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Field Study Of A Radiant Heating System For Sleep Thermal Comfort And Potential Energy Saving

Christopher L. K. Wang
Ryerson University

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Field Study of a Radiant Heating System for Sleep Thermal Comfort and
Potential Energy Saving

By

Christopher Leung Kwong Wang

BASc, University of Windsor, 2006

A thesis presented to Ryerson University in partial fulfillment of the
requirements for the degree of

Master of Applied Science

In the Program of

Building Science

Toronto, Ontario, Canada, 2011

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ABSTRACT

Field Study of a Radiant Heating System for Sleep Thermal Comfort and Potential Energy Saving

Christopher Leung, Kwong Wang, BAsC

Master of Applied Science, Building Science 2011

Ryerson University

As sleep is unconscious, the traditional definition of thermal comfort with conscious judgment does not apply. In this thesis sleep thermal comfort is defined as the thermal condition which enables sleep to most efficiently rejuvenate the body and mind.

A comfort model was developed to simulate the respective thermal environment required to achieve the desired body thermal conditions and a new infrared sphere method was developed to measure mean radiant temperature. Existing heating conditions according to building code conditions during sleeping hours was calculated to likely overheat a sleeping person and allowed energy saving potential by reducing nighttime heating set points.

Experimenting with existing radiantly and forced air heated residential buildings, it was confirmed that thermal environment was too hot for comfortable sleep and that the infrared sphere method shows promise. With

the site data, potential energy savings were calculated and around 10% of energy consumption reduction may be achieved during peak heating.

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1.0 Introduction

1.1. Background

As a designer of radiant ceiling heating systems, the author received reports of individuals waking up in the middle of the night with a cold sweat while sleeping under an operating radiant ceiling system.

These radiant ceiling heating systems were the embedded hydronic type designed to meet the Ontario Building Code (Province of Ontario, 2006) heating requirements of an indoor air temperature of 22°C and according to ASHRAE handbook: Systems and Equipment (ASHRAE, 2008). A separate ventilation distribution system provides ventilation as per 100% outdoor air systems.

During winter operation, the radiant heating panel temperature may reach above 30°C during the on cycle and gradually decrease back to room temperature during the off cycle. Thus, a residential ceiling radiant system differs significantly from traditional forced air or radiator system in which both the radiator and the forced air system deliver heat via natural or forced convection with sufficiently more air movement than a radiant ceiling system.

As such, radiant heating systems do not necessarily follow existing thermal comfort standards such as ASHRAE 55 (ASHRAE, 2010), which stipulates that dissimilar vertical ceiling/floor radiant field should not exceed 5°C. Often,

radiant heating systems will exceed this difference to meet required heating output.

Existing thermal comfort research has been performed on the awake condition with people wearing typical daily attire performing daily tasks. These include researchers in North America, Europe and Asia (Fanger, Ipsen, Langkilde, Olesen, Christensen, & Tanabe, 1985); (Atmaca, Kaynakli, & Yigit, 2007); (Imanari, Omori, & Bogaki, 1999). However, the sleep thermal comfort criteria have not been sufficiently explored.

The sleep thermal comfort condition has not been thoroughly developed by building scientists, psychologists and physiologists. The inherent differences between the external science of building science to the physical and mental aspects of physiology and psychology of the human body meant that research produced independently by one generally does not enter sufficient detail into the other aspects. This is also true regarding terminology which overlaps, but is inconsistent (Bakken, 1980). Coupled with the generally accepted definition of thermal comfort which explicitly defines that the mind expresses satisfaction in a conscious state, there is not a coherent method of determining sleep thermal comfort.

Utilizing existing research on sleep efficiency, limiting this research to a steady state condition and filling many of the voids in-between the available knowledge by educated guesses, the author has developed a thermal model to

calculate the thermal condition that are comfortable for sleep. A simple numeric calculation of the radiant ceiling condition has been developed and will be utilized to create the parameter for a heating system to be operated at during hours of sleep and be the basis of an investigation into potential energy savings.

1.2. Research Direction

The purpose of this research is to investigate and document the thermal environment a typical radiant heated residential building will create in relation to the concept of sleep thermal comfort, where the existing thermal environment created will be compared to existing thermal comfort standards and other research. The activities of a person at home are very much different than a person at office work where thermal comfort standards are typically derived. According to the Bureau of Labor Statistics (United States Department of Labor, 2008) it was found that on average a person at home spends roughly 1.73 hours doing chores, 3 hours doing leisure activities and 8.6 hours sleeping. Since sleeping represents a significant proportion of the time spent at home it is important to explore more fully the conditions for sleep thermal comfort and to determine the thermal environment a heating system will have to create to achieve it.

The physiological and psychological aspects of sleep during various stages are inherently variable and while HVAC systems may be modulated to compensate for these changes, it is unfeasible to create a thermal environment for each occupant. Additionally, physiological and psychological responses to the thermal environment during sleep have not been clearly established. Therefore, the focus of this research will be based on a steady state thermal condition to provide the optimized sleep thermal comfort.

In summary this research will investigate the thermal comfort aspect of radiant heating, the definition of thermal comfort during sleep, the potential energy savings by reducing system output and the development of tools to better measure the thermal environment.

1.3. Objectives

The objectives of this research are to:

1. Determine the thermal conditions for sleep thermal comfort.
2. Develop a fast acting mean radiant temperature determination tool.
3. Explore the potential energy savings in association with improving sleep thermal comfort.

4. Compare the thermal environment created by a radiant heating system and a forced air system according to the ASHRAE 55 (ASHRAE, 2010) requirements.

Comfortable sleep or quality of sleep contribute significantly to the performance and productivity of a person after he or she wakes up (Haskell, Palca, Walker, Berger, & Heller, 1981) and thus is of significant importance.

Existing literature recognizes the reduction of metabolism during sleep and the need to decrease body core temperature. This differs significantly from the human state which existing thermal comfort standards were developed and as such the sleep thermal comfort condition is specifically excluded from ASHRAE 55 (ASHRAE, 2010).

This research aims to develop the sleep thermal criteria and develop a simplified comfort model. This is much more of a concern in radiant ceiling heated buildings as the person asleep lying down will directly face the radiant panel on the ceiling. A field test will be performed in a ceiling radiant heated building and the results will be analyzed.

During sleep, metabolism rate and core temperature requirements will reduce. Coupled with the significantly increased clothing (a blanket) level during sleep, these conditions suggest that there will be possible heating output reduction of the heating system to meet the comfort requirements. With the comfort model

and the field data, the energy consumption reduction can be calculated and simulated.

As radiant heating systems are marketed as fast acting with heat transfer via radiation being instantaneous, it is important that the deployed sensors have a fast enough response to record the thermal environment as it is being created or changed.

The existing method of determining mean radiant temperature by the black globe setup as per ISO 7726 (ISO, 2002) has a latency of around 30 minutes. This is considered insufficient and an alternative sensor needs to be developed. To test the new sensor, an additional field test in another building heated by forced air will be completed as verification.

Many studies such as (Chen, 1990) have found radiant heating to be more comfortable than traditional systems such as forced air or high temperature radiator systems. With field testing of two buildings, one heated radiantly and the other by forced air, a limited comparison could be made of the thermal comfort level with the collected field data although the envelope of the two buildings are vastly different.

1.4. Outline of Thesis

The background, research direction and objectives are discussed and summarized in the previous sections of Chapter 1.0. Chapter 2.0 reviews

existing standards, modeling principles and field studies in the fields of thermal comfort, sleep and experimental equipment.

With the insights outlined in chapter 2.0 the development of the thermal model is discussed in chapter 3.0. The sleep thermal criteria as extrapolated from Chapter 2.0 are summarized and are inputted into the thermal model. The results from the thermal model are discussed and compared to similar research.

Chapter 4.0 presents the necessity of developing a mean radiant sensor and discusses the theory for its operation. Chapter 5.0 discusses the experimental design and equipment setup. The experimental results regarding thermal comfort, sleep thermal comfort and the performance of new mean radiant temperature sensor is presented in chapter 6.0.

Chapter 7.0 consists of an idealized calculation of potential energy savings if the mechanical system was operated at a reduced level to achieve desire sleep comfort conditions. Chapter 8.0 summarizes the contribution of this thesis and provides concluding statements.

2.0 Literature Review

The common and accepted definition of thermal comfort as developed by Fanger (Fanger P. , 1970) and adopted by ASHRAE 55 (ASHRAE, 2010) is the *“condition of mind which expresses satisfaction with the thermal environment and is assessed by subjective evaluation”* and is determined by established thermal comfort components: Mean Radiant Temperature, Air Dry Bulb Temperature, Metabolic Rate, Clothing Insulation, Air Velocity and Humidity.

As thermal comfort is a subjective evaluation, it has been extensively studied by Fanger and others about people who are awake in laboratory conditions. As a subjective evaluation, the thermal comfort for people who are asleep, sedated or unconscious is undefined and under-researched. Research from psychology and physiology has identified internal body conditions that are favorable for efficient sleep such as core and distal temperature requirements. But these conditions were derived without a clear connection to the thermal environment.

This chapter explores existing thermal comfort standards and identifies existing research into thermal comfort for the awake condition. From the psychological and physiological perspective, it reviews the body thermal condition for comfortable sleep and combines its conditions with the thermal environment.

2.1. Thermal Comfort Standards

Existing thermal comfort standards for awake conditions such as ASHRAE 55 (ASHRAE, 2010) and ISO EN7726 (ISO, 2002) are based on Fanger's studies. ASHRAE 55 will be used to evaluate the thermal comfort level achieved by the radiant heated building and the forced air heated building.

2.2. Thermal Environment Comfort Criteria

Thermal comfort criteria for awake as defined by ASHRAE Standard 55-2004 (ASHRAE, 2010) are:

1. Allowable radiant asymmetry: 5% percent dissatisfied, 4°C
2. Allowable air temperature stratification: 5% percent dissatisfied, air temperature between head and feet less than 3°C
3. Allowable floor temperature range: 19°C-29°C
4. Allowable operative temperature: 20% percent dissatisfied
5. Allowable peak to peak variation in operative temperature: 1.1°C
6. Allowable ramp and drift in operative temperature:

Table 1 ASHRAE 55 Ramp and Drift Criteria

Time Period	0.25 h	0.5 h	1.0 h	2.0 h	4.0 h
Maximum Operative Temperature Change Allowed	1.1°C	1.7°C	2.2°C	2.8°C	3.3°C

The measuring criteria as per ASHRAE 55 is as follows:

1. Air speed to be measured over 3 minutes
2. Measure according to ASHRAE 70-1991, 113-1990 or ISO 7726
3. Measure in the centre of the room for rooms with undefined occupancy
4. Sensors to be located 1.0 m away from all surfaces

2.3. Existing Radiant Thermal Comfort Research

To establish radiant thermal comfort condition for persons who are awake, Fanger et al (Fanger, Banhidi, B.W.Olesen, & Langkilde, 1980) used 16 human subjects to calibrate a thermal manikin, which was then used for further experiment. It was found that a radiant asymmetry of 4°C was the threshold at which 5% of their test subjects became uncomfortable. However, the experiment was performed at a thermally neutral state where the air temperature was reduced when the radiant ceiling panel temperature was increased to obtain a uniform operative temperature. As such the discomfort due to the radiant ceiling is attributed to the feeling of cold feet due to the reduced air temperature.

The comfort model developed by Fanger et al is not representative of radiant heating systems operated to building code (Province of Ontario, 2006) compliance, which is based on maintaining an air temperature of 22°C for heating. The air temperature will not be lowered to achieve the desired operative temperature and therefore the feeling of cold feet by reduced air temperature may not occur.

If the building envelope is well insulated with minimal leakage, the radiant output will be sufficient to heat the surrounding surfaces which in turn increase the temperature of the air meaning that the air temperature will be very close to the mean radiant temperature.

As a continuation of their previous study, Fanger et al (Fanger, Ipsen, Langkilde, Olesen, Christensen, & Tanabe, 1985) attempted to study the effects of radiant asymmetry and discovered that ceiling heating causes the most thermal discomfort for a person who is conscious and seated. However, as discussed earlier asymmetric radiant heating will cause cold sensation on the other side of the body and that the magnitude of discomfort documented for heated ceilings are in part caused by the cold feet sensation of the test subjects.

Their findings are incorporated into the thermal comfort standards such as ASHRAE 55 (ASHRAE, 2010). The standard claims that average air velocity may be increased to offset operative temperature increases and the average air velocity should be 0.8 m/s or less. This also brings out another perspective of

thermal comfort; thermal sensation of each body part is different and may be more significant during sleep when the head of the subject is exposed while the body is generally covered with bedding.

Many researchers have approached radiant heating systems by comparing its comfort criterion and energy performance to traditional systems such as forced or free convection. Qingyan Chen (Chen, 1990) reviewed the said systems and performed a subsequent indoor particulate and energy simulation. The results obtained by Chen were not entirely representative of current realistic systems because forced air was the primary system, with the radiant system operating as a secondary heating system. The ventilation rate used was in the vicinity of 1.0 ACH instead of the 0.33 ACH as designed in current buildings.

However the indoor particulate simulation may be indicative as the reduction of ventilation air was found to be proportional in the three cases tested by Chen (Chen, 1990). It was found that the forced air heating system had generally worse particulate concentration due to the re-circulating nature of the system. Thus it suggests that particulate concentration of a radiant system should not be worse than a traditional forced air system.

Similarly, Z. Zhang et al (Zhang & Pate, 1986) performed field measurements and experimentation in a research house with dual force air and radiant ceiling heating. It was found that the room air temperature followed quite closely to the radiant ceiling heating cycle and did not lag in response. The experimental

study suggested that the radiant ceiling successfully warmed the floor, which in turn heated the air convectively providing a thermal stratification less extreme than via traditional forced air systems. The radiant distribution was found to be fairly uniform and the radiant panels achieved thermal comfort for the occupants.

Similar to the research performed by Fanger et al., Imanari et al (Imanari, Omori, & Bogaki, 1999) experimented and simulated radiant ceiling heating and cooling systems in Tokyo. A total of 74 subjects were hired to participate in two sets of experiments in a room set up with a radiant and forced air system. Following the methods as proposed by Fanger et al. and by the PMV thermal comfort scaling, it was discovered that 72% of the participant found radiant ceiling cooling comfortable compared to 55% for traditional forced air system. Although the participant who found radiant ceiling heating comfortable is lower at around 38%, a similar relationship is established with forced air heating where only 35% of the participant felt comfortable. If neutral sensations were included, this will be 90% and 82% respectively. This also indicates that people are more sensitive to heat gain as more people found cooling to be more comfortable than heating.

A simulation of the radiant heating system was performed for a floor in an office building in Tokyo by Imanari et al. (Imanari, Omori, & Bogaki, 1999); it was found that over the course of a year, radiant heated and cooled ceiling

systems used roughly 10% less energy than traditional systems of the same technology level. However, more savings shall be possible given the advances in our understanding of thermal comfort and the subsequent development of controls.

Others such as McIntyre and Griffiths (McIntyre & Friffiths, 1972) tried to establish the heat sensory relationship between radiative temperature and air temperature. Sixteen male subjects were exposed to several thermal conditions in which the radiant and air temperatures were varied. A questionnaire with Fanger's seven-point comfort scale was utilized and a regression analysis of the results established that the subjects were more sensitive to air temperature where if there is a decrease of 1°C in air temperature, a rise of 1.5°C of radiant temperature is required to achieve the same thermal sensation.

Coupled with the lower heat transfer coefficient for convective heat transfer in relatively still air of $3.1 \text{ W}/(\text{m}^2\text{K})$ and $4.0 \text{ W}/(\text{m}^2\text{K})$ to roughly $4.7 \text{ W}/(\text{m}^2\text{K})$ for radiative heat transfer (ASHRAE, 2009) it means that a radiant system will be inherently less disturbing than a forced air system for the same heating load; as less radiant temperature difference is required for the same heating output as forced air temperature difference.

Note that McIntyre et al. (McIntyre & Friffiths, 1972) research is based on thermal sensation which is not thermal comfort or thermal neutrality. Thermal sensation is inversely proportional to thermal comfort as the greater the

thermal sensation will result in less thermal comfort. It was also noted that the subjects felt a decrease in freshness with the increase of both radiant and air temperature. Although the experiments show a slight freshness feeling in favor to radiant system, the difference is small and the sample size is insufficient to conclusively prove this.

Similarly, Toftum (Toftum, 2002) confirms that increased temperature promote heightened sensation to noise, air quality and discomfort. In his review of 5656 set of ASHRAE field data on draft movement, it was found that about 40% of people preferred more air movement even when they feel cool. This holds true where air movement is less than 0.00 m/s to 0.25 m/s. It was also noted that a sedated person is much more sensitive to draught, and hence to a forced air system.

In summary, existing research and practices have determined that radiant heated systems can often provide relatively instantaneous heating and cooling; and is more energy efficient as an overall heating system while also being less disturbing to the recipients. However, existing research has a focus on the awake condition and there was no thermal comfort standard to deal specifically with sleep and radiant temperature. A more fundamental understanding of sleep efficiency and thermal comfort from the psychological and physiological stand point shall be investigated.

2.4. Sleep Thermal Regulation and Thermal Comfort

Many studies have been completed on the thermal environment implication to psychological and physiological responses. Most are in relation to medical conditions or necessities, such as sleep deprivation thermal regulatory changes (Savourey & Bittel, 1994) and thus needs to be interpreted against thermal comfort criteria.

Current defined criteria for steady state thermal comfort include specific requirements for air temperature, radiant temperature, air velocity, humidity, clothing level and metabolic rate. Here, air temperature includes air temperature stratification and radiant temperature includes radiant asymmetry and indoor air quality. Additional criteria include state of mind, draft, noise and indoor lighting.

If the definition of thermal comfort is broadened, it has the connotation of the thermal condition which the person may perform work most efficiently at the least fatigue and that is thermally neutral. As the purpose of sleep can be seen as a phase of our daily cycle to allow our body to rest and our mind to reset; thus for sleep, thermal comfort can be taken to mean the condition which best allows sleep to occur and rejuvenate the body and mind.

Considering thermal condition on sleep quality, Candas et al (Candas, Libert, & Muzt, 1982) researched transient thermal conditions on Slow Wave Sleep (SWS) and Rapid Eye Movement (REM) stages of sleep. Over a period of 5 days, five semi-nude male subjects slept in a weather chamber kept at an operative

temperature of 32°C and an air temperature of 25°C. The air temperature was increased to 41°C or decreased to 28.5°C during each specific stage of sleep and the subject skin temperature and response was recorded.

