Measurements and Analysis of Aerobiological Distribution Causing Nosocomial Infections

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A dissertation submitted to the Graduate Division in partial fulfillment of the requirements for the degree of

Doctor of Philosophy in Biomedical Engineering

South Dakota School of Mines and Technology
Rapid City, South Dakota

July 11, 2013

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ABSTRACT

Approximately 2,000,000 patients develop nosocomial infections annually in the United States (U.S.) and circa 98,987 of these individuals die [1]. These data lack specific underlying infectious agent identification as the records relates anatomically to where the infections manifest. However, there are some sources that attribute almost 10,000 deaths to airborne *Aspergillus* spores [2] and the typical spike in infections during construction projects confirm a relation between infection rates and the presence of elevated levels of airborne dust and particles. The access to cheap antimicrobials have allowed the health care industry to ignore the more costly "do-it-right" approaches but the increasing occurrence antimicrobial resistance by bacteria may cause a paradigm shift. The Center for Disease Control (CDC) also estimates that 25% of hospital acquired infections (HAI) occur because of medical staff mistakes and studies show that poor thermal comfort, limited ventilation and high humidity conditions cause more frequent human mistakes [3-5].

The building pressurization and heating, ventilation, and air conditioning (HVAC) systems substantially impact the building climate and the spread of airborne contaminants within a health care facility. Current commissioning and retro-commissioning procedures for HVAC systems are limited by existing measurement techniques and allow most systems to operate at a much lower level of efficiency than their intended design. The consequences are deficient indoor climate and poor indoor air quality (IAQ) that impact HAI rates. To overcome these issues more accurate airflow measurements, methods to diagnose and solutions to repair the problems are needed.

A new measurement technique developed through this effort is called Time-Stepped Enthalpy (TSE) has a calculated accuracy of ±2.91% and the measured airflow volume, heat transfer, and other system performance parameters are within ±3% from 300 cubic feet per minute (cfm) through 1400 cfm. Applying TSE and standards from the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) a software capable of comparing existing performance to the intended design was completed and tested. Furthermore, a new method to robotically seal supply, return and exhaust duct air leakage was developed and in an operating room (OR) the duct leakage was reduced by 16%, 10% and 39% respectively.

ACKNOWLEDGEMENT

I would like to express my gratitude to Dr. Charles Tolle for his guidance and support during my time at South Dakota School of Mines and Technology. I would also like to thank my committee members Dr. Sookie Bang, Dr. Carter Kerk, Dr. Dimitrios Anagnostou and Dr. Yanxiao Zhao for their support and understanding through my Ph.D. studies and dissertation efforts.

Furthermore, thanks to Lloyds Systems LLC for financially supporting this effort, Carlisle HVAC for providing duct sealants, Bob Carlson for third party measurements data, Anthony Gervasi for software development, John Bradley for assisting in the laboratory, and to South Dakota School of Mines faculty for assisting and directing this research. Without their effort, the completion of the project would not have been possible. Additional gratitude is given the team at United Test and Balance, Inc. for assisting in data collection and for the use of instrumentation at the South Dakota Hospital. Without their effort, the completion of the project would not have been possible.

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1. INTRODUCTION

1.1 BACKGROUND

Nosocomial or hospital acquired infections (HAI) is a substantial challenge for the health care industry. Data from Center of Disease Control (CDC) Nosocomial Infections Surveillance Systems reports that the annual estimated number of U.S. deaths related to the nearly 2 million infections occurring is 98987 [1-3]. Of these deaths 35967 is attributed pneumonia, 30665 bloodstream, 13088 urinary tract, 8205 surgical site and 11062 other sites. However, the underlying infectious agent or pathogen is not identified and reported, thus making it challenging to identify underlying causes. The principal avenues of transmission of HAIs or the germs causing these infections are contact, droplet, airborne and common-vehicle [4]. The most common and most preventable transmission is direct and indirect contact. Direct contact involves body to body surface contact with a physical transfer of microorganisms, while indirect contact (or cross-contamination) involves body surface contact with a contaminated surface. Hand hygiene both before and after contact significantly reduces the possibility of infection transmission and helps reduce transmission via common contact surfaces too. Droplet transmission occurs when droplets containing microorganisms from an infected person are propelled through the air and land on the mouth, eyes, or nose of another person. Droplets are primarily generated when a person is coughing, sneezing, or talking. Droplets do not remain suspended in the air. Airborne transmission occurs when a droplet containing microorganisms evaporates and remains suspended in the air for a long time or when dust particles containing infectious agents circulate. A good precaution when working in close contact (3 feet or less) with an infected person is to use personal safety equipment that covers nose, mouth and eyes. Common vehicle transmission refers to contaminated items such as food, water, medications, devices, and equipment.

1.1.1 Infectious Disease Medicine

Nosocomial or HAI sorts under infectious disease medicine that focuses on germs. A germ is a microorganism known as a pathogen or infectious agent and may be a virus, bacterium, fungus, parasite or prion that causes disease in its host [5]. A virus is a small infectious agent that can replicate only inside the living cells of an organism. Viruses are the most abundant biological entities on Earth where they outnumber all the others combined [6-7]. A virus consists of protein and lipids covered deoxyribonucleic acid (DNA) or ribonucleic acid (RNA). Viruses can infect all types of organisms, from animals and plants to bacteria and archaea. Viral infections in humans provoke an immune response that usually eliminates the infecting virus. Vaccines target this immune response to provide an artificially acquired immunity to the specific viral infection.

Bacteria constitute a large domain of prokaryotic microorganisms which do not have a nucleus, mitochondria, or any other membrane-bound organelles. Their proteins, DNA and metabolites are located together in the same area enclosed by cell membrane, rather than separated in different cellular compartments. The small size of a few micrometers and their various shapes allow approximately 40 million bacterial cells in a gram of soil [8]. Fungi are members of a large group of eukaryotic microorganisms such as yeasts, molds and mushrooms that produce mycotoxins. A mycotoxin is a secondary metabolite produced by these organisms and two famous examples are antibiotics and alcohol [9-10]. It is estimated that over five 5 million species exist and, along with bacteria, fungi are the major decomposers in most terrestrial and some aquatic ecosystems.

Parasitism is a non-mutual relationship between organisms of different species where the parasite benefits at the expense of the host. Most parasites have the ability resist the host innate defense and then evade the host's immune response for successful establishment. Parasites show a high degree of specialization, and reproduce at a faster rate than their hosts. Viruses and bacteria are defined as microparasites and can be directly transmitted between hosts of the same species [11].

Prions are a family of rare progressive neurodegenerative disorders that affect both humans and animals. These pathogenic infectious protein particles are transmissible and induce abnormal folding of specific normal cellular proteins called prion proteins that are found most abundantly in the brain. The abnormal folding of the prion proteins leads to brain damage and the characteristic signs and symptoms of the disease [12].

1.1.2 Infections

An infection or immune response occurs when a germ successfully invade bodily tissues to combat their presence, multiplication and toxins they produce [13]. A short-term infection is an acute infection whereas a long-term infection is a chronic one. Infection can take place via many potential routes through horizontal or vertical transmission. Horizontal transmission passes the infecting organism from person to person in the same generation while vertical transmission happens from a mother to a child. As stated earlier, infectious organisms may have direct or indirect contact exposure pathways. Direct contact occurs when an individual comes into contact with the reservoir. Indirect contact occurs when organisms are able to withstand the harsh environment outside the host for long periods of time. Infection may occur through airborne, droplet, endogenous, tunnel and opportunistic scenarios. An airborne infection is one that is contracted by inhalation of microorganisms or spores suspended in air on water droplets or dust particles [14]. Droplet infection occurs when respiratory pathogens are exhaled by someone already infected and the suspended liquid particles are inhaled by another host. Endogenous infection is the reactivation of organisms present in a dormant focus. A tunnel infection may manifest itself when an artificial passage into the body has been created. Opportunistic infection establishes when an organism that does not ordinarily cause disease becomes pathogenic under certain circumstances. As HAIs usually relate to bacterial, viral and fungi, the remaining part of this chapter emphasis on these.

Bacterial and viral infections can cause similar symptoms including coughing, sneezing, fever, inflammation, vomiting, diarrhea, fatigue, and cramping, all of which are methods the immune system tries to get rid of the body of infectious organisms. Viral infections are usually systemic and affect several different body parts but local variants exist such as in viral conjunctivitis and herpes. Most viruses cause disease and attack specific cells in the liver, respiratory system, or blood. When virus target bacteria they are called bacteriophages [15]. Virus can only reproduce by attaching themselves to cells and hijacking the cellular machinery. Quite frequently the cells are reprogrammed to make new viruses until the cells burst and die or the cells are turned into malignant or cancerous cells. Usually bacterial infections are classified by the pathogen in tandem with symptoms and medical signs produced. Bacterial infections are symptomatic, subclinical or latent where only symptomatic causes localized redness, heat, swelling and pain. Some infections produces pus and milky-colored liquids such as during a tooth abscess or as seen as a pimple. Bacterial infections may cause local pain at the affected area whereas pain rarely manifest during viral infections.

The fungi that produce toxins are known as toxigenic fungi (or molds) and are of primary interest in health care facilities. Five families of molds are producing the most toxic spores or mycotoxins that affect human health: *Aspergillus*, *Cladosporium*, *Penicilium*, *Fusarium*, and *Stachybotrys*. The mycotoxins are thought to be used to kill nearby competing molds or bacteria to create a competitive advantage. Some mycotoxins cling to the surface of mold spores while others exist within spores. Mycotoxins may have deleterious effects on humans if ingested, inhaled or by skin contact. Mild toxins usually cause hay fever-like allergic reactions, while potent ones trigger deadly illnesses. The *Stachbotrys chartarum* produces the most deadly mycotoxins that have been tied to liver damage, pulmonary edema, brain or nerve damage and death. Only 16 of 160 species of *Aspergillus* cause illness in humans, but *A. flavus* and *A. parasiticus* produces Aflatoxin B1 that is one of the most potent carcinogens known. Carcinogens have long incubation time, thus making it difficult to link the exposure to health care facilities. However, it is also known that approximately 14% of *Aspergillus* caused HAI's is attributed *A. flavus*. Three other HAI causing molds are *A. fumigatus* (66%), *A. niger* (5%) and *A. terreus* (5%) [16]. This diminishes the more minor effects of the *Cladosporium*, *Fusarium*, and *Penicillium* families that usually relate to asthma and infections of the lungs, liver, and kidneys.

1.1.3 Immune System

The immune system keeps infectious microorganisms out of the body and attempts to destroy any germs that successfully invade. It consists of an innate and an adaptive portion. The innate immune system is immediate and functions as the first line of defense against invading organisms. The adaptive immune system is slow and it may take up to 15 days to trigger a full first exposure response. However, for re-exposures a quick 10-day response with a 100-fold immune intensity may result. It is this phenomenon that is the underlying feature in a vaccine. These slow adaptive immune responses confirm the importance of the innate immune system as it prevents a pathogen from causing considerable harm at the initial stage of an infection [17].

The elements of the innate immune system include anatomical barriers, secretory molecules and cellular components. Among the mechanical anatomical barriers are the skin and internal epithelial layers, the movement of the intestines and the oscillation of broncho-pulmonary cilia. Associated with these protective surfaces are chemical and biological agents designed to recognize molecules shared by groups of related microbes that are essential for the survival of those organisms and are not found associated with mammalian cells. The response triggers various responses that transport neutrophils, eosinophils, monocytes, macrophages, mast cells, basophils, eosinophils and natural killer cells to the infected area.

The adaptive or acquired immune system is antigen specific and reacts only with the organism that induced the response to eliminate or prevent pathogen growth. An antigen is defined as a substance that reacts with antibody molecules and antigen receptors on lymphocytes. An immunogen is an antigen that is recognized by the body as foreign and stimulates an adaptive immune response. The actual fragment of an antigen that reacts with an antibody and lymphocyte receptor is an epitope. The immune system recognizes an antigen as foreign when epitopes of an antigen bind to epitope-specific receptor molecules on the surface of B-lymphocytes (BCR) or T-lymphocytes (TCR). An estimate is that the adaptive immune system may recognize more than 10^7 different epitopes and generate 10^9 unique antibodies [18].

Both the blood and lymph vessel networks transport the immune system or white blood cells around the body. The lymph flows from the interstitial fluid through lymphatic vessels up to the thoracic duct or right lymph duct to mix into the subclavian blood veins. All blood cells are produced from stem cells in the bone marrow via a process called hematopoiesis. The stem cells convert into hemocytoblasts that differentiate into the precursors for erythrocytes (red blood cells), leukocytes (white blood cells), and thrombocytes (platelets). The leukocytes are further subdivided into granulocytes (neutrophils, eosinophils, and basophils) and agranulocytes (monocytes, B-lymphocytes and T-lymphocytes).

An infection may become deadly when an overwhelming immune response is triggered. Chemicals released into the blood to fight infection trigger widespread inflammation that may result in organ damage. Blood clotting in small blood vessels are triggered by the innate immune system as it help fight pathogenic microbes. However, extensive blood clotting reduces blood flow to limbs and internal organs, depriving them of nutrients and oxygen to a point where one or more organs fail. Still the main cause of death is septic shock where the infection leads to a life-threatening drop in blood pressure which quickly leads to the failure of lungs, kidneys, and liver [19].

1.1.4 Treatment Options

To assist the innate and adaptive immune system in fighting an infection several treatments have been discovered and developed. All *penicillins* are β-lactam antibiotics, which kills or stops the growth of most Gram-positive bacteria inside the body. A Gram stain test is used to classify two distinct types of bacteria based on the existence of an outer membrane cell wall as it prevent the penetration of the stain. Compared with Gram-positive bacteria, Gram-negative bacteria are more resistant against antibodies, because of their impenetrable wall. It is not just the Gram-negative bacteria that are a challenge to treat. The antibiotics have no effect on viruses, so several antiviral drugs have been developed. However, some viruses including those that cause AIDS and viral hepatitis evade these drugs and result in chronic infections. Furthermore, more and more bacteria are becoming *penicillin* resistant thus new treatment methods are needed to fight infections. Subsequently, new treatment options are developed including honey based creams for wound care. An additional concern is the use of antimicrobials has been a cheap method to sanitize and disinfect surfaces like bodies, equipment or tools. This extensive use has made many bacteria resistant to antimicrobials and is a major concern for those who work to reduce nosocomial infection rates.

1.2 HEALTH CARE FACILITIES

There are already several building codes that govern the design of health care facilities [20-21]. However, substantial room for improvement exists to better support the flow of patient services with reduced nosocomial infection rates in mind. Although there are many areas of possible improvements, the main focus in this effort is the impact on nosocomial infections by the HVAC system and building pressurization [22-23]. Virus, bacteria and fungi dominate the cause of nosocomial infections and due to the physical size of a variety of these germs distribution is possible through the HVAC system and/or in tandem with varying building pressurization. System operation at or close to the intended design allow sufficient moisture control to prevent mold growth. However, as molds need both humidity and food to proliferate an additional objective is complete source removal of potential molds foods. Such dual impact method will further reduce the probability of the establishment of any mold colonies within the building envelope.

1.2.1 Heating Ventilation and Air Conditioning (HVAC) Systems

Figure 1.1 illustrates a typical HVAC system where the fan pulls air from the building space as Return Air (R/A) and a portion becomes Exhaust Air (E/A) to meet American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) 62.1-2007 ventilation requirements. Outside Air (O/A) is pulled in to replace the E/A volume and will mix with the remaining R/A. The mixture of R/A and O/A have various air velocities and temperature profiles while propagating through the filter bank to be heated or cooled according to seasonal demand and/or any ASHRAE 55-2004 settings. After changing the airflow enthalpy, at the coils, the fan pushes the conditioned air through the ductwork as Supply Air (S/A). The S/A flow is delivered to the various rooms by interconnected ductwork and associated diffusers.

The filtered and conditioned air flow provides thermal comfort to support healthy and comfortable indoor environments for the building occupants.

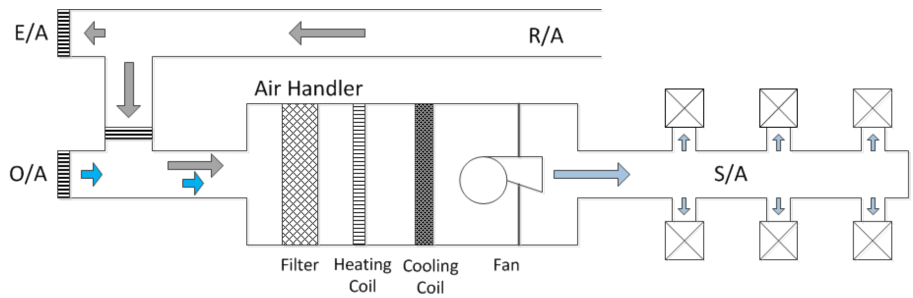

Figure 1.1: A typical draw through HVAC system.

High Efficient High-efficiency particulate arresting (HEPA) filtration of airflows in a heath care facilities is very important to prevent germs to spread within the building envelope. U.S. government standards require a HEPA filter to remove from the airflow 99.97% of particles that have a size of 0.3 micrometres (µm) [22]. This particle size is the most challenging to filter, therefore larger or smaller particles are filtered at a better efficiency than 99.97%. This effect is accomplished by many intertwined fibers where particles become trapped either by interception, impaction, or diffusion. Interception happens when air is pushed through a filter and the particles that come within one radius of a fiber will stick to that fiber. Larger particles cannot avoid the fibers and become embedded in the fibers by impaction when the air is pushed through the filter. The smallest airborne particles <0.1µm become trapped in the fibers in a process called diffusion. Tiny air particles constantly collide with gas molecules and the force of contact cause the particles to either stick to or embed in the fibers [24]. The filter medium is usually composed borosilicate fibers treated with a wet-strength water-repellant binder.

A HEPA filter is not shown in Figure 1.1 but is often located either after the fan or at each diffuser. The 2003 CDC Guidelines recommend that filter frames are metal as wooden ones can compromise the air quality if it becomes and remains wet, thus allowing the growth of fungi and bacteria. Wet wooden frames also expands and thus may compromise the filter seal to allow air bypass. Some manufacturers report that high humidity and high moisture will affect the overall resistance of a HEPA filter. Water droplets or mist form within the air stream and can collect to a point of saturation on the filter media. At this point, the moisture will block the effectiveness of the airflow and actually disintegrate the media structure by breaking down the latex based adhesives used to provide the filter fiber structure. Whereas other manufactures are not very concerned about wet HEPA filters and the post-performance, consensus is that ultra violet light should be installed by the filter to kill off any germs captured and to prevent any microbial growth [25-26].

1.2.2 Building Pressurization

The HVAC system in tandem with toilet, bathroom, kitchen, and dryer exhausts are major air movers in a building. Furthermore, the balance between O/A intake, intrinsic building pressurization and air escaping from the building envelope has significant impact on air movement. The intrinsic building pressurization has two main contributors: wind pressure and the stack effect. These effects that drive natural ventilation of buildings and will be discussed in detail in the next two sub-chapters [27]. Very strong winds may be created within a building and its direction and force may change depending on the pressure differentials, door and window positions. These winds are not HEPA filtered thus have a major effect on how construction duct, particles, lint and contaminants that harbor germs are circulating in a building. Furthermore, the building pressurization also affects the resistance of the HVAC airflow where high head pressure reduces airflow. Within the building envelope other air pressure disturbances exist such as a moving elevator that functions as a large piston moving a bubble of air up or down depending on the elevator direction.

1.2.2.1 Wind Pressure

With reference to the atmospheric pressure wind induces a positive differential on the upwind face of the building and negative pressure differentials are created on the sides and at the rear. The windward roof face may also be at a negative pressure unless the pitch angle exceeds 30°. Within an urban environment, the wind speed at building height may in some cases be less than half of that measured at a meteorological station. Therefore, wind speed corrections are used to determine approximate wind pressures for buildings in urban settings. The wind pressure (P_w) created when wind strikes a building is expressed in Pascal and is described in formula 1.1. P_w is dependent on the air density (p), wind pressure coefficient (C_p) and the wind velocity at building height (V_r).

$$P_w = \frac{\rho \cdot C_p \cdot v_r^2}{2} \qquad (1.1)$$

If the wind pressure were isolated and was the only active variable, the pressure differential would vary across the building envelope with air infiltration on the positive pressure side and possible equilibrium for the rest of the envelope. However, by combining the wind pressure component with the stack effect and the HVAC system, a very complex pressure differential environment results.

1.2.2.2 Stack Effect

The temperature differential between the inside and outdoor air temperatures creates the stack effect. When the indoor temperature is higher than ambient, the indoor air is less dense and lighter. At the level where the pressure differential is in balance a neutral pressure plane exist while it has a negative below and a positive pressure above this plane. If a door or window was opened at the neutral pressure plane the airflow would be negligible.

The stack pressure (P_s) is calculated using the ideal gas law using the following variables: air density (p), gravity (g), height of opening (h), outside temperature (t_{ext}) and inside temperature (t_{int}). The temperature difference or stack induced pressure at an opening at any arbitrary datum height is found by Formula 1.2.

$$P_s = -\rho \cdot g \cdot 273 \cdot h \cdot \left(\frac{1}{t_{ext}} - \frac{1}{t_{int}}\right) \qquad (1.2)$$

When the indoor temperature is higher than ambient the stack pressure is negative pulling in outside air while it becomes positive when ambient is warmer than the inside air. This reversal of pressure differentials in tandem with the other variables may sometimes be difficult to manage.

1.3 MOTIVATION

The motivation behind this research is to reduce the outrageous number of nosocomial infections that occur in the U.S. today. Over 2,000,000 patients develop HAI annually and approximately 100,000 of these patients die either by septic shock, widespread inflammation or blood clotting. The manifested infections sites are bloodstream, pneumonia, ventilator-associated pneumonia, urinary tract and surgical site, but available data makes it difficult to assign the cause as to why these infections actually occur. Is it due to poor sanitation, sterilization, mistakes by medical staff, movement of patients and equipment, patient isolation, HVAC system or building pressurization? Current commissioning and retro-commissioning procedures for HVAC systems are limited by existing measurement techniques and allow most systems to operate at a much lower level of efficiency than their intended design. Furthermore, there are no assessments related to the combined building pressurization and HVAC system operation. The consequences are nosocomial infections, waste of building energy, poor indoor IAQ and increased system operating costs. Current measurements of airflow are completed using either by Pitot tubes, hotwire anemometer or tracer-gas, but these methods are often inaccurate or cost prohibitive.

To address the current nosocomial infections rates, new approaches to prevention are needed. This research aims to assess core capabilities for a complete system approach on how germs may propagate within a health care facility building envelope and to identify what is needed to reduce the medical staff mistakes causing an estimated 25% of all reported HAIs.

The first step in this process was to assess the ability to properly measure HVAC airflow since Pitot and hotwire aneometer measurements are only ±10% accurate. The first objective is therefore to determine if a new method to measure HVAC airflow within ±3% can be devised. If this objective is met, would it then be possible to use computational analysis to perform a complete system performance assessment to identify underlying factors that directly and indirectly cause issues. Then, can repair procedures of any shortcomings be identified? If not, can methods be invented to resolve any problem area identified?

