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ANALYSIS OF ENERGY SAVINGS THROUGH THE USE OF DISPLACEMENT VENTILATION IN DOMESTIC BUILDINGS

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ANALYSIS OF ENERGY SAVINGS THROUGH THE USE OF DISPLACEMENT VENTILATION IN DOMESTIC BUILDINGS

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ABSTRACT

This study investigates the conditions under which it might be possible to implement a displacement ventilation system in a residential building. An experimental study of the impact on a mechanical air conditioning system of the vertical location of the inlet and outlet vents was performed. The four ventilation configurations of low inlet high outlet, low inlet low outlet, high inlet high outlet, and high inlet low outlet were compared. These four configurations were compared under 13 different heat load scenarios in a full scale instrumented model room. It was found that, for higher heat loads, the low inlet high outlet configuration was able to maintain approximately the same temperature in the occupied region as for a lower heat load, while developing a strong two layer stratification within the room such that the outlet temperature was significantly higher than the ambient temperature in the lower occupied region of the room. This was achieved because this ventilation configuration was able to stratify the temperature within the room and force the heat into the upper unoccupied region. From this zone the outlet was able to more efficiently remove the unwanted heat. The increased outlet temperature means that the inlet temperature can be closer to the temperature required for thermal comfort meaning that less pre-cooling of air is required. The results show that, even with only a 2.5 meter ceiling height, comparable to most residential applications, the displacement ventilation configuration was able to

reduce the need for mechanical conditioning. This would have a noticeable impact on the energy requirements of a residential building.

DEDICATION

I would like to dedicate this thesis to my advisory committee. Throughout this process their help and advice were invaluable. Dr. Nigel Kaye helped me from the beginning, putting this project together and putting the research team together. I thank him for all of his help and support.

Dr. Vincent Blouin was pulled in from another department to be my advisor for this project. His knowledge of Computational Fluid Dynamics and of the benefits to indoor air quality was invaluable to this projects success.

Dr. Leidy Klotz was my first point of contact for graduate school and directed me to Dr. Kaye. His direction and guidance during my undergraduate has shaped the way I view construction and sustainability. His view of the construction industry will forever shape my professional career.

To all three of my advisors I offer my thanks and appreciation. This is for all of the time and energy that you have invested in this project and in me as a person and a professional.

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CHPTER 1. LITERATURE REVIEW

1. General Overview

A significant percentage of the energy use in the United States is consumed through the use of buildings. In 2011, the percentage of the total energy used for commercial and residential buildings was 18% and 22% respectively (U.S. Energy Information Administration, 2011). These significant components of the overall energy use have driven substantial research for the reduction of demand within the built world. Heating, ventilation, and air conditioning, HVAC, have taken a substantial component of this research due to it being one of the most significant consumers of energy within buildings. HVAC consumes 20% of all energy within a commercial building and 40% of the energy in a residential building (Levine et al., 2007).

Much of the research on HVAC design has focused on comparing various ventilation strategies. These include natural, mixed, displacement, impinging jet ventilation, and others. Natural ventilation involves the design of the building requiring no mechanical systems. When correctly designed these systems utilize wind and/or thermal energy to provide a comfortable occupancy air quality (Linden, 1999). When used in the appropriate environment, these systems can provide sufficient air quality with little to no energy required. To increase the versatility of natural ventilation, some researchers like Gladstone have looked into driving the natural flow with inputs like heated floors or cooled ceilings (Gladstone & Woods, 2001). These additions have shown an ability to reduce energy use while maintaining sufficient indoor air quality, IAQ.

A significant amount of research has focused on comparing two of the major mechanical ventilation strategies, mixed and displacement. Mixed ventilation, MV, involves the air being forced into the conditioned space with a sufficient velocity to thoroughly mix the air within the space. This produces a constant ambient temperature throughout the space. This also results in the contaminant being evenly dispersed throughout the space. In contrast, displacement ventilation, DV, requires the input to the room to be let in with a relatively slow velocity filling the room from a low elevation. The conditioned air fills from the bottom and slowly flows upward. The resulting stratification of the air temperature drives the pollutants to the top layer of the air, which is then extracted through the outlet near the top of the room.

Other ventilation strategies have been studied for various applications. Varodompun summarized the various systems (Varodompun & Navvab, 2007). Some of these systems can be seen in Figure 1-1 and Figure 1-2. Some systems incorporate distributed supply such as underfloor air distribution systems. Others strategically locate ventilation inlets directed at the occupant to only condition the general vicinity of the occupant, cutting down energy use. Others still vary the way in which the given system is executed. For example, there are numerous ways to produce a mixed ventilated space. Impinging jet ventilation combines the benefits of mixed ventilation with some of the benefits of displacement ventilation.

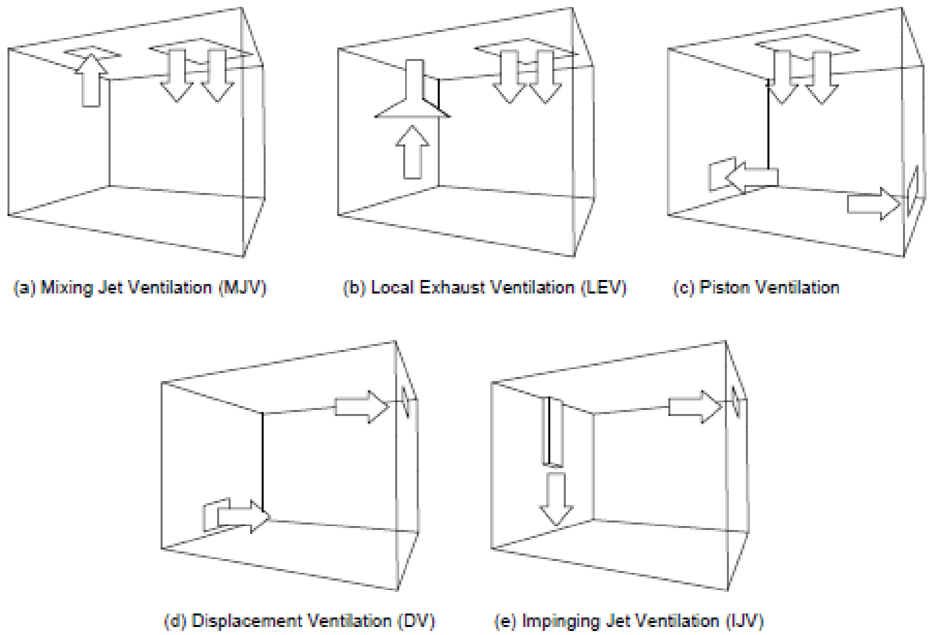


Figure 1-1: Examples of ventilation types.

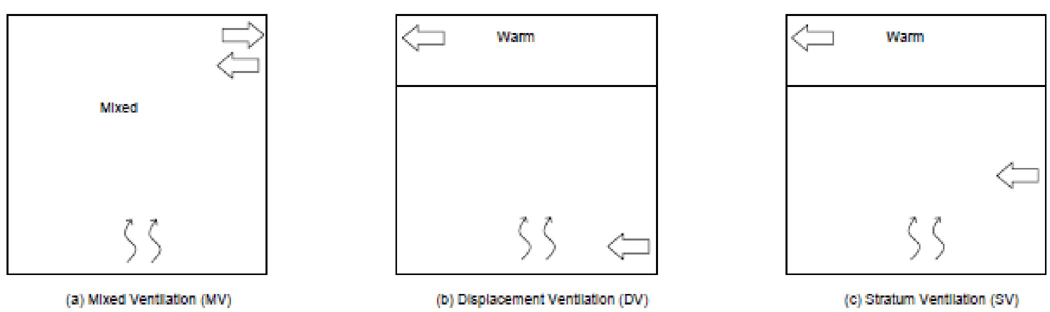


Figure 1-2: Example of mixed ventilation compared with displacement ventilation.

Each of these types of ventilation has been studied from different perspectives. Chen thoroughly describes the benefits and drawbacks to a few different research strategies (Q. Chen, 2009). Analytical and empirical models combine the conservation of mass and energy to predict the behavior of the fluid. Experiments attempt to simulate the actual situation either in a full-scale or small-scale replica. The resultant situation is then observed and data is collected. Also, computer simulations can be used; these combine various equations to simulate the situation in Computational Fluid Dynamics (CFD) models. Full-scale experimentation can produce the best results, but it is expensive and time consuming. So frequently, the experimentation is used to validate a CFD model, which can be run at a much lower cost and time expense. Also, small-scale experiments can be run at a reduced cost, but the accuracy of results can again be questionable. Empirical and analytical models are also useful tools but they require assumptions which have innate error.

Another issue that researchers have encountered is the substantial variability in the simulations. The potential for different furniture locations and different occupancy demands can substantially affect the success of a system. Also the location and type of contaminants will affect which system has the best resultant indoor air quality, IAQ. Causone (2010) investigates some of these variations by combining floor heating and cooling with DV. He finds some of the limitations that must be considered in the design of these systems, such as the increased vertical air temperature difference which can lead to occupancy discomfort. Another consideration that is typically ignored by other studies is the effects from the radiation off walls and heat sources investigated by Chow (Chow & Holdo, 2010). In order for any researcher to reduce the problem down to a manageable size, considerations like the heat transfer by radiation are often ignored. With each study another step is taken toward a clear understanding of these systems.

2. Displacement vs. Mixed Ventilation

Displacement and mixed ventilation receive a substantial amount of interest when it comes to HVAC research. Mixed ventilation is the standard practice in most HVAC designs, and displacement ventilation is one of the most used alternatives to mixed ventilation, especially in Scandinavian countries (Lee & Lam, 2007; Serra & Semiao, 2009). This has led to extensive comparison between the two systems. It has been generally concluded that DV is much more efficient in cooling a room and MV is more efficient in the heating phase (Varodompun & Navvab, 2007).

Mixed ventilation, MV, allows for the inlet and outlet to be located at the most accessible location from the heating and cooling systems. This reduces the expense of ductwork and the length that the ventilated air must be forced through. A mixed system also evenly distributes a contaminant within the space. This significantly reduces the contaminant level near the source. This requires a difference in the inlet temperature and velocity from the ambient conditions to fully mix the air in the room and produce a uniform temperature within the room. Figure 1-1 (a) depicts what mixed ventilation looks like, see mixed jet ventilation.

Displacement ventilation, DV, utilizes the effects of buoyancy to move the warm waste air up through the space to ceiling level outlets. This allows the inlet air to be nearer the ambient temperature and for the flow to be at a relatively low velocity. The inlet and outlet locations can be seen in Figure 1-1. Stratification is created in the air. A lower level develops with a cooler temperature and a lower contaminant level. The top layer has a higher temperature and contaminant level. The stratification of the air can be seen in Figure 1-2. The temperature difference causes any pollutant to be driven into the top layer. The outlet then extracts the hotter more polluted air.

Varodompun compared MV with DV using a full-scale model to validate a Computational Fluid Dynamics, CFD, simulation, which was then used to compare the two systems. It was found that DV can perform significantly better than MV under certain metrics, like ventilation effectiveness (VEF). However, metrics that take stratification discomfort into consideration can make DV less appealing. This study suggests the use of impinging jet ventilation(IJV). This study finds that IJV allows for the benefits of the DV without some of the drawbacks. This system can be seen in Figure 1-1. The difference in velocity and direction from DV allows the IJV to be much more effective in the heating phase than DV (Varodompun & Navvab, 2007).

Lin also looks at the variation in IAQ and thermal comfort between floor-supply DV and standard MV. CFD simulations were used to analyze an example classroom, office, retail space, and industrial workshop. The simulations were run to represent the external ambient environment of Hong Kong. To evaluate the effectiveness of each system the percent persons dissatisfied (PPD), the percentage of dissatisfied people due to draft (PD), and the temperature distribution were all found and compared. It was found in all cases that the DV did have an acceptable temperature difference between toe and head level. Also the PD was found to be 10% and the PPD was calculated at less than 20%. It was also found that the MV had adverse effects on the comfort of the occupants, due to the high velocity of the inlet. All of which seems to support the use of DV to reduce energy consumption and improve IAQ (Zhang Lin, Chow, Fong, Qiuwang Wang, & Ying Li, 2005).

Lau took the comparison to the environmental conditions in the United States, using the environmental conditions of five major US cities. Here the focus was on the energy savings and determining where DV would reduce energy use and where it may not. This study identified DV's reduced ability to dehumidify the air in the heating phase, furthering the support for not heating with DV. This study also concluded here that in a humid environment, like South Carolina, DV is not as much an energy saver with the additional energy load of dehumidifying the air. The reduced energy demand of the chiller is offset but the increased demand from the fan and boiler. This contradicts other studies that found up to a 34% energy savings for DV over MV (Lau & Chen, 2006).

Causone looks at the addition of floor heating and cooling to DV systems to improve their effectiveness. A full-scale room was constructed to run the experiments. The heated floor was found to maintain high values of ventilation effectiveness. It was also found that the only contaminant that evaded the ventilation were contaminants located on cold vertical surfaces. This needs to be considered early in the design to avoid problem contaminants. The floor cooling was found to produce a temperature difference between the foot and head outside the comfort standard. This is a result of the cooled floor reducing the effects of convection between the warm ceiling and the floor (Causone, Baldin, Olesen, & Corgnati, 2010).

Lin looked at the energy use of DV compared to MV. An office, classroom, and retail space were all analyzed to find the resultant savings. The systems were analyzed to find the total energy consumption in each situation for a typical year conditioning the space in Hong Kong. TRNSYS was used to calculate the energy consumption. This study found that DV had a 19% savings over MV and that a third alternative, stratum ventilation had an additional 25% savings over DV. The stratum system supplies the air at the breathing level and extracts high on the wall. The stratum ventilation can be seen in Figure 1-2 (c) (Z. Lin et al., 2011).

Serra compared DV and MV through a CFD model. The environmental conditions were modeled to match a Mediterranean climate with both heating and cooling considered. The simulations used a small office with a desk and one occupant to determine the efficiency of each system. The simulations found that the DV provided improved ventilation efficiency with a reduction in energy use. Independently the simulations found that the DV performed much more efficiently in the cooling phase than the MV. However, in the heating phase, the DV resulted in a flow short-circuit. This is when the room is not thoroughly ventilated and there is a direct stream from the inlet to the outlet, resulting in unconditioned regions (Serra & Semiao, 2009).

Lee investigated the effects of a single plume rise in a displacement ventilated space. The environment of Hong Kong was considered in a CFD model. The simulations concluded that the major factors were the ceiling height, the design temperature, and reference room temperature. A lower ceiling height and/or lower design room temperature a higher velocity is needed at the inlet, reducing the benefit of the system (Lee & Lam, 2007).

Each of these cases shows potential for variation in results considering where you set your scope and what factors you consider. Also, how you incorporate those factors can greatly vary the result. Consider the variation in results between Lin and Lau. Lau found DV to have no substantial energy savings when the humidity of the air was considered (Lau, 2006). However, Lin found a savings of around 20% between the two systems (Lin, 2011). Determining the exact cause of this variation is difficult with one considering the climate of Hong Kong and the other is looking at US conditions. It is also not clear if Lin considered the humidity treatment that Lau included. This makes it difficult to conclude which system has the most efficient treatment of the air. Each case is different and the designer must determine the best system for the building in question.

3. Commercial vs. Residential Applications

DV has been adopted into some commercial applications, particularly in the Scandinavian region of Europe. This is reflected in the research. Many studies have been done on retail, office, and other commercial applications. These studies look at the costs and benefits of DV over MV. However, the literature seems to lack research into residential applications for DV.

Lin looked at the application of DV on an office, classroom, retail shop, and industrial workshop in two different papers discussed above (Z. Lin et al., 2011; Zhang Lin et al., 2005). These two studies ignored the potential residential application, which represents a larger component of the national energy use, at least in the US. Also discussed above, Lau investigated the application of DV to industrial workshops (Lau & Chen, 2006). A study by Nahor also investigated the commercial applications for DV using CFD (Nahor, Hoang, Verboven, Baelmans, & Nicolai, 2005).

There are a few articles that focus on the residential application of DV. The first is the paper by Varodompun discussed above (Varodompun & Navvab, 2007). The other two are a pair from Gao. Both use the same full-scale experimental set up to validate CFD models. They use the environmental conditions standard for Hong Kong and the simulations are run on a bedroom layout. The first studies the benefit of floor-based air-conditioning systems (FAC) over the more standard ceiling-based system (CAC). An energy savings of about 7% is found with this simple change (Gao, Lee, & Hua, 2009). The second study looked at the variations within the FAC. An angle of deflection 45° was found to be more efficient than perpendicular to the wall. Also, a mount height of 1.1m was found to be preferable to .6m. Though these systems are not the above discussed DV and MV systems they do display a similar variation in location of inlet and outlet. Also they show general agreement that there is a benefit to cooling from a lower elevation than what is currently considered standard practice.

The extent of research devoted specifically to the residential application of DV is inadequate. The research that is in place points to DV being able to reduce the energy demand of the HVAC system. It has also led to a commercial application of DV. Further DV research with a residential focus could find a practical application of DV for the residence.

4. Locational Application

Another major consideration that permeates the research is the variation between different geographical locations. DV may be the best system available for one region of the world, but could be the worst option in another. There has been extensive investigation into the effectiveness of DV in the Hong Kong region. The general consensus found that DV worked well in the climate conditions of Hong Kong (Gao & Lee, 2009; Gao et al., 2009; Z. Lin, Chow, Tsang, Fong, & Chan, 2005; Z. Lin et al., 2011; Zhang Lin et al., 2005). Another study, by Serra, looked at applications in the Mediterranean (Serra & Semiao, 2009). But the research applied directly to the United States is relatively limited. The above discussed studies by Lau and another by

Varodompun look specifically at applications in the United States (Lau & Chen, 2006; Varodompun & Navvab, 2007). However, the understanding of DV systems in the United States is also limited by the infrequency that these systems are used in the US. Therefore, further study looking at the performance of DV in the US is needed.

5. Thermal Comfort and Indoor Air Quality

Besides energy consumption, indoor air quality (IAQ) is the other major parameter that is frequently investigated in the literature. Each researcher uses a different metric to determine the success of the ventilation strategy and each metric considers different components of a comfortable indoor environment. Some of the main considerations to the thermal comfort that can be measured are the temperature (Zhang Lin et al., 2005), the relative humidity (Varodompun & Navvab, 2007), the concentration of pollutants (Z. Lin et al., 2005) and the air velocity at different locations in the room. Also, the mean age of the air can be calculated using equations similar to those used by Lin (Lin,Z. 2005). From these and other measured values, metrics can be calculated. Each metric attempts to create a value that can be universally compared between indoor environments. Each study uses different metrics that they have created or adopted from other studies. Lin uses the predicted percentage dissatisfaction, PPD, which calculates the expected percentage of occupants that would be dissatisfied with the indoor condition. Gan also used the PPD in their study (G. H. Gan, 1995). Lin additionally used the percentage dissatisfied due to draft, PD (Z. Lin et al., 2005; Zhang

Lin et al., 2005). Varodompun used the ventilation effectiveness (Varodompun & Navvab, 2007). Gao used the Draft Risk, which was also used by Causone (Causone et al., 2010; Gao & Lee, 2009; Gao et al., 2009). Gao uses the air diffusion performance index, ADPI, as an additional metric to compare systems with each other. The ADPI is a standard that evaluates the effectiveness of the diffusion of air. This value is based on the air speed and temperature. Ng uses the ADPI to find the ideal supply temperature (Ng, Kadirgama, & Ng, 2008). Another metric is the intake fraction, IF, this was used in the study by Russo. The IF evaluates the concentration of pollutants in the air.

The use of different metrics means the results are not easily comparable between studies. However, when almost every metric comes to the same conclusion regardless of what is considered, the validity of the conclusion is generally supported. Every study found some level of benefit to cooling with DV, however the extent of the benefit was not always agreed upon.

6. Small-Scale, Full-Scale, or CFD

Each study also investigates their specific problem with different simulation methods. One method is called small-scale experimentation. This method uses a small scale replica of the subject space to reduce cost and waste. Many times a saline solution is substituted for air to better view the movement and conditions within the experiment. This was used by Hunt to predict the time needed to flush a pollutant from an enclosure (Hunt & Kaye, 2006). This was also the method used by Gladstone to simulate a displacement ventilated space with a heated floor (Gladstone & Woods, 2001). Chenvidyaharn also used a small-scale simulation to study the height of the interface between warm upper air and cool lower air under different conditions. It was found that increasing the supply flow rate or the strength of the heated floor will raise the interface and affect the temperature of both zones (Chenvidyakarn & Woods, 2008). This was also the method of choice for Linden, Walker, and Sandbach in their respective studies (Linden, 1999; Sandbach & Lane-Serff, 2011a; Walker, Tan, & Glicksman, 2011).

The alternative to small-scale is a full-scale experiment. These take many different forms. Some use fully constructed buildings and others may build a sample space to run the experimentation. The full scale method allows for the observer to see exactly what is occurring in that case. There is no need to calculate and decipher what the data means for the real application; it is the real application. This method was used frequently in the literature (Kobayashi & Chen, 2003; Lau & Chen, 2007; Novoselac, Burley, & Srebric, 2006; Rees, McGuirk, & Haves, 2001; Sandbach & Lane-Serff, 2011b). It does however have the drawback that it is harder to control the external ambient conditions.

The newest of these methods is Computational Fluid Dynamics (CFD). This uses computer software to run an experiment in significantly less time with significantly less cost. These studies require some calibration and the results do not always exactly replicate the experimental results. This is due to numerous different methods to calculate results within the software. Most of these types of studies seem to use some experimental results to confirm the accuracy of the CFD values that are being produced. Once the CFD model has been verified, then additional simulations can be run with a higher confidence in the results. This has become the most common method of study within this field over the past decade (Angioletti, Di Tommaso, Nino, & Ruocco, 2003; Bolster & Linden, 2007; Causone, Olesen, & Corgnati, 2010; Cehlin & Moshfegh, 2010; Cho, Awbi, & Karimipanah, 2008; Chow & Holdo, 2010; Deevy, Sinai, Everitt, Voigt, & Gobeau, 2008; Gladstone & Woods, 2001; He, Yang, & Srebric, 2005; Holford & Woods, 2007; Karimipanah & Awbi, 2002; Kaye, Ji, & Cook, 2009; Rohdin & Moshfegh, 2011; Russo & Khalifa, 2010; Varodompun & Navvab, 2007; Wu, Wu, Feng, & Zhang, 2007; Xing, Hatton, & Awbi, 2001; Xu, Yang, Yang, & Srebric, 2009; Zhang, Lee, & Chen, 2009; Zhong, Kang, & Wang, 2008).

7. Computational Fluid Dynamics Settings

There are numerous settings within any CFD program that can significantly change the accuracy of the results. The calculations can be run with different algorithms, but one of the frequently used is the SIMPLE algorithm (G. H. Gan, 1995; G. H. Gan, 1995; G. Gan, 1998; Gao & Lee, 2009; Gao et al., 2009; Park & Holland, 2001). Another setting that is frequently discussed is the turbulence model. The standard practice used to be the standard k-e (SKE) and it is still used in many studies. However, frequent studies have focused on finding another method that could improve the accuracy of the simulations. Rohdin did a study to compare the three variations of k-e, SKE, RNG, and RKE. It was found that RNG, renormalized group k-e, was the most in agreement with the measured results (Rohdin & Moshfegh, 2011). These findings are also supported by numerous other studies (G. Y. Chen & Xu, 1998; Q. Chen, 2009; G. Gan, 1998). These findings have led to frequent use of this model instead of the SKE (Gao & Lee, 2009; Ji & Cook, 2007; Ji et al., 2007; Ji, Cook, & Hanby, 2007; Z. Lin et al., 2011; Wang & Zhao, 2006).

8. Open questions

The literature is unclear on the question of whether DV or MV is energy efficient for residential buildings. The answer to this question is complex as it must account for geographical location, surrounding terrain, building construction type, air quality standards, and the measures used to quantify performance (PPD, PD, ADPI, etc.). However, there is a broad consensus that DV has some energy saving benefit compared to MV. Therefore, it would be useful to understand the circumstances under which a DV flow can be established. That is, for what ventilation configuration, heat load, vent velocity, and heat load geometry (localized or distributed) will a thermal stratification develop within a residential room. The goal of this thesis is to answer this question.

In Chapter 2 the experimental simulations are discussed, in addition to the hypothesis statement. Chapter 3 discusses the experimental setup and sample results. In Chapter 4 the results are presented and discussed. Finally Chapter 5 presents the conclusions and plans for future work.

CHPTER 2. INTRODUCTION

1. Introduction

Displacement ventilation (DV) is an energy saving strategy for heating, ventilation and air conditioning, HVAC, systems. DV is typically used in large commercial spaces with high ceilings. The major savings are achieved by reducing the space that is conditioned down to just the occupied space. The intent of this study is to identify if this strategy could be utilized in a residential application with much lower ceilings.

When DV is properly implemented in the cooling phase, a defined separation in temperature occurs between a lower conditioned zone and a higher warm zone. This separation can be caused by a point source heat load or the natural buoyancy in the air. Ideally, the system could be designed where the buoyancy alone would drive and maintain that separation with any configuration of heat loads. However, with a lower ceiling it is likely that point loads would help maintain that stratification.

2. Potential Variables

One of the major complications with determining if DV is effective in a residential application is the numerous configurations and variables that play a role in how well the separation in conditioned zones is held constant.

The location of both the inlet and outlet with respect to each other and the geometry of the loads within the space both significantly affect the stratification stability. In theory, the inlet should be low and the outlet high. This would allow for the system to input the cold conditioned air at floor level and extract the heated contaminated air at ceiling level. However, the system may perform just as well or better with both inlet and outlet at the ceiling level. This system could drop the cold conditioned air into the space and then extract the warm contaminated air near the ceiling. Also, the proximity of inlet and outlet may have a significant affect.

