate model structure to predict the indoor air temperature, but there has been little research completed into the most suitable application for whole building ROMs.

CHAPTER THREE: METHODOLOGY

ASHRAE BESTEST Case Study

With energy simulation growing in popularity, ASHRAE has designated predefined models used to verify the accuracy of energy simulation software, otherwise known as BESTEST case studies. Not only do these models verify the accuracy of the software, they can also diagnose where the inconsistencies originate from being either algorithmic, modeling limitations, user input or coding errors [23]. ASHRAE has designed multiple scenarios to test different energy modeling applications, the case that was used to verify the second order ROM was BESTEST Case 900.

BESTEST Case 900 is a simple single zone structure with two windows, both (2.0m by 3.0m) south facing. The building is constructed out of heavy materials equating to a large effective thermal capacitance, meaning the building can store a large amount of thermal energy and has a longer time constant. Figure 5 is an illustration of the building's geometric properties as well as its orientation.

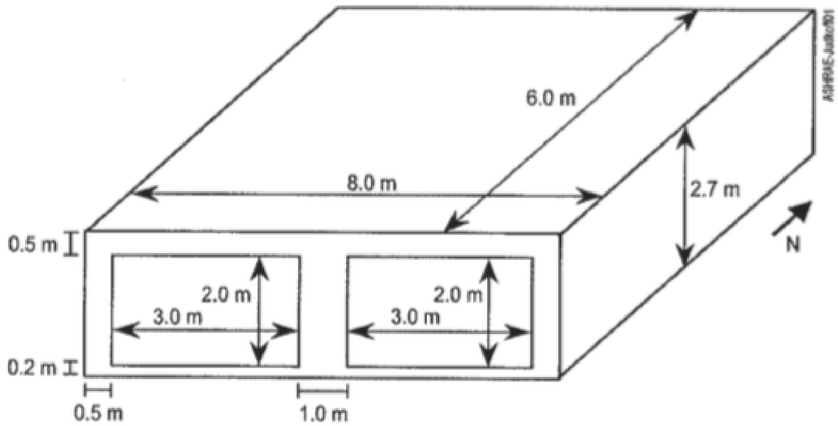

Figure 5. BESTEST Case 900 Geometry and Orientation [23]

The single zone building is constructed out of typical building materials, a complete summary of all the material properties can be seen below in Table 3. All the other thermal and physical characteristics of the building have been defined by ASHRAE and can be found in ASHRAE Standard 140-2007 [23].

Table 3. Summary of Case 900 Material Properties [23]

Component	Area [m^2]	UA [W/K]
Wall	63.60	32.58
Floor	48.00	1.892
Roof	48.00	15.25
South Window	12.00	36.00
Infiltration		18.44[a]
Total UA (with South Glass)		104.17
Total UA (without South Glass)		68.17
	ACH	**Volume [m^3]**
	0.500	129.60

[a]UA corresponds to infiltration gains calculated using ACH x volume x (specific heat of air) x (density of air)

This case study was chosen due to its similarities to the COBE Building at the University of Idaho. The high thermal mass is typical of large mixed-use buildings meaning the time constants will be the same order of magnitude between the BESTEST and the COBE building. Unlike the COBE Building, the BESTEST case study has a constant internal gain from equipment, i.e. plug load, of 200W and no occupancy.

Structure of the Case Study Reduced Order Model

The first step of modeling the BESTEST building was determining what ROM structure would best describe the building. The building could be modeled using a high order thermal circuit, but that does little to simplify the energy modeling process. ROMs are typically excited through either an impulse, ramp, or step input [9], and the response is analyzed to estimate the system's parameters, which was used to determine the order of the system. EnergyPlus was used to simulate a step input for the BESTEST case, which was accomplished by modifying the model's weather file and HVAC controls. A step input is when an input changes from one value to another value instantaneously [9], and this was implemented with small changes to the input file of the BESTEST model. These alterations were adjusting the weather file to have a constant outdoor temperature of 25°C and changing the building's cooling set point to 0°C. Additionally, all other ambient conditions in the weather file were adjusted such that they were constant over the testing periods. This was done to ensure no other environmental conditions excited the system, thus eliminating all potential sources for external noise. The zone temperature was held at the cooling set point long enough to guarantee the building was at a uniform temperature throughout and the transient response of the system had dissipated, once this was realized the HVAC was completely shut off mimicking a step input. This modeling approach is very useful for characterizing systems, but due to system limitations is not realistic to use with actual buildings [5]. The response of the BESTEST EnergyPlus step input can be seen in Figure 6.

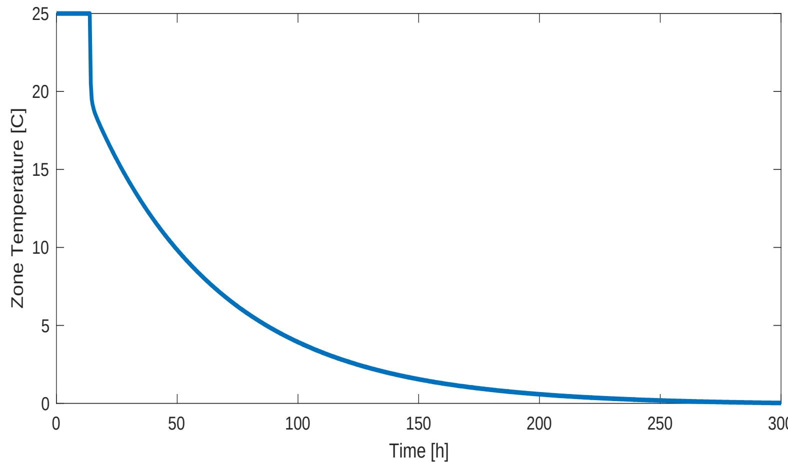

Figure 6. Energy Plus Step Input Response

The response exhibits two distinct regions, one with a rapid temperature decay of about 5°C, and the other which is much slower. Research by Antonopoulos and Koronaki [17] have shown that the thermal delay of the indoor air temperature is linearly related to indoor thermal capacitance of the space. Antonopoulos and Koronaki defined the thermal delay as “the time…needed for the mean temperature of the indoor air, partitions, and contents to increase over the mean value of the outdoor temperature oscillations by a specific amount under specified building heating” [17]. The BESTEST case study does not include any interior mass or partitions, the only matter occupying the zone is the indoor air, which has a low thermal capacitance. This low thermal capacitance initially dissipates the stored energy rapidly, but as the temperature differential between the zone air and the wall temperature approaches zero, the only pathway for heat transfer is through the wall construction, which substantially reduces the heat flux rate. The behavior in Figure 6 agrees with the prior research, and due to the response a second order thermal network was chosen as the structure for the BESTEST case study ROM.

The BESTEST ROM is composed of two thermal resistances and two thermal capacitances, with the model’s inputs being the ambient outdoor temperature, solar and internal heat gains. A set of differential equations were developed using Kirchoff's node law and formulated into a Simulink block diagram for analysis. The differential equations that describe the system can be seen in Equation 3.1 and 3.2, a full list of the variables and their definitions can be seen in Table 2.

$$C_1 \frac{dT_1}{dt} = \frac{T_\infty - T_1}{R_1} + \frac{T_2 - T_1}{R_2} \tag{3.1}$$

$$C_2 \frac{dT_2}{dt} = \frac{T_1 - T_2}{R_2} + \dot{Q}_{Loads} \tag{3.2}$$

The system's response is a function of the ambient temperature and the internal heat gains which possess their own independent set of dynamics. To gain a full understanding of the system, all the internal heat gains were initially set to zero to study the response when only excited by the ambient temperature. The model parameters were determined without the excitation of the internal and solar loads, but they were reintroduced into the model to compare their effects in later iterations. The system parameters were estimated using two different approaches, one using an assumed solution to the second order differential equation (ODE) and optimized with a numerical solver using least-squares optimization approach, and the other through a parameter optimization package in MATLAB. Both methods used the response generated in energy plus as the baseline data. The optimization package in MATLAB creates a constrained optimization problem and solves using standard optimization techniques. Whereas, the numerical solver approach aims to minimize the sum of the square error between the baseline data and the model response, as predicted by set of differential equations.

The assumed solution for the 2nd order ODE takes the form:

$$T_2(t) = A_1 e^{-t/\tau_1} + A_2 e^{-t/\tau_2} \tag{3.3}$$

Where T_2 is the zone temperature, and A_1 and A_2 are constants determined though initial conditions $T_2(0)$ and $\dot{T}_2(0)$, and τ_1 and τ_2 are the two individual time constants of the second order system. The above equation is represented in terms of the two time constants, which need decomposed into the *Rs* and *Cs* to have a direct comparison to the MATLAB optimization parameter estimation. The decomposed time constants of the assumed solution are shown below in Equation 3.4. The full derivation of this equation can be seen in [24], the $\pm\tau$ corresponds to τ_1 and τ_2 in Table 4 below.

$$\pm\tau = \frac{1}{2}(C_2R_2 + C_2R_1 + C_1R_1)\left[1 \pm \sqrt{1 - \frac{4C_2R_2C_1R_1}{(C_2R_2 + C_2R_1 + C_1R_1)^2}}\right] \tag{3.4}$$

The results for both parameter optimization methods are shown in Table 4.

Table 4. MATLAB and Numerical Solver Solutions for Thermal Parameters of a 2R2C Reduced Order Model

	R_1	R_2	C_1	C_2	τ_1 [hrs.]	τ_2 [hrs.]
MATLAB	0.0375	0.3298	5.123E6	2.024E3	53.35	0.19
Numerical Solver	0.0493	0.2931	3.944E6	2.933E3	54.11	0.24
% Difference	**23.0%**	**12.5%**	**29.9%**	**31.0%**	**1.41%**	**20.28%**

The numerical solver approximation used two equations to approximate four parameter values making the method underconstrained and allowing for multiple solutions, which explains the difference between the two methods. The absolute value of the percent difference between the two parameter values was calculated to compare the two parameter estimation methods. Even though a large percent difference exists between the individual thermal parameter results, the difference between the time constants τ_1 and τ_2 is within a range of acceptability. The fast time constant (τ_2) is associated with the internal air temperature and the quick frequencies in the system response. Even though there is an approximate 20% difference between the numerical solver and MATLAB optimization parameter values, the actual difference is only 0.05 hrs., or three minutes, which should not affect the model's overall results.

The thermal parameters were computed assuming the system had no other external forcing function other than the outdoor air temperature, which is an over simplification of the actual building. When there is only one system input the model shows good fidelity when calculating the annual floating zone temperature of the space, i.e. the unconditioned zone temperature. When other ambient conditions were added back into the model such as wind speed and direction, all solar parameters, humidity, sky cloud coverage, etc., the model failed to accurately predict the zone temperature when compared to the EnergyPlus model. This can be explained due to the lack of higher frequency excitations when optimizing the thermal parameters [25]. These higher frequencies are associated with the fast-changing ambient conditions and the quick response of HVAC equipment that severely change the thermal parameters, which were omitted when only the ambient temperature was included as the system inputs.

Solving for the parameters using both methods allowed for a comparison between the known solution and the parameter optimization in Simulink. Determining the parameters using both, software optimization and an assumed solution, for higher order models is computationally extensive and time consuming. The COBE building thermal model is not a simple 2R2C model, making software optimization the only viable method to obtain the parameters, which we illustrated is nearly as accurate as the explicit solution.

CHAPTER FOUR: UNIVERSITY OF IDAHO'S COLLEGE OF BUSINESS AND ECONOMIC BUILDING STUDY

Prior Research

In 2014, the Integrated Design Lab (IDL) of Boise conducted a study to research virtual commissioning as part of a grant from Avista Power Company [6]. This research developed a process of virtually commissioning the COBE building using an EnergyPlus model and a duplicate controller which of the HVAC controller used at facility. This was accomplished by enabling communication from the EnergyPlus simulation to the building controller using the Building Controls Virtual Test Bed (BCTVB). BCVTB is a "middle-ware" which translates the outputs from an EnergyPlus simulation to inputs that building controllers can understand [7]. The variables chosen for the study included outdoor air temperature, outdoor air damper position, mixed air temperature, and return air temperature. The COBE's Alerton controller required inputs from other equipment and feedbacks from all thermal zones, which was not practical to model in EnergyPlus due to computational limitations. These inputs were bypassed by adjusting the logic to allow the controller to operate without each individual feedback.

COBE Building Information and HVAC Equipment

The College of Business and Economics building is a 50,000-sq. ft. mixed use building constructed in 2001. This building is composed of a mixture of office spaces, class rooms, and a unique trading simulation lab used for real time market trading and analysis. The building is conditioned with two variable air volume (VAV) air handler units (AHUs), one servicing only the basement, and the other servicing the top three floors. The air handlers rely on a district heating and cooling system to provide the chilled and hot water the AHUs use for conditioning the spaces. Non-fan powered re-heat terminal units are located in each zone of the building. Figure 7 shows the actual building and the Energy Plus model's geometry.