1.4 BROADER IMPACT AND SIGNIFCANCE

The World Health Organization (WHO) views HAI and antimicrobial resistance (AMR) as overlapping areas and associates high rates with weak health care systems. The fear is that AMR coupled with HAI may drive the death rates to unprecedented levels. It is asserted that the use of antimicrobials is the key driver of resistance. The AMR pressure come extensive use of antimicrobials. In addition, the overuse of antibiotics for minor infections, misuse due to lack of access to appropriate treatment, and underuse due to lack of financial support to complete treatment courses deteriorate the value of this option. It is also asserted that poor infection control is the key driver of HAIs. Infection control is acknowledged universally as a solid and essential basis towards patient safety and supports the reduction of HAIs and recognizes the severe consequences. AMR and HAI are serious global health and economic issues where countries with less well organized health care settings such as Brazil, Morocco and Malaysia have almost twice as high HAI rates peaking at 19.1% as is observed in most developed countries [28]. In Europe the HAI rates vary from 3.5% to 14.8% with an average of 7.1%. This translates in to 4.1 million affected patients, 16 million extra days of hospital stay and 147,000 deaths. The annual economic impact of only the direct cost is approximately 9 billion U.S. dollars [28]. Compared to the U.S. having a population that is 2.4 times smaller than Europe the HAI numbers are 2 million patients affected where about 100,000 of these die. However, the economic impact in the U.S. is estimated to be in the range from $28.4 to $33.8 billion (2007 dollars) [1-3]. HAI deaths are expected to increase in the future as more and more Gram positive and negative bacteria become AMR. To date, 28 states and the District of Columbia require reporting of HAIs using CDC's National Healthcare Safety Network (NHSN). Over 9000 hospitals, in all 50 states now use CDC's NHSN to track HAIs. An example of data reported is illustrated in Table 1.1 comparing an average hospital with a large medical center.

Table 1.1: Estimates to relate HAI to size of Hospitals in the U.S.

	Hospital	**Medical Center**
Number of beds	165	1000
Number of HAI Deaths/year	17	106
Number of HAIs/year	348	1122
HAI Cost per CDC	$12.8M	$78M
HAI Cost per Pennsylvania State	$64.5M	$391M

An important observation from the table is that the CDC economic impact figures are approximately five (5) times lower than those calculated by representatives from the State of Pennsylvania. Thus the economic impact numbers may be substantial higher than those used by CDC and WHO.

It has been suggested that prophylactic antimicrobials are at least as effective as ultraclean air and exhaust ventilated operation rooms in reducing deep sepsis for total joint replacements [29]. The push has therefore been to "exhaust" cheap prophylactic antimicrobials. However, the increasing AMR has shown that this option is becoming an expensive option for the health care sector in general. It has been established that indoor air is potential source of microorganisms that can contaminate surgical wounds [30-32]. Operating room air is often contaminated with microorganisms that are usually attached to other airborne particles such as dust, lint, skin squames, or respiratory droplets. Many of these microorganisms are potential pathogens and will settle down on instruments, surfaces or be suspended in the air over a long period of time. Greater numbers of airborne microorganisms occur with increased numbers of persons, especially if doors are being opened and the persons are moving, talking, or have uncovered skin areas. Therefore, traffic control and closed door (to prevent corridor air contamination) surgeries in tandem with producing a minimum of 20 changes of highly filtered air per hour should reduce the suspended air particle counts. The particle count can be further reduced by installing "laminar flow" ventilation units to provide marginal air turbulence. Another significant discovery was that spores of Clostridium difficile easily spread via airborne means where spores were isolated from the air near 7 of 10 patients with active disease [33]. In tandem with developing methods to reduce transmission pathways another important aspect is that CDC reports that 25% of all HAI's related to medical staff mistakes. An established fact is that productivity decrease and mistake rates increase if the HVAC system does not properly ventilate or control thermal comfort and humidity. Studies show a close correlation between human performance and higher ventilation rates [34]. Furthermore, other studies report a significant correlation between the human perceptions of thermal environment to productivity loss [35]. Therefore, a crucial component in reducing medical mistakes that cause nosocomial infections is to operate the HVAC system at or close to its design specifications.

2. ACCURACY AND PRECISION ANALYSIS

This chapter reports the findings related to the first objective of this research effort. The full title of this journal paper is "Accuracy and Precision Analysis of Time Stepped Enthalpy" and it was submitted to An International Journal devoted to Investigations of Energy Use and Efficiency in Buildings.

2.1 ABSTRACT

In this effort an analysis was performed to quantify the impact instrumentation and other system uncertainties have on the accuracy and precision of the new and innovative Time Stepped Enthalpy (TSE) method. All conversions of energy taking place in the system under test (SUT) were identified and quantified to verify system performance. All data was collected using a SUT built around a negative air machine with a variable frequency drive (VFD) and a high-efficiency particulate air (HEPA) filter. A 13.2 kilo Watt (kW) electric heating element was installed to provide the needed energy to heat the cool outside air (O/A). The SUT was used to measure the known airflow volume with three different methods: TSE, hotwire anemometer (HWA) and an averaging Pitot tube (APT). Measurements were made for airflow volumes in the range from 200 to 1200 cubic feet per minute (cfm) before the data was analyzed and compared. Measured TSE data yields ±3% accuracy and ±3% precision for airflow volume in the range from 300 cfm through 1200 cfm. The calculated accuracy based on instrument uncertainties yields ±2.91%. This shows a strong correlation between the calculated and measured results.

2.2 INTRODUCTION

The importance of Heat Ventilation and Air-Conditioning System (HVAC) energy efficiency in health care, commercial and industrial buildings is continually increasing. The annual energy consumption in the United States alone exceeds $51 billion dollars [36-37]. Federal, state, county and city governments consume an additional $18 billion annually in building-related energy [38]. HVAC systems running inefficiently often cause poor IAQ and thermal comfort [39-43]. For certain buildings these consequences are more economically devastating than the actual waste of energy. Reduced worker productivity and comfort may be related to latent heat buildup, mold growth, odors, high CO_2 levels, particulate contamination, noise pollution etc. [44-46]. A wide variety of measurement techniques are currently used to inaccurately measure airflows and other important parameters in HVAC systems. This inability to properly obtain operational data through measurements has moved the majority of the industry down an energy modeling path [47-50]. Future high-performance HVAC systems will rely on variable air volume distribution and more advanced control systems. However, the challenges in utilizing effective measurement techniques slow down the innovation and development of these super-efficient HVAC systems. The industry was in search of a quick, easy and accurate method to measure HVAC parameters such as airflow and heat transfer when the time-stepped enthalpy concept was invented.

2.2.1 Time-Stepped Enthalpy Method

The TSE method is the main underlying method to be evaluated in this effort [51-56]. TSE is based on psychrometrics, thermodynamics and other parameters to quantify the energy released into or extracted from the HVAC system. Equation 2.1 is the formula for the introduction of electric heat into a HVAC system. The total heat released is the voltage (V) multiplied by the current (I) and the conversion factor between watts (W) and British thermal units per hour (BTU/hr).

$$Q_E = 3.41214163 \cdot I \cdot V \left[\frac{BTU}{hr}\right] \qquad (2.1)$$

The measured energy Q_E is substituted as Q_A in the total heat formula in Equation 2.2 along with the delta enthalpy reading obtained by running the system at full system capacity and then with no heat transfer present.

$$Q_A\left(\frac{BTU}{hr}\right) = V_{airflow}\left(\frac{cf}{\min}\right) \cdot 4.5\left(\frac{\min \cdot lb}{hr \cdot cf}\right) \cdot \Delta h\left(\frac{BTU}{lb}\right) \quad (2.2)$$

The airflow volume is then found by rearranging Equation 2.2 and inserting the measured values of total heat energy delivered and delta enthalpy.

2.2.2 Error Sources

Several already described sources of errors exist for the TSE method. The first is related to the inverse multivariable nature of the total heat equation used to calculate airflow. For small delta enthalpy values, e.g. 1 BTU/lb., a variation of only 0.1 BTU/lb. may translate to a 10% error. In contrary, with delta enthalpy of 10 BTU/hr. the same variance yields accuracy within 1% [57]. Secondly, management of both the dew point moisture, due to cooling of air and the relative humidity is required to consistently use the static pressure to aid system diagnostics. It is recommended that static pressure measurements are performed only when the cooling coil is dry and when the relative humidity is below 55% [57]. Next, Figure 1 illustrates the changes in volume airflow and fan power consumption for typical HVAC system operation.

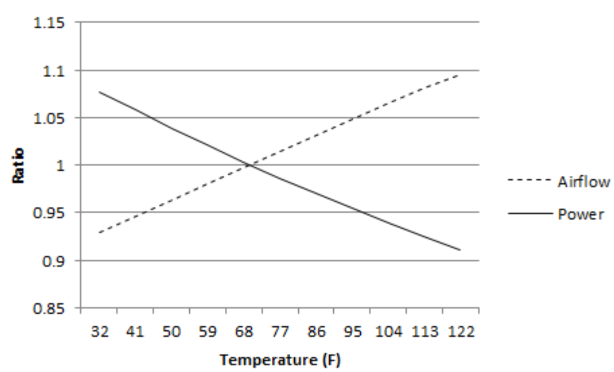

Figure 2.1: Airflow and Power Consumption Ratio

It can be seen that both the airflow volume and the fan power consumption may vary 15.2% to 18.3% between the low and high extreme points depending on which reference values are used. Diligence is required for the development of the testing protocol required to manage these dynamic variations. This is especially important for TSE as the wet/dry bulb temperatures and enthalpy are the manipulated parameters. Furthermore, system effects such as air exchange rate, fan motor drift and unwanted thermal feedback loops are also error sources unless proper care is taken. A final error source worth mentioning is the inaccuracies of the instruments that are used to measure the needed parameters. This is further described in section 2.2.3.

2.2.3 Root Sum Squares

The root sum squares (RSS) is a statistical analysis method to associate the probable error of a measurement method. It is a simple mathematical formula used to estimate the probable error by only taking one set of readings. If a parameter N is a function (f) of other parameters $x_1, x_2, \ldots, x_n$ N is computed by measuring $x_1, x_2, \ldots, x_n$, and each measurement of $x_1, x_2, \ldots, x_n$ has a certain degree of error or uncertainty; the overall error, in the computed value of N, E_{RSS} is given by Equation 2.3 [58-59].

$$E_{RSS} = \sqrt{\left(\Delta x_1 \frac{\partial f}{\partial x_1}\right)^2 + \left(\Delta x_2 \frac{\partial f}{\partial x_2}\right)^2 + \cdots + \left(\Delta x_n \frac{\partial f}{\partial x_n}\right)^2} \qquad (2.3)$$

To illustrate the application of RSS, an example focusing on the heating element is investigated. The energy converted into the airflow using an electric heating element has an efficiency of 100%. The electric power is measured using two instruments, a voltage and an ampere meter with accuracy of ±1% and ±1.5% respectively. Using these instruments the current was measured to 44.1A and the voltage to 207.2V. These two measurements are the only ones necessary to find the probable measured error. Equation 2.3 is rewritten for the partial derivations of voltage and current and the result is Equation 2.4.

$$\Delta P = \sqrt{\left(\Delta V \cdot \frac{\partial P}{\partial V}\right)^2 + \left(\Delta I \cdot \frac{\partial P}{\partial I}\right)^2} \qquad (2.4)$$

Then executing the partial derivations and substituting these into Equation 2.4 yields Equation 2.5.

$$\Delta P = \sqrt{(\Delta V \cdot I)^2 + (\Delta I \cdot V)^2} \qquad (2.5)$$

Utilizing the measurements and the accuracy data provided for the instruments, the delivered energy into the system may be expressed 9138 W ± 165 W or ±1.8%. By utilizing RSS on all relevant system processes, the probable measurement error (E_{RSS}) of TSE was calculated to ± 2.91%.

2.3 DATA COLLECTION

The challenge to properly assess feasibility and measurement accuracy of new measurement methods is to create a controlled environment where all dependent and independent variables may be monitored and possibly adjusted if necessary. Initial studies using TSE indicated that the measured airflow volumes corresponded well with HWA [60-62] and Pitot tube [63-65] measurements in laminar flows. Although such observations were encouraging, the frequent challenges of obtaining laminar flows that yield accurate and precise results using those techniques prompted the need for a more in depth analysis to determine the actual performance of TSE.

2.3.1 Instrumentation

The SUT as seen in Figure 2.1 was built around a VariAire negative air machine and three electric heating elements. The study monitored all energy flows into and out of the SUT. To measure the electric power into the fan motor and the electric heating elements, two Fluke multimeters were used: Fluke-324 Clamp Multimeters ±1.5% to measure current and Fluke 87-V Digital Multimeter with ±1% accuracy to measure the voltage. Multiplying the voltage and current provide the electric power into the system. A Vailala LM70 ±1% was used to measure the properties of the incoming outside air (O/A) and a second one was used to measure the same in the exhausted air (E/A). E/A and supply air (S/A) are considered the same in this SUT as S/A is ventilated outdoors. The static pressure pre-HEPA and post-HEPA were also monitored using two Omega Engineering sensors with ±1% accuracy. Three different instruments were used to measure the system airflow volume to allow a comparison to TSE. These instruments were a handheld TPI 575C1 vane/hotwire anemometer (VA/HWA) accuracy ±2% and ±5% along with a Kemo DMB-610 averaged Pitot tube (APT) at ±3%.

2.3.2 Test Setup

A laboratory test setup was designed to allow measurements in a controlled environment. Homogeneous outside air (O/A) was pulled in through a 16 feet long and 14 inch diameter pipe by a negative air machine. O/A enthalpies, relative humidity, dry, and wet bulb temperatures were continuously monitored during data acquisition. The pipe length provided laminar flow (Re < 40) to allow measurements using HWA and averaging Pitot tubes as references as shown in the upper portion of Figure 2.2.

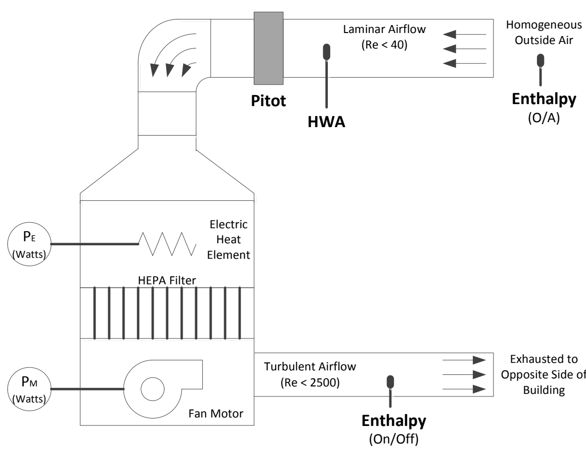

Figure 2.2: Image of the experimental laboratory setup

The airflow then bends and enters the negative air machine where an electric heating element was installed to turn on or off conditioning of the airflow emulate an air-handler. A HEPA filters the air and provide a static pressure drop similar to operational units in the field. In the bottom sits the fan motor that drives the air through the system and the airflow is blown through a smaller diameter pipe to the rear of the building. It is here that the downstream and upstream enthalpies are measured. All energy into the system is closely monitored and accounted for and a fan motor VFD enables measurements for a wide range of airflow volumes. All energy entering or exiting the system under test must be identified and quantified. The O/A energy is quantified by its enthalpy, the electric heat element by its input current and voltage and the fan motor system as illustrated in Figure 2.3.

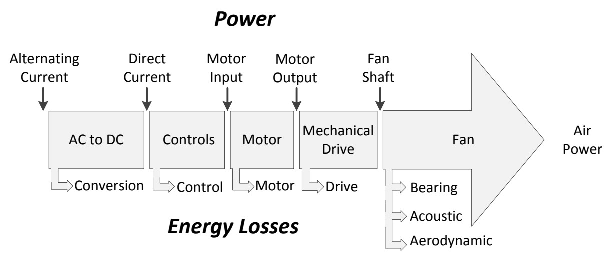

Figure 2.3: Energy Conversion Losses for Fan Motor

110 VAC (measured by a multimeter) is rectified to a DC bus by the VFD; this is then converted to ~230 three phase (measured by the VFD) to power the fan motor. The remaining energy, less what is consumed by the control circuitry, is the motor power input. The motor, mechanical drive and fan have five major energy losses leaving the remaining energy as air power. All other energy lost is converted into heat by the friction between the airflow and the walls, joints, filter and seams in the system. Air power gets converted to total pressure and airflow volume. This theoretical pathway is used to verify the airflow volume to be measured.

2.3.3 Measurements

The airflow volume in this study was measured using three independent methods in tandem with a close monitoring of the system energy to allow an accurate theoretical numeric quantification of the parameter of interest. As seen in Figure 1 the first measurement point is to monitor the energy and humidity content of the O/A used in the study. The second measurement point is an airflow volume reference using a HWA and the log linear rule for traverse points for a round duct bigger than 10 inches as illustrated in Figure 2.4.

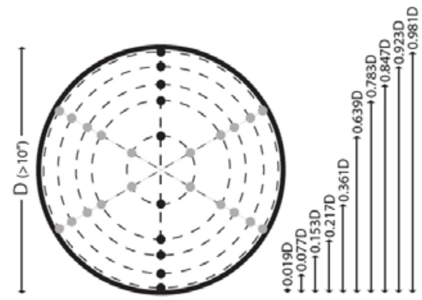

Figure 2.4: Required HWA measurement points.

To obtain acceptable results, a round duct of 14 inches in diameter calls for 10 air velocity measurements across the center of the profile. The cross section area for a 14 inch round duct is 1.068 sq. ft. and will be used to calculate the airflow volume contributions for each measurement point using Equation 2.6.

$$Volumeetric\ Flow\ Rate = A_{duct} \cdot \frac{1}{n}\sum_{i=1}^{n} v_i \qquad (2.6)$$

Using Equation 2.6 and the HWA airflow velocities obtained for the measurement series at an airflow volume of 1000 cfm, Table 2.1 results.

Table 2.1: HWA measurements for 1000 cfm test case.

Position	Velocity	Airflow
0.266	4.3	804
1.078	5.6	1048
2.142	6	1122
3.038	5.7	1066
5.054	6.1	1141
8.946	6.1	1141
10.962	5.3	991
11.858	5.4	1010
12.922	5.3	991
13.734	4.6	861
Average	**5.44**	**1018**

The HWA measurement summed up to 1018 cfm thus, only 18 cfm off the actual airflow volume and therefore has an error of approximately 1.8%. The averaged Pitot tube reference measurement was next. The integrated instruments provide the actual airflow volume directly in cfm. In comparison to the HWA measurement at this instant of time, the Pitot tube displayed an average airflow volume of 980 cfm, thus giving an error of -2.0%. The average Pitot tube uses a multitude of measured values to improve accuracy. However, if there are non-random (cyclic) high frequency fluctuations in the airflow velocities, as often is the case in HVAC systems, averaging may cause larger errors. Both electric energies into the systems are measured and monitored as poor quality electric power variations could void test accuracy. The final measurement occurs at the fan blast zone where it is desired to obtain enthalpy readings for when the electric heat is turned off and when it runs at 100%. Recorded data for an airflow volume of 400 cfm yields Table 2.2.

Table 2.2: Collected TSE data for 400 cfm series.

N	hA	hon	hoff	kW	CFM
1	12.13	13.46	22.18	19.61	400
2	12.08	13.57	21.94	19.61	406
.	-	-	-	-	-
.	-	-	-	-	-
35	11.22	13.26	22.1	19.46	389

The air power is the amount of energy delivered by the electric circuit translated into the static and velocity pressure that drive airflow volume. Air power can be quantified just by measuring the airflow volume in cfm in tandem with the total pressure. These measurements are then substituted into Equation 2.7 to determine actual air power.

$$Air\ Power = \frac{Airflow \cdot Total\ Pressure}{6356} \quad (2.7)$$

The efficiency of the fan motor system is found by dividing the air power with the system input power. The input power is measured and monitored by volt and ampere meters. It was found the efficiency of the fan motor system was in the range of 37% to 58%.

2.4. DATA ANALYSIS

A measurement system is valid if it is both accurate and precise. Validity refers to the agreement between the value of a measurement and its true value. Validity is quantified by comparing the measurements with values that are as close to the true values as possible. Poor validity degrades the precision of a single measurement, and it reduces the ability to characterize relationships between variables in descriptive studies. Reliability refers to the reproducibility of a measurement. The reliability is quantified by simply taking several measurements on the same subjects. Poor reliability degrades the precision of a single measurement and reduces the ability to track changes in measurements in the clinic or in experimental studies. Figure 2.5 visually illustrates the difference between accuracy and precision where the bull's eye to the left has a series of four that is accurate but not precise, and the one to the left is precise but not accurate.

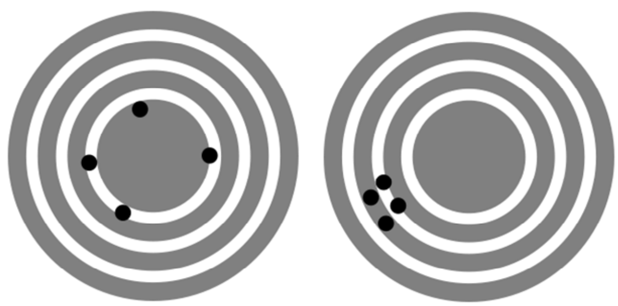

Figure 2.5: Difference between accuracy and precision.

As stated to the left it is high accuracy but low precision whereas to the right it is low accuracy but good precision. A powerful measurement method should have both high accuracy and precision. The mean or average value measures the accuracy of the method and the standard deviation is a measure of precision. The wider the spread the more volatile the series of data is. The smaller the spread the less volatile the series is. Thus, the standard deviation is a measure of dispersion away from the mean, or average, value. It tells how likely it is that a randomly chosen sample will be at or near the mean. A large standard deviation indicates a bad measurement method or device as a bias exist (non-random or directed effects caused by a factor or factors unrelated to the independent variable). Whereas the accuracy is the mean or average value of a measurement series that is mainly affected by random variability errors. Table 2.3 summarizes the reference airflows and the mean value of repeated measurements (N) with its associated standard deviation.

Table 2.3: Average and standard deviation for TSE.

Reference	Mean	Std. Dev.	N
200	216	21	28
400	398	13	35
600	579	4	37
800	817	15	36
1000	994	20	43
1200	1180	40	47
1400	1370	34	33

Using the mean, the TSE accuracy in this study is ±3% except for airflows below 300 cfm where it is ±8%. The reason for this decreased accuracy for low airflow is attributed to the modulating behavior of the fan motor's VFD. Further investigation is needed to affirm this cause. The standard deviation is used to assess the precision of TSE and it was quantified to 3% from 300 through 1200 cfm with an abnormality of precision of only 10% for low airflows (>300cfm). To manifest the resilience of TSE, the outdoor enthalpy did change as much as 7.5% during the measurement sequences. Figure 5 illustrates the accuracy of TSE, HWA and APT compared to reference values. Reference values were found using energy conversion calculations as mentioned in section 2.3. These values were also confirmed using the fan curve of the fan in the SUT and equations that describe fan performance or "fan laws" which relate pressure, fan speed, and air volume. As the data studied are of a one-dimensional nature, the graphs in the following figures have volume airflow on both the x and y axis. This is the best method to visualize accuracy and precision by comparing averaged values throughout the airflow or air velocity range.

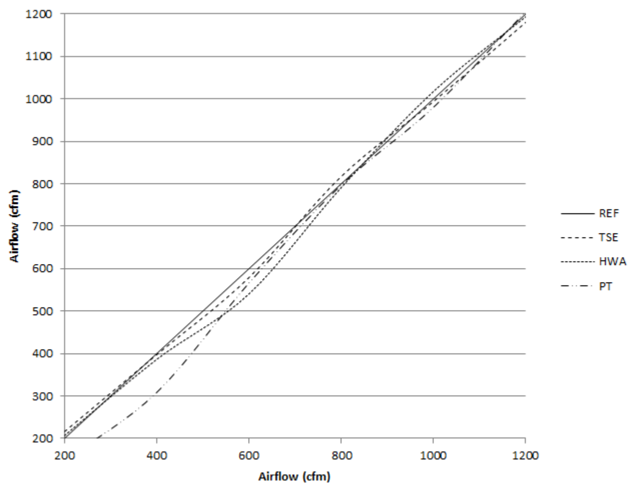

Figure 2.6: TSE, HWA and PT from 200 to 1200cfm

From the figure it can be seen that the accuracy or the systematic offset is reasonably small for TSE. The averaged Pitot tube shows substantial inaccuracy from 200 through 700cfm whereas HWA has its major inaccuracy in the range from 400cfm through 800cfm. In this laboratory study as shown in Figure 2.6 it is clear that TSE is more accurate than HWA and APT. The accuracy of TSE may be better seen in the higher resolution graphs in Figure 2.7 and 2.8.