In a practical application, there will be numerous configurations of heat loads. One day the system may be conditioning one person with minimal other heat loads. The next day the system may be conditioning 50 people with an abundance of additional heat loads. Also the location of these loads within the space could have a substantial effect on how well the stratification is maintained. Therefore, regardless of the heat load configuration the system must be capable of maintaining some consistency, at least within the occupied space.

The inlet conditions would also play a major role in the stability of the stratification. The velocity and temperature of the air entering the room will play a role in the amount of mixing that occurs. This variable determines if these systems can be designed with standard heating and cooling units or if these systems would require a unit specific to these applications.

Another consideration is the environmental factors that are specific to an area. The humidity in one area of the country or the prevailing seasonal temperatures will determine the viability of such an approach.

Even the geometry of the space itself could alter the efficiency of DV as an energy saving strategy. Factors like the height of the ceiling or the volume of the room could change the system’s ability to stratify the air at the appropriate height and maintain a conditioned occupied zone.

3. Research Objectives

The major focus of this study was to determine if the location of the inlet and outlet, alone could have an effect on the efficiency of the system. This is done by rearranging the inlet and outlet while maintaining all other conditions in the model room. The four configurations can be seen in Figure 2-1. Figure 2-2 shows the panels that were used to switch the location of both the inlet and outlet.

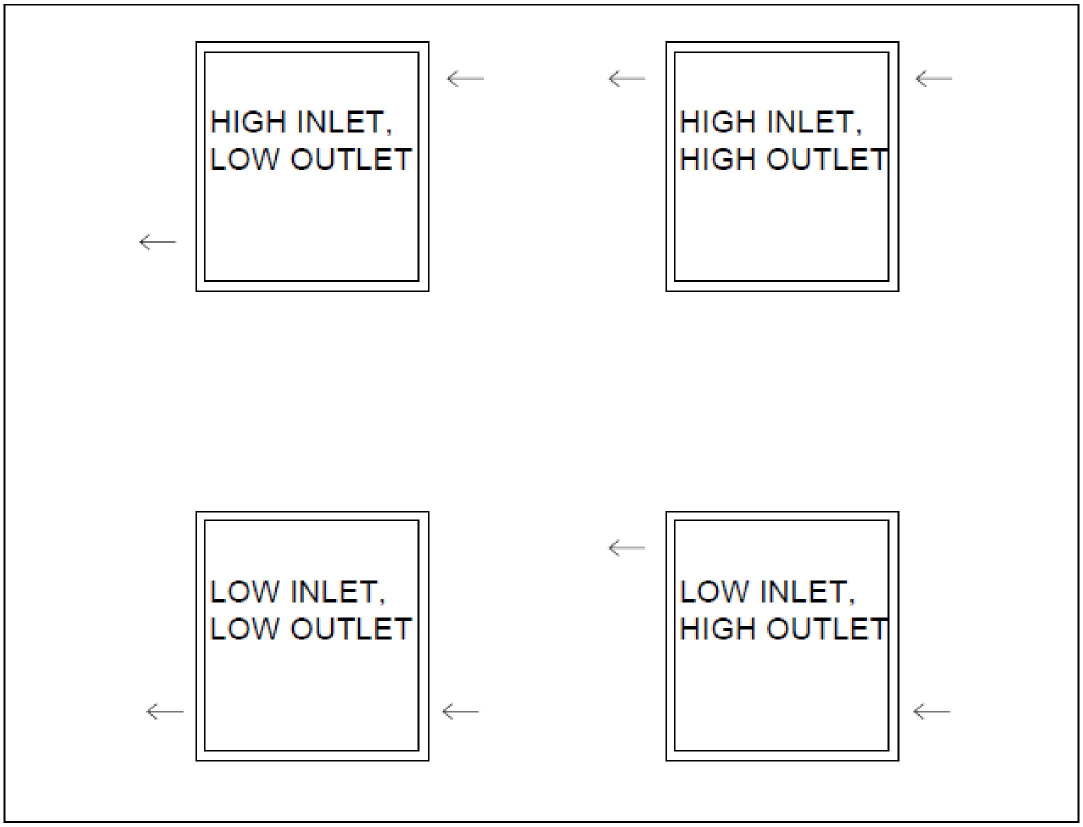


Figure 2-1: The four configurations of the inlet and outlet locations.

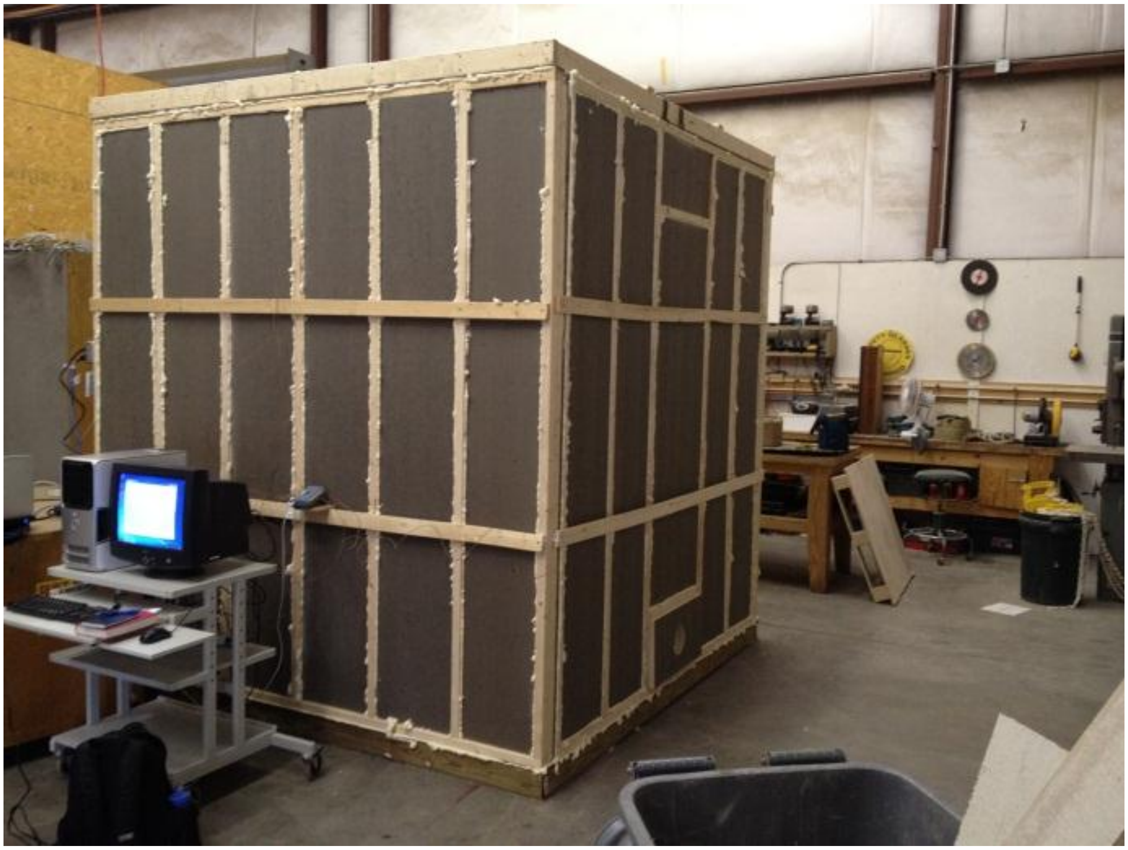


Figure 2-2: The constructed cube. The panels on the inlet wall, right side, that can be switched to change the location of the inlet.

In additon to the configurations, the amount and distribution of heat loads was studied. A point load with and without a distributed load was run at different input wattages. This showed the system's ability to withstand various loads and determined the configurations that resulted in a strong stratification. This also allowed for point load effects to be compared to distributed loads. By better understanding what type of loads assist, hinder or have no effect on the stratification, the appropriate applications of DV can be identified.

4. Hypothesis and Experiments

A model room was constructed and installed with various sensors as shown in Figures 2-3 & 2-4. A humidity sensor and velocity sensor were installed at the inlet to monitor these conditions. A thermocouple was placed on either side of the floor to regulate the thermostats for the under floor heating. Power monitors were installed to monitor the power consumption of the air conditioner, hot plate, and the floor heating systems independently. A thermocouple was installed in each outlet location. Thermocouples were installed evenly through the height of the experimental room in one corner, shown in Figure 2-4. Additionally, a thermocouple was installed at 55” from the floor in the two corners adjacent to the vertical array of thermocouples.

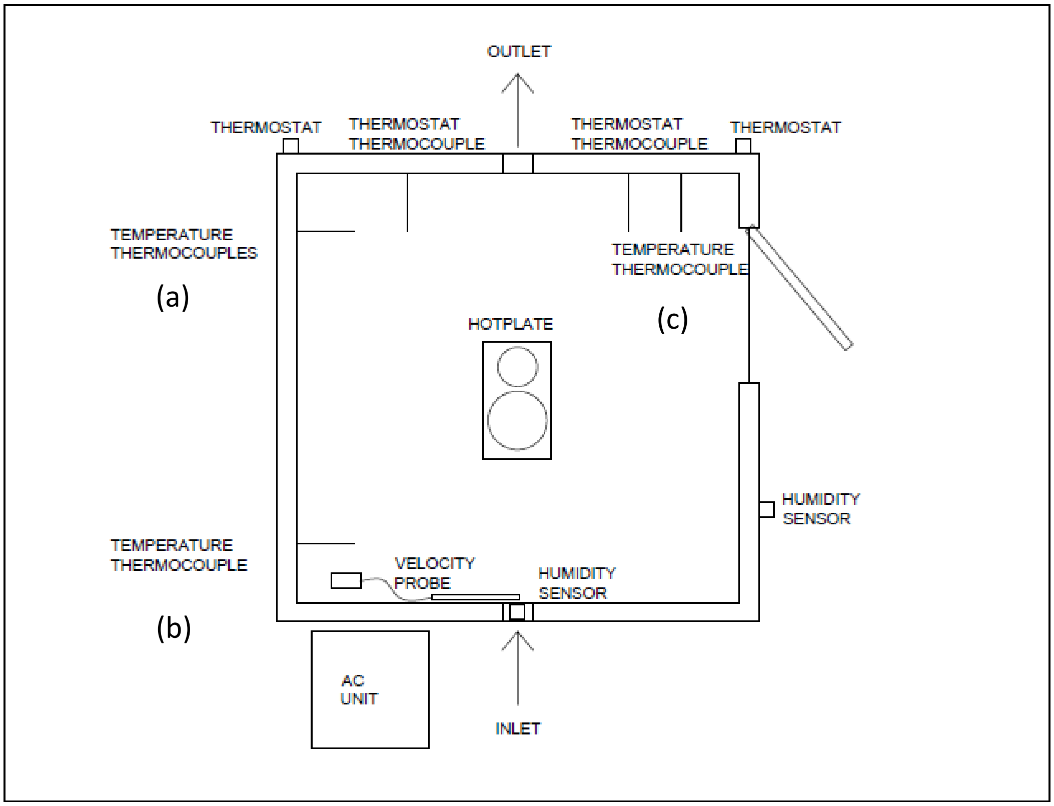


Figure 2-3: Cross-section of the cube from above.

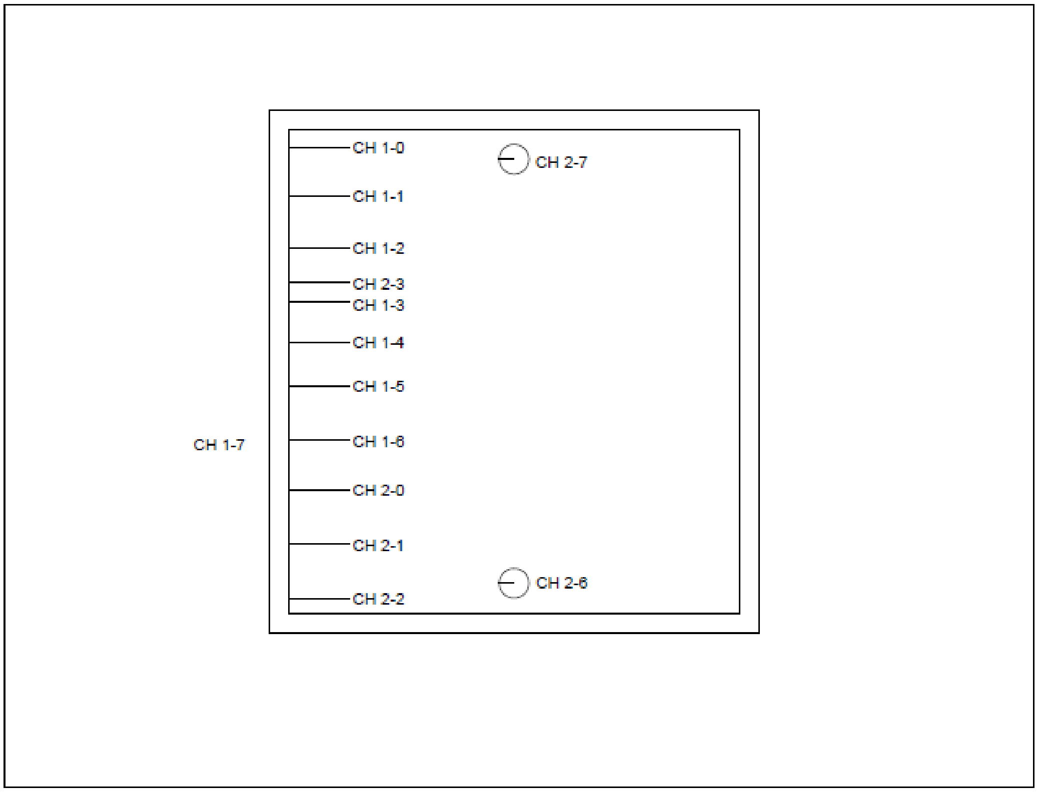


Figure 2-4: Cross-section of the experimental room from the side.

The variables considered were the four inlet/outlet configurations, the hot plate on high and low, the floor heating on high, low, and off, and the air conditioning fan on high and low. The combination of these variables results in 48 experiments and 12 different comparisons of the four main configurations. A few of these sets were also run multiple times to assure the ability of the experiments to be replicated.

It was theorized that the low inlet and high outlet with a lower inlet velocity would result in the strongest stratification when in the cooling phase. This is based on the fact that heat rises which means that a low inlet would place the cold air at the occupant's feet where it would slowly rise as it is heated and contaminated. Then once the air reached the outlet, it would be extracted, reducing the concentration of both heat and pollutants within the space. The benefits of the low inlet, high outlet configuration would be predominantly the improved IAQ and additionally a reduced energy demand from the mechanical system. These benefits should ideally outweigh any cost of the systems implementation.

CHPTER 3. EXPERIMENTAL SETUP

1. Introduction

A series of full scale experiments were run to establish the conditions under which a vertical stratification would be maintained in a small air conditioned room. The apparatus consisted of an eight-foot wooden experimental room connected to an air conditioning unit and installed with both localized and distributed heat sources. The model room was instrumented with temperature, velocity and humidity probes to measure the internal and external climatic conditions. The Power monitors were connected to the heat sources and air conditioning unit to measure the heating and cooling loads. These provided a resolution of 0.1 watts, a measurement range of 0-1800 watts, an accuracy of ±1.5%, and took measurements every one minute. The room was constructed with a wood frame structure, particle board lining the interior and R-19 foam insulation between the studs. The data collection was taken on three computers with two thermocouple readers, two humidity sensors, a velocity probe, and three power monitors. The thermocouple readers provided a resolution of 0.0001˚C, a measurement range from -210˚C to 1200˚C, an accuracy of ±0.29˚C, and took measurements every 10 seconds. The humidity sensor provided a resolution of 0.1˚C and 0.05%RH, a measurement range from -20˚C to 70˚C and 0% to 99% RH, an accuracy of ±0.5˚C and ±2%RH, and took measurements every 30 seconds. The velocity probe provided a resolution of 0.1m/s, a measurement range from 0.2m/s to 20m/s, an accuracy of ±3%, and took measurements every 10 seconds. The cooling load was supplied by a portable air conditioning unit that was retrofitted to have the supply ducted into the cube. The distributed load was simulated with a floor heating system and the point load was simulated with a hot plate.

2. Experimental Room Materials

The frame of the experimental room was constructed out of typical yellow pine 2x4s. The studs were spaced 16 in on center. One side of the frame was lined with 7/16" OSB particle board and between each stud, foam board insulation was placed. The specs for the insulation can be seen in the Appendix. The voids between the insulation and the studs were filled with spray foam. A typical wall is shown in Figure 3-1.

Figure 3-1: The Base wall insulated on the floor and one framed wall braced, standing up

Six walls were constructed in this fashion. One wall was installed with a door which was also insulated as shown in Figure 3-2 and Figure 3-3. Two other walls were built with the center stud missing for easy installation of the inlet and outlet of the air-conditioning system. The resulting void was braced with cross members to stabilize the cube. Below the bottom brace and above the top brace an identical rectangle was left between studs. This allowed for the insulation pieces at both inlet and outlet positions to be interchangeable. The inlet and outlet can then be moved from a bottom position on the wall to a top position and vice versa. This can be seen in Figure 3-4. The six walls were then attached to each other, as seen in Figure 3-4, to create the experimental room. The walls were framed using typical screws and the OSB was attached using typical nails. The walls were attached to each other using the same typical screws. This was done for ease of deconstruction.

Figure 3-2: The insulation on the door of the experimental room

Figure 3-3: The insulation on the door to the experimental room

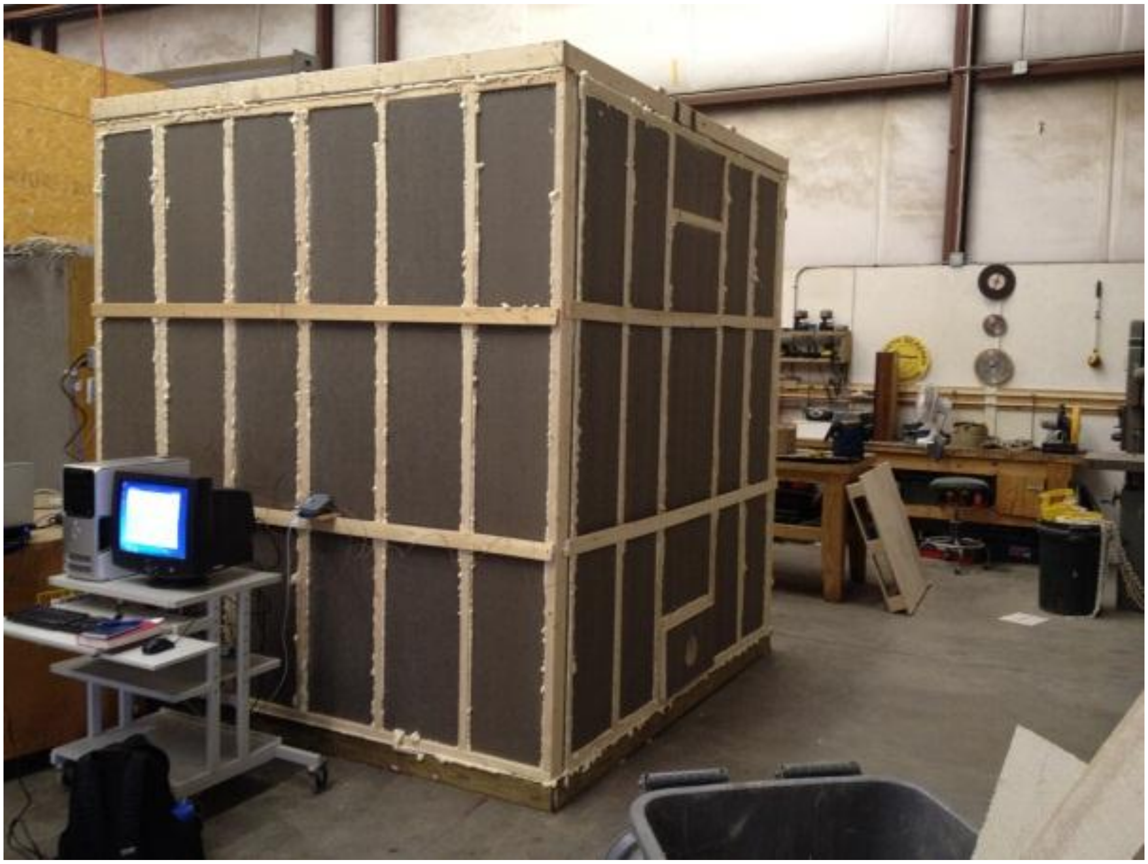


Figure 3-4: The finished experimental room, with the two interchangeable inlet panels shown

Figure 3-5: The final experimental room

Figure 3-6: The unfinished experimental room with the insulation missing between the two studs where the two inlet location were installed

Heating and Cooling Mechanism

Tests were run by imposing a heat load, simulating the occupants and other sources of heat in a residence. This was accomplished with a hot plate for the point loads and a distributed floor heating system for the distributed loads.

Distributed Floor Heating

The floor was heated using a SunTouch system, visit www.warmyourfloor.com for more information. The mesh and wiring was installed between two layers of cement board to prevent any fire hazard and to more evenly distribute the heat. This can be seen in Figure 3-7. The system was designed to be installed in the mortar below bathroom tile. However, for this application maintaining the ability to change the layout was important. So the heating element was installed in a way such that it could be moved and reconfigured. This was done for the sake of future studies.

Figure 3-7: The under floor distributed heating system partially covered in cement board

The under floor heating system is controlled by a thermostat that typically has a thermocouple installed into the floor system that it monitors. However, in this application, a constant power output was desired in order to establish a steady state internal temperature field. The power output of the system was controlled by the thermostat. Therefore the thermocouple was placed on top of the cement board and when it read the temperature to be at the set point the element turned off. The thermocouple placement can be seen in Figure 3-8. Then the element turned back on when the temperature dropped back below the set point. This resulted in the floor system approximately operating as a constant heat flux source, see Figure 3-9. A standard non-programmable thermostat was used because the power output was monitored by other means. The thermostat had labeled set point temperatures at min, 60, 70, 80, 90, and max. This allowed for some variation to the input level. The heating element achieved this set point temperature by turning on and off.

Figure 3-8: The thermocouples that are attached to the thermostats for the under floor distributed heating system

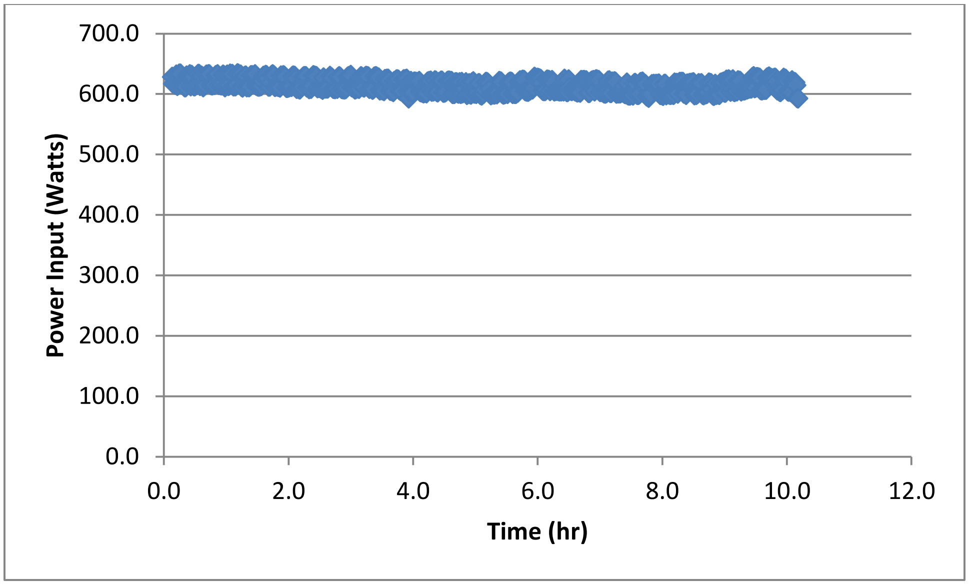


Figure 3-9: The power inlet to the distributed floor heating system over ten hours

Hotplate

Localized heating, such as from a person or electrical appliance, was simulated by a portable GE double burner hotplate, with a maximum output of 1500 watts. Just as the thermostat regulated the floor heating system, an internal temperature sensor regulated the hotplate. The hotplate runs to a much higher cut off point than the under-floor heating system so the energy input is much closer to constant at about 600 watts with the smaller burner set to max. An occupant can produce a load between 115 and 580 W depending on size and activity level, according to the 2009 ASHRAE Handbook - Fundamentals. Therefore, when the hotplate is set to low, the resultant ≈300 W is similar to the heat load of one occupant with reasonable activity or two occupants at a more sedentary state.

Portable AC Unit

The cooling load was provided by a portable air conditioning unit that was retrofitted to allow the cooled air to be ducted into the cube. A few different cooling units were considered, but the PAC N100EL portable air conditioner, a DeLonghi product, was the easiest to attach ductwork to. The unit has three different fan speeds and temperature set points from 61˚F to 89˚F. The output of the portable air conditioning unit was attached to a duct reducer that surrounded the rectangular output and contracted down to a six inch round duct. The void between the reducer and the air conditioning unit was sealed with two sided sealant tape and the duct and reducer were screwed together. Then a piece of insulated flex duct was run from the reducer to the six inch vent inlet hole in the side of the experimental room. At the void in the wall a six inch round duct connecter was fitted into the insulation. This gave the flex duct a rigid connection point on the cube. The flex pipe was connected to the duct connector and secured with duct tape.