Figure 7. COBE Building and Energy Plus Geometry

The building employs thermostat setbacks when the building is unoccupied which helps to reduce overall energy consumption. The setbacks were enabled in 2014-2015 when the original IDL's research was conducted, but it was later determined that the controller logic had been altered removing the setbacks sometime after the study. Because the ROM is going to be compared against the previous EnergyPlus model these setbacks were included in the ROM. Table 5 has both the occupied and unoccupied thermostat set points for heating and cooling.

Table 5. Heating and Cooling Schedule

Occupancy Status	*Heating Set Point [℃]*	*Cooling Set Point [℃]*
Occupied	21.0	24.0
Unoccupied	15.6	26.7

The space occupancy varies throughout the day due to the nature of the building, which impacts the overall internal loads of the zones. ASHRAE standards were used in determining all internal loads (i.e. occupant heat gains, plug loads, etc.) and all infiltration, fenestration schedules for each zone. Figure 8 shows the internal, fenestration, and infiltration gains during a normal week that class is in session. The internal loads change during the summer months due to fewer occupants but follow the same trend as the ones illustrated below.

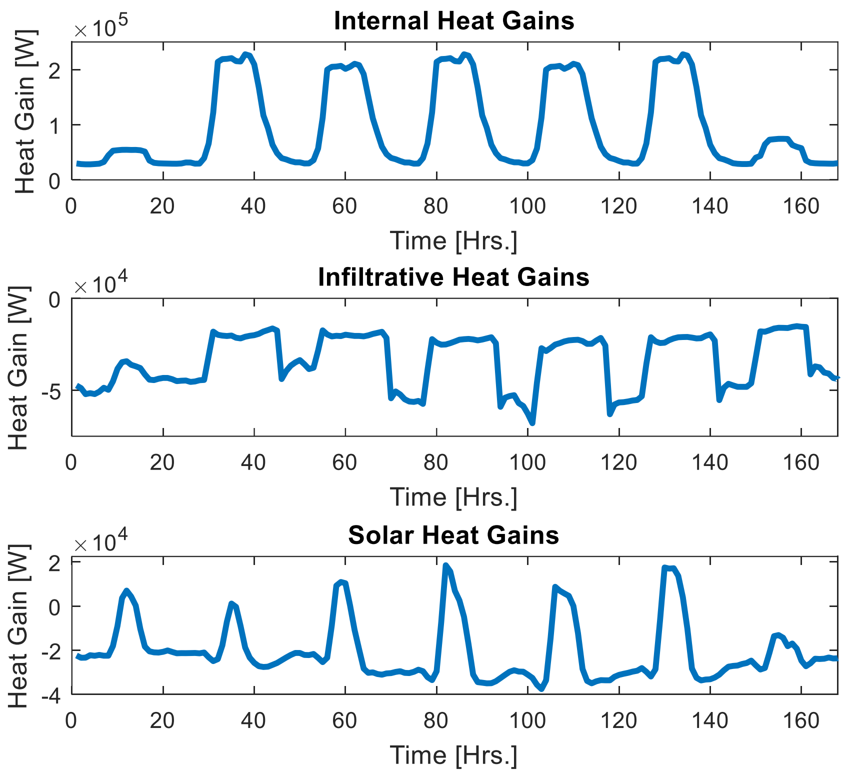

Figure 8. Weekly Schedule of COBE Heat Gains

The model was calibrated according to ASHRAE's standards utilizing actual meteorological data from the Pullman/Moscow Airport and actual building energy consumption data collected form the COBE's building controller. The COBE Building was chosen for the previous research because the building's controllers communicate through a standard building automation and control network protocol: BACnet. This communication protocol was essential for the research so that the energy model could interact with the controllers in a standard way. The team continued using this building for the ROM virtual commissioning research so that the new method could be compared against the calibrated baseline EnergyPlus model from the previous research.

Integrated Design Lab Results

Once the Energy Plus model had been calibrated, the IDL focused on the outside air damper position, which controls how much outside air is vented into the building. There are several set control points that determine how the damper is modulated, with the main one being the economizer lock out temperature. When the ambient temperature is below the economizer lock out temperature, there is no restriction on the amount of outside air allowed into the mixing chamber. When the ambient temperature is above the lock out temperature, the damper restricts the flow such that only the minimum required ventilation air is vented into the AHU. This system has a large impact on the total energy consumption of the building which is why this parameter was chosen for the study. The current operational set point at the COBE building was compared to the air percent calculated by EnergyPlus, and it was discovered that the COBE was not allowing the economizer to capture the maximum amount of free cooling, thus consuming more energy than need be. The recommendation was modeled in EnergyPlus and it was found that by adjusting the economizer lock out temperature the building would consume 7% less energy annually. By comparing the optimum set points to the current building operations, recommendations were made that could reduce the total energy consumption of the COBE building. Through virtual commissioning, actual realized energy savings could be achieved, but EnergyPlus still has a learning curve limiting its usability. A simpler energy modeling approach would increase virtual commission's potential to reduced overall commercial building energy consumption.

COBE Reduced Order Thermal Model

As discussed earlier, the COBE building is composed of over 50 thermal zones which is typical for commercial buildings of this size. However, for the purpose of this research it was hypothesized that an adequate level of fidelity can be captured by considering the COBE building as a single thermal space. This simplification was necessary, otherwise each zone would need to be modeled with a separate thermal circuit, making the number of parameters to be identified in the thousands. Additionally, building zones are designed based on space usage and there is no reason to believe that thermal capacitances should align with the zone design. Modeling the COBE building as a single zone does not decrease the overall accuracy of the ROM, in fact it has been shown that most multi-zone buildings can be reduced to a single zone if the entire building is approximately at the same temperature [18]. The thermostat set points may vary a degree or two between each space, but this temperature difference is not enough to drive enough interzonal heat exchange to rationalize modeling each zone.

Additional assumptions were made to model the COBE Building using a thermal circuit, and they are as follows: (1) The air inside of the zone is well mixed and at a homogenous temperature throughout. (2) Heat transfer is one-dimensional, meaning the heat passing though the outer walls of the COBE building are doing so such that the direction of heat transfer is only perpendicular to the outer wall surface. (3) The wall temperature is homogenous throughout meaning there is no temperature gradient in the wall making the heat flux a constant value for the entire surface.

The COBE Building was modeled as a 3R3C model describing the internal building dynamics, coupled with a 2R1C model describing the dynamics of the ground and foundation of the structure. The 2R1C model was added into to compensate for heat transfer through the foundation that the 3R3C model was unable to accurately capture. A diagram of the model and descriptions of the parameters can be seen below in Figure 9.

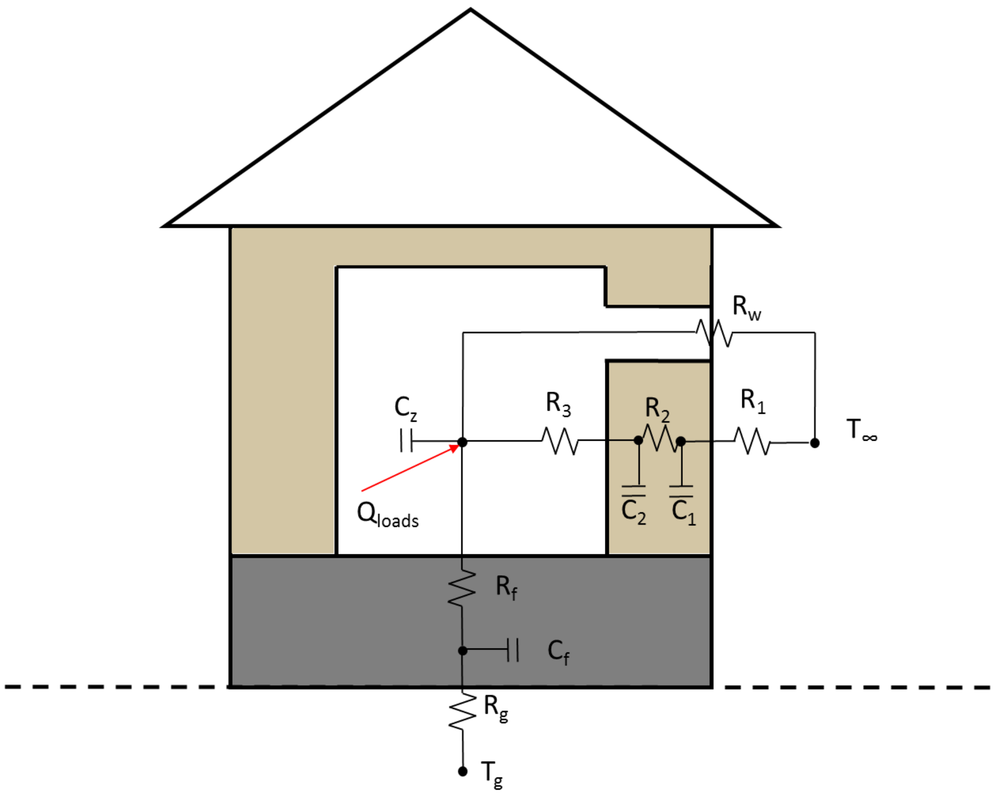

Figure 9. Diagram of Thermal Network used in Modeling COBE Building

Where:

Variable	Description	Units
T_∞	The outside ambient temperature	[°C]
T_g	The temperature of the ground	[°C]
R_1	The thermal resistance between the outside and wall	[°C/W]
R_2	The thermal resistance of the wall	[°C/W]
R_3	The thermal resistance of the wall and the between the zone	[°C/W]
R_w	The thermal resistance of the windows	[°C/W]
R_f	The thermal resistance between the center of the foundation and the central interior zone	[°C/W]
R_g	The thermal resistance of the ground and the midsection of the building's foundation	[J/°C]
C_1	The thermal capacitance of the wall	[J/°C]
C_2	The thermal capacitance of the wall	[J/°C]
C_z	The thermal capacitance of the zone	[J/°C]
C_f	The thermal capacitance of the foundation	[J/°C]
$\dot{Q}_{Loads}$	The heating and cooling loads of the system (solar, internal, infiltration, HVAC)	[W]

This thermal circuit can be expressed using a set of linear differential equations to describe the states of the structure. The selected states are the temperature of the foundation (T_f), the internal zone temperature (T_z), and the temperatures (T_1) and (T_2) which fall in-between the exterior façade and the interior wall. The state space representation of can be seen bellow in its matrix formulation. It should be noted that the states are the effective temperatures of the modeled capacitances and they represent the overall average temperature for each lump.

$$\begin{bmatrix} \dot{T}_1 \\ \dot{T}_2 \\ \dot{T}_z \\ \dot{T}_f \end{bmatrix} = \begin{bmatrix} -\left(\frac{R_1+R_2}{R_1R_2C_1}\right) & \frac{1}{R_2C_1} & 0 & 0 \\ \frac{1}{R_2C_2} & -\left(\frac{R_2+R_3}{R_2R_3C_2}\right) & \frac{1}{R_3C_2} & 0 \\ 0 & \frac{1}{R_3C_z} & -\left(\frac{R_3R_f+R_wR_f+R_wR_3}{R_3R_wR_fC_z}\right) & \frac{1}{R_fC_z} \\ 0 & 0 & \frac{1}{R_fC_f} & -\left(\frac{R_f+R_g}{R_fR_gC_f}\right) \end{bmatrix} \cdot \begin{bmatrix} T_1 \\ T_2 \\ T_z \\ T_f \end{bmatrix} + \begin{bmatrix} \frac{1}{C_1R_1} & 0 & 0 \\ 0 & 0 & 0 \\ \frac{1}{C_zR_w} & 1 & 0 \\ 0 & 0 & \frac{1}{C_fR_g} \end{bmatrix} \begin{bmatrix} T_\infty \\ \dot{Q}_{Loads} \\ T_g \end{bmatrix} \tag{3.3}$$

The loads included in the model were the internal loads, and the solar gains which were outputted from EnergyPlus and used as an input into the ROM. These values can be determined computationally using ASHRAE standards but to avoid any additional errors, EnergyPlus outputs were utilized. All loads were applied at the center of the zone which is an oversimplification of the system. It is known that the solar loads will be distributed throughout the interior of the structure, and the distribution pattern is determined by the geometry, reflectance, and many other parameters of the building. Additionally, the conditioned air will be distributed throughout the entire space and not just supplied to the center of the zone, but without knowing the exact distribution pattern, the loads have to be applied at the center. Additionally, the solar gains do not only come from the radiation directly admitted into the space via the window, they also come from the thermal storage properties of the exterior wall construction. As the day progresses, the building materials store energy and their internal temperature increases, thus increasing conductive heat transfer throughout the entire surface of the exterior wall. These gains are happening simultaneously over the entire exterior wall, not just at the center of the zone. The Simulink block diagram of the above system of equations representing the thermal model can be seen in Appendix A.