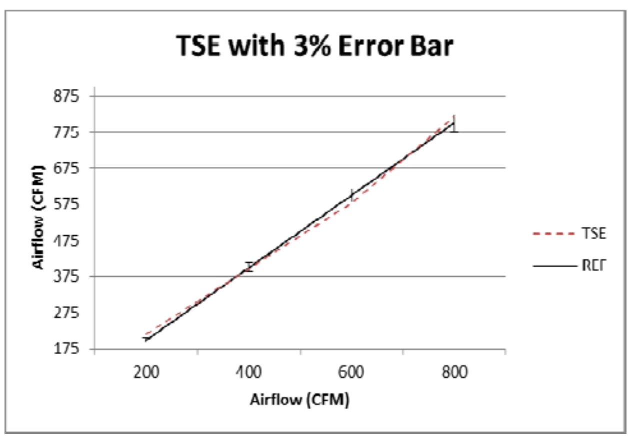

Figure 2.7: TSE compared to reference with error bars: 200cfm to 800 cfm

In Figure 2.7 the graph compares to the reference volume airflow in the range from 200cfm through 800cfm. It is here seen that the accuracy for the method breaks down at around

300cfm being equivalent to an air velocity of 281 fpm. However, from 300cfm through 800cfm the accuracy improved gradually. This can be seen as the dotted TSE line is well within the 3% error bars. Another interesting observation is the modulating behavior of the TSE graph. This may be a cyclic phenomenon that warrants more investigation. In Figure 2.8 it is the range from 800 cfm through 1400 cfm that is depictured.

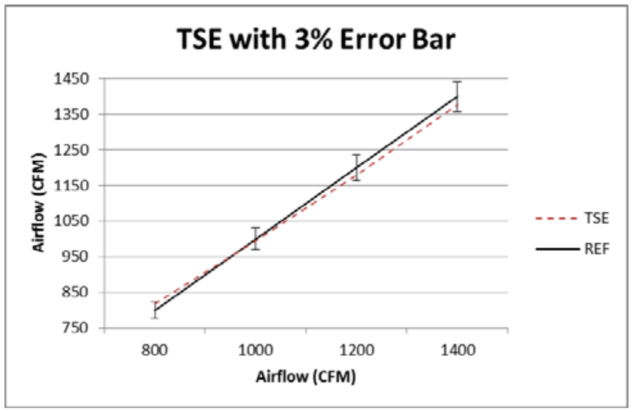

Figure 2.8: TSE compared to reference with error bars: 800cfm to 1400 cfm.

From this graph the same cyclic behavior can be seen up to 1100cfm when it ceases to exist. Above 1100cfm the graph appears to be linear with the reference airflow thus possibly becoming a systemic or bias error. Although the error percentage is approximately as calculated by the mean square method at ±3%, the data shown in these figures indicates a possible promise of an improved accuracy through calibration.

2.5 CONCLUSION

The presented research indicates by measurements that TSE has an accuracy of ±3% and a precision of 3% from 300cfm through 1200 cfm. However, both accuracy and precision have poor performance for airflows below 300cfm where these are ±8% and 10% respectively. Furthermore, TSE outperform both HWA and APT in this study. HWA becomes inaccurate and imprecise in the range from 400 to 800 cfm where the cause is most likely an increase in the Reynolds number or turbulence due to the construction of the laboratory setup. This is a known issue with HWA measurements where turbulent airflow may throw off the quality of these measurements. The observed issues with APT occur from

200 through 700 cfm where substantial discrepancies manifest themselves in both accuracy and precision. This is also expected as AABC does not recommend Pitot tube measurements in ducts with airflows less than 1000 cfm. Although the TSE has been proven to be a very accurate and precise method to measure airflow in this laboratory setting, diligence must be taken for use in the field. The underlying sensitivity in the equations used can cause substantial errors. It is also very important to properly understand the limitations of the instruments used to collect the data as substantial errors occur if measurements are not properly conducted. Several environmental conditions and system effects could also cause inaccurate measurements despite setting the system to the required testing configuration, e.g. thermal feedback from other heating and cooling zones, varying radiant heat load, building pressurization variations, fan motor RPM stability, outdoor thermal feedback, air exchange rates and so forth. However, TSE is a powerful new method that may supplement other airflow measurement techniques such as HWA, Pitot Tubes and tracer gas.

3. SOFTWARE APPLICATION JOURNAL PAPER

This chapter reports the findings related to the second objective of this research effort. The full title of this journal paper is “Time-Stepped Enthalpy Software Application” and it was submitted to International Society for Computers and Their Applications (ISCA) Journal of Computers and Applications.

3.1 ABSTRACT

Current commissioning and retro-commissioning procedures for Heating, Ventilation and Air-Conditioning (HVAC) systems are limited by existing measurement techniques and allow most systems to operate at a much lower level of efficiency than their intended design. The consequences are waste of building energy, poor indoor air quality (IAQ) and increased operating costs. Current operations measure airflow by use of Pitot tubes, hotwire anemometer or tracer-gas but these methods are often inaccurate or cost prohibitive. A newly developed measurement technique known as Time-Stepped Enthalpy (TSE) measures actual airflow volume [56], heat transfer, and other system performance parameters. It has been coupled with a very easy-to-use and scalable cloud software (SW) application.

3.2 INTRODUCTION

The average yearly energy consumption in U.S. commercial buildings is almost $51 billion, thus the return on investment for even small fractions of introduced energy efficiency is in the billions of dollars [36-37]. It is a usual practice for HVAC engineering contractors to install systems constructed of modules from various manufactures. The control system, often purchased from yet another manufacturer, links all the components together without first verifying that the system is operating at optimal efficiency before the control parameters are set. Figure 3.1 illustrates a typical pull-through HVAC system for warm climates where the fan pulls air from the building space as return air (R/A) and a portion becomes exhaust air (E/A) to meet American Society of Heating, Refrigerating and Air Conditioning

Engineers (ASHRAE) 62.1-2007 ventilation requirements. Fresh outside air (O/A) is pulled in to replace the E/A volume and will mix with the remaining R/A. The mixture of R/A and O/A has various air velocities and temperature profiles while being pulled through the filter bank by the fan. Some systems do not have E/A as it is desired to keep the air conditioned zone under positive pressure to prevent infiltration of air. The fan then pushes the airflow to be heated or cooled (determined by the thermostat setting and/or ASHRAE 55-2004) before it propagates into the ductwork as supply air (S/A). The S/A flow is delivered to the various rooms by interconnected ductwork and associated diffusers. The filtered and conditioned flow provides humidity adjusted thermal comfort to support healthy and productive environments for the building occupants.

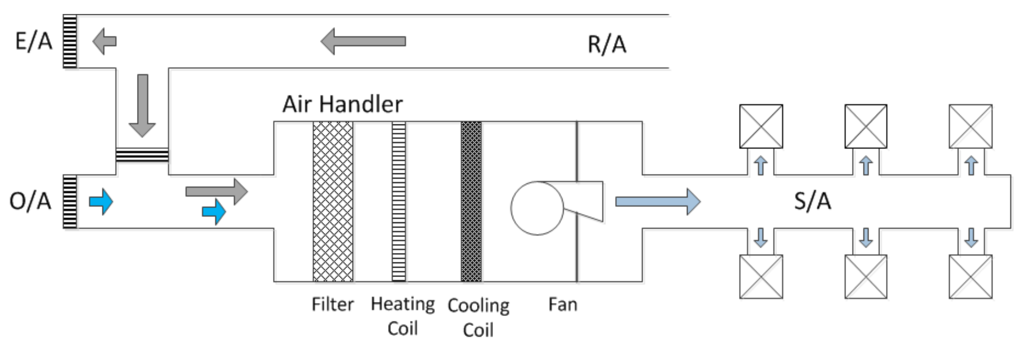

Figure 3.1: A typical pull through HVAC system.

The design cycle of a high-performance automobile relies on modeling, design analysis, module implementation and system integration. At each successive step in the process the design is subjected to rigorous performance testing. The final product is then fine-tuned to meet design objectives. Additional re-tuning of a car takes place as the mileage accumulates and parts wear out. The segment of the HVAC industry that addresses building energy efficiency is fragmented. Two dominant provider groups have emerged: the energy audit service consultant and the contractor that takes a project from start to finish. The energy auditor holds certifications such as Professional Energy Manager (PEM), Certified Energy Manager (CEM), Carbon Reduction Manager (CRM), Certified Energy Auditor (CEA), Leadership in Energy and Environmental Design (LEED), Test and Balance Engineer

(TBE) and Testing Adjusting and Balancing Bureau (TABB). Often the auditors are engineers that use ASHRAE's level 1 through 3 energy audit approaches. Contractors working the project from start to finish most often base their efforts on energy use modeling [67-69] and ensure that the project is completed as planned. Independent of the building energy conservation measures (ECM) used, the underlying fundamentals only yield a promise of savings with no actual guarantees.

3.3 TIME-STEPPED ENTHALPY METHOD

The TSE method is the first breakthrough for the building energy audit industry that allows for a comparison of actual system performance to intended design [56, 70-73]. Additional measurements enable a complete HVAC system diagnosis with information that may assist the building owner to bring the system performance towards the design intent. This revolutionary capability has been combined with all major elements of the dominate state-of-the-art energy programs being implemented in the industry. An additional enhancement is the implementation of the Environmental Protection Agency (EPA) Energy Star Score and green-house gas (GHG) accounting in tandem with the ability to estimate LEED credit potential for the building under test. These are all features made available through a very easy-to-use graphical user interface (GUI) coupled with an instantly generated independent third-party report [72-74].

3.3.1 Time-Stepped Enthalpy Method

The TSE method (patents pending and copyrighted) is the underlying technique for this study [75]. TSE is based on psychrometrics and thermodynamics and utilizes various measurements to quantify the energy released into or extracted from the system (Equation 3.1). The energy value is then substituted into the total heat formula (Equation 3.2) to calculate the airflow volume. For the hydronic cooling system investigated in this study only three measurements were required to quantify the energy extracted from the buildings airflow: water fluid flow (V_{water}) and the temperatures in and out of the heat transfer device. These measurements are substituted into the water system Equation 3.1.

$$Q_W\left(\frac{BTU}{hr}\right) = V_{water}\left(\frac{G}{\min}\right)\cdot 60\left(\frac{\min}{hr}\right)\cdot 8.33\left(\frac{lb}{G}\right)\cdot 1\left(\frac{BTU}{lb\cdot{}^\circ F}\right)\cdot \Delta T\left({}^\circ F\right) \quad (3.1)$$

The constants provided in the equations are derived for systems running at sea level and would be adjusted for those running at higher elevations. Nevertheless, the total heat (Q_W) is found by converting water flow in gallons per minute to gallons per hour multiplied by the weight of one standard gallon of water, the heat content constant, and the delta temperature of the water flow. The total heat in and out of the airstream (Q_A) transferred from the up-stream heat ladened airflow is equal to the Q_W transferred in the water fluid flow to satisfy the first two laws of thermodynamics. If a glycol mixture is used then the flow's heat content capacity is corrected to the specific gravity value. To find the actual airflow, the calculated value of Q_w is substituted for total heat Q_A in Equation 3.2. Delta enthalpy is measured by means of an enthalpy meter when the system is running at full system capacity (h_1) and then with no heat transfer present (h_2).

$$Q_A\left(\frac{BTU}{hr}\right) = V_{airflow}\left(\frac{cf}{\min}\right)\cdot 4.5\left(\frac{\min\cdot lb}{hr\cdot cf}\right)\cdot \Delta h\left(\frac{BTU}{lb}\right) \quad (3.2)$$

3.3.2 Airflow Error

The percentage error for a delta enthalpy of 1 BTU/lb, due to the inverse multivariable nature of the formula (Equation 3.2), translates into a 10% error with a variation of only 0.1 BTU/lb. From Figure 3.2 it can be seen that the expected accuracy with a 0.1 BTU/1b variance converges to 1% error when the delta enthalpy is 10 BTU/lb.

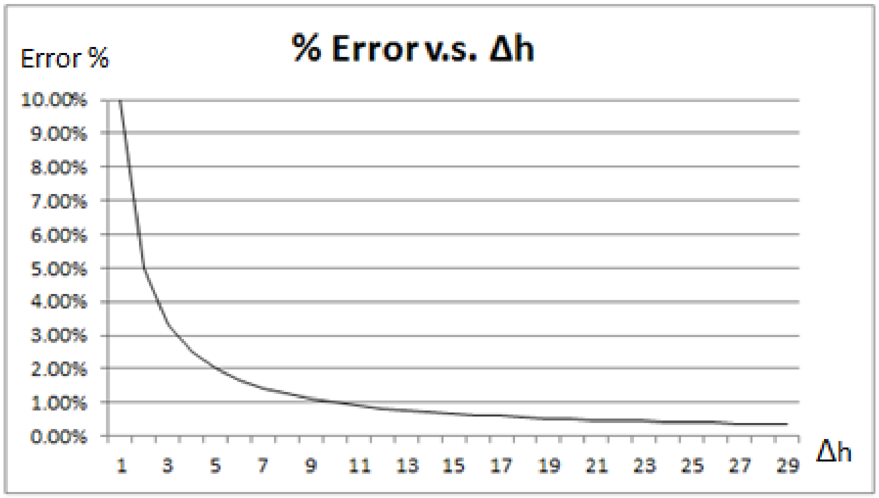

Figure 3.2: The airflow error percentage converges to less than one percent for a delta enthalpy above ten when experiencing a 0.1 variation over time.

The delta enthalpy in this study was in the range of 7.23 to 9.36 BTU/lb., which made the airflow volume calculations less susceptible to large volume swings as the enthalpy variances were small. This met the intrinsic sensitivity goal of the TSE method where the error in airflow volume should converge to one percent.

3.3.4 Wet versus Dry Cooling Coil

Static pressure (SP) measurements are used to support the diagnostic features of the TSE method. However, to properly utilize SP data for decision making it important to account for the dynamically changing environment, e.g. relative humidity and dew point. These properties change when the airstream is gradually cooled or heated and have a direct effect on SP readings. Cooling the airstream at or below its dew point makes the evaporative cooling coil accumulate precipitation as moisture is extracted from the airstream. It is for this reason that cooling coils have a drain pan and piping that drains water from the air-handler. Both types of humidity cause increased air resistance through the system due to molecular friction and decreased free area. Studies performed as part of this effort indicate that variations between wet and dry conditions range from four to seven percent. Therefore it is recommended that SP measurements be taken only when the cooling coil is dry and when the relative humidity is below 55%.

3.3.5 Repeatability Accuracy

Several repeatability studies have been performed with very good results. One study that focused on marginal enthalpy conditions in a heat-recovery system was published in an Australian Institute of Refrigeration, Air Conditioning and Heating (AIRAH) conference paper [52]. The main findings were that the method accuracy converged to ±4 percent with delta enthalpy in the range from 1.0 to 1.4 BTU/lb. This is significant as enthalpy variances of 0.1 BTU/lb would alone be the cause of an inaccuracy equal to 10% for a delta enthalpy of 1.0 BTU/lb. Furthermore, the TSE method converges to an error of less than 1% at enthalpy variances of 0.1 BTU/lb when the delta enthalpy is ten or more. The published results show that the TSE method is repeatable when test conditions are the same.

3.4 CLOUD COMPUTING

Certified building energy efficiency analysts such as the Professional Energy Managers (PEM) collect voluminous amounts of data for analysis. Weeks of effort will then be presented in a report that takes a few months to publish. The information provided in these energy estimates most often falls short of anything but a promise of savings. To the contrary, the future of building energy efficiency services will require accurate assessments of actual energy savings so decisions may be made based on information rather than a best guess or an energy modeling software that comes with intrinsic assumption errors. These requirements, in tandem with accurate measurements for repeatable results, will prevail to properly assess the health of an HVAC system in a timely manner. To accomplish this monumental task that secures quality assurance and an objective instantaneous independent third party report, various SW architectures were compared using scoreboards. A cloud computing architecture, as illustrated in Figure 3.3, quickly emerged as a viable solution that supports all these objectives in tandem with the goal of being globally scalable.

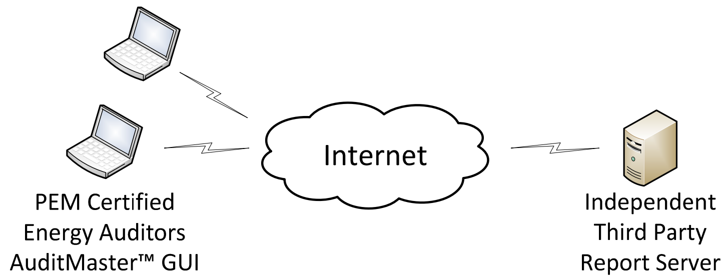

Figure 3.3: Cloud Computing System Architecture

Figure 3.3 shows the architecture for the system presented in this paper. A client application is running on a PEM certified energy auditor's laptop. The PEM collects data locally and then upload it to the application server for an instantaneously generated third party report.

3.4.1 AuditMaster™

The client SW that runs on the PEM's laptop is reconfigured depending on the HVAC system design. The application brings the PEM through a sequence of dialogs to ensure that all pertinent system information is collected along with the required measurements. Figure 3.4 depicts a few select GUI dialogs of the client SW. When pertinent data and measurements are uploaded, a computational analysis is completed within ten minutes. The generated report that is automatically e-mailed to all concerned parties lists main operational performance parameters along with the design values and actual measured results.

Figure 3.4: Select AuditMaster™ GUI pages.

The format is user friendly and easy to read. It supports quick reviews of actual systems performance compared to design intent. To comprehend the importance of this information it is very crucial to understand that a control system cannot compensate for lack of heat transfer, duct air leakage or other system deficiencies that may be occurring. Figure 3.5 depicts how results are presented in the report.

AHU 1	**Design Value**	**Actual Value**	**Deviation**	**Perf. %**
AHU 1 Airflow				
Total Fan Airflow (CFM)	20900	11637	9263	**56**
Compare to Design				
Total Fan Airflow (CFM)	20900	11637	9263	**56**
Total Outside Airflow (CFM)	2400	809	1591	**34**
Total Return Airflow (CFM)	18500	10828	7672	**59**
Duct Leakage (CFM)	0	1525	1525	**86**
Fan Motor HP/BHP	40	13.0	27.0	**33**
Thermal Performance				
Total Heat Transfer (BTU/hr)	764430	392243	372187	**52**
Sensible Heat Transfer (BTU/hr)	764430	287568	476862	**38**
Latent Heat Transfer (BTU/hr)	0	104674	104674	**N/A**
Upstream DB/RH (F/%)	-	77/48	-	-
Downstream DB/RH (F/%)	-	54/87	-	-
Coil (BTU/hr)	764430	392243	372187	**52**
Hydronic Flow (GPM)	153.0	98.1	54	**65**
Hydronic Delta Temp	10.0	8.0	2	**80**
Water Coil Carryover (FPM)	477	266	211	**56**
Ventilation				
Total Outside Airflow (CFM)	2400	809	1591	**34**
Total Return Airflow (CFM)	18500	10828	7672	**59**
Coil Cleaning				
Chiller Coil (BTU/hr)	764430	392243	-	7.84
Estimated EPA Energy Star Score	-	82%	-	-
Totals				

Figure 3.5: Intended design with actual performance.

3.4.2 Airflow

Due to variable environment and system parameters the most challenging parameter to measure in the HVAC system is airflow. As building pressure, VAV demand or damper position changes so does the airflow volume. Duct leakage reduces SP while clogged coils or filters increase SP. According to the Bernoulli equation, higher SP usually translates into less airflow. Most HVAC systems have nonlinear airflows that alternate between laminar and turbulent flows within a duct cross section. Furthermore, the temperature profiles vary throughout the system as O/A mixes with R/A and heat is added or removed from the airflow at the coils, fan motors and internal heat sources. All measurement techniques except for tracer gas and TSE rely on the existence of laminar flow for accurate measurements. Therefore most airflow measurements have errors above the rated accuracy of the device. Before executing airflow measurements the system should be set to design specifications and the variable frequency drive (VFD) set to run at maximum capacity. The AuditMaster™ SW allows the user to select and enter the measurement technique preferred: TSE, Pitot tube, hotwire anemometer or tracer gas. Entered values are used to determine the system airflow that is used to calculate other parameters such as O/A volume, duct leakage, fan motor operation, sensible heat, latent heat, coil water carryover, etc.

3.4.3 Ventilation

Ventilation is required for buildings to keep the CO_2 at acceptable levels, to expel odors, VOC, and humidity. However, ventilation is costly as conditioned air is expelled either through natural building openings, bathroom, hall, toilet or HVAC exhaust air. For this reason many HVAC systems now have heat recovery systems installed on E/A to capture a portion of the expelled energy and reapply this to the incoming unconditioned O/A. Tracer gas is known in the industry to be the most accurate method to measure actual O/A percent. Other methods to find O/A percent use CO_2 concentrations and dry bulb temperatures of O/A, R/A, and S/A. The O/A percentage is multiplied by the measured airflow to find actual O/A airflow. The implementation supports all three alternatives to determine O/A as there is limited applicability for the methods across the board. The dry bulb method will yield fairly accurate results when the fan motor is outside the airstream and all heat transfer in and out of the system is either kept constant or turned off. Using CO_2 concentrations to calculate O/A is effective when the occupancy load is held constant or if the building is not occupied at the time of test. Tracer gas will yield accurate results in both scenarios but is substantially more complex, error prone and time consuming measurement to make.

3.4.4 Airflow Heat Transfer

The heat transfer into the airflow determines the load to the chillers or boilers, maintains thermal comfort, and assists in system diagnosis. If thermal comfort is not met or if latent heat is not properly removed the occupants will either complain or they will change the thermostat settings and cause the system to use more energy. Many chillers run 24/7 to reduce the mean time between failure, to manage humidity, and to take advantage of the cooler air and lower electricity rates at night. Other chillers now also store overcooled glycol produced at night to further leverage the mentioned benefits. Therefore the main advantage of bringing the system up to intended design is to allow the chiller to always run at its optimal energy efficiency point, let VFDs work optimally by ramping down pumps and fan motors quickly and more frequently, allow the system to always provide proper thermal comfort and prevent any IAQ issues. For chillers that cycle on and off both the duty cycle and runtime at optimal energy efficiency will substantially reduce the energy usage.

3.4.5 Indoor Air Quality

A properly designed, installed and operated HVAC system will normally be energy efficient and mitigate most IAQ issues. The application compares the analysis results to the essence of ASHRAE standards 62.1-2007, 55-2004, 55.2-1999 and 91.1-2007: ventilation, thermal comfort, air filters, and building energy. It is important to recognize that dense (efficient) filters require the system to use more energy to maintain the same airflow. IAQ issues include thermal comfort, humidity, airborne particulate, ventilation, odors, and VOC. A too humid environment will cause building materials to expand, and promote mold growth, bacterial growth, musky odors and uncomfortable air. It hastens the deterioration of the HVAC system over time. Lack of proper filtering and a dusty building environment cause the inhalation of allergens and causes respiratory illness. Examples of common allergens are airborne pollen, dust mite particles, mold spores, mycotoxins, cat and dog dander and latex dust. Building dust provides harbor for bacteria that can easily spread throughout the building. Poor ventilation will increase CO_2 levels and lead to fatigue and inability to concentrate. The AuditMaster™ SW prepares an analysis that quickly identifies issues and provides data for the selection of proper remediation methods.