3. Instrumentation

The experimental room was instrumented with thermocouples, humidity sensors, a velocity probe and two particle counters. The instrumentation was connected to three computers for data recording. The first computer, a Dell Dimension E521, was attached to the first thermocouple reader, which was reading thermocouples CH 1-0 through CH 1-7. The second computer, a Gateway laptop, was attached to the second thermocouple reader, which was reading thermocouples CH 2-0 through CH 2-7. The velocity probe was either attached to a Dell XPS L502X or the readings were manually recorded. The remaining instruments recorded their own data, which was then downloaded after each simulation.

Thermocouples

Fifteen thermocouples were attached to two thermocouple readers on the exterior of the experimental room and run through holes in the side of the experimental room; the hole was then filled with spray foam insulation. Visit http://www.omega.com/ppt/pptsc.asp?ref=5TC for more information on the thermocouples and http://www.omega.com/ppt/pptsc.asp?ref=OM-USB-TC for more information on the thermocouple readers. Each thermocouple was calibrated using the program's calibration and a manual calibration to verify the electronic calibration. The thermocouples were attached with a small zip tie to a fishing line run one foot from each wall inside the experimental room; see Figure 3-10 and Figure 3-11. Ten thermocouples were evenly spaced over the height of the experimental room at (a) in Figure 2-3, approximately every 28 cm with approximately 7.6 cm clearence below the bottom and above the top thermocouple. One thermocouple was also located 1.4 m above the floor at (b) and (c) in Figure 2-3. One was placed at each of the potential outlets, located at 16.5 cm and 231 cm above the floor. The remaining thermocouple was run in the experimental room at 1.7 m. According to ASHRAE Standard 55-2010 1.7

m is the height used to represent the top of the occupied zone. The difference in temperature between this height and 28 cm from the floor is one criteria used by ASHRAE Standard 55 to assure occupant comfort. Each thermocouple is attached to one of two thermocouple readers that are attached to one of the three computers. The computer continuously records the temperature values every 10 seconds throughout each experiment.

Figure 3-10: The thermocouple closest to the floor of the experimental room

Figure 3-11: A thermocouple inside the experimental room attached to the fishing line with a small zip tie

Power Monitors

The energy input from the under-floor heater and the hotplate was monitored with 3 *Watts Up? Pro* power monitors, see www.wattsupmeters.com for more information. The efficiency of the heating systems was assumed to be 100% as any losses would also manifest themselves as heat. A third power monitor was attached to the portable air conditioner (AC). This allowed for the monitoring of the power used when the AC was on and when the AC was only pumping exterior air into the cube. These power monitors record the data throughout each test, after which the data is downloaded to one of the computers and the power monitor is reset ready for the next test.

Inlet vent velocity

The velocity at the AC inlet vent was measured continuously during select tests using an Extech Heavy Duty Hot Wire Thermo-anemometer, see http://www.extech.com/instruments/product.asp?catid=1&prodid=39 for more details. The velocity was measured at one point multiple times over 4 or 5 hours to assure that the velocity did not vary with time. Figure 3-12 shows the consistency of the inlet velocity. It was noticed that the velocity varied greatly with the location on the inlet. Therefore velocity readings were taken at nine different locations on the inlet. These readings were then weighted by the area they represented to find a reasonable value for the inlet velocity. This was done for the inlet at the low location with low fan speed and high fan speed, as well as for the inlet at the high location with both fan speeds. Therefore, one value was found for the inlet velocity for the four different inlet conditions. These four values can be seen in Table 3-1.

Table 3-1: The average velocity for each inlet scenario

<table>
<tr><td></td><td></td><td colspan="2">Fan Speed</td></tr>
<tr><td></td><td></td><td>Low</td><td>High</td></tr>
<tr><td rowspan="2">Inlet Location</td><td>Low</td><td>2.79 m/s</td><td>3.02 m/s</td></tr>
<tr><td>High</td><td>2.47 m/s</td><td>2.84 m/s</td></tr>
</table>

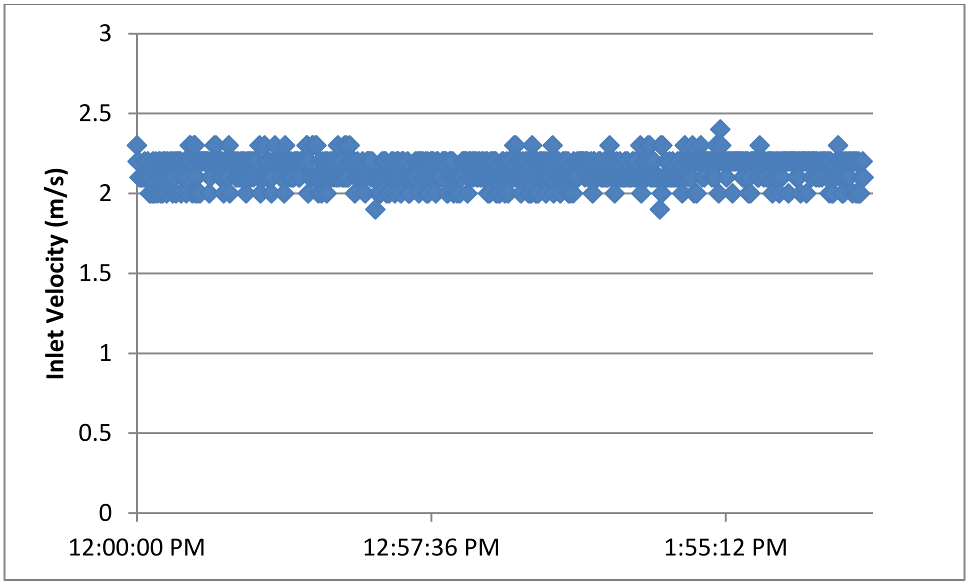


Figure 3-12: The velocity at the inlet over two hours

Relative Humidity

The humidity on the inside and the outside of the experimental room were monitored with humidity sensors, see http://www.omega.com/ppt/pptsc.asp?ref=OM-70_Series&Nav=dase01 for more info. One was hung on the outside of the experimental room and the other was placed inside the inlet. This provided the outside conditions and the inlet temperature and humidity for each run. The location of each sensor can be seen in Figure 2-3. These sensors record the data throughout each test and then the data was downloaded and the sensor reset after each run.

4. Calibration

Each set of instruments were calibrated to assure their accuracy. The power monitors, humidity sensors, and velocity probe all either came with software to calibrate the system or were calibrated by the manufacturer.

Thermocouple Calibration

The thermocouples were calibrated a few different ways. The computer program, *Instacal,* was used to calibrate the system electronically. This program is the manufacture recommended calibration format and was available free from their website. The thermocouples were also manually calibrated with ice water. The ice water calibration was done to assure that each thermocouple read zero degrees Celsius when in a well-mixed ice bath. After the calibration, the thermocouples were assured to read within 0.2 degree Celsius of each other. This in conjunction with the computer calibration assured the accuracy of the temperature readings from the thermocouples.

System Calibration

Although the room was insulated with R-19 insulation, there was still heat transfer through the walls that had to be accounted for. The insulation greatly reduced this heat transfer but the wood studs offer channels for the heat to enter or escape the cube.

This heat loss was initially estimated by using the R-values of the materials. The heat loss, HL, was calculated with $HL=AU(\Delta T)$, where A is the area in m^2, U is the heat transfer rate of the material ($W/(K*m^2)$), and ΔT is the difference in temperature from the inside of the experimental room to the outside. Table 3-2 shows these areas and R-values. The R-values are reported in imperial units in the US. The values in Table 3-2 are the converted values. Since most of the simulations were run with relatively low internal to external temperature differences, the heat loss should be negligible.

Table 3-2: The R-values and areas of the different regions of the experimental room

	Insulated Area (m²)	Uninsulated Area (m²)	Door and Frame (m²)	Outside Air Film R-Value	Inside Air Film R-Value
Floor	4.77	0.44	0.00	0.19	0.19
Wall 1	5.02	0.64	0.00	0.26	0.26
Wall 2	5.02	0.81	0.00	0.26	0.26
Wall 3	3.43	0.57	1.65	0.26	0.26
Wall 4	5.02	0.81	0.00	0.26	0.26
Roof	4.77	0.44	0.00	0.19	0.29
R-Value	3.63	1.14	3.68		

Sealed System

To verify the calculated value, the room was sealed up and the heat was turned on. Once the system reached a steady state, that is the internal temperature remained steady over time, the amount of heat put into the room must be balanced by the heat leaving the room through the walls. The room was heated to 5.68 ˚C higher than the temperature in the lab. This was the average difference over an hour and forty five minutes. The resulting heat input was averaged over the same time and was found to be 183W. The calculation would account for 110W. Therefore, the remaining 73W can be attributed to the materials performing at a lower R-value than their rating and to some leakage through the walls. To account for these losses an AU* value of 61 was used as the adjusted factor to calculate HL.

5. Repetitions

Three scenarios were repeated to test the consistency of results. The power inputs and outside humidity to these scenarios are listed in

Table 3-3 and the temperature readings are shown in Table 3-4. All of the repetitions were run with the low inlet, high outlet configuration at steady state. It was found that after 10 hours running, most experiments reached a steady state. Steady state was considered the time when the room was no longer heating up or cooling down. This was controlled by the thermostats for both heating elements. The thermostats were able to turn the heating element on and off once the desired set points were reached. The low AC fan speed, low hot-plate, and high distributed floor configuration was run an additional five times. This resulted in six sets of the same configuration. Also, the low AC fan speed, low hot-plate, and no distributed floor configuration and the low AC speed high hot-plate, and no distributed floor configuration were run an additional two times. This provided two sets of three runs with the same scenario.

Table 3-3: The power inputs and outside humidity readings for the repeated experiments

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	Low	High	624.26	275.79	1280.37	55.44
Low	High	Low	Low	High	620.51	273.69	1289.77	38.75
Low	High	Low	Low	High	618.56	277.06	1289.71	43.55
Low	High	Low	Low	High	622.49	270.66	1169.74	49.67
Low	High	Low	Low	High	610.85	308.66	1114.65	43.06
Low	High	Low	Low	High	612.20	285.67	1152.74	65.48
Average					618.14	281.92	1216.16	49.33
Standard Deviation					5.49	14.03	79.29	9.83
Low	High	Low	Low	Off	0.00	316.60	1230.38	54.66
Low	High	Low	Low	Off	0.00	324.86	1239.69	43.62
Low	High	Low	Low	Off	0.00	348.64	1217.97	43.21
Average						330.03	1229.34	47.16
Standard Deviation						16.64	10.90	6.50
Low	High	Low	High	Off	0.00	568.18	1269.77	38.01
Low	High	Low	High	Off	0.00	565.04	1347.61	44.11
Low	High	Low	High	Off	0.00	568.34	1248.93	56.97
Average						567.18	1288.77	46.37
Standard Deviation						1.86	52.01	9.68

Table 3-4: The temperature readings for the repeated experiments

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Temperature Reading (˚C)															
					CH 1-0	CH 1-1	CH 1-2	CH 2-3	CH 1-3	CH 1-4	CH 1-7	CH 2-4	CH 1-5	CH 1-6	CH 2-0	CH 2-1	CH 2-2	CH 2-6/7	Inlet	Outside
Inlet	Outlet				94"	84"	74"	67"	63"	55"	55"	55"	46"	35"	25"	14"	3"	Outlet		
Low	High	Low	Low	High	31.07	30.99	30.91	31.02	30.89	30.65	31.70	30.33	30.80	30.89	30.91	30.91	30.85	31.59	17.79	22.68
Low	High	Low	Low	High	29.19	29.20	28.79	28.97	28.42	28.43	29.38	28.29	28.52	28.51	28.78	29.00	28.99	29.51	14.53	24.40
Low	High	Low	Low	High	29.55	29.55	29.20	29.37	28.84	28.77	29.73	28.84	28.89	28.88	29.08	29.24	29.16	29.88	15.10	24.87
Low	High	Low	Low	High	28.78	28.84	28.43	28.53	27.96	27.88	28.89	27.92	28.00	28.01	28.29	28.52	28.56	29.06	14.27	21.70
Low	High	Low	Low	High	31.05	31.07	31.21	31.33	31.15	30.84	31.59	30.66	30.75	30.51	30.49	30.39	29.95	31.60	17.40	26.59
Low	High	Low	Low	High	30.77	30.58	30.39	30.18	29.56	28.73	30.12	29.35	28.00	27.61	27.93	28.13	27.88	30.25	15.94	22.76
Average					30.07	30.04	29.82	29.90	29.47	29.22	30.24	29.23	29.16	29.07	29.25	29.36	29.23	30.32	15.84	23.84
Std Dev					1.02	0.96	1.17	1.13	1.31	1.23	1.17	1.10	1.29	1.34	1.20	1.08	1.05	1.07	1.48	1.79
Low	High	Low	Low	Off	21.94	21.83	21.87	22.42	21.81	21.77	22.02	21.34	21.77	21.64	21.71	21.66	21.51	21.72	15.10	21.33
Low	High	Low	Low	Off	21.90	21.82	21.89	22.39	21.85	21.74	21.94	21.25	21.71	21.59	21.65	21.59	21.44	21.74	15.22	22.15
Low	High	Low	Low	Off	20.80	20.56	20.48	20.81	20.35	20.24	20.86	20.83	20.23	20.13	20.25	20.29	19.97	21.33	13.84	23.13
Average					21.55	21.40	21.41	21.87	21.33	21.25	21.60	21.14	21.24	21.12	21.21	21.18	20.97	21.60	14.72	22.20
Std Dev					0.64	0.73	0.81	0.93	0.86	0.88	0.65	0.28	0.87	0.86	0.83	0.77	0.87	0.23	0.76	0.90
Low	High	Low	High	Off	24.94	24.84	24.96	25.51	24.98	24.98	24.98	24.05	25.03	24.83	24.67	24.27	23.85	24.99	14.61	22.81
Low	High	Low	High	Off	29.62	29.54	29.72	30.05	29.78	29.62	29.76	28.65	29.67	29.51	29.25	28.76	28.30	30.08	18.73	25.70
Low	High	Low	High	Off	26.21	25.55	25.19	25.22	24.89	24.73	25.97	25.38	24.64	24.38	24.20	23.63	23.34	25.95	14.40	22.12
Average					26.92	26.64	26.62	26.93	26.55	26.44	26.90	26.03	26.45	26.24	26.04	25.55	25.16	27.01	15.91	23.54
Std Dev					2.42	2.54	2.69	2.71	2.80	2.75	2.52	2.37	2.80	2.84	2.79	2.79	2.73	2.71	2.44	1.90

Figure 3-13 shows the six repetitions of the low inlet, high outlet, low AC fan speed, low hot-plate, and high distributed floor heat configuration. The overall temperature of the profiles seems to vary significantly. However, when the inlet temperature is considered, the variation is perfectly in line. The inlet temperatures are listed in the key with each line type. As the inlet temperature increases, so too does the internal temperature. The only exception is the run with an inlet temperature of 15.9 C. This run was able to achieve a harsh stratification, while none of the others were.

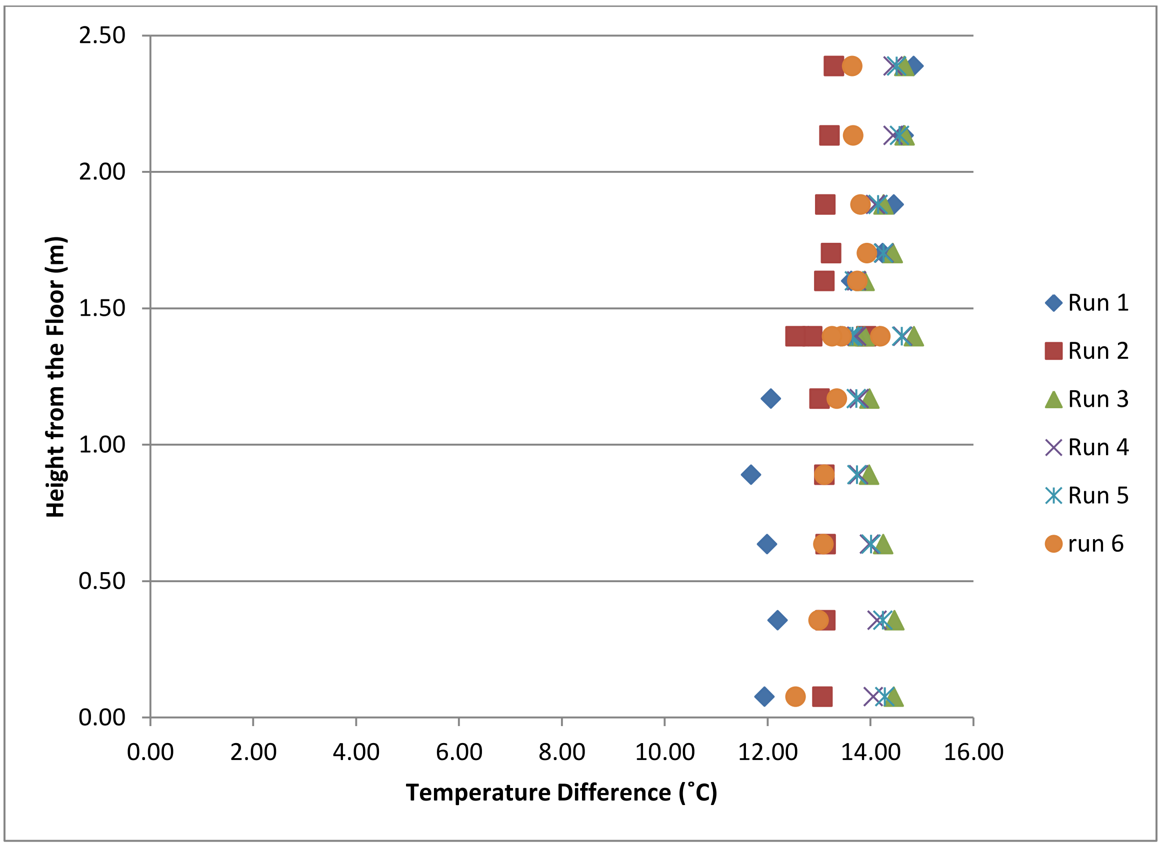


Figure 3-13: The six temperature profiles for the repetitions of the low inlet, high outlet, low AC fan speed, low hot-plate, and high distributed floor heat configuration are shown above. The temperature difference is each temperature reading less the inlet temperature.

Figure 3-14 shows the three repetitions of the low inlet, high outlet, low AC fan speed, low hot-plate, and no distributed floor heat configuration. All three of these runs line up with the same curve. The 13.8 C inlet temperature led to a lower overall temperature. However, the other two runs had about the same inlet temperature and resulted in almost identical curves.

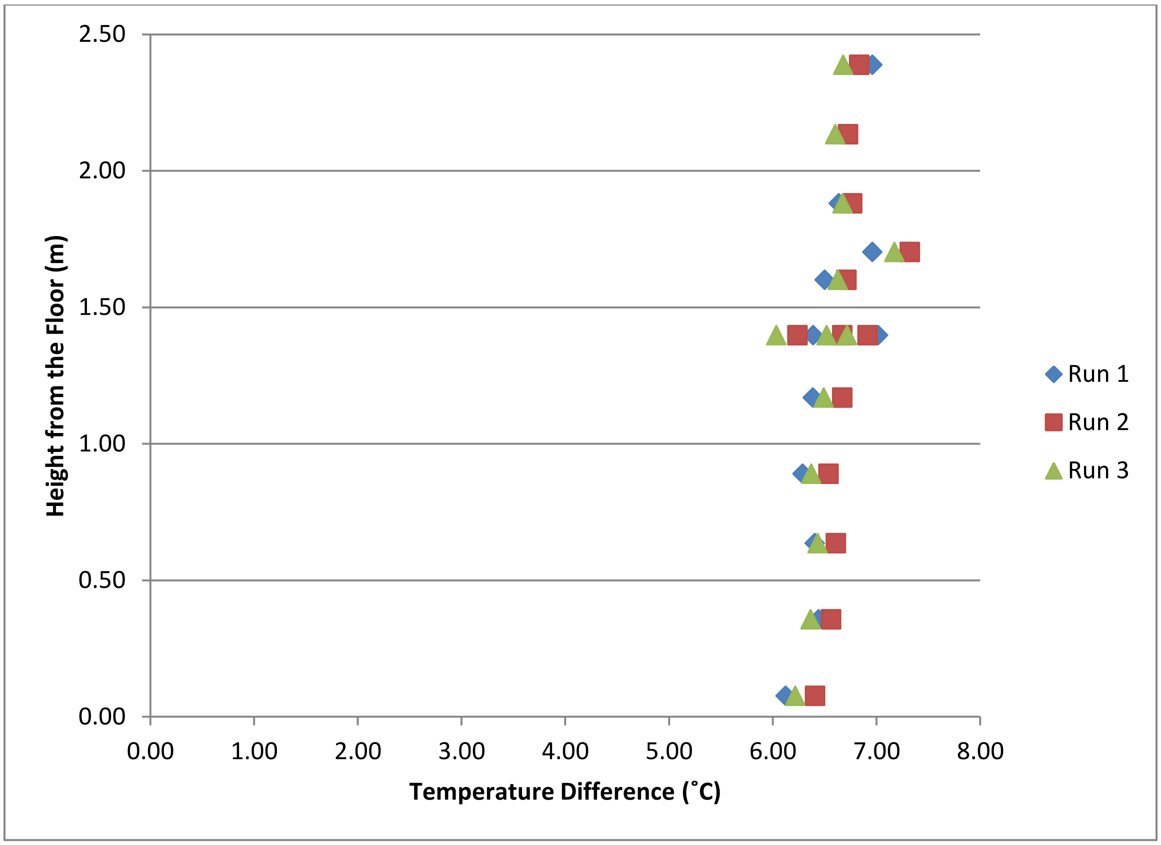


Figure 3-14: The three temperature profiles for the repetitions of the low inlet, high outlet, low AC fan speed, low hot-plate, and no distributed floor heat configuration are shown above. The temperature difference is each temperature reading less the inlet temperature.

Figure 3-15 shows the three repetitions of the low inlet, high outlet, low AC fan speed, low hot-plate, and no distributed floor heat configuration. Just as with the other runs when the inlet temperature varies the curve shifts in the direction the inlet temperature does. The two runs with nearly the same inlet temperature have nearly the same curve.

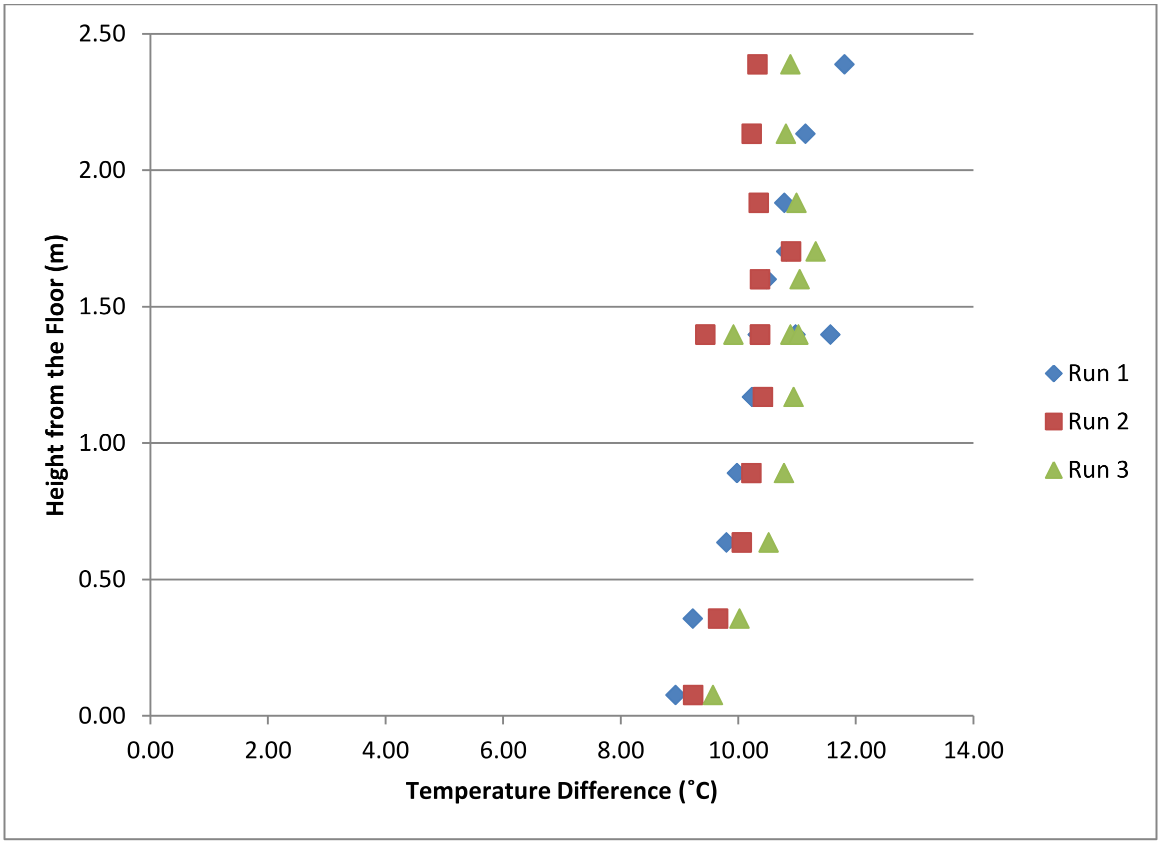


Figure 3-15: The three temperature profiles for the repetitions of the low inlet, high outlet, low AC fan speed, high hot-plate, and no distributed floor heat configuration are shown above. The temperature difference is each temperature reading less the inlet temperature.

The repetitions show that there is some variation that is influenced by the external environment. The inlet temperature varies with the outside temperature and humidity. Therefore, the curves are almost exclusively shifted with the inlet temperature. The presence or lack of a harsh stratification seems to also vary with the conditions. However, with and without the harsh stratification the low inlet, high outlet configuration was able to handle a greater heat load in almost all cases.

