THE IMPACT OF CONTROL SETPOINTS, ECONOMIZERS AND ENERGY RECOVERY SYSTEMS ON BUILDING ENERGY USE

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Abstract

This paper presents one possible method of applying simulation to examine the impact of building HVAC system setpoints such as air/water temperature, flow rate and outdoor air fraction on represented building operation for a typical system design and operation. The method utilizes the performance characteristics of the system components along with the outdoor and indoor air conditions to determine the optimum collection of setpoints and operating conditions. The idea is to use simulation predictions to get the best building system setpoints with the best energy performance. The objective is to determine the range of energy usage and the potential for reducing or minimizing building energy requirements by dynamically adjusting setpoints. The analysis focuses on the energy consumption condition in four different climate zones. At the same time, an improved method to determine the optimum outdoor air fraction and mix air temperature of the economizer and the energy recovery ventilator in certain outside weather environment are discussed and demonstrated. Potential energy savings due to improved control algorithms are presented and discussed.
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the two-year study.
1. **Introduction**

As we all realize, building consumes a significant amount of energy, while heating, ventilating and air–conditioning (HVAC) systems contribute more than half of the building energy consumption according to the data from Commercial Buildings Energy Consumption Survey (CBESC) made by U.S. Energy Information Administration in 2003. The increasing demand of air-conditioning and the energy crisis during the last decades have led to a surge of attention and there is no doubt the improvement of the HVAC control system is one of the effective solutions to realize sizable energy-saving for the building sector. The aim of HVAC control is to provide a comfortable, safe, healthy and productive environment for occupants with least energy. However, the biggest impact factor effecting on the building energy consumption which is the HVAC control has the least installations units in buildings (WBCSD, 2009). That means a significant saving potential exists for building systems energy consumption during operation with the help of current fancy control technology such as intelligent, adaptive or model predictive control. The development of this kind of technology has led to the possibility of the improvement of building operational performance as well as energy usage.
Figure 1. Energy Consumption by End Use of Buildings (CBECS 2003, EIA)

Figure 2. Installed units versus system impact on energy consumption in buildings (WBCSD, 2009)
The challenge is that firstly we should get to know how much energy can be saved and how to achieve the saving through improved control logic. However, it is difficult to evaluate the potential or effectiveness of the new control strategies without first gaining a better understanding of the range of operating conditions possible for any particular building/HVAC system combination. That is, the amount of energy savings is a function of both the actions of the new control strategy and the fundamental capabilities of the HVAC system. In its most basic form, a building control system can do no more than monitor sensors, apply logic and manipulate actuators. Thus, the main objective of the work described in this paper is to clearly identify and define the space within which the building/HVAC combination is capable of operating in order to enable the determination of both energy saving potential and optimal setpoints and control logic. While this is not specifically an optimization effort, i.e. a single optimal solution is not the goal since it is understood that setpoints and control logic may need to be adjusted on a dynamic basis based on different requirements, the primary metric utilized, namely total building energy usage, can be considered as an objective function here. Comparisons of building energy usage with different system setpoints may give us new ideas on how to operate building HVAC system more wisely.

Meantime, building economizer and energy recovery systems which have evolved and boomed in recent years can also be very useful for reducing energy requirements associated with the conditioning of outdoor air required for ventilation, and may also be able to provide free cooling under some conditions. That’s the reason why building economizer cycles and energy recovery systems have become a mandatory requirement in certain outdoor weather condition and HVAC operation condition. New ASHARE standard 90.1 also strongly encourages buildings in certain locations to use economizer and heat recovery technology. However, implementing either
economizer or energy recovery requires various dampers and mechanical systems, along with a control strategy or more energy may be consumed instead of saving energy. So some work must be implemented to determine how and when the systems should be operated to obtain the most favorable outcome.

The content is organized as follows. Section II reviews recent studies on HVAC control, simulation, modeling, related software and energy recovery and economizer system. Section III presents simulation work. Section IV gives the results and discussion. Section V presents the conclusions. Last section shows some recommendations for the future work.

II. Literature Review

HVAC Control
For many years, control has been a very active research area of the development in the HVAC field, aiming at operation of HVAC systems in terms of reducing overall system operating cost, ensuring thermal comfort of occupants, and satisfying indoor air quality. But a tension exists between building energy efficiency, thermal performance and occupant health and comfort, whereby the means employed to minimize energy usage may negatively impact thermal and air quality within the building’s conditioned space. A properly operated HVAC system finds the often-delicate balance between optimizing occupant comfort while controlling operating costs. Comfort is really an important issue as it can directly affect occupant’s concentration and productivity. At the same time, controlling these comfort and health parameters directly affects HVAC system operating costs in terms of energy, maintenance and equipment life. Back into 1990s, a study had shown that the optimal control can save 11% of utility cost compared to the conventional control strategy for a two – zone HVAC system (House and Smith, 1995). Most
efforts in the control of building HVAC systems are typically based on the modeling and simulation. In the building field, simulation can be applied to reveal the inter-actions between the building itself and its occupants, HVAC systems, and the outdoor climate which will be presented in detail in the following part. A large amount of work has been done to show how important building simulation is in the study of energy performance and the design and operation of energy-efficient buildings (Clarke et al., 2002), and a lot of work has also been done in the field of building energy consumption simulation, as (Pan, Huang and Wu, 2007) and (Li, Pan and Chen, 2009) analyzed and displayed the building energy break-down with calibrated models in 2007 and 2009 respectively, however, more effort is needed to understand how to obtain optimum operating parameters, particularly for building control systems. Simulation and modeling do provide a good opportunity to evaluate the dynamic and energy performance of HVAC system control strategy in a convenient and low cost way. The control strategy can also be pre-tuned before being utilized in the real system with the help of simulation and modeling.

**Modeling**

The significance of the developing models is that they can be used for simulation and optimization. In order to carry on the simulation and optimization successfully, all the individual components of the HVAC system that have influence on the objective function must be included with the mathematical models, and validation of simulation models is another key issue in order to have an accurate and intensive simulation result.

The challenge is also very obvious: it is always not an easy task to build models for HVAC system as there are so many components and the prime parameters for control, the temperature and humidity, are very prone to be effected by uncontrolled factors. Modeling certain part of the HVAC system such as chiller, cooling coil and heat exchangers has been a hot filed of research
over the past decades. For example, a simple engineering model for a cooling coil unit (CCU) based on energy conservation and heat transfer principles were derived by (Wang, Cai et al. 2004).