Parameter Estimation

The model parameters were estimated using a Simulink® Optimization package that iterates through different parameter values until the model best predicts values when compared to a user inputted time series [26], with the time series being the zone temperature as predicted by EnergyPlus. The estimated parameter values are shown in Table 6.

Table 6. Optimized Model Parameter Values

Thermal Resistance		*Thermal Capacitance*	
R_1	6.617 E2	C_1	7.029 E8
R_2	1.272 E-1	C_2	2.583 E12
R_3	3.021 E-4	C_z	4.520 E7
R_w	3.768 E-4	C_f	1.296 E9
R_f	2.909 E-5		
R_g	2.968 E-4		

The model parameters shown above best predicted the ROM's zone temperature when compared to the results of EnergyPlus. The zone temperatures from EnergyPlus have been plotted against those from the ROM in Figure 10. The top and bottom figures show the hourly zone temperature for the first week of February and the first week of August respectively.

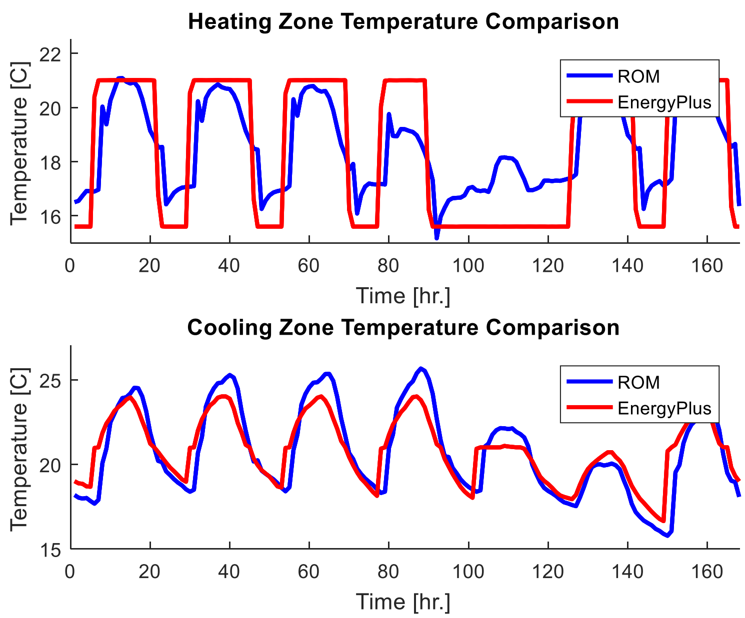

Figure 10. Energy Plus vs. ROM Zone Temperature for first week of February (Top) and the first week of August (Bottom).

During February, the zone temperature settles at its set point whereas the lumped RC model does not accurately predict this behavior. This response is believed to be an indication that the building is a higher order system during the heating season than originally modeled. The ROM has better fidelity during the cooling season, which is illustrated in the August zone temperature figure, meaning the order of the model is dependent on whether the building is operating in heating or cooling mode.

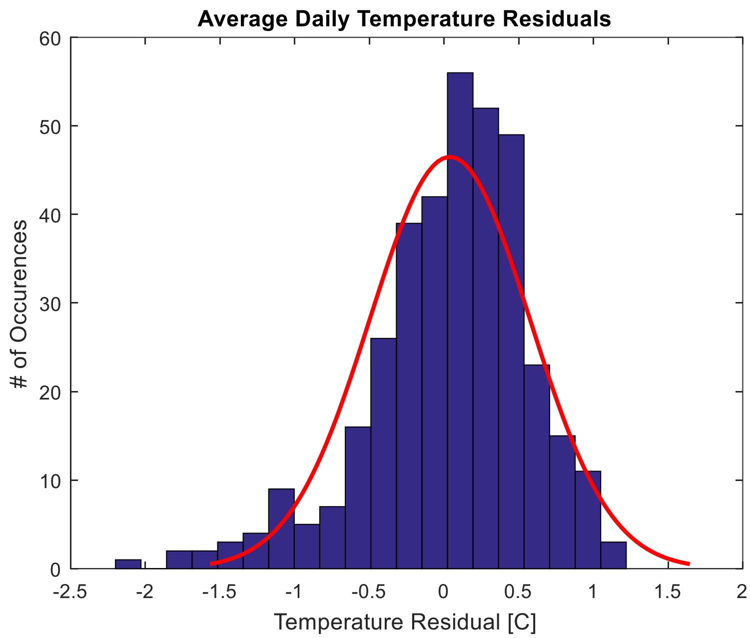

Figure 11. Histogram of Average Daily Residuals of ROMs and EnergyPlus' Predicted Zone Temperature

The daily average temperature difference between EnergyPlus and the ROM have been plotted in a histogram, seen above in Figure 11, centered around an average of 0.039°C. This is an indication that the ROM over-predicts the zone temperature by an average of 0.039°C. This should translate into a higher magnitude of cooling need to compensate for the over-prediction of the zone temperature when compared to EnergyPlus. While the reduced order thermal model predicts zone temperature, the indoor zone temperature is not a direct indication of energy consumption. In order to 'convert' these temperature predictions to energy, an HVAC and controller model are need. To use this model as a tool of virtual commissioning, the HVAC model needs to describe the mechanical systems as accurately and simply as possible. A diagram illustrating the flow of the fully integrated model can be seen below in Figure 12.

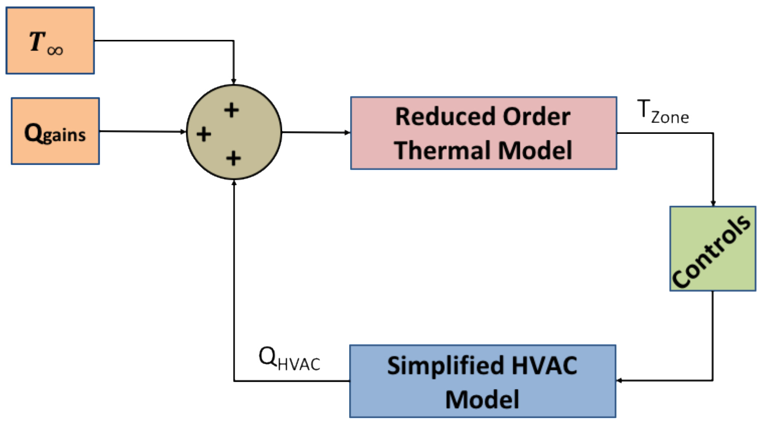

Figure 12. Integrated Reduced Order Thermal Model Flow Diagram

The thermal model will predict the zone temperature of the buildings, which will be passed through to a controller to inform the simplified HVAC model how to condition the space. The HVAC model will predict the magnitude and duration of zone conditioning which will be relayed back to the thermal model and applied at the center of the zone like the other zone loads, which completes the feedback loop.

COBE HVAC Model

As discussed earlier, the COBE Building is a mixed use educational facility. It has over fifty zones having varying occupancy, internal loads and thermostat set points. The HVAC equipment used at the COBE building relies on a district heating and cooling VAV system with non-fan powered terminal reheat. A district chiller and boiler provide each building with chilled and hot water, which is utilized as the working fluid in the main AHUs. The COBE building has two AHUs, one that services only the basement, and the other that services the three above ground floors. For simplicity, the HVAC model was altered to only have one AHU to service the entire building. Figure 13 shows a diagram of a typical air handler unit.

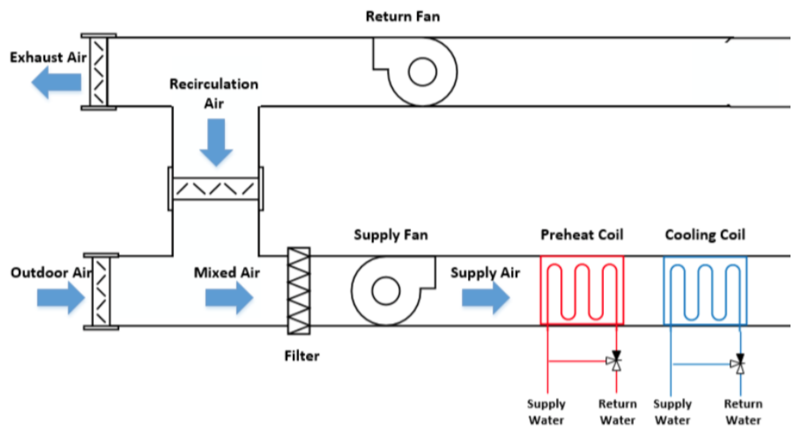

Figure 13. AHU Diagram [27]

There are four main parts of a typical HVAC system similar to the one used at the COBE building: the supply and return air fans, air dampers, heating and cooling coils, and the terminal reheat box (not illustrated above). The supply air fan provides the necessary flow to meet the minimum outdoor ventilation air standard and to condition the zone to the required thermostat set point. The supply air fan is connected to a variable frequency drive (VFD) that controls the fan speed, which is the most efficient way of controlling the air flow. The VFD allows the main air handler to modulate the air flow when the zones are being conditioned to the appropriate set points, decreasing the power consumption of the unit, making them far more efficient than their constant speed counterparts. The next element of the HVAC system are the dampers, which are used to vary the amount of outside air and return air vented into the mixing chamber. The building controller sets the damper position depending on the ventilation demand and the outside air temperature modulating the flows of the two streams entering the mixing chamber. After the air passes through the mixing chamber it is conditioned to the supply air temperature set point, which oftentimes is the same as the mixed air set point. The AHU heating and cooling coils are only used when the dampers cannot meet the mixed air set point. The last component of these systems are the terminal units which are located at the individual zone and are used to reheat the air before it enters the space. Each box has a separate hot water heating coil supplied from the central system. Each one of the above-mentioned components needs to be modelled individually to have an accurate representation of the entire system.

Economizer Controls

The HVAC system used in the COBE building relies on an economizer to capture free cooling during time when the ambient conditions permit. Economizers are mechanical dampers that modulate their blade position to control the amount of outside air vented into a building. Economizers typically have four different operational modes: heating, modulating, integrating, and mechanical cooling mode [28]. When the outdoor temperature is less than 1°C (heating mode) the economizer only allows the minimum air required for ventilation. The outdoor air is mixed in with the return air in the mixing chamber and then heated to the necessary temperature to meet the demand of the space. During mild outdoor temperature (1°C to 13°C, i.e., modulated economizer mode) the full cooling demand of the building can be met by modulating the fraction of outdoor air mixed with the return air. This operational mode allows the economizer to provide the most amount of free cooling to the building. The next operation mode is integrated economizer mode which occurs when the outdoor temperature is too high for the full load to be met by outdoor air (13°C to 24°C), during this temperature band some mechanical cooling must take place to meet the cooling demand of the building. The last operation mode occurs when the outdoor air temperature is above the economizer's high limit shut off. During this mode, the economizer only allows the minimum required outdoor air to meet ventilation requirements and the space conditioning is accomplished through mechanical cooling. Figure 14 below offers an illustration of all economizer operational modes throughout a typical year.

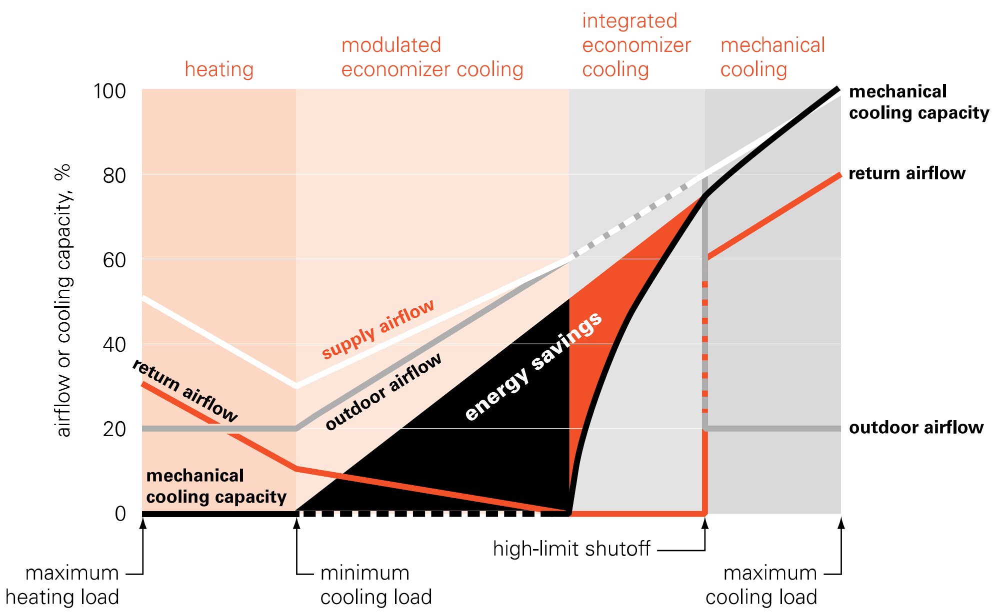

Figure 14. Typical Economizer Operation [28]

The above operational modes are the basis for the economizer controls utilized in the HVAC model. The outside air damper position was determined through applying a heat balance to the AHU mixing chamber with the control volume encompassing all air streams flowing into and out of the mixing chamber. Below is a simplified diagram of the mixing chamber showing the individual energy streams included in the analysis.