It was found that in 50% of all cases when a skin warming of 0.8°C was detected the subjects woke up. While for sleep interruption such as a sleep stage change, during the period of skin warming, 30% of the subjects were interrupted by the heat during SWS and 66% of the subjects were interrupted during REM. The amount of interruption was found to be similar when the operative temperature was increased by 1.6°C/min or decreased by 0.8°C/min, suggesting that sleep termination was more of a state function than a path function.

In the small scale experiment conducted by Candas et al., there are insufficient data to conclude what temperature increase caused the change in sleep condition as there were instances where no sleep condition changed even when 1.2°C of temperature rise was detected on the skin. This research also supports the notion that cooling is the main source of discomfort during sleep where it questions Fanger's 4°C radiant asymmetry limit as it is based on a subsequent reduced air temperature with radiant temperature increase to maintain a uniform operative temperature. Note that the temperature increase ramp in Candas et al. research is significantly higher than the allowable ramp of ASHRAE 55 (ASHRAE, 2010) of 1.1°C/15min.

Candas et al induced the thermal conditioning via forced convection, which differs from radiant heating, however their documentation is via skin temperature, which is affected more instantaneously by radiant heating, thus the suggestion that rapid temperature change will induce sleep thermal discomfort should be applicable in this research.

Other research into sleep quality and thermal conditioning utilized a sleep efficiency index to gauge how good sleep is under two thermal environments. Okamoto-Mizuno et al (Okamoto-Mizuno, Tsuzuki, Mizuno, & Iwakic, 2005) experimented with 16 subjects and subjected them to two environments over a period of several months in a climate chamber. The first environment the subjects were exposed to involved a typical warm environment of 26°C at 50% RH and was transited to a hot humid condition of 32°C at 80% RH during mid-sleep.

The second environment was the reverse where subjects were exposed to a condition of 32°C at 80% RH and transited to an environment of 26°C at 50% RH during mid-sleep. Although the change in relative humidity will not change as such with a radiant heating system, this research may be of importance as Okamoto-Mizuno et al found that subjects had a higher sleep efficiency index during the environment where the warm condition was transited to the hot and humid condition. This finding is in agreement with research presented by Lack et al (Lack, Gradisar, Someren, Wright, & Lushington, 2008) where the body

needs to dissipate heat prior to entering sleep. In both environments during the hot and humid condition, the sleep efficiency index decreased.

Bischof et al (Bischof, Madsen, Clausen, Madsen, & Wildschodtz, 1993) preformed a pilot study in microclimate within a bed for thermal comfort during sleep when conscious thermal control such as removing article of clothing or adjusting the indoor temperature is unavailable. Five test subjects were subjected to various conditions such as spot cooling on the head and different levels of clothing. It was found that for the subjects, the comfort regime is with a skin temperature of approximately 32.5°C with Fanger's PMV measurements. This skin temperature value is also similar to those found by Huang (Huang, 2008) between 31.5°C to 32.9°C.

It was also found that the motoric level of a person during sleep can be strikingly different during different sleep periods and that localized skin temperature of a person may be significantly different. Sleep comfort was found to be more comfortable with a lower temperature during the evening prior to sleep.

Based on percentage of stage-3 and stage-4 sleep and the percentage of rapid eye movement sleep, this paper established that the thermal climate outside of the immediate micro- climate zone is largely irrelevant to thermal comfort. By the PMV measurements, Percentage of REM and percentage of stage-3 and

stage-4 sleep all suggest a skin temperature of 35°C or more will cause discomfort and reduction in the quality of sleep.

Bischof et al. (Bischof, Madsen, Clausen, Madsen, & Wildschodtz, 1993) looked at transient state for sleep comfort where several additional factors were also considered. This includes rate of change of air temperature, rate of change of radiant temperature. Other research such as Lack et al. (Lack, Gradisar, Someren, Wright, & Lushington, 2008) stresses the importance of pre and post sleep conditioning such as helping the subject fall asleep by allowing heat to dissipate from the body.

Lack et al (Lack, Gradisar, Someren, Wright, & Lushington, 2008) studied the requirement of sensible heat loss from the distal area of the human body to enter sleep and the correlation between core body temperature and different types of insomnia. Although the authors were unable to determine if insomnia is a cause or effect of body thermoregulation, it was shown that during sleep the body core temperature drops approximately 2°C. Thirteen subjects with different insomnia were tested and that subjects with Early Morning Awakening insomnia have their core body temperature cycle shifted forward while subjects with Sleep Onset Insomnia had their body core temperature cycle delayed.

Insomnia from hyper arousal was also explored and subjects were found to have higher core temperature and lower distal temperatures than typical good sleepers. This research reinforced the inverse relationship where distal

temperature increase promotes core temperature decrease suggesting that the body is required to reject core heat for initiating and maintaining good quality sleep. It is argued that a higher core temperature represents a longer sleep-wake state, and thus poorer quality of sleep.

The importance of heat rejection from distal areas such as the hands and the feet for entering good sleep was discussed and concluded to be the primary process that drives the circadian rhythm of core body temperature

The distal area skin temperature was further investigated by Sakoi et al (Sakoi, Tsuzuki, Kato, Ooka, Song, & Zhu, 2007) by exposing 35 different thermal conditions on both a manikin and on test subjects involving different air temperatures, relative humidity and radiant fields on a seated position. It was found that thermal comfort on the head is dependent on both the skin temperature and sensible heat loss while thermal comfort on the foot is only dependent on skin temperature.

The experiment by Sakoi et al. (Sakoi, Tsuzuki, Kato, Ooka, Song, & Zhu, 2007) with subjects shows a trend that more thermal discomfort on the head is reported when radiant heating is applied from above and cooling is from below than cooling from above and heating from below. Similarly a preference in comfort by heating from the front of the body and cooling from the back is established. Regardless of radiant heating and cooling output, thermal comfort via skin temperature is found to be in the 33.5°C range for subjects at semi-

rest, however skin temperature was not found to necessarily represent thermal comfort. The forehead can lose up to 80 W/m^2 without the person feeling discomfort; this is in general agreement with Huang (Huang, 2008) who suggest that soldiers exposed to the cold outdoor environment were comfortable without face masks when sleeping in sleeping bags, except for more extreme environments.

Agreeing with prior research, it was found that the subjects find less discomfort with a cold ceiling than a warm one. It was found that the subjects felt the most comfortable if they were able to lose around 36 W/m^2 of sensible energy regardless of the magnitude of the radiant field asymmetry tested. This is similar to the values developed for sleeping bags from soldiers of 34.9 W/m^2 of heat loss quoted by Huang (Huang, 2008) or up to 43.5 W/m^2 with a heat debt of 334.944 kJ.

The radiant temperature differential recorded was not as per the *radiant temperature asymmetry* as defined by Fanger et al but as a differential to the room air temperature without a view factor and the operative temperature was not kept constant. Their human experimentation suggests that less than 20% of the subjects was uncomfortable for the cold floor case of a room with air temperature of 25.5°C where the ceiling was at 33.5°C and the floor was at 16.5°C ($\pm 8^\circ\text{C}$); and for the warm floor case where the air temperature was at 28.0°C and the ceiling was at 14°C and the floor was at 42°C ($\pm 14^\circ\text{C}$).

Regarding sleep thermal comfort, being asleep differs from being awake by the inability of the person to consciously change their thermal environment, such as turning on a fan, or putting on or taking off an article of clothing, and that body conditions are not uniform but varies during different stages of sleep. Thus unlike thermal comfort while awake, it is unreasonable to assume that a static thermal condition will provide sleep thermal comfort.

Atmaca et al (Atmaca, Kaynakli, & Yigit, 2007) researched the effect of radiant heat transfer to individual body parts and their correlation to thermal comfort for a seated person. The second part of this research focused on simulating the internal radiant temperature fields of building envelopes exposed to strong solar radiation.

Their research concludes that asymmetric radiant fields cause the most thermal discomfort for a person. They found that a correlation between thermal comfort and radiant surface to room air temperature differential exist and that thermal comfort measured on a PMV scale is a function of skin surface temperature.

It was also found that with a radiant surface temperature of 12°C above ambient average room air temperature, the warming effect is noticeable.

Similarly, it was found that typically a skin heated to a temperature of 33.6°C via radiant heating will show the same level of warm feeling. Exposed skin had

been found to have the greatest temperature distortions compared to the clothed portion of the body.

Based on equivalent physiological response to thermal conditions of conscious and unconscious humans and a uniform sleep comfort condition, Z. Lin and S. Deng (Lin & Deng, 2008) developed a model to predict sleep thermal comfort. Their model predicted that a neutral radiant and air temperature of 29.6°C will provide a naked person sleep thermal comfort while with typical sleep attire, the neutral temperature drops to 24.2°C. This agrees well with previous experimental studies by others.

In summary, it was found that during sleep, the skin and core temperature of a person is much more dynamic than that of a person who is awake. This may invalidate the typical steady state heat transfer model used in building science research.

Recognizing that building heating systems will not likely be operated as per for a single person, the skin temperature value of 32.5°C will be accepted as the thermal condition at which sleep will be sufficiently efficient and a heat loss rate of 36 W/m² found to be comfortable for sedated persons will be assumed to hold true for sleeping persons.

2.5. Localized Body Response to Thermal Environment

Sleeping in a radiant ceiling heated environment will expose a person's face directly to the radiant panel while the body will be covered with covering of high insulation value, thus logically, the exposed body parts will participate more in heat transfer than the covered parts. Huizenga et al (Huizenga, Zhang, Arens, & Wang, 2004) studied 109 human subjects about their core and distal temperatures with localized body part being heated or cooled by the means of forced air via ventilated sleeves. It was found that in a thermally neutral environment, heating or cooling of distal areas of the body such as the hands will promote an inverse effect on the temperature of the person's core, while heating or cooling the core area of the body directly will promote positive correlation in core temperature rise and fall. The face of the subjects were found to be the most sensitive to localized heating or cooling and produces instantaneous response in core temperature compared to several minutes of lag for other distal areas such as the hands. Although the experiments were performed with forced air instead of radiant system, the body's reaction to localized distal and core tempering is likely to be similar.

2.6. Thermal Modeling between the Body and the Thermal Environment

There are two types of thermal models that are widely used; these are the single node model and the two -node model. The single node model is suggested by ASHRAE Fundamentals (ASHRAE, 2009) and have been widely used

by researchers such as (Goldman, 1988)(Holand, 1999) and was originally proposed by Fanger in (Fanger P. , 1970) It treats the human body as a single finite unit with heat transfer acting directly on its surfaces.

The two node models for heat transfer assumes the human body as two concentric mass of flesh each as one node with the outer layer representing the skin and distal area of the body while the center representing the core of the body. Notable two node models include the improved Pierces model developed by Gagge (Gagge, Stolwijk, & and Nishi, 1971) and the Kansas State University model (Azer, 1977).

Both models have been expanded from a single body to segregated bodies to represent limbs for a better representation of the actual human body. The two node model should give better results but because of the lack of knowledge regarding how the human body responds during sleep, it could not be employed with confidence.

2.7. Mean Radiant Temperature

The mean radiant temperature requires special attention as it is inherently very hard to determine. The mean radiant temperature concept as first suggested by Korsgaard (Korsgaard, 1949) has been adopted by ASHRAE (ASHRAE, 2010) and EN ISO (ISO, 2002) as an index of the thermal environment within a room or a space.

ASHRAE Fundamentals (ASHRAE, 2009) has defined the mean radiant temperature as: *“the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual nonuniform enclosure”*. Radiant heat transfer is in the long wave spectrum and the requirement of the human body means that the mean radiant temperature is location specific.

There are two accepted approaches to determine the actual mean radiant temperature. The first is the black globe method and the second is the surface temperature method.

2.7.1. Black Globe Thermometer

The black globe thermometer method of determining mean radiant temperature has been well established (T. BEDFORD & C. G. WARNER, 1934) and was quoted by ASHRAE fundamentals (ASHRAE, 2009). The black globe thermometer is originally a copper water closet float sphere painted black with a thermometer inserted in the center.

The black globe was found to be representative of the view factor ratio of a human body and the relationship of the globe temperature and the mean radiant temperature given by:

$$T_{MRT} = \left[(T_g + 273)^4 + \frac{1.10 \times 10^8 \times V_a^{0.6}}{\varepsilon_{globe} \times D^{0.4}} \times (T_g - T_a) \right]^{0.25} \quad (1)$$

Where;

T_{MRT} = Mean Radiant Temperature – K

T_g = Globe Temperature – °C

V_a = Air Velocity – m/s

ε_{globe} = Emisivity of Black Globe – 0.95

T_a = Air Temperature – °C

D = Diameter of Globe – m

The globe temperature was found to be representative of the operative temperature for a person who is seated (ASHRAE, 2009). However, it was found to overly weigh ceiling and floor radiant exposures for a standing person and likewise should overly weigh the floor and wall exposures in the case of a reclined person.

Derivative and configuration of the Black Globe method were employed to remove the convective component of the heat transfer thus eliminating the need of measuring air velocity by the dual globe method. This method (ISO, 2002) utilizes one black globe and one polished globe where the polished globe is used as the representation of heat transfer due to convection. The other variation is an air temperature controlled globe used to eliminate the convective heat exchange portion of the heat transfer faces similar restrictions.

The black globe is known to overestimate the mean radiant temperature in the cases with direct sunlight (ISO, 2002).

2.7.2. Surface Temperature Measurement Approach

The other method requires the knowledge of the surface temperatures and the calculation of view factors to the human body. The mean radiant temperature may be determined by the following equation from ASHRAE Fundamentals:

$$T_{MRT}^4 = \sum_{i=0}^n T_n^4 \times F_n \quad (2)$$

Where;

T_{MRT} = Mean Radiant Temperature – K

T_n = Wall Surface Temperature – K

F_n = View Factor – To be determined by site plan layout

The surface temperature method has been used to calibrate black globe sensors in climatic chambers and should be the most accurate method due to its proximity to the actual definition of mean radiant temperature.

The surface temperatures can be determined by an array of surface temperature sensors, or by mapped infrared cameras. The surface temperature method requires a sufficient amount of temperature sensors to construct an accurate depiction of the surfaces and suffers when the surfaces have irregular temperature gradients. While the infrared camera approach can handle more complex temperature profiles, infrared cameras are much more expensive.

The view factor is a geometric correlation between each surface and the human body and is dependent on the position of the person. Being a spatial function, view factor can be described as the ratio of heat a person receives in the direction of the particular surface to which the view factor corresponds. View factors can be developed by partial differential equations but have been linearized to basic room shapes by Fanger (Fanger P. , 1970) for people who are standing and for people who are seated.

Alternatively, view factor can be determined by the Nusselt sphere method (Nusselt, 1928) which is a simpler method that compares the projected surface area of the specific surfaces onto an imaginary sphere located over the host object to the sphere's cross sectional area. With Nusselt's view factor theory, a simple photographic method on a polished sphere may be used to determine view factor.

2.8. Summary

In summary, radiant heating systems were found to be generally more comfortable than forced air systems. However, it was established that vertical radiant asymmetry is a likely cause of thermal discomfort in radiant heating systems. Existing thermal comfort standards were found to specifically exclude the sleep condition.

The sleep thermal comfort was explored in terms of human thermal regulation and its cardiac implications at different stages. It is inherently complex and a widely accepted understanding is unavailable to construct an accurate model.

Researchers have analyzed field studies data and simplified the model to a steady state and found that an average heat loss rate of 36 W/m^2 at a skin temperature of 32.5°C is the ideal body thermal condition to provide optimal sleep comfort.

Commonly used body thermal model, the single node and two node model were explored. Existing steady state thermal model based on thermal neutrality was found not to be a good representation during sleep as existing literatures established that for sleep thermal comfort, a gain or deficit of heat is required at different stages of sleep.

The existing tools to quantify a thermal environment were reviewed and the method of determining mean radiant temperature was found not to be ideal for dynamic building system control and measurement due to its significant lag of 30 minutes.

3.0 Heat Transfer Model

The heating system should be able to produce the ideal thermal environment in the room for a person to sleep well. To determine this ideal thermal environment, a model of the body's thermal environment was constructed (body model). Conditions for optimal comfortable sleep were used as inputs in the body model in order to determine the room's ideal thermal environment. Finally, room's ideal thermal environment was used as input data to engineer the heating system required. The main requirement is for the total heat dissipation by a body to be 36 W/m^2 and that the skin temperature of a sleeping person should be 32.5°C . A set of equations describing all available heat transfer mechanisms was utilized to determine the approximate combination of mean radiant temperature and mean air temperature that fulfills these criteria.

3.1. Heat Transfer Model

Thermal comfort is typically calculated for the steady state as the awake state is typically chaotic and hard to predict. This may not be adequate for the sleep state. As suggested by existing research, sleep is not a thermally neutral state and the metabolism rate during sleep is 0.7 met or 41 W/m^2 for an average American male (ASHRAE, 2010) and the desired heat loss rate for sleep is 36 W/m^2 as developed above.

As the body produces more heat than it losses, there is a slight heat gain over a typical sleep period of 8 hours. With a 50th percentile US male, this results in a body temperature rise of roughly 0.8°C which closely match the 1°C of body temperature rise over the sleep period to wake up the person as described by Lack et al. (Lack, Gradisar, Someren, Wright, & Lushington, 2008).

Although a steady state model is not representative of the dynamic process of sleep as discussed earlier, without a better understanding and the knowledge of the sleep thermal gain and deficit interaction; the comfort criteria are only a skin temperature and a heat transfer rate, a single node model at steady state is utilized in this research. As with previous studies such as (Holand, 1999), it is sufficient to determine the thermal environment required with only the desired skin temperature and heat transfer rate.

3.2. Setup of Heat Exchange Model

3.2.1. Thermal Environment

A thermal environment was created to exchange heat with the body model. This simplified the thermal environment into the main heat transfer mechanisms. This includes radiant, convective, conductive and mass heat transfers and these processes are shown in Figure 1.

Radiant heat transfer from the body model to the thermal environment was calculated from the surface temperature of the exposed body or the covering

and to the radiant aspect of the thermal environment represented by the Mean Radiant Temperature.

Convective heat transfer from the body model to the thermal environment was calculated from the surface temperature of the exposed body or the covering and to the convection aspect of the thermal environment represented by the mean dry bulb air temperature and the mean air velocity.

Latent heat transfer from the body via respiration was calculated with the mean air temperature and relative humidity of the air.

Conductive heat transfer from the body model to the thermal environment was calculated from the surface temperature of the covered body to the surface of the exposed covering surface. Conductive heat transfer acts across the insulative body covering layer from the body contact on the underside to the environment contact on the topside.

The body was broken into two sections, the head and the body. The reason for this is that the body may be covered by coverings while the head is mainly exposed.