The HVAC system modeling methodologies field is the part attention will be focused on. There exist four different approaches presented by Hensen in 1996: 1). Pure conceptual system modeling approach, which means that the HVAC primary and secondary systems is idealized and a capacity limitation may be imposed upon them. This kind of approach is widely used in a number of current building simulation software such as Design Builder based on the EnergyPlus, in which there are options from “simple” HVAC system to “compact” one. If “simple HVAC system” is chosen, the simulation engine will run in the “ideal load system” mode, in which case, the plants will provide exact the same cooling/heating energy as the summed zone loads at each time step and the energy consumption is calculated by using constant conversion factors-efficiencies or COPs. The distribution energy is usually omitted (Wang and Ma, 2008). 2). System-based modeling approach. For this method, the user must preconfigure the system so the system capacity, type and parameters are very flexible. But the limitation is that only predefined control strategies could be specified or selected. 3). Component-based system modeling approach. This represents the case where a system is configured by a network of inter-connected component models. 4). Component-based multi-domain system modeling approach. This one is different from the former in a way that component is further divided into multiple interrelated balance concepts, e.g. heat balance, mass flow balance concept. More flexible in terms of possible system arrangement and control strategies could be achieved compared with the modeling strategies above, but generally with longer calculation time as a penalty. HVACSIM+ and TRABSYS is an example of component-based simulation tool. In 2010, Trcka and Hense
came up with a fifth category: Equation-based system modeling approach, where a system is represented by a basic modeling unit physically smaller than a component in the form of an equation or a low-level physical process model. This methodology is recently emerged and motivated by the need to improve the building performance simulation (BPS) tools developed in the early 1970s. Examples of equation-based tools include SPARK (Simulation Problem Analysis and Research Kernel) developed by Lawrence Berkeley National Laboratory, EKS (Energy Kernel System) in the UK and Modelica. This modeling approach has become popular in other industries such as chemical engineering.

Simulation

Building simulation began in the 1960s and became a hot topic of the 1970s within the energy research community. For nowadays, computer simulation is not only playing an increasingly significant role in building design stage like sizing and configuration design, but is also being adopted for system performance analysis more and more widely. Simulation is taken as one of the oldest but very effective tools to engineers in every discipline. Building simulation can be applied to reveal the inter-actions between the building itself and its occupants, HVAC systems, and the outdoor climate. Future development and application of information technology in the building industry will lead to a completely new building design philosophy and methodology (Hong, Chou and Bong, 2000). In 2003, (Mathews and Botha 2003) conducted simulation with three cases and proved that simulation does indeed have the ability to improve the thermal and energy management of building HVAC systems. A lot of work has been done in the field of building energy consumption simulation but the attention of system control is far from enough. Simulation does provide a good opportunity to evaluate the dynamic and energy performance of HVAC system control strategy in a convenient and low cost way. The control strategy can also
be pre-tuned before being utilized in the real system with the help of simulation. Traditionally, less attention has been put on buildings operation compared with the design of a system and its construction/installation. What’s more, the simulation software that uses models has been evolving steadily over recent years. HVAC component and subsystem models are now generally well understood and have been the subject of a number of researches (e.g., Clark, 1985). Simulation has extended to the use to the building operation process although it has been traditionally regarded as a design tool. In 2003, (Mathews and Botha, 2003) conducted simulation with three cases and proved that simulation does indeed have the ability to improve the thermal and energy management of building HVAC systems. Recent research also showed performing building simulation analysis enabled diagnosis of malfunctioning or incorrectly commissioned equipment within the building and thus also assisted with future commissioning and tuning of the building performance (Osborne, 2011). In the literature (Homoud, 1994), with optimization for lighting, cooling and heating, a saving of 5%-30% in annual energy consumption can be achieved. All the content above proved simulation in building energy sector has effects on building energy efficiency improvements.

**Optimization**

Optimization is an area of mathematics that is concerned with finding the “best” points, curves, surfaces, etc. (Hull 2003). Both significant energy saving and improvement on indoor environment can be achieved by HVAC system optimization (Nizet et al. 1984). Finding the optimal solution to an optimization problem is the key issue for a supervisory control application. However, it is never easy to realize. The components in HVAC systems, such as pumps, fans, chillers and so on, are interacted with each other, and the fact is that the saving of energy input of one subsystem might result in the increase of energy input of the other part with respect to the
changes of certain control variables. Therefore, the optimal solution for the related control variable is the trade-off between the energy inputs for all subsystems. For example, for a specific HVAC system, the optimal chilled-water temperature setpoint is the trade-off between the electrical power of both chillers and chilled-water pumps, while the optimal condenser-water temperature setpoint is the trade-off between the electrical power of both chillers and cooling tower fans. The objective of optimization is usually not hard to be decided. The difficulty is to determine what variables should be optimized and which kind of method should be adopted. It is obvious that it is inefficient and impractical to optimize all of the different control variables at the same time as a whole. For a particular system, it is quite necessary and essential to identify and divide the variables which have effects on trade-offs clearly. All the variables can be divided into three groups: uncontrollable variable, independent variables and dependent variables (Lu, Cai et al. 2005). The uncontrolled variables can be measured or treated as constants for the optimization problem. Independent variables can be taken as the input variables of optimization problem. Dependent variables are determined by independent variables with the regards of constraint.

**Software Package**

Realization of modeling and simulation is impossible without the help of related software. Simulation programs, such as EES, are very valuable tools to study building energy consumption under different conditions. These programs can be used to investigate the effect of different building designs on their energy consumption. Simulation programs, intended for research, have been around for many years but their development has not kept with other software commonly used by engineers. The number of such kind of software is quite limited. For now, the robust model adopted by virtual building is based on equations provided in HVACSIM+ and
EnergyPlus, while the computations are run in MATLAB, Simulink, and Simscape; the equations from HVACSIM+ are used for modeling HVAC components as well as room conditions while EnergyPlus is mainly used for components. The details of the modeling and simulation tools suitable for using in this study are provided as below.

**Matlab**: As previously mentioned, MathWorks software is the primary tool used for the building model computations. Among them, Matlab is no doubt the most mentioned and used software. It has been shown that, recently, a number of studies using Matlab environment has been grown rapidly (Dong, 2010). Matlab is capable of solving the systems of equations involved in the virtual building. However, Matlab does not provide the desired ease of use in terms of creating and modifying various virtual buildings. Therefore, several other modeling tools developed by MathWorks are used in conjunction to create the virtual building model. The success of this tool is probably due to the multidisciplinary in a single environment. What’s more, Development of control algorithms in tools like Matlab/Simulink is preferable as the code generated can be uploaded to controller as required. Matlab (matrix laboratory) is a high-level technical computing language and interactive environment for algorithm development, data analysis and numerical computation. Matlab can be applied in a wide range of fields such as control design, test and measurement, modeling and simulating processes. Especially, architectural engineers can use it to develop dynamic models of HVAC systems for controller development and calibration. However, matlab is not ideally suited for this case because it is complicated and slow at solving problems in which the effects of one or more parameters must be determined. For this study, as annual hourly energy simulation was carried, the calculation time should be a matter taken into consideration.
**EES:** The usefulness of Engineering Equation Solver has been showed in Bertagnolio’s work, the EES result is compared to that of the TRANSYS and the relative error is quite optimistic. In my view, one of the advantages of adopting models built in EES is the number of parameters is very limited and easily estimable. What’s more, EES, the solution of a set of algebraic equations, can efficiently solve hundreds of coupled non-linear algebraic equations and initial value differential equations. A major difference between EES and existing equation solving programs is the many built-in mathematical and thermal physical property functions, which are very helpful in solving problems in thermodynamics, fluid mechanics, and heat transfer for HVAC system. What’s important, its built-in look-up parametric table is really useful for searching the optimum value from different setpoints pool. The engineering equation solver (EES) has become a popular tool for engineering analysis. The CoolPack refrigeration model from Denmark was developed using EES.