3.4.6 Diagnostics

A powerful feature in the AuditMaster™ SW is its diagnostic algorithm. It uses psychrometrics, various industry standards, mold characteristics, known problem areas, and actual measured performance characteristics to diagnose the HVAC system in regards to IAQ, operation, life cycle, maintenance, and energy efficiency. The SW prepares a report and recommends adjustments to the control settings so that the HVAC system performs to design requirements. An additional application for the diagnostic algorithm is to allow manufacturers to verify that delivered equipment meets specification after it has been installed in the field. Conflicts between manufacturers and contractors occur rather frequently so a system diagnosis should be required on all new construction, retrofits and renovations.

3.4.7 Energy Efficiency Measures (EEM)

The report provides a list of suggested EEM's with associated cost savings based on the diagnostic algorithm implemented. The saving models are currently based on heating and cooling degree days and are adjusted by the latent heat factor of the O/A. Common EEM's are pulley adjustments, fan motor resizing, and duct sealing, air- and system balancing. In contrary to all other performance percentages provided in Figure 5 the reporting on the coil cleaning EEM is the maximum possible improvement. The reason is that this percentage is the remaining system inefficiency not attributed other causes.

3.4.8 EPA Energy Star Score

The EPA energy star score for a building is the only recognized benchmark that compares energy efficiency in similar constructions. However, the regression models used to calculate the ENERGYSTAR score use heating and cooling degree days as a parameter and ignore the humidity in the outside air (latent heat). These models also ignore important architectural and building orientation parameters that affect the lowest possible energy consumption baseline. These effects could have been mitigated by considering the rated heating and cooling capacity for a building. Therefore, the benchmark is useful to a certain extent but susceptible to errors when comparing similar buildings across various climate zones. Since the LEED status of a building affects the resale value and the USGBC relies on the EPA Energy Star for LEED credits the LEED regression models have been implemented into the AuditMaster™ SW.

3.5 AUDITMASTER™ CASE STUDY

The data presented in this paper was obtained at a hospital in South Dakota as part of a field study using the TSE method. All data was captured using the AuditMaster™ client SW. Although the current version of the SW supports both hydronic and direct expansion cooling systems as well as hydronic and electric heating systems, the effort as shown in this case study focuses only on hydronic cooling systems. A PEM would normally not perform data logging as shown here, but it has been included to illustrate what TSE is. The instruments used in this case study were: Vaisala gm70, Testo 175-T3, GE pt878, Fluke 430 and Shortridge 88L. All were configured to log data and time stamp these in one minute steps. Outdoor enthalpy typically increases at a slow pace throughout the day as the energy delivered by the sun accumulates. The histogram labeled 1 in Figure 3.6 shows where the enthalpy increases from approximately 21.8 BTU/lb to almost 22.0 BTU/lb in 23 minutes. The histogram labeled 2 has captured the TSE method where the cooling coil runs at full capacity until it is de-energized. The timed response of the system is seen as the energy is slowly removed. It is the time difference between the energized and non-energized states of the system that yields the pertinent delta enthalpy, thus given the name TSE method. During the process it is very important that the system steady state is reached to ensure that the transient conditions do not skew the calculated results. As seen in Figure 3.6 the delta enthalpy in this case would be calculated using the values at time 13 and 21. By using Δh and Q_A the airflow may be calculated as presented in the graph in Figure 3.7.

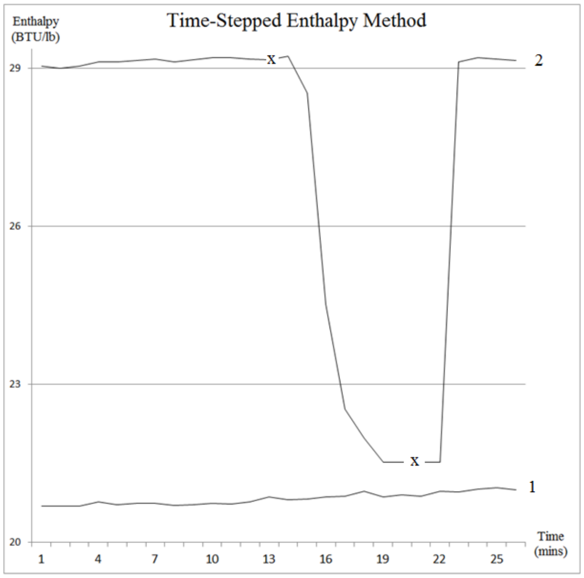

Figure 3.6: Graph 1 shows the increasing outdoor enthalpy while graph 2 plots the time-stepped enthalpy method.

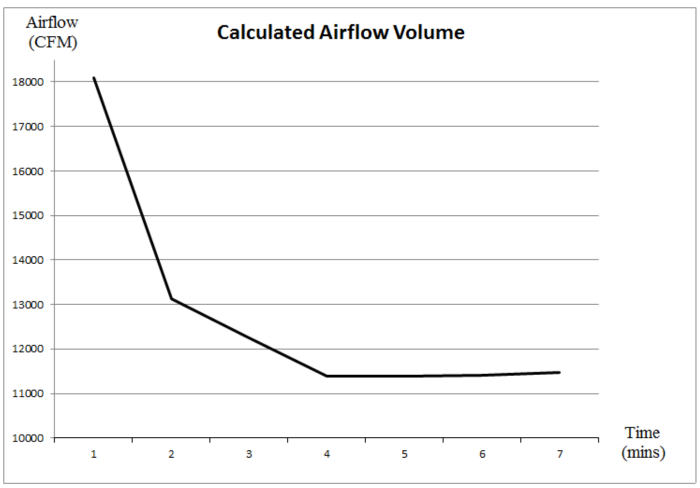

Figure 3.7: Airflow plotted as a function of time using the TSE method.

From Figure 3.7 it can be seen that the system reaches steady state after approximately six minutes. The airflow converges towards 11,642 cubic feet per minute (CFM) using these two numbers. During the walk-through phase of this study it was documented that all air-handlers were well maintained and a certified test and balance company had adjusted all air and hydronic flows for both heating and cooling systems. The PEM collected data on paper while concurrently entering it into the AuditMaster™ client SW. After the initial walk-through, which normally takes four hours for this size building, an energy audit template was generated for the system to allow design information to be entered when offsite. With a well prepared energy audit template in tandem with the identification of measurement points the actual measurement sequences could be completed on six air-handlers in less than ten hours. As the PEM completed the last measurement entry the data was uploaded and an instant retro-commissioning report was generated and e-mailed to all parties. Figure 3.5 depicts part of the report summary and Table 3.1 lists the operational performance parameters of the six air-handlers under test. From Table 3.1 it can be seen that the hydronic flows have been somewhat properly balanced. However, the airflows are substantially short of intended design partly because the fan motors do not have the appropriate horsepower.

Table 3.1 Compared performance to design for six main air-handlers at a South Dakota hospital.

AHU	Airflow	Ventilation	Heat Transfer	Hydronic Flow	Motor HP
1	56%	34%	52%	64%	33%
2	87%	18%	99%	127%	107%
3	70%	135%	66%	112%	63%
4	64%	80%	70%	113%	75%
5	56%	4%	53%	93%	42%
6	61%	3%	55%	110%	55%

Measurements using traditional Pitot tubes may yield large errors, thus the air balancer will either overstate or understate the airflow which results in improper diagnostics. For example, where the fan airflow is overstated and the test and balance contractor uses a flow hood to measure the diffusers delivered airflow, the results provided are substantially less than expected. The difference between the fan airflow and the accumulated total of diffuser airflow is assumed to be duct leakage. The normal prescriptive measure calls for sealing the air-ducts with assumed leakage while the actual issue was an improper fan airflow measurement. To keep up with the cooling demand, and to satisfy ASHRAE standard 55-2004, the control system restricts the outdoor air for all but air-handler 3. Reduced O/A will make the CO_2 levels build in violation of ASHRAE 62.1-2007. High CO_2 levels directly affect the health and performance of the buildings medical professionals. A well maintained system that cannot maintain system set-point for a myriad of reasons will sacrifice ventilation for increased thermal control without key personnel being informed. The AuditMaster™ analysis not only identifies and diagnoses issues like the one previously described but it also provides very accurate assessments on wasted energy. In addition to the direct utility cost of the wasted energy are penalties for HVAC equipment that is required to run longer due to system inefficiencies. AuditMaster™ identifies the interventions needed to resolve system problems and also provides an estimate of ECM improvement costs. Subsequently, the return on investment can be calculated enabling building owners, facility managers and building engineers to make educated decisions based on reliable information. To further elevate the utility of the AuditMaster™ SW the EPA Energy Star Score has been incorporated along with their carbon credit accounting values. This score is used to determine the LEED credit potential the building has available by utilizing the United States Green Building Council (USGBC) guidelines. Using the provided data the energy score calculated was only 50% being average in its class.

3.6 CONCLUSION

The presented research and associated SW shows that the underlying fundamentals of complex buildings, and more specifically complex HVAC systems, may be accomplished in a fashion that brings energy efficiency assessment to a new tier utilizing a cloud computing application. Not only has the AuditMaster™ application, built around the TSE method, allowed simplification of the tedious data collection and solution process, it has combined psychrometric calculations, thermodynamics, associated energy savings, ECM's, energy models and cost models. The solution enables extremely accurate results with measured and calculated parameters that are instantly used to generate a report via the cloud where accuracies converge to 1% under proper test conditions. Proper test conditions include, but are not limited to, a delta enthalpy of at least ten, dry cooling coil and relative humidity of less the 55%. The process to collect and complete a typical HVAC energy performance audit can be completed very quickly. A walk-through including the actual performance assessment for a building with up to six air-handlers serving 112,000 square feet may now be completed in two days versus several weeks without the AuditMaster™. The instant report generated is multi-faceted; it spans building commissioning and retro-commissioning, Government Energy Reduction Mandates and HVAC energy performance assessments.

4. INTERVENTION JOURNAL PAPER

This chapter reports the findings related to the third objective of this research effort. The full title of this journal paper is "Robotic Duct Sealing Enables Adequate Comfort, IAQ and Energy Efficiency" and it was submitted to International Journal of Advanced Robotic Systems.

4.1 ABSTRACT

Air leakage occurs in all Heating Ventilation and Air-Conditioning (HVAC) ducts and affects the system's ability to provide adequate thermal comfort, indoor air quality (IAQ) and energy efficient operation. This effort has a system perspective of core challenges and issues related to the variety of HVAC configurations and environmental conditions. Two new methods to quantify and seal duct leakage are also presented in tandem with two case studies that address any similarities and differences. In a hospital operating room (OR) the complaints were a lack of thermal comfort for the surgeons, return fan noise and limited cooling capacity for proper OR hygiene. At a separate location the exhaust system was required to ventilate odors and humidity from bathrooms, halls and toilets. Robotic application of long-term reliable synthetic polymer sealants mitigated all reported issues and quantified the duct leakage reductions to 16%, 10% and 39% of overall airflows.

4.2 INTRODUCTION

The importance of HVAC energy efficiency in health care, commercial and industrial buildings is continually increasing where the annual energy consumption in the United States alone exceeds $51 billion dollars [36-37]. Federal, state, county and city governments consume an additional $18 billion annually in building-related energy [38]. HVAC systems running inefficiently often cause poor IAQ and thermal comfort. For certain buildings these consequences are more economically devastating than the actual waste of energy. Reduced worker productivity and comfort may be related to latent heat buildup, mold growth, odors, high CO_2 levels, particulate contamination, noise pollution etc. [76]. The HVAC airflow is either directed through low-pressure ducts or high-pressure pipes. Friction between the airstream and the duct surface is a significant portion of the overall fan load. Usually smaller ducts are installed because of lower initial costs in terms of material, installation and space requirements, while ignoring the increased operational cost. Other considerations are shape and leakage class where round ducts are preferred because the smaller surface area per unit cross section. Less surface area translates into less leakage and thermal losses. Duct leakage typically originates at seams, joints, penetrations and service openings while poor workmanship and static pressure (SP) amplify the leakage. Multiple service openings are typically cut by duct cleaning contractors to enable manual contact vacuuming. Figure 4.1 shows a service opening left open after duct cleaning, eliminating ventilation and temperature control in six offices.

Figure 4.1: A service opening left open by a duct cleaner.

Duct leakage is a major energy efficiency and IAQ concern. Not all duct leakage is created equal, as some leakage occurs within the air-conditioned envelope and has marginal impact unless the control sensors are affected. Other times leakage goes directly to the outside or to non-air-conditioned spaces causing substantial heating and cooling energy losses. Supply air (S/A) leaks cause insufficient heating and cooling, encouraging the occupants to adjust the thermostat setting up or down as compensation. S/A leaks often disturb dust in drop-ceilings and mezzanine or entresol floors allowing varying building pressurization to spread these airborne particles. Return air (R/A) leaks may pull air from outside or from non-conditioned spaces into the duct system reducing both system efficiency and capacity. The negative pressure will also draw with it humidity, dust, mold spores, insulation fibers and other contaminants. Air leakage in toilet and bathroom exhaust systems often requires larger fans that vent unnecessary conditioned air outdoors. Inability to properly ventilate these rooms may create foul odors and mold problems in the habitat.

4.2.1 Classification

Leakage class (CL) is a benchmark that links SP to acceptable air leakage rates while ignoring contributions from connections to grilles, diffusers, and registers. To simplify the calculations of acceptable leakage, factors that correspond to CL 2, 4, 8, and 16 are provided [77]. Diligence is needed as erroneous calculations may cause unrealistic sealing goals. The CL does not quantify actual duct leakage or air exchange under natural conditions. CL is an estimate with various underlying assumptions. However, the main conclusion is that all air ductwork leaks and increasingly so when the SP gets higher in a duct. The tightness of system joints, seams, penetrations, fittings, access doors, dampers, and terminal boxes affect the air leakage rate. All ductwork should be sealed based on actual SP: Class A – (4"+) seal penetrations, transverse joints and longitudinal seams, Class B: (3"+) seal transverse joints and longitudinal seams and Class C: (2"+) seal traverse joints. Figure 4.2 depicts common duct interconnects and seams.

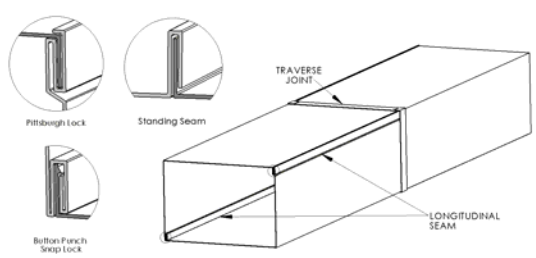

Figure 4.2: Duct joints and seams - Pittsburg Lock, Standing Seam and Button Pinch Snap Lock.

4.2.2 Duct Sealing Methods

It is now required to seal HVAC ducts for most new building construction. However, few ducts were sealed in older buildings as energy was considered low-cost. Most ducts were sealed either internally during assembly or externally after installation. In developed economies, at least half of the buildings that will be in use in 2050 have already been built [76]. This huge building inventory needs to be leakage tested and sealed using one of three currently available sealing methods: manual, aerosolized adhesives or robotics.

4.2.2.1 Manual Brush and Spray Duct Sealing

As preferred for new construction, select contractors seal ductwork externally in existing buildings using brushes and/or airless sprayers. External application works well for negative pressure ducts such as return and bathroom exhaust as the force vectors point inwards. External application is sometimes used on easily accessible and/or suspended positive pressure ductwork when visual appearance is not important. The main challenge with external application is gaining access to the ductwork due to hard ceilings, wall/floor penetrations, external duct insulation and obstructions located too close to the ductwork. It is usually cost prohibitive to expose the ductwork unless made accessible through renovations and retrofits.

4.2.2.2 Aerosolized Adhesive Duct Sealing

Aerosolized adhesive duct sealing has been around for almost two decades but has failed to gain much traction despite being the only alternative to manual sealing for an extended time period [78]. The method is implemented by temporarily capping off all supply and returns grilles and injecting pressurized small aerosol particles suspended in the airflow. As the air makes a sharp turn to exit through a leak, the particles collide with and adhere to the edges to seal leakage openings up to 5/8 inches across. There are no studies on the long-term integrity of the adhesive seals but the manufacturer warranty is limited to only three (3) years for commercial buildings [78]. As the method utilizes “blind” application, fire-dampers, duct sensors, reheat coils, variable air volume (VAV) boxes and duct walls will be contaminated with adhesive. Some issues may be mitigated by taping and covering devices ahead of injecting the adhesive but the effect of the deposited glue on the duct walls and other components along with the SP and hygiene of the system is unanswered. A quick laboratory test showed that dust quickly accumulates onto the glue and promotes an almost impossible to remove particulate as shown in Figure 4.3.

Figure 4.3: Source removal is impossible after adhesive sealing.

A major concern is therefore the ability for ductwork sealed with aerosolized adhesives to pass the industry's cleanliness protocols using the NADCA vacuum test [79] or by an industrial hygienist's particle counting procedure. The residual glue on the duct walls has also been reported to prevent duct cleaning as tethers and hoses adhere to the surface preventing mobility. All organic particulate is mold food requiring only spores and humidity to enable growth. Humidity exists in a duct system as grains of moisture, condensate, water leakage or blow through water; all of which are common occurrences in these types of systems. If duct cleaning is attempted, the source removal process consist of a combination of vacuum suction of up to 110" w.g., aggressive rotating high torque driven brushes or compressed air driven whips certainly will affect the integrity of the adhesive seals. All of these could possibly bring the duct leakage back to status quo. Some duct cleaners also report that the adhesive breaks down and peels off in flakes after a few years. Sealing ducts under positive pressure but operating under negative pressure (e.g. return and exhaust) creates an additional concern. As the force vectors are reversed after sealing, the integrity of the seal is questionable.

4.2.2.3 Robotic Duct Sealing

Robotic duct sealing is a new application of a mature technology. A quick training program is required for new operators but the user threshold is low. Application of sealants depends on the sealing class: A, B or C. Different robot equipment is needed for vertical and horizontal ducts. Varying ductwork sizes require robots of different dimensions or adjustable spray attachments for nozzles to reach the application surface. A duct sealing robot system consists of an airless sprayer, a robot and a sprayer attachment as seen in Figure 4.4.

Figure 4.4: A typical HVAC robotic system setup.

The sealant is pumped by the airless sprayer from a 5-gallon pail to the sprayer applicator attached to the robot. It is the operator's workmanship using video feedback that prevents sealant contamination on fire dampers, sensors, reheat coils and VAV boxes. Sealing around these devices and any access holes should be done manually. Other known challenges for robotic sealing are proper positioning of the spray nozzles, the ability to concurrently cross spray traverse joints, and to fill gaps wider than ¼ inch. New synthetic polymers have a smooth non-tacky finish with low friction coefficients so the SP improves and less particulate buildup occurs in the ductwork. These new generations of duct sealants do not crack in comparison to early market ones made of watered down high-viscosity sealants.

4.2.3 Duct Sealing Verification Testing

Leakage checks on all new and existing HVAC ducts are highly recommended as all installed ducts leak air. Depending on the contractor's background, certain leakage measurement methods are preferred to others independent of accuracy or applicability. Most measurement methods are not intrinsically very accurate and not even feasible to use because of the nature of the applied techniques [80-82]. Commonly used methods are smoke, static pressure, tracer gas, Pitot tubes, hot wire anemometer (HWA), and Time-Stepped Enthalpy (TSE) testing. All methods but smoke and SP testing require air flow hooding of all registers during normal air handler operation to determine the total air flow delivered. Errors for passive flow hoods may be substantial due to added airflow resistance, stratified flows and positioning whereas active flow hood measurements are typically within ±3% [83]. In most low and medium pressure HVAC environments the airflow velocity pressure (VP) is small compared to the SP. However, for high pressure ducts VP can exceed the SP as shown in Figure 4.5.

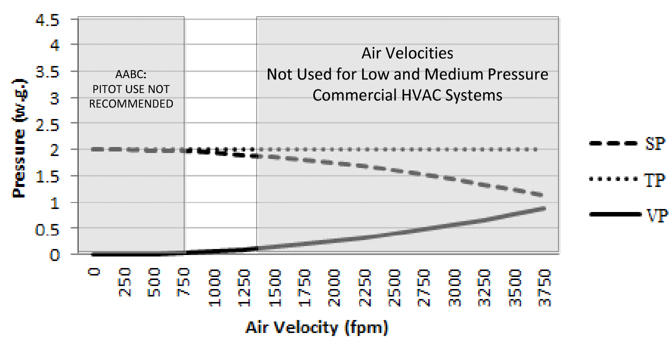

Figure 4.5: TP, SP and VP as air velocity increase.

4.2.3.1 Orifice Testing

An orifice leakage test pressurizes the duct up to its pressure class rating and measures the airflow required to sustain this pressure [84]. The measured pressure drop over the orifice restriction is proportional to air volume leakage at this SP. Although an orifice measurement is accurate, the assumptions that that dampers are airtight, SP operates at its class rating and is homogeneous for the section of ductwork under test are substantial error sources. Furthermore, it is often impractical or impossible to test the entire system. Usually individual sections are tested, and the result from one section is sometimes applied to estimate the total duct leakage – this practice is not recommended. In an attempt to determine where the leakage originates, smoke bombs are applied with varying success. Figure 4.6 depicts SP variations throughout a HVAC system to show why an orifice SP test weigh leakage upstream less than downstream.

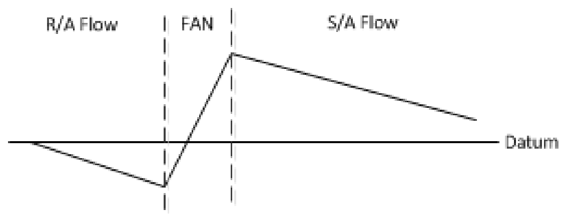

Figure 4.6: SP profile of an operational HVAC system.

4.2.3.2 Pitot Tubes

The Pitot tube measures a fluid velocity by converting the kinetic energy of the flow into potential energy [63-65]. The conversion takes place at the stagnation point (z1), located at the Pitot tube entrance. The stagnation pressure (Ps) is measured and comparing to the flow's SP (p2) using a differential manometer.