3.3. Body Heat Exchange Model

Concerning the heat exchange of a body with the thermal environment, the body is separated to two parts, the exposed head and the covered body. The main heat transfer between the body and the thermal environment is the

respiratory heat transfer, radiant heat transfer and convective heat transfer. On the covered body, there is an additional conduction heat transfer between the body and the thermal environment through the blanket. Values for coefficients were determined by regression analysis of empirical data tables from ASHRAE and other sources.

Figure 1 Body Heat Transfer Diagram

$$\dot{Q}_{cond}(T_2 - T_3) = \dot{Q}_{rad}(T_3 - T_{MRT}) + \dot{Q}_{conv}(T_3 - T_A) \quad (3)$$

$$\dot{Q}_{tot} = \dot{Q}_{rad}(T_1 - T_{MRT}) + \dot{Q}_{conv}(T_1 - T_A) + \dot{Q}_{rad}(T_3 - T_{MRT}) + \dot{Q}_{conv}(T_3 - T_A) + \dot{Q}_{res}(T_E - T_A) \quad (4)$$

Where,

$$\begin{aligned} \dot{Q}_{cond}(T_2 - T_3) &= \text{Heat conduction between body coverings from skin contact to outer surface} \\ &\quad - W/m^2 \end{aligned}$$

$\dot{Q}_{\text{rad}(T_3-T_{\text{MRT}})} = \text{Heat lost from body to radiant environment by covering surface} - W/m^2$

$\dot{Q}_{\text{conv}(T_3-T_A)} = \text{Heat lost from body to convective environment by covering surface} - W/m^2$

$\dot{Q}_{\text{rad}(T_1-T_{\text{MRT}})} = \text{Heat lost from head to radiant environment} - W/m^2$

$\dot{Q}_{\text{conv}(T_1-T_A)} = \text{Heat lost from head to convective environment} - W/m^2$

$\dot{Q}_{\text{res}(T_E-T_A)} = \text{Respiratory heat loss to environment} - W/m^2$

$\dot{Q}_{\text{tot}} = \text{Desired heat loss from body} - 36 W/m^2$

$T_1 = T_2 = \text{Desired skin temperature} - 32.5^\circ\text{C} = 305.5^\circ\text{K}$

3.3.1. Sleep Efficiency Thermal Condition Criteria

Sleep efficiency and its relationship with the thermal condition do not have a well-established definitive relationship. From the available literature it is suggested that:

1. The thermal environment should help the body lose heat prior to the person entering sleep.
2. Rapid heating of a person at sleep resulting in more than 1°C of skin temperature rise will likely awake the person regardless of starting temperature.

3. A skin temperature of 35°C or higher will most definitely cause discomfort during sleep where the optimal skin temperature is 32.5°C.
4. A person asleep is less subjective to the change in temperature and humidity than when awake, but achieves better sleep in dryer conditions.
5. Mean air velocity shall be less than 0.8 m/s
6. A person at rest feels thermally neutral when they lose 36 W/m² of body heat. Given that a person at rest produces 41 W/m² therefore the person feels thermally neutral with a slight rise in body temperature over time.
7. Heating cycle should induce less than 0.8 °C of skin warming over a relative short period.

3.3.2. Body Analytical Data

The biometric data (ASHRAE, 2009) for an average 50th percentile American male was used as the human physical model. Since the thermal comfort criteria established above are based on skin surface temperature and a specific heat loss rate per unit area which both were independent of dimension and mass; the actual physical body characteristic does not matter.

However it is prudent to keep to the 50th percentile American male for the deduction of the convective heat transfer characteristic length; and to be consistent of the view factors correlation of the human body with the black globe sensor later on in the field study.

3.3.2.1. *Human Surface Skin Area and Body Mass*

Average adult male body surface area was found to be 1.8 m² and mass at 80 kg according to ASHRAE 55 (ASHRAE, 2010). Heat capacitance of human flesh was recorded at 3.47 kJ/kgK.

3.3.2.2. *Body Metabolic Rate*

Metabolic rate of an average sleeping male is: 0.7 met while equivalent 1 met equates to 58.2 W/m² according to ASHRAE 55 (ASHRAE, 2010). This means that a sleeping person will have an equivalent heat generation of roughly 41 W/m².

3.3.3. *Radiant Heat Exchange*

The Radiant heat exchange is assumed to be:

1. Occurring between the person and his/her surrounding represented by the mean radiant temperature
2. Steady state condition, the person is motionless
3. Plane to plane radiant heat transfer, minor role for view factor as a person or furniture get moved around.

3.3.3.1. *Sensible Heat Exchange*

$$\dot{Q}_{rad} = \sum_{i=0}^n \varepsilon_n \times \varepsilon_{MRT} \times A_n \times \sigma \times (T_n^4 - T_{MRT}^4) \quad (5)$$

Where;

$\dot{Q}_{rad}$ = Radiant heat lost to the radiant environment – W/m^2

ε_n = Emissivity of body section – Dimensionless

ε_{MRT} = Emissivity of surrounding – 0.95

A_n = Area of individual body part – m^2

T_n = Surface temperature of individual body part – K

T_{MRT} = Mean Radiant Temperature as experienced by body part – K

σ = Stefan – Boltzmann Constant (5.6704×10^{-8}) – $W/(m^2 K^4)$

3.3.3.2. *Radiative Heat Transfer Comparison*

ASHRAE Fundamentals quotes the linearized radiative heat transfer coefficient to be:

$$h_{rad} = 4 \times \varepsilon \times \sigma \times \frac{A_r}{A_D} \left(273.2 + \frac{T_{cl} + T_{MRT}}{2} \right)^3 \quad (6)$$

Where:

$\frac{A_r}{A_D} = 0.7$ for a seated person, 0.73 for a standing person

3.3.4. *Convective Heat Exchange*

The convective heat exchange is assumed to be:

1. Laminar flow
2. Horizontal cylinder approximation with the mass and density of an average American male
3. Dry air calculations
4. Steady state condition, the person is motionless

The convective latent heat exchange is assumed to be negligible. As per Guyton (Guyton & Hall, 1992) it was stated that for a typical person, sweating of the skin begins when the skin temperature reaches 37°C or greater. As the design condition of the skin is 32.5°C, evaporative heat exchange will not be considered.

The convective sensible heat exchange is calculated using equation below:

$$\dot{Q}_{conv} = \sum_{i=0}^n \frac{A_n N_u k}{L_n} \times (T_{s_n} - T_a) \quad (7)$$

$$\dot{Q}_{conv} = \sum_{i=0}^n \left\{ 0.6 + \frac{0.378 \left(\frac{g \beta \rho^2 |T_{s_n} - T_a| L_n^3 c_p}{\mu k} \right)^{1/6}}{\left[1 + \left(\frac{0.559 k}{c_p \mu} \right)^{9/16} \right]^{8/27}} \right\}^2 \frac{k A_n}{L_n} \times (T_{s_n} - T_a) \quad (8)$$

Where;

$$\dot{Q}_{conv} = \text{Total convective heat transfer} - W$$

$$A_n = \text{Surface of part} - m^2$$

N_u = Nusselt Number

κ = Thermal conductivity – W/mK

L_n = Characterstic length of part – m

R_a = Rayleigh Number

g = Gravitational Acceleration – Constant 9.81 m/s²

β = Coefficient of Thermal Expansion – m³/K

ρ = Density of Air – kg/m³

T_{s_n} = Surface Temperature of part – K

T_a = Average Air Temperature – K

C_p = Specific Heat at constant pressure – kJ/kgK

μ = Absolute Viscosity = $\nu\rho$; $-\frac{\text{kg}}{\text{ms}} = \text{PaS}$

ν = Kinematic Viscosity – m²/s

Nusselt Number

$$N_u = \left\{ 0.6 + \frac{0.378R_a^{1/6}}{\left[1 + \left(\frac{0.559}{Pr} \right)^{9/16} \right]^{8/27}} \right\}^2 \quad (9)$$

Rayleigh Number

$$R_a = G_r P_r \quad (10)$$

Grashof Number

$$G_r = \frac{g\beta\rho^2|\Delta T|L^3}{\mu^2} \quad (11)$$

Prandtl Number

$$P_r = \frac{c_p\mu}{k} \quad (12)$$

Density of Air

Based on the Gas Law, the density of air is:

$$\rho = \frac{P_d}{R_d T_a} + \frac{P_w}{R_w T_a} \quad (13)$$

Where;

P_d = Dry air pressure (partial) – Pa

R_d = Gas Constant of dry air – 287.042 J/kgK

P_w = Water vapor pressure (partial) – Pa

R_w = Gas Constant of water vapor – 461.524 J/kgK(263K – 323K)

Given that for the dry bulb air temperature range explored is $283^\circ K < T_a < 303^\circ K$; the water vapor pressure at saturation is 1.228 kPa and 4.246 kPa respectively. With an atmospheric pressure of 101.325 kPa, water vapor represent 0.76% and 2.72% respectively. Given that moist air will not be saturated, the contribution to density is minimum therefore dry air calculation is acceptable.

Leradi's (Ierardi, 2000) regression fitting of SFPE Handbook of Fire Protection Engineering 2nd Edition Table B-2 for the density of dry air:

$$\rho = 360.77819T_a^{-1.00336} = \text{Density of Air} - \text{kg/m}^3$$

For a maximum error of -1.35% at an average error of 0.25% between 100K and 1600K

Coefficient of Thermal Expansion

$$\beta = \frac{dV}{dT_a} = \text{Change in volume in relationship with temperature} - 1/K \quad (14)$$

Since

$$V = \frac{1}{\rho} = \text{Volume of dry air} - m^3 \quad (15)$$

Taking Leradi's density empirical equation;

$$V = \frac{1}{360.77819T_a^{-1.00336}} \quad (16)$$

$$V = 0.002771786177T_a^{1.00336} \quad (17)$$

Therefore;

$$\beta = 0.002781099379T_a^{0.00336} \quad (18)$$

Specific Heat

Leradi's (Ierardi, 2000) regression fitting of SFPE Handbook of Fire Protection Engineering 2nd Edition Table B-2 for the Specific Heat Capacitance of dry air:

$$C_p = 1.9327 \times 10^{-10} T_a^4 - 7.9999 \times 10^{-7} T_a^3 + 1.1407 \times 10^{-3} T_a^2 - 4.4890 \times 10^{-1} T_a + 1.0575 \times 10^3 - \frac{kJ}{kgK} \quad (19)$$

For a maximum error of -0.38% at an average error of 0.10% between 100K and 1600K

Thermal Conductivity

Leradi's (Ierardi, 2000) regression fitting of SFPE Handbook of Fire Protection Engineering 2nd Edition Table B-2 for the thermal conductivity of dry air:

$$k = 1.5207 \times 10^{-11} T_a^3 - 4.8574 \times 10^{-8} T_a^2 + 1.0184 \times 10^{-4} T_a - 3.9333 \times 10^{-4} - \frac{W}{mk} \quad (20)$$

For a maximum error of -0.83% at an average error of 0.3% between 100K and 1600K

Absolute Viscosity

The absolute viscosity was regression analyzed from Leradi's (Ierardi, 2000) regression fitting of SFPE Handbook of Fire Protection Engineering:

Table 2 Dynamic Viscosity of Air at Various Temperatures

Temperature (K)	Dynamic Viscosity (kg/ms)
100	0.000006924
150	0.000010283
200	0.000013289
250	0.00001488
300	0.00001983
350	0.00002075
400	0.00002286

Figure 2 Regression Analysis of Dynamic Viscosity of Air at Different Air Temperatures

The regression analysis produces the following equation:

$$\mu = -2.3822 \times 10^{-13}T_a^3 + 1.3125 \times 10^{-10}T_a^2 + 3.6983 \times 10^{-8}T_a + 2.2925 \times 10^{-6} - \frac{kg}{ms} \quad (21)$$

Convective Heat Transfer Comparison

ASHRAE Fundamentals(ASHRAE, 2009) quotes (Mitchell, 1974) a convective heat loss coefficient of 3.1 W/m²K for a seated person with an air velocity 0.0 m/s to 0.2 m/s; (Colin & Houdas, 1967) 5.1 W/m²K for a reclining person in moving air with an air velocity 0.0 m/s to 0.15 m/s; (Seppänen, McNall, Munson, & Sprague., 1972) 4.0 W/m²K for a standing person in moving air with an air velocity 0.0 m/s to 0.15 m/s.

Given the geometrical differences between that used in the developments of these coefficients to the horizontal sleep condition, the calculated convective heat transfer coefficients between $1 \text{ W/m}^2\text{K}$ to $4 \text{ W/m}^2\text{K}$, as shown in Table 5, are considered reasonable.

3.3.5. Conductive Heat Exchange

The human body when sleeping is generally in contact with the various bedding material, this includes the bed material and the coverings. For a person standing or sitting, conductive heat transfer is less of a concern as the contact surface area such as the feet to the floor is small and the contact area is often insulated, e.g. carpets, clothing. Therefore conductive heat transfer was considered only for the sleep duration.

The assumptions used in the conductive heat exchange calculation include:

1. Micro thermal environment between covering and the human body is completely efficient, that the air pockets are thin and the skin conducts with the blanket directly.
2. Conduction is unidirectional, from a uniform source to a uniform environment.
3. Steady state condition, the person is motionless

Upward Covering Conduction

The body of a sleeping person is generally covered by some sort of covering; heat will have to travel through the covering from the body via conduction prior to being exchanged radiantly and convectively.

$$\dot{Q}_{cond} = \sum_{i=0}^n \frac{\kappa_n A_n (T_h - T_c)}{D} \quad (22)$$

Where;

$\dot{Q}_{cond}$ = Total conductive heat transfer – W

κ_n = Thermal conductivity of the material – W/mK

A_n = Wetted area of heat transfer – m²

T_h = Temperature of hot side, equal to T_2 – K

T_c = Temperature of cold side, equal to T_3 – K

D = Thickness of material – m

Downward Bedding Conduction

The mattress or bedding a person sleeps on in the northern hemisphere typically is of significant insulation value. ASHRAE 55 (ASHRAE, 2010) addresses the downward heat transfer by a person sitting on a chair by applying an additional overall insulation value to the entire body. For an executive chair, this is considered to be an additional 0.15 clo for persons dressed 0.5 to 1.2 clo. But a thin chair has much less wetted area between the chair and the body compared to person on a bed and is significantly thinner than a mattress.

A mattress made of separate spring compartments, foam or other material with significant air pockets, a 200 mm thick mattress will have an insulative value of roughly RSI 5 to RSI 10 ($\text{m}^2\text{K/W}$) depending on the filling, with additional insulation value provided by the box frame and other common bed frame accessories. Compared to winter coverings of an insulative value of 1 to 13.5 TOG (RSI 0.1 to RSI 1.35), heat loss via downward conduction was considered insignificant.

3.3.6. Respiration Heat Exchange

Cain et al (Cain, Livingstone, Nolan, & Keefe, 1990) found that the exhaled air temperature and humidity for five male subjects are relatively independent of ambient temperatures. The exhaled air in an ambient temperature of between 20°C to -40°C is between 34°C and 30°C respectively.

The ambient air was dry while the exhaled air was either at saturation, super saturation or near saturation. The average heat loss was found (Cain, Livingstone, Nolan, & Keefe, 1990) to be at around 25 W with an ambient temperature of 0°C to -40°C . The average respiratory rate was found to be 8.7 L/min. The heat exchange by respiration is given by:

$$\dot{Q}_{res(T_E-T_A)} = \rho \dot{V} C_p (T_E - T_A) + \rho \dot{V} (\gamma_E - \gamma_A) h_{vap} + \rho \dot{V} \gamma_A C_{pv} (T_E - T_A) \quad (23)$$

$$\dot{Q}_{res(T_E-T_A)} = \rho \dot{V} [C_p (T_E - T_A) + (\gamma_E - \gamma_A) h_{vap} + \gamma_A C_{pv} (T_E - T_A)] \quad (24)$$

Where removing the averages to the specific heat capacitance;

$$Q_{res(T_E-T_A)} = \rho \dot{V} \left[(C_{p_E} T_E - C_{p_A} T_A) + (\gamma_E - \gamma_A) h_{vap} + \gamma_A (C_{pv_E} T_E - C_{pv_A} T_A) \right] \quad (25)$$

Where;

$\dot{V}$ = Respiration volumetric rate $\frac{8.7L}{min}$ or $1.45 \times 10^{-4} m^3/s$ (Cain, Livingstone, Nolan, & Keefe, 1990)

h_{vap} = Latent heat of vaporization for water – kJ/kg

C_{p_E} = Heat capacitance of dry air at T_E – $\frac{kJ}{kgK}$

C_{p_A} = Heat capacitance of dry air at T_A – $\frac{kJ}{kgK}$

T_E = Temperature of expelled air from lungs – 303K

(Cain, Livingstone, Nolan, & Keefe, 1990)

γ_A = Humidity ratio of inhaled air into lungs, assumed at 0% RH conservatively

– 0 Kg_w/Kg_A

C_{pv} = Heat capacitance of water vapor, linearized between 275K and 325K

Humidity Ratio

Humidity ratio of expel air from lungs – at saturation $\frac{kg_w}{kg_A}$, curve fitted from:

Table 3 Maximum Amount of Moisture Air Can Hold at Various Temperatures

Temperature (K)	Maximum Humidity Ratio (kgw/kgd)
273	0.003767
278	0.005387
283	0.007612
288	0.01062
293	0.014659
298	0.019826
303	0.027125

Maximum Moisture Air Can Hold Vs. Temperatures

Figure 3 Regression Analysis of Maximum Moisture Content Air Can Hold at Various Dry Air Temperatures.

$$\gamma_E = 6 \times 10^{-11} \times e^{0.0656T_E} \quad (26)$$

Heat Capacitance of Water Vapour

Heat capacitance of water vapour at T_E , curve fitted from

Table 4 Specific Heat Capacitance of Water Vapour at Various Temperatures

Temperature (K)	Specific Heat Capacitance (kJ/kgK)
250	1.855
275	1.859
300	1.864
325	1.871
350	1.88

Specific Heat Capacitance of Water Vapour at Various Temperatures

Figure 4 Heat Capacitance of Water Vapour Regression Analysis at Various Temperatures

$$C_{pv_E} = 5 \times 10^{-9}T_E^3 - 3 \times 10^{-6}T_E^2 + 0.0009T_E + 1.7725 \quad (27)$$

3.4. Operative Temperature

Operative temperature is often used to represent a thermal environment.