6. Sample Data Results

Each experiment ran for at least 10 hours. All of the data outputs were combined into one Excel file for analysis. Figure 3-16 shows the raw Excel output from the humidity sensor. The time, temperature, relative humidity, and dew point every 30 seconds are listed. The Excel output from the thermocouple reader is shown in Figure 3-17. The time and the temperature at each of eight channels are listed every 10 seconds. Figure 3-18 shows the Excel output from the power monitors. The time from the beginning of the experiment is listed with the wattage and the voltage every minute. Figure 3-19 shows the Excel output from the velocity probe. The time, date, and two channel values are listed every 10 seconds. Channel 1 is the velocity in meters per second and Channel 2 is the temperature in ˚C.

	A	B	C	D	E	F	G	H
1	Logger Memory: Temperature/Humidity Logger							
2								
3								
4	Summary							
5								
6	Channel	Min	Max	Average	Standard Dev	Mean Kinetic Temperature		
7	Temp (F)	56.1	63.5	61.3609	2.17	61.5034		
8	Hum (%RH	48.05	78.45	62.3411	1.44	-----		
9	Dew Point	38.25	56.24	48.3801	1.76	-----		
10								
11								
12	Sample	Temp (F)	Hum (%RH	Dew Point	Date Time			
13	1	58.5	72.8	49.81	8:51:37 AM			
14	2	59.1	65.95	47.75	8:52:07 AM			
15	3	59.4	64.4	47.4	8:52:37 AM			
16	4	59.45	63.95	47.26	8:53:07 AM			
17	5	59.45	63.95	47.26	8:53:37 AM			
18	6	59.45	65.5	47.9	8:54:07 AM			
19	7	59.4	66.15	48.11	8:54:37 AM			
20	8	59.4	65.85	47.99	8:55:07 AM			
21	9	59.3	65.95	47.94	8:55:37 AM			
22	10	59.3	66.25	48.06	8:56:07 AM			
23	11	59.4	66	48.05	8:56:37 AM			
24	12	59.45	65.9	48.06	8:57:07 AM			
25	13	59.45	60.7	45.88	8:57:37 AM			

Tables / Temp (2) / Temp / System1 / System2 / **Humidity Inlet** / Humidity Ou

Figure 3-16: Data output from the humidity sensor

	A	B	C	D	E	F	G	H	I	J	K
1	Header Size: 8										
2	Version: 2										
3	Sampling Interval: 10										
4	Sampling Rate: 0.1										
5	Sample Count: 3606										
6	Device Serial Number: 0										
7	Culture Info: en-US										
8	Sample N	Date/Time	CH 1-0	CH 1-1	CH 1-2	CH 1-3	CH 1-4	CH 1-5	CH 1-6	CH 1-7	Events
9	1	9:01:48 AM	80.8939	79.5213	78.614	77.1739	76.3718	75.7004	74.9081	78.3188	DAQ Start
10	2	9:01:58 AM	81.0254	79.2867	78.8026	77.3251	76.6074	75.9296	75.2152	78.4869	
11	3	9:02:08 AM	81.0221	79.1695	78.7993	77.4616	76.7625	76.1794	75.561	78.66	
12	4	9:02:18 AM	80.7837	79.2817	78.7419	77.5507	76.8557	76.3231	75.7955	78.7492	
13	5	9:02:28 AM	80.3893	79.1109	78.6319	77.6335	77.0799	76.6587	76.0833	78.9913	
14	6	9:02:38 AM	80.2017	79.1665	78.7614	77.7812	77.316	76.8833	76.3701	79.2	
15	7	9:02:48 AM	79.9577	79.2098	78.8435	78.0161	77.4115	77.1887	76.7819	79.3524	
16	8	9:02:58 AM	79.9871	79.379	78.9622	78.2553	77.589	77.3286	76.9101	79.4792	
17	9	9:03:08 AM	79.8503	79.3936	79.0117	78.4383	77.8779	77.5217	77.2329	79.6851	
18	10	9:03:18 AM	79.7873	79.3927	79.1946	78.5409	77.8735	77.6572	77.3774	79.8088	
19	11	9:03:28 AM	79.8723	79.5734	79.3521	78.6778	78.0876	77.8156	77.5294	79.8713	
20	12	9:03:38 AM	79.8365	79.6036	79.4936	78.8917	78.2739	78.0473	77.7443	80.0316	
21	13	9:03:48 AM	79.904	79.6892	79.6969	79.0783	78.4781	78.2916	78.0042	80.2938	
22	14	9:03:58 AM	79.8613	79.6827	79.7396	79.2142	78.6817	78.5264	78.2661	80.4481	
23	15	9:04:08 AM	79.9091	79.7202	79.9	79.3915	78.8724	78.7482	78.3675	80.5649	
24	16	9:04:18 AM	79.9583	79.8858	79.9854	79.561	79.0666	78.9048	78.6045	80.7343	
25	17	9:04:28 AM	80.1226	80.0075	80.1264	79.6529	79.0795	79.0045	78.6783	80.8132	

Tables / Temp (2) / Temp / **System1** / System2 / Humidity Inlet / Humidity Outside / PMHP / PMFloor / PMAC

Figure 3-17: Data output from the thermocouple reader

	A	B	C	D	E	F
1	Time	Watts	Volts			
2						
3	0	562.4	119.3			
4	0.000694	565	119.4			
5	0.001389	0	123.6			
6	0.002083	0	123.6			
7	0.002778	0	123.7			
8	0.003472	0	123.8			
9	0.004167	0	123.6			
10	0.004861	0	123.7			
11	0.005556	598.3	121.6			
12	0.00625	564.9	119.5			
13	0.006944	560	119.4			
14	0.007639	558.2	119.3			
15	0.008333	556.9	119.3			
16	0.009028	557	119.1			
17	0.009722	505.6	119.2			
18	0.010417	556.7	119.2			
19	0.011111	558.2	119.2			
20	0.011806	561.7	119.2			
21	0.0125	512.6	119.4			
22	0.013194	0	121.2			
23	0.013889	0	121			
24	0.014583	0	121			
25	0.015278	0	120.9			

Tables / Temp (2) / Temp / System1 / System2 / Hu

Figure 3-18: Data output from the power monitor

	A	B	C	D	E	F
1	date	time	chan 1	unit 1	chan 2	unit 2
2	9/17/2012	8:44:15 PM	3.1	NO UNIT	16.9	degree C
3	9/17/2012	8:44:25 PM	3	NO UNIT	16.7	degree C
4	9/17/2012	8:44:35 PM	3.1	NO UNIT	16.6	degree C
5	9/17/2012	8:44:45 PM	3.1	NO UNIT	16.6	degree C
6	9/17/2012	8:44:55 PM	3.2	NO UNIT	16.5	degree C
7	9/17/2012	8:45:05 PM	3	NO UNIT	16.4	degree C
8	9/17/2012	8:45:15 PM	3.1	NO UNIT	16.4	degree C
9	9/17/2012	8:45:25 PM	3.1	NO UNIT	16.4	degree C
10	9/17/2012	8:45:35 PM	3.1	NO UNIT	16.4	degree C
11	9/17/2012	8:45:45 PM	3	NO UNIT	16.3	degree C
12	9/17/2012	8:45:55 PM	3.1	NO UNIT	16.3	degree C
13	9/17/2012	8:46:05 PM	3	NO UNIT	16.3	degree C
14	9/17/2012	8:46:15 PM	3.1	NO UNIT	16.3	degree C
15	9/17/2012	8:46:25 PM	3	NO UNIT	16.3	degree C
16	9/17/2012	8:46:35 PM	3	NO UNIT	16.3	degree C
17	9/17/2012	8:46:45 PM	3	NO UNIT	16.3	degree C
18	9/17/2012	8:46:55 PM	3	NO UNIT	16.3	degree C
19	9/17/2012	8:47:05 PM	2.9	NO UNIT	16.3	degree C
20	9/17/2012	8:47:15 PM	2.8	NO UNIT	16.3	degree C
21	9/17/2012	8:47:25 PM	3	NO UNIT	16.3	degree C
22	9/17/2012	8:47:35 PM	3.1	NO UNIT	16.3	degree C
23	9/17/2012	8:47:45 PM	3.3	NO UNIT	16.2	degree C
24	9/17/2012	8:47:55 PM	3.2	NO UNIT	16.2	degree C
25	9/17/2012	8:48:05 PM	3.3	NO UNIT	16.2	degree C

Sheet1 / Sheet2 / Sheet3

Figure 3-19: The Excel output file from the velocity probe, Channel 1 is the velocity in m/s and Channel 2 is the temperature in ˚C

All of the thermocouples' readings were plotted on one graph seen in Figure 3-20. Then an hour was identified as the closest to steady state. This hour was then used to find an average value for each thermocouple temperature. The graph of the thermocouple data over that hour can be seen in Figure 3-21. Then the average temperatures were put into the table in Figure 3-22. Additionally the average wattage taken from each power monitor, the average humidity and temperature outside the experimental room, and the graph of the temperature profile were all put into the Excel sheet in Figure 3-22.

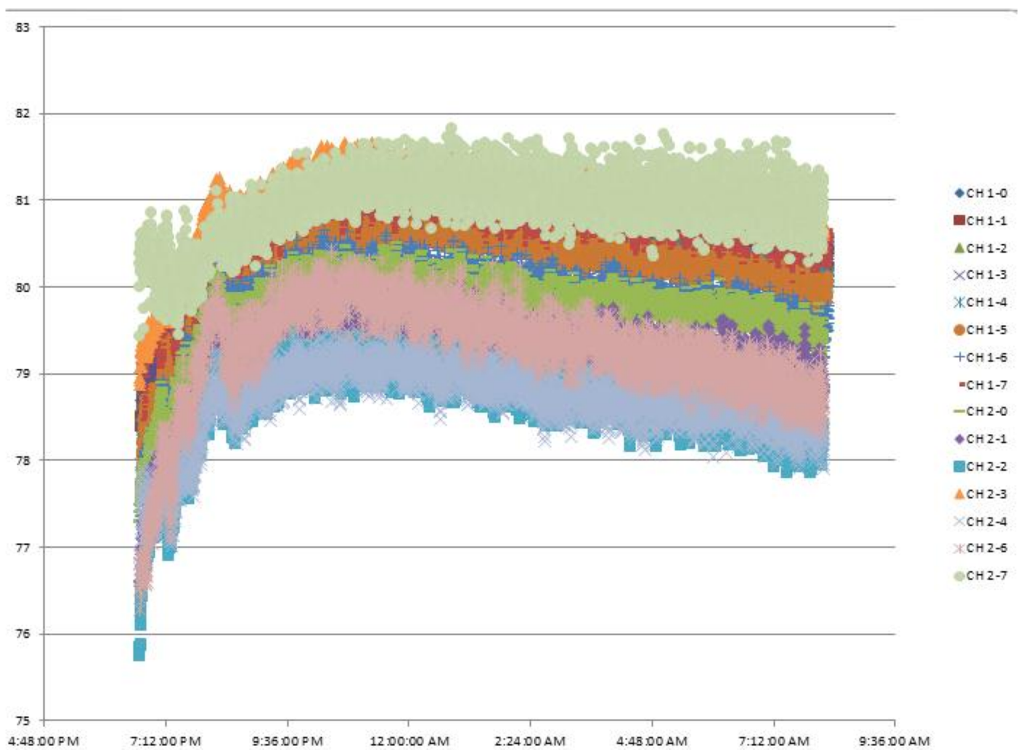


Figure 3-20: Temperature data for the full run of an experiment

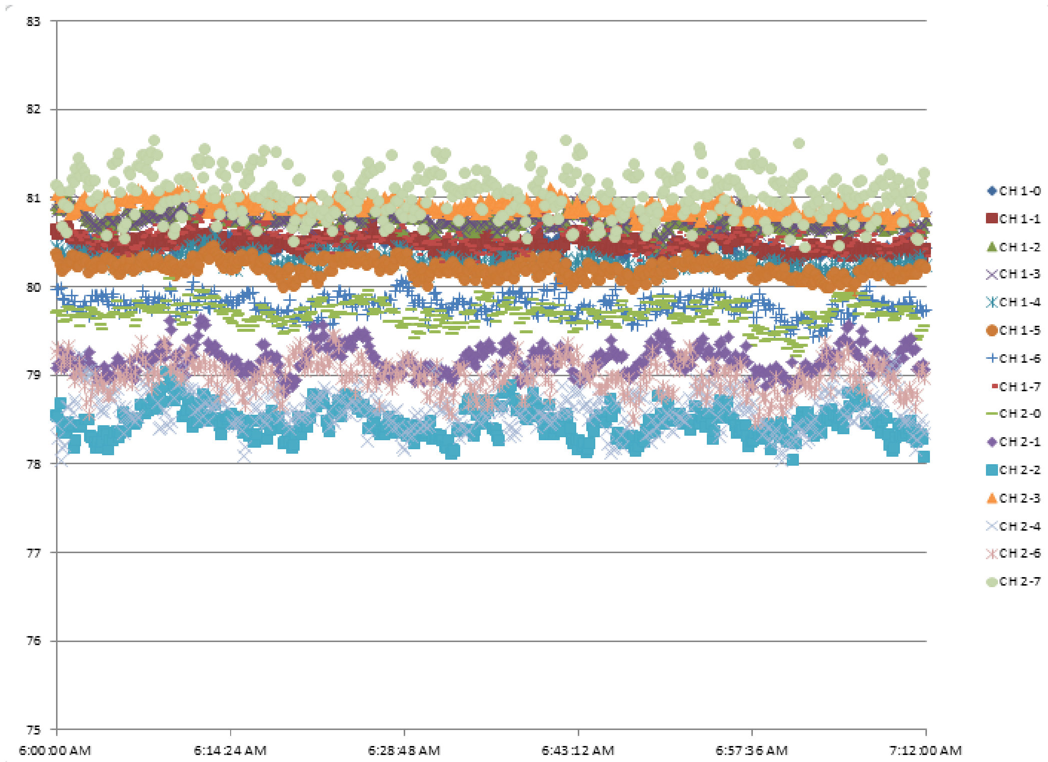
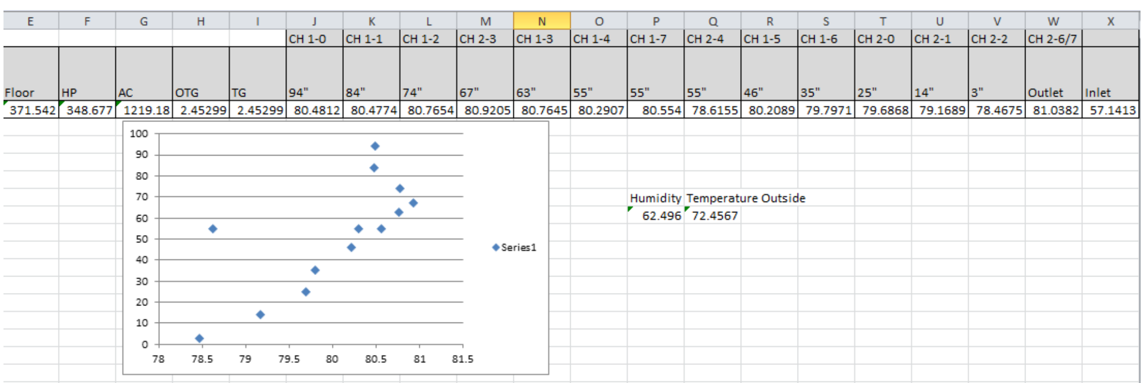


Figure 3-21: One hour of temperature data



E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X
					CH 1-0	CH 1-1	CH 1-2	CH 2-3	CH 1-3	CH 1-4	CH 1-7	CH 2-4	CH 1-5	CH 1-6	CH 2-0	CH 2-1	CH 2-2	CH 2-6/7	
Floor	HP	AC	OTG	TG	94"	84"	74"	67"	63"	55"	55"	55"	46"	35"	25"	14"	3"	Outlet	Inlet
371.542	348.677	1219.18	2.45299	2.45299	80.4812	80.4774	80.7654	80.9205	80.7645	80.2907	80.554	78.6155	80.2089	79.7971	79.6868	79.1689	78.4675	81.0382	57.1413

Humidity	Temperature Outside
62.496	72.4567

Figure 3-22: The table used to summarize the results of each experimental run

The temperature profiles in Figure 3-22 along with the power input to the heated floor and hot plate were used to compare the different scenarios. The power input of the heating systems was considered the heat load in the space. The resultant temperature profile reflected the ventilation configuration's ability to handle that heat load. Some scenarios were able to take a larger heat load and provide a cooler occupied zone.

CHPTER 4. RESULTS

1. Introduction

The four ventilation configurations were compared with each other for each heat load configuration. Three configurations were run multiple times to assure the repeatability and consistency of the results. One was run six times and two others were run three times each.

The CFD analysis was limited to one heat load scenario that compared the four ventilation configurations using computer software. This data was then compared to experimental results, allowing for the accuracy of the CFD analysis to be assessed.

2. Experiments Run

The experiments that were run in the test room were selected to provide an array of heat loads to the room. This allowed the results to show multiple scenarios and not just an idealized case.

The list of heat loads is shown in Table 4-1, each heat load is run with the four ventilation configurations. This resulted in 52 sets of data. Also, Table 4-2 shows the three experimental setups that were run multiple times. This was done to assure the repeatability of the experiments and to show the influence of the external variables. This added 9 additional sets of data for these repetitions.

Table 4-1: The list of the experimental scenarios to be observed in the experimental room

	AC Speed	Hot Plate Setting	Floor Setting
1	Low	High	Off
2	Low	Low	Off
3	Low	Low	Low
4	Low	Low	High
5	Low	High	Low
6	Low	High	High
7	Low	Off	High
8	High	High	Off
9	High	Low	Off
10	High	Low	Low
11	High	Low	High
12	High	High	Low
13	High	High	High

Table 4-2: The list of experimental setups that were run multiple times.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Number of Iterations
Inlet	Outlet				
Low	High	Low	Low	High	6
Low	High	Low	Low	Off	3
Low	High	Low	High	Off	3

3. Results

Each heat load scenario had four ventilation configurations. The power inputs and the external humidity for each observation are listed in Table 4-3 through Table 4-15. Each table has the data for the four observations with the same heat load configuration. The data shows that although the systems are all on the same setting, the actual power inputted into each set varies. It was observed that the configurations with the low inlet flushed the heat source more quickly, which led to these scenarios having a greater input of heat. This is a result of the sensors in the distributed floor heater and the hot-plate not reaching their cut-off temperature as quickly. Also these scenarios have a higher velocity at the inlet which adds to the speed of the heat being flushed.

Table 4-3: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, high hot-plate input, and nothing from the floor.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	High	Off	0.00	568.34	1248.93	56.97
Low	Low	Low	High	Off	0.00	563.15	1091.53	49.06
High	Low	Low	High	Off	0.00	453.97	831.73	60.89
High	High	Low	High	Off	0.00	472.91	1101.89	52.29

Table 4-4: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, low hot-plate input, and nothing from the floor.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	Low	Off	0.00	348.64	1217.97	43.21
Low	Low	Low	Low	Off	0.00	350.53	1046.43	57.06
High	Low	Low	Low	Off	0.00	199.05	1088.95	51.48
High	High	Low	Low	Off	0.00	240.86	1060.32	58.82

Table 4-5: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, low hot-plate input, and low floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	Low	Low	600.12	286.78	1026.84	58.37
Low	Low	Low	Low	Low	440.96	291.38	1127.59	51.23
High	Low	Low	Low	Low	356.47	209.63	1068.31	59.29
High	High	Low	Low	Low	299.80	213.31	1139.53	52.99

Table 4-6: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, low hot-plate input, and high floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	Low	High	612.20	285.67	1152.74	65.48
Low	Low	Low	Low	High	614.43	252.66	1193.57	63.70
High	Low	Low	Low	High	621.61	191.19	1121.40	64.28
High	High	Low	Low	High	612.33	190.42	1129.75	56.51

Table 4-7: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, high hot-plate input, and low floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	High	Low	290.71	564.70	1085.96	61.17
Low	Low	Low	High	Low	244.52	566.29	1104.84	56.99
High	Low	Low	High	Low	106.37	459.69	1131.45	54.92
High	High	Low	High	Low	167.66	483.83	1074.21	62.80

Table 4-8: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, high hot-plate input, and high floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	High	High	613.84	554.06	1095.71	48.60
Low	Low	Low	High	High	562.02	547.25	1227.93	44.92
High	Low	Low	High	High	519.12	436.07	1116.29	55.18
High	High	Low	High	High	492.73	456.41	1196.85	54.46

Table 4-9: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, high hot-plate input, and nothing from the floor.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	High	High	Off	0.00	563.54	1292.96	62.05
Low	Low	High	High	Off	0.00	578.03	1245.93	61.34
High	Low	High	High	Off	0.00	422.36	1318.63	55.01
High	High	High	High	Off	0.00	429.76	1296.30	53.91

Table 4-10: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, low hot-plate input, and nothing form the floor.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	High	Low	Off	0.00	351.46	1267.50	49.81
Low	Low	High	Low	Off	0.00	191.19	1188.97	57.63
High	Low	High	Low	Off	0.00	182.63	1260.95	51.12
High	High	High	Low	Off	0.00	187.78	1209.12	57.64

Table 4-11: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, low hot-plate input, and low floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	High	Low	Low	371.54	348.68	1219.18	62.50
Low	Low	High	Low	Low	247.62	335.18	1283.77	55.94
High	Low	High	Low	Low	519.08	215.18	1084.60	50.73
High	High	High	Low	Low	335.29	190.17	1225.53	65.53

Table 4-12: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, low hot-plate input, and high floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	High	Low	High	115.85	565.13	1319.23	64.87
Low	Low	High	Low	High	210.73	568.32	1235.88	55.32
High	Low	High	Low	High	86.31	420.57	1323.54	52.12
High	High	High	Low	High	178.51	413.56	1245.58	59.16

Table 4-13: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, high hot-plate input, and low floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	High	High	Low	625.46	335.28	1247.17	59.08
Low	Low	High	High	Low	617.41	327.57	1265.98	64.65
High	Low	High	High	Low	592.29	203.90	1316.38	58.99
High	High	High	High	Low	614.71	208.10	1258.39	67.75

Table 4-14: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for high AC speed, high hot-plate input, and high floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	Low	High	High	High	528.01	536.27	1041.90	62.36
Low	High	High	High	High	500.23	553.78	1274.26	62.47
High	High	High	High	High	553.22	460.60	1172.26	55.71
High	Low	High	High	High	621.10	477.01	1120.67	47.79

Table 4-15: The observed power inputs to the floor heating, hot-plate, and portable AC unit, as well as the observed humidity in the warehouse, outside of the experimental room for low AC speed, no hot-plate input, and high floor input.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Floor Power Input (W)	Hot-Plate Power Input (W)	AC Power Input (W)	Humidity Outside (%)
Inlet	Outlet							
Low	High	Low	Off	High	625.78	0.00	1358.37	41.19
Low	Low	Low	Off	High	630.30	0.00	1229.93	48.75
High	Low	Low	Off	High	630.10	0.00	1426.24	42.02
High	High	Low	Off	High	634.11	0.00	1222.52	49.12

Table 4-16 shows all of the temperature observations for each scenario. The height of each observation is listed across the top of the table. These temperature profiles show the gradient through the height of the room and whether or not a harsh stratification occurred in the room. When this did occur, it allowed the warm air to separate from the cooler air and rise above the occupied zone.

Table 4-16: The temperature measurements for all of the 52 scenarios at the 14 points observed. Each point is the average measurement over the most stable hour of the observation.