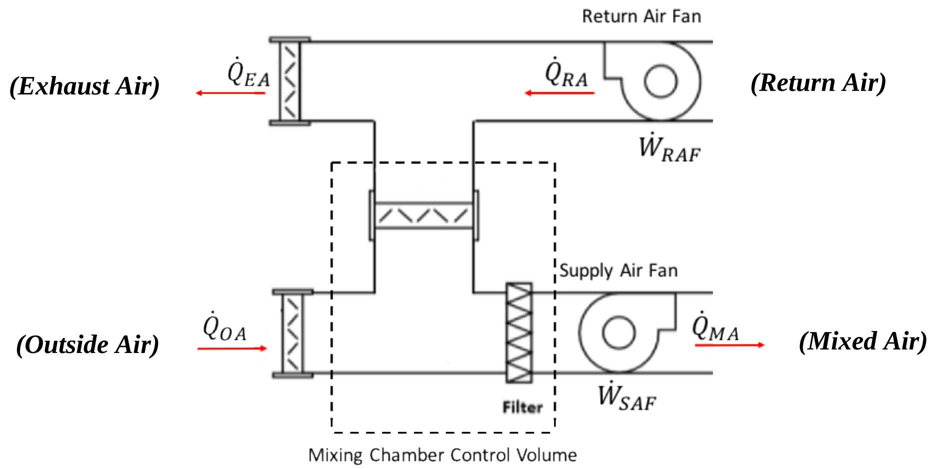

Figure 15. Control Volume used to Derive Economizer Control Scheme

The amount of energy contained in the supply air stream is a combination of the return air and the outside air streams. The volume of air exhausted was assumed to be equivalent to the volume of outside air vented in, a necessary assumption to avoid adverse building pressurization issues. It was also assumed that the damper position and the percent air flow through the damper have a linear relationship, which is not typically the case. However, building controllers are programed with the damper position vs. flow curve, making the system operate in a linear fashion, thus justifying the assumption. The following equations were used to determine the outside air damper position. RA denotes the return air steam, OA denotes the outdoor air stream, and MA denotes the mixed air stream for following equations.

$$\dot{Q}_{RA} + \dot{Q}_{OA} = \dot{Q}_{MA} \tag{4.1}$$

Equation 4.1 shows the heat rate supplied contained in the mass flow rate of air is the sum of heat rate contained in the fraction of non-vented return air and the outside air contained in the flows of each stream. Only the sensible heat will be accounted for in this analysis for simplicity, this assumption is relatively accurate due to the arid climate of

Moscow, ID, which would not be the case in humid locations. The amount of sensible heat in each of the mixing air streams is determined through Equation 4.2.

$$\rho_{RA} C_p \dot{V}_{RA} T_{RA} + \rho_{OA} C_p \dot{V}_{OA} T_{OA} = \rho_{MA} C_p \dot{V}_{MA} T_{MA} \tag{4.2}$$

Where ρ is the density of air [kg/m^3], C_p is the specific heat of air [kJ/kgK], and $\dot{V}$ is the volumetric flow rate of the air stream [m^3/s]. Incompressible flow is assumed; thus the volumetric flow of mixed air must be equal to the total flow through the air handler unit.

$$\dot{V}_{Total} = \dot{V}_{MA} \tag{4.3}$$

A linear relationship between damper position and percent air flow is assumed, making it possible to express the equation with percent air flow fraction, which has a direct correlation to percent open or close damper position.

$$\dot{V}_{RA} = (\%RA)\dot{V}_{Total} = (\%RA)\dot{V}_{MA} \tag{4.4}$$

$$\dot{V}_{OA} = (\%OA)\dot{V}_{Total} = (\%OA)\dot{V}_{MA} \tag{4.5}$$

The supply air must have the same volumetric flow rate as the sum of the outside air and the return air, otherwise the building would experience adverse pressurization issues.

$$\dot{V}_{MA} = \dot{V}_{OA} + \dot{V}_{RA} \tag{4.6}$$

Combining and simplifying Equation 4.6 with Equation 4.4 and 4.5 yields a relationship showing the sum of the return and outdoor air stream must equal unity, which is expected per the law of conservation of mass.

$$\%OA + \%RA = 1 \tag{4.7}$$

The specific heat of air is quasi-constant in the temperature region typical of HVAC operation, simplifying Equation 4.2. In addition, Equation 4.2, 4.4 and 4.5 can be combined yielding:

$$\rho_{RA}(\%RA)\dot{V}_{MA}T_{RA} + \rho_{OA}(\%OA)\dot{V}_{MA}T_{OA} = \rho_{MA}\dot{V}_{MA}T_{MA} \tag{4.8}$$

Which we can further simplify and eliminate the percent return air variable using Equation 4.7:

$$\rho_{RA}(1 - \%OA)T_{RA} + \rho_{OA}(\%OA)T_{OA} = \rho_{MA}T_{MA} \tag{4.9}$$

The density of the air can be found using Equation 4.10, which is accurate for air at temperatures and densities seen in typical HVAC operation [24]. Where *H* is the site elevation [m] and *T* is the temperature of the air stream [°C].

$$\rho = 353\frac{e^{-H/8320}}{T+273} \tag{4.10}$$

Rearranging Equation 4.9 and solving for the percent outside air yields:

$$\%OA = \frac{\rho_{MA}T_{MA} - \rho_{RA}}{(\rho_{OA}T_{OA} - \rho_{RA}T_{RA})} \tag{4.11}$$

The percent outside air fraction is a function of the each of the air streams thermos-physical properties which are a function of the air temperature. There is an 8.4% difference between the density of air at 0°C and 25°C, even though incompressibility was assumed, the propagation of this error can be avoided by explicitly solving for the air density at the various operational temperatures. Additionally, to avoid an iterative solution method, it was assumed that the mixed air density was the average between the return air and the outside air density, allowing the percent outside air to be calculated directly.

The above equations were implemented in Simulink block diagrams starting with the economizer subsystem, seen in Figure 16. The economizer model has four separate subsystems: ECON ON/OFF, Air Properties, ECON Command Control, Mixed Air Temperature, all of which will be discussed in detail below. The 'Air Properties' subsystem uses Equation 4.9 and the site elevation to calculate the density of the supply, return, and outside air streams.

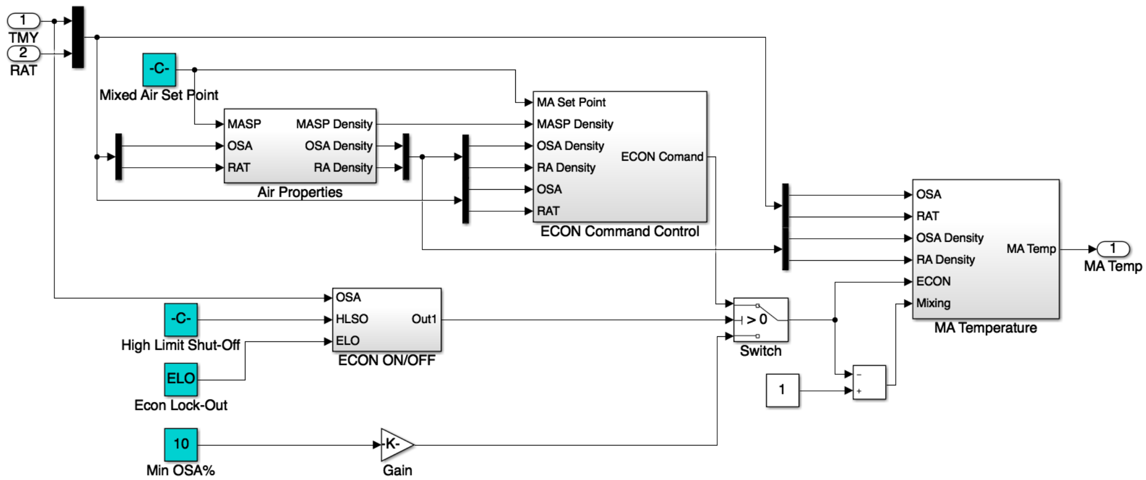

Figure 16. **Economizer Simulink Model**

The subsystem 'ECON ON/OFF' model, shown in Appendix B, uses the ambient temperature, high limit shut off temperature, and the economizer lock out temperature as system inputs. This system compares the current outdoor temperature to the economizer set point temperatures, if the ambient temperature falls in between the high and low set point it turns the economizer on, otherwise an off signal is sent to ECON Command Control subsystem. When the signal indicates the economizer is on, the ECON command control optimizes the damper, using Equation 4.10, to meet the mixed air temperature set point corresponding to the modulated economizer operation mode as discussed above. If the signal is 'off' only the minimum outside ventilation air is allowed into the building. The ECON Command Control subsystem also includes a limiting function which restricts the range of calculated values between zero and one limiting the percent outside air to fall in between zero and one-hundred percent. The block diagram of this subsystem can also be seen in Appendix B. The final subsystem of the economizer model calculates the mixed air temperature. This model calculates the temperature of mixed air depending on the amount of outside and return air vented into the mixing chamber. This model uses the heat balance developed above and feedbacks from the zone air temperature as calculated by the reduced order thermal model. Block diagrams for all the discussed economizer subsystems can be seen in Appendix B. The economizer model is used to optimize free cooling during time when the outside conditions permit. After the economizer, the mixed air is conditioned to meet the supply air set point temperature.

AHU Preheat Coil and Cooling Coil Model

Once the air has been mixed in the mixing chamber, the supply air fan drives the air though the AHU’s heating and cooling coils conditioning the air stream to the supply air set point temperature. When the AHU is operating in modulated economizer mode, the temperature differential between the mixed air set point and the supply air set point is nominal and the energy required to condition the supply air is minimized. During the other modes of operation, the AHU’s heating and cooling coils will have to condition the supply air to the correct set point. The driving force in moving air throughout a building is the pressure differential between the supply air and the zone air. This process is non-adiabatic and the air stream collects residual energy as it passes though the supply air fan, which increases the air temperature. This rise in temperature can be determined through computational methods, but for simplicity and accuracy the temperature rise was taken from the EnergyPlus model and included in the Simulink model.

The AHU heating and cooling model compares the current mixed air temperature to the supply air set point and calculates the amount of heating or cooling needed to condition the air to the correct temperature. Errors associated with simultaneous heating and cooling were avoided by including a dead-band, typical of actual HVAC control systems. The dead band was modeled to be ±1°C, if the differential between the mixed air and the supply air set point temperature is greater than one degree Celsius, the heating and cooling coils are turned on, otherwise the coils are off. The amount of heating or cooling required was determined using first principles and is shown in Equation 4.12.

$$\dot{Q}_{AHU} = \rho_{SA} C_p \dot{V}_{SA} (T_{SASP} - T_{MA}) \tag{4.12}$$

Where T_{SASP} is the supply air set point temperature [°C] and T_{MA} is the current mixed air temperature [°C]. The relay in Figure 17 corresponds to the dead band discussed above. When the absolute value of the difference between the supply air temperature and the supply air temperature set point is more than one, the relay outputs a zero which forces the heating and cooling to go to zero. This correlates to the supply air temperature set point being exclusively met by modulating the outside air damper without the need for additional heating or cooling from the main heating or cooling coil.