Operative temperature combines the radiant heat transfer with convective heat transfer into a single temperature value:

$$T_O = \frac{(T_{MRT} \times h_r + T_a \times h_c)}{(h_r + h_c)} \quad (28)$$

Where;

T_O = Operative temperature – K

T_{MRT} = Mean radiant temperature – K

T_a = Air temperature – K

h_r = linearized radiant heat transfer coefficient – $\frac{W}{m^2 K}$

h_c = linearized convective heat transfer coefficient – $\frac{W}{m^2 K}$

It is often simplified to:

$$T_O = \frac{(T_{MRT} + T_a)}{2} \quad (29)$$

3.5. Calculation Results

3.5.1. Respiratory Heat Loss

The respiratory heat loss was area weighted to the surface area of an average male. The respiratory heat loss composes between 4.13 W/m² to 1.66 W/m² between the air temperatures calculated which in turn is 11.5% and 4.6% of the desired heat loss rate of 36 W/m².

3.5.2. Convective and Radiative Heat Loss

The equations were solved for a general heat loss of 36 W/m^2 with a skin temperature of 32.5°C with MathCAD. The combination of mean radiant temperature, insulation value and mean air temperature to achieve this is displayed in the table and chart below.

Table 5 Mean Radiant Temperature, Mean Air Temperature and Bedding Insulation Value to achieve 36 W/m^2 of heat loss with a skin temperature of 32.5°C

Air Temperature 288K									
Insulation level (TOG)	MRT (K)	Surface Temperature (K)	Operative Temperature (K)	h_{rad} ($\text{W/m}^2\text{K}$)	h_{conv} ($\text{W/m}^2\text{K}$)	Q_{rad} (W/m^2)	Q_{conv} (W/m^2)	Q_{res} (W/m^2)	T_o (K)
0.00	311.17	305.50	299.58	6.00	3.76	-34.00	65.87	4.13	302.24
1.00	305.59	302.31	296.79	5.75	3.54	-18.82	50.69	4.13	298.88
3.00	294.35	295.94	291.18	5.26	2.97	8.33	23.54	4.13	292.06
5.00	283.54	289.56	285.77	4.82	1.84	29.01	2.87	4.13	284.77
7.00	273.17	283.19	280.58	4.41	2.55	44.16	-12.29	4.13	278.61
9.00	259.43	276.81	273.72	3.95	3.29	68.65	-36.78	4.13	272.41
Air Temperature 291K									
0.00	308.71	305.50	299.85	5.93	3.55	-19.03	51.43	3.60	303.62
1.00	303.07	302.26	297.04	5.68	3.29	-4.60	37.00	3.60	300.24
3.00	291.88	295.78	291.44	5.19	2.54	20.24	12.16	3.60	293.25
5.00	281.83	289.30	286.42	4.77	1.88	35.59	-3.19	3.60	285.65
7.00	269.66	282.82	280.33	4.32	2.99	56.82	-24.43	3.60	280.46
9.00	254.25	276.34	272.62	3.83	3.56	84.57	-52.17	3.60	274.64
Air Temperature 294K									
0.00	306.36	305.50	300.18	5.86	3.30	-5.03	37.94	3.09	304.98
1.00	300.70	302.21	297.35	5.61	2.98	8.44	24.48	3.09	301.56
3.00	289.80	295.63	291.90	5.13	1.85	29.91	3.01	3.09	294.26
5.00	279.33	289.04	286.67	4.70	2.57	45.63	-12.72	3.09	287.23
7.00	265.61	282.46	279.81	4.22	3.30	71.03	-38.11	3.09	282.32
9.00	248.42	275.88	271.21	3.70	3.79	101.51	-68.60	3.09	276.92

Air Temperature 297K									
0.00	304.14	305.50	300.57	5.80	3.01	7.86	25.55	2.59	306.33
1.00	298.54	302.16	297.77	5.55	2.59	20.05	13.36	2.59	302.85
3.00	288.38	295.48	292.69	5.09	1.81	36.17	-2.76	2.59	295.05
5.00	276.28	288.79	286.64	4.62	2.98	57.82	-24.41	2.59	288.81
7.00	261.08	282.11	279.04	4.11	3.56	86.38	-52.97	2.59	284.23
9.00	241.90	275.43	269.45	3.56	3.98	119.28	-85.87	2.59	279.27
Air Temperature 300K									
0.00	302.12	305.50	301.06	5.74	2.63	19.40	14.49	2.11	307.65
1.00	296.71	302.11	298.35	5.49	1.99	29.69	4.20	2.11	304.10
3.00	286.28	295.33	293.14	5.04	2.51	45.60	-11.71	2.11	296.23
5.00	272.79	288.56	286.39	4.53	3.28	71.42	-37.53	2.11	290.45
7.00	256.05	281.78	278.02	3.99	3.77	102.65	-68.76	2.11	286.21
9.00	234.57	275.00	267.28	3.41	4.15	137.74	-103.8	2.11	281.72
Air Temperature 303K									
300.3	305.50	301.69	5.69	2.08	29.14	5.21	1.66	308.9	300.38
295.5	302.07	299.25	5.46	1.57	35.81	-1.47	1.66	305.0	295.51
283.6	295.20	293.35	4.97	2.92	57.11	-22.77	1.66	297.5	283.69
268.9	288.33	285.95	4.43	3.53	86.08	-51.74	1.66	292.1	268.91
250.4	281.46	276.75	3.86	3.96	119.67	-85.32	1.66	288.2	250.49
226.2	274.59	264.64	3.25	4.31	156.76	-122.4	1.66	284.2	226.28

Note that MRT is mean radiant temperature, h_{rad} is radiant heat transfer

coefficient, h_{conv} is the convective heat transfer coefficient, Q_{rad} is the radiative heat transfer, Q_{conv} is the convective heat transfer, Q_{res} is the heat transfer by respiration and T_o is the operative temperature.

Mean Radiant Temperature Vs Mean Air Temperature to achieve 36 W/m² of heat loss with a skin temperature of 32.5°C

Figure 5 Acceptable Mean Radiant Temperature and Air Temperature at Different Insulation Levels to Achieve Heat Transfer Requirement and Skin Temperature Requirements.

Figure 5 shows the combinations of mean radiant temperature and air temperature at different insulation to help achieve a heat loss rate of 36 W/m^2 and a skin temperature of 32.5°C . The contours are not linear as the magnitude of the radiant heat transfer coefficient is based on the fourth power absolute temperature difference between the surfaces. The magnitude of the convective heat transfer coefficient is influenced by the relative temperature difference between the environment and the surfaces.

While both the radiant heat transfer and convective heat transfer is based on the interfaced surface which is the skin at 32.5°C (305.5K), the respiratory heat transfer is based on the core temperature which with mixing, the exhaled temperature is at 34°C (307K).

Given on the chart is a constant operative temperature line at 21°C (298K), which can be used to gauge if the mean radiant temperature or the air temperature has a greater influence. The cases with the blanket at 0 tog to 3 tog shows that the mean radiant temperature has a large impact. While the case with a blanket of 5 tog and above shows that air temperature is a more significant contributor. This is consistent with Table 5 showing that the convective heat transfer is the least between 3 to 5 togs of bedding insulation.

With typical indoor environments where the mean radiant temperature and air temperature cannot be too low, blankets should be between 3 to 5 tog. Figure 5 have been reconfigured into Figure 6 to illustrate this effect better.

Mean Radiant Temperature Vs Bedding Insulation Value to achieve 36 W/m² of heat loss with a skin temperature of 32.5°C

Figure 6 Mean Radiant Temperature Vs. Respective Bedding Insulation Level to Achieve Heat Transfer Requirement and Skin Temperature Requirements.

The mean radiant temperature of a heated building would not likely be lower than 290K (17°C) and at reasonable air temperatures of 288K to 303K (15°C to 30°C), the maximum insulation value of the blanket is 3 tog to 4 tog.

As a traditional summer blankets is around 2 tog and cold climate blankets are around 12 tog, the model suggests that in modern homes with tight envelopes, the typical heating set point will likely result in overheating during sleep.

3.6. Analysis and Discussion

The convective heat transfer coefficient calculated is similar to those quoted by ASHRAE fundamental (ASHRAE, 2009) for a seated person. It may be an overestimation due to the assumption that the body is represented by a free cylinder in the calculation while as a body asleep on a bed is more akin to a cylinder on a plane. This means that wetted area of the body is smaller than assumed and the geometry is less suitable at promoting convective flow.

Similarly as the body is lying on a bed, it suggests that a proportion of the body area is not acting in radiative and convective heat transfer. As discussed above, the conductive heat transfer into the bed itself is minimal and hence the 36 W/m² of heat loss rate may have to be weighted into the body area that is exposed.

The model suggests that blankets and coverings over 5 tog may induce overheating for people residing in buildings with tight envelopes that maintain

a relative high operative temperature during the night. Inversely this also means that the building may be lowered to an operative temperature of 15°C during sleep to conserve heating energy without impacting sleep thermal comfort.

4.0 Development of Mean Radiant Temperature Sensor

The importance of mean radiant temperature has been fully discussed by researchers (Fanger, Banhidi, B.W.Olesen, & Langkilde, 1980); (Fanger, Ipsen, Langkilde, Olesen, Christensen, & Tanabe, 1985) and (Korsgaard, 1949).

Mean radiant temperature and directional radiant temperature is typically measured by a globe thermometer or surface temperature method. However the typical response rate of a globe thermometer is approximately 20-30 minutes (ISO, 2002) which is too slow to document more rapid HVAC system operation changes or external factors such as solar incidence.

Surface temperature method will provide a more rapid assertion of the mean radiant temperature due to the quicker response of the sensors. But to deploy such an array of surface sensors to calculate mean radiant temperature is subjective of the actual surface temperature uniformity each surface sensor represents and may be impractical for larger or more public spaces for limited available sensor lengths and interference with the lifestyle of the occupancy.

A device based on the actual reception of radiant energy to determine mean radiant temperature such as the black globe is desirable but with a quicker response time. The black globe however requires an additional sensor to

measure the air velocity to derive the mean radiant temperature. The hot sphere anemometer used to derive the air velocities are fragile and is a sensitive piece of equipment that may not be ideal for deployment in high traffic areas.

Recognizing the infeasibility of measuring, monitoring and controlling mean radiant temperature, researchers (Jones & Chapman, 1994) have attempted to simplify mean radiant temperature determination. A simple singular sensor to determine mean radiant temperature would be appreciated.

An infrared sphere device is developed to measure mean radiant temperature. It was tested against surface temperature method and the black globe method in determining mean radiant temperature.

4.1. Traditional Method of Determining Mean Radiant Temperature

Both traditional methods of determining mean radiant temperature are deployed, the black globe method with hot sphere anemometer and the surface temperature method with surface temperature sensors and a view factor determination.

An array of direct surface mounted thermocouples will be attached on major surfaces in a dispersed manner. The exact location of the thermocouples will be determined with an infrared thermal camera to provide a more realistic average

of the surface temperatures at a much quicker response speed than a globe mean radiant temperature sensor.

The data to be collected will be used to develop a mean radiant temperature of the room and be used as a datum along with the globe mean radiant temperature to evaluate the effectiveness of the infrared sphere approach.

4.2. Infrared Sphere Approach

An alternative method of measuring the mean radiant temperature will be pursued in conjunction with the traditional globe sensor. Directional radiant temperature sensors are available but have a viewing range of around 20°.

The theory for this approach is based on the Nusselt Sphere approach (Nusselt, 1928) where the view factor of a plane area to a small plane element is the ratio of the plane area projection on an arbitrary hemisphere's two dimensional base area divided by the hemisphere's base area. As Farrell (Farrell, 1976) had shown that an area ratio by photography of a reflective hemisphere can estimate the angle factor between the radiant plane and the plane element with the main error being the roundness of the sphere and the truncation of the hemisphere due to the optical view angle.

For mean radiant temperature, sphere of highly polished low emissivity reflector with two radiant temperature sensors reversed facing will be used to determine the reflected room radiant temperature. The reflected room radiant

temperature reading theoretically is the actual mean radiant temperature with the influence of the reflector emissivity.

Unlike the black globe thermometer which is dependent on the radiant heat transfer and convective heat transfer that heats up the volume within the sphere, a low emissivity sphere theoretically is less susceptible to the actual sphere temperature. Emissivity is a function of temperature but is quite uniform; therefore will not be a function of air temperature and air velocity. As with the black globe thermometer, a linear view factor adjustment is not required as the reflector geometry inherently factor in the radiative flux levels.

4.2.1. Reflector Geometry

The infrared reflector will have similar geometry requirements as to parabolic reflectors employed in lighting fixtures. However, parabolic conic low emissivity reflectors are not readily available on the market while perfect hemispherical stainless steel, aluminum and copper balls are available.

The intent is to use the highly reflective surface of a polished metal ball to allow commercially available infrared sensors to view all of the radiant surfaces of the room. Two sensors will be positioned to read the reflected heat from a polished sphere. Each of the sensors will be tasked with reading half the room's quadrant.

Figure 7 Infrared Temperature Sensor with Convex Reflector

The intended infrared sensors have a view aspect ratio of 1:2, which means that viewing cone is 14.036° (28.072° for the entire cone). For this, the sensor will have to be located so that the extremity of the cone is receiving reflected energy from orthogonal to the sensor. This is calculated via Figure 8 Infrared Sensor Sphere Design Calculation as shown below.

Figure 8 Infrared Sensor Sphere Design Calculation

Where;

R = Radius of Sphere = 50.8mm

β = Viewing angle of IR sensor 2: 1 = 14.036°

α = Relected angle

θ = Reflection law line angle

x = Distance to viewing cone extremety

L = Distance of front of sensor to center of sphere

They are related geometrically by:

$$\beta + 90^\circ = 2 \times \alpha$$

$$L = R \cos \theta + x \cos \beta$$

$$R \sin \theta = x \sin \beta$$

With a 2:1 sensor, this gives:

$$L = 203.37 \text{ mm}$$

4.2.2. Assumptions and Limitations

The manufacturer of the infrared sensors was unable to provide sensor vignette information and it is assumed that the sensor vignette was weighted equally for center spot area and fringes of the reading cone.

The stainless steel sphere is assumed to be perfectly spherical and is polished to an emissivity of 0.03. This is assumed to be accounted for with the calibration of the sensor.

The construction and workmanship of the apparatus is assumed to be sufficiently within tolerances given the available tools.

4.2.3. Construction of the Apparatus

The construction of the apparatus is based on a stainless steel sphere with a radius of 101.6 mm (4"). It is elevated on a pedestal 76.2 mm (3") above the cantilever arm forming the base. The cantilever arm extends away from the centre of the sphere and supports two extensions that in turn support each infrared sensor.

The extensions are constructed so that the infrared sensor position in relations to the sphere can be fine-tuned. Each sensor is allowed to be adjusted vertically up and down; the nose of the sensor may be pitched up and down; and yawed left and right. A slot allows the sensor to be moved closer or further away from the sphere. In total, 4 degrees of freedom is afforded with only side to side displacement unavailable.

5.0 Experimental Design and Setup

5.1. Intent of Field Study

The intent of the field study includes several folds; the first is to compare the actual thermal environment created by a ceiling radiant heated room to the desired thermal environment calculated by the heat transfer model. The second is to field test a new mean radiant temperature sensor developed in this research and the third is to allow a comparison of radiant ceiling heating versus forced air heating.

To determine the sleep thermal comfort environment in a radiant heated room, thermal environment sensors were deployed in a room located at Caledon, Ontario, Canada to monitor the thermal environment. In the same room, two widely accepted methods of determining mean radiant temperature was deployed with the proposed Infrared Sphere as discussed in Section 4.2.

The three mean radiant temperature determination setup were also repeated in a forced air heated room located in Richmond Hill, Ontario, Canada. Additional sensors were deployed and data collected to allow for the comparison of the thermal comfort aspect of the forced air heating system versus the ceiling radiant heating system. The heating systems of both field test locations were operated normally.

5.2. Data Acquisition and Post Processing

Applying existing thermal comfort definitions and criteria to a person asleep may be challenging as some of the conditions may not exist. For example, the allowable air temperature stratification as defined by ASHRAE 55-2004 is 3°C from head to feet, where as if a person is asleep on the horizontal plane, the air temperature stratification difference from head to feet will always be relatively miniscule.

Similarly, the term radiant asymmetry has no meaning since a person who is asleep will be lying on a horizontal surface with no radiant exchange beneath him as he will be in contact with that surface and only be exposed to the radiant surface above him.

Thus for the sleep condition, some adaptation of the existing definitions will be required. This dictates what data can be acquired and which calculation is meaningful.

5.3. Intended Variables Acquired

The environmental variables collected were as follows:

Mean Air Temperature and Air Temperature Stratification

Mean air temperature and air temperature stratification was measured by an array of radiant shielded air temperature sensors equally spaced vertically 600 mm apart.

A single air temperature sensor located 600 mm above the finished floor at the center of mass of a sleeping person was used to determine the air temperature of the room.

Mean Radiant Temperature and Radiant Temperature Asymmetry

Mean radiant temperature was recorded with the black globe method and the infrared method as described in Section 2.7.1 and Section 4.2. The mean radiant temperature was also developed by the surface contact method as described in Section 2.7.2.

As the surface contact method inherently produces interior surface temperatures, it was used to calculate radiant asymmetry in the traditional three axis as per ASHRAE 55 (ASHRAE, 2010)

Mean Air Velocity

As the air velocity of the room will be low given the ventilation system and expected to be between 0.05 m/s and 0.5 m/s, an Omni-directional hot wire anemometer was deployed to record the mean air velocity at the center of mass of a sleeping person.

Outdoor Temperature and Relative Humidity

A radiant shielded outdoor temperature sensor and relative humidity sensor were deployed in a clearing at each field study location.

HVAC System Response

A temperature sensor was deployed to measure the supply temperature into the radiant temperature panel at the Caledon field test location. The supply of warm water into the panel was used as an indicator for heating initiation. For the Richmond Hill field test location, the highest air temperature sensor on the air temperature stratification array is within the down draft of the supply vent and therefore was indicative of heating cycle.

Meteorological Data

Meteorological data was extracted from the nearest weather station at Toronto Pearson Airport (YYZ) located at 43°40'39" North; 79°37'51" West for the duration of the monitoring.

Humidity and Air Quality

Humidity and air quality was measured by a humidity sensor, CO₂ sensor and VOC sensor respectively. The humidity sensor was located at the center of mass of a sleeping person. The CO₂ sensor and VOC sensor were a part of the building management system control suite at the Caledon field test location. It is wall

mounted at shoulder height and CO₂ and VOC data were only available for the Caledon site.

5.4. Aliasing and Sampling Theorem

The sampling of the data should be frequent enough to capture subtle changes such as solar exposure and equipment response cycles. For example, a simple thermostat that calls for heat will initiate a heating equipment and delivery equipment. Each of these will require time before performing nominally. In addition, smarter equipment such as ECM pumps has exercise cycles or heat pumps have self -diagnostic cycles that may affect the heating system in strange ways.