Ventilation Configuration		AC Speed	HP Setting	Floor Setting	Temperature Reading (˚C)															
					CH 1-0	CH 1-1	CH 1-2	CH 2-3	CH 1-3	CH 1-4	CH 1-7	CH 2-4	CH 1-5	CH 1-6	CH 2-0	CH 2-1	CH 2-2	CH 2-6/7	Inlet	Outside
Inlet	Outlet				94"	84"	74"	67"	63"	55"	55"	55"	46"	35"	25"	14"	3"	Outlet		
Low	High	Low	High	Off	26.21	25.55	25.19	25.22	24.89	24.73	25.97	25.38	24.64	24.38	24.20	23.63	23.34	25.95	14.40	22.12
Low	Low	Low	High	Off	27.24	26.77	26.04	24.91	23.99	22.87	24.89	24.53	22.53	22.32	22.38	21.85	21.46	24.12	12.64	24.61
High	Low	Low	High	Off	21.08	21.13	21.14	21.27	21.14	20.90	20.39	19.19	20.86	20.65	20.66	20.53	20.34	19.75	7.57	22.18
High	High	Low	High	Off	24.88	24.85	24.70	24.89	24.57	24.41	24.58	23.37	24.53	24.27	24.38	24.23	24.02	24.44	14.14	24.53
Low	High	Low	Low	Off	20.80	20.56	20.48	20.81	20.35	20.24	20.86	20.83	20.23	20.13	20.25	20.29	19.97	21.33	13.84	23.13
Low	Low	Low	Low	Off	20.37	19.34	18.72	18.80	18.42	18.16	19.21	18.83	18.06	17.82	17.77	17.58	17.41	19.29	11.13	21.70
High	Low	Low	Low	Off	18.23	18.08	18.01	18.32	17.95	17.92	18.67	19.31	18.06	18.09	18.35	18.36	18.49	18.86	13.14	24.22
High	High	Low	Low	Off	18.88	18.90	18.95	19.31	18.94	18.78	18.79	17.81	18.73	18.53	18.91	18.80	18.72	18.66	12.00	22.35
Low	High	Low	Low	Low	27.81	27.55	27.43	27.45	26.78	26.03	27.11	26.49	25.12	24.64	24.95	24.97	24.53	27.26	12.96	22.62
Low	Low	Low	Low	Low	29.90	29.65	29.24	29.12	28.66	27.58	28.30	27.97	26.50	25.35	25.58	25.68	25.37	26.00	14.23	24.85
High	Low	Low	Low	Low	24.49	24.56	24.55	24.83	24.52	24.43	24.12	23.30	24.44	24.32	24.48	24.43	24.48	23.61	12.60	22.82
High	High	Low	Low	Low	24.94	24.96	24.95	25.26	24.96	24.82	24.58	23.85	24.81	24.72	24.92	24.87	24.91	24.70	14.31	24.62
Low	High	Low	Low	High	30.77	30.58	30.39	30.18	29.56	28.73	30.12	29.35	28.00	27.61	27.93	28.13	27.88	30.25	15.94	22.76
Low	Low	Low	Low	High	33.11	32.91	32.70	32.77	32.15	30.79	32.07	31.58	29.75	29.21	29.55	29.48	29.08	29.45	14.05	24.21
High	Low	Low	Low	High	30.00	30.07	30.07	30.16	30.08	30.08	29.61	28.43	29.84	29.84	29.98	29.96	30.12	28.72	15.20	23.56
High	High	Low	Low	High	28.88	28.91	28.81	28.99	28.75	28.47	28.38	27.40	28.70	28.48	28.76	28.76	28.85	28.62	15.73	24.20
Low	High	Low	High	Low	29.22	28.76	28.67	28.67	28.27	27.15	28.37	27.65	25.99	25.20	25.39	25.45	25.14	28.25	13.45	22.37
Low	Low	Low	High	Low	31.06	30.75	30.30	30.12	29.61	28.40	29.21	28.10	26.72	25.75	26.17	26.04	25.61	27.46	13.49	23.56
High	Low	Low	High	Low	26.87	26.96	26.94	27.08	26.81	26.70	26.36	25.02	26.82	26.55	26.57	26.41	26.21	25.69	11.71	24.53
High	High	Low	High	Low	26.28	26.21	26.17	26.41	26.15	26.01	25.95	24.76	26.01	25.82	25.88	25.77	25.64	25.51	13.04	22.26
Low	High	Low	High	High	30.87	29.71	29.21	29.09	28.92	28.61	29.63	28.64	28.39	27.74	27.62	27.45	27.07	29.77	9.96	20.55
Low	Low	Low	High	High	37.50	37.27	36.79	36.19	35.19	33.78	35.18	35.73	33.45	33.01	32.93	32.67	32.29	32.70	14.95	25.41
High	Low	Low	High	High	31.10	31.10	31.09	31.22	31.11	30.93	30.74	29.46	30.90	30.67	30.85	30.88	30.83	29.44	12.94	22.69
High	High	Low	High	High	31.83	31.87	31.78	31.94	31.75	31.56	31.25	29.89	31.67	31.35	31.43	31.38	31.36	31.14	16.39	25.25
Low	High	High	High	Off	27.42	27.40	27.57	27.75	27.61	27.25	27.52	26.47	27.27	27.17	26.80	26.24	25.81	27.49	16.59	24.37
Low	Low	High	High	Off	27.41	27.45	27.32	27.16	27.16	27.10	27.50	27.10	27.12	26.92	27.41	27.45	27.32	27.12	15.83	23.57
High	Low	High	High	Off	23.30	23.25	23.28	23.59	23.31	23.32	23.98	24.89	23.56	23.49	23.62	23.62	23.75	24.16	17.09	24.23
High	High	High	High	Off	21.00	20.87	20.98	22.96	21.12	21.18	21.77	23.05	21.35	21.34	21.58	21.53	21.61	22.09	15.79	24.03
Low	High	High	Low	Off	22.59	22.48	22.54	22.75	22.48	22.19	22.75	22.54	22.20	22.10	22.08	22.03	21.95	22.72	15.63	23.21
Low	Low	High	Low	Off	20.75	20.75	20.87	21.17	20.81	20.57	20.77	20.32	20.50	20.29	20.28	20.13	20.09	21.56	13.81	21.49
High	Low	High	Low	Off	20.07	20.11	20.16	20.46	20.14	20.12	20.58	21.25	20.28	20.26	20.37	20.38	20.58	20.85	15.91	24.24
High	High	High	Low	Off	17.50	17.66	17.81	18.42	17.84	17.76	17.97	18.50	17.83	17.84	17.96	18.00	18.38	17.73	13.36	21.19
Low	High	High	Low	Low	26.93	26.93	27.09	27.18	27.09	26.83	26.97	25.90	26.78	26.55	26.49	26.20	25.82	27.24	13.97	22.48
Low	Low	High	Low	Low	27.86	27.86	27.83	27.88	27.69	27.51	27.89	27.03	27.49	27.34	27.08	26.73	26.34	26.62	15.78	24.01
High	Low	High	Low	Low	23.03	23.08	23.14	23.29	23.10	22.92	22.39	21.26	22.95	22.85	22.89	22.90	22.99	21.92	9.62	19.38
High	High	High	Low	Low	22.04	22.00	22.35	22.85	22.64	22.71	23.26	24.49	22.89	22.88	23.04	23.02	23.13	23.50	15.13	21.91
Low	High	High	Low	High	28.77	28.72	28.86	29.04	28.80	28.51	28.78	27.89	28.42	28.18	27.83	27.44	27.09	28.83	17.12	22.74
Low	Low	High	Low	High	28.40	28.36	28.46	28.63	28.38	28.19	28.70	28.63	28.06	27.74	27.33	26.85	26.45	27.93	15.47	24.41
High	Low	High	Low	High	24.36	24.20	24.36	24.74	24.59	24.49	25.33	26.37	24.69	24.75	24.96	24.94	25.04	25.41	17.22	24.69
High	High	High	Low	High	23.35	23.25	23.62	24.18	23.94	24.05	24.68	25.91	24.24	24.21	24.41	24.34	24.38	24.87	15.57	22.08
Low	High	High	High	Low	30.03	30.02	30.10	30.23	29.94	29.78	30.55	29.27	29.66	29.38	29.37	29.26	28.96	30.61	15.70	24.29
Low	Low	High	High	Low	31.65	31.40	30.66	30.23	29.87	29.36	30.38	30.02	29.16	28.91	28.94	28.79	28.48	27.88	13.27	22.50
High	Low	High	High	Low	30.55	30.63	30.67	30.84	30.58	30.51	29.92	28.63	30.61	30.38	30.37	30.45	30.66	29.42	17.31	24.22
High	High	High	High	Low	29.61	29.71	29.74	29.93	29.75	29.62	29.04	27.92	29.62	29.45	29.55	29.64	29.81	29.71	16.10	21.81
Low	Low	High	High	High	37.96	37.65	37.19	36.76	36.01	34.44	35.76	35.17	33.69	33.16	33.42	33.51	33.24	34.79	18.96	23.64
Low	High	High	High	High	35.72	35.36	35.13	35.07	34.53	33.39	34.83	34.24	32.50	31.92	32.07	32.29	32.06	34.73	18.07	23.29
High	High	High	High	High	29.62	29.66	29.67	29.85	29.75	29.56	29.00	27.39	29.54	29.22	29.31	29.29	29.29	29.16	11.04	23.38
High	Low	High	High	High	28.05	28.22	28.28	28.37	28.26	28.09	27.39	25.60	28.12	27.87	27.90	27.78	27.77	26.29	10.70	20.73
Low	High	Low	Off	High	27.12	26.60	26.25	26.10	25.58	24.86	26.61	25.63	26.61	24.87	25.14	25.33	25.21	27.08	15.73	24.97
Low	Low	Low	Off	High	27.09	26.96	26.42	26.12	25.58	24.95	26.64	26.28	24.51	24.50	24.70	24.94	24.88	24.18	15.14	21.10
High	Low	Low	Off	High	28.70	28.78	28.92	29.12	28.93	28.71	28.05	27.29	28.73	28.73	28.75	28.85	29.27	27.80	18.14	25.58
High	High	Low	Off	High	25.20	25.30	25.34	25.81	25.32	25.12	24.56	23.89	25.17	25.14	25.28	25.43	25.84	24.73	14.42	21.16

4. Energy Balance

An energy balance was calculated for each experimental run. This was done to assure that all the energy going in and coming out of the experimental room was accounted for. The three flows of energy considered were the power input to the floor, the flow of air through the experimental room, and the losses through the floor and walls.

The power input was monitored for the hotplate and the distributed floor heat with a power monitor. It was assumed that both were 100% efficient and all the power going in was transferred to heat.

The heat loss from the air flow through the test chamber was calculated using

$$H = Q\rho C_p(T_{Outlet} - T_{Inlet}) \tag{1}$$

where Q is the volume flux of air through the chamber, ρ is the density of air (taken to be 1.225 kg/m^3), C_p is the specific heat of air (1.0035 kJ/kg.K) and T_{Outlet} and T_{Inlet} are the temperatures observed at the outlet and inlet, respectively. The value used for the specific heat capacity assumes that the air is dry and at sea level. The combined effects of these accounting for humidity and altitude would alter C_p by less than 1% and were therefore ignored. The flow rate was measured at the inlet to the room by measuring the velocity at a range of points along a pair of vertical and horizontal lines passing through the center of the vent inlet. The flow rate was then calculated by numerical integration of the velocity profile,

$$Q \approx \sum udA \tag{2}$$

where u is the average velocity measured at a particular radial location and dA is the area of a circular strip centered on the radial distance at which the velocities were measured.

The difference between the heat lost due to air flow and the input energy should be the amount of energy lost through the walls and floor. This is a result of the conduction through the wall and the convection around the exterior surface of the room. Also the radiation from the experimental room to the walls of the lab could contribute to these losses. Since the lab was a conditioned space there should be only very small variations in surface temperatures throughout the space and, therefore, losses from radiation were considered negligible, and ignored.

The losses through the walls of the experimental room were calculated two different ways. The first correction for losses, L, was calculated using

$$L = AU * \Delta T \tag{3}$$

This equation uses a calibration factor that was found by heating the experimental room up to steady state and observing the heat required to maintain that steady state. The calibration factor is simply the average W/K needed to maintain that steady state at different ΔT. This AU value was found to be 37.5 W/K. The issue with this method is the assumption that the relationship between the ΔT and the energy transfer is linear. In actuality it is more of a logarithmic relationship. When the calibration factor was calculated the ΔT was between 8 and 15K. The experimental run ΔT varied from 0 to 12K, typically on the lower end.

To attempt to correct for this, the losses were also calculated using

$$L = A_C U_C * \Delta T_C + A_W U_W * \Delta T_W + A_F U_F * \Delta T_F \tag{4}$$

where the indices *C*, *W*, and *F* refer to Ceiling, Walls, and Floor, respectively. The R-values of each wall and the surface area were used to calculate the AU. The R-value is a characteristic of each material used in the wall. These values are available from the manufacturer. The total R-value for each section of the wall is found by adding all of the R-values of the materials in the wall and the additional R-value for the air that hugs each side of the wall. The inverse of this total R-value is the U, and each U times its area gives the AU for that wall. The ceiling AU was found to be 0.442 W/K, the wall AU was found to be 2.023 W/K, and the floor AU was found to be 0.456 W/K. The factors were broken up in this fashion because of the difference in temperature in the experimental room. In some cases the ceiling had a higher ΔT than the floor, affecting its heat loss. The units of the AU were converted to W/K since R-values are given in Imperial units in the US. The U was found to be 0.442 W/K. This reduced the average error from 28.1% to 19.8%.

Each scenario and its resultant energy balance using Equation 3 can be seen in Table 4-17 and Table 4-18. The same balance using Equation 4 can be seen in Table 4-19 and Table 4-20. These balances show a substantial amount of energy leaving the experimental room that is not accounted for entering the room. This is assumed to be a result of the losses varying drastically with the varying outdoor conditions. The temperature difference may remain the same but the temperature outside the warehouse will change the rate the energy radiates to the outside. Additionally, the thermal mass of the experimental room will accumulate or dissipate heat if a true steady state is not reached. In many of the experimental runs the temperature is slowing dropping a couple degrees over a few hours. This drop in temperature would be paralleled with the thermal mass of the room releasing energy to the space. That heat would be leaving but not be accounted for entering the experiment.

Table 4-17: The calculated energy balance using Equation 3 for all scenarios with low AC fan speed

Ventilation Configuration		AC	Hot Plate	Floor	P in (watts)	Energy Flow Out (watts)	Losses (watts)	Balance (watts)	Percent Error
Inlet	Outlet								
Low	High	Low	High	Off	568.34	720.85	103.08	-255.60	45.0%
Low	Low	Low	High	Off	563.15	716.22	-23.60	-129.46	23.0%
High	Low	Low	High	Off	453.97	672.97	-55.10	-163.91	36.1%
High	High	Low	High	Off	472.91	569.06	-3.61	-92.53	19.6%
Low	High	Low	Low	Off	348.64	467.34	-100.58	-18.12	5.2%
Low	Low	Low	Low	Off	350.53	509.16	-119.99	-38.65	11.0%
High	Low	Low	Low	Off	199.05	316.40	-222.10	104.75	52.6%
High	High	Low	Low	Off	240.86	368.02	-133.98	6.82	2.8%
Low	High	Low	Low	Low	886.90	892.52	135.13	-140.75	15.9%
Low	Low	Low	Low	Low	732.34	734.84	103.58	-106.08	14.5%
High	Low	Low	Low	Low	566.10	608.64	58.64	-101.18	17.9%
High	High	Low	Low	Low	513.11	573.68	7.15	-67.72	13.2%
Low	High	Low	Low	High	897.86	893.65	240.34	-236.12	26.3%
Low	Low	Low	Low	High	867.09	961.35	260.84	-355.10	41.0%
High	Low	Low	Low	High	812.80	747.52	236.41	-171.13	21.1%
High	High	Low	Low	High	802.74	712.37	166.03	-75.65	9.4%
Low	High	Low	High	Low	855.41	923.55	182.17	-250.31	29.3%
Low	Low	Low	High	Low	810.81	871.90	177.75	-238.84	29.5%
High	Low	Low	High	Low	566.05	772.36	76.24	-282.54	49.9%
High	High	Low	High	Low	651.49	689.04	137.30	-174.85	26.8%
Low	High	Low	High	High	1167.89	1236.57	305.19	-373.87	32.0%
Low	Low	Low	High	High	1109.27	1107.70	351.01	-349.44	31.5%
High	Low	Low	High	High	955.20	911.42	305.62	-261.85	27.4%
High	High	Low	High	High	949.14	815.22	233.04	-99.12	10.4%
Low	High	Low	Off	High	625.78	708.74	32.70	-115.66	18.5%
Low	Low	Low	Off	High	630.30	564.10	171.15	-104.95	16.6%
High	Low	Low	Off	High	630.10	534.08	116.32	-20.31	3.2%
High	High	Low	Off	High	634.11	569.79	151.11	-86.79	13.7%

Table 4-18: The calculated energy balance using Equation 3 for all scenarios with high AC fan speed

Ventilation Configuration		AC	Hot Plate	Floor	P in (watts)	Energy Flow Out (watts)	Losses (watts)	Balance (watts)	Percent Error
Inlet	Outlet								
Low	High	High	High	Off	563.54	736.40	102.33	-275.19	48.8%
Low	Low	High	High	Off	578.03	762.57	138.40	-322.94	55.9%
High	Low	High	High	Off	422.36	449.46	-23.06	-4.04	1.0%
High	High	High	High	Off	429.76	399.76	-92.51	122.51	28.5%
Low	High	High	Low	Off	351.46	479.09	-31.73	-95.90	27.3%
Low	Low	High	Low	Off	191.19	523.10	-34.89	-297.01	155.3%
High	Low	High	Low	Off	182.63	313.75	-145.29	14.16	7.8%
High	High	High	Low	Off	187.78	277.32	-121.28	31.74	16.9%
Low	High	High	Low	Low	720.22	897.02	157.47	-334.27	46.4%
Low	Low	High	Low	Low	582.80	732.28	127.98	-277.46	47.6%
High	Low	High	Low	Low	734.26	781.28	129.33	-176.35	24.0%
High	High	High	Low	Low	525.46	532.01	35.99	-42.53	8.1%
Low	High	High	Low	High	680.98	791.03	209.73	-319.78	47.0%
Low	Low	High	Low	High	779.05	841.41	135.19	-197.55	25.4%
High	Low	High	Low	High	506.88	520.58	5.35	-19.05	3.8%
High	High	High	Low	High	592.07	591.06	79.35	-78.34	13.2%
Low	High	High	High	Low	960.74	1007.39	204.15	-250.81	26.1%
Low	Low	High	High	Low	944.98	987.06	274.95	-317.03	33.5%
High	Low	High	High	Low	796.19	768.88	230.70	-203.38	25.5%
High	High	High	High	Low	822.81	864.48	288.21	-329.88	40.1%
Low	Low	High	High	High	1064.28	1069.65	434.69	-440.07	41.3%
Low	High	High	High	High	1054.01	1125.45	393.19	-464.63	44.1%
High	High	High	High	High	1013.81	1151.00	222.77	-359.95	35.5%
High	Low	High	High	High	1098.11	990.30	265.97	-158.16	14.4%

Table 4-19: The calculated energy balance using Equation 4 for all scenarios with low AC fan speed

Ventilation Configuration		AC	Hot Plate	Floor	P in (watts)	Energy Flow Out (watts)	Losses (watts)	Energy Balance (watts)	Percent Error
Inlet	Outlet								
Low	High	Low	High	Off	568.34	720.85	7.92	-160.44	28.2%
Low	Low	Low	High	Off	563.15	716.22	-1.55	-151.52	26.9%
High	Low	Low	High	Off	453.97	672.97	-4.30	-214.71	47.3%
High	High	Low	High	Off	472.91	569.06	-0.28	-95.87	20.3%
Low	High	Low	Low	Off	348.64	467.34	-7.89	-110.81	31.8%
Low	Low	Low	Low	Off	350.53	509.16	-9.02	-149.62	42.7%
High	Low	Low	Low	Off	199.05	316.40	-17.24	-100.11	50.3%
High	High	Low	Low	Off	240.86	368.02	-10.41	-116.74	48.5%
Low	High	Low	Low	Low	886.90	892.52	10.46	-16.08	1.8%
Low	Low	Low	Low	Low	732.34	734.84	8.06	-10.56	1.4%
High	Low	Low	Low	Low	566.10	608.64	4.66	-47.20	8.3%
High	High	Low	Low	Low	513.11	573.68	0.66	-61.22	11.9%
Low	High	Low	Low	High	897.86	893.65	18.84	-14.62	1.6%
Low	Low	Low	Low	High	867.09	961.35	20.23	-114.49	13.2%
High	Low	Low	Low	High	812.80	747.52	18.59	46.69	5.7%
High	High	Low	Low	High	802.74	712.37	13.15	77.23	9.6%
Low	High	Low	High	Low	855.41	923.55	14.12	-82.26	9.6%
Low	Low	Low	High	Low	810.81	871.90	13.84	-74.94	9.2%
High	Low	Low	High	Low	566.05	772.36	5.92	-212.22	37.5%
High	High	Low	High	Low	651.49	689.04	10.72	-48.27	7.4%
Low	High	Low	High	High	1167.89	1236.57	24.00	-92.68	7.9%
Low	Low	Low	High	High	1109.27	1107.70	27.42	-25.84	2.3%
High	Low	Low	High	High	955.20	911.42	23.92	19.86	2.1%
High	High	Low	High	High	949.14	815.22	18.26	115.66	12.2%
Low	High	Low	Off	High	625.78	708.74	2.82	-85.78	13.7%
Low	Low	Low	Off	High	630.30	564.10	13.61	52.60	8.3%
High	Low	Low	Off	High	630.10	534.08	9.34	86.68	13.8%
High	High	Low	Off	High	634.11	569.79	12.08	52.25	8.2%

Table 4-20: The calculated energy balance using Equation 4 for all scenarios with high AC fan speed

Ventilation Configuration		AC	Hot Plate	Floor	P in (watts)	Energy Flow Out (watts)	Losses (watts)	Energy Balance (watts)	Percent Error
Inlet	Outlet								
Low	High	High	High	Off	563.54	736.40	7.53	-180.39	32.0%
Low	Low	High	High	Off	578.03	762.57	10.87	-195.41	33.8%
High	Low	High	High	Off	422.36	449.46	-1.87	-25.22	6.0%
High	High	High	High	Off	429.76	399.76	-7.44	37.44	8.7%
Low	High	High	Low	Off	351.46	479.09	-2.55	-125.08	35.6%
Low	Low	High	Low	Off	191.19	523.10	-2.85	-329.06	172.1%
High	Low	High	Low	Off	182.63	313.75	-11.35	-119.77	65.6%
High	High	High	Low	Off	187.78	277.32	-9.46	-80.08	42.6%
Low	High	High	Low	Low	720.22	897.02	11.99	-188.78	26.2%
Low	Low	High	Low	Low	582.80	732.28	9.67	-159.14	27.3%
High	Low	High	Low	Low	734.26	781.28	10.23	-57.26	7.8%
High	High	High	Low	Low	525.46	532.01	2.55	-9.10	1.7%
Low	High	High	Low	High	680.98	791.03	15.96	-126.01	18.5%
Low	Low	High	Low	High	779.05	841.41	9.99	-72.35	9.3%
High	Low	High	Low	High	506.88	520.58	0.30	-14.00	2.8%
High	High	High	Low	High	592.07	591.06	5.89	-4.88	0.8%
Low	High	High	High	Low	960.74	1007.39	15.68	-62.33	6.5%
Low	Low	High	High	Low	944.98	987.06	21.60	-63.68	6.7%
High	Low	High	High	Low	796.19	768.88	18.18	9.13	1.1%
High	High	High	High	Low	822.81	864.48	22.65	-64.31	7.8%
Low	Low	High	High	High	1064.28	1069.65	34.16	-39.53	3.7%
Low	High	High	High	High	1054.01	1125.45	30.70	-102.15	9.7%
High	High	High	High	High	1013.81	1151.00	17.47	-154.66	15.3%
High	Low	High	High	High	1098.11	990.30	20.79	87.02	7.9%
Low	High	Low	Off	High	625.78	708.74	2.82	-85.78	13.7%
Low	Low	Low	Off	High	630.30	564.10	13.61	52.60	8.3%
High	Low	Low	Off	High	630.10	534.08	9.34	86.68	13.8%
High	High	Low	Off	High	634.11	569.79	12.08	52.25	8.2%

5. Profiles

The temperature profiles for each set of four ventilation configurations were plotted for comparison. Each plot shows the four profiles on top of each other and displays the temperature gradient that is achieved. While observing the profiles, it should be pointed out that the power input is consistently higher when the inlet is low. This is because the heat inputs were both temperature controlled rather than power level controlled. Therefore, the higher the inlet velocity and the closer the inlet is to the heating elements, the greater the ability of the air flow to flush heat off the floor and the more energy is required to maintain the desired temperature. The increase in energy put into the system resulted in a higher overall temperature in some cases.

Figure 4-1 shows the temperature profiles for the four ventilation configurations with the AC fan on low, the hot-plate on low, and the distributed floor turned off, and Table 4-21 shows the calculated energy balance for each of these configurations. In this case, there is about 100 W of additional heat going into the two configurations with the low inlet. The low inlet, high outlet configuration did result in a higher overall temperature, but the low inlet, low outlet configuration was able to maintain about the same average temperature even with the higher input heat.

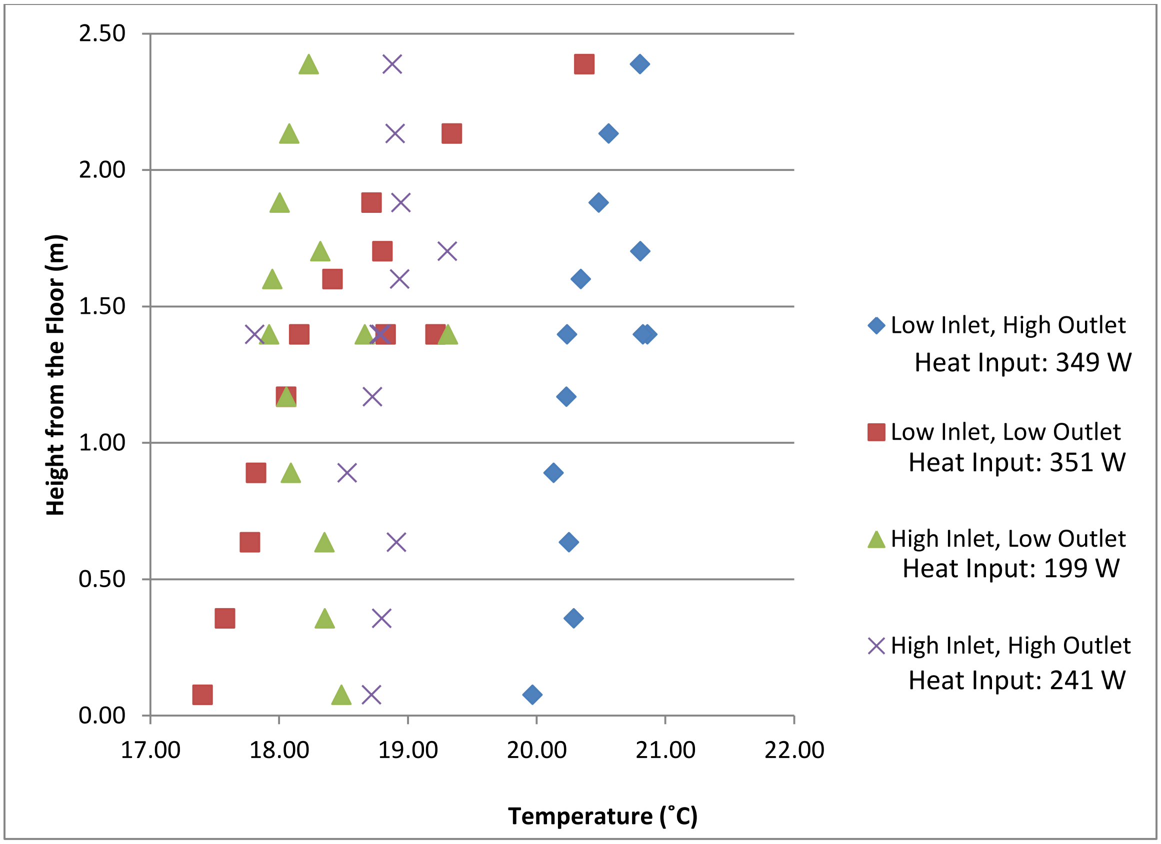


Figure 4-1: The temperature profile for the four ventilation configurations with low AC speed, low hot-plate input, and no distributed floor heat. Here the low inlet, low outlet configuration seems to outperform the others.