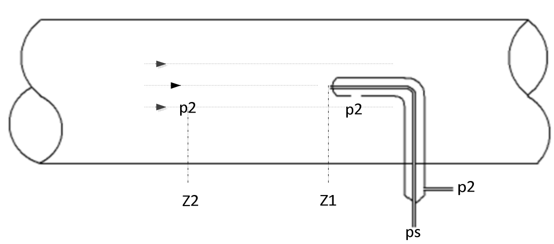

Figure 4.7: Cross-section of a Typical Pitot Static Tube

The Bernoulli equation states that the energy along a streamline is constant for incompressible flows at speeds below 30% of sonic velocity. Therefore, the summation of the velocity, static and hydrostatic pressures is constant as shown in Figure 4.5. By using Figure 4.7 and evaluate two different points along a streamline, Z_1 and Z_2, the Bernoulli yields Equation 4.1.

$$\frac{1}{2}\rho v_{z2}^2 + p_{z2} + yh_{z2} = \frac{1}{2}\rho v_{z1}^2 + p_{z1} + yh_{z1} \quad (4.1)$$

The VP is 0.5 times the fluid density (ϱ) multiplied by the square of the flow velocity (v) added to the static pressure (P_Z). The hydrostatic pressure is described as the specific weight (y) times the elevation height (h). For small distances the hydrostatic pressures cancel (yh_{Z2}=yh_{Z1}) and at the stagnation point (Z_1) the fluid velocity is zero (v_{Z1} = 0). Therefore, by applying more trivial naming conventions Equation 1 reduces to Equation 4.2. The flow velocity (v) is found by the square root of the differential manometer output times two divided with the fluid density.

$$v = \sqrt{\frac{2 \cdot (p_{stagnation} - p_{static})}{\rho}} \quad (4.2)$$

Pitot tube measurements require laminar and homogenous flows. However, these conditions seldom exist in HVAC ductwork resulting in large errors. To compensate for varying velocity profiles a number of (n) air velocity (v_i) measurements are averaged and multiplied to the cross section (A_{duct}) to find the volumetric flow rate as shown in Equation 4.3.

$$Volumeetric\ Flow\ Rate = A_{duct} \cdot \frac{1}{n}\sum_{i=1}^{n} v_i \quad (4.3)$$

Two measurement point distributions are used with Equation 4.3 in an attempt to take airflow friction into account: Centroids of Equal Areas or Log-Tchebycheff. The Log-Tchebycheff point distribution is illustrated in Figure 4.8.

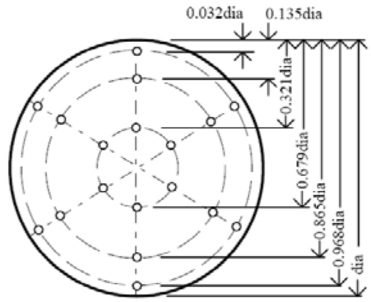

Figure 4.8: Measuring points for log-Tchebycheff traversing.

The ISO Standard 3966 describes the accepted method for traversing rectangular ducts with a minimum of 25 points. The uncertainty analysis also mentions a number of systematic and random errors: velocity fluctuations, turbulence (Reynolds number), inclination, positioning and flow rate calculations. Even the use of the Log-Tchebycheff distribution is questionable because HVAC ducts have numerous fittings and flow disturbances.

4.2.3.3 Hot-Wire Anemometer

Hot wire anemometers (HWA) use a very fine electrically heated wire cooled by a fluid flow [61-62]. The convective heat transfer changes the electrical resistance of platinum and tungsten. The voltage output from an HWA is the result of an electronic circuit within the device that holds either the current, voltage or temperature constant. The differential equation describing the physical phenomena has six terms: conduction loss, heating energy, thermal energy storage, forced convection, natural convection loss, and radiation. Natural convection and radiation losses are negligible so a single wire HWA may be realized by Equation 4.4.

$$-k_w A_w \frac{\partial^2 T_w}{\partial x^2} + \frac{I^2 \chi_w}{A_w} - \pi d_w h(T_w - T_a) - \rho_w c_w A_w \frac{\partial T_w}{\partial t} = 0 \quad (4.4)$$

The parameters for this expression are: wire conduction constant (k_w), wire cross-section (A_w), wire temperature (T_w), wire length (x), electric current (I_w), wire resistivity (χ_w), wire diameter (d_w), heat transfer coefficient (h), fluid temperature (T_a), wire material density (ρ_w), specific heat wire (c_w) and time (t). Figure 4.9 is the schematics for a constant temperature Wheatstone bridge.

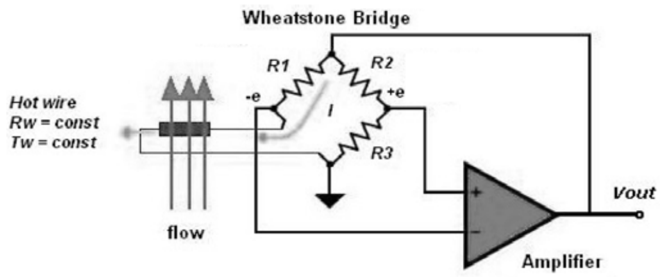

Figure 4.9: A constant temperature HWA schematics.

The Wheatstone bridge will keep the electric current proportional to the mass flow velocity. The King's law shown in Equation 4.5 is an empirical solution to Equation 4.4 and the mathematical expression of Figure 4.9. The voltage drop (V) is an expression of the fluid velocity (v). The constants A, B and the exponent n are empirically determined and are ambient specific. Assumptions are uniform temperature across wire length, uniform and normal air flows across the wire, constant fluid density and temperature, and a Reynolds number less than 140.

$$V^2 = (T_w - T_a)(A + Bv^n) \qquad (4.5)$$

A and B are found by measuring the voltage (V) obtained for a number of known flow velocities and determined by the least squares fits. The exponent (n) is assumed to be in the range of 0.45 to 0.5. However, direct calibration is recommended for all HWA devices and the ASTM Standard D 3464-75 specifies to average measurements from 4 to 20 sampling points due to the variable velocities in a cross section.

4.2.3.4 Tracer Gas

Tracer gas testing uses a gaseous taggant that is dispersed into the moving air before the concentration is measured to find the air volume using Formula 4.7 or 4.8 [85-87]. The gasses used are typically odorless, non-reactive, non-toxic, colorless, and detectable by recognized techniques. Helium, Hydrogen and CO_2 use increase in popularity as the Koyto protocol aims to phase out halon tracer gasses. A downside with CO_2 is that people produce it and plants absorb it; both may affect the measurement accuracies. Tracer gas can accurately measure any gas flow rates in ducts, stacks, or pipes and therefore often used to calibrate less accurate measurement methods. Until now tracer gas has been the only method capable of accurately measuring the amount of outside air supplied to a building under actual operating conditions. The theory for tracer gas originates in the mass balance equation.

$$\frac{dm}{dt}_{TG}^{Upstream} + \frac{dm}{dt}_{TG}^{Injection} = \frac{dm}{dt}_{TG}^{Downstream} \tag{4.6}$$

Equation 4.6 shows the tracer gas mass rate upstream added to the injected mass rate, equals the combined mass rate downstream. Substituting a mass flow rate equal to the density times the volumetric flow before solving for volumetric flow (Q_{AIR}) yields Equation 4.8.

$$Q_{AIR} = \frac{\frac{dm}{dt}_{TG}^{Injection}}{\rho_{TR} \cdot \left(c^{Downstream} - c^{Upstream}\right)} \tag{4.7}$$

By integrating Equation 4.7 as a function of time (t), the airflow may be determined using concentrations measured as parts per million (ppm). Equation 4.8 results if the upstream concentration changes linearly during injection and a trapezoid formula is used to integrate over n segments where i=1… n while both pre- and post- background concentrations are held constant.

$$Q_{AIR} = \frac{m_{TR} \cdot 10^6}{\rho_{TR} \cdot T \left[\begin{array}{c} \frac{1}{2n} \sum_{i=1}^{n} \left(c_{i-1}^{Downstream} + C_i^{Downstream} \right) \\ - \frac{1}{2} \cdot \left(\frac{1}{m} \sum_{j=1}^{m} c_j^{pre} + \frac{1}{L} \sum_{k=1}^{L} c_k^{Post} \right) \end{array} \right]} \tag{4.8}$$

Despite this simplification the method requires extensive knowledge and training to allow accurate and repeatable results. Errors often occur because of limited air and tracer gas mixing and a short measurement time-window. Therefore, this expensive measurement setup often takes substantial time to get accurate.

4.2.3.5 Time-Stepped Enthalpy

TSE is a new method [51-57] to accurately measure airflow in HVAC systems. TSE use psychrometric and thermodynamic measurements to quantify the energy released into or extracted from an HVAC system. Only three measurements are required for a hydronic cooling system to quantify the energy extracted from the building's HVAC airflow: fluid volume flow (V_{fluid}) and the two temperatures in and out of the heat transfer device. These measurements are substituted into the hydronic system Equation 4.9 to quantify the total heat.

$$Q_F\left(\frac{BTU}{hr}\right) = V_{fluid}\left(\frac{G}{\min}\right)\cdot 60\left(\frac{\min}{hr}\right)\cdot 8.33\left(\frac{lb}{G}\right)\cdot 1\left(\frac{BTU}{lb\cdot {}^\circ F}\right)\cdot \Delta T({}^\circ F) \quad (4.9)$$

The constants provided in the equation are derived for systems running at sea level and therefore need to be corrected for higher elevations. The total heat Q_F is found by first converting the fluid flow in gallons per minute into gallons per hour which is multiplied with the weight of one standard gallon of water times the heat content variable times the delta temperature of the fluid flow. The total heat Q_A transferred into or extracted from the airflow is equal to the Q_F transferred by the fluid flow to satisfy the first two laws of thermodynamics. If a glycol mixture is used then the flow's heat content capacity is corrected to the specific gravity value. To find the airflow Q_F is substituted as the total heat Q_A into Equation 4.10 along with delta enthalpy of the airflow: h_1 is measured when the system is running at full capacity and h_2 is measured when no heat transfer is taking place. All constant heat contributions will automatically cancel out as one of the two measured parameters is a baseline.

$$Q_A\left(\frac{BTU}{hr}\right) = V_{airflow}\left(\frac{cf}{\min}\right)\cdot 4.5\left(\frac{\min\cdot lb}{hr\cdot cf}\right)\cdot \Delta h\left(\frac{BTU}{lb}\right) \qquad (4.10)$$

4.2.3.6 Measurement Techniques Compared

Table 4.1 summarizes the main parameters of the measurement methods. The table shows that both tracer gas and TSE provides sufficient accuracy for duct leakage assessments in all HVAC airflows. Pitot tubes and HWA require non-frequent laminar flows for good accuracy making measurements susceptible to large errors. Orifices SP is intrinsically a very accurate method but results are often inaccurate (as explained in section 4.2.3.1).

Table 4.1: Comparison of measurement methods [55,65].

Method	Accuracy	Time	Flow	Calibration
Smoke	N/A	60	Any	N/A
SP	Low	60	No	No
Pitot	±10%	60	Laminar	Yes
HWA	±10%	60	Laminar	Yes
Tracer	±1%	180	Any	No
TSE	±3%	15	Any	No

4.3 HVAC ROBOTIC DUCT SEALING

Robotic inspection has been used for many decades to visualize difficult to reach places. As inspection robots became better other payloads were added to execute various functions and tasks. Duct cleaning was implemented early using either rotating brushes or compressed air whips. Newer HVAC robots allow more advanced airless sprayer attachments to apply sealants in ductwork. Existing systems may operate horizontally in ducts from five inches up to five feet in diameter and vertically from eight inches up to nine feet in diameter. In the presented efforts both horizontal and vertical HVAC robots were used.

4.3.1 HVAC Robotics

A horizontal HVAC robot was utilized for the S/A and R/A case studies. The system was configured to seal ducts from 8 inches up to 30 inches in diameter and is shown in Figure 4.4.

The combined cleaning and sealing process often requires the fan motor to be shut off while a HEPA filtered negative air machine creates an airflow that extract dust and debris from the ductwork. The same airflow prevents camera overspray when applying sealants while driving in reverse.

4.3.2 Self-Centering Spray-Head

High quality duct sealants are expensive synthetic polymers and it is desired to minimize the amount of applied material. This motivation pushed the development of systems that automatically self-center in the ducts by feeding sensor data to a proportional-integral-derivative (PID) controller. Another sensor measures the height of the duct and uses this input to position the rotating head in the three dimensional center of the duct. For round ducts the spraying application takes place with a constant rotation. For rectangular or oval ducts a speed profile for even application may be desired to allow quick and effective robotic sealing. This feature is very valuable for class A sealing where the robot applies sealant everywhere in the ductwork. For class B sealing the nozzles are stationary to apply sealants at the longitudinal seams first, followed by a second pass with rotating application for traverse joints. Class C only uses the rotating application at all traverse joints. A similar approach for self-centering is used for vertical duct robots although the propulsion system is different. Figure 10 depicts finished Class A sealing using robotic application.

Figure 4.10: Class A robotic application of sealant.

4.3.3 Manual Sealing of Near Openings

Robotic sealing of ducts sometimes requires manual application in certain places. It is especially important where service openings are made into the system, by fire-dampers, VAV and reheat coils.

4.4 DATA COLLECTION/ANALYSIS

The two case studies presented were selected from a pool of studies to show the versatility of HVAC robotic duct sealing and to pinpoint similarities and differences between duct sealing of exhaust, R/A, and S/A systems. In the selected cases good conditions existed for accurate airflow measurements and all instruments were rated at ±3% or better. When applicable, flow hood measurements were applied to quantify the amount of leakage before and after duct sealing. A retrofit project was initiated after the medical staff had complained about inadequate system performance in OR. After a new higher-capacity direct expansion (DX) system with a desiccant wheel dehumidifier was put in service the complaints continued. Measurements indicated duct leakage as the cause of the problems and sealing of the R/A and S/A ductwork was prescribed. The results from the bathroom, hall and toilet exhaust system presented, were obtained at a separate location. Issues that triggered this effort were insufficient ventilation to meet building codes and to prevent odors and microbial growth.

4.4.1 Return Duct System

The R/A volumes were measured to 4796 cfm at the air-handler unit (AHU) and 3701 cfm at the intake, thus a duct leakage of 1095 cfm. After sealing the AHU air volume remained the same while increasing to 4450 cfm at the intake. The difference of 346 cfm is the remaining duct leakage and show a reduction in the unaccounted R/A of 749 cfm or 16%. It is assumed that the remaining leakage is from the return isolation damper separating the new and old systems as this damper was not sealed. Further reductions may be accomplished by installing a bubble damper in place of the existing leaking one. As the R/A leaks pulled in warm air from the boiler room the temperature difference before sealing was 6.4F and only 0.8F afterwards. Less negative pressure airflow also drew less dust, mold spores, insulation fibers and other contaminants into the R/A system, thus reducing the filter loads.

4.4.2 Supply Duct System

The S/A volumes were 5560 cfm at the AHU and 4515 cfm at registers thus a duct leakage of 1050 cfm. After sealing the AHU air volume remained the same while increasing to 5065 cfm at the registers. The difference of 550 cfm is the remaining duct leakage and show a reduction in the unaccounted S/A of 550 cfm or 10%. The remaining leakage of 495 cfm is attributed to the unsealed return isolation damper. Therefore, to further reduce the leakage it is recommended that a bubble damper is installed in place of the existing leaking one. The combined effect of sealing 1244 cfm air leakage in the R/A and S/A improved system capacity to maintain the required 60-62 °F temperature in the OR even when the return fan was off.

4.4.3 Exhaust Duct System

Residential building codes call for minimum ventilation rates to rid odors and humidity from showers, baths and toilets. Continuous ventilation is typically 20 cfm while it is 50 cfm for intermittent operation. The continuous ventilating system was initially exhausting 1,582 cfm while the 40 ventilation units were supposed to draw approximately 800 cfm. The system was unbalanced and only 621 cfm was ventilated from bathrooms, halls and toilets and thus 961 cfm of conditioned air was unintentionally pulled out from the building. Before sealing, 20 cfm constant airflow regulators were installed in all the exhaust vents. A vertical robot was used to seal the main duct by first using backdrop fillers for the large gaps and a final seal of synthetic polymer. The ventilation dropped to 974 cfm, a reduction of 608 cfm or 39%. The affinity laws were used to size a new smaller fan motor to be installed. The smaller fan motor and the reduction of conditioned air exhausted will quickly finance the IAQ benefits obtained.

4.5 CONCLUSION

The presented research show that robotic application of reliable long-term synthetic polymer sealants mitigated thermal comfort, exhaust fan noise pollution, cooling capacity, OR hygiene, odor ventilation and humidity ventilation. Quantitative analysis using different measurement techniques at ±3% or better accuracy quantified the duct leakage reductions of approximately 16%, 10% and 39% of overall airflow. It was found that duct sealing not only improves energy efficiency and reduces system load but also has a substantial impact on the indoor environment. Although the verification and validation methods presented in the effort rely on Pitot tubes, HWA and flow hoods, it is important to know how to make accurate measurements using these instruments. Tracer gas and TSE measurements are both good options to assess HVAC systems and to calibrate other non-accurate methods. Manual or robotic application of properly designed sealants is the only recommended duct sealing approach for hospitals or health care facilities to facilitate proper duct hygiene and long-term seal envelope integrity.

5. CONCLUSIONS

In this research the impact of the HVAC system on nosocomial infections in healthcare facilities has been investigated. As the HVAC system impacts many aspects of the building environment such as building pressurization, indoor air quality, people/equipment movement, and humidity control, the scope of this research encompass these and more parameters. It is a known fact that germs are spread by dust and particles in hospitals various ways such as through the HVAC system, building pressurization, elevators and moving people. In the U.S. alone approximately 2,000,000 patients develop nosocomial infections annually where 100,000 of these patients die. It is estimated that almost 10,000 of these deaths is related to *Aspergillus* spores spread by dust and particles. The CDC also estimates that 25% of nosocomial infections occur because of medical staff mistakes and studies show that poor thermal comfort, limited ventilation and lack of humidity control cause medical staff to make mistakes.

To reduce nosocomial infections related to the indoor environment it is necessary to view everything within the building envelope as well as the zones around exhaust and fresh air intakes as part of the HVAC system. The HVAC system can be viewed as the lungs of the building and proper performance at the intended design specifications usually prevents the proliferation of biological pollutants. Mold, mildew, and similar pollutants thrive in moist, humid environments. Proper functioning bathroom exhaust fans will further help prevent the proliferation of mold and other pathogens by driving warm, moist air out. Pollutants may also originate from past or current construction, remodeling or system updates. People continually bring in pollutants on their clothing as does the HVAC system's fresh air intake and any opening in the building envelope with negative pressure, compared to the ambient outdoor pressure. Building pressurization varies and depends on indoor and outdoor temperatures, wind and wind-direction and internal stack conditions. All buildings should therefore have a system to always ensure a slight positive pressure to prevent infiltration of outside air. Proper filtering of the incoming air and change to new sterile clothing would be good procedures. Any procedure that is prescribed to remove dust and particles, being duct cleaning, construction dust management or dust downs, large truck mount negative air machines should NEVER be used as the replacement air will bring in outdoor contaminants. All negative air systems should be standalone HEPA negative air machines that use air from within the building envelope. A cleaner indoor environment resulting in the reduction of dust, particles and humidity within the building envelope would also reduce the use of disinfectants that are often blamed for making bacteria stronger and more immune to treatments.

In tandem with better management of influx of dust and particles into the envelope it is equally important to remove any existing reservoir of dust and particles within. Furthermore, an additional precaution is to limit agitation of any such reservoirs by controlling building pressurization, manage elevator pressure gradients and to properly seal all duct work using sealants with a smooth non-sticky finish. Care must also be taken to ensure that the HVAC air filters are kept fully operational with no pass through air and that all HEPA filters are installed in dry zones and with associated ultraviolet light to kill any germs. Wet HEPA filters sometimes compromise the filter performance enabling a free passage for germs to circulate within the building envelope. Therefore, a good practice would be to supplement system installed HEPA filters with standalone HEPA machines placed at strategic locations so any airborne dust and particles are captured. Building codes for health care facilities should all adopt the requirement to have an overhead supply air and a floor return to utilize gravity to bring any dust and particles in the room down and into the return system to prevent these from floating around over time in the heat plumes and eventually allowing deposits onto any horizontal surfaces.

The three journal papers that make up the core of this research aim to fill the gaps in technology and innovation that enable the implementation of new best practices for dust and particle influx, elimination of existing reservoirs, minimizing agitation and management of indoor humidity to prevent proliferation of germs. The first objective was to devise a new method to accurately measure volume airflow in the health care facilities. For this purpose TSE was invented and studied to yield accuracy and precision of ±3% as described in chapter 2. The importance of this objective was to measure a fundamental parameter needed compare actual HVAC performance with intended design to allow for system optimizing in regards to thermal comfort, proper ventilation and humidity control. There are all important parameters to reduce the probability of medical staff mistakes and proliferation of germs. The second objective was to develop computational analysis software that could properly diagnose the HVAC system in regards to humidity, dust agitation sources, filter performance and indoor air quality as described in chapter 3. Appendix A provides a user dialog sequence for an analysis of a South Dakota hospital and the associated instantly generated report is included as Appendix B. The third and final objective was to fill any gaps in capabilities in remediation of contaminants or agitation that may occur. Chapter 4 describes the current shortcomings in the ability to determine total duct leakage and current methods applied to seal detected air leakage. A method to properly measure actual total duct leakage was devised in tandem with a new robotic applicator for a no sticky sealant to be applied internally in the ductwork. Such solution was required as manual application struggle with access to the seams and joints while the aerosolized glue method requires the replacement of the ductwork in health care facilities as the sticky surfaces would permanently create a dirty environment that would harbor dust and particles supporting proliferation of germs within the building envelope.

There are many other pathways to nosocomial infections such as bloodstream infection, ventilator-associated pneumonia, urinary tract infection, surgical site infection, etc., that may or may not be related to the indoor environment. Also, many people do not grasp that most bacteria may already exist on the patient's body including some that either are antibiotic or antimicrobial resistant and that it is the treatment that compromises the immune system to trigger opportunistic infections by these. Thus it is not necessarily poor hygiene or factors at the hospital that cause the infection. Rather it is the weakened natural immune system that allows the already present bacteria to flourish. Research is therefore currently focusing (with very little success) on possible vaccines for the most common problem germs. A more radical bacteriophages therapy pioneered by George Eliava early last century has resurfaced. Prior to the discovery and widespread use of antibiotics, it was suggested that bacterial infections could be prevented and/or treated by the administration of bacteriophages. A bacteriophage is a virus that infects and replicates within bacteria. Time will show if this approach may advance into a viable infection treatment option without risking new viruses that start infect healthy cells in humans or animals.

Another subject that may be addressed as a final comment is in regards to that the building envelope always should have a little positive pressure to prevent infiltration of outside contaminants. This creates an issue when patients infected with one of more than two-dozen types of bacteria now resistant to one or more antibiotics. To protect the surroundings, rooms from the isolated ones should be kept under negative pressure to prevent any potential spread. However, with negative pressure other germs may propagate from the outside, making the isolated patient even more vulnerable with the potential for new infections. Therefore, care should be taken when isolating patients and such patients should be placed in a wing with a separate HVAC system or the room should be kept under negative pressure to protect the surroundings while the patient should be placed in a positive pressure containment perimeter to prevent additional infectious disease exposure.