At the Caledon site, the control thermostat operates at a refresh rate of 1/60 hertz and the control valves have open to the fully open state from the fully closed state in 45 seconds. Therefore the data logger was set at a scan rate of once every 10 seconds and the code may be found in the Appendix.

5.5. Sensor Deployment Locations

Thermal environment monitoring equipment was placed within the building at strategic locations within the room. Most sensors pertaining to the thermal comfort of a person were deployed at or close to the centre of mass of where that person would be.

5.6. Equipment Calibration

The thermocouples were calibrated to 0.2°C with an ice bath prior to being deployed in the field study. Room air temperature and relative humidity sensor was verified with a mercury sling hygrometer and the black globe thermister was checked according to the air temperature sensor reading; all readings were found to be within 0.5°C. The Infrared sphere infrared sensors were adjusted manually in 4 degrees of freedom (pitch, yaw, vertical displacement, horizontal displacement) to achieve the desired aspect ratios.

5.7. Deployed Equipment

The equipment deployed for the two field studies are listed in Table 6:

Table 6 Deployed Sensors at Caledon and Richmond Hill

Item No.	Item	#	Manufacturer and Model	Range and Accuracy	Caledon	Richmond Hill
1	Mean Radiant Temperature Globe Temperature Sensor	1	Campbell Scientific - BLACKGLOBE L	-5°C to 95°C ±0.5°C	Yes	Yes
2	Temperature Sensor	19	TSL-30-TT Type T 30 AWG thermocouple	±0.5°C or ±4%	Yes	Yes
3	Air Temperature Sensors (stratification)	5	TSL-30-TT Type T 30 AWG thermocouple	±0.5°C or ±4%	Yes	Yes
4	Omni-directional Low Air Velocity Sensor	1	Sensor Electronic - SensoAnemo 5130SF	0.05 m/s to 0.5 m/s, ±0.02 m/s or ± 1%	Yes	Yes
5	Radiant Temperature Sensor	2	Micro-Epsilon - Optris CT	-40°C to 900°C, ±1°C or ±1%	Yes	Yes
6	Humidity Sensor	1	Vaisala - INTERCAP HMP60	0%RH to 100%RH, ±3%	Yes	Yes
7	Data Logger	1	Campbell Scientific - CR3000		Yes	Yes
8	Outdoor Temperature Sensor	1	HOBO U12	-40°C to 70°C, ±2.5%	Yes	Yes
9	Infra-red Camera	1	FLIR	±1°C or ±1%	Yes	Yes
10	Particulate Sensor	1	OptiNet –Aircuity BAS	0.3-2.5 µm ±25%	Yes	No
11	Carbon Monoxide	1	OptiNet –Aircuity BAS	0-150 PPM ±5%	Yes	No
12	Carbon Dioxide	1	OptiNet –Aircuity BAS	0-2000 PPM ±75ppm below 1500 PPM, ±5% above 1500 PPM	Yes	No
13	Total Volatile Organic Compounds	1	OptiNet –Aircuity BAS	0-125 PPM ±25%	Yes	No

5.8. Experimental Setup

5.8.1. Setup at Caledon

5.8.1.1. *Site Background*

The building is a single family dwelling approximately 500 meter squared located North West of Bolton, Ontario in the municipality of Caledon East at the coordinates of 43°57'4"North -79°52'27" West. The building is one story above grade with a basement semi below grade. The envelope is designed and built with a high degree of air tightness. Glazing is triple glazed argon low E with a RSI value of 1.2 while walls are designed with an RSI value of 6.2.

The residence is thermally maintained by radiant heated and cooled ceiling where hydronic pipes are embedded in sprayed concrete. Warm or cool water is circulated within the embedded pipes to provide heating or cooling. The main heat generation for the system is via a well source heat pump rated at 17.6 kW backed up with a wood gasification boiler rated at 30 kW. Cooling is provided by heat exchange with ground water.

Ventilation is provided to the building via a 100% outdoor air system delivering ventilation air into each occupied space utilizing an energy recovery ventilator. Exhaust air is drawn from each occupied space.

The building is controlled by a building management system which operates the ventilation system by room independent Volatile Organic Compound sensors

and operates the heating and cooling system by room independent air temperature. Each room is independently controlled for heating and cooling output. Hydronic flow is controlled by an Electronic Commuted Motor pump and hydronic supply temperature follows an Outdoor Reset logic. All of which are logged.

The thermal environment the room was monitored is located on a semi recessed basement with a large 2.6 m wide grade to ceiling 2.1 m tall window facing North North East. The room is 6.9 m long by 3.6 m wide where the occupant sleeps in the middle of the room.

The main source of heating is via the radiant ceiling. A minute amount of heat is injected into the room by the ventilation system at 5 L/s supplied and at room air temperature. This creates a very small heat gain into the room and does not create a forced convection over the body; therefore it was not included in the model.

An initial site visit was performed to determine the existing condition of the building and the current state of operation. Several measurements were used to validate the site condition to the original design drawings of the building. This includes checking the ventilation rate, determining the radiant view factor via (Farrell, 1976) and checking the air leakage of the building envelope via a blower door test.

5.8.1.2. *Instrumentation*

The surface temperature of the walls, ceiling and floor were measured with an infrared camera to determine the placement of surface mounted temperature sensors and is presented in Figure 9. The mechanical system has been shut off for a day with an outdoor temperature of approximately -13°C at the time of the recording. The lowest surface temperature was recorded to be 15.7°C at the junction of the exterior wall. The floor labeled D and interior walls labeled C and F were found to be fairly uniform and so minimal sensor was deploy to represent these surface temperatures as shown in Figure 14, Figure 15 and Figure 16.

The left panel of the window was found to be at a slightly higher temperature than right panel due to being more directly exposed to the radiant heating panel. The small wall directly below the window was found to be fairly uniform and therefore a single sensor was deployed on each to measure its representative temperature.

The ceiling contained both the heating panel and free ceiling area. The free ceiling area was progressively warmer when closer to the radiant panel most likely by surface conduction. Three sensors were deployed with a closer sensor element spacing when nearer the radiant panel. The radiant panel showed significant temperature changes across it and therefore 6 sensors were

dispersed to determine the average radiant panel temperature as per Figure 10, Figure 12 and Figure 13.

Similarly, the exterior side wall labeled A was found to be progressively warmer when it neared the radiant panel. The temperature was found to be fairly constant until it reaches within 30" or 762 mm of the radiant panel as shown in Figure 9. Thus a single sensor was deployed to represent the lower portion of the wall and three equally space sensor were deployed to represent the wall area close to the radiant panel.

Figure 9 Caledon Room Surfaces Condition and Surface Identification

Sensors were placed at the locations as indicated in Figure 10 Sensors Location in Caledon Test Room; adjustments were made due to site conditions. The surface temperature sensors were placed according to areas representative of the mean panel temperature determined by an infra-red camera.

An outdoor temperature and relative humidity sensor was located in a shaded region of the premises. The black globe, hot sphere anemometer, infrared sphere and air temperature stratification array was deployed at the centre of the room and at the height of the centre of mass of a sleeping as per Figure 10.

Figure 10 Sensors Location in Caledon Test Room

Figure 11 Caledon Window Wall Sensor Locations

Sensor 18 (Figure 11) represents the temperature of the left plane window area while sensor 19 is used to represent the right plane window area and sensor 20 was used to represent the remaining wall area.

Figure 12 Caledon Ceiling Sensor Locations.

Sensors 01, 02, 03, 04, 08 (Figure 12) and 17 (Figure 13) temperature reading were averaged and used to represent the temperature of the radiant panel. Sensor 05, 06 and 07 were area weighted with each sensor representing the surface temperature from the boundary or midpoint between the previous sensor and itself to the boundary or midpoint between the next sensor and itself. The area weighted average was used to represent the temperature of the free ceiling.

Figure 13 Caledon Exterior Side Wall Sensor Locations

Sensor 14, 15 and 16 (Figure 13) were area weighted with each sensor representing the surface temperature from the boundary or midpoint between the previous sensor and itself to the boundary or midpoint between the next sensor and itself. The area weighted average was used to represent the temperature of the wall.

Figure 14 Caledon Interior Side Wall Sensor Locations

Figure 15 Caledon Floor Sensor Locations

Figure 16 Caledon Back Wall Sensor Locations

5.8.1.3. Determination of View Factor for Radiant Heat Transfer

The view factor is determined by Nusselt method (Nusselt, 1928) with a photograph of a polished stainless steel sphere as per Figure 17 shown below. The focal point of the photo is approximately 558.8 mm (22") above the sphere and the sphere is 101.6 mm (4") nominal diameter.

The conical view angle of the photographic lens will have a reduction of 0.3 % of the actual diameter of the sphere. This error is discarded. The view factor is

calculated and tabulated in Table 7 Caledon Room View Factor Calculation below.

Figure 17 Caledon Photo of Stainless Steel Sphere with Surfaces Denoted A B C D E F G and G'

Table 7 Caledon Room View Factor Calculation

Area of Circle (Unit Area)	4232.3052			
Surface	Name	Unit Area	Surface Area m ²	View Factor F
A	Interior Side Wall	787.0391	16.548	0.185959
B	Window Wall	235.2227	4.7252	0.059830
C	Window	131.3297	3.9677	0.0310302
D	Exterior Side Wall	628.4138	16.548	0.1484802
E	Back Wall	166.5718	8.693	0.0393572
F	Floor	1206.058	24.980	0.284965
G	Unheated Ceiling	710.5298	15.462	0.1678824
G'	Radiant Panel	349.1487	9.518	0.082496

5.8.2. Setup at Richmond Hill

5.8.2.1. Site Background

The building is a two-storey single family dwelling with a basement below grade located in Richmond Hill, Ontario. The building envelope is designed and built with typical construction in the 80s. Glazing is double glazed clear air while walls are believed to be as per building code specification of an RSI value of 3.52. The building is thermally maintained by a forced air heating system with a design infiltration rate of 1.5 ACH at 50 Pa. The main source of heating is by an atmospheric furnace. Control is by centrally located thermostat with temperature differential dead band.

The room in which the field study was performed is located on the second floor with three casement table height to ceiling height windows facing South South West. The windows are 914 mm wide and 1.524 m tall. The room is 4.9 m long

by 3.66 m wide where the occupant sleeps in the North West corner of the room, see Figure 18.

An initial site walkthrough was performed to determine the existing condition of the room and the current state of operation.

5.8.2.2. *Instrumentation*

The surface temperature of the walls, ceiling and floor were measured with an infrared camera to determine the placement of surface mounted temperature sensors. The outdoor temperature was approximately -4°C at the time of the recording.

The surface temperature distribution over the exterior walls varied with the contour of the stud wall clearly visible. The floor labeled D and interior walls labeled A and F on Figure 18 were found to be fairly uniform and so minimal sensor was deployed to represent these surface temperatures as shown in Figure 20, Figure 21 and Figure 24.

Significant infiltration was detected around windows and the exterior wall junction with the ceiling. As no discernable temperature gradient exist on the interior surfaces, the number of deployed surface temperature sensors was increased and dispersed to achieve a better averaging effect.

With a southern exposure, extra care was exhibited in protecting each surface temperature sensor from direct solar exposure via the application of foil tape

and the spacing of the sensors so that only one of the sensors out of a group representing a surface will be exposed directly at a time.

The electronic ballast of the compact fluorescent lighting was found to induce a current in the thermocouples and therefore affect the accuracy of the sensors near the lighting fixture when turned on. Therefore, the lighting fixtures were disconnected for the entire duration of the field testing.

Sensors were placed at the locations as indicated on Figure 19 to Figure 24; adjustments were made due to site conditions. The surface temperature sensors were placed according to areas representative of the mean panel temperature determined by an infra-red camera.

An outdoor temperature and relative humidity sensor was located in a shaded region of the premises. The black globe, hot sphere anemometer, infrared sphere and air temperature stratification array was deployed at the centre of the room and at the height of the centre of mass of a sleeping person.

Figure 18 Richmond Hill SSW Room Windows and Wall Layout

Figure 19 Richmond Hill Window Wall Sensor Locations

Each of the windows has a temperature sensor deployed. The small wall directly below the window was found to be fairly uniform and therefore two sensors were deployed on each to measure its representative temperature as shown in Figure 19. The ceiling and floor have relatively uniform temperature; therefore, three sensors were deployed on each as shown in Figure 21 and Figure 22.

Figure 20 Richmond Interior Side Wall Sensor Locations

Figure 21 Richmond Hill Floor Sensor Locations

Figure 22 Richmond Hill Ceiling Sensor Locations

Figure 23 Richmond Hill Exterior Side Wall Sensor Locations

The sensors located on the exterior side wall were dispersed to achieve a more reliable averaging effect. Note the bottom right corner is obstructed by furniture.

Figure 24 Richmond Hill Back Wall Sensor Locations

The sensors located on the back wall were dispersed to achieve a more reliable averaging effect. Note the bottom left corner is obstructed by furniture.

5.8.2.3. Determination of View Factor for Radiant Heat Transfer

The view factor is determined by Nusselt method (Nusselt, 1928) with a photograph of a polished stainless steel sphere as per Figure 25 shown below. The focal point of the photo is approximately 558.8 mm (22") above the sphere and the sphere is 101.6 mm (4") nominal diameter.

The conical view angle of the photographic lens will have a reduction of 0.3 % of the actual diameter of the sphere. This error is discarded. The view factor is calculated and tabulated in Table 8 below.

Figure 25 Richmond Hill Photo of Stainless Steel Sphere with Surfaces denoted A B C1 C2 C3 D E and F

Table 8 Richmond Hill NNW Room View Factor Calculation

Area of Circle (Unit Area)	8776.2934			
Surface	Name	Unit Area	Surface Area m ²	View Factor F
D	Exterior Front Wall	1604.583	6.5551	0.182775
B	Exterior Side Wall	996.6059	9.4758	0.113522
A	Interior Back Wall	1289.582	11.0551	0.146894
E	Interior Side Wall	515.2421	9.4758	0.05869
G	Ceiling	1851.213	15.6	0.210868
F	Floor	2124.66	15.6	0.242016
C1	Left Window	180.2901	1.5	0.020537
C2	Middle Window	157.9221	1.5	0.017989
C3	Right Window	58.89531	1.5	0.006709

5.8.3. Check Up and Retrieval Site Visit

Checkup on the site monitoring was performed every week on the equipment setup such as disconnected sensors or delaminated temperature sensors from the walls. The quality of the recorded data were also monitored and errors reduced by improving signal quality.

Data from disconnected sensors or delaminated sensors were excluded from the calculations until the sensors were reconnected or reattached. Noise from electromagnetic induction into the thermocouples were reduced manually while solar inducted spikes in temperature sensor readings were reduced with the application of foil tape and additional sensors.

5.8.4. Equipment Programming and Data Analysis

The deployed thermocouples in the surface temperature array and the air temperature stratification array were set to scan every 10 seconds. In the post processing every 6th data point was utilized to represent each minute for the analysis. The excess data were used to backfill sequences of the data chain that was compromised by selecting the preceding value or averaging of the sequence.

The relative humidity probe, thermistor used in the black globe, and the infrared temperature sensors were set to scan with averaging of two repetitions every 30 seconds. In the Post processing every 2nd data point was used to represent each minute for the analysis.

Similarly, the outdoor temperature and humidity sensor was set to scan every 30 seconds and post processed to each minute for the analysis.

As described above, disconnected sensors or malfunctioning sensors were excluded from the analysis until that data collection point was repaired.

5.8.5. Schedule

The field study was conducted during the heating season. The field study at the Caledon site was performed between the periods of February 1st to February 20th while the field study at the Richmond Hill site was performed between the periods of April 17th to April 30th.

6.0 Experimental Results

6.1. Analysis and Discussion of Data

6.1.1. Sleep Thermal Comfort

The operative temperature as determined by the surface sensor method is presented in Figure 28. As anticipated earlier, it was found that the air temperature is less than the operative temperature, which in turn is less than the mean radiant temperature.

As such, if a radiant system is operated to building code (Province of Ontario, 2006) requirements to maintain indoor air temperature at 22°C, this will create an indoor operative temperature of 23°C and thus will cause discomfort by overheating to a sleeping person. This suggests that the operation of the HVAC system may be reduced during sleep thus providing an energy savings as will be explored in chapter 7.

6.1.2. Thermal Comfort

The thermal environment of the Caledon NNW room was reviewed according to ASHRAE thermal comfort criteria (ASHRAE, 2010).

The mean air velocity was determined to be 0.021 m/s and is less than 0.8 m/s therefore should not have any influence on thermal comfort. Relative humidity was found to be in between 22% to 29%, both are measured at 0.6m above the floor. Air temperature stratification was less than 3°C difference from floor to

ceiling and less than 2°C difference from 1.8m, the height of an average American male, to the floor.

The air temperature of the room was found to follow the mean radiant temperature without significant lag within 10 minutes by approximately 2°C less. Hence, the operative temperature inside the room is approximately 1°C above the air temperature and 1°C below the mean radiant temperature.

6.1.2.1. *Caledon NNW Room*

Figure 26 Caledon NNW Room Temperature of Various Surfaces

Figure 26 shows the average surface temperature as determined by area weighted average of thermocouple readings. The surface names correspond to the names identified in Figure 9 and the location and number of sensors are indicated in Figure 11 to Figure 16.

As the Back Wall faces the Window Wall and Window, the Interior Side Wall faces the Exterior Side Wall, and the Floor faces Ceiling and Radiant Panel; the radiant asymmetry is approximately: $\sim 3^{\circ}\text{C}$ for the window wall to the interior wall, $\sim 0.5^{\circ}\text{C}$ between the side walls and $\sim 6^{\circ}\text{C}$ between the ceiling and the floor.

Except for the window and the radiant panel, the remaining surface temperatures are fairly uniform, which implies a fairly well sealed and insulated envelope. Furthermore, the temperature ramps and drift over a day is fairly minimal in the order of around 2°C per day, much less than the 1.1°C per 15 minutes allowed by ASHRAE 55.

Radiant asymmetry for the horizontal axis are well within thermal comfort requirements (ASHRAE, 2010) while worst case vertical asymmetry is about 6°C which is slightly over the prescribed limit of 5°C and suggests that the room is thermally comfortable.

Figure 27 Caledon NNW Room Air Temperature Stratification.

Figure 27 shows the air temperature stratification inside the room at Caledon. The temperature as shown indicates a developed air temperature profile that responds directly to the heating input.

Using ASHRAE 55's (ASHRAE, 2010) definition of air temperature stratification between the head and foot of a 1.8m tall person, the air temperature difference is about 2°C and this is well within the comfort limit of 3°C.