**HVACSIM +:** It is really old-fashioned software running in Dos environment. Although the HVACSIM+ executable is not used, many of the equations used in HVACSIM+ have been written into the MathWorks codes. The blocks created in Simscape for individual components such as a fan have the HVACSIM+ equations embedded in their code. Aside from these, HVACSIM+ is pretty good at simulating dynamic responses benefiting from its characteristic that can adjust the time step by itself. Some HVAC components modeling equation from the HVACSIM+ Building Systems and Equipment Simulation Program Reference Manual are adopted.

**EnergyPlus:** It is a very famous energy simulation program which enables building professionals to model energy and water use in buildings in order to get optimized performance.
For this study, the components equations from the manual are utilized in the engineering equation solver for the simulation.

**Economizer and Energy Recovery System**

For a typical economizer function, the relative fractions of outdoor air and return air are modulated to produce a desired mixed air temperature entering the air handler. For cooling condition, outdoor air intake is minimized when its temperature or enthalpy exceeds fixed setpoints or the indoor air condition. The resulting mixed air must lie on the line connecting the outdoor and indoor air conditions depicted on the psychrometric chart like in Figure 3 & 4. The introduction of ventilation air from the outside may impact heating or cooling loads since the outdoor air needs to be conditioned to maintain the indoor comfort conditions. This means that in extreme climate conditions (winter, summer) additional energy may be required to heat or cool the ventilation air as compared to a sealed building. An economizer cycle utilizes outdoor air when appropriate to provide “free cooling”, the rough equivalent of opening the windows on a nice day rather than using mechanical cooling. The former study has demonstrated that the energy savings associated with economizer could be very significant (Brandemuehl et al., 1999). The energy saving of the robust control strategy on cooling coil energy consumption which was evaluated by over one year’s comparison tests on two air-handling units in a building was 88.47% in contrast to the energy consumption using the conventional control (damper at fixed position) in Hong Kong (Wang et al., 2004).

For Energy recovery systems, they are more complicated and required with more equipment. They may utilize rotating wheels, heat pipes or run-around coils to transfer heat and moisture between the outdoor entering and the exhaust air leaving a building. Generally, energy recovery systems capture some of the sensible or latent energy content of the building exhaust air and
transfer it to the outdoor air intake, without mixing or mass transfer. In essence, they can preheat or precool the makeup air, thereby reducing coil loads. Some units may transfer both sensible and latent energy (enthalpy wheels) while others are sensible or latent only, and can be placed in series. As shown in Figure 3 & 4, when both sensible and latent energy exchange are possible, the resulting preconditioned makeup air stream must fall somewhere within the rectangle defined by the indoor and outdoor air conditions, subject to the limitations imposed by the efficiencies of the energy recovery functions. Energy recovery systems exploit the “free heating or cooling” provided by exchanging energy between the exhaust and outdoor airstreams. In both of these systems, some control logic must be implemented to determine how and when the systems should be operated to obtain the most favorable outcome. During recent years, energy recovery ventilators (ERV) have increased in popularity throughout the world; this kind of equipment has been widely used in Europe for quite a long time (Lazzarin et al., 1998). There are presently two kinds of energy recovery ventilators available: rotary energy wheels (Stiesch et al., 1995) and membrane-based energy recovery ventilators (Zhang et al., 1999). The applicability and benefits of adopting ERV have been discussed by many researchers (Zhong et al., 2009). A study conducted in Hong Kong has showed that the yearly net cooling energy saving with a membrane-based energy recovery ventilator (MERV) can reach 12.4% (Zhang et al., 2001). The energy saving performance is related to many parameters such as ventilation air flow rate (Liu et al., 2007) but is mainly determined by the enthalpy efficiency which is made up of sensible load efficiency and latent load efficiency. Operating temperature and humidity have an influence on the enthalpy efficiency as has been observed by several energy wheel manufacturers and researchers (Simonson et al., 1999). One of the previous researches has shown that the average of efficiency of ERV is linear with the indoor temperature set-points of the air conditioner in
different seasons and for different weather conditions (Zhou et al., 2007). So proper control strategies in different climate conditions must be raised to maximize the benefits and avoid costing even more energy. Un-controlled operation of ERV may increase the cooling energy consumption by 5% (Rasouli et al., 2010). Different control options have been presented (Mumma et al., 2001). A useful figure called specific net energy saving was also introduced to determine whether a heat-recovery system will save energy or not (Roulet et al., 2001).

**Figure 3.** Economizer and energy recovery psychrometric states in summer time. ($E_1$ denotes the air condition from the economizer and $E_2$ denotes the air condition from the energy recovery system)
Figure 4. Economizer and energy recovery psychrometric states in winter time. (E’₁ denotes the air condition from the economizer and E’₂ denotes the air condition from the energy recovery system)

Above all, a proper outdoor air fraction and other related parameters must be determined to ensure the economizer and the energy recovery system to achieve energy saving goal instead of costing more energy.

III. Research

Weather Data
The simulations are performed in four different cities. These are: State college, Pennsylvania; Miami, Florida; Phoenix, Arizona and Minneapolis, Minnesota. The weather files used in this thesis are in the simulation use typical meteorological year (TMY) format and are obtained from
the United States Department of Energy website (2010). A typical meteorological year (TMY) is a collation of selected weather data for a specific location, generated from a data bank much longer than a year in duration. It is specially selected so that it presents the range of weather phenomena for the location in question, while still giving annual averages that are consistent with the long-term averages for the location in question. TMY annual weather data information is also used in the Energyplus program. The weather data is given for each hour throughout the year. The simulation is run at intervals of one hour.

The four different cities are chosen based on their typical weather patterns. Minneapolis (7A) was chosen for its cold, dry climate, State college (5A) for its mild climate, Phoenix (2B) for its hot, dry climate and Miami (1A) for its hot, humid climate. The cumulative outdoor dry bulb temperature, relative humidity and enthalpy distributions for each of these cities are given in Figure 5, 6 and 7 respectively.
From Figure 5 it can be seen that each of the four cities has a different temperature profile. In Minneapolis, the temperature gets quite cold, and most of the year (about 7000 hours or 80%) the temperature is below 20°C (68°F). In the other three cities it does not get nearly as cold, with most of the time being above 0°C (32°F). Miami has a moderately flat profile indicating that the temperature doesn’t vary much throughout the year. Phoenix has the hottest climate, reaching values of almost 50°C (122°F). Miami also has a warm climate ranging between 0°C and 35°C (97°F). The slope of the cumulative temperature curve in Miami is large at first and the temperature reaches 20°C quickly, but it then flattens out in the middle stretch. Another slight increase occurs at the end of the profile and the temperature reaches about 35°C (95°F). The
maximum and minimum values for temperature taken from the weather files are presented in Table 1.

![Cumulative outdoor relative humidity distributions for state college, Minneapolis, Miami and Phoenix](image)

**Figure 6.** Cumulative outdoor relative humidity distributions for state college, Minneapolis, Miami and Phoenix

The cumulative outdoor humidity ratio is shown in Figure 6 for each city. State college, Minneapolis and Miami have very similar profiles, showing many hours at the lower humidity levels and gradually increasing up to values around 50%. Phoenix, however, shows only a few hours at the higher humidity levels and many hours at lower humidity levels. The maximum and minimum humidity in each city levels can also be found in Table 1.