Table 4-21: The percent error calculated for each configuration with the AC fan on low, the hot plate on low, and the distributed floor turned off

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	31.8%
Low	Low	42.7%
High	Low	50.3%
High	High	48.5%

Figure 4-2 shows the temperature profiles of the four ventilation configurations with the AC fan on low, hot-plate on high, and the distributed floor off, and Table 4-22 shows the calculated energy balance for each of these cases. This configuration like the one in Figure 4-1 had an additional 100W of power input when the inlet was low. Here you see that both low inlet configurations were able to maintain a temperature comparable to the high inlet, high outlet configuration. However, the high inlet, low outlet was able to bring the overall temperature down a couple of degrees. Also, the low inlet, low outlet configuration was able to achieve a sharp stratification at about 1.6m above the ground. This shows that the desired energy saving stratification can be achieved in a single story environment.

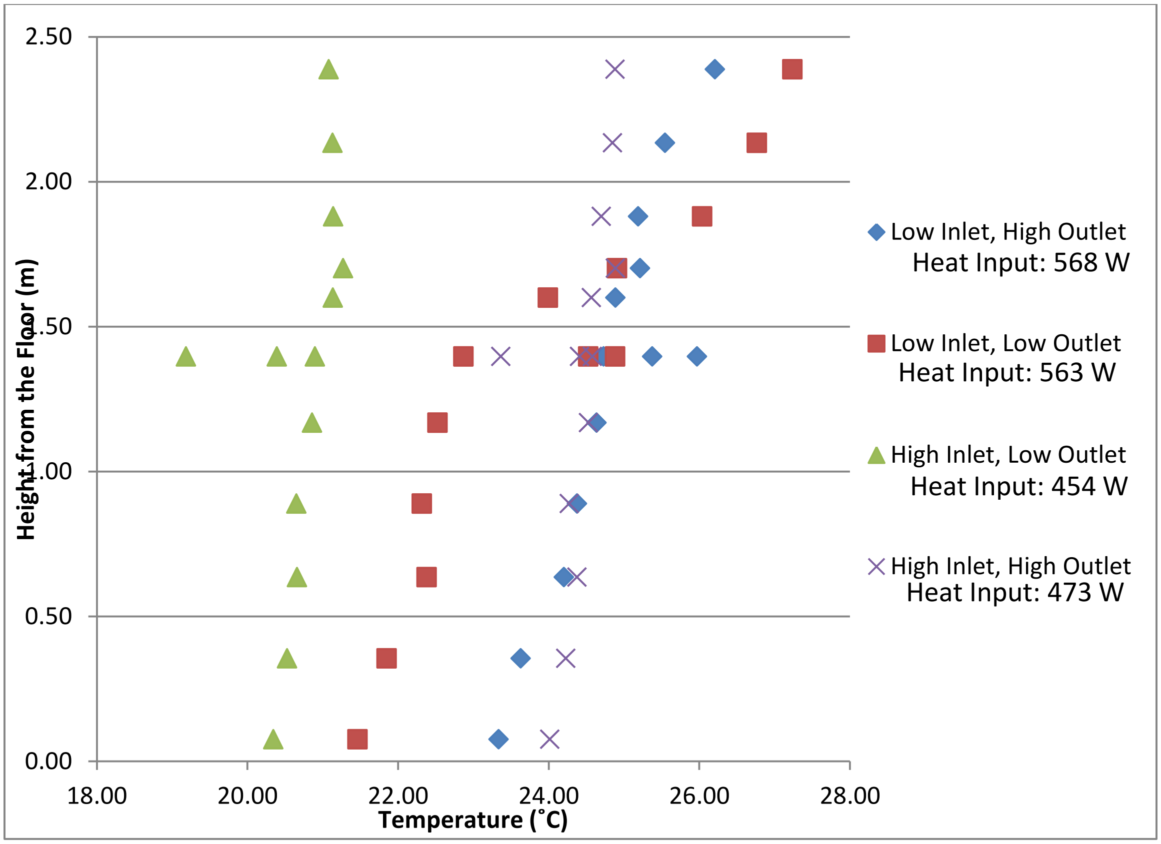


Figure 4-2: The temperature profile for the four ventilation configurations with low AC speed, high hot-plate input, and no distributed floor heat. Here the only configuration that was able to achieve a step in the temperature profile was the low inlet, low outlet.

Table 4-22: The percent error calculated for each configuration with the AC fan on low, the hot plate on high, and the distributed floor turned off

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	28.2%
Low	Low	26.9%
High	Low	47.3%
High	High	20.3%

Figure 4-3 shows the temperature profiles of the four ventilation configurations with the AC fan on low, hot-plate on low, and the distributed floor on low, and Table 4-23 shows the calculated energy balance for each of these configurations. In these scenarios, the low inlet configurations took about 80 additional watts off of the hot plate and about 200 additional watts. However, the lower region of the room is about the same temperature as the high inlet configurations. Also, the power input to the low inlet, high outlet configuration is about 150 watts greater than the low inlet, low outlet. Regardless of the greater load, the low inlet, high outlet configuration has a lower temperature profile.

Also, the two low inlet configurations show the sharp temperature stratification. When properly executed this provides the desired reduction in conditioned space.

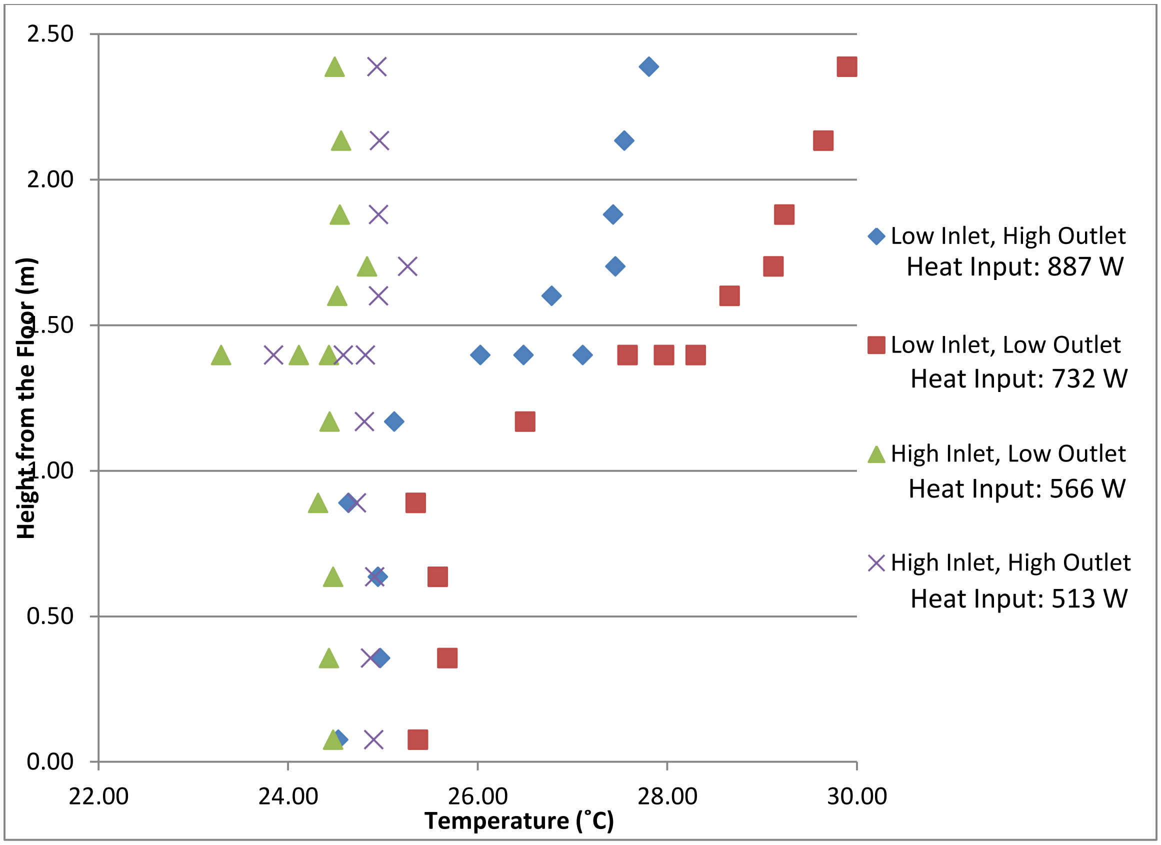


Figure 4-3: The temperature profile for the four ventilation configurations with low AC speed, low hot-plate input, and low distributed floor heat. Here the low inlet, high outlet configuration is able to take a much larger heat load but is able to provide approximately the same temperature in the occupied zone.

Table 4-23: The percent error calculated for each configuration with the AC fan on low, the hot plate on low, and the distributed floor on low

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	1.8%
Low	Low	1.4%
High	Low	8.3%
High	High	11.9%

Figure 4-4 shows the temperature profiles of the four ventilation configurations with the AC fan on low, hot-plate on high, and the distributed floor on low, and Table 4-23 shows the calculated energy balance for each of these configurations. The heat load of the Low inlet configurations is between 200 and 300 W greater than the high inlet configurations. However, the lower occupied region is again approximately the same temperature as the high inlet configurations, and the heat is restricted to the upper region. The low inlet, high outlet configuration was able to flush the upper heat and maintain an overall lower temperature than the low inlet, low outlet even with an additional 50 watt load.

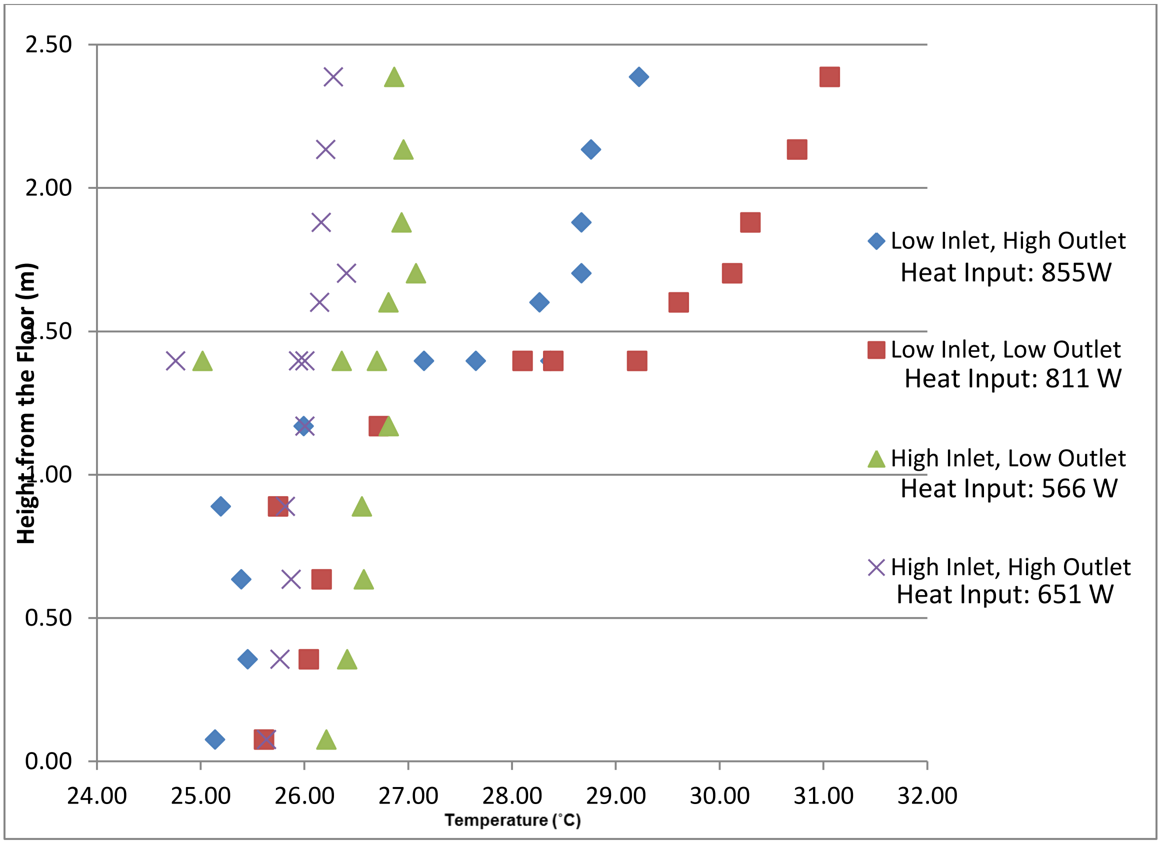


Figure 4-4: The temperature profile for the four ventilation configurations with low AC speed, high hot-plate input, and low distributed floor heat. Here the low inlet, high outlet configuration is able to handle a larger heat load than the other configurations but provides a cooler occupied region.

Table 4-24: The percent error calculated for each configuration with the AC fan on low, the hot plate on high, and the distributed floor on low

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	9.6%
Low	Low	9.2%
High	Low	37.5%
High	High	7.4%

Figure 4-5 shows the temperature profiles of the four ventilation configurations with the AC fan on low, hot-plate on low, and the distributed floor on high, and Table 4-25 shows the calculated energy balance for each of these configurations. In these scenarios there is only about a 50 watt increase in the low inlet, low outlet configuration over the high inlet scenarios, and a 100watt increase with the low inlet, high outlet. Again it is observed that the occupied temperature is the same or lower with the low inlet. The low inlet, high outlet was able to cool the occupied space more than all other configurations and keep the higher region only 2˚C higher. This led to an overall similar temperature with a higher heat load and a cooler occupied region.

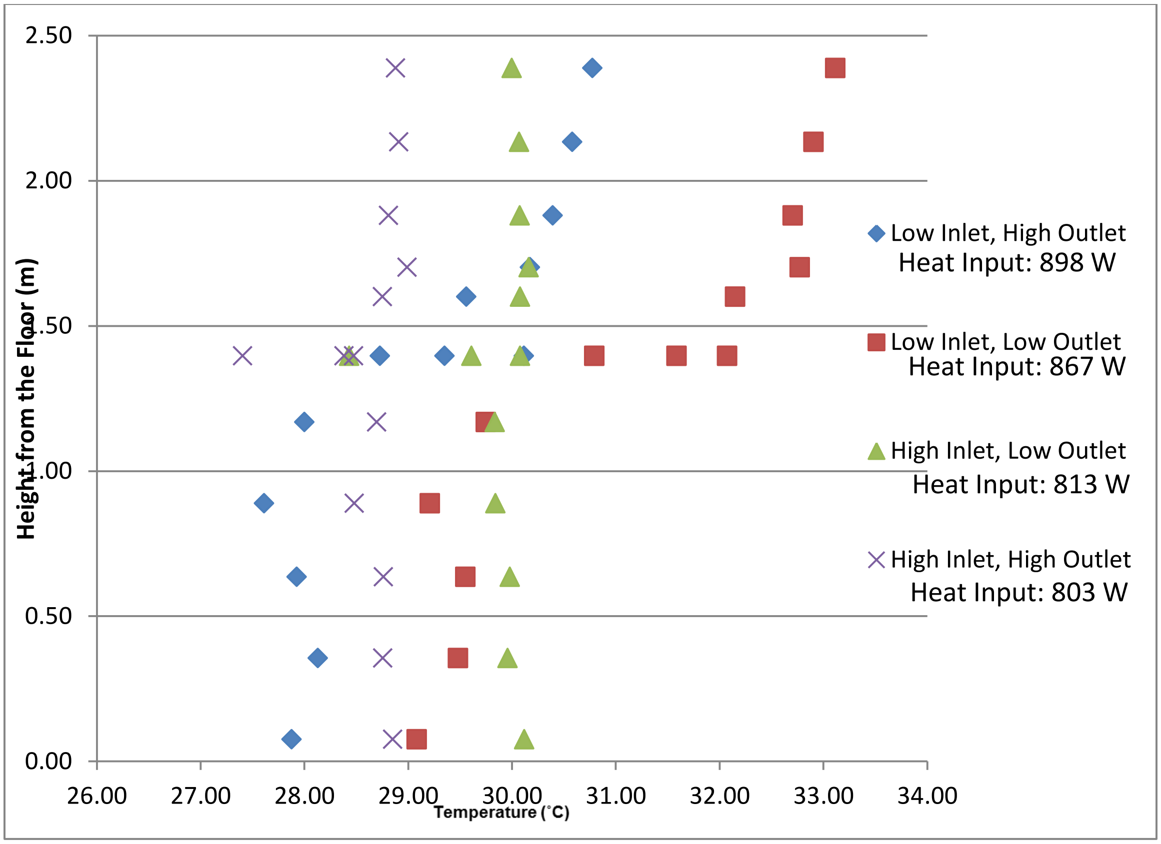


Figure 4-5: The temperature profile for the four ventilation configurations with low AC speed, low hot-plate input, and high distributed floor heat. Here the low inlet, high outlet configuration was able to absorb a slightly larger heat load while providing an occupied temperature cooler than the other configurations.

Table 4-25: The percent error calculated for each configuration with the AC fan on low, the hot plate on low, and the distributed floor on high

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	1.6%
Low	Low	13.2%
High	Low	5.7%
High	High	9.6%

Figure 4-6 shows the temperature profiles of the four ventilation configurations with the AC fan on low, hot-plate on high, and the distributed floor on high, and Table 4-26 shows the calculated energy balance for each of these configurations. The low inlet, high outlet had an additional load of about 200 watts; while the low inlet, low outlet configuration had about 150 watts of additional load. The low inlet, low outlet configuration was able to respond to the higher load by forcing the heat into the upper region and maintain only a 2˚C increase in the lower region. However, the low inlet, high outlet configuration was able to cool the space about 3˚C lower than the high inlet configurations. It also did this without creating a harsh stratification.

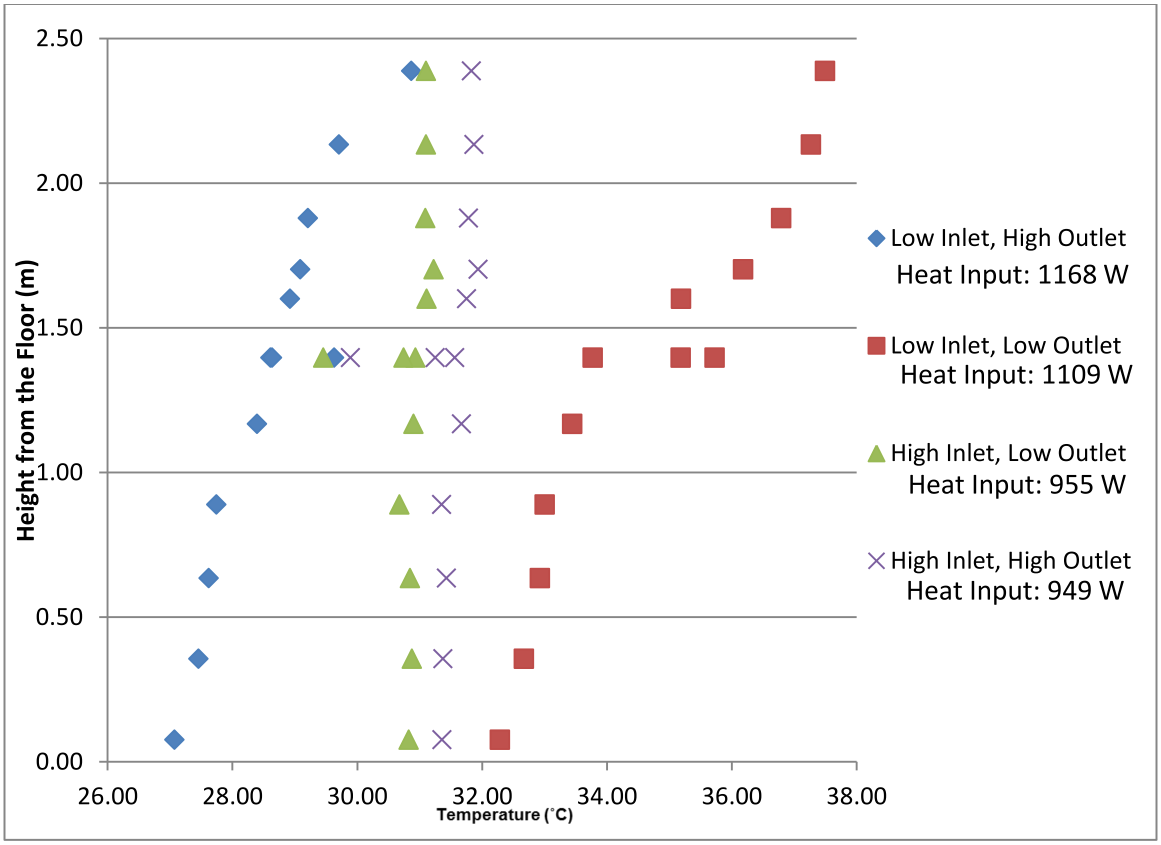


Figure 4-6: The temperature profile for the four ventilation configurations with low AC speed, high hot-plate input, and high distributed floor heat. Here the low inlet, low outlet configuration is able to flush more heat from the floor heating elements and provides a cooler temperature throughout its profile.

Table 4-26: The percent error calculated for each configuration with the AC fan on low, the hot plate on high, and the distributed floor on high

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	7.9%
Low	Low	2.3%
High	Low	2.1%
High	High	12.2%

Figure 4-7 shows the temperature profiles of the four ventilation configurations with the AC fan on low, the hot-plate off, and the distributed floor on high, and Table 4-27 shows the calculated energy balance for each of these cases. These configurations resulted in approximately the same heat load. Both low inlet configurations resulted in similar occupied temperatures to the high inlet, high outlet. However, the high inlet, low outlet configuration was about 4˚C higher than the other three scenarios.

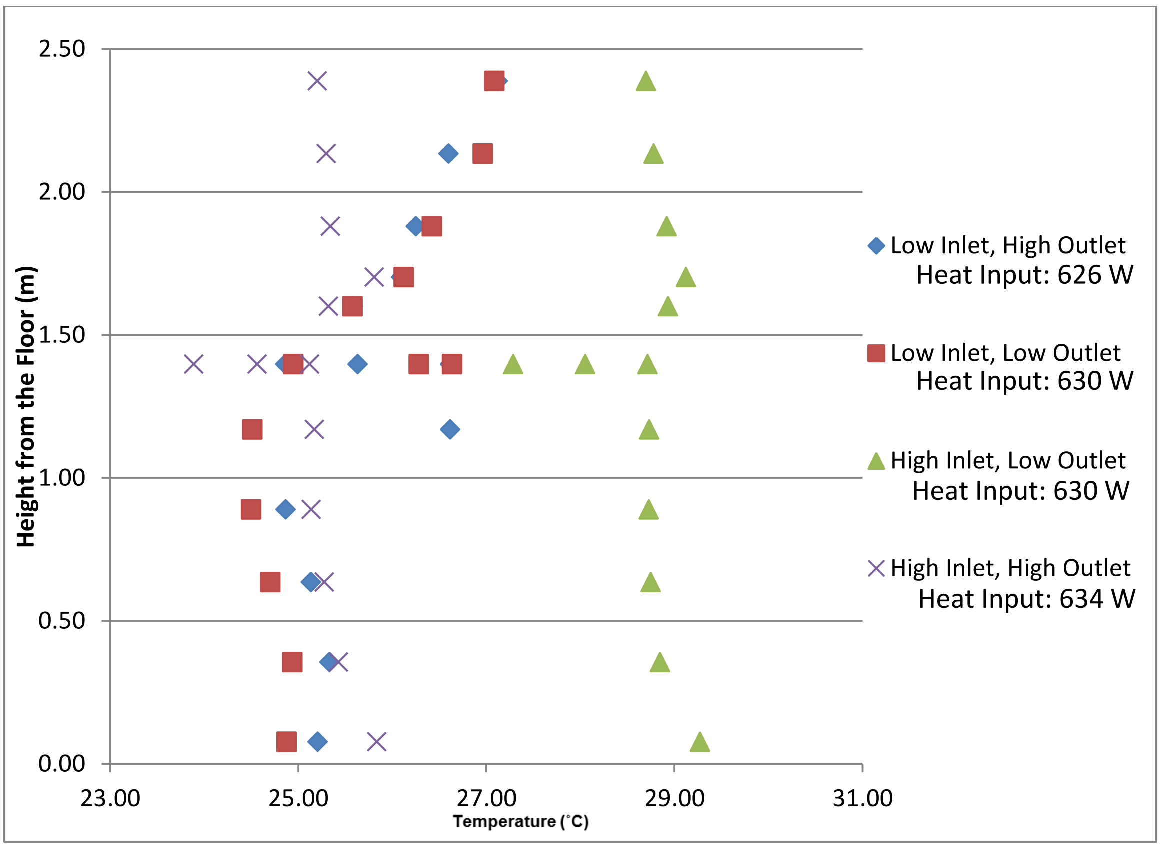


Figure 4-7: The temperature profile for the four ventilation configurations with low AC speed, no hot-plate input, and high distributed floor heat. Here the heat loads were nearly the same for all cases, and the low inlet configurations and the high inlet, high outlet configuration were all able to provide about the same occupied conditions.