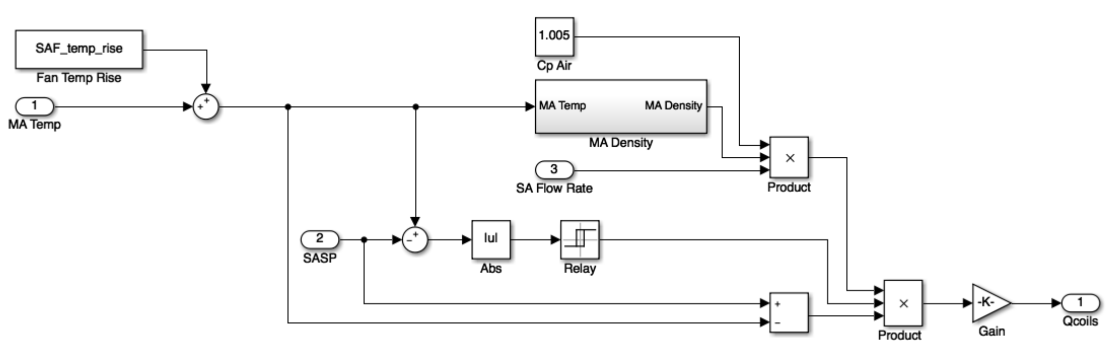

Figure 17. AHU Coil Heating and Cooling Block Diagram

The energy consumption calculated in this model does not directly feedback to the thermal model, but it does contribute to the overall energy consumption of the building. The air will undergo an additional conditioning phase at the terminal reheat box before reaching the space.

Terminal Reheat Model

After the supply air has been conditioned to correct set point temperature it is distributed throughout the building. The COBE building's supply air set point temperature is 12.78°C (55°F), meaning during the winter and shoulder seasons the air is going to need to be reheated before being introduced into the zone. This secondary conditioning is accomplished through the terminal reheat boxes located in each zone. The terminal boxes at the COBE building are known as single-duct VAV non-fan powered terminal boxes with reheat. This model of reheat unit only has a hot water heating coil which is supplied from the same central plant as the main heating coil in the AHU. Along with the heating coil there is a terminal box damper, and a flow sensor. As the zone temperature fluctuates the controller modulates the damper position to vary the amount of air delivered into the zone, with the flow sensor serving as a failsafe to ensure the supply air does not fall below the minimum requirements for ventilation. If the heating load of the building is not met by modulating the flow rate by means of the damper position, the building controller modulates a valve allowing more hot water to flow through the coil, increasing the temperature of the air supplied to the zone [29]. A typical terminal box control loop diagram can be seen below in Figure 18.

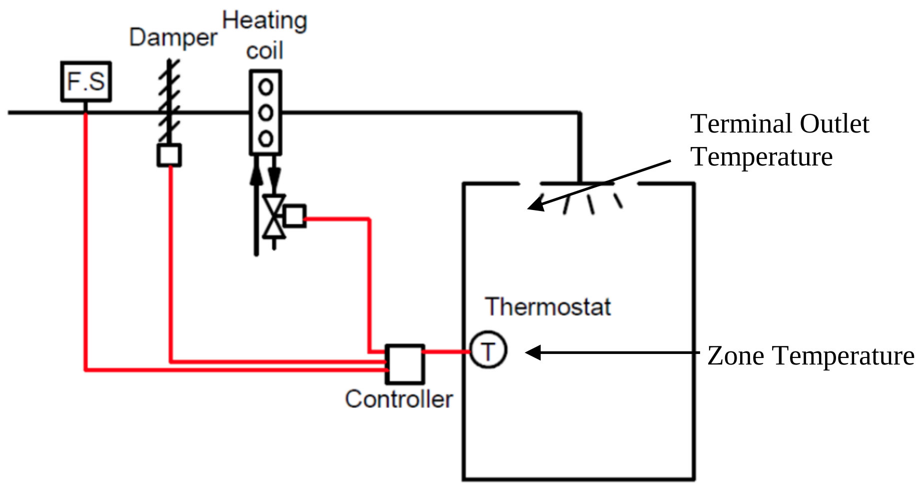

Figure 18. Typical Terminal Box Feedback Diagram [29]

Terminal dampers are an essential component to multi-zone VAV HVAC systems, without them the air flow would be entirely controlled at the main air handler. The COBE ROM was simplified to a single zone building so the terminal damper feedback loop was not modeled directly but its main function was captured and incorporated into another aspect of the model. In multi-zone system, the terminal box dampers control the flow supplied to the zone, and in part controlling the amount of heating or cooling introduced to the zone. Each zone will have different heating or cooling loads and the main AHU is controlled such that the largest load will still be met. In zones that call for less heating or cooling, the terminal dampers are modulated to avoid over-conditioning the space. The optimal way to control the terminal dampers is by allowing the highest flow rate into the room which reduces the system's overall pressure losses. If all the zones are being adequately conditioned and the terminal dampers are set at their minimum position, the main AHU's VFD reduces the overall supply air flow rate. In the single zone COBE model, when the zone is being over-conditioned, a feedback signal is sent to the main air handler to reduce the supply air flow. The modulation does not occur at each zone, instead happening at the main air handler making the terminal damper modulation obsolete.

Unlike the AHU's heating and cooling coils, there are two separate perspectives that had to be accounted for: the amount of energy supplied to the zone, and the amount of energy consumed while conditioning the air. The supply air is entering the terminal reheat box at the supply air set point temperature, which is below the cooling set point making terminal reheat essential even during the cooling season. Even though the terminal reheat unit is only capable of supplying heat to the air stream, all the energy contained in the air needs to be accounted for when coupling the HVAC model with the ROM. The amount of energy being supplied to the zone from the terminal unit can be seen below in Equations 4.13 and 4.14.

$$\dot{Q}_{Supplied\ Zone\ Heat} = \{\rho_{SASP} C_p \dot{V}_{SA}(T_{TO} - T_{zone}), \qquad if\ T_{TO} > T_{zone}\} \tag{4.13}$$

$$\dot{Q}_{Supplied\ Zone\ Cool} = \{\rho_{SASP} C_p \dot{V}_{SA}(T_{TO} - T_{zone}), \qquad if\ T_{TO} < T_{zone}\} \tag{4.14}$$

Where, T_{TO} is the terminal outlet temperature [°C] and T_{zone} is the current zone temperature [°C]. The amount of energy supplied to the zone is proportional to the temperature differential between the terminal outlet temperature and the zone temperature. It should be noted that the cooling energy supplied to the zone is negative which is to keep the HVAC model compatible with the thermal model.

When the terminal outlet temperature is less than the current zone temperature, the zone is being cooled. But due to the operational modes of the terminal reheat boxes, the air may still need reheated even though the room is being cooled. During cooling mode, the terminal outlet temperature will be in modulated between the range of 12.78°C and approximately 18°C. The terminal outlet temperature will be modulated from 18°C up to 33°C during the heating mode. The amount of energy consumed while conditioning the air is then proportional to the temperature differential between the supply air set point and the terminal outlet temperature, which can be seen below in Equation 4.15.

$$\dot{Q}_{Terminal\ Reheat} = \rho_{SASP} C_p \dot{V}_{SA} (T_{TO} - T_{SASP}) \tag{4.15}$$

To accurately predict the amount of energy supplied to the space, the terminal outlet temperature needed to be modeled.

<u>Terminal Outlet Temperature</u>

The terminal outlet temperature (TOT) is one of the main driving forces of energy consumption. Built into the building controls is the logic containing how to modulate the flow through the heating coils to achieve a warmer terminal outlet temperature. The ROM and HVAC models are only trying to simulate the indoor zone temperature and the total energy consumption of the building respectively, regardless of the mechanism used to obtain a higher terminal box outlet temperature. With this knowledge, the flow through the heat exchanger coils was simplified by only modeling the outlet temperature itself. Introducing heat exchanger efficiencies would over complicate the nature of the model and limit its usability to individuals with more advanced technical experience.

As it can be seen in Equation 4.15, the amount of heating or cooling supplied to the zone is proportional temperature difference between the supply air set point and the terminal outlet temperature. The terminal outlet temperature is modulated depending on whether the room's thermostat set point is being met. If the zone required additional heating, the controller opens a valve allowing more hot water to flow through the coils effectively increasing the outlet temperature, thus increasing the amount of heat supplied to the space. To model this behavior correctly, all conditions had to be accounted for which are shown in Table 7.

Table 7. Terminal Reheat Operational Modes

Case	***Mode***	***Description***	***Terminal Reheat***	***TOT Range [°C]***
1	Cooling	The zone's cooling demand is being met and the space is being conditioned to the appropriate set point	No	12.78
2	Cooling	The zone's cooling demand is not being conditioned to the appropriate set point	No	12.78
3	Cooling	The zone is being over cooled	Yes	12.78 – 18
4	Heating	The zone's heating demand is being met and the space is being conditioned to the appropriate set point	Yes	18 – 33
5	Heating	The zone's heating demand is not being met and the space is not being conditioned to the appropriate set point	Yes	33

Most of the time the heating or cooling demand will not be met exactly so the terminal outlet temperature will have to be modulated and for each case the terminal outlet temperature will have different ranges of operation. A further description and explanation of each case is as follows:

Case (1 & 3) – This case typically occurs during the shoulder season and occasionally the winter months. The COBE building has high internal loads due to the occupancy and a server room and needs occasional cooling during the winter months. When the internal loads are high there will be no need to reheat the supply air before it enters the zone. When the internal loads are being met exactly, the terminal outlet temperature is set to the supply air set point (12.78°C). However, most of the time the loads will not be being met exactly so the air will need to be reheated to avoid over cooling the space. The minimum and maximum outlet cooling temperatures were modeled to be 12.78°C and 18°C respectively.

The minimum air temperature the unit can supply is the supply air set point temperature because the terminal units are only capable of supplying heat to the air stream. The maximum temperature was chosen to fall in between the occupied and the unoccupied cooling set point temperatures, which can be seen above in Table 7.

Case (4) - This case typically occurs during the shoulder seasons and the zone's heating demand is being met. During this operational mode, the terminal outlet temperature is modulated from 18°C up to the maximum heating outlet temperature of 33°C.

Case (2 & 5) - These cases typically occur during the peak heating and cooling season. There are operational limits to the upper and lower terminal outlet unit can achieve, which is due to system efficiencies. During the peak cooling season, the supply air is entering the zone at the supply air set point temperature (12.78°C), as well, during the peak heating season the supply air is entering the zone at approximately 33°C. This value was chosen by comparing the results that EnergyPlus predicted. It should be noted that the actual COBE building may be able to achieve a higher terminal outlet temperature, but to avoid discrepancies when comparing this model to the EnergyPlus, the temperature was chosen to reflect the value from the previous COBE model. If the set points are still unable to be met by modulating the air temperature to the absolute maximum and minimum for heating and cooling respectively, the VFD signal is turned on and the supply air flow rate is increased thus, supplying more heating or cooling to the space. This function will be discussed in further details in the following section.

For all the individual cases the terminal outlet temperature varies from one another. There are two operational modes that the outlet temperature needs modulated, Case 3 and Case 4; Equation 4.16 and 4.17 show the how the TOT temperature was calculated for both.

$$TOT_{Cooling} = Max\ Cool\ Temp - [(T_{zone} - Cool\ Tstat) \cdot 2.25] \quad (4.16)$$

$$TOT_{Heating} = Min\ Heat\ Temp + [(Heat\ Tstat - T_{zone}) \cdot 2.25] \quad (4.17)$$

Where $CoolT_{stat}$ and $HeatT_{stat}$ are the current cooling and heat set points respectively [°C]. The max cool temp and the min heat temp are both 18°C. Equation 4.16 and 4.17 show that the larger the temperature differential between the zone temperature and the current set point, the higher or lower the terminal outlet temperature becomes for heating and cooling respectively. A small temperature indicates that the zone is being conditioned well, and that no additional heating or cooling is needed at the current time. But when the differential is large, that is an indication that the current energy being supplied to the zone is not sufficient to meet the demand of the zone, meaning more energy needs to be introduced to condition the zone to the current set point. The value for the linear gain was determined by trial and error. All the discussed logic above was modeled in Simulink and the block diagram of the system can be seen below in Figure 19.

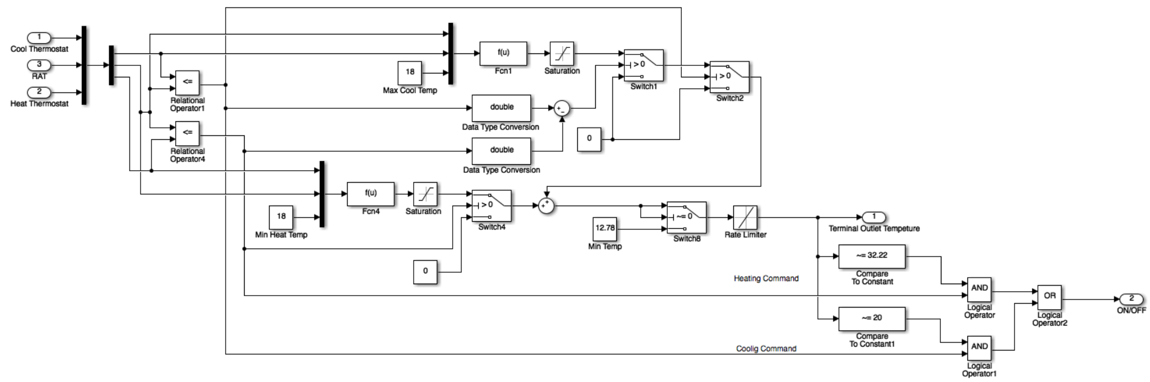

Figure 19. Terminal Outlet Temperature Simulink Block Diagram

The heating or cooling demand will not be met during the peak seasons by modulating the terminal outlet temperature alone. When the zone calls for more cooling than can be supplied by the TOT, the supply air flow rate has to be increased to meet the demands.