The cumulative outdoor enthalpy distribution for each city is shown in Figure 7. The values were calculated through the EES built-in function based on outdoor air dry bulb temperature,
relative humidity and barometric pressure (assumed to be 101KPa constantly). The equation is shown as below:

\[
h = h('AirH2O', T=T, R=\text{rh}, P=101)
\]

As shown in the figure and table, Minneapolis has the lowest enthalpies, because of its low temperatures and humidity levels. State college shows a relative small difference in its range of enthalpies, because of its mild climate. Phoenix is higher than state college, because of its warmer temperatures. Miami has the highest enthalpies, because of its high temperatures and humidity levels.

*Figure 7. Cumulative outdoor enthalpy distributions for state college, Minneapolis, Miami and Phoenix*
Table 1. Maximum and minimum temperatures, relative humidity and enthalpies for each of the four cities

<table>
<thead>
<tr>
<th>City</th>
<th>Outdoor Temperature</th>
<th>Outdoor Relative Humidity</th>
<th>Outdoor Enthalpy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Min</td>
<td>Max</td>
<td>Min</td>
</tr>
<tr>
<td>State College</td>
<td>-19.5°C</td>
<td>33.3°C</td>
<td>15%</td>
</tr>
<tr>
<td>Minneapolis</td>
<td>-31.1°C</td>
<td>37.2°C</td>
<td>19%</td>
</tr>
<tr>
<td>Miami</td>
<td>3.3°C</td>
<td>33.9°C</td>
<td>20%</td>
</tr>
<tr>
<td>Phoenix</td>
<td>-2.8°C</td>
<td>46.1°C</td>
<td>4%</td>
</tr>
</tbody>
</table>

Model description

The primary objective of the simulations is to quantify the maximum potential savings, which requires models of the performance characteristics of each component, the chiller, the cooling tower and chiller water pump and the supply air fan plus the energy input value related to lighting and other electrical equipment. Meantime, energy performance simulation was conducted to compare the difference between a system with and without energy recovery ventilator to determine whether or not the addition of an energy recovery system can improve the energy-saving. The objective function is:

\[
E_{\text{total}} = E_{\text{lighting}} + E_{\text{equipment}} + E_{\text{chiller}} + E_{\text{pump}} + E_{\text{fan}}
\]

where:
E_{Total} = total energy power density

E_{Lighting} = lighting power density input

E_{Equipment} = Equipment power density input

E_{Chiller} = chiller power density input

E_{Pump} = pump power density input

E_{Fan} = fan power density input

The first two terms are specified as follows according to ASHRAE Standard 90.1 IP:

E_{Lighting} = 1.0 \, \text{w/ft}^2

E_{Equipment} = 1.0 \, \text{w/ft}^2

To minimize the effect from the building itself on the simulation results, the zone is simplified as much as possible. The case that is used in this simulation is assumed to be an office zone has a dimension of 30 feet ×25 feet with a 12 feet high wall and 9 feet ceiling height. An overall envelop thermal transfer rate is assumed to be 0.064 Btu/h-ft²-°F according to the table 5.5-5 in ASHRAE 90.1-2007. The infiltration rate through the exterior walls is set at 0.1cfm/ft², which is based on information from Mcquiston and Spitler (1992). This infiltration occurs 24 hours a day. The ventilation rate is assumed to be 1.5 cfm/ ft². For the lighting, equipment and occupant heat gains are all assumed to be equal to 1w/ft². Also, the effectiveness of the energy wheel is assumed to be constant throughout the year while it is not true in real word. It should change as the outdoor temperature and humidity change throughout the year. For this case, the effectiveness is set at 70% and the effectiveness for the heat exchanger is assumed to be 75%.
The system schematic is presented in figure 8. As the diagram shows, one zone of a multiple zone VAV system with energy recovery ventilator was studied for this simulation analysis. For HVAC component energy consumption analysis, polynomial fits were used with generic coefficients, with the important variables being chilled water supply temperature, coil loads, chilled water flow rate, outdoor air fraction, supply airflow rate, supply air temperature and room temperature (EnergyPlus, 2010). For the simulation software, EES (Engineering Equation Solver) was selected because of its built-in high-accuracy thermodynamic and heat transfer parameters and capability for solving design problems in which the effects of one or more parameters must be determined. Equation-based simulation models are created through the EES to build the models. One of the main advantages of this approach is the ease of developing and maintaining model. Equation-based simulation models use generalized solution techniques to solve arbitrarily complex sets of differential and algebraic equation. The other advantage of this continuous simulation approach is that the system is described by the general set of equation instead of a mathematical algorithm that is only valid for a certain set of conditions. Previous research work proves that the simplicity of the models and the use of an equation solver to run the simulation ensure good robustness and full transparency (Bertagnolio, Andre and Lemort, 2010). Table 2 summarizes the model parameters. It is worth mentioning that the system efficiency is more important than the efficiency of individual components, when the energy performance of HVAV system is evaluated. And the effectiveness of the energy recovery system is assumed to be constant throughout the year while it is not true in real word. It should change as the outdoor temperature and humidity change throughout the year.
Table 2. Reference characteristics of equipment

<table>
<thead>
<tr>
<th>Components</th>
<th>Selected parameter values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Centrifugal Chillers</td>
<td>Capacity – 33hp COP - 2.75</td>
</tr>
<tr>
<td></td>
<td>Rated T_lcw - 44.01°F</td>
</tr>
<tr>
<td></td>
<td>Rated T_ecf - 84.92°F</td>
</tr>
<tr>
<td></td>
<td>Rated V_chw - 4.24ft³/min</td>
</tr>
<tr>
<td></td>
<td>Rated V_cdw - 4.87ft³/min</td>
</tr>
<tr>
<td>Variable Volume Fan</td>
<td>Flow rate - 4555.85ft³/min</td>
</tr>
<tr>
<td></td>
<td>Rated Power - 2.46hp</td>
</tr>
<tr>
<td></td>
<td>Pressure Rise - 0.087psi</td>
</tr>
<tr>
<td></td>
<td>Fan Efficiency - 70%</td>
</tr>
<tr>
<td>Variable Speed Pump</td>
<td>Flow rate - 2.54ft³/min</td>
</tr>
<tr>
<td></td>
<td>Rated Power - 0.67hp</td>
</tr>
<tr>
<td></td>
<td>Pump head - 43.51psi</td>
</tr>
<tr>
<td></td>
<td>Pump Efficiency - 66%</td>
</tr>
<tr>
<td>Energy Recovery Ventilator</td>
<td>Rated Power - 2.68hp</td>
</tr>
<tr>
<td></td>
<td>Average Efficiency - 70%</td>
</tr>
</tbody>
</table>

*T-Temperature, V-Flow Rate, lcw-leaving chilled water, ecf-entering condenser fluid, chw-chilled water, cdw-condenser water
Table 3. Description of sample case