Figure 28 Caledon NNW Room Various Temperature Comparison.

Figure 28 shows the operative temperature derived by the air temperature and the mean radiant temperature by the surface temperature method. The outdoor air temperature, average radiant panel temperature, supply water temperature, indoor air temperature and area weighted average surface temperature are also included in Figure 28 for comparison and suggest that the operative temperature is pretty much independent of the outdoor temperatures.

The supply water temperature was governed by a heat pump and a wood boiler, therefore the supply water temperature shows two distinct phases. The sinusoidal temperature profile such as the one between Feb 05th and Feb 07th shows the wood boiler firing and its discharge cycle. The relatively consistent temperature profile with periodical drop offs such as those between Feb 08th and Feb 11th is the operation of the heat pump.

The first eight days' data are shown in Figure 29 below to illustrate the anti-aliasing theorem.

Figure 29 First Eight Days' Data of Figure 28

The data shown in Figure 29 have been down sampled to allow it to be plotted clearly. The down sample is to 1 reading per hour and shows a pretty consistent supply water temperature output when operating the heat pump between the 2nd and the 6th.

However, if the same data are plotted at a sampling rate of 1 reading per 30 minute as shown in Figure 30, the supply water temperature shows a pattern such as a consistent temperature drop at around 6 pm in the period which the

heat pump is running (Feb 2 – Feb 4).

Figure 30 Same Data as Figure 29 Down Sampled to 30 Minutes Intervals

There seems to be a periodic temperature drop off at around the same time each day. However, the pattern does not repeat itself on the 8th. Therefore it suggests that down sampling rate was insufficient to reveal the actual operation of the system.

The sampling rate was increased to a reading every 5 minutes and is plotted in Figure 31 below.

Figure 31 Same Data as Figure 29 Down Sampled at 5 Minutes Intervals

At a higher sampling rate, a distinct pattern of the periodic supply water temperature drop off is seen. The periodical drop off actually represents the heat pump self check cycle every ~4 Hours as confirmed with the manufacturer. The sampling rate as per the anti aliasing theorem was successful in allowing this cycle to be identified.

The operative temperature remains between 19°C to 22°C even when the outdoor air temperature varies from -22°C to 10°C. The operative temperature of the room during the operation with only the heat pump was within ~0.5°C

each day. This suggests that the mechanical system is sufficient in providing the necessary output to match the load determined by the building envelope.

The maximum rate of operative temperature change is approximately $1^{\circ}\text{C}/6$ hours, which is less than ASHRAE 55 (ASHRAE, 2004) (ASHRAE, 2010) limit of $3.3^{\circ}\text{C}/4$ hours. In short, the interior space of the room would be considered comfortable.

Figure 32 Relative Humidity and Contaminants

Figure 32 shows the relative humidity and contaminants at the Caledon site.

Although contaminants are beyond the scope of this research, it is indicative of the separate ventilation aspect of radiant heated buildings.

Ventilation air is provided at 5 L/s directly into the room and 5 L/s are extracted from the room. The particulate spike with 250 particle per cubic meter on the 1st of February corresponds to the two researchers (one male and one female) setting up the equipment with light activity. While the 150 particle per cubic meter spike on the 5th of February corresponds to one male researcher retrieving data and the 100 particle per meter cube spike on the 6th and the 7th of February corresponds to the to the female owner doing some light cleaning.

The CO₂ level is consistent at roughly 500 ppm for the duration of this study. This is much less than the ASHRAE (ASHRAE, 1999) and OSHA (Carbon Dioxide) standard of a ceiling tolerance of 1000 ppm and 5000 ppm respectively.

The indoor relative humidity is in the low 20% range as the humidifier was not operated. All the humidity within the building was generated by the people inside and the activities that they have completed. Moisture was also recovered from the exhaust air via the Energy Recovery Ventilator.

6.1.2.2. *Richmond Hill SSW Room*

The field data measurements were repeated at a forced air heated suburban

house located in Richmond Hill, Ontario. The house was constructed during the 1980s and is typical of that era. The sensors were deployed in a SSW facing room with three casement windows.

Air Temperature Stratification - Richmond Hill SSW Room

Figure 33 Richmond Hill SSW Room Air Stratification Comparison.

The overall air temperature stratification shows that the vertical stratification is within 1°C, however significant ramp changes in temperature exist such as the 5°C rise over 6 hours on the 23rd; 4°C rise over 3 hours at 6 pm on the 20th and 3°C rise over 2 hours on the 28th.

The operation of the mechanical system can also be deduced as on the 20th, the furnace can operate continuously to meet the heating load when the outdoor condition is -3°C with an air temperature rise of 4°C. While on the 25th the furnace was cycled 4 times to meet the heating load when the outdoor condition is 6°C, each with a temperature rises of around 1.5°C.

The air temperature stratification for a selected period is shown in Figure 34.

Figure 34 Richmond Hill SSW Room Air Stratification Comparison Over Selected Dates

Figure 34 shows clearly the chaotic air temperature regime and stratification inside the room with almost instantaneous step changes. The step increases are consistently at 6 pm as shown on the 18th, 19th, 20th and the 21st as the outdoor temperature begins to drop therefore suggesting the operation of the force air heating system.

The floor surface temperature is also shown and is indicative of the solar heat gain of the room. The indoor air temperatures correspond well with the floor temperature but the influence of the outdoor air temperature can also clearly be seen.

At instances where the floor temperature curve does not follow the outdoor air temperature curve, the indoor air temperature curve shows distinctive minor peaks corresponding to the outdoor temperature. This suggests that there is significant infiltration.

Wall Surface Temperature - Richmond Hill SSW Room

Figure 35 Richmond Hill SSW Room Surface Temperature Comparison.

The room surface temperatures are fairly uniform, on the 23rd of April; the surface temperatures are raised steadily by 5°C over 4 hours to an average room surface temperature of 24°C.

The window temperature are especially high, this may be due to the direct solar exposure onto the window thermocouples which were not direct solar radiation protected with foil tape as the wall sensors were.

This may skew the subsequent mean radiant temperature and therefore the operative temperature to become higher. According to Table 8, the combined view factor of the windows attribute to 0.045 or 4.5%. This suggest that on a peak day like April 13 with recorded operative temperature of 24.9°C and an average wall temperature of 24.4°C; the windows contributed a maximum of 0.5°C to the operative temperature even when reached a temperature of 39°C. This suggest that the solar impact on the window sensor is minimal to the overall mean radiant temperature determination and even less so to the operative temperature determination.

The large change in surface and air temperature suggests that the operative temperature would also have large changes and is shown in Figure 36.

Figure 36 Richmond Hill Temperature Comparison

The operative temperature follows closely to the pattern of the air temperature. The maximum ramp of operative temperature is approximately $1^{\circ}\text{C}/15$ minutes for a maximum of 4°C change. It is close to the acceptability limit of ASHRAE 55's tolerance of $1.1^{\circ}\text{C}/15$ minutes and exceeds the maximum drift of $3.3^{\circ}\text{C}/4$ hour. Therefore the Richmond Hill building may be considered thermally uncomfortable according to ASHRAE standard 55.

6.1.3. Forced Air and Radiant Ceiling Heating Thermal Comfort Comparison

Although the two buildings are vastly different in construction and build quality, some merits are still present in a comparison of the thermal comfort

conditions created in each case. The ramp and drift of each field test location is calculated graphically and is presented in Table 9 below. The heating system response time could not be calculated as the Caledon site was being operated continuously while the Richmond Hill site could not be determined as the collected data shows a lot of interference thought to be from leakage and solar radiation. A comparison of how long it took from the peak heating delivery by either the forced air or the radiant panel to the maximum resultant operative temperature was graphically derived and shown in Table 9 below.

Table 9 Comparison of Forced Air Heating and Radiant Ceiling Heating

Criteria	Richmond Hill Forced Air	Caledon Radiant Ceiling
Vertical Radiant Temperature Difference	3°C	5°C
Horizontal Radiant Temperature Difference	15°C	4°C
Air Temperature Stratification	1°C	2°C
Mean Air Velocity	0.4 m/s	0.021 m/s
Max Ramp and Drift	~1°C/15minutes	~0.05°C/15minutes
Peak Heating Delivery to Peak Thermal Environment	~45 Minutes	~15 Minutes

The results are consistent with existing research (Haddad & Purdy, 2007) in that radiant heating systems provides better thermal environments and energy efficiency in buildings with better envelopes. This is generally due to the lower temperature output at the primary heat generation site, therefore, allowing higher coefficient of performance or efficiency.

The heat distribution is generally more efficient due to the energy density water can hold in comparison to air. While delivery by radiant means is

omnidirectional and is instantaneous when compared to the slow non-uniform distribution of buoyancy and diffusion of forced or free convective heating.

While an analysis of the overall heating efficiency of these two heating systems cannot be completed due to the insufficient data, the radiant heating system is expected to heat more efficiently and effectively than forced air as per other radiant heating system studies (Murakami, Kato, & Kim, 2001)

6.1.4. Effectiveness of the Infrared Sphere at Determining Mean Radiant Temperature

6.1.4.1. Comparison of Mean Radiant Temperature from Infrared Sphere, Black Globe and Surface Sensor Method

Three methods of field determination of the Mean Radiant Temperature were employed in two different rooms in two separate buildings. One of the rooms is NNW exposed while the other one is SSW exposed. The Surface Sensor method and the Infrared Sphere method produce nearly identical mean radiant temperature at both locations while the Black Globe method produced a higher Mean Radiant Temperature at the SSW facing room during non clouded period of day time. The black globe overestimation of mean radiant temperature during solar exposure is consistent with findings from previous studies (Jones & Chapman, 1994). (Jones & Chapman, 1994) In both buildings the Mean Radiant Temperature is indicative of the Average Surface Area Weighted Temperature that is normally used for HVAC control.

Figure 37 Caledon NNW Room Comparison of Mean Radiant Temperature by Surface Sensor Method, Black Globe method and Infrared Sphere method.

The AUST (Average Weighted Surface Temperature) with all surfaces and only the recipient surfaces excluding the radiant panel are shown in Figure 37. The mean radiant temperature developed by the infrared sphere method and the surface temperature method are very similar with a discrepancy of around 0.25°C , which is well within the accuracy of the equipment and the geometry and calculation errors. The black globe method underestimates the mean radiant temperature compared to the others by around 1°C . Note that the Caledon site faces North North West and therefore have very little solar exposure especially during winter and does not show the characteristic overestimation of mean radiant temperature during the day. This discrepancy may be attributed to the thermistor used inside the black globe in comparison to the thermocouples readings for the surface measurements.

These results were also reproduced at the Richmond Hill site as shown in Figure 38 below.

Figure 38 Richmond Hill SSW Mean Radiant Temperatures Comparison

There is a more pronounced difference between the different methods of determining mean radiant temperature. Under direct sunlight, the black globe method estimate is significantly higher than the surface temperature method by 4°C as shown at noon on the 23rd.

This is consistent with previous researchers who found that the black globe overestimates mean radiant temperature under direct sunlight exposure (ISO, 2002). The infrared sphere method overestimated the mean radiant temperature by approximately 1°C under direct sunlight as shown at noon on

the 23rd and suggests that the infrared sphere method is more resilient to sun exposure than the black globe.

Figure 39 Richmond Hill SSW Room Mean Radiant Temperature Comparison of Selected Dates.

Figure 37 and Figure 39 shows very little discrepancy between the mean radiant temperature developed by the infrared sphere method in comparison to the surface sensor method and the black globe method. This suggests that the Infrared Sphere method may be a possible substitute for the Black Globe method and Surface Sensor method in the determination of Mean Radiant Temperature and Average Surface Area Weighted Temperature under certain circumstances.

6.1.4.2. *Revisit the Definition of Mean Radiant Temperature*

The mean radiant temperature that is defined by Fanger and as worded in (ASHRAE, 2009) is: “the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual nonuniform enclosure.” The definition of radiant heat transfer can be and have been viewed as the radiant heat transfer via long wave radiation as shown in the proposed calculation method of Mean Radiant Temperature by surface temperatures and view factors.

This means that short wave radiation generally from solar sources is omitted, which affects the accuracy of the apparatus in determining Mean Radiant Temperature. In essence, a Mean Radiant Temperature that is determined by sufficiently distributed surface temperature sensor grid will be different than a Mean Radiant Temperature determined by the Black Globe method. The difference is caused by the working mechanism of each method. The black globe works by finding the neutral temperature point at the recipient’s location where the radiant heat flux gain/loss at the globe is matched by the convective heat loss/gain of the globe, while the surface sensor method works by measuring the long wave radiation heat flux going towards the recipient. Direct sunlight on several surface temperature sensors will elevate only a portion of the heat flux going towards the recipient while direct sunlight on the black

globe will significantly elevate the heat flux entering at the hemispherical direction of the solar incident.

Likewise, the Infrared Sphere method like the surface sensor method measures the reflected long wave infrared heat flux off the polished sphere. Since the infrared sensors are only sensitive to long wave radiation and that the emissivity of polished stainless steel is higher for short wave radiation than long wave; the Infrared Sphere method will not represent the effect of direct sunlight incidence well.

The Mean Radiant Temperature determined by the Black Globe method is within reason of the other two methods during night time while increases significantly when under exposure from direct sunlight.

The meteorological data as obtained from Environmental Canada in

Table 11 suggests April 20th, 25th and 26th is clouded over with precipitation with no direct solar gains and Figure 38 shows the mean radiant temperature derived by the Black Globe method to be similar to the other methods without the characteristic spike at noon. This is similar to (ISO, 2002) on the performance of the black globe which finds that the black globes overestimates mean radiant temperature when under direct sunlight.

Table 10 Meteorological Data for Caledon Test Period (T in table means Trace amount detected)

Date	Max Temp	Min Temp	Mean Temp	Total Rain	Total Snow	Total Precipitation	Snow on Ground	Direction of Max Gust	Speed of Max Gust
	Deg C	Deg C	Deg C	mm	cm	mm	cm	Deg	km/h
1-Feb-11	-7.1	-11.8	-9.5	0	1.6	1	7	7	39
2-Feb-11	-8.4	-14.9	-11.7	0	12.8	9.4	15	6	48
3-Feb-11	-3.8	-17.1	-10.5	0	0	0	21	24	50
4-Feb-11	-4.6	-10.2	-7.4	0	0	0	20	24	46
5-Feb-11	-0.9	-9.1	-5	0	11.6	9.8	20	30	59
6-Feb-11	0.6	-3.2	-1.3	0	2.6	1.4	17		<31
7-Feb-11	0.4	-10.6	-5.1	0	1.2	0.7	17	35	54
8-Feb-11	-9.6	-16	-12.8	0	T	T	16	34	48
9-Feb-11	-6.3	-13.7	-10	0	0	0	17	27	54
10-Feb-11	-9.2	-15.3	-12.3	0	0	0	17	28	46
11-Feb-11	-4.4	-18	-11.2	0	T	T	16	23	33
12-Feb-11	-1.9	-7	-4.5	0	0.2	T	16	30	56
13-Feb-11	6.8	-6.1	0.4	0	0.2	0.2	16	24	52
14-Feb-11	6.8	-10.3	-1.8	T	0	T	12	33	78
15-Feb-11	-3.6	-13.6	-8.6	0	0	0	8		<31
16-Feb-11	5.5	-6.3	-0.4	0	T	T	5		<31
17-Feb-11	11.3	-1	5.2	1.2	0	1.2	3	22	33
18-Feb-11	10.8	-1	4.9	0	T	T	T	27	96
19-Feb-11	-0.9	-6.7	-3.8	0	T	T	T	30	83
20-Feb-11	-3.3	-10.1	-6.7	0	4.6	4.2	T		<31

Table 11 Meteorological Data for Richmond Hill Test Period (T in table means Trace amount detected)

Date	Max Temp	Min Temp	Mean Temp	Total Rain	Total Snow	Total Precipitation	Snow on Ground	Direction of Max Gust	Speed of Max Gust
	Deg C	Deg C	Deg C	mm	cm	mm	cm	Deg	km/h
17-Apr-11	5.8	-0.2	2.8	0.6	0.4	1	T	28	69
18-Apr-11	4.3	-0.3	2	T	0	T		30	46
19-Apr-11	6.2	-1.7	2.3	4.2	T	4.2		8	56
20-Apr-11	7.7	2.4	5.1	13	0	13		27	61
21-Apr-11	8.4	-0.8	3.8	0	0	0		32	52
22-Apr-11	8.6	-0.2	4.2	1.2	0	1.2		13	44
23-Apr-11	22	4.1	13.1	6.6	0	6.6		27	59
24-Apr-11	12.6	4	8.3	0	0	0			<31
25-Apr-11	11.8	3.4	7.6	6.2	0	6.2			<31
26-Apr-11	11.9	8.3	10.1	12.6	0	12.6		9	61
27-Apr-11	22.6	7.2	14.9	4.6	0	4.6		22	69
28-Apr-11	16	5.9	11	4.6	0	4.6		25	95
29-Apr-11	9.4	4.3	6.9	T	0	T		28	37
30-Apr-11	13.8	2.4	8.1	0	0	0			<31

This proposes the question, should Mean Radiant Temperature account for direct solar radiative heat transfer as its original definition will allow? Or should direct solar radiation be removed from its definition specifically. Solar radiation is not something a HVAC system could control in the traditional sense, but can explicitly be controlled by the occupant by drawing a curtain. While HVAC system can influence surface temperatures, air temperature and velocity, these items cannot be controlled by the occupants directly.

6.2. Summary

In summary, the Infrared Sphere method in determining mean radiant temperature by this research shows an agreement with the other methods of

determining mean radiant temperature for the two field test locations. Some discrepancies between the Black Globe method and the infrared sphere method is recorded which is also documented by previous research (ISO, 2002). Further research should be completed to validate the Infrared Sphere method prior to its adoption.

This research has also illustrated the importance of the definition of mean radiant temperature to the methods used for its determination. An in-depth review of its intended application is recommended.

7.0 Potential Energy Reduction Due to Nighttime Set Point Reduction

7.1. Introduction

The collected data from the Caledon residence allows the deduction of potential energy savings via the analysis of the installed mechanical system. Assuming the equipment manufacturer's specification data are accurate and reading the supply water temperature off Figure 28 to achieve the desired Operative Temperature, the total reduction of energy consumption and the decreased power consumption due to the increase of coefficient of performance can be determined.