<u>Supply Air Flow Rate</u>

The supply air flow rate was modeled the same way as the terminal outlet temperature. The supply air flow rate is turned on and off through a 'VFD' signal that originates in the TOT model. When the cooling or heating demand cannot be met by modulating the outlet temperature, the VFD signal is turned on which allows the flow rate to be increased, if the signal if off, the flow rate allowed into the space is the minimum required ventilation air. For both heating and cooling, the flow rate is controlled in the same way, which is through applying a liner gain to the temperature differential between the zone temperature and the current set point, which can be seen in Equation 4.18.

$$\text{SA Flow} = [(T_{zone} - T_{stat}) \cdot 0.2] + Min\ Flow\ Rate \qquad (4.18)$$

The linear gain was determined through an iterative process. The minimum and maximum flow rate of the system was determined by examining the EnergyPlus' prediction for the flow rate. Once again, the actual COBE Building may achieve a higher or lower flow rate, but since the model's effectiveness is based off a comparison between EnergyPlus, so those values are more important. The full block diagram of the supply air flow rate subsystem can be seen in Appendix C.

Once the flow rate of the system is accurately calculated, the amount of heat or cooling entering the space can be determined. This modeled energy flow is a feedback for the thermal model and is applied at the center of the zone.

CHAPTER FIVE: RESULTS

The ROM was utilized to predict the performance of the COBE building and several variables were used to compare the model's overall fidelity. Moscow, Idaho's actual meteorological year (AMY) weather file for the 2014 – 2015 calendar year was used as one of the ROM's inputs, along with the heating and cooling demand, and the solar gains as calculated by EnergyPlus. The COBE's performance was simulated for a year and the results were compared to both the single zoned and the fully zoned EnergyPlus model of the COBE building. The single zone EnergyPlus model was used to compare individual output parameters such as zone heating and cooling loads, supply air flow rate, etc., whereas the fully zoned EnergyPlus was compared to the ROM for the virtual commission recommendations. This comparison method was chosen due to the simplification made earlier in the modeling process. The ROM was optimized to thermally perform similar to the single zone EnergyPlus model, and the parameters were optimized to match the response of the single zone EnergyPlus model, and as such the individual output parameters should represent the single zone model more accurately. The commissioning recommendations are going to be compared against the fully zoned model of the COBE building. The fully zoned COBE building has been calibrated according to ASHRAE's standards and is a more accurate representation of the actual building than the single zone model.

Common variables and energy consumption metrics between the two models were selected for comparison. They are as follows: zone air temperature, supply air flow rate, zone heating and cooling loads, total building energy demand, and the percent energy savings from control-based commissioning. The process flow indicates what EnergyPlus model was used for comparison between each variable can be seen in Figure 20. It should be noted that the heating and cooling loads do not account for system efficiency. It is the ideal load that will keep the space conditioned at the given set point, given all ambient and internal effects. The zone heating and cooling load was chosen to account for errors caused by the VAV box system efficiency. The ROM did not include any measure of efficiency, making the modeled value more representative of a load, and not a consumption. If this parameter was not chosen for comparison, it would have to be assumed that VAV box system efficiency is independent of the heating or cooling load, which is typically not the case. The losses in the terminal box are from the heat exchanger in the unit, which is dependent of both fluid flows, supply air and district hot water. The inlet and outlet temperature of the heat exchanger water was not modeled directly, so determining the VAV system efficiency in the ROM was not feasible. Making it a necessity to compare the ROM predicted terminal reheat energy usage to the EnergyPlus model's predicted heating and cooling load of the zone.

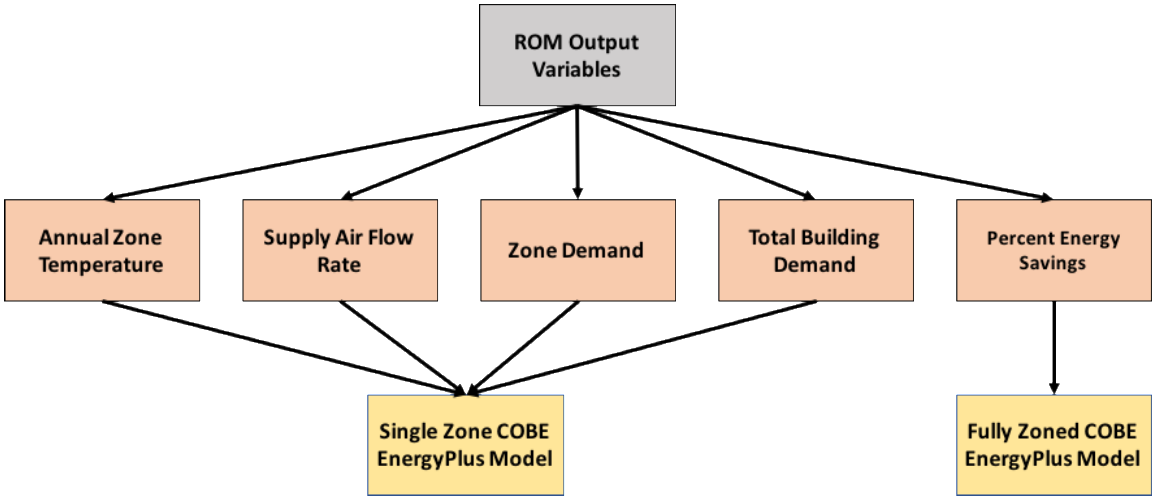

Figure 20. Process Flow of ROM Variable Result Comparison with EnergyPlus Model

ASHRAE has designated the comparison method that should be used for energy modeling. The two recommended modes of comparison, by ASHRAE's Guideline 14 [30], are the coefficient of variation of the root mean square error (CVRMSE) and the normalized mean bias error (NMBE). ASHRAE Guideline 14 considers a building model calibrated with hourly data to have a CVRMSE within the range of ±30%, and NMBE in the range of ±10%. The CVRMSE and the NMBE are shown below in Equations 5.1 and 5.2 respectively.

$$\text{CVRMSE} = \frac{\sqrt{\sum_{i=1}^{N_i}\left[\frac{(y_i - \hat{y}_i)^2}{n-p}\right]}}{\bar{y}} \tag{5.1}$$

$$\text{NMBE} = \frac{\sum_{i=1}^{N_i}(y_i - \hat{y}_i)}{(n-p)\bar{y}} \tag{5.2}$$

Where y_i and $\hat{y}_i$ are the ROM and EnergyPlus predicted value respectively, n is the number of sample data points, "p is the number of parameters or terms in the baseline model, as developed by mathematical analysis of the baseline data" [30], (in this case $p=1$), and $\bar{y}$ is the arithmetic mean of the EnergyPlus observations. The CVRMSE value is representative of how well the two values trend together throughout the year, whereas the NMBE is an indication of how accurate the overall magnitudes compare to one another. Both values must fall within the range set by ASHRAE to be considered 'calibrated'. Typically, this standard is used to compare the energy consumption predicted by the energy model and the actual building energy consumption, as reported on the energy bills, but this method should still remain valid when comparing one energy model to another.

The amount of heating or cooling supplied to the zone is, in part, a function of the supply air flow rate. This variable was compared by looking at the difference between the daily average values between the ROM and the single zone EnergyPlus model, otherwise known as the residuals. A histogram of the daily average residuals, as well as a normal distribution centered around the average, is shown below in Figure 21.

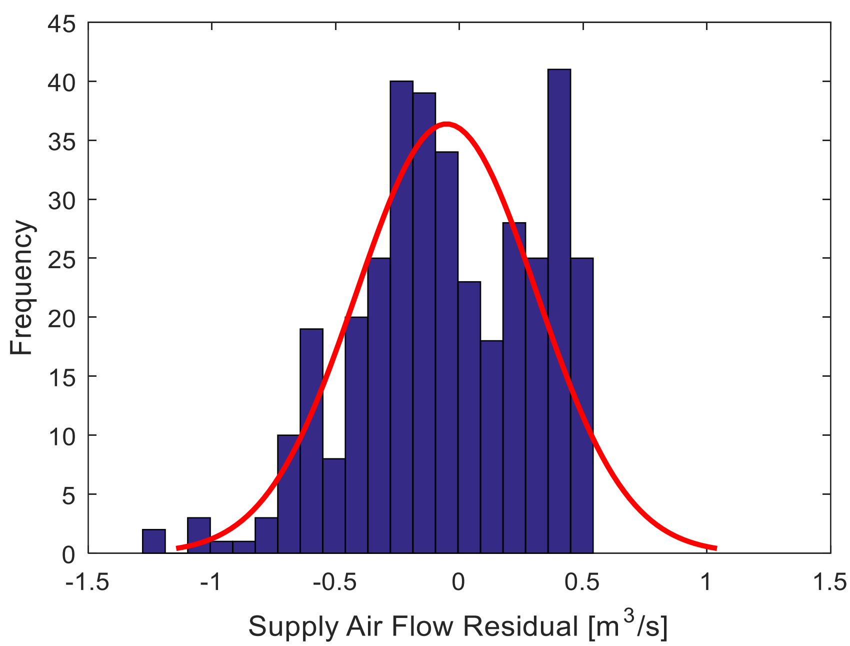

Figure 21. Daily Average Supply Air Flow Rate Residual

The daily average difference between EnergyPlus and the reduced order model is -0.095 m^3/s. The CVRMSE and the NMBE for the supply air flow rate were 6.057% and -0.561% respectively. These results are an indication that the flow rate was modeled correctly, and the linear gain factor are similar to the ones used in EnergyPlus.

The next variable compared was the amount of energy supplied to the zone. As discussed earlier, EnergyPlus does not decouple this parameter from the VAV energy consumption, making the comparison only possible if the HVAC heat and cooling zone loads are used from EnergyPlus. These heating and cooling loads are going to be compared to the ROM predicted value of energy supplied to the zone, as seen in Equation 4.12 and 4.13. The ROM zone energy supply and EnergyPlus heating and cooling loads are comparable because the heating and cooling supplied to the zone should be the amount of energy required to maintain the space at the thermostat set point, which is also known as the heating or cooling load. This individual parameter is an indication of the thermal parameter's overall accuracy. If the parameters were optimized correctly, the model's overall heat transfer coefficient should be identical between the ROM and EnergyPlus. A histogram of the residuals, as well as a normal distribution, can be seen below in Figure 22.

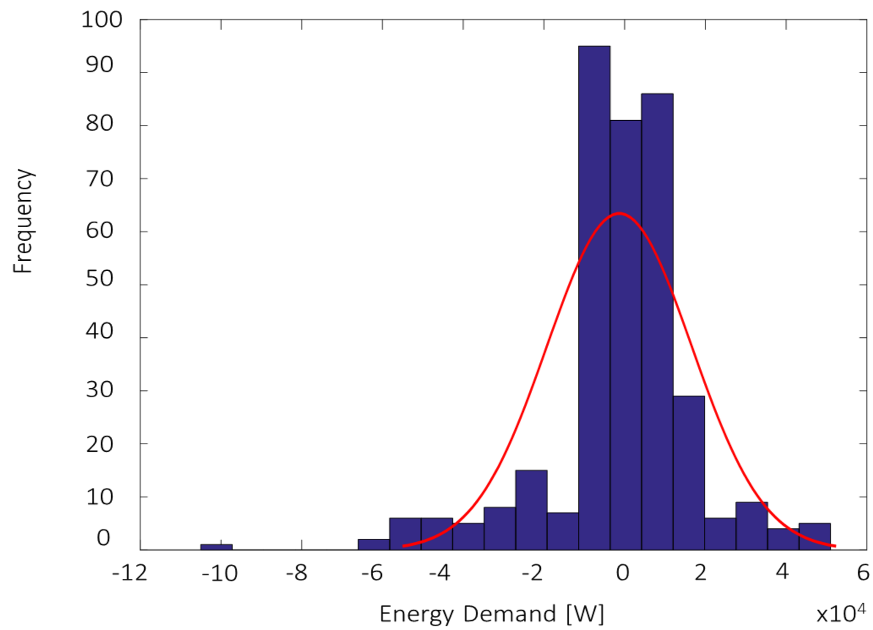

Figure 22. Daily Average Zone Demand Residual

The CVRMSE and the NMBE for the Zone Demand was 63.7% and 2.38% respectively. The zone demand showed poor CVRMSE performance, but that should be expected. ROMs lump masses together and assume each mass has an equivalent temperature, varying the magnitude of heat transfer at any given time when compared to EnergyPlus. The NMBE shows the two models use similar overall magnitudes of energy throughout the year which indicates the model is performing. The average daily zone demand residual was approximately -1,336 watts where the COBE building's total average zone demand was approximately 25,000 watts. These results are indication that the ROM is predicting a zone load of 1,336 watts less than EnergyPlus, which is expected due to the ROM zone temperature being over predicted, as seen in the previous chapter. This over prediction of zone temperature equates to the HVAC system having to add less thermal energy to condition the zone and match the thermostat set points.