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zone Area</td>
<td>$S = 750 \text{ ft}^2$</td>
</tr>
<tr>
<td>Overall Envelope Heat Transfer Rate</td>
<td>$UA = 0.064 \text{ Btu/h-ft}^2 \cdot ^\circ \text{F}$</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>$T_a = 90^\circ \text{F}$ (summer condition)</td>
</tr>
<tr>
<td></td>
<td>$T_a = 30^\circ \text{F}$ (winter condition)</td>
</tr>
<tr>
<td>Ambient Pressure</td>
<td>$P = 1 \text{ atm}$</td>
</tr>
<tr>
<td>Zone Relative Humidity</td>
<td>$RH = 50%$</td>
</tr>
<tr>
<td>Zone Air Setpoint Temperature</td>
<td>$T_z = 75^\circ \text{F}$ (summer condition)</td>
</tr>
<tr>
<td></td>
<td>$T_z = 72^\circ \text{F}$ (winter condition)</td>
</tr>
<tr>
<td>Outdoor Air fraction</td>
<td>$F_o = 30%$</td>
</tr>
<tr>
<td>Solar Heat Gain</td>
<td>$q_S = 1.5 \text{ w/ft}^2$ (summer condition)</td>
</tr>
<tr>
<td></td>
<td>$q_S = 0.8 \text{ w/ft}^2$ (winter condition)</td>
</tr>
<tr>
<td>Lighting Heat Gain</td>
<td>$q_l = 1.0 \text{ w/ft}^2$</td>
</tr>
<tr>
<td>Equipment Heat Gain</td>
<td>$q_e = 1.0 \text{ w/ft}^2$</td>
</tr>
<tr>
<td>Occupants Heat Gain</td>
<td>$q_o = 1.0 \text{ w/ft}^2$</td>
</tr>
<tr>
<td>Ventilation Air Flow rate</td>
<td>$M_v = 1.5 \text{ cfm/ft}^2$</td>
</tr>
<tr>
<td>Infiltration Air Flow Rate</td>
<td>$M_i = 0.1 \text{ cfm/ft}^2$</td>
</tr>
<tr>
<td>Heat Exchanger Effectiveness</td>
<td>$U_1 = 75%$</td>
</tr>
<tr>
<td>Energy Recovery Effectiveness</td>
<td>$U_2 = 70%$</td>
</tr>
</tbody>
</table>

Cooling load is defined as the sum of all loads, internal and external, sensible and latent, which are needed to be balanced from the indoor zone to keep a comfort environment. In other words, cooling load is actually the sum of heat gains transferred from outer space such as sun, occupant, equipment etc. to the zone air. As a result, there are different types of heat gains, solar, heat transmission through the walls, human, lights, ventilation and infiltration. Depending on the building characteristics, these heat gains are converted to cooling loads after some time delay. If the zone condition is to be maintained at a constant value, the chiller working load should be equal to the cooling load in theory. But in the real world, the HVAC system itself has a certain “load” due to duct friction, fan and pump losses etc. In practice, due to HVAC system control and occupants moving, the zone condition experiences a swing, i.e. a temperature fluctuation.
But what is the most important and obvious is that the ambient environment and system
parameters (chiller water temperature and flow rate) have a significant effect on the plant (chiller,
boiler, for example) performance.

\[ Q_z = q_s + q_i + q_t + q_o + q_e + q_l \]

where:

\( q_s \) = solar load

\( q_i \) = infiltration air load

\( q_t \) = envelope thermal load

\( q_o \) = occupants load

\( q_e \) = equipment load

\( q_l \) = lighting load

As shown above, the zone load is made up of solar load, lighting load, equipment load,
occupants load, infiltration air load and envelope thermal load (heat gains to zone were assumed
as positive). Since the heat gain from lighting, equipment occupants and solar was already set up,
the load values of infiltration and envelope thermal conduct can be determined from the
thermodynamic relationships as described below, the zone load can be figured out for the energy
consumption simulation.
The component energy consumption was simulated with polynomials in EES as described below:

\[
q_i = m_i \cdot c_{\text{p,air}} \cdot (T_z - T_a)
\]

\[
q_t = UA \cdot (T_z - T_a)
\]

where:

\[
Q_{\text{avail}} = Q_{\text{ref}} \times \text{ChillerCapFTemp}
\]

\[
V_{\text{water}} = \text{mass flow rate of chilled/hot water}
\]

\[
f_{\text{pl}} = \text{air part load factor}
\]

\[
m_{\text{design}} = \text{fan design flow rate}
\]

\[
P_{\text{rise}} = \text{fan pressure rise}
\]

\[
E_{\text{tot}} = \text{fan total efficiency}
\]

\[
E_{\text{chiller}} = \frac{Q_{\text{avail}} \cdot \text{ChillerEIRFTemp} \cdot \text{ChillerEIRFPLR}}{\text{COP}_{\text{ref}}}
\]

\[
E_{\text{pump}} = v_{\text{water}} \cdot \frac{\text{PumpHead}}{\text{TotalEfficiency}}
\]

\[
E_{\text{fan}} = f_{\text{pl}} \cdot m_{\text{design}} \cdot \frac{P_{\text{rise}}}{e_{\text{tot}} \cdot r_{\text{air}}}
\]

where:

\[
Q_{\text{avail}} = Q_{\text{ref}} \times \text{ChillerCapFTemp}
\]

\[
V_{\text{water}} = \text{mass flow rate of chilled/hot water}
\]

\[
f_{\text{pl}} = \text{air part load factor}
\]

\[
m_{\text{design}} = \text{fan design flow rate}
\]

\[
P_{\text{rise}} = \text{fan pressure rise}
\]

\[
E_{\text{tot}} = \text{fan total efficiency}
\]
\( P_{\text{air}} \) = density of air

In the heating situation, the fuel input was calculated with this equation (Wienese, A., 2001):

\[
F_{\text{boiler}} = m_\text{hw} \cdot c_p\text{water} \cdot \left[ \frac{T_{\text{hws}} - T_{\text{hwr}}}{BE \cdot VHI} \right] \cdot 3600
\]

where:

BE = Boiler Efficiency

VHI = Fuel Heat Value

\( m_\text{hw} \) = hot water mass flow rate

\( c_p\text{water} \) = specific heat capacity of water

\( T_{\text{hws}} \) = hot water supply temperature

\( T_{\text{hwr}} \) = hot water return temperature

The simulations are conducted on an hour-by-hour basis. The operating hours are from 9:00 to 18:00. Fan efficiency is selected as 70% as shown in table 2. For the gas boiler energy consumption, the energy consumed in the form of natural gas is converted to electricity by the unit conversion from BTU/h to KW. The heat rate of natural gas is 1000 BTU/ft\(^3\).
Some other important assumptions are listed as below:

For the cooling and heating coils, cooling/heating and dehumidification/humidification of the incoming fresh air is performed here. The temperature effectiveness in a heating or cooling is governed by the effectiveness relationship. An effectiveness of 75% is assumed as presented previously in table 3.

Fan and pump energy is an important factor in the annual energy consumption of an HVAC system. Fan (pump) performance can be characterized by its efficiency, which itself is dependent on operational air-flow rate. Mostly, rated volumetric flow rate, pressure rise and efficiency are available from the manufacturer. But for this research, these numbers are assumed as shown in table 1 with reasonable values.