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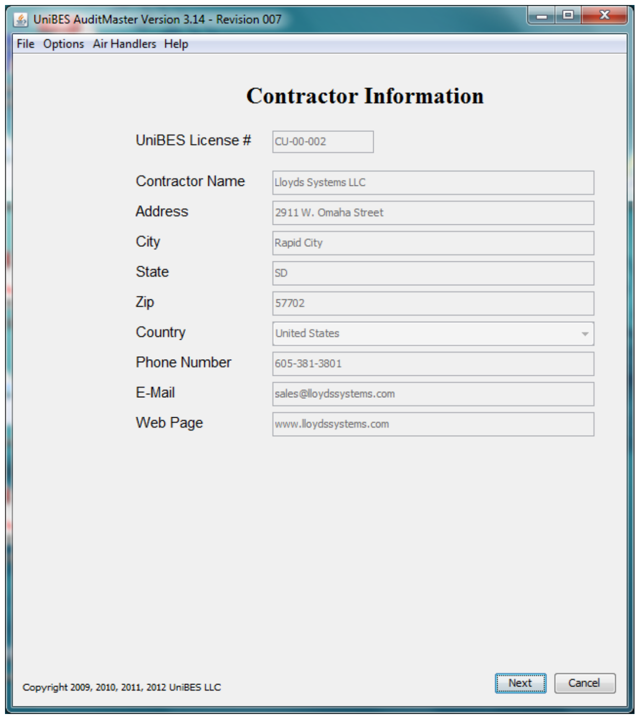
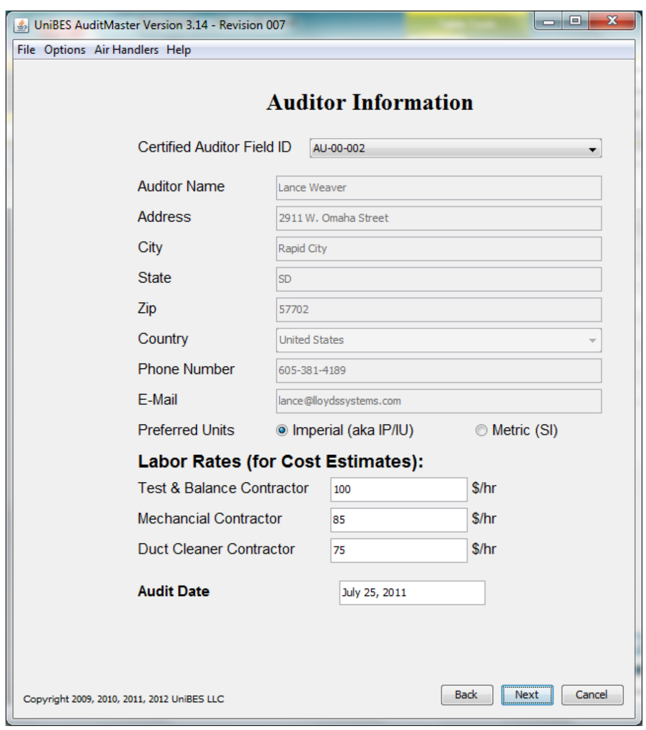
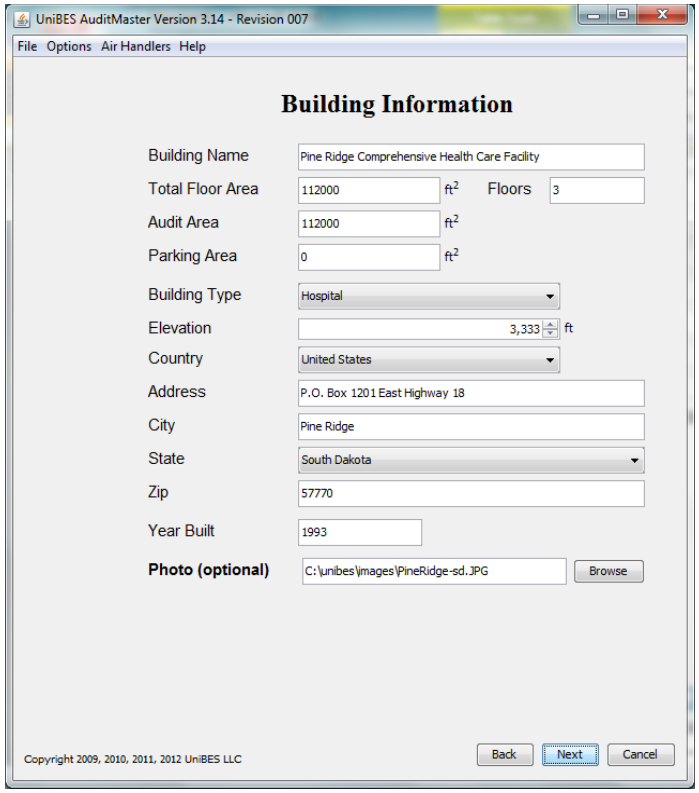
Appendices