The last variable compared was the total energy consumption of the building, which includes the energy supplied to precondition the air stream after the mixing chamber and at the terminal boxes. The total energy consumption does not include any mechanical energy consumed by the supply fans or the return fan, this variable is just the thermal energy supplied to the air stream. The first method of comparison was by plotting the energy signature of the COBE building as predicted by both models. An energy signature is a plot of the energy consumption vs. the average ambient temperature, typically tabulated daily [31]. Characterizing a building using an energy signature offers a quick method to determine how the building is performing and is a way to graphically illustrate the amount of heating or cooling required for any given outdoor temperature conditions. The heating and cooling energy signatures of the COBE building, as predicted by the reduced order model, were compared against the EnergyPlus, which is shown in Figure 23 and Figure 24 respectively.

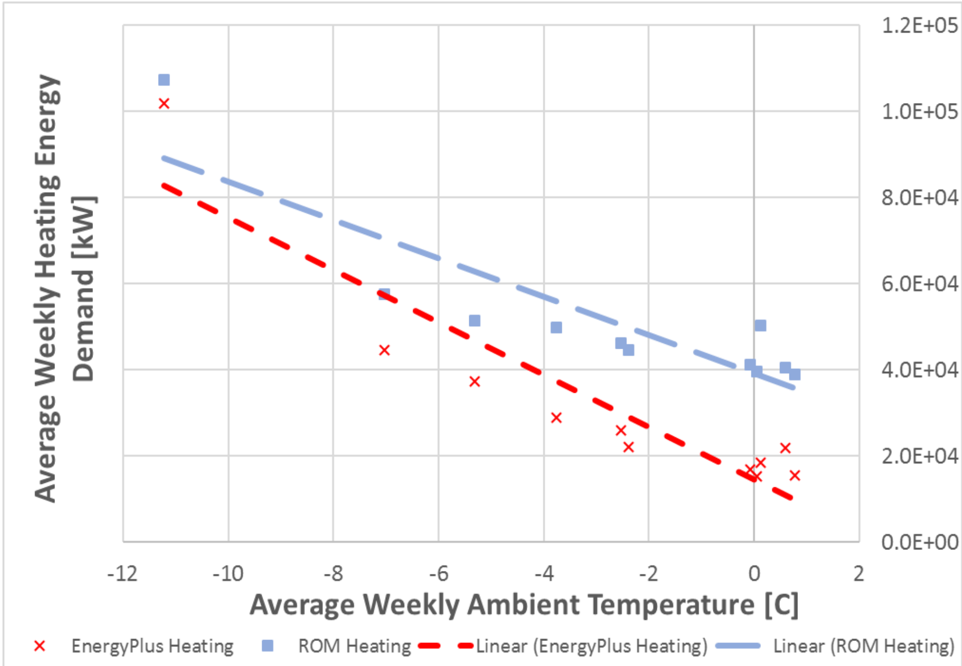

Figure 23. **EnergyPlus and Reduced Order Model Heating Energy Signature of COBE Building.**

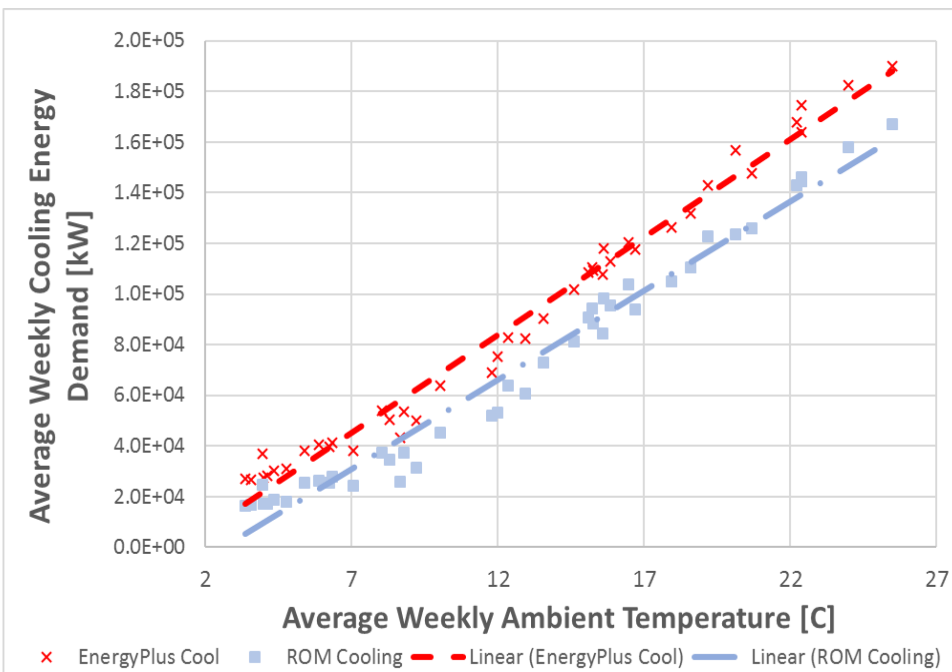

Figure 24. **EnergyPlus and Reduced Order Model Cooling Energy Signature of COBE Building.**

The balance temperature of the building is the ambient temperature at which the building does not require heating or cooling after adjusting for internal loads was determined to be approximately 2.5°C for both the EnergyPlus and the reduced order model. Both heating and cooling energy signatures from the reduced order model are similar to the predicted signatures from EnergyPlus. This is an indication that both models have similar thermal properties. However, the cooling energy signature for the reduced order model mirrors that of the EnergyPlus' model better than the heating energy signature. This is thought to originate from the terminal reheat VAV box sub model. The histogram of the daily average total energy demand residuals was plotted and the results can be seen below in Figure 25.

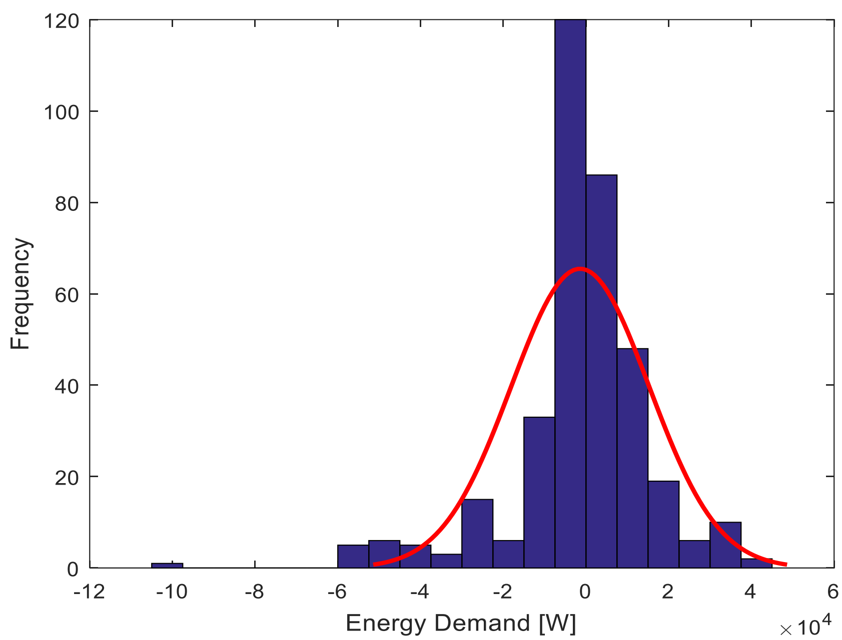

Figure 25. Daily Average Total Energy Demand Residual Histogram

The total energy demand residual had an average of approximately -1,401 watts. The overall CVRMSE and the NMBE for the total energy demand when compared to EnergyPlus was 42.4% and 1.7% respectively. This is indicating that the ROM's heating and cooling demand is, on average, 1,401 watts less per hour than the EnergyPlus model. This error also originates from the over estimation of the ROMs zone temperature, as discussed above. The ROM accurately predicted the magnitude of total energy consumption when comparing the results to the previous EnergyPlus model. The CVRMSE value was less than allowed by the ASHRAE standard, but due to how the model lumps masses together, is still an accurate result. The ROM is verified to be an accurate representation of the COBE building and the next step is using this model as a tool for virtual commissioning by looking for recommendations that can yield realized energy saving at the COBE building in Moscow.

Virtual Commissioning

Remote access was granted to the EMS for the COBE building at the University of Idaho, and while logging onto the system it was noticed that the building was operating in “occupied” mode at a time when educational buildings typically are unoccupied. While operating in occupied mode, the HVAC conditions the building to different temperature set points and uses more electricity than unoccupied mode. This controller setting was investigated multiple times by logging into the system at typical unoccupied times, and problem persisted which points to the discrepancy not being by chance, but an overall operational and control issue. The reduced order model was used to determine the energy savings associated with programming thermostat setbacks into the controller. Table 8 shows the current and recommend thermostat settings, which were the values used in the study.

Table 8. Current and Recommended Thermostat Set Points

	Occupancy Status	*Heating Set Point [℃]*	*Cooling Set Point [℃]*
Current Thermostat Set Points			
	Occupied/Unoccupied	20.0	22.78
Recommended Thermostat Set Points			
	Occupied Unoccupied	21.0 15.6	24.0 26.7

The values for the current set points were determined by examining the trend logs of zone temperatures from the COBE's EMS system and the values represent an average of all the zones. The recommended thermostat set points were determined using ASHRAE's standards. The COBE building was simulated using the ROM with both the current and recommended thermostat set points and it was found by utilizing thermostat setbacks during unoccupied times, the HVAC energy consumption would be decreased by 9.6% annually. This study was also conducted using the fully calibrated COBE EnergyPlus model that was develop during the previous Avista Research Grant. With the full EnergyPlus model, it is predicted to save approximately 9.97% of heating and cooling energy by adjusting the thermostat set points. This energy savings does not include pump or fan power savings; the reported value is only the amount of energy consumed while conditioning the space.

The above process illustrates the effectiveness of using ROM to virtually commission buildings. The building's thermo-physical properties do not need to be known, which is difficult to predict after many years of degradation. Instead, this approach uses building data as a means to optimize the effective building parameters to estimate the total energy consumption. This method can be used to determine the potential energy savings for any control signal that is captured in the ROM. A preliminary study was conducted with the economizer high limit shut off set point and the results indicated that the ROM agrees with the energy savings predicted from EnergyPlus. This needs to be investigated further before any definite conclusions can be drawn. This modeling approach used building data generated from a calibrated EnergyPlus model, not actual building data from the COBE building, which is the natural progression of this research. This approach needs to be applied to an actual building using EMS data from the real building and determine if the model parameters are obtainable.

CHAPTER SIX: CONCLUSION

This research has shown that using ROM for virtual building commission is a viable option for whole building commissioning. This approach lessens the time and money constraints that are prohibitive for many building owners. The difference between the results of the fully-zoned EnergyPlus model and the reduced order model was insignificant when predicting the amount of energy savings from thermostat setbacks. Approximately 104,000 kWh annually can be saved with temperature setbacks during unoccupied periods. There are areas within the research that need further developing to increase the accuracy and usability of this process as a whole.

One aspect that needs further development is how to accurately model the HVAC system and how to couple it with the ROM more effectively and accurately. The thermal mode accurately predicted the zone temperature of the COBE building when compared to the single zone EnergyPlus model. But the zone temperature is only an intermediate variable, the critical parameter is the overall energy consumption of the building. The simplified HVAC model translated the indoor zone temperature into energy consumption using a controller (thermostat). Initially this research focused on whether a ROM was capable of accurately predicting the zone temperature given some heating or cooling load. These loads were going to be computed and supplied as an input to the thermal model, but it was determined that wouldn’t be an accurate representation of the building. The energy supplied to the zone is only part of the total energy consumption, with the other half coming from the energy needed to precondition the air at the main air handler and the terminal reheat box. This indicated that an HVAC model needed to be developed. The HVAC model was designed to be simple and following first principals. It was not developed as strictly as the thermal model was, and as such, lacks the fidelity seen in the ROM. The HVAC model only relies on the effective zone temperature of the modeled zone capacitance as a feedback from the ROM. The HVAC system supplies energy based on this one feedback and that introduces errors because the effective zone temperature is not a direct correlation to what the apparent zone temperature is.