Effectiveness of the main components are related to design and operating conditions. When the operating conditions fluctuate near design conditions, the effectiveness change is really small. To simplify analysis, effectiveness for various components is assumed to be constant.

HVAC components such as chiller and pumps are composed of a number of sub-components such as engine, evaporator, compressor, condenser and throttling valve, but these sub-components are not included for this study as in the energy balance equation derived for the simulation, only the inter-connections are of interest.
Figure 8. HVAC system schematic
Economizer Control Strategy

The challenge for controlling either an economizer or energy recovery system lies in ensuring the most favorable outdoor air fraction or energy transfer, on some basis. The basic economizer strategy (ASHRAE Standard 90.1-2010) employed includes the following conditions, under which the outdoor air intake damper would be opened beyond the minimum required for ventilation:

- fixed dry bulb-outdoor air dry bulb temperature less than a set value
- differential dry bulb- outdoor air dry bulb temperature less than return air temperature
- fixed enthalpy- outdoor air enthalpy less than a set value
- differential enthalpy- outdoor air enthalpy less than return air enthalpy

While the fixed and differential setpoints control logic can be simple to implement, they do not always provide for minimum energy consumption for several reasons in real word. As we know, the differential modes require at least two sensors, and the enthalpy-based strategies require the measurement of relative humidity or some other moisture content indicator, while the combination strategies require multiple temperature and enthalpy sensors. Lacking these sensors limits the control strategies that can be implemented. Even when the sensors are installed, sensor errors and other measurement problems can seriously compromise performance, particularly for enthalpy measurements, which are notoriously unreliable (Zhou et al., 2008). Several studies have illustrated the potential degradation of economizer energy performance due to sensor uncertainty (Seem et al., 2010). That said, this paper will not explore that issue any further, but
will focus on the theoretical basis for economizer control assuming properly functioning and reasonably accurate sensors.

Figure 9. Typical logic for economizer control in summer time

Region A: 100% outdoor air based on lower enthalpy for the wet coil

Region B: 100% outdoor air based on lower dry bulb temperature for the dry coil

Region C, D and E: set to minimum for outdoor air

Region F: some fraction of outdoor air may be beneficial
Note that lines of constant dry bulb temperature are nearly vertical on a psychrometric chart, increasing from left to right on the horizontal axis, while moisture content increases on the vertical axis, and lines of constant enthalpy slant diagonally downward from left to right. Two air condition points define the boundaries for optimum economizer control, one being the return air temperature \( T_r \) (which is assumed to be equal to the zone air temperature) and the supply air temperature \( T_s \) (which is assumed to be saturated). If the condition of the air entering the cooling coil is above the moisture content defined by the supply air temperature intersection with the saturation curve, condensation will occur on the coil (wet coil) while if the moisture content is below the line, the condensation will not occur (dry coil). Thus, it will be advantageous to use 100% outdoor air if the outdoor air condition lies in region A, based on lower enthalpy for the wet coil, or region B, based on lower dry bulb temperature for the dry coil. In regions D and E, the use of outdoor air is not advantageous, while in region F, some fraction of outdoor air may be beneficial. However, through the simulation, outdoor air conditions in region C, being warmer but dryer than the return air, may provide reduced cooling loads with partial economizer operation, which is shown in Figure 9 as above. More details will be provided in the later result and discussion part.

The air handling process must be clarified for the achievement of simulation. In order for the mixed air to be cooled and dehumidified to the desired supply air temperature setpoint \( T_s \), the cooling coil must have chilled water flowing through it so that its temperature is somewhat colder than the desired supply air temperature. The temperature of the cooling coil is called the apparatus dew point temperature \( T_d \), because water vapour will condense from the mixed air if its dew point temperature exceeds \( T_d \), which would be true for this example. The important point, and the reason for this example, is to emphasize that the condition of the discharge air
leaving the cooling coil must lie on the line connecting $T_m$ and $T_d$, known as the condition line, but generally not all the way to $T_d$ for real cooling coils. Physically this is because the coil is finite in size and not all of the air passing through it will be cooled uniformly, a phenomena termed bypass factor, which can range from 10 to 20% or higher. The control system will modulate the chilled water flow through the coil to maintain the dry bulb temperature of the air leaving the cooling coil at the setpoint, $T_s$, and any dehumidification will be a by-product of the sensible cooling of the mixed air. The moisture content of the air leaving the cooling coil is determined by the intersection of the condition line with the supply air temperature line. Finally, the supply air is delivered to the zone, absorbing heat (sensible load) and moisture (latent load), and under equilibrium conditions, maintaining the zone temperature at the setpoint.

The economizer for the simulation employed the differential enthalpy control strategy, one of the basic strategies from ASHRAE Standard 90.1-2010. To make this control logic simple, for the cooling condition, when the outdoor air enthalpy is less than that of return air, a 100% fresh air will be applied while for heating, a minimum outdoor air will be supplied when outdoor air enthalpy is less than that of return air. Obviously, a control strategy based only on temperature will be more simple and straightforward, but as enthalpy takes into account both the sensible and latent energy of the air, it should have a more favorable energy-saving outcome. What’s more, economizer merely based on temperature may increase the energy demand. For example, all of the following conditions have the same enthalpy but different dry bulb temperature:

74°F 50% RH Enthalpy: 28 BTU/lbm

69°F 67% RH Enthalpy: 28 BTU/lbm

64°F 88% RH Enthalpy: 28 BTU/lbm
In humid climates such as Miami, it is important to factor in the effect of humidity on the energy of the outside air. If you are trying to maintain 75°F and 50 percent humidity inside, then just because the air is 5-10°F cooler outside does not mean it will provide free cooling. With enthalpy control, if the enthalpy of the outside air is less than return air, then the economizer will increase the outside air percentage to provide cooling which is the right choice. In ASHARE 90.1, differential dry bulb control strategy is prohibited in 1a, 2a, 3a 4a, these areas all have a relative warm and humid weather.

**Simulation Process**

The setpoints were changed as described in table 4 and 5. For the summer condition simulation, five parameters, condenser entering temperature, chilled water supply temperature, chilled water mass flow rate, supply air temperature and flow rate are set as variables. Ten different values are selected for each parameter so there are 50 different scenarios in total. As only hot water supply temperature and mass flow rate, supply air temperature and flow rate were changed in the winter condition, 40 group of total power density resulted from the simulation. But here as the whole year total energy consumption is the object of study, the summer cooling and winter heating will be simulated simultaneously with a condition judgment statement coded in EES. When outdoor air temperature is greater than 80 °F, the cooling will be on and when outdoor air temperature is less than 55 °F, the heating will be simulated.