Table 4-27: The percent error calculated for each configuration with the AC fan on low, the hot plate turned off, and the distributed floor on high

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	13.7%
Low	Low	8.3%
High	Low	13.8%
High	High	8.2%

Figure 4-8 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on low, and the distributed floor off, and Table 4-28 shows the calculated energy balance for each of these configurations. The low inlet, high outlet configuration had an additional load of approximately 150 watts over the other three configurations. No harsh stratification was achieved and the low inlet, high outlet configuration resulted in a higher overall temperature in response to the additional load. Also, the high inlet, high outlet configuration was able to maintain a lower temperature when compared to the other scenarios with the same load.

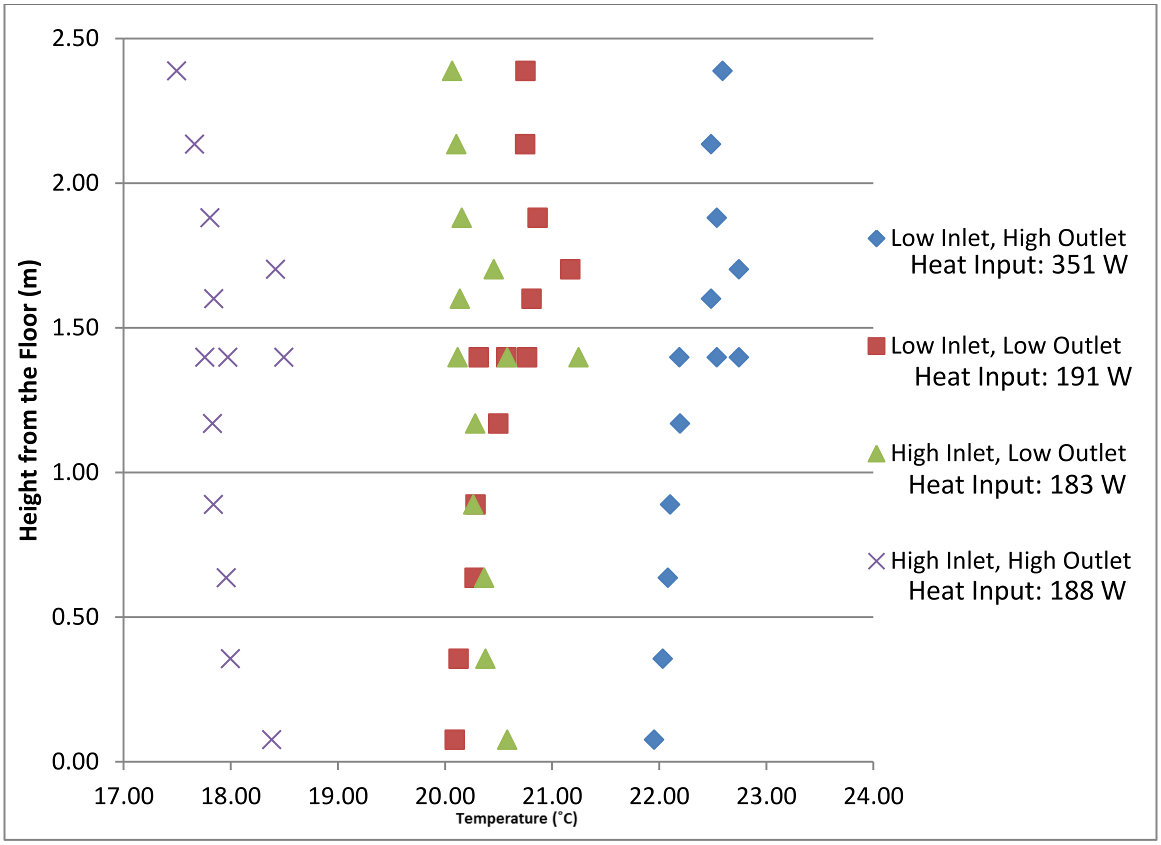


Figure 4-8: The temperature profile for the four ventilation configurations with high AC speed, low hot-plate input, and no distributed floor heat. Here the low inlet, high outlet configuration took almost double the heat load from the heating elements, but only had about a two degree increase in the temperature of the occupied region. Also the high inlet, high outlet was able to outperform the low, low and high low configurations.

Table 4-28: The percent error calculated for each configuration with the AC fan on high, the hot plate on low, and the distributed floor turned off

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	35.6%
Low	Low	172.1%
High	Low	65.6%
High	High	42.6%

Figure 4-9 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on high, and the distributed floor off, and Table 4-29 shows the calculated energy balance for each of these configurations. Both low inlet configurations had approximately 150 additional watts in heat load when compared to the high inlet configurations. No harsh stratification was achieved, resulting in a higher temperature in the low inlet configurations. Again the high inlet, high outlet configuration was able to cool the room more than the high inlet, low outlet configuration with the same heat load.

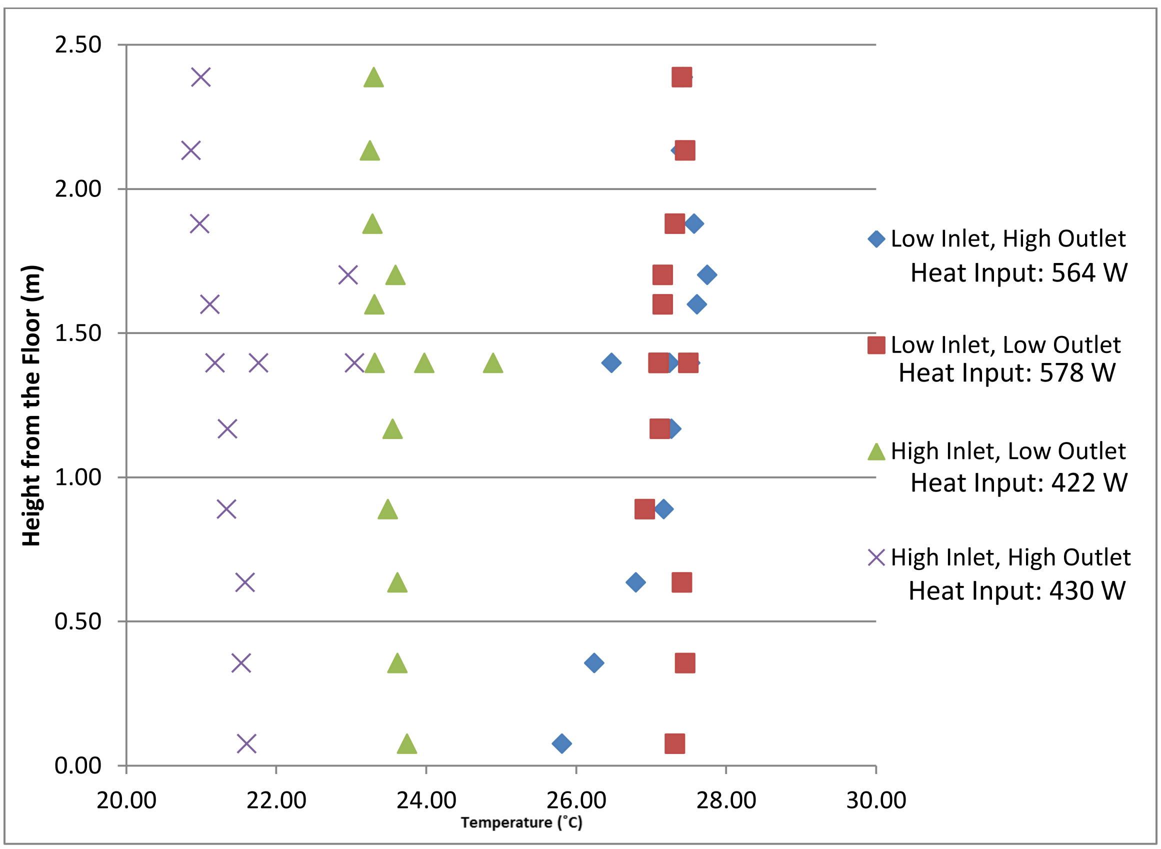


Figure 4-9: The temperature profile for the four ventilation configurations with high AC speed, high hot-plate input, and no distributed floor heat. Here the additional heat load for the low inlet configurations resulted in a higher temperature across the profile.

Table 4-29: The percent error calculated for each configuration with the AC fan on high, the hot plate on high, and the distributed floor turned off

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	32.0%
Low	Low	33.8%
High	Low	6.0%
High	High	8.7%

Figure 4-10 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on low, and the distributed floor on low, and Table 4-30 shows the calculated energy balance for each of these configurations. Here the heat loads were more complicated. The point source load was greater on the low inlet configurations, but the distributed load was greater on the high inlet, low outlet configuration. The two displaced configurations had approximately the same overall load, as did the two same level configurations. In both comparisons, the high inlet configurations were able to cool the room further.

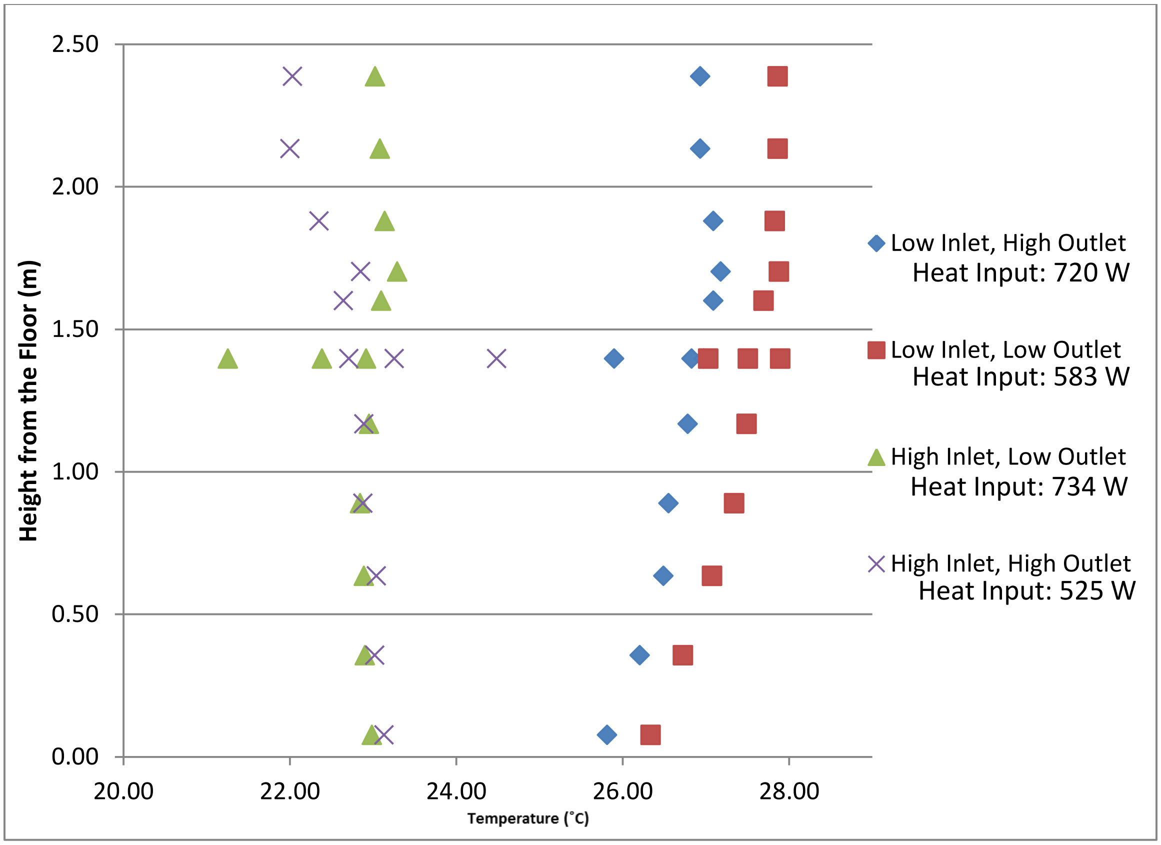


Figure 4-10: The temperature profile for the four ventilation configurations with high AC speed, low hot-plate input, and low distributed floor heat. Here the high inlet configurations were able to outperform the low inlet configurations.

Table 4-30: The percent error calculated for each configuration with the AC fan on high, the hot plate on low, and the distributed floor on low

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	26.2%
Low	Low	27.3%
High	Low	7.8%
High	High	1.7%

Figure 4-11 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on high, and the distributed floor on low, and Table 4-31 shows the calculated energy balance for each of these configurations. This scenario resulted in approximately 120 additional watts in the low inlet configurations. Both were able to at least slightly cool the space more than the high inlet configurations. The low inlet, high outlet configuration was able to do this without a harsh stratification.

It should also be noted that the high inlet configurations resulted in a much warmer temperature at the elevations closest to the heat source. The low inlet configurations were able to more quickly respond to the floor heat loads.

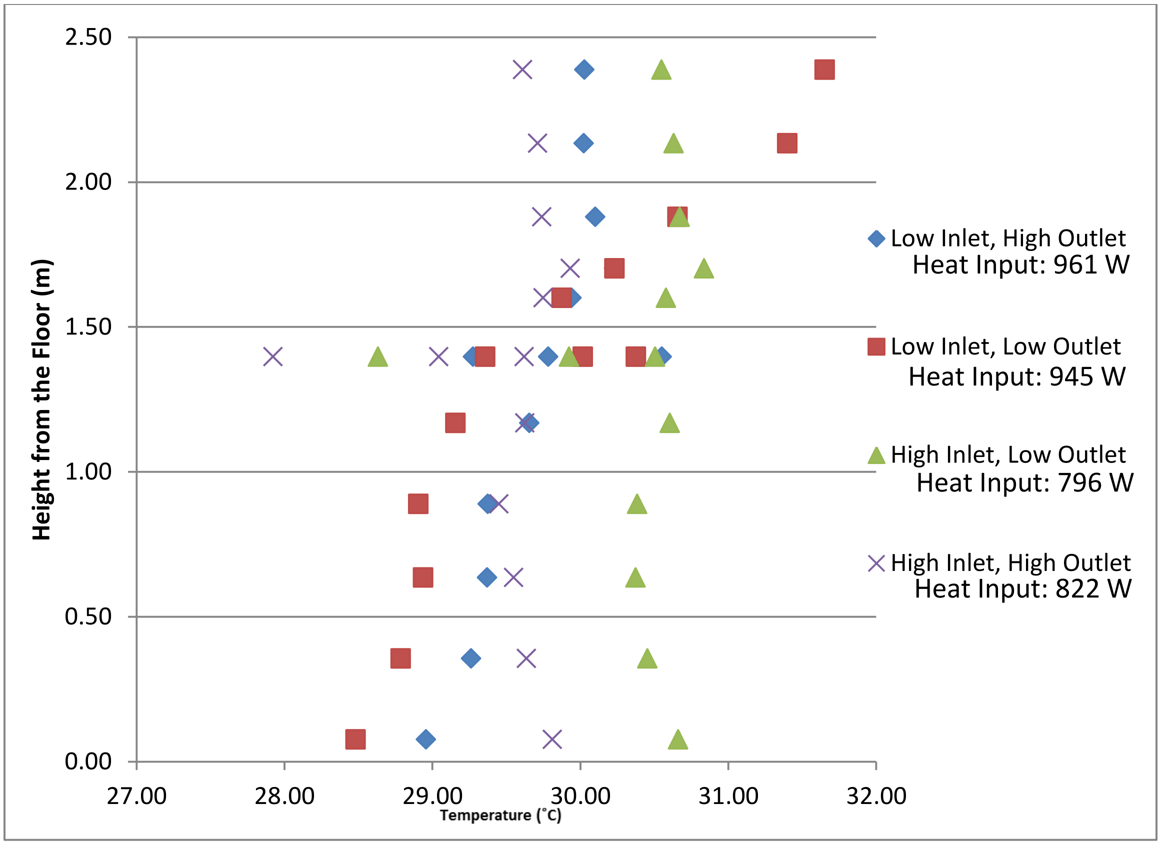


Figure 4-11: The temperature profile for the four ventilation configurations with high AC speed, high hot-plate input, and low distributed floor heat. Here the low inlet configurations took larger heat loads and provided a cooler occupied region.

Table 4-31: The percent error calculated for each configuration with the AC fan on high, the hot plate on high, and the distributed floor on low

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	6.5%
Low	Low	6.7%
High	Low	1.1%
High	High	7.8%

Figure 4-12 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on low, and the distributed floor on high, and Table 4-32 shows the calculated energy balance for each of these configurations. The distributed load is approximately the same in all cases, but the point load is about 150 watts higher with the low inlet configurations. This caused the overall temperature to rise in the low inlet configurations and the harsh stratification to be lower than previously seen.

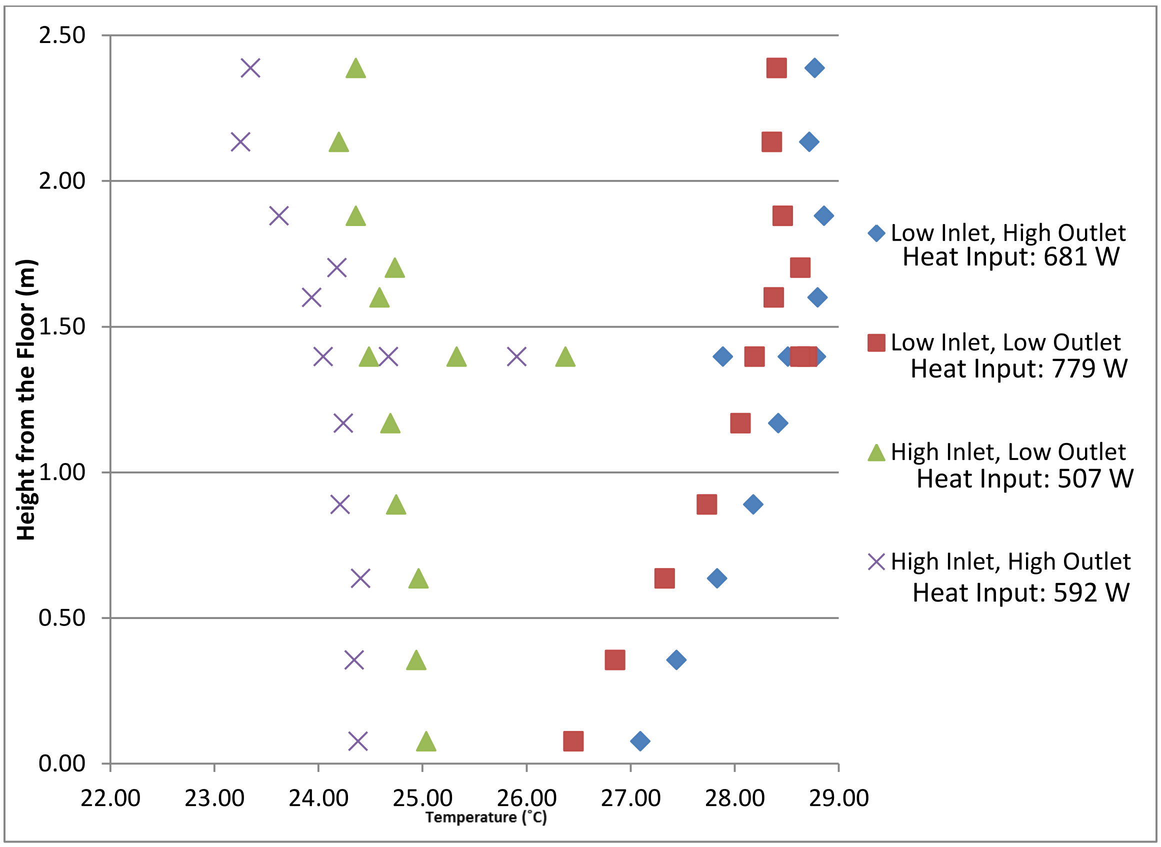


Figure 4-12: The temperature profile for the four ventilation configurations with high AC speed, low hot-plate input, and high distributed floor heat. Here the low inlet configurations took larger heat loads which resulted in a higher temperature in the occupied region.

Table 4-32: The percent error calculated for each configuration with the AC fan on high, the hot plate on low, and the distributed floor on high

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	High	18.5%
Low	Low	9.3%
High	Low	2.8%
High	High	0.8%

Figure 4-13 shows the temperature profiles of the four ventilation configurations with the AC fan on high, hot-plate on high, and the distributed floor on high, and Table 4-33 shows the calculated energy balance for each of these configurations. These scenarios had approximately the same heat loads and resulted in the low inlet configurations having a higher overall temperature with a harsh stratification at about 1.4m above the floor.

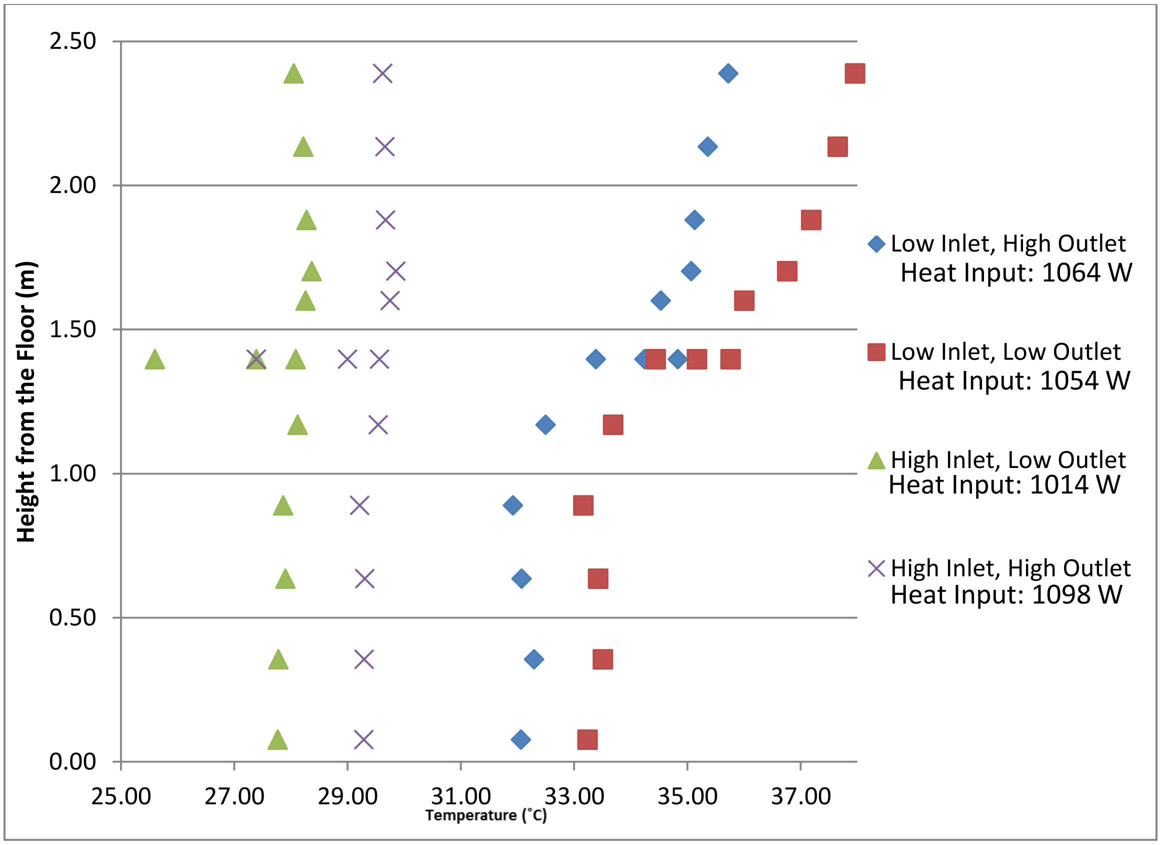


Figure 4-13: The temperature profile for the four ventilation configurations with high AC speed, high hot-plate input, and high distributed floor heat. Here all configurations had about the same heat load and the high inlet configurations outperformed the low inlet configurations.

Table 4-33: The percent error calculated for each configuration with the AC fan on high, the hot plate on high, and the distributed floor on high

Ventilation Configuration		Percent Error
Inlet	Outlet	
Low	Low	3.7%
Low	High	9.7%
High	High	15.3%
High	Low	7.9%

The results showed that with a lower inlet velocity the low inlet configurations were able to handle a higher heat load with either a cooler or a comparable occupied temperature. However, when the velocity was increase, the consistency of this was reduced. Also, the low inlet, high outlet configuration was able to consistently cool the space more than the low inlet, low outlet configuration when the velocity was set to low.

It was also noticed that the high inlet, high outlet was able to dump the cool air into the space and extract the warmer air. It was able to outperform the high inlet, low outlet configuration in most cases.

In all cases the vertical variation was as high as 2˚C. This shows that any variation of less than 2˚C in the horizontal direction cannot be attributed to the temperature gradient, but could be a result of the standard variation within the room. Therefore, only the stratifications exceeding 2˚C are clear examples of the energy saving stratification within the room.