Additionally, the HVAC applies all the heating and cooling energy to the zone node, which is an oversimplification as discussed above. For any zone, there are typically more than one air diffuser per zone distributing the air throughout the entire space, including the walls, floor and ceiling of the zone. The energy in the air is absorbed by all the mass in the room changing its overall temperature, as well as the air. How exactly the air is distributed in the zone needs to be further researched to improve the accuracy of the model. Additionally, how the solar and conductive loads are applied to the thermal model needs to be further researched. The solar loads do not only come from conduction through the windows, it also originated from short and long wave radiation. The radiation heats up any surface it directly comes into contact with and is reflected throughout the zone. Assuming the radiation only effects the zone temperature causes additional errors that need accounting for.

This model only calculates the energy consumed conditioning the space to the thermostat set point temperature, and it does not include the energy consumed by the other mechanical systems, or the plug loads of the building. Typically, when calibrating energy models the only data available is the monthly energy consumption as reported on the energy bills. These bills include all energy consumed at the site and does not differentiate the plug loads from the energy consumed conditioning the space. To make this a more suitable modeling approach all the plug loads and other energy consumption need to be included in the model.

Finally, without access to accurate data from the COBE building, outputs from the previously developed EnergyPlus model had to be used to find the thermal parameters of the ROM. Ideally, this method of building modeling would be a standalone process and would not rely on an EnergyPlus model; all the ROM inputs would be calculated, or measured, or estimated using ASHRAE standard 90.1 [32]. The next natural progression of this research would be using actual building data to optimize the model parameters. Promising results were achieved determining the parameters using the buildings response for the BESTEST case study, and it was shown that the thermal parameters could be determined both ways, through optimization and through numerically fitting the parameters to best fit the building's thermal decay of energy. The differences between the time constants of the two models were insignificant and would have minimal effects on the overall building energy consumption. But the parameters were optimized to the building's temperature decay as modeled using EnergyPlus and not an actual building. The method of using the buildings temperature decay needs to be further investigated, and eventually needs to be accomplished using actual building data to see if the parameters can be found from large temperature setbacks similar to the process used in the BESTEST case study.

Overall though this method still holds relevance to the virtual commissioning process and with further research conducted in the above-mentioned areas can be as accurate and reliable as a fully commissioned EnergyPlus model as used to accurately predict expected energy saving from control-based building commissioning.

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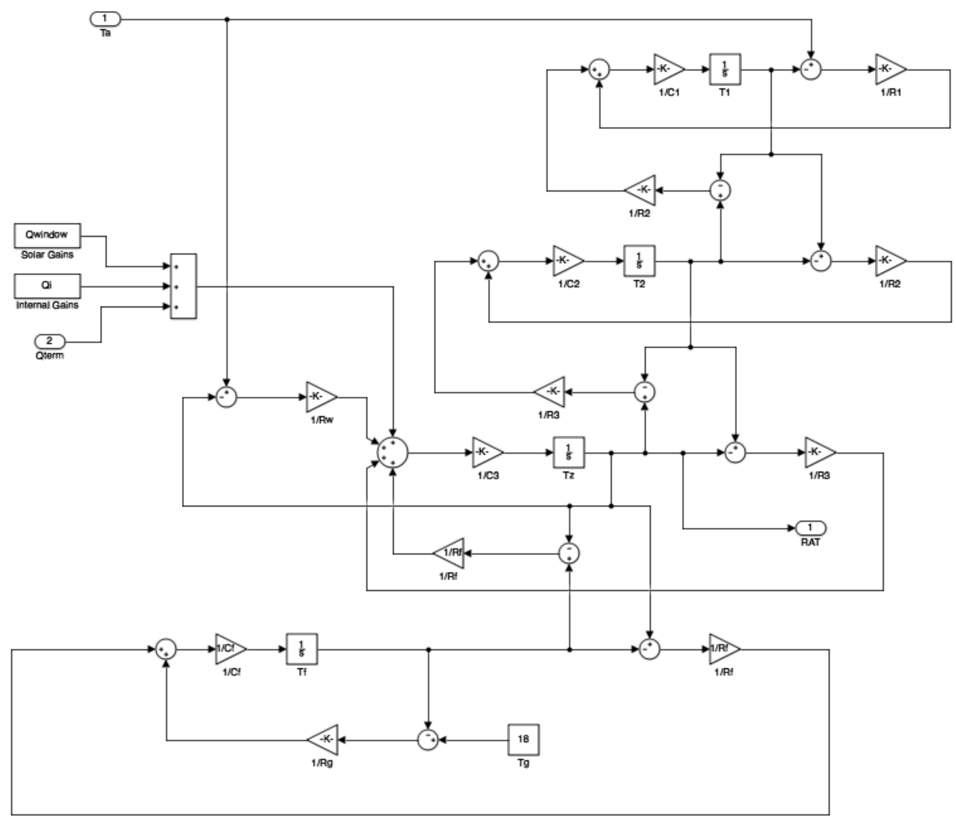
APPENDIX A:

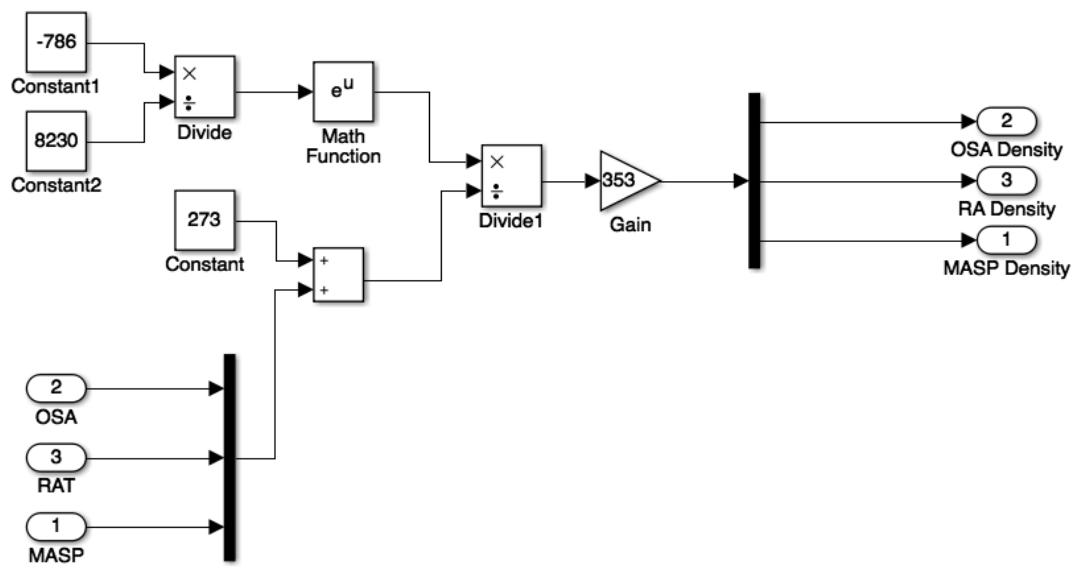
Thermal Reduced Order Model Simulink Block Diagram



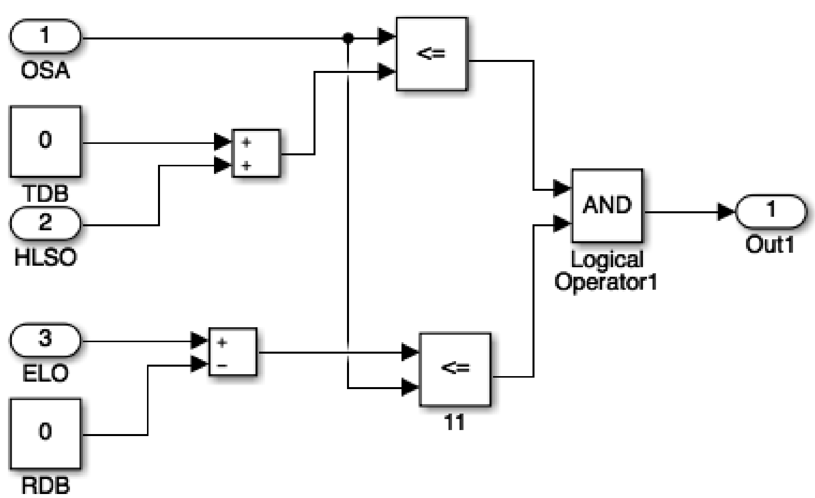
APPENDIX B:

Economizer Subsystem Simulink Block Diagrams

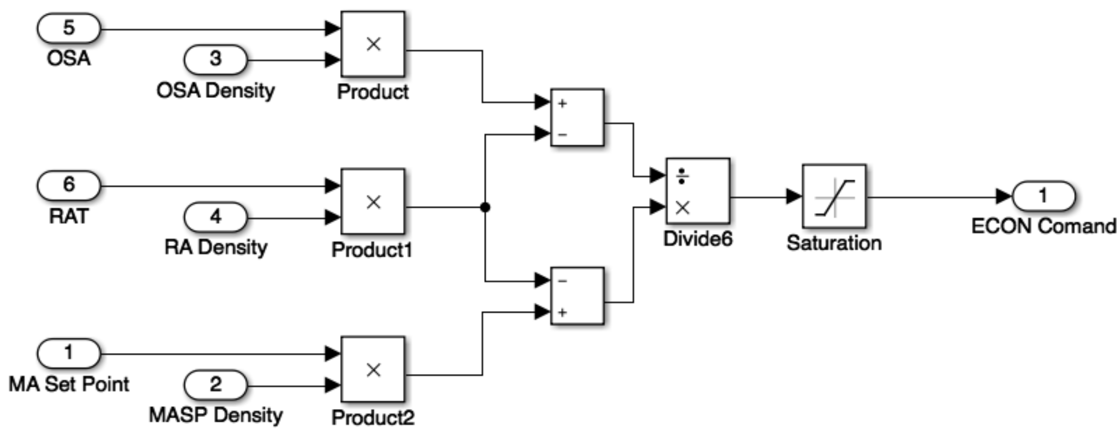
Air Properties Subsystem



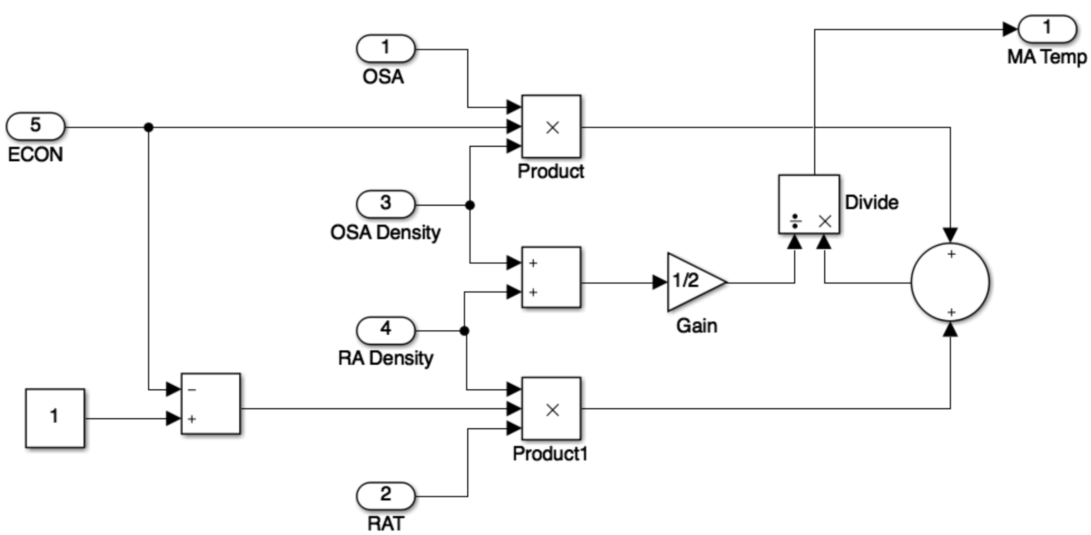
Econ On/Off Subsystem



ECON Command Control Subsystem



Mixed Air Temperature Subsystem



APPENDIX C:

Supply Air Flow Rate Model Simulink Block Diagram