APPENDIX A: AUDITMASTER™ USER DIALOGS






UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Building Information
Building Name
Pine Ridge Comprehensive Health Care Facility
Total Floor Area
112000
ft²
Floors
3
Audit Area
112000
ft²
Parking Area
0
ft²
Building Type
Hospital
Elevation
3,333
ft
Country
United States
Address
P.O. Box 1201 East Highway 18
City
Pine Ridge
State
South Dakota
Zip
57770
Year Built
1993
Photo (optional)
C:\unibes\images\PineRidge-sd.JPG
Browse
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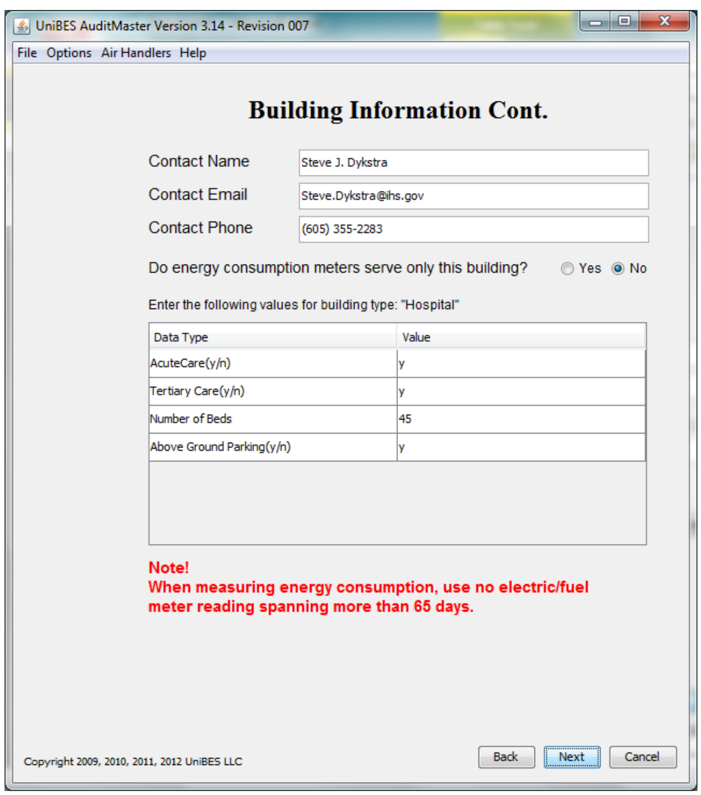
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Building Information Cont.
Contact Name
Steve J. Dykstra
Contact Email
Steve.Dykstra@ihs.gov
Contact Phone
(605) 355-2283
Do energy consumption meters serve only this building?
Yes
No
Enter the following values for building type: "Hospital"
Data Type
Value
AcuteCare(y/n)
y
Tertiary Care(y/n)
y
Number of Beds
45
Above Ground Parking(y/n)
y
Note!
When measuring energy consumption, use no electric/fuel meter reading spanning more than 65 days.
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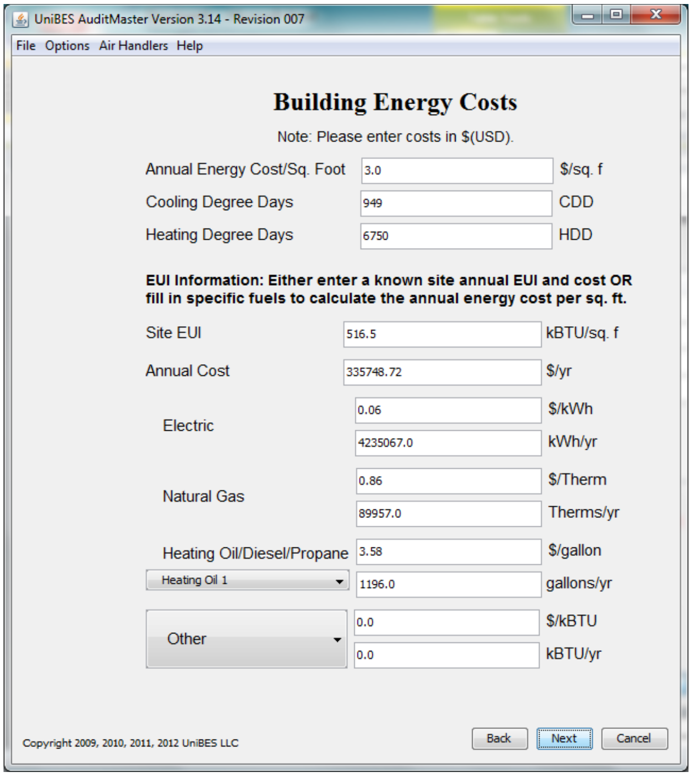
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Building Energy Costs
Note: Please enter costs in $(USD).
Annual Energy Cost/Sq. Foot 3.0 $/sq. f
Cooling Degree Days 949 CDD
Heating Degree Days 6750 HDD
EUI Information: Either enter a known site annual EUI and cost OR fill in specific fuels to calculate the annual energy cost per sq. ft.
Site EUI 516.5 kBTU/sq. f
Annual Cost 335748.72 $/yr
Electric 0.06 $/kWh
4235067.0 kWh/yr
Natural Gas 0.86 $/Therm
89957.0 Therms/yr
Heating Oil/Diesel/Propane 3.58 $/gallon
Heating Oil 1 1196.0 gallons/yr
Other 0.0 $/kBTU
0.0 kBTU/yr
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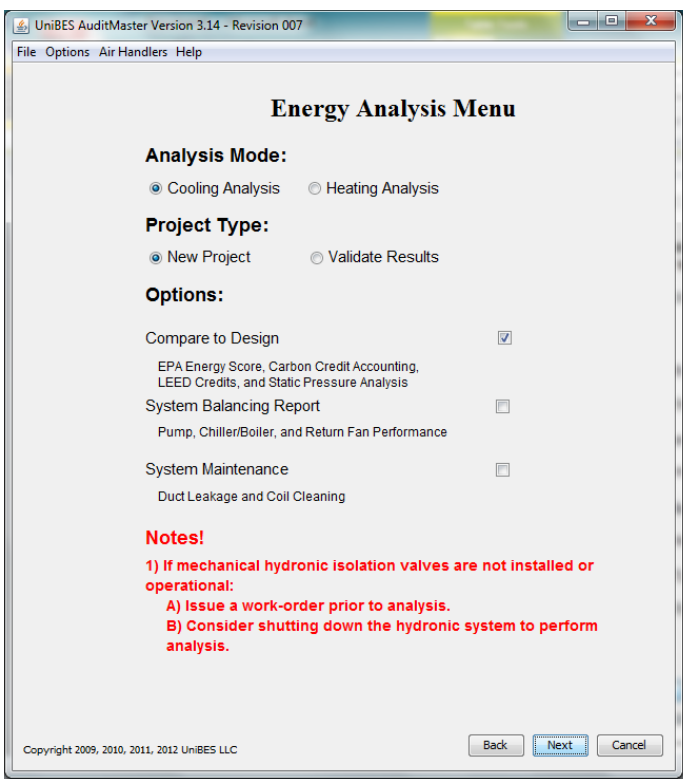
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Energy Analysis Menu
Analysis Mode:
Cooling Analysis Heating Analysis
Project Type:
New Project Validate Results
Options:
Compare to Design
EPA Energy Score, Carbon Credit Accounting, LEED Credits, and Static Pressure Analysis
System Balancing Report
Pump, Chiller/Boiler, and Return Fan Performance
System Maintenance
Duct Leakage and Coil Cleaning
Notes!
1) If mechanical hydronic isolation valves are not installed or operational:
A) Issue a work-order prior to analysis.
B) Consider shutting down the hydronic system to perform analysis.
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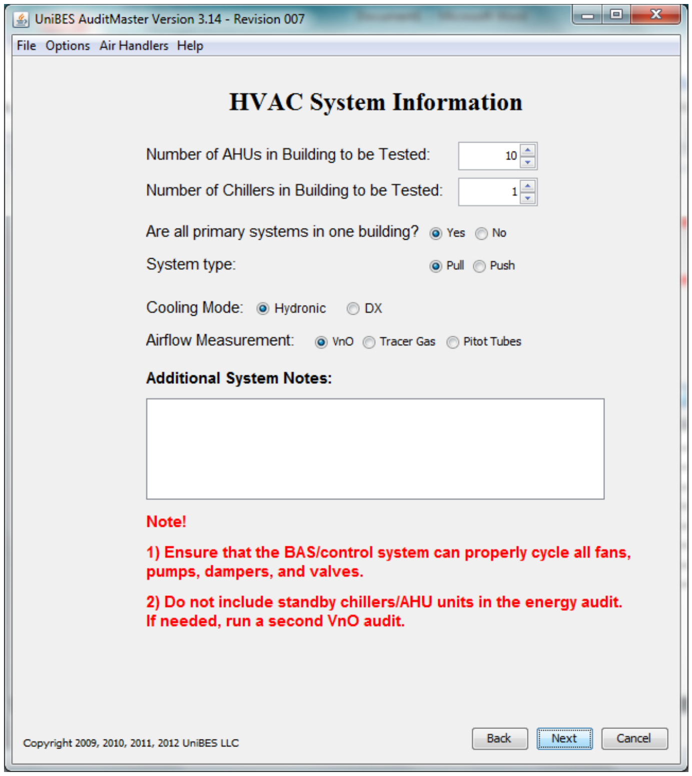
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
HVAC System Information
Number of AHUs in Building to be Tested: 10
Number of Chillers in Building to be Tested: 1
Are all primary systems in one building? Yes No
System type: Pull Push
Cooling Mode: Hydronic DX
Airflow Measurement: VnO Tracer Gas Pitot Tubes
Additional System Notes:
Note!
1) Ensure that the BAS/control system can properly cycle all fans, pumps, dampers, and valves.
2) Do not include standby chillers/AHU units in the energy audit. If needed, run a second VnO audit.
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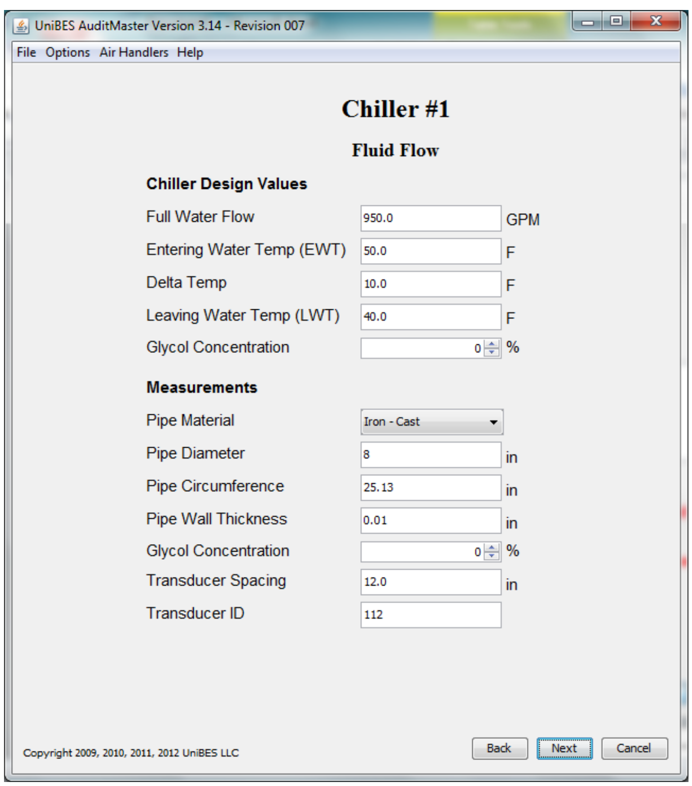
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Chiller #1
Fluid Flow
Chiller Design Values
Full Water Flow 950.0 GPM
Entering Water Temp (EWT) 50.0 F
Delta Temp 10.0 F
Leaving Water Temp (LWT) 40.0 F
Glycol Concentration 0 %
Measurements
Pipe Material Iron - Cast
Pipe Diameter 8 in
Pipe Circumference 25.13 in
Pipe Wall Thickness 0.01 in
Glycol Concentration 0 %
Transducer Spacing 12.0 in
Transducer ID 112
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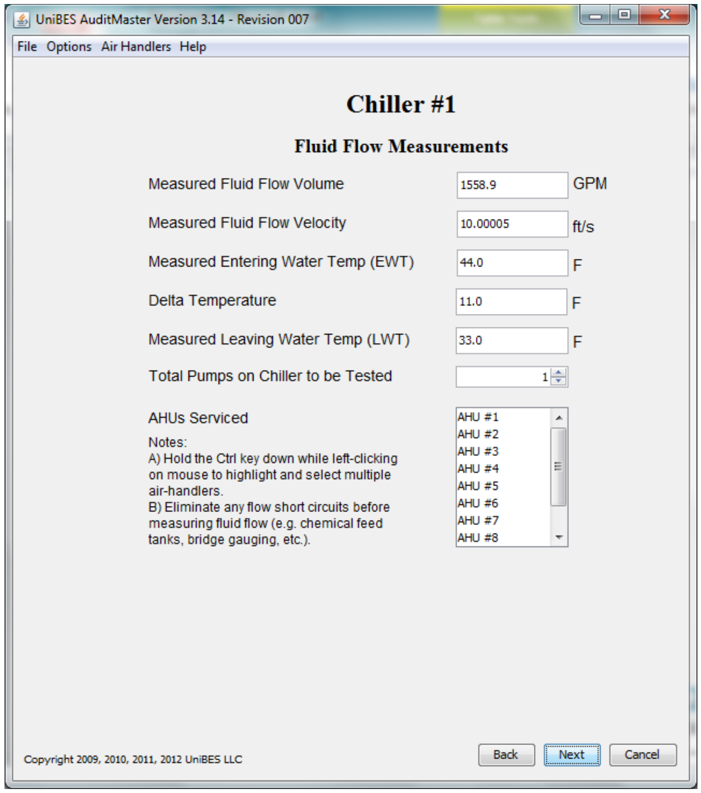
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Chiller #1
Fluid Flow Measurements
Measured Fluid Flow Volume 1558.9 GPM
Measured Fluid Flow Velocity 10.00005 ft/s
Measured Entering Water Temp (EWT) 44.0 F
Delta Temperature 11.0 F
Measured Leaving Water Temp (LWT) 33.0 F
Total Pumps on Chiller to be Tested 1
AHUs Serviced
AHU #1
AHU #2
AHU #3
AHU #4
AHU #5
AHU #6
AHU #7
AHU #8
Notes:
A) Hold the Ctrl key down while left-clicking on mouse to highlight and select multiple air-handlers.
B) Eliminate any flow short circuits before measuring fluid flow (e.g. chemical feed tanks, bridge gauging, etc.).
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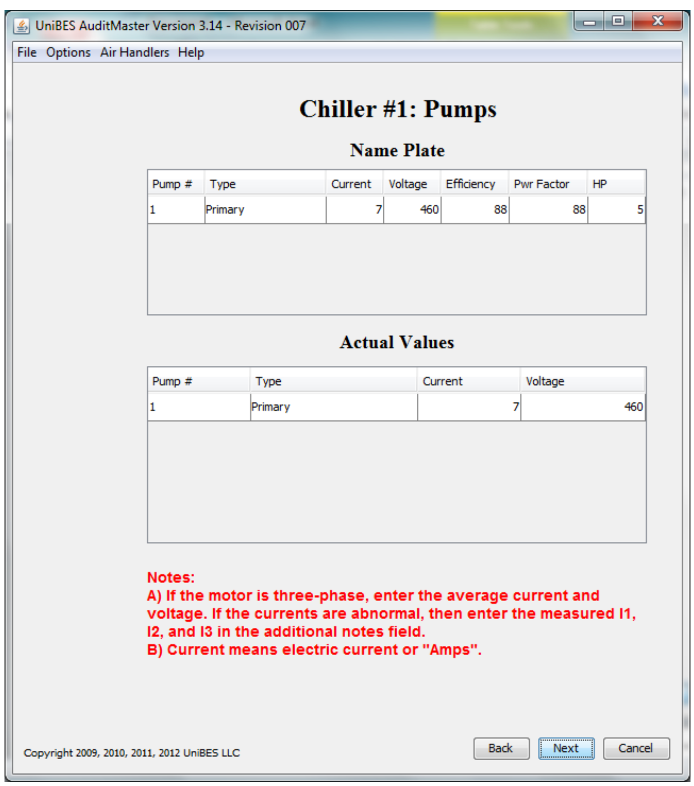
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Chiller #1: Pumps
Name Plate
Pump # Type Current Voltage Efficiency Pwr Factor HP
1 Primary 7 460 88 88 5
Actual Values
Pump # Type Current Voltage
1 Primary 7 460
Notes:
A) If the motor is three-phase, enter the average current and voltage. If the currents are abnormal, then enter the measured I1, I2, and I3 in the additional notes field.
B) Current means electric current or "Amps".
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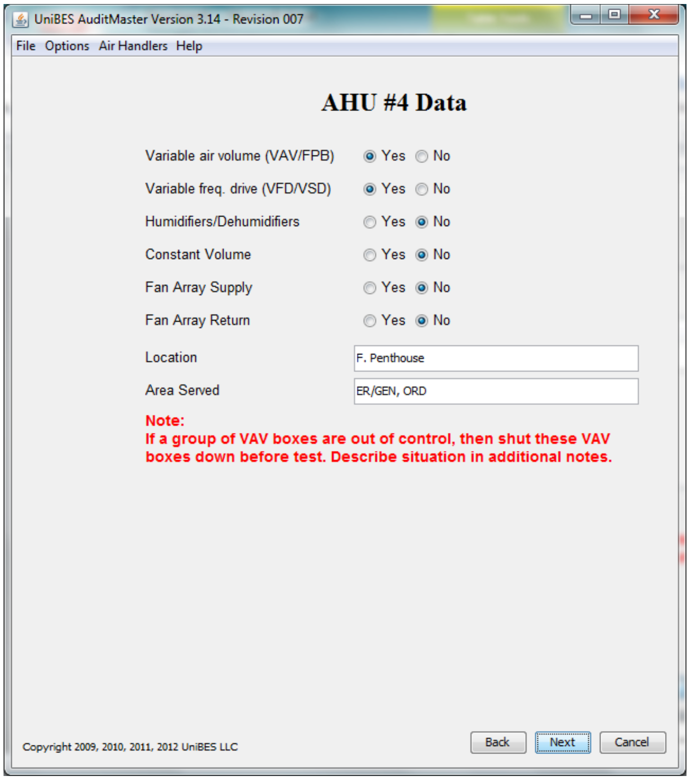
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4 Data
Variable air volume (VAV/FPB) Yes No
Variable freq. drive (VFD/VSD) Yes No
Humidifiers/Dehumidifiers Yes No
Constant Volume Yes No
Fan Array Supply Yes No
Fan Array Return Yes No
Location F. Penthouse
Area Served ER/GEN, ORD
Note:
If a group of VAV boxes are out of control, then shut these VAV boxes down before test. Describe situation in additional notes.
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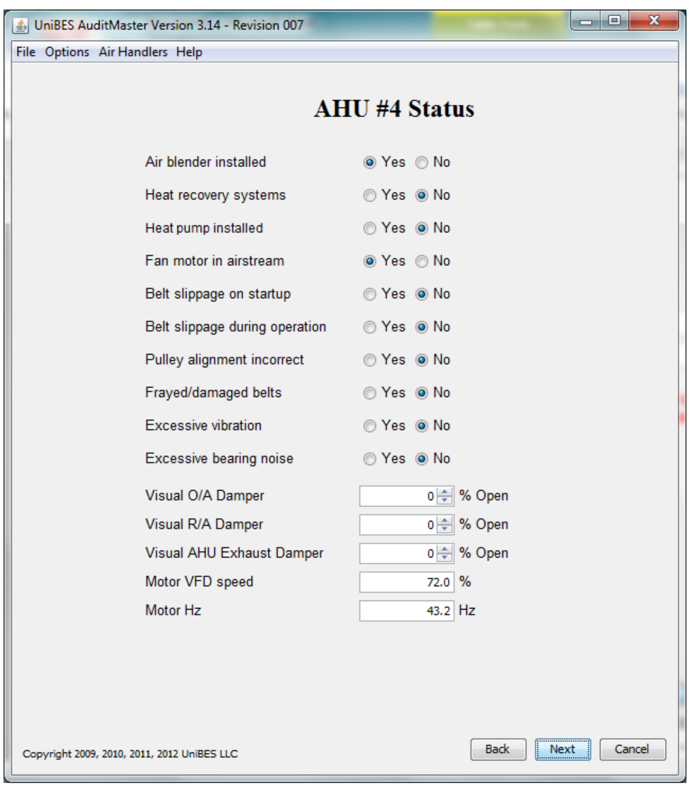
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4 Status
Air blender installed Yes No
Heat recovery systems Yes No
Heat pump installed Yes No
Fan motor in airstream Yes No
Belt slippage on startup Yes No
Belt slippage during operation Yes No
Pulley alignment incorrect Yes No
Frayed/damaged belts Yes No
Excessive vibration Yes No
Excessive bearing noise Yes No
Visual O/A Damper 0 % Open
Visual R/A Damper 0 % Open
Visual AHU Exhaust Damper 0 % Open
Motor VFD speed 72.0 %
Motor Hz 43.2 Hz
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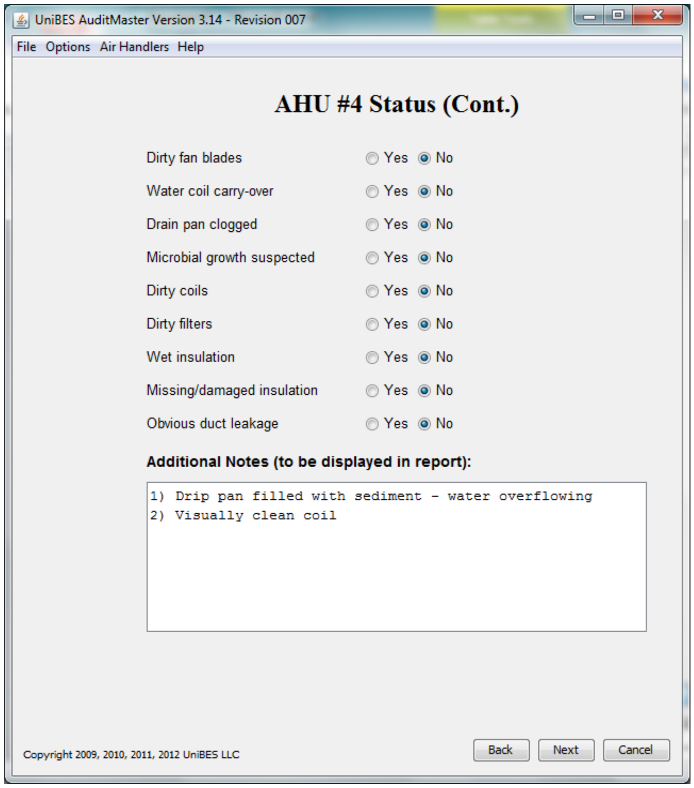
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4 Status (Cont.)
Dirty fan blades Yes No
Water coil carry-over Yes No
Drain pan clogged Yes No
Microbial growth suspected Yes No
Dirty coils Yes No
Dirty filters Yes No
Wet insulation Yes No
Missing/damaged insulation Yes No
Obvious duct leakage Yes No
Additional Notes (to be displayed in report):
1) Drip pan filled with sediment - water overflowing
2) Visually clean coil
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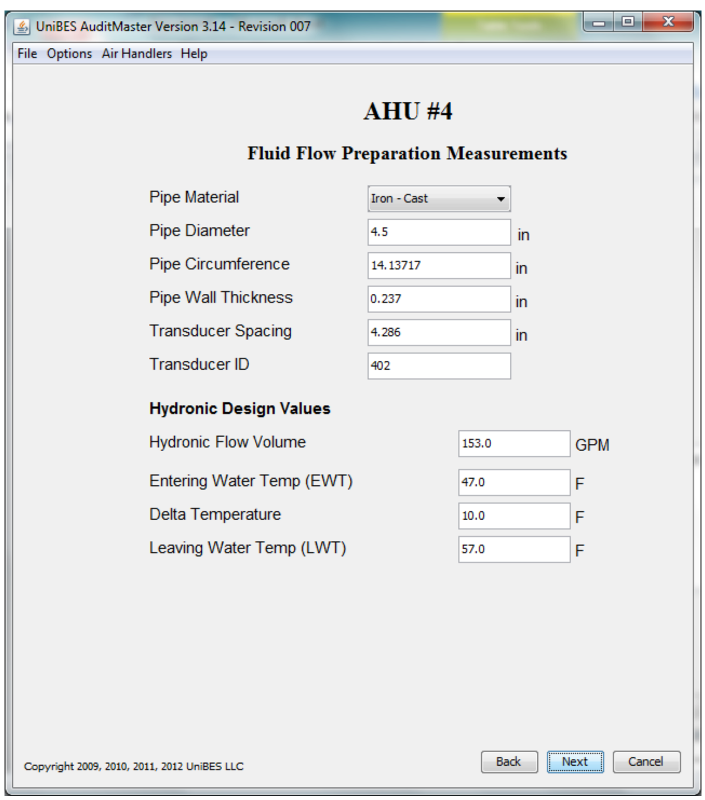
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Fluid Flow Preparation Measurements
Pipe Material Iron - Cast
Pipe Diameter 4.5 in
Pipe Circumference 14.13717 in
Pipe Wall Thickness 0.237 in
Transducer Spacing 4.286 in
Transducer ID 402
Hydronic Design Values
Hydronic Flow Volume 153.0 GPM
Entering Water Temp (EWT) 47.0 F
Delta Temperature 10.0 F
Leaving Water Temp (LWT) 57.0 F
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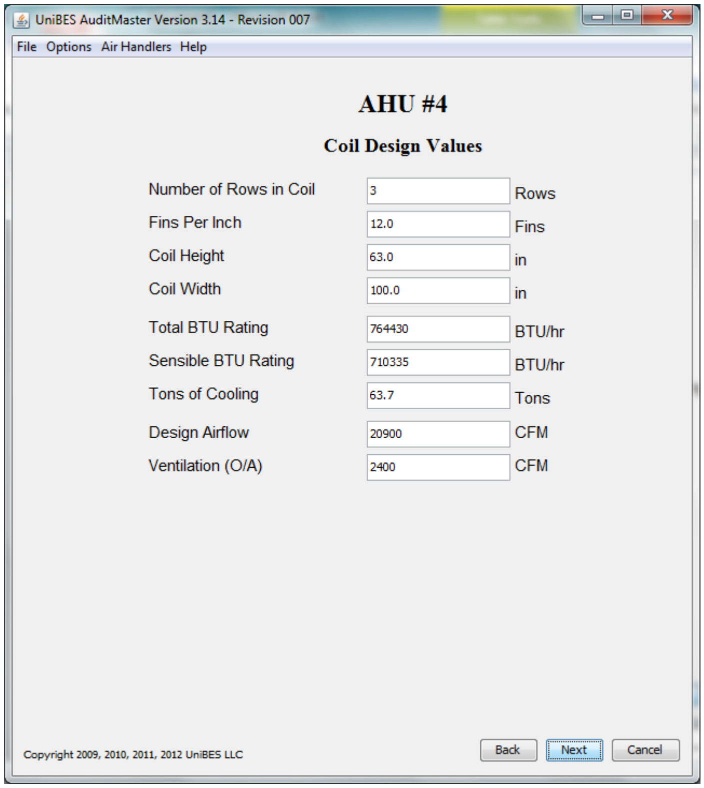
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Coil Design Values
Number of Rows in Coil 3 Rows
Fins Per Inch 12.0 Fins
Coil Height 63.0 in
Coil Width 100.0 in
Total BTU Rating 764430 BTU/hr
Sensible BTU Rating 710335 BTU/hr
Tons of Cooling 63.7 Tons
Design Airflow 20900 CFM
Ventilation (O/A) 2400 CFM
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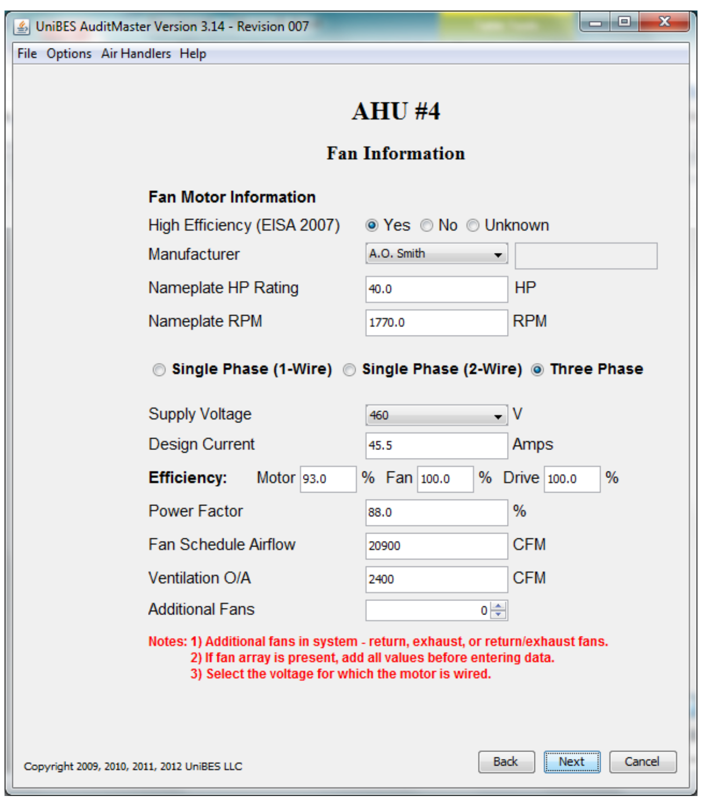
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Fan Information
Fan Motor Information
High Efficiency (EISA 2007) Yes No Unknown
Manufacturer A.O. Smith
Nameplate HP Rating 40.0 HP
Nameplate RPM 1770.0 RPM
Single Phase (1-Wire) Single Phase (2-Wire) Three Phase
Supply Voltage 460 V
Design Current 45.5 Amps
Efficiency: Motor 93.0 % Fan 100.0 % Drive 100.0 %
Power Factor 88.0 %
Fan Schedule Airflow 20900 CFM
Ventilation O/A 2400 CFM
Additional Fans 0
Notes: 1) Additional fans in system - return, exhaust, or return/exhaust fans.
2) If fan array is present, add all values before entering data.
3) Select the voltage for which the motor is wired.
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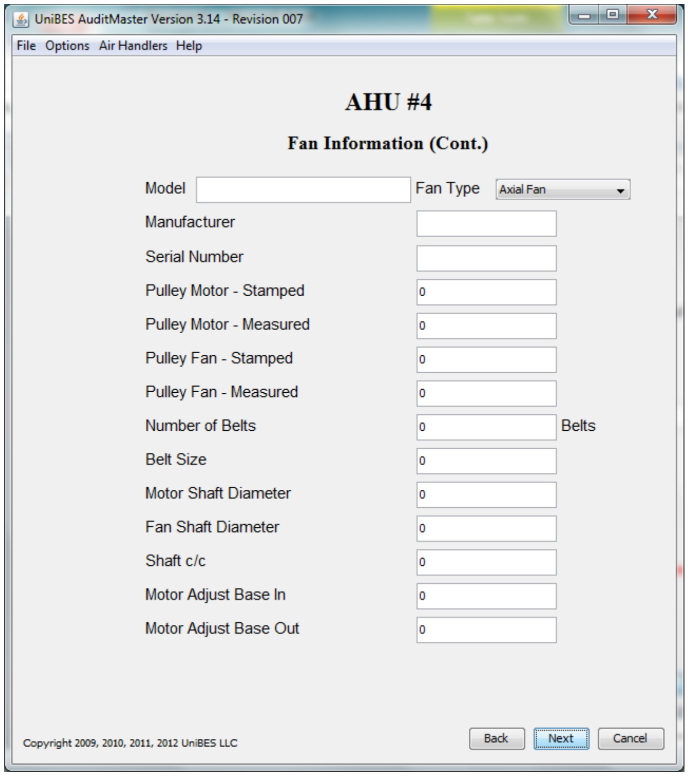
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Fan Information (Cont.)
Model
Fan Type
Axial Fan
Manufacturer
Serial Number
Pulley Motor - Stamped 0
Pulley Motor - Measured 0
Pulley Fan - Stamped 0
Pulley Fan - Measured 0
Number of Belts 0 Belts
Belt Size 0
Motor Shaft Diameter 0
Fan Shaft Diameter 0
Shaft c/c 0
Motor Adjust Base In 0
Motor Adjust Base Out 0
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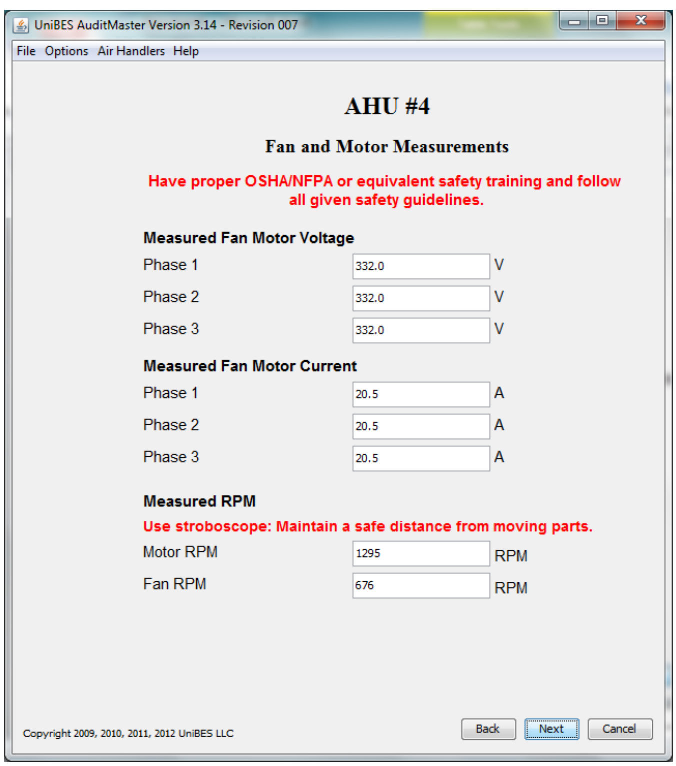
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Fan and Motor Measurements
Have proper OSHA/NFPA or equivalent safety training and follow all given safety guidelines.
Measured Fan Motor Voltage
Phase 1 332.0 V
Phase 2 332.0 V
Phase 3 332.0 V
Measured Fan Motor Current
Phase 1 20.5 A
Phase 2 20.5 A
Phase 3 20.5 A
Measured RPM
Use stroboscope: Maintain a safe distance from moving parts.
Motor RPM 1295 RPM
Fan RPM 676 RPM
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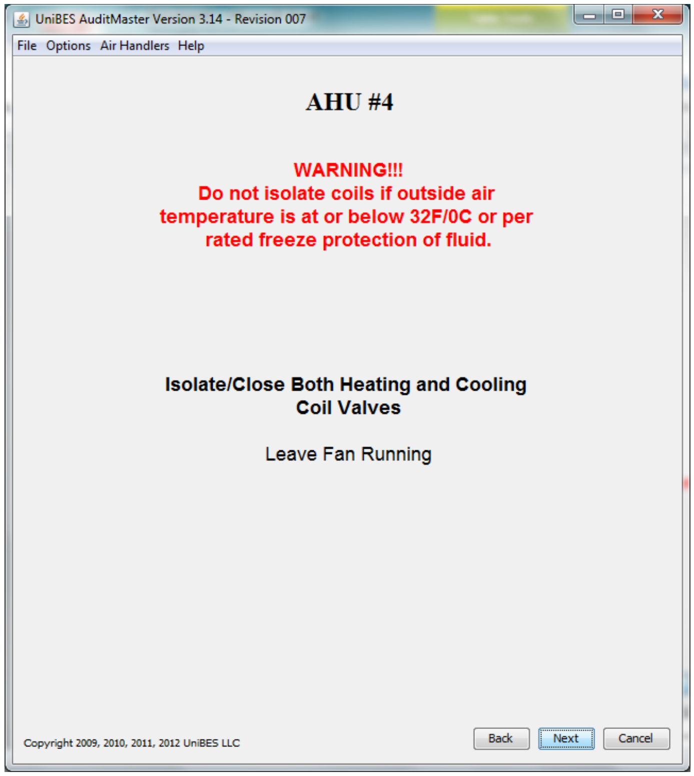
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
WARNING!!!
Do not isolate coils if outside air temperature is at or below 32F/0C or per rated freeze protection of fluid.
Isolate/Close Both Heating and Cooling Coil Valves
Leave Fan Running
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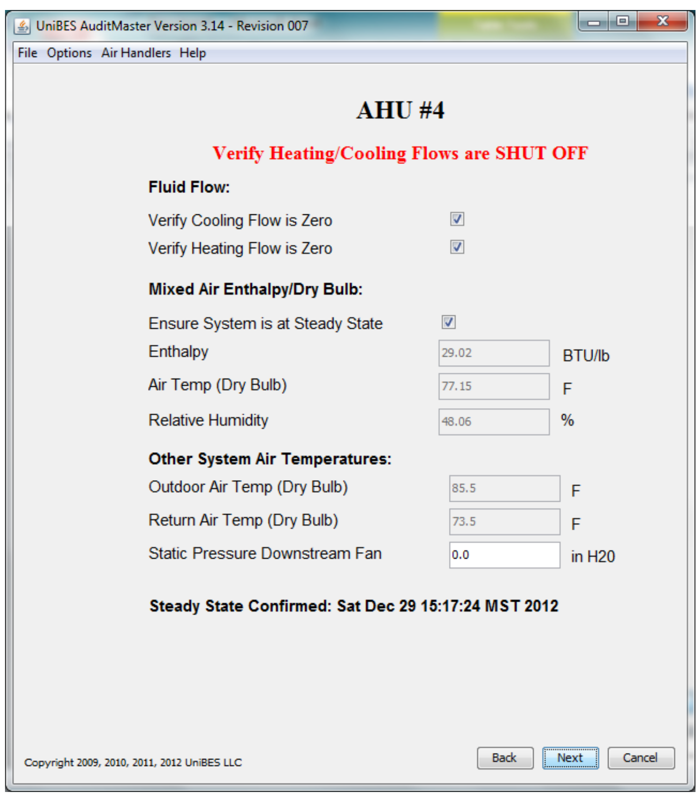
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Verify Heating/Cooling Flows are SHUT OFF
Fluid Flow:
Verify Cooling Flow is Zero
Verify Heating Flow is Zero
Mixed Air Enthalpy/Dry Bulb:
Ensure System is at Steady State
Enthalpy 29.02 BTU/lb
Air Temp (Dry Bulb) 77.15 F
Relative Humidity 48.06 %
Other System Air Temperatures:
Outdoor Air Temp (Dry Bulb) 85.5 F
Return Air Temp (Dry Bulb) 73.5 F
Static Pressure Downstream Fan 0.0 in H20
Steady State Confirmed: Sat Dec 29 15:17:24 MST 2012
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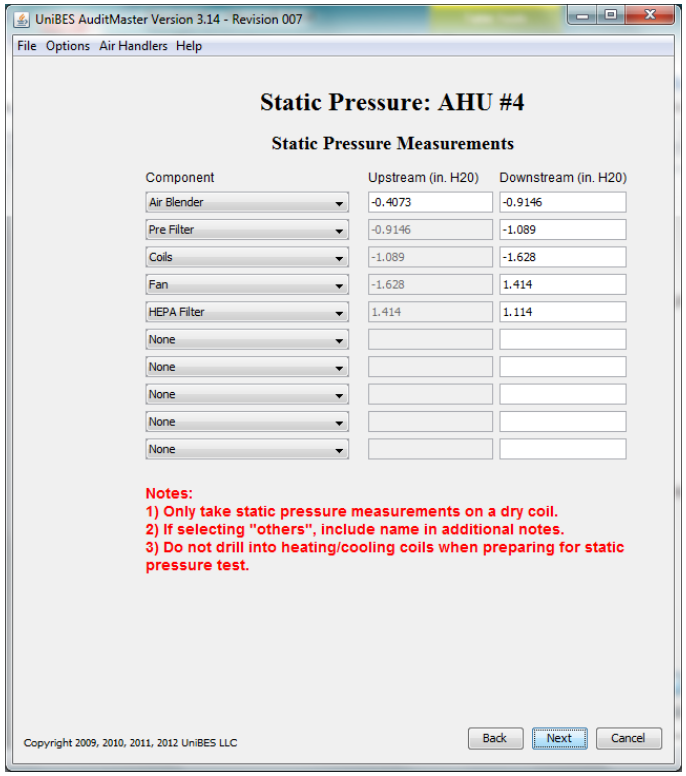
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Static Pressure: AHU #4
Static Pressure Measurements
Component
Upstream (in. H20)
Downstream (in. H20)
Air Blender -0.4073 -0.9146
Pre Filter -0.9146 -1.089
Coils -1.089 -1.628
Fan -1.628 1.414
HEPA Filter 1.414 1.114
None
None
None
None
None
Notes:
1) Only take static pressure measurements on a dry coil.
2) If selecting "others", include name in additional notes.
3) Do not drill into heating/cooling coils when preparing for static pressure test.
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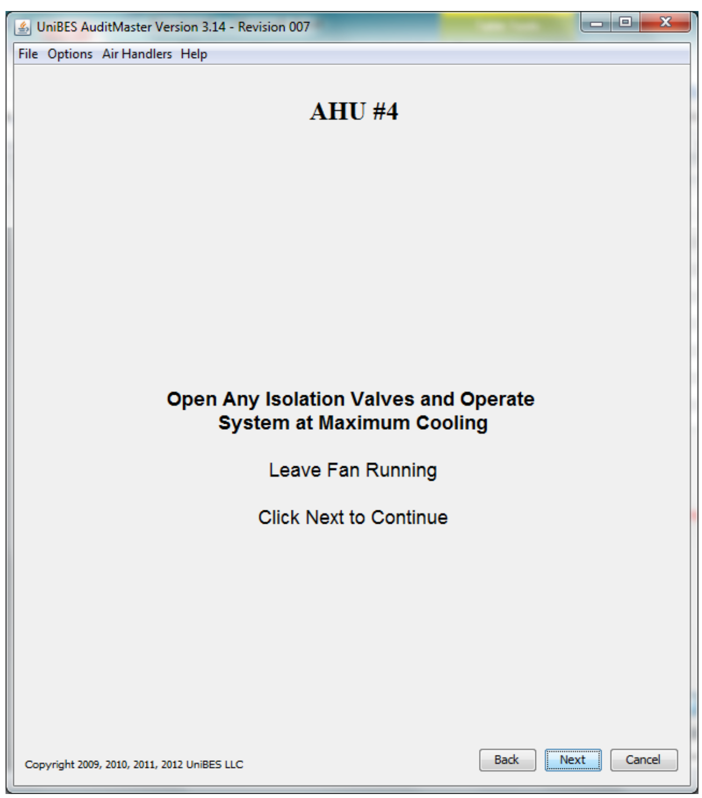
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Open Any Isolation Valves and Operate
System at Maximum Cooling
Leave Fan Running
Click Next to Continue
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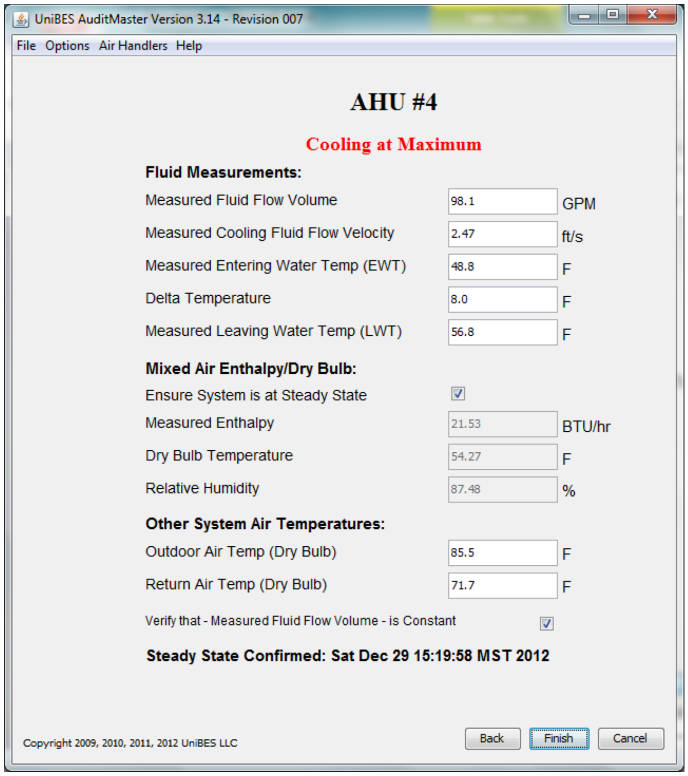
Next
Cancel


UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU #4
Cooling at Maximum
Fluid Measurements:
Measured Fluid Flow Volume 98.1 GPM
Measured Cooling Fluid Flow Velocity 2.47 ft/s
Measured Entering Water Temp (EWT) 48.8 F
Delta Temperature 8.0 F
Measured Leaving Water Temp (LWT) 56.8 F
Mixed Air Enthalpy/Dry Bulb:
Ensure System is at Steady State
Measured Enthalpy 21.53 BTU/hr
Dry Bulb Temperature 54.27 F
Relative Humidity 87.48 %
Other System Air Temperatures:
Outdoor Air Temp (Dry Bulb) 85.5 F
Return Air Temp (Dry Bulb) 71.7 F
Verify that - Measured Fluid Flow Volume - is Constant
Steady State Confirmed: Sat Dec 29 15:19:58 MST 2012
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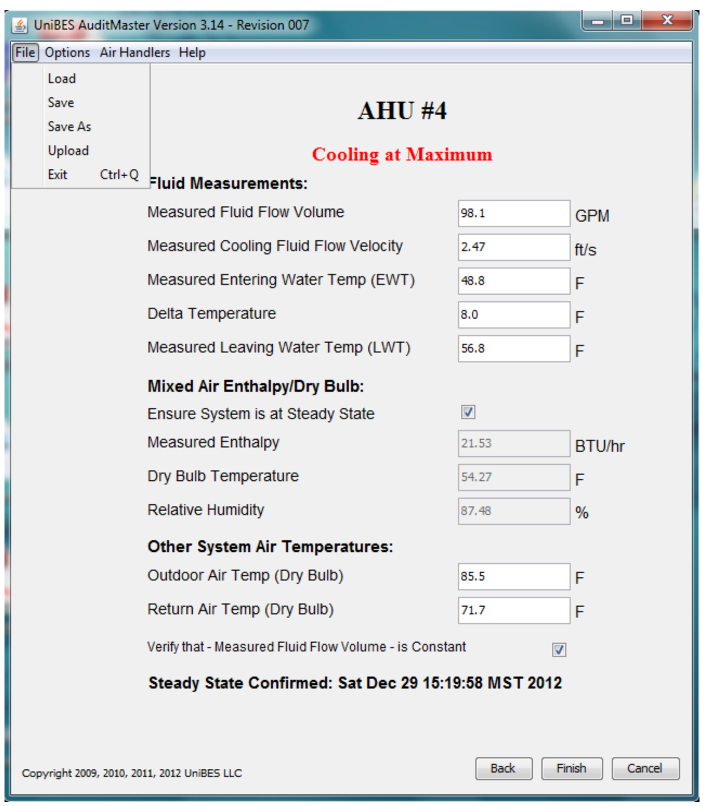
Finish
Cancel


UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Load
Save
Save As
Upload
Exit Ctrl+Q
AHU #4
Cooling at Maximum
Fluid Measurements:
Measured Fluid Flow Volume 98.1 GPM
Measured Cooling Fluid Flow Velocity 2.47 ft/s
Measured Entering Water Temp (EWT) 48.8 F
Delta Temperature 8.0 F
Measured Leaving Water Temp (LWT) 56.8 F
Mixed Air Enthalpy/Dry Bulb:
Ensure System is at Steady State
Measured Enthalpy 21.53 BTU/hr
Dry Bulb Temperature 54.27 F
Relative Humidity 87.48 %
Other System Air Temperatures:
Outdoor Air Temp (Dry Bulb) 85.5 F
Return Air Temp (Dry Bulb) 71.7 F
Verify that - Measured Fluid Flow Volume - is Constant
Steady State Confirmed: Sat Dec 29 15:19:58 MST 2012
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Finish
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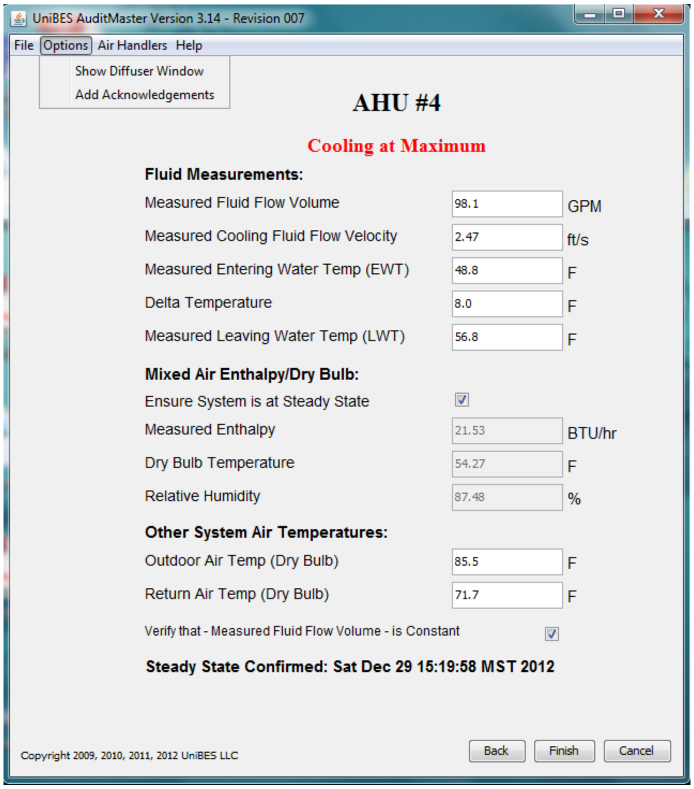
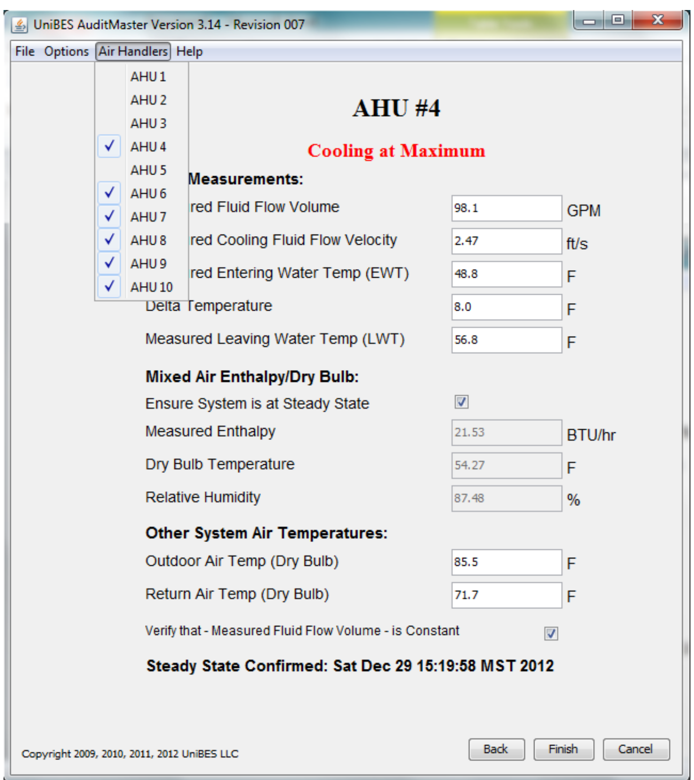

UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
Show Diffuser Window
Add Acknowledgements
AHU #4
Cooling at Maximum
Fluid Measurements:
Measured Fluid Flow Volume 98.1 GPM
Measured Cooling Fluid Flow Velocity 2.47 ft/s
Measured Entering Water Temp (EWT) 48.8 F
Delta Temperature 8.0 F
Measured Leaving Water Temp (LWT) 56.8 F
Mixed Air Enthalpy/Dry Bulb:
Ensure System is at Steady State
Measured Enthalpy 21.53 BTU/hr
Dry Bulb Temperature 54.27 F
Relative Humidity 87.48 %
Other System Air Temperatures:
Outdoor Air Temp (Dry Bulb) 85.5 F
Return Air Temp (Dry Bulb) 71.7 F
Verify that - Measured Fluid Flow Volume - is Constant
Steady State Confirmed: Sat Dec 29 15:19:58 MST 2012
Copyright 2009, 2010, 2011, 2012 UniBES LLC
Back Finish Cancel


UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
AHU 1
AHU 2
AHU 3
AHU 4
AHU 5
AHU 6
AHU 7
AHU 8
AHU 9
AHU 10
AHU #4
Cooling at Maximum
Measurements:
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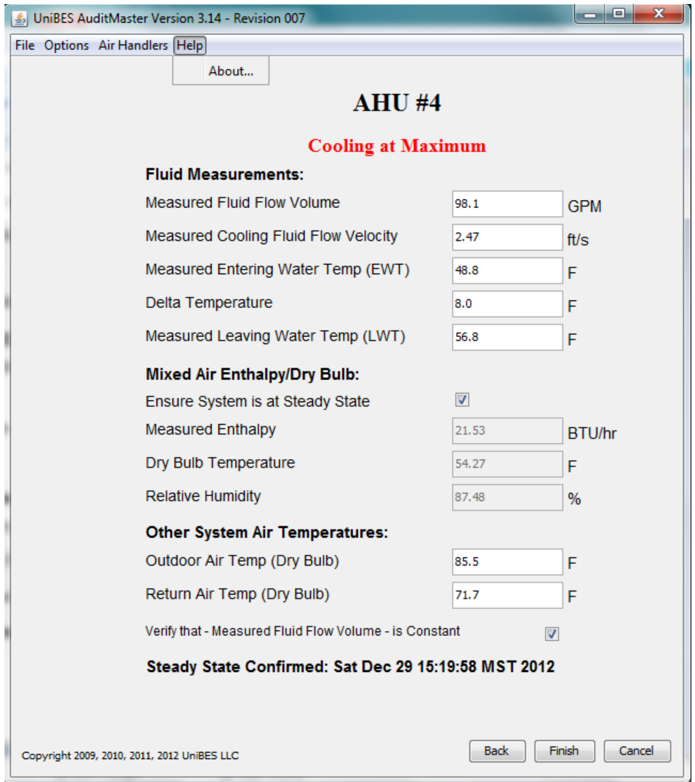
UniBES AuditMaster Version 3.14 - Revision 007
File Options Air Handlers Help
About...
AHU #4
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Verify that - Measured Fluid Flow Volume - is Constant
Steady State Confirmed: Sat Dec 29 15:19:58 MST 2012
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APPENDIX B: AUDITMASTER™ REPORT

UniBES LLC
United Building & Energy Sciences LLC
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Low-Cost High-Impact HVAC Energy Performance Analysis
July 25, 2011

Estimated Energy Star Score: 50%

Building	Service Provider
Pine Ridge Comprehensive Health Care Facility	Lloyds Systems LLC
Steve J. Dykstra	Lance Weaver
P.O. Box 1201 East Highway 18	2911 W. Omaha Street
Pine Ridge, SD 57770	Rapid City, SD 57702
(605) 355-2283	605-381-4189
Steve.Dykstra@ihs.gov	lance@lloydssystems.com

This report has been prepared by professionals holding the following degrees: PhD EE, PhD ME, MSTM, MSEE, BSEE, BSME, BSCS, BSM and BSAE. The team holds certifications in ASCS, CEM, CEA, NEMI, CxA, LEED, PEM, AABC TBE, TABB, CRM, INFRARED ANALYST LEVEL III, VIBRATION ANALYST LEVEL I. One of the team members also participated in the development of the ITI Energy program.

Report Reference No. 9531-076370

Executive Summary Report

Lloyds Systems LLC utilized AuditMaster™ procedures and software to execute a HVAC energy performance analysis to accurately measure actual field energy efficiency and usage for the **Pine Ridge Comprehensive Health Care Facility, Pine Ridge, SD on July 25, 2011**. The UniBES proprietary, copyrighted and patent pending measurement procedures identify prescriptive energy conservation measures (ECMs) which can be used to correct deficiencies and optimize the HVAC system performance, thereby reducing energy use and building operational costs. Furthermore, by identifying the actual HVAC performance, engineers have the necessary information to meet current ASHRAE, LEED and IAQ requirements, such as building load design, thermal comfort, outside air demand ventilation rates and humidity control. **This building does not monitor energy consumption with independent meters.** This report provides results from the **initial energy audit study** for this building.

The **Pine Ridge Comprehensive Health Care Facility** is **10,405** square meters divided on **3** floors where **10,405** square meters were audited. The base electric rate is **$(USD)0.06** per kWh. Current annual HVAC energy usage is **$(USD)32.27** per square meter with an energy use index (EUI) of **5559** kJ per square meter, which has been used to calculate the EPA Energy Star Score (www.energystar.gov) to be 50% compared to similar buildings. Energy Star measured greenhouse gas consumption is **2,416,883 tons CO2e** or a carbon credit equivalent potential of **2,416,883** to become net zero. Cooling and heating degree day numbers for the region are **949** and **6750**, respectively (www.degreedays.net). Hourly contractor rates are as follows: Test and Balance **$(USD)100**, Mechanical **$(USD)85** and Coil Cleaning **$(USD)75**. This information is used along with measured data to determine cost for services and module replacements, as well as potential savings estimates as calculated in this report. Please do not hesitate to contact UniBES LLC at report@unibes.com if you have any questions regarding the calculations or savings estimates.

The HVAC energy performance analysis executed on AHU4 used **Hydronic Cooling Mode** with enthalpy as the measured parameter. Analysis modules presented in this report are: **AHU Airflow, A Comparison to Design Specifications, Ventilation Ratio and Static Pressure**. Module(s) not included are: **Duct Leakage Analysis and Coil Cleaning Analysis**. By implementing the prescriptive ECMs as outlined in this report, the utility-paying customer can expect to see a dramatic reduction in utility expenditures. These savings are accomplished by reduction of demand charges, time of use, peak rates, RATCHET clause charges, etc., typically billed at a substantial increase above the provided **$(USD)0.06** base rate. Additional savings may be realized through available rebate programs such as ENERGY GRID IQ (www.energygrid.com). Additionally, the building owner will increase the building's resale value and better attract new tenants, as the life of installed equipment is extended and maintenance and wear reduced. Implementing system changes and service based on actual performance data will improve tenant thermal comfort along with indoor air quality.

Actual Operational Performance Data

	AHU4				
Airflow	56%				
Ventilation O/A	138%				
Heat Transfer	52%				
Hydronic Flow	65%				
Fan Motor HP	33%				

Chiller 1 Balance Report and Pump Performance

Chiller 1 Parameters			
	Design Value	Actual Value	Deviation
Units Fed	0	0	0
Fluid Flow (LPM)	3,596.0	5,901.0	2,305.0
EWT (C)	10.0	6.7	3.3
LWT (C)	4.4	0.6	3.9
Δ T (C)	5.6	6.1	0.6
COP	-	-	-

Chiller 1 Pump Performance Analysis					
ID	Actual kW	Name Plate HP	Actual BHP	Motor Operation	System Fluid Flow %
Primary	4.9	5.0	3.8	76%	164.0

Report Reference No. 9531-076370

AHU 4	Design Value	Actual Value	Deviation	Perf. %	Saving/yr $(USD)	Intervention	Cost $(USD)	ROI	Carbon Credits	LEED Credits	IAQ
AHU 4 Airflow											
Total Fan Airflow (CMM)	592	329	263	**56**	$4634	Change Pulley*	$1420	0.3 Yrs	47	EA 2.1 (2p)	-
Compare to Design										-	
Total Fan Airflow (CMM)	592	329	263	**56**	-	-	-	-	-	-	-
Total Outside Airflow (CMM)	68	94	26	**138**	-	-	-	-	-	-	1
Total Return Airflow (CMM)	524	235	289	**45**	-	-	-	-	-	-	-
Fan Motor HP/BHP	40.00	12.93	27.07	**33**	-	-	-	-	-	-	-
Thermal Performance											
Total Heat Transfer (kJ/hr)	806,516	414,058	392,458	**52**	-	Systems Balancing	$500	-	-	-	2
Sensible Heat Transfer (kJ/hr)	749,443	4,527	744,916	**1**	-	-	-	-	-	-	-
Latent Heat Transfer (kJ/hr)	57,073	409,531	352,458	**717**	-	-	-	-	-	-	2,5
Upstream DB/RH (C/%)	-	25/48	-	-	-	-	-	-	-	-	-
Downstream DB/RH (C/%)	-	12/87	-	-	-	-	-	-	-	-	-
Coil (kJ/hr)	806,516	414,058	392,458	**52**	-	-	-	-	-	-	-
Hydronic Flow (LPM)	579	371	208	**65**	-	-	-	-	-	-	-
Hydronic Delta Temp (C)	5	4	1	**81**	-	-	-	-	-	-	-
Water Coil Carryover (MPM)	145	81	64	**56**	-	-	-	-	-	-	5
Ventilation											
Total Outside Airflow (CMM)	68	94	26	**138**	-	Air Balancing	-	-	-	-	1
Total Return Airflow (CMM)	524	235	289	**45**	-	-	-	-	-	-	-
Estimated EPA Energy Star Score	-	50%	-	-	-	-	-	-	-	EA 1 (0p)	-
Totals					**$4634**		**$1920**		**47**	**2p**	

LEED - New: EA 1, EA 2.1; Reaudit: EA 1, EA 2.2, EA 2.3, EA 6; Monitor: EA 3.2

Intervention ***

- Change Pulley - *May increase duct air leakage
- System Balancing - chiller, boiler, and airflow
- Pulley Size Required = N/A cm
- RPM Required = 2324.49 RPM
- BHP Required = N/A BHP

Potential IAQ Issues

1. ASHRAE 62.1-2007 Ventilation
2. ASHRAE 55-2004 Thermal Comfort
3. ASHRAE 55.2-1999 Air Filters
4. ASHRAE 91.1-2007 Building Energy
5. Microbial Growth Possible

Explanations

- SIR - Savings to Investment Ratio
- ROI - Return on Investment (yrs)
- Carbon Credits Reporting
- Seal Ducts - use RS 100 on main trunks
- Clean Coil - use pH-neutral solutions

Air Handler 4

AHU 4 is a PULL system with the configuration in the table below showingsystem configurations and maintenance observations. The air handler is located in F. Penthouse and serves the area: ER/GEN, ORD.

Configuration Option	YES/NO	System Observations	YES/NO
All Sub-Systems in One Building	YES	Frayed Belts	NO
Variable Air Volume	YES	Excessive Vibration	NO
Variable Frequency Drive	YES	Excessive Bearing Noise	NO
Humidifiers/Dehumidifiers	NO	Dirty Fan	NO
Constant Volume	NO	Water Coil Carry-Over	NO
Air Blender	YES	Drain Pan Clogged	NO
Heat Recovery System	NO	Microbal Growth Suspected	NO
Fan Motor in Airstream	YES	Dirty Coils	NO
Fan Array Supply	NO	Dirty Filters	NO
Fan Array Return	NO	Wet Insulation	NO
Heat Pump Installed	NO	Missing/Damaged Insulation	NO
		Obvious Duct Leakage	NO
		Belt Slippage on Start	NO
		Belt Slippage During Operation	NO
		Pully Alignment Incorrect	NO

The static pressure measurements are used to assist in system diagnostics and all measurements were taken with a dry coil.

Static Pressure Measurements				
Up/Down	**Component**	**SP Pa.**		**ΔP Pa.**
Up Stream	Air Blender	-0.1014		
Dn Stream	Air Blender	-0.2278	ΔP	0.1264
Up Stream	Pre Filter	-0.2278		
Dn Stream	Pre Filter	-0.2713	ΔP	0.0435
Up Stream	Coils	-0.2713		
Dn Stream	Coils	-0.406	ΔP	0.1347
Up Stream	Fan	-0.406		
Dn Stream	Fan	0.3522	ΔP	0.7582
Up Stream	HEPA Filter	0.3522		
Dn Stream	HEPA Filter	0.2775	ΔP	0.0747

The system fan operation performance table can be found below.

AHU 4 Fan Performance Analysis					
ID	**Actual kW**	**Name Plate HP**	**Actual BHP**	**Motor Operation**	**VFD Hz**
AHU Fan	10.3	40.0	12.9	32%	43.2

Additional Notes:

1) Drip pan filled with sediment - water coming out of unit versus drain.
2) Coil appeared clean visually (may be impact or internal defects).

VITA

BERNT A. ASKILDSEN – BORN: BERGEN, JULY 21, 1968

EDUCATION

□ Ph.D. Biomedical Engineering, SDSM&T, Rapid City, 2013

□ Master of Science Technical Management, SDSM&T, Rapid City, 2003

□ Master of Science Electrical Engineering, SDSM&T, Rapid City, 2000

□ Bachelor of Science Electrical Engineering, SDSM&T, Rapid City, 1996

□ Bergen Maritime High School, Bergen, 1988

CAREER OVERVIEW

□ LLOYDS SYSTEMS LLC, Executive Vice President, 2006 – Present

□ SDSM&T EE/CENG DEPARTMENT, Instructor, 2006 – 2012

□ REALTRONINCS CORPORATION, Vice President, 2001 – 2006

□ COMUNIQ INC., System Design Engineer, 1997 – 2001

□ SDSM&T EE/CENG DEPARTMENT, Teaching Assistant, 1997 – 1998

□ MRT MICRO INC., Application Engineer, 1996 – 1997

□ NORWEGIAN AIRFORCE, Second Lieutenant, 1988-1990

HONORS/ORGANIZATIONS

SBIR Panel Reviewer, National Science Foundation, 2010 - Present □ Genome Scholar, National Institute of Health, 2007 □ Outstanding Recent Graduate Award, 2006 □ Industry Advisor Council Member, Computer Science, SDSM&T, 2004 □ Entrepreneur Scholarship Recipient, Genesis of Innovation, 2003 □ Panel Member/Presenter, National SBIR Conference Rapid City, 2001 □ Magnet Award Recipient, Rapid City Economic Development Partnership, 2000 □ Order of the Engineer, 1996 □ Eta Kappa Nu, 1996 □ Tau Beta Pi, 1996 □ Institute of Electrical and Electronics Engineers (IEEE) member, 1994 - present □ Outstanding Leadership Award, BTS Student Society, 1994

PEER-REVIEWED PUBLICATIONS

□ B. Askildsen, C. Tolle, et al., "Accuracy and Precision Analysis of Time Stepped Enthalpy," An International Journal devoted to Investigations of Energy Use and Efficiency in Buildings, submitted June 2013.

□ B. Askildsen, C. Tolle, et al., "Robotic Duct