7.1.1. Datum Data and Assumptions

The system is idealized for heating only as per schematic below:

Figure 40 Simplified Heat Flow Schematic of Mechanical System, (Detailed Mechanical Schematic May be Found in the Appendix)

The assumptions and data used for the calculations are as follows:

1. The heat source for the heat pump is ground water from a well and has been measured previously to be at around 5°C year round
2. The sleep cycle is considered to last 8 hours
3. Thermal mass storage effects are considered negligible
4. The heat exchanger loss and other losses are assumed to constitute a 10°C temperature loss

5. The heat pump is a Climatemaster Genesis GSW 060 5 ton heat pump. At the installed specification, the unit is specified to provide the output as shown below:

Table 12 Heat Pump Performance

Entering Source Temperature		40°F (4.4°C)
Total Heating Produced	Leaving Sink Temperature	Coefficient of Performance
45.1 MBH	126.0°F	2.66
46.8 MBH	106.2°F	3.48
48.1 MBH	86.4°F	4.56
49.1 MBH	66.5°F	5.87

6. The design day outdoor condition is used and assumed to be consistent for 24 hours of the analysis. Existing engineering documents states heating load to be 12.9 kW by ENERPASS
7. The heat pump heat sink pump is a Grundfos UPS 32-40 BF pump. 100 W
8. The radiant circulator pump is a Wilo 1.25X3-35 pump. 320 W
9. The heat pump heat source is provided by the well pump which is a Berkley MGS 15 GPM staged pump with P43B0005A2 motor. ½ hp shaft power, 745 W

7.1.2. Calculation Procedure

7.1.2.1. Power Reduction

The power consumption of the heating system to meet the heating load can be broken down into two parts: heat generation and heat delivery. The heat

generation portion is the power consumed to generate the heat to heat the building and the heat delivery portion is the power consumed to deliver the heat to the space such as by the motors that drives the pumps.

The heat generation portion is affected by the heat generation efficiency and in this case the coefficient of performance of the heat pump while the delivery portion of the power consumption does not change.

The power consumed can be expressed as follows:

$$P_{Input} = \frac{P_{Out}}{COP} + P_{source} + P_{Sink} + P_{Delivery} \quad (30)$$

Where;

$$P_{Input} = \text{Power Consumed} - kW$$

$$P_{Out} = \text{Heat Pump Heating Output} - kW$$

$$COP = \text{Coefficient of Performance} - \text{Dimensionless}$$

$$P_{source} = \text{Heat Source Pump Input Power} - kW$$

$$P_{Sink} = \text{Heat Sink Pump Input Power} - kW$$

$$P_{Delivery} = \text{Heat Delivery Pump Input Power} - kW$$

Note that both Heating Output and COP are determined by reading Figure 28 with the desired operative temperature and supply water temperature. With

the water temperature, Heating Output and COP can be determined by interpolating Table 12.

Figure 41 Panel Supply Temperature vs. Operative Temperature

Note the sudden drop of supply temperature by around 2°C is due to the self-check cycle of the heat pump. The data have been extrapolated to 15°C to estimate the potential energy savings.

7.1.2.2. Total Energy Consumption over a Day

The heat generation at the heat pumps exceeds the heating load of the building. This would mean that the heat pump and associated source and sink pumps will cycle on and off. The peak heating load is determined by reading off Figure 28 when the heat pump is operating in equilibrium to provide heating

(February 3rd). Since the heat pump is operating in equilibrium, the heating output of the heat pump will be the heating load of the building. This is used to verify the ENERPASS calculated heating load as discussed in Section 7.1.1. The linearized building heat transfer coefficient is also determined with the peak heating load:

$$U_{Building} = \frac{P_{Load}}{(T_o - T_e)} \quad (31)$$

Where;

$$U_{Building} = \text{Heat Transfer Coefficient of Building} - kW/^{\circ}C$$

$$P_{Load} = \text{Heating Load of Building} - kW$$

$$T_o = \text{Indoor Operative Temperature} - ^{\circ}C$$

$$T_e = \text{Exterior Air Temperature} - ^{\circ}C$$

The building heat transfer coefficient is important to determine the reduction of heating load with the reduction of indoor temperature. Assuming no startup and other loss, the total energy consumed can be calculated by:

$$Q_{Day} = \left(\left[\frac{P_{Load(day)}}{P_{Out(day)}} \right] t_{day} \right) \times \left(\frac{P_{Out(day)}}{COP} + P_{source} + P_{Sink} \right) + \left(\left[\frac{P_{Load(night)}}{P_{Out(night)}} \right] t_{night} \right) \times \left(\frac{P_{Out(night)}}{COP} + P_{source} + P_{Sink} \right) + P_{Delivery} \times (t_{day} + t_{night}) \quad (32)$$

Where;

$$Q_{Day} = \text{Total Heating Input Energy for a Day} - kJ$$

$$t_{day} = \text{Daytime Hours in Seconds} - 57600 \text{ Seconds (16 Hours)}$$

$t_{night} = \text{Nighttime Hours in Seconds} - 28800 \text{ Seconds (8 Hours)}$

$P_{Load(day)} = \text{Daytime Heating Load} - kW$

$P_{Load(night)} = \text{Nighttime Heating Load} - kW$

$P_{Out(day)} = \text{Heat Pump Daytime Output} - kW$

$P_{Out(night)} = \text{Heat Pump Nighttime Output} - kW$

7.1.2.3. Results and Discussion

Several cases of night time operative temperature reduction were calculated and are presented in Table 13.

The normalized heating load “Day Case” was found to be 12.74 kW, which is similar to engineering heating load derived from ENERPASS as of 12.9 kW (Section 7.1.1).

A reduction to an operative temperature of 15°C during the night for sleep as suggested in Section 3.6 will result in 10.12% of energy reduction and reduce the runtime of the heat pump by 14.49% while increasing the performance of the heat pump by 27.93%.

Table 13 Calculated heating load and energy reduction for night time setback from 22°C down to 15°C.

Case	Operative Temp (C)	COP	Heating Load kW	Heat Produced (kW)	Heat Pump Power Input kW	System Power Input kW	Qday kWh	Temp Reduction With Time Factor %	Runtime %	Energy Reduction %	COP Increase %
Datum	20.50	3.55	13.71	13.71	3.86	5.02	118.3		100.00%		
Day Case	22.00	3.38	12.74	13.62	4.03	5.20	117.2	0.00%	93.54%	0.00%	0.00%
Night 21C	21.00	3.45	12.43	13.66	3.96	5.12	115.7	0.82%	90.94%	1.35%	2.29%
Night 20C	20.00	3.56	12.11	13.71	3.85	5.02	113.9	1.63%	88.37%	2.82%	5.43%
Night 19C	19.00	3.69	11.80	13.75	3.73	4.90	112.1	2.45%	85.81%	4.35%	9.19%
Night 18C	18.00	3.82	11.49	13.80	3.61	4.77	110.4	3.27%	83.24%	5.85%	13.32%
Night 17C	17.00	3.98	11.18	13.85	3.48	4.65	108.7	4.08%	80.67%	7.32%	17.82%
Night 16C	16.00	4.14	10.86	13.91	3.36	4.52	107.0	4.90%	78.09%	8.74%	22.69%
Night 15C	15.00	4.32	10.55	13.97	3.24	4.40	105.4	5.72%	75.51%	10.12%	27.93%

The datum case is extrapolated from actual site data where the building was heated to an operative temperature of 20.5°C as can be seen on Feb 2nd as shown on Figure 28 with an outdoor temperature of -23.4°C. The building is assumed to operate with 16 hours at the daytime setting and 8 hours at the night time setting. The indoor daytime operative temperature is the design heating temperature of 22°C.

For example, if the operative temperature was reduced to 15°C during the night. The night time heating load is reduced proportionally by the reduction of indoor operative temperature and outdoor design air temperature difference. The design heating load is from 22°C indoors to -18.8°C outdoors for a

temperature difference of 40.8°C, while the reduced night time temperature to 15°C will create a smaller temperature difference of 33.8°C. Thus the nighttime heating load is reduced to 10.55 kW and the day time heating load remains at 13.71 kW.

As the indoor air temperature is reduced, the leaving load temperature at the heat pump also reduces and therefore the COP of the heat pump increases. The COP as interpolated from Table 12 would increase to 4.32 from the daytime case of 3.55 for a 27.93% increase. This result in more heat produced if the heat pump is operated continuously from 13.71 kW to 13.97 kW.

This means that the heat pump does not need to operate continuously to meet the heating load resulting in the heat pump running 75.51% percent of the time. To produce the heating output, the heat pump consumes 3.24 kW and with the pumping energy it consumes 4.40 kW. Weighting the reduction to the night hours and day hour runtimes, the total heating energy required for a day is 105.4 kWh compared to the base case of 118.3 kWh, resulting in an overall energy reduction of 10.12%.

8.0 Conclusion

8.1. Summary of Findings

8.1.1. Sleep Thermal Comfort

Most current research in thermal comfort is in part derived from the Rohles-Nevins data set (Rohles, Woods, & and Nevins, 1973) and the steady state equations by Fanger (Fanger P. , 1967) and Gagge (Gagge, Stolwijk, & and Nishi, 1971) for the human sweat and shivering control.

However, a closer inspection of current literature by psychologist and physiologists such as Someren (Someren, 2000) suggests that a period of heat surplus or deficit can be said as comfortable and in instances help the patron complete specific tasks better. This is especially true for sleep (M.Raymann, Swaab, & Someren, 2008) and our understanding of sleep thermal regulation and comfort is insufficient and the data from Rohles-Nevins (Rohles, Woods, & and Nevins, 1973) and (Gagge, Stolwijk, & and Nishi, 1971) were derived from steady state awake persons.

The existing heating thermal environment requirements were identified to likely cause discomfort during sleep for being too warm. The conditions for sleep thermal comfort had been calculated and are in agreement with previous studies. A thermal model was developed based on existing literature to achieve a desired body surface temperature of 32.5°C and a heat loss rate of 36 W/m².

Based on this model, the operative temperature of a room may be reduced down to 15°C depending on the bedding insulation level.

8.1.2. Energy Efficiency

An idealized calculation of the energy reduction due to the room set point reduction to achieve various desired operative temperature levels was performed. The calculation was performed over a heating design day and is based on the reduction of operative temperature during 8 hours of night and maintaining heating design temperature for the rest of the day. The equipment performance data used in the calculation is not obtained on site but is the specified performance by the manufacturers.

The calculation shows that there is potentially up to 10% energy saving by reducing the set point during the night when people are asleep. With a reduced set point, heat pumps have improved performance and reduced run time which resulted in an estimated ~1.4% energy reduction per degree set point reduction during the night.

This suggests that additional energy reduction could be achieved in residential buildings when they are unoccupied. It also suggests the use of smart control units to implement this set point reduction during the night.

8.1.3. Infrared Sphere Method to Determine Mean Radiant Temperature

The Infrared sphere method developed to determine the mean radiant temperature was found to be more resistant to direct sunlight than the black globe method and is very similar to the mean radiant temperature estimated by the surface temperature method.

The success of the Infrared Sphere method also suggests the reflector approach is viable and more practical for HVAC control and less invasive HVAC mean radiant temperature or area weighted average surface temperature may be developed.

The reflector theory allows specialized reflector shapes to allow the infrared sensor to be more sensitive in certain directions. This allows offsetting the sensors onto the walls instead of being centrally located where the occupants are as a black globe would require. The relative simplicity of the Infrared Sphere also lends itself as a much more economical and rugged design than the Black Globe method, which requires an expensive, sensitive and fragile anemometer to determine air velocity and temperature to determine the Mean Radiant Temperature.

In summary, the infrared sphere approach in determination of the Mean Radiant Temperature shows promise and should be further explored. More tests and comparisons should be performed prior to accepting the Infrared Sphere method as an accurate approach to determine the Mean Radiant Temperature.

More importantly, the definition of Mean Radiant Temperature should be further refined.

8.1.4. Thermal Comfort Comparison

Agreeing with previous research, the radiant heated building was found to be more comfortable than the forced air heated building according to ASHRAE 55 (ASHRAE, 2010) criteria.

The thermal comfort comparison is not entirely valid as the two building envelopes are different and the field studies did not occur coincidentally. However certain conclusions can be drawn such as the system response rate to a heating demand. The radiant heated system has a quicker response than the forced air system and in a more uniform manner.

8.2. Contributions

1. The concept of sleep thermal comfort has been explored and relevant research from psychology and physiology has been incorporated in the development of a sleep thermal model. The model was found to be in general agreement with existing standards for sleeping bags design.
2. The radiant heating system was compared with traditional forced air system in two separate buildings in terms of the thermal environment created according to existing thermal comfort standards. The field measurements of a radiant heated building have shown that the operative

temperature achieved is closer to an average of 22°C; this would suggest that overheating of an average sleeping person is likely and that there is a possibility of energy consumption reduction which can be achieved by lowering the operative temperature during the hours of sleep.

3. The Infrared Sphere approach to determine mean radiant temperature has been developed and tested in two different field study locations with different heating systems and building envelopes. At both instances, the mean radiant temperature determined by the Infrared Sphere approach agrees well with the mean radiant temperature derived via traditional methods.
4. The thermal model was used to calculate the energy reduction associated with the reduction of operative temperature for sleep comfort.

8.3. Recommendation of Future Work

The limitation of this research is that it only looked at established comfort criteria as the basis of design. It is inherently limited to radiant ceiling systems thus will not apply to wall and floor systems. The experiment was setup in an existing house with an operational radiant heated ceiling; this house is not a typical house with insulation, air tightness and glazing being superior to code, thus may not be representative of other installations. The recommended future works are:

1. Currently there is no single accepted definition of sleep thermal comfort nor is it defined in detail. This has caused variations in its determination. For example Bischof et al (Bischof, Madsen, Clausen, Madsen, & Wildschodtz, 1993) used the Predicted Mean Vote method and Huang (Huang, 2008) used the ASHRAE Thermal Neutrality method, but the results attained are similar. However, at higher level of clothing, the thermal comfort condition differences become much more pronounced. Further research into the methodology of determining sleep thermal comfort is necessary for the progression of the science.
2. The definition of sleep thermal comfort should have strong theoretical backing and be supported by strenuous empirical data. The thermal comfort model currently used has not been extensively tested and the conditions it abides to were derived from limited small scale studies. This suggests that this model may not be accurate but is found to be in overall agreement with other studies. Further research into the definition of determining sleep thermal comfort is suggested.
3. The Infrared Sphere approach to determine mean radiant temperature have been tested in two distinct different field study locations with different heating systems and envelope. At both instances, it had agreed with the mean radiant temperature derived via traditional means. This

suggests that the infrared method may be a viable method to determine mean radiant temperature, and further validation is required.

4. The development of the Infrared Sphere approach also highlighted the existing inconsistency and limitation of the definition and determination of mean radiant temperature. More so, this research questions the usability of mean radiant temperature for HVAC control and the inherent difficulties in designing interior thermal environments to meet specific location based comfort standards. This prompts the development of a gross environment based standard which is a function of the space instead of being a function of the occupants. Thus allowing building control systems to operate on sensors which are inherently gross environment based.
5. What could be drawn from this study is that design parameter of the radiant system is different than traditional forced air or convector systems and a separate design and operating guideline should be developed.
6. The building management system at the Caledon site theoretically can implement the set point reduction. However the building management system was offline during the field study and was being operated manually. Further work should be completed such as energy simulation and actual field studies to verify actual energy reductions.

7. A gap in knowledge was identified between engineering researchers and psychology/physiology researchers. Engineering researchers focused on factors external to the body which engineering normally controls and fails to report on internal factors when determining thermal comfort. Likewise psychology/physiology researchers fail to quantify the external factors in detail and focus mainly on the internal factors. As thermal comfort is an interaction between the human body and its environment, a more integrated research with both engineering and psychology/physiology researchers will be beneficial to the overall understanding of the science.
8. The appropriateness and applicability of existing thermal comfort standards such as ISO 7730 and ASHRAE 55 for residential buildings is questioned. This is especially acute for the sleep condition and for high performance buildings with efficient envelopes.

9.0 Appendix

Terminology and Definitions

Operative Temperature	The operative temperature is a hypothetical temperature representing both the combination of the radiant environment and the convective environment where both radiant temperature and convective temperature is equal. The operative temperature is at the condition which this hypothetical condition is exchanging the same amount of heat as per the actual non uniform thermal environment.
Plane Radiant Temperature	Plane radiant temperature is the area weighted average temperature of a surface with view factor. Traditionally this is calculated by the radiant flux acting on a small plane element facing the measured surface. Where the element is located where the center of mass of a seated or standing person should be, 0.6 meters and 1.2 meters above the floor.
Radiant Temperature Asymmetry	Radiant Temperature Asymmetry is defined as the difference of radiant temperature as recorded by two back to back small plane measurements setup to represent the center of mass of a person. Each small plane element is measuring the radiant flux as received by the surfaces it is facing. Therefore, radiant temperature asymmetry translates directly to different radiant

	temperatures. Note that existing data developed with a relatively constant operative temperature, id est., air temperature was increased or decreased according to the radiant environment.
Vertical Radiant Temperature Difference	The largest temperature difference between the ceiling surface and the floor surface
Horizontal Radiant Temperature Difference	The largest temperature difference between opposing wall/window surface
Mean Radiant Temperature	The mean radiant temperature is the average radiant temperature an average male body will interface within a certain location within a space.
Air Temperature	Temperature of Air located at the center of mass of a person.
Mean Air Velocity	Omnidirectional air velocity located at the center of mass of a person.
Thermal Comfort	The condition of mind that expresses satisfaction with the thermal environment
Radiant Environment	The environmental condition which a person radiantly exchange heat to and from.