One thing to note is that
**Table 4. Case description for the summer and winter condition**

<table>
<thead>
<tr>
<th>Cases (summer)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (group 1-10)</td>
<td>Increase condenser entering temperature (50-60 °F)</td>
</tr>
<tr>
<td>2 (group11-20)</td>
<td>Increase chilled water supply flow rate (0.4-0.7 lb./s)</td>
</tr>
<tr>
<td>3 (group 21-30)</td>
<td>Increase chilled water supply temperature (45-55 °F)</td>
</tr>
<tr>
<td>4 (group 31-40)</td>
<td>Increase supply air flow rate (500-700 cfm)</td>
</tr>
<tr>
<td>5 (group 41-50)</td>
<td>Increase supply air temperature (60-65 °F)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Cases (winter)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (group 1-10)</td>
<td>Increase hot water supply temperature (185-195 °F)</td>
</tr>
<tr>
<td>2 (group11-20)</td>
<td>Increase hot water supply flow rate (0.4-0.7 lb./s)</td>
</tr>
<tr>
<td>3 (group 21-30)</td>
<td>Increase supply air flow rate (500-700 cfm)</td>
</tr>
<tr>
<td>4 (group 31-40)</td>
<td>Increase supply air temperature (85-95 °F)</td>
</tr>
</tbody>
</table>
IV. Results and Discussion

The simulation figure depicts the total power density as a function of different setpoint settings. The case is default with differential enthalpy economizer but without ERV system. The total power density consumed in each city is shown in the Figure below. Each city stands for a typical weather conditions.

Energy Saving Prediction

Figure 10, 11, 12 and 13 illustrate the annual power density for four different weather cases from largest to the smallest for the year around. The different colors present the breakdown of the electricity usage. As we can see, HVAC system (including chiller, cooling tower pump, chiller water pump and supply air fan) is the biggest electric consumer in the model, which accounts for around 60% of total energy consumption, while both lighting and equipment account for around 15% of the total power consumption, respectively. According to the table 5, the maximum annual power density can reach 36.121kwh/sf-year at Miami when the chilled/hot water flow rate at the biggest value and a small supply air flow rate can decrease the energy consumption to 26.712kwh/sf-year at state college.
Figure 10. Annual power density in Minneapolis

Figure 11. Annual power density in Phoenix
Figure 12. Annual power density in Miami

Figure 13. Annual power density in State College
As the figures and table shown, Miami has the largest annual power density while state college
has the smallest. The reason is for a hot and humid weather, although heating is not necessary
for most of time, cooling and dehumidifying are required for almost all the year round. While
the mild weather in state college provides a number of opportunities for free cooling. The
energy consumptions at Phoenix and Minneapolis are similar but Phoenix still needs a little more
energy than Minneapolis for humidifying. In general, the annual difference is not very big due to
the limitations of the simplicity of models and obviousness of variation was diminished by the
rise and fall in energy use occurred in the whole year.

Mentioning the potential energy saving, to make comparisons, the annual energy maximum
value and the minimum value for the four proposed cases are computed and listed in Table 5. As
can be seen in the table, saving potential value for Minneapolis is the largest. This is due to the
reason that the chiller and boiler saving potential here is the biggest while chiller takes the largest
percentage of the total energy consumption. The biggest saving percentage occurs under cold
and dry weather situation which value can be as high as 8.55%.

Figure 15 is the building energy usage breakdown. The percentage of the total power that is
required by HVAC system to ventilate and condition (fans, pumps, chillers and boilers) is 67%
in Minneapolis, 71% in Miami, 68% in Phoenix and 63% in state college. Among the HVAC
system itself, chiller and boiler is the largest power consumer while the pump consumes least
energy.
### Table 5. Comparison of power density with different setpoints

<table>
<thead>
<tr>
<th>City</th>
<th>Annual Power Density maximum (Kwh/sf-year)</th>
<th>Annual Power Density minimum (Kwh/sf-year)</th>
<th>Potential Energy Saving (Kwh/sf-year)</th>
<th>Saving Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minneapolis</td>
<td>34.689</td>
<td>31.723</td>
<td>2.966</td>
<td>8.55%</td>
</tr>
<tr>
<td>Phoenix</td>
<td>33.423</td>
<td>31.645</td>
<td>1.778</td>
<td>5.32%</td>
</tr>
<tr>
<td>Miami</td>
<td>36.121</td>
<td>35.340</td>
<td>0.781</td>
<td>2.16%</td>
</tr>
<tr>
<td>State College</td>
<td>28.644</td>
<td>26.712</td>
<td>1.932</td>
<td>6.75%</td>
</tr>
</tbody>
</table>

To gain a deeper understanding of the simulation results, plots were generated showing the cumulative percent of the total data population described at each yearly energy density value, working from smallest to largest. The distribution of the data is close to a normal cumulative distribution which agrees with previous assumption that most total energy consumption for buildings has a normal distribution. It can be seen that the considered cases show a significant energy usage difference between the best and worst cases. Thus, there is a significant savings potential, which can potentially be exploited by adjusting the HVAC system setpoints.

The second thing should be noticed is that the standard deviation and mean numbers for these four conditions. The mean number stands for the average energy usage during a whole year. So based on the results, Miami (hot and humid weather) has the largest average power density while
state college (mild weather) has the smallest one. The difference on the running time of economizers may contribute to such an outcome.

In statistics and probability theory, the standard deviation shows how much variation or dispersion from the average exists. A low standard deviation indicates that the data points tend to be very close to the mean; a high standard deviation indicates that the data points are spread out over a large range of values. So for here, a larger standard deviation means the energy usage is relatively unstable when changing the setpoints, but at the same time it also indicates a larger saving potential. Minneapolis has the biggest standard deviation which is consistent with the results in table 6. And Miami has a relatively stable data so in the hot and humid weather condition, changing system setpoints will not bring a significant fluctuation in the energy usage.
Figure 14. Cumulative distribution for the annual power density in four different climates: (a) cool and dry, (b) hot and dry, (c) hot and humid and (d) mild weather condition.
To sum up, the power saving potential in HVAC sector is pretty huge. To obtain an estimate of an office building with 50,000ft² floor areas, assuming that the average cost saving due to optimization building and HVAC design are 8% according to the optimum results, and the

**Figure 15.** Distribution of annual power density for the no energy recovery system in Minneapolis, Miami, Phoenix and State College

To sum up, the power saving potential in HVAC sector is pretty huge. To obtain an estimate of an office building with 50,000ft² floor areas, assuming that the average cost saving due to optimization building and HVAC design are 8% according to the optimum results, and the
average energy cost is $0.1 per kWh, and the annual energy consumption without setpoint optimization is 30kWh/ft², then the saving is $12,000 per year.

Sensitivity Analysis

Sensitivity Analysis (SA) is defined as the study of how uncertainty in the output of a model (numerical or otherwise) can be apportioned to different sources of uncertainty in the model input (Saltelli et al, 2000). To evaluate the effects of these key parameters on the energy performance in different climate conditions, sensitivity analysis was carried out, which provides a good opportunity of giving a hierarchical rating to a large number of energy model inputs based on their relative importance to building energy consumption.

When mentioned the method of sensitivity analysis, Lomas and Eppel (1992) documents three SA Techniques-Differential Sensitivity Analysis (DSA), Monte Carlo Analysis (MCA) and Stochastic Sensitivity Analysis (SSA), the DSA is most commonly used due to its simplicity and easy-to-understand. For this research, the DSA method is picked to assess the relative influences of selected inputs on the energy consumption.