6. Summation of Results

The experimental results showed that the low inlet, high outlet configuration was able to maintain a stratification within the space more effectively than the other ventilation configurations. It consistently resulted in a greater floor to ceiling temperature difference compared to the other configurations. This is summarized in Figure 4-14 that shows a plot of the floor to ceiling temperature difference for each of the four ventilation configurations and for each heat load. Also, a line is drawn at where

$\Delta T=2^{\circ}C$. This line separates the cases where the stratification is less than the expected variation, below the line, and the cases where a significant stratification has occurred, above the line. Table 4-34 lists the different heat load configurations in order of the ΔT for the low inlet, high outlet configuration. Figure 4-14 shows, in general, that for low heat loads (generally the earlier cases) there is no significant vertical temperature variation for any ventilation configuration. This is most likely a result of the inflow cool air containing enough kinetic energy to keep the room well mixed. However, for higher heat loads the low – high configuration, there is a consistent vertical stratification, as the stabilizing effect of the heating is able to resist the mixing due to the ventilation flow. The low-low configuration also produces significant vertical variation in temperature. However, this is due to the cooling flow short circuiting and leaving the upper portion of the room warm but unventilated.

Table 4-34: The list of the heat load scenarios, organized from the lowest internal ΔT to the largest for the low inlet, high outlet configuration

Series Number	AC Speed	HP Setting	Floor Setting	Total Load (W)
1	High	Low	Off	228.26
2	Low	Low	Off	284.77
3	High	Low	Low	640.68
4	High	High	Low	881.18
5	High	High	Off	498.42
6	High	Low	High	639.74
7	Low	Off	High	630.07
8	Low	High	Off	514.59
9	Low	Low	High	845.12
10	Low	Low	Low	674.61
11	Low	High	High	1045.37
12	High	High	High	1057.55
13	Low	High	Low	720.94

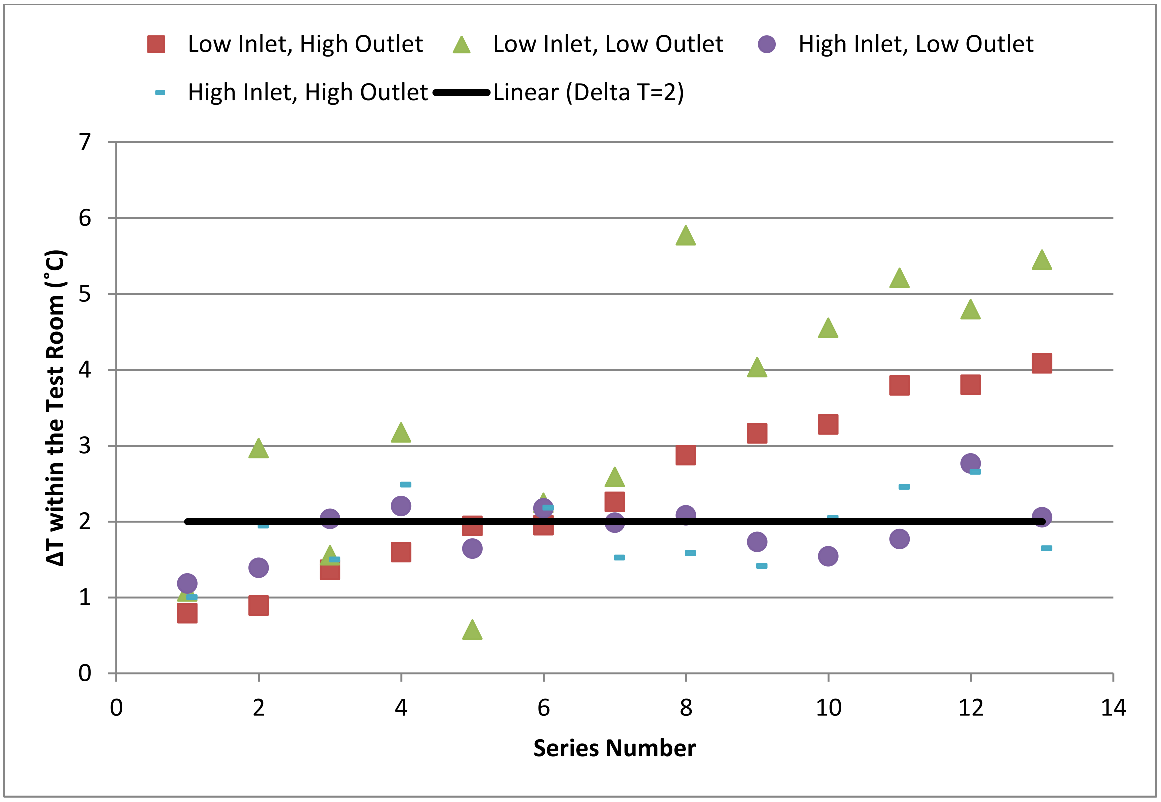


Figure 4-14: The four ventilation configuration's internal ΔT for each series number, listed in Table 4-34, and the total heat load for each series. The low inlet configurations were able to achieve a ΔT greater than 2˚C. This shows that the stratification is achieved by something other than the standard variation in temperature in the room.

The relationship between temperature difference and heat load for the low-high configuration is more clearly seen in Figure 4-15 that shows just the low inlet, high outlet ΔT (left axis) plotted alongside the total heat load (right axis) for each heat load scenario. This shows that, in general, as the heat load increases so does the ΔT for the low-low configuration. The only significant exceptions to this are cases 4 and 8. In case 4 there is a high heat load but also high ventilation flow breaking down the stratification resulting in a relatively low temperature difference for the given moderate to high heat load. In case 8 the temperature difference is relatively high for a lower heat load. In this case there is a lower ventilation flow rate and the entire heat load is in the concentrated (point source) hot plate (it is well established that point heat sources lead to stronger stratifications than distributed heat sources, see Linden 1999).

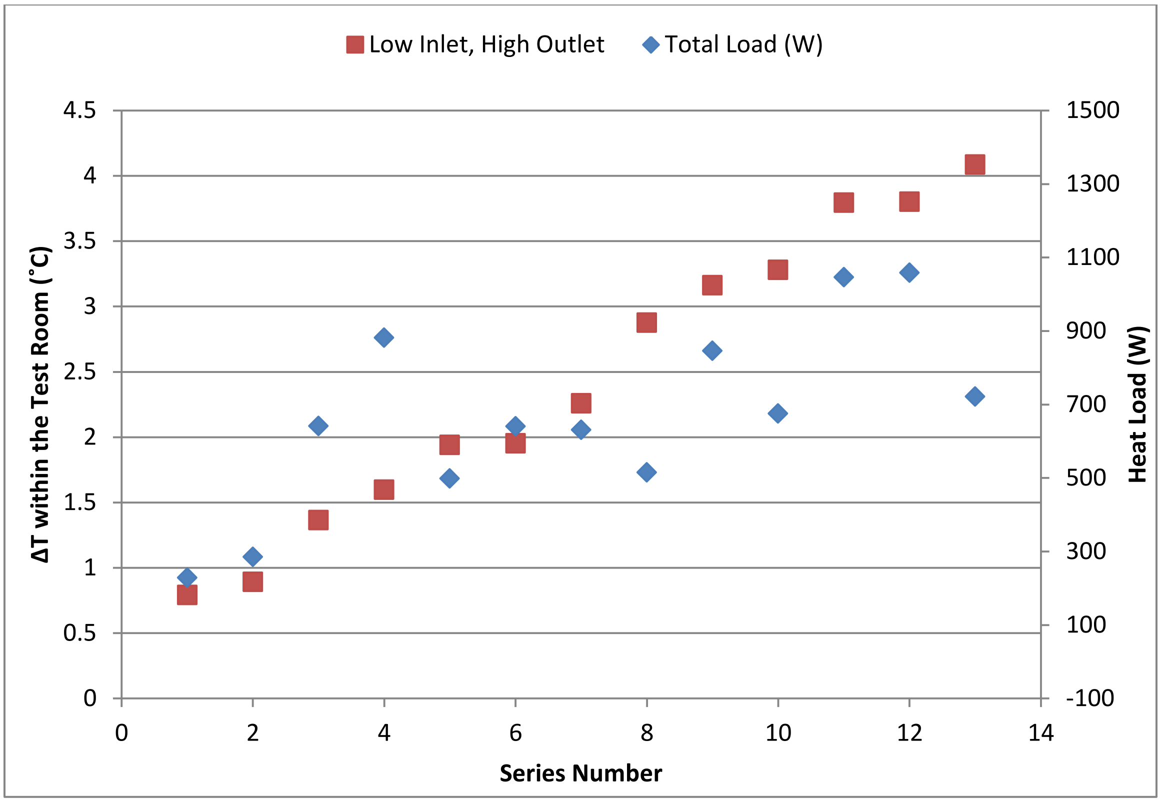


Figure 4-15: The total heat load for each heat load configuration graphed with the low inlet, high outlet ΔT. Each heat load configuration is listed in Table 4-34 with its series number. With few exceptions the increase in the ΔT parallels the increase in heat load. The exceptions can be attributed to the effect of the inlet velocity.

CHPTER 5. DISCUSSION AND CONCLUSIONS

1. Conclusions

The results of the experimental runs show that the low inlet, high outlet configuration was better able to maintain a thermal stratification compared to the other ventilation configurations when in cooling. A cooler occupied zone was not shown in every scenario, but in general the low-high configuration resulted in a two-layer stratification within the room. This means that the temperature of the lower occupied region will be closer to the inlet temperature than the outlet temperature meaning that the inlet temperature can be higher than for the case of mixed ventilation, thereby saving energy. The thermal stratification also indicates that the low-high configuration likely has the ability to flush contaminants vertically through the room and hence improve the IAQ. Since IAQ is increasingly becoming an important issue in design, this alone could justify this ventilation configuration being implemented on a larger scale. The amount of pollutants and VOCs that are currently in the residential environment could have a reduced impact on the occupant if the pollutants are prevented from accumulating in the lower occupied region of the room. This improved IAQ would also provide a strong case for DV in a hospital application. One of the potential drawbacks to DV is that it is less effective in the heating than the cooling phase. For a hospital, the system is in the cooling phase almost all of the time. Therefore, future studies into the improved IAQ could be extremely applicable to hospital ventilation.

Also, when the hot plate and heated floor were closer in magnitude to each other, the low inlet, high outlet configuration was able to achieve a cooler temperature in the occupied zone. In an actual application the heat load will remain the same regardless of the ventilation configuration. However, the more efficient configuration could aid the system in achieving a cooler occupied zone more quickly. This would lead to the system not needing to work as hard to condition the space, and therefore, it would save energy and improve efficiency.

Although it is clear that the low inlet, high outlet configuration has the potential to reduce the load on the mechanical system, there are some complications with implementing this strategy. The scenario studied was a simplified case. In a realistic application, the space would have various geometries put together. The space would be broken into multiple inlet/outlet zones and these zones would interact with each other. If the residence was multiple stories, it is unknown how the floors would interact and mix. It is also unknown how the heating phase would affect the efficiency. The same buoyant flow of air would not occur in heating. One of the reasons DV works well with large commercial spaces is that the high concentration of occupants and significant internal heat gains keeps the system in cooling mode most of the time. These and other practical application issues need to be investigated before this system is recommended for residential use.

2. Cost/Benefit

The major cost to displacing the inlet and outlet in a residential application is running the ductwork vertically in the walls. This can be done with a soffit down the wall, or restricting the ductwork to the void in the wall. Both of these scenarios come with an additional cost, somewhere in the magnitude of a few hundred dollars. The soffit takes away square footage in your space and additional framing, finishing, and ductwork costs. To run the ductwork in the walls, the ductwork sizing is altered and the ductwork will typically require more material to carry the same amount of air. This means that the costs of the soffit are eliminated but the higher cost of the ductwork balances this savings. Therefore, either scenario will have an impact on the bottom line. Further, the additional ducting, regardless of how it is implemented, will increase the head loss in the duct system; this will increase the load on the fan driving the flow. The additional power demand will be minimal, since the fan only consumes a small amount of energy when compared to the entire HVAC system. There is also a large upfront cost due to the diffusers that are typically used with DV. These diffusers allow the air to enter the room at a low velocity without reducing the overall flow rate within the system. One of these diffusers can cost anywhere from $300 to $500 dollars. The larger benefit would be if the system could be reduced in size. If the system could be dropped by one ton, there would be an upfront savings of buying a smaller unit. This savings would be somewhere in the magnitude of a few hundred dollars. The bigger impact of this reduction in size would be the annual savings in energy consumption. Depending on the size of the system, this reduction could result in a savings of a couple hundred dollars every year.

The deciding factor depends on the occupant. To some, the improved IAQ and reduction in air born pollutants would outweigh the costs. However, to others they may not see a financial pay back from the improved IAQ and therefore may not see the benefit in the low inlet, high outlet configuration. Therefore, the real cost/benefit must rely on the concrete financial savings.

This comparison will require case studies that can quantify the system size reduction or the energy savings of the same size system. Ideally, the system should run constantly to increase its efficiency. Therefore, it would be preferable that the system could be reduced in tonnage. This would result in upfront savings of a couple and long term energy savings.

To society as a whole, there is always a benefit to reducing the energy demand. As prices for energy increase, a decrease in demand could begin to offset these increasing costs. One way to reduce this overall demand is to improve the design of our residential sector. Buildings are the largest consumer of energy and residential buildings consume about a third of that energy. However, this is an area that tends to be ignored in the research and to not get the attention in design that is devoted to the other areas of construction.

3. Future Work

These experimental runs show that the ventilation configuration and conditions at the inlet can impact occupant comfort within a space. However, they have also shown numerous variables that need to be evaluated in more detail. There are a few of these areas that should be investigated prior to any other study. Determining the proper inlet velocity for DV could increase the quality of the results acquired. Also, finding a cutoff ceiling height above which DV is effective and below which it is not, would greatly improve the likelihood of DV being implemented in residential settings. Being able to quantify the amount of energy that could be saved by using DV would significantly increase the support for these systems. Lastly, having a concrete analysis of the IAQ improvements throughout a space could alone justify the installation of these systems.

In order for the desired strong stratification to occur, the turbulence in the air needs to be reduced. One way to reduce this turbulence is to reduce the turbulence at the inlet. This can be achieved by reducing the velocity at the inlet. However, this reduces the amount of air entering the room and therefore reduces the system's ability to quickly condition the space. Alternatively, a large diffuser can be attached at the inlet. This can disperse the air at a lower velocity, but still allow the same amount of air to enter the space. Various diffuser geometries and resultant velocities should be investigated further.

This study limited the ceiling height of the space to about 2.8 meters. The current applications of displacement ventilation are in large auditoriums where the unoccupied zone is substantial. If future studies could investigate at what ceiling height the benefits of DV no longer outweigh the costs, designers could have a criterion for when DV is most effective.

One of the major benefits of DV is the system's ability to flush pollutants and improve the IAQ. Future work could study these benefits in more detail and determine if the benefits of this pollutant flushing justify the implementation of DV systems in residential application.

The potential energy savings from using DV is in the ability to reduce the size of the system installed, due to a more efficient circulation of air. Determining the actual amount by which a system could be reduced would allow for an actual up front cost savings to be calculated. More importantly, this would allow for an annual cost savings to be determined. This could quickly justify the use of DV in a residential application.

Some of the other areas of this study that need to be investigated further include the short circuits between the inlet and outlet, the variability of the inlet conditions, other heights for the inlet and outlet, stabilizing the stratification within the room, and running the same experiments for the heating condition.

When both the inlet and outlet were located at the same vertical position they were directly across from each other. In some cases, this may have led to a short circuit in the system. This occurs when the inlet air travels directly to the outlet, without properly conditioning the space. The amount of mixing that occurred in these experiments would have prevented this. However, when the mixing is reduced by slower inlet speeds, the short circuit could become a larger problem. Future work should consider displacing the inlet and outlet horizontally to reduce the likelihood of these short circuits occurring.

The inlet temperature varied with the external conditions and affected the consistency of the results. In future studies, something should be done to eliminate this variation. One way to do this would be insulating the ductwork from the AC unit to the inlet. This in conjunction with a more controlled environment to run the experiments could significantly improve the consistency of the experimental runs and the results.

Also, the velocity at the inlet varied between the low and the high inlet configurations. This was a result of the longer distance and more ductwork the air had to travel along to get to the high inlet when compared to the low inlet. This could be resolved by balancing the static pressure for the lower inlet by stretching out the ductwork. If done properly, the inlet velocity for both high and low inlets could be equivalent.

These studies only investigated the difference between low and high inlet and outlet locations. In future studies, the various locations throughout the height of the wall should be investigated, particularly an outlet just above the occupied zone. This could help the system to only condition the occupied zone. It could also help control the location of the temperature step in the stratification when it is achieved.

In the experiments run the stratification within the room separated within the occupied zone. In order for DV to be an acceptable ventilation strategy within residential housing, the separation in temperature zones needs to occur above the occupied region. The effects of ceiling height, heat loads, and inlet velocity on the location and stability of this separation needs to be investigated further. The proper combination of these three variables and possibly others, needs to be understood in order to achieve a stable separation in the air temperature to save money on energy while still providing a comfortable occupied region.

Also, this study focused on the cooling side of air conditioning. In some climates HVAC load is dominated by the heating side. In these applications, a different ventilation configuration may be more beneficial. Future work should look into what ventilation configuration is most advantageous when the system is in heating mode.

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APPENDIX A - INSULATION TECHNICAL INFORMATION SHEET

The detailed information on the insulation used for the experimental setup.

TECHNICAL INFORMATION SHEET

917
5/9/2011

Firestone BUILDING PRODUCTS

RESISTA™

DESCRIPTION:
Firestone Resista flat and tapered roof insulation consists of a closed-cell polyiso foam core laminated to a specially coated, inorganic, fiberglass facer. Resista roof insulation provides outstanding fire, wind and thermal performance for single-ply and modified bitumen commercial roofing applications.

All Firestone polyiso insulations use EPA accepted blowing agents and qualify under the Federal Procurement Regulation for Recycled Material. Flat and tapered RESISTA with IsoGard™ foam technology incorporates a HCFC-free blowing agent that does not contribute to the depletion of the ozone (ODP-free). Additionally, Firestone Resista is a formaldehyde free material.

Resista is a mold resistant material per ASTM D3273

METHOD OF APPLICATION:
Insulation shall be neatly fitted to all roof penetrations, projections and nailers. No more insulation shall be installed than can be covered with membrane and completed before the end of each day's work or before the onset of inclement weather.
Note: See Firestone Technical Information Sheet 950 for Insulation Attachment Patterns.

RESISTA FLAT AND TAPERED MUST BE INSTALLED USING: Firestone Fasteners and plates or Firestone approved insulation adhesives.

DO NOT use hot asphalt with Resista™

WARRANTY:
Firestone Resista is acceptable in approved Firestone Red Shield or Platinum Roofing Systems ranging from 5 to 30 years.

RESISTA Flute Span Over Metal Decks				
Thickness	1.0"	1.25"	1.5"-3.8"	4.0"
Span	2.625"	3.675"	4.375"	4.5"

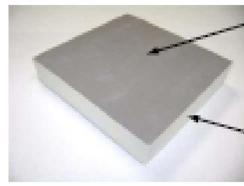
SPECIFICATION COMPLIANCE:
ASTM C1289, Type II, Class 2
UL Classified
FM Class 1 Approved
Manufactured in an ISO 9001 Registered Facility
CAN/ULC-S704, Type 2 pending

FM APPROVED

CLASSIFIED C UL US

PIMA QualityMark™ The Certified LTTR value

This sheet is meant to highlight Firestone's products and specifications and is subject to change without notice. Firestone takes responsibility for furnishing quality materials, which meet Firestone's published product specifications. Neither Firestone nor its representatives practice architecture. Firestone offers no opinion on and expressly disclaims any responsibility for the soundness of any structure. Firestone accepts no liability for structural failure or resultant damages. Consult a competent structural engineer prior to installation if the structural soundness or structural ability to properly support a planned installation is in question. No Firestone representative is authorized to vary this disclaimer.



PRODUCT DATA

Insulation Thickness (inches)	Insulation Thickness (mm)	LTTR* R-Value
1.00	25.4	6.0
1.25	31.7	7.5
1.50	38.1	9.0
1.75	44.5	10.5
2.00	50.8	12.1
2.30	58.4	14.0
2.50	63.5	15.3
2.80	71.1	17.2
3.00	76.2	18.5
3.25	82.6	20.1
3.50	88.9	21.7
3.75	95.3	23.4
4.00	101.6	25.0

**Long Term Thermal Resistance (LTTR) values provide a 15-year time-weighted average in accordance with CAN/ULC-S770.*

POLYISO PHYSICAL PROPERTIES

Physical Property	ASTM Test	English Values	Metric Values
Compressive Strength, nominal	D 1621	20 psi	138 kPa
Compressive Strength, min	D 1621	25 psi	172 kPa
Density	D 1622	2 pcf	32 kg/m^3
Dimensional Stability	D 2126	<2%	<2%
Moisture Vapor Transmission	E 96	<1.0 Perm	<57.5 $ng/(Pa{\cdot}s{\cdot}m^2)$
Water Absorption	C 209	<1% by Volume	<1% by Volume
Service Temperature		-100° to 250° F	-73° to 121° C

AVAILABLE SIZES:
Flat Boards: 4' x 4' (1.22 m x 1.22 m)
4' x 8' (1.22 m x 2.44 m)
Thickness ranging 1.0" (25.4 mm) to 4.0" (101.6 mm)

Tapered Boards: 4' x 4' (1.22 m x 1.22 m)
4' x 8' (1.22 m X 2.44 m) (special order)
Thickness ranging 0.5" (12.7 mm) to 4.0" (101.6 mm)
Slopes ranging 1/16" per foot (.5%) to ½" per foot (4%)
See chart on page 2.

STORAGE AND PRECAUTIONS:
- Keep insulation dry at all times.
- Elevate insulation above the deck or ground.
- Cover insulation with waterproof tarps.
- Combustible. Refer to MSDS for more information.
- Do not install over wet, damp or uneven substrates.

Manufacturing Locations:

Florence, KY   Youngwood, PA   Salt Lake City, UT

Firestone Building Products Company
250 W. 96th Street, Indianapolis, IN 46260
Sales: (800) 428-4442 • Technical (800) 428-4511
www.firestonebpco.com

S723-RFS-257

TECHNICAL INFORMATION SHEET

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5/9/2011

LEED INFORMATION:

Firestone Resista can help contribute to overall L.E.E.D. certification due to its thermal efficiency (R-Value), recycled content, and zero ozone depletion. Resista can help achieve valuable credits in the following categories:

- Energy and Atmosphere
  - Benefits of commissioning building energy systems include lower energy use and lower operating costs.
  - Firestone Resista contains the highest R-Value per inch to help assist in the following credits:
    - EA Pre-requisite 2: Minimum Energy Performance
    - EA Credit 1: Optimize Energy Performance (1-19 points)
- Material & Resources
  - Benefits include the reduction of waste, as well the preservation of landfill space and raw materials.
  - Firestone Resista contains recycled content, can be re-used, and is manufactured strategically across the United States.
    - MR Credit 2: Construction Waste Management (1-2 points)
    - MR Credit 3: Material Re-Use (1-2 points)
    - MR Credit 4: Recycled Content (1-2 points)
    - MR Credit 5: Regional Material (1-2 points)

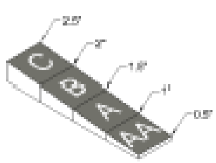
RECYCLE CONTENT OF RESISTA

Thickness (Inches)	Thickness, (millimeters)	% Post Consumer	% Pre-Consumer	TOTAL RECYCLED CONTENT
1.00	25.40	0%	5%	5%
1.25	31.75	0%	5%	5%
1.50	38.10	0%	6%	6%
1.75	44.45	0%	7%	7%
2.00	50.80	0%	7%	7%
2.30	58.42	0%	8%	8%
2.50	63.50	0%	8%	8%
2.80	71.12	0%	8%	8%
3.00	76.20	0%	9%	9%
3.25	82.55	0%	9%	9%
3.50	88.90	0%	9%	9%
3.75	95.25	0%	10%	10%
4.00	101.60	0%	10%	10%

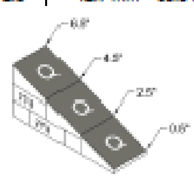
LEED Reference Guide information can be obtained on line at http://www.usgbc.org/ "LEED is trademark of the U.S. Green Building Council."

TAPERED RESISTA (Available Sizes):

Panel Code	Min-Max Thickness		Slope		Pieces Per Bundle (4'x4')
A	0.5" - 1.0"	12.7 mm – 25.4 mm	1/8"/ft.	1%	64
AA	1.0" - 1.5"	25.4 mm – 38.1 mm	1/8"/ft.	1%	38
B	1.5" - 2.0"	38.1 mm – 50.8 mm	1/8"/ft.	1%	26
C	2.0" - 2.5"	50.8 mm – 63.5 mm	1/8"/ft.	1%	20
G	1.0" - 2.0"	25.4 mm – 50.8 mm	1/4"/ft.	2%	32
H	2.0" - 3.0"	50.8 mm – 76.2 mm	1/4"/ft.	2%	18
I	3.0" - 4.0"	76.2 mm – 100.6 mm	1/4"/ft.	2%	12
X	0.5" - 1.5"	12.7 mm – 38.1 mm	1/4"/ft.	2%	48
Y	1.5" - 2.5"	38.1 mm – 63.5 mm	1/4"/ft.	2%	24
Z	2.5" - 3.5"	63.5 mm – 88.9 mm	1/4"/ft.	2%	16
Q	0.5" - 2.5"	12.7 mm – 63.5 mm	1/2"/ft.	4%	32



1/8"/ft. Tapered Section



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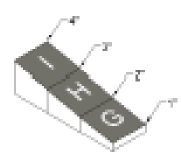
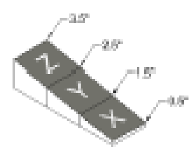


1/4"/ft. Tapered Section



1/4"/ft. Tapered Section

Contact Firestone's Estimating Services Department for a project quotation or for layout guidance. For site assistance or contractor support, contact Firestone's Tapered Project Management Team by calling: 1-800-428-4442 or by emailing: estimatingservicesdept@firestonebp.com

S723-RFS-257