Convective Environment	The environmental condition which a person convectively exchange heat to and from.
Thermal Environment	The combined environmental condition of a radiant environment and a convective environment.
Relative Humidity	The ratio of actual moisture vapor pressure at the center of mass of a person over the theoretical maximum vapor pressure limit at the relevant air temperature.
Thermally Neutral	The state of a person's thermal interaction with a thermal environment that heat gain and heat loss via convection, conduction and radiation is balanced and there is no net heat gain or loss.
Time Constant	Time period for measurement measuring sensor to reach 63% of final value after a step change.
Response Time (90%)	Response time at which a mechanical system will reach 90% of its settings after a step change is initiated by the control. Equals to 2.3 times the time constant for an exponential system control.
Step Change	A change of a constant control variable to another constant control variable such as in a set point.
Temperature Cycle	The linearized rate of change of change of operative

and Drifts	temperature during a single cycle.
Thermal Sensation	Thermal sensation is measured by the ASHRAE 7 point scale where the scale ranges from +3 to -3 with each point respective to hot, warm, slightly warm, natural, slightly cold, cool and cold

Caledon Mechanical Schematic Drawing

Equipment Specification Sheets

Performance Data (60Hz) — GSW060 - Heating

SOURCE				LOAD																								
EWT °F	Flow			EWT °F	Flow 7.5 gpm								Flow 11.3 gpm								Flow 15.0 gpm							
	GPM	WPD			HC MBtuh	Power KW	HE MBtuh	LWT °F	COP	WPD		HC MBtuh	Power KW	HE MBtuh	LWT °F	COP	WPD		HC MBtuh	Power KW	HE MBtuh	LWT °F	COP	WPD				
		PSI	FT							PSI	FT						PSI	FT						PSI	FT	PSI	FT	
20	15.0	7.9	18.2	60	36.8	2.53	28.2	69.8	4.26	1.6	3.7	37.2	2.44	28.9	66.6	4.48	3.1	7.1	37.4	2.39	29.3	65.0	4.58	5.0	11.6			
				80	35.0	3.17	24.2	89.3	3.24	1.3	3.0	35.2	3.04	24.9	86.2	3.39	2.6	5.9	35.4	2.98	25.2	84.7	3.48	4.2	9.7			
				100	33.9	4.01	20.2	109.0	2.48	0.9	2.1	34.0	3.86	20.9	106.0	2.59	1.9	4.5	34.1	3.78	21.2	104.5	2.64	3.5	8.1			
				120	33.6	5.08	16.3	129.0	1.94	0.6	1.5	33.6	4.89	17.0	126.0	2.02	1.6	3.6	33.6	4.79	17.3	124.5	2.06	2.9	6.7			
30	7.5	3.1	7.2	60	38.2	2.55	29.5	70.2	4.39	1.6	3.7	38.7	2.46	30.3	66.8	4.62	3.1	7.1	38.9	2.41	30.7	65.2	4.73	5.0	11.6			
				80	36.3	3.19	25.4	89.7	3.34	1.3	3.0	36.6	3.06	26.1	86.5	3.50	2.6	5.9	36.7	3.00	26.5	84.9	3.58	4.2	9.7			
				100	35.2	4.04	21.4	109.4	2.55	0.9	2.1	35.4	3.88	22.1	106.3	2.67	1.9	4.5	35.4	3.81	22.4	104.7	2.73	3.5	8.1			
				120	35.0	5.12	17.5	129.3	2.00	0.6	1.5	34.9	4.92	18.1	126.2	2.08	1.6	3.6	34.9	4.83	18.5	124.7	2.12	2.9	6.7			
	11.3	5.0	11.5	60	40.5	2.57	31.7	70.8	4.61	1.6	3.7	41.0	2.48	32.5	67.2	4.84	3.1	7.1	41.2	2.43	32.9	65.5	4.96	5.0	11.6			
				80	38.5	3.22	27.5	90.3	3.50	1.3	3.0	38.8	3.09	28.2	86.9	3.68	2.6	5.9	38.9	3.03	28.6	85.2	3.76	4.2	9.7			
				100	37.3	4.08	23.4	109.9	2.68	0.9	2.1	37.4	3.92	24.1	106.6	2.80	1.9	4.5	37.5	3.84	24.4	105.0	2.86	3.5	8.1			
				120	37.0	5.16	19.4	129.9	2.10	0.6	1.5	37.0	4.97	20.1	126.5	2.18	1.6	3.6	37.0	4.87	20.4	124.9	2.23	2.9	6.7			
	15.0	7.4	17.0	60	42.3	2.58	33.5	71.3	4.80	1.6	3.7	42.8	2.49	34.3	67.6	5.04	3.1	7.1	43.0	2.44	34.7	65.7	5.16	5.0	11.6			
				80	40.2	3.23	29.2	90.7	3.65	1.3	3.0	40.5	3.10	29.9	87.2	3.82	2.6	5.9	40.7	3.04	30.3	85.4	3.92	4.2	9.7			
				100	39.0	4.09	25.0	110.4	2.79	0.9	2.1	39.1	3.93	25.7	106.9	2.91	1.9	4.5	39.2	3.86	26.0	105.2	2.98	3.5	8.1			
				120	38.7	5.18	21.0	130.3	2.19	0.6	1.5	38.7	4.99	21.6	126.8	2.27	1.6	3.6	38.7	4.89	22.0	125.2	2.32	2.9	6.7			
40	7.5	2.7	6.2	60	44.1	2.62	35.1	71.8	4.93	1.6	3.7	44.3	2.48	35.8	67.8	5.23	3.1	7.1	44.3	2.42	36.1	65.9	5.37	5.0	11.6			
				80	43.1	3.32	31.8	91.5	3.80	1.3	3.0	43.4	3.14	32.6	87.7	4.05	2.6	5.9	43.5	3.06	33.1	85.8	4.17	4.2	9.7			
				100	41.8	4.21	27.5	111.2	2.91	0.9	2.1	42.1	3.99	28.5	107.5	3.10	1.9	4.5	42.3	3.89	29.0	105.6	3.19	3.5	8.1			
				120	40.3	5.28	22.2	130.7	2.23	0.6	1.5	40.6	5.03	23.5	127.2	2.37	1.6	3.6	40.8	4.91	24.1	125.4	2.44	2.9	6.7			
	11.3	4.4	10.2	60	46.7	2.64	37.7	72.4	5.17	1.6	3.7	46.9	2.50	38.3	68.3	5.49	3.1	7.1	47.0	2.44	38.6	66.3	5.64	5.0	11.6			
				80	45.6	3.35	34.2	92.2	3.99	1.3	3.0	45.9	3.17	35.1	88.1	4.25	2.6	5.9	46.0	3.08	35.5	86.1	4.38	4.2	9.7			
				100	44.3	4.25	29.8	111.8	3.06	0.9	2.1	44.6	4.03	30.9	107.9	3.25	1.9	4.5	44.8	3.92	31.4	106.0	3.35	3.5	8.1			
				120	42.6	5.33	24.5	131.4	2.35	0.6	1.5	43.0	5.07	25.7	127.8	2.49	1.6	3.6	43.2	4.95	26.3	125.8	2.56	2.9	6.7			
	15.0	6.6	15.2	60	48.8	2.66	39.7	73.0	5.38	1.6	3.7	49.0	2.51	40.4	68.7	5.71	3.1	7.1	49.1	2.45	40.7	66.5	5.87	5.0	11.6			
				80	47.7	3.36	36.2	92.7	4.16	1.3	3.0	48.0	3.18	37.1	88.5	4.42	2.6	5.9	48.1	3.10	37.5	86.4	4.56	4.2	9.7			
				100	46.3	4.26	31.7	112.3	3.18	0.9	2.1	46.6	4.04	32.8	108.3	3.38	1.9	4.5	46.8	3.94	33.4	106.2	3.48	3.5	8.1			
				120	44.5	5.35	26.3	131.9	2.44	0.6	1.5	45.0	5.09	27.6	128.0	2.59	1.6	3.6	45.1	4.97	28.2	126.0	2.66	2.9	6.7			
50	7.5	2.2	5.2	60	50.0	2.67	40.9	73.3	5.50	1.6	3.7	50.3	2.53	41.6	68.9	5.83	3.1	7.1	50.3	2.46	41.9	66.7	5.99	5.0	11.6			
				80	48.9	3.38	37.4	93.0	4.24	1.3	3.0	49.2	3.20	38.3	88.7	4.51	2.6	5.9	49.4	3.11	38.7	86.6	4.65	4.2	9.7			
				100	47.5	4.28	32.9	112.7	3.25	0.9	2.1	47.8	4.06	34.0	108.5	3.45	1.9	4.5	48.0	3.95	34.5	106.4	3.56	3.5	8.1			
				120	45.7	5.37	27.4	132.2	2.49	0.6	1.5	46.1	5.12	28.7	128.2	2.64	1.6	3.6	46.3	5.00	29.3	126.2	2.72	2.9	6.7			
	11.3	4.0	9.2	60	53.0	2.69	43.8	74.1	5.77	1.6	3.7	53.2	2.55	44.5	69.4	6.12	3.1	7.1	53.3	2.48	44.8	67.1	6.29	5.0	11.6			
				80	51.8	3.41	40.2	93.8	4.45	1.3	3.0	52.1	3.22	41.1	89.2	4.74	2.6	5.9	52.3	3.14	41.6	87.0	4.88	4.2	9.7			
				100	50.3	4.32	35.6	113.4	3.41	0.9	2.1	50.7	4.10	36.7	109.0	3.63	1.9	4.5	50.9	3.99	37.2	106.8	3.74	3.5	8.1			
				120	48.4	5.42	29.9	132.9	2.62	0.6	1.5	48.9	5.16	31.2	128.6	2.77	1.6	3.6	49.1	5.04	31.9	126.5	2.85	2.9	6.7			
	15.0	6.1	14.1	60	55.4	2.70	46.1	74.8	6.00	1.6	3.7	55.6	2.56	46.9	69.8	6.37	3.1	7.1	55.7	2.49	47.2	67.4	6.55	5.0	11.6			
				80	54.1	3.42	42.5	94.4	4.64	1.3	3.0	54.5	3.24	43.4	89.6	4.93	2.6	5.9	54.6	3.15	43.9	87.3	5.08	4.2	9.7			
				100	52.5	4.34	37.7	114.0	3.55	0.9	2.1	52.9	4.11	38.9	109.4	3.77	1.9	4.5	53.1	4.01	39.5	107.1	3.89	3.5	8.1			
				120	50.6	5.44	32.0	133.5	2.72	0.6	1.5	51.0	5.18	33.3	129.0	2.88	1.6	3.6	51.3	5.06	34.0	126.8	2.97	2.9	6.7			
60	7.5	2.1	4.8	60	55.5	2.71	46.3	74.8	6.00	1.6	3.7	55.8	2.57	47.0	69.9	6.37	3.1	7.1	55.9	2.50	47.3	67.4	6.55	5.0	11.6			
				80	54.3	3.43	42.6	94.5	4.64	1.3	3.0	54.6	3.25	43.5	89.7	4.93	2.6	5.9	54.8	3.16	44.0	87.3	5.08	4.2	9.7			
				100	52.7	4.35	37.9	114.1	3.55	0.9	2.1	53.1	4.13	39.0	109.4	3.77	1.9	4.5	53.3	4.02	39.6	107.1	3.89	3.5	8.1			
				120	50.7	5.46	32.1	133.5	2.72	0.6	1.5	51.2	5.20	33.4	129.1	2.88	1.6	3.6	51.4	5.08	34.1	126.9	2.97	2.9	6.7			
	11.3	3.8	8.7	60	58.8	2.73	49.5	75.7	6.31	1.6	3.7	59.1	2.59	50.2	70.5	6.69	3.1	7.1	59.2	2.52	50.6	67.9	6.88	5.0	11.6			
				80	57.5	3.46	45.7	95.3	4.87	1.3	3.0	57.9	3.27	46.7	90.2	5.18	2.6	5.9	58.0	3.19	47.1	87.7	5.34	4.2	9.7			
				100	55.8	4.39	40.8	114.9	3.73	0.9	2.1	56.2	4.16	42.0	110.0	3.96	1.9	4.5	56.4	4.05	42.6	107.5	4.08	3.5	8.1			
				120	53.7	5.51	34.9	134.3	2.86	0.6	1.5	54.2	5.24	36.3	129.6	3.03	1.6	3.6	54.4	5.12	37.0	127.3	3.12	2.9	6.7			
	15.0	5.7	13.3	60	61.4	2.74	52.1	76.4	6.56	1.6	3.7	61.7	2.60	52.8	70.9	6.96	3.1	7.1	61.8	2.53	53.2	68.2	7.15	5.0	11.6			
				80	60.1	3.48	48.2	98.0	5.06	1.3	3.0	60.4	3.29	49.2	90.7	5.39	2.6	5.9	60.6	3.20	49.7	88.1	5.55	4.2	9.7			
				100	58.3	4.41	43.3	115.5	3.88	0.9	2.1	58.8	4.18	44.5	110.4	4.12	1.9	4.5	59.0	4.07	45.1	107.9	4.25	3.5	8.1			
				120	56.1	5.53	37.3	135.0	2.98	0.6	1.5	56.6	5.27	38.7	130.0	3.15	1.6	3.6	56.9	5.14	39.3	127.6	3.24	2.9	6.7			
70	7.5	1.9	4.5	60	59.7	2.79	50.2	75.9	6.27	1.6	3.7	60.0	2.61	51.1	70.6	6.73	3.1	7.1	60.2	2.53	51.5	68.0	6.96	5.0	11.6			
				80	58.7	3.53	46.7	95.7	4.87	1.3	3.0	59.0	3.31	47.7	90.4	5.23	2.6	5.9	59.2	3.20	48.2	87.9	5.42	4.2	9.7			
				100	57.7	4.46	42.5	115.4	3.80	0.9	2.1	58.0	4.18	43.7	110.9	4.06	1.9	4.5	58.1	4.05	44.3	107.8	4.20	3.5	8.1			
				120	OPERATION NOT RECOMMENDED																							
	11.3	3.6	8.2	60	63.3	2.82	53.7	76.9	6.59	1.6	3.7	63.6	2.64	54.6	71.3	7.07	3.1	7.1	63.7	2.55	55.0	68.5	7.31	5.0	11.6			
				80	62.2	3.56	50.0	96.6	5.12	1.3	3.0	62.5	3.33	51.1	91.1	5.49	2.6	5.9	62.6	3.23	51.6	88.4	5.69	4.2	9.7			
				100	61.1	4.49	45.8	116.3	3.99	0.9	2.1	61.4	4.22	47.0	110.9	4.27	1.9	4.5	61.6	4.09	47.6	108.2						

Description	Value	UPS 32-40 F B	
Product name:	UPS 32-40 F B		
Product Number:	96402699		
EAN number:	5700390678029		
Technical:			
Speed Number:	3
Head max:	13.1 ft		
Approvals on nameplate:	CUL		
Model:	C		
Materials:			
Material, pump housing:	Bronze 2.1176.01 DIN W.-Nr.
Material, impeller:	Stainless steel 1.4301 DIN W.-Nr. 304 AISI		
Installation:			
Range of ambient temperature:	32 .. 104 °F		
Maximum operating pressure:	145 psi		
Standard, pipe connection:	USA Oval		
Type of connection:	F		
Size, pipe connection:	32		
Pressure stage, pipe connec.:	145 psi		
Port-to-port length:	11" in		
Liquid:			
Liquid temperature range:	14 .. 248 °F		
Electrical data:			
Power input speed 1:	85 W		
Power input speed 2:	100 W		
Max. power input:	120 W		
Mains frequency:	60 Hz		
Rated voltage:	1 x 115 V		
Current in speed 1:	0.95 A		
Current in speed 2:	1.04 A		
Max. current:	1.08 A		
Cos phi in speed 1:	0.78		
Cos phi in speed 2:	0.84		
Cos phi:	0.97		
Capacitor size - run:	16 µF/400 V		
Enclosure class (IEC 34-5):	IP44		
Insulation class (IEC 85):	F		
Motor protection:	CONTACT		
Thermal protec:	internal		
R a:	17.8 - 23.2 ohm		
R s1:	3.95 - 5.2 ohm		
R s2:	9.4 - 12.4 ohm		
Controls:			
With relay:	with relay		
Pos term box:	1.30H		
Others:			
Net weight:	39.2 lb		
Gross weight:	46.7 lb		
Shipping volume:	2.08 ft³		
Sales region:	Namreg

WILO Stratos Submittal

Stratos 1.25 x 3-35

Project:	
Engineer:	
Contractor:	
Submitted By:	Date:
Approved By:	Date:

Tag #	Model #	Flow	Head	Volt/Phase	Control	IF Module	Comments
	Stratos 1.25 x 3-35			230/1/60

Control Modes

- ☐ Δp-v Pressure Differential Variable - factory default setting (set pressure is half at zero flow)
- ☐ Δp-c Pressure Differential Constant (set pressure constant)
- ☐ Δp-t Temperature Influenced Differential Pressure Control (IR Device Required to Activate)
- ☐ Automatic Night Set Back Activated - factory default setting
- ☐ Manual Speed Control (RPM x 100)

IF Interface Modules (optional)

- ☐ External Min/0 to 10 Volt DC (0 - 10 Vdc input remotely controls head set point or pump speed)/DP (Dual Pump)
 - External Min (control override) remote, volt free n/c contact
- ☐ External Off/OC (Operation Contact)/DP (Dual Pump)
 - External Off (control override) remote, volt free n/c contact,
 - Operation Contact, volt free n/o contact
- ☐ LON/DP (LonWorks BMS communication and Dual Pump)

Literature Number StratosSD1.25x3-35 ver 07.05

5.4 4" MOTOR SPECIFICATIONS

TABLE 5-1: Single Phase Motor Specifications (115 and 230 Volt, 60 Hz, 3450 RPM)

Motor Type	PENTEK Part Number	Rating					Full Load		Maximum (S.F. Load)	
		HP	kW	Volts	Hz.	Service Factor	Amps	Watts	Amps	Watts
4" 2-Wire	P42B0005A1	1/2	0.37	115	60	1.6	7.4	845	9.5	1088
	P42B0005A2	1/2	0.37	230	60	1.6	3.7	834	4.7	1073
	P42B0007A2	3/4	0.55	230	60	1.5	5.0	1130	6.4	1459
	P42B0010A2	1	0.75	230	60	1.4	Data not yet Available			
	P42B0015A2	1-1/2	1.1	230	60	1.3				
CSIR 3-Wire	P43B0005A1	1/2	0.37	115	60	1.6	Y - 11.0 B - 11.0 R - 3.0	637	Y - 12.6 B - 12.6 R - 0	916
	P43B0005A2	1/2	0.37	230	60	1.6	Y - 5.5 B - 5.5 R - 0	745	Y - 6.3 B - 6.3 R - 0	1033
	P43B0007A2	3/4	.55	230	60	1.5	Y - 7.2 B - 7.2 R - 0	1014	Y - 8.3 B - 8.3 R - 0	1381
	P43B0010A2	1	0.75	230	60	1.4	Y - 8.4 B - 8.4 R - 0	1267	Y - 9.7 B - 9.7 R - 0	1672
CSCR 3-Wire	P43B0005A2	1/2	0.37	230	60	1.6	Y - 4.1 B - 4.1 R - 2.2	727	Y - 4.9 B - 4.4 R - 2.1	959
	P43B0007A2	3/4	0.55	230	60	1.5	Y - 5.1 B - 5.0 R - 3.2	1012	Y - 6.3 B - 5.6 R - 3.1	1303
	P43B0010A2	1	0.75	230	60	1.4	Y - 6.1 B - 5.7 R - 3.3	1209	Y - 7.2 B - 6.3 R - 3.3	1532
	P43B0015A2	1-1/2	1.1	230	60	1.3	Y - 9.7 B - 9.5 R - 1.4	1693	Y - 11.1 B - 11.0 R - 1.3	2187

XE-Series Motors

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