The sensitivity coefficient presented below is defined as the percentage change of the output divided by the percentage change of the input. Figure 16 provides a more vivid picture of the proposed procedure.

![Diagram](image)

**Figure 16.** Input – Output analysis of simulation system
The equations used for the sensitivity analysis are shown as below:

\[
\text{SensitivityCoefficient} (SC) = \left| \frac{\Delta O}{\Delta I} \right|
\]

where:

\[
\Delta O = \frac{O_{\text{pert}} - O_{\text{base}}}{O_{\text{base}}}
\]

\[
\Delta I = \frac{I_{\text{pert}} - I_{\text{base}}}{I_{\text{base}} - I_{\text{min}}}
\]

These two terms are the changes of the output and input relative to the base model and input, respectively. \(O_{\text{base}}\) and \(I_{\text{base}}\) are the base model output and input, respectively, \(O_{\text{pert}}\), \(I_{\text{pert}}\) and \(I_{\text{min}}\) are the perturbed model output, input and potential minimum value of input, respectively.

In this paper, the interested simulation outputs include whole building annual electricity and gas energy uses, as well as the chiller, the pump and the fan energy uses. So to sum up the winter and summer conditions, there are five interested input variables, they are condenser entering temperature, chilled/hot water supply temperature, chilled/hot water supply flow rate, supply air flow rate and supply air temperature. Then, the range of each parameter was determined according to the actual building operation situation. The range is shown in table 4 and table 5. Perturb one parameter at a time while keeping other parameters constant, the sensitivity coefficient can be calculated based on the simulated output.
So based on the sensitivity-analysis method described above, and the simulation results for these four different climates, the variables in descending order of their relative importance to the annual whole building energy use for the four different climates is shown in Table 7.

**Table 5.** Sensitivity analysis results for (a) mild, (b) cool and dry, (c) hot and dry and (d) hot and humid weather condition

(a)

<table>
<thead>
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<th>Parameters (P)</th>
<th>Sensitivity Coefficients (SC)</th>
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<td>chilled/hot water supply flow</td>
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<td>chilled/hot water supply temperature</td>
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(b)

<table>
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<th>Parameters (P)</th>
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According to the data shown above, with regards to the mild weather, the supply air flow rate has the largest sensitivity coefficient, which means minimizing the supply air flow rate is shown to be the most effective measure to save the energy usage. On the opposite, chilled/hot water temperature has the smallest value, which indicates the variation on the chilled/hot water temperature settings will have the least influence on the power consumption. While for the rest three outdoor conditions, the supply air setpoint temperature is in the driving position.
The results suggest that control strategies that are capable of dynamically adjusting setpoints in response to environmental and occupant conditions can potentially save a substantial amount of energy as compared to fixed setpoints.

**Energy Recovery and Economizer**

![Annual power density comparison between system with and without ERV](image)

**Figure 17.** Annual power density comparison between system with and with ERV

The comparisons of the different systems are plotted in figure 17. It is shown that among the four locations, system with ERV in state college consumes the least energy, and system with ERV in Miami consumes the most. The most significant change happens in Minneapolis where the energy consumption decreases by 14.6% with the addition of the ERV. The difference in the
power density between the two cases for state college is really minimal. Small change is expected here as it is mild weather and the energy captured by the ERV system is relatively limited. Miami and Minneapolis both experienced similar decrease (around 10%) with the ERV. This reduction results because the decrease in the energy for chiller, pump and fan is larger than the energy use by the fan in the ERV.

Another finding through the simulation is a potential improved control setpoint for economizer. When economizer operates in hot and dry weather environment like Phoenix, based on conventional control method, a minimum outdoor air is recommended. If the outdoor air is assumed at 30 Celsius, and the return air is set at 26 Celsius for the summer cooling, outdoor humidity and supply air humidity are assumed at 20% and 60% respectively. As the cooling coil needs to decrease the mix air enthalpy to meet the requirement of supply air, the enthalpy difference can be used to predict the power consumed by the HVAC system. Through simulation results as shown in figure 18, however, the least enthalpy difference between supply air and mix air into cooling coil occurs at a certain fraction on the line connecting the return air state point and outdoor air point. In other word, the minimum coil load will occur if the mixed air fraction is such that moisture content of the air is equal to the moisture content of saturated air, that is, if the cooling coil is at the borderline between wet and dry conditions (Figure 19). Under these conditions, the coil only removes the sensible heat load, and the drier outdoor air precisely balances the room latent load. The condition line is horizontal in this case, and the increase in sensible cooling load due to the introduction of outdoor air is more than compensated by the reduction in latent load.
Figure 18. Simulation result shows the mixed air point in Figure 3 is the optimum mix air point.

$\Delta H =$ enthalpy of mix air - enthalpy of supply air
V. Conclusion

In this paper, the engineering equation solver computer program was used to perform simulations on a conditioned zone with various collections of setpoints in four different cities standing for four different outdoor environments. Then simulations were performed with and without energy recovery system to determine the effect of this kind equipment on the building energy use. At last, a method was presented and demonstrated for determining the optimum operating strategy for economizer in certain outside weather condition.
The analysis reveals that a large potential of energy reduction exists in the building. Whole building energy saving from fine tuning HVAC system can reach as high as around 9 percentages of annual heating and cooling power costs in cold and dry weather area. Different control system setpoints provide different degree of energy savings. Minimizing the supply air flow rate is shown to be the most effective measure to save power usage under mild weather while better selected supply air temperature will be the most important factor, while a large chilled/hot water flow rate is also should be taken careful consideration. The results suggest that control strategies that are capable of dynamically adjusting setpoints in response to environmental and occupant conditions can potentially save a substantial amount of energy as compared to fixed setpoints.

The addition of ERV into the HVAC system of a building is shown to have a large impact on the energy in a dry climate, such as Minneapolis and Phoenix. In a very humid weather, such as Miami, there is also a reduction in the power consumption, but not so large as in the relatively dry weather. For a mild climate, such as state college, there is no significant change. In summary, an energy recovery system can help improve the building energy performance in dry and humid climates in an obvious scale while not so obviously in a mild climate. So there should be a careful tradeoff, when making decisions about whether adopting such a kind system or not.

This simulation method also enables the determination of the best outdoor air fraction for economizer to minimize the cooling coil load in hot and dry climate. The method differs from the more typical approaches which are based on either a comparison of return air and outdoor air dry bulb temperatures or enthalpies. Furthermore, the use of partial outdoor air fractions to
maintain a dry coil expands the useful application of economizer cycle and energy recovery for warm, dry outdoor air conditions.

VI. **Recommendations for Future Work**

The prediction of building energy consumption has led to a surge of attention, however, there are still some investigations needed. For the future work, the following recommendations are made for future work:

- The number of setpoints studied is limited and the more detailed model could be studied.
- Latent load should be considered in the future work.
- The results should be conducted with cross comparison with other software output.
- More comprehensive climate regions should be further extended and investigated.
- Sensitivity analysis might consider the simultaneous variation of parameters and interaction term.
- The energy recovery system efficiency could include the effects from outdoor air temperature and humidity. Perform an energy consumption simulation with variable effectiveness values of energy recovery system to see the effect of changing effectiveness due to outdoor temperature and humidity on the result.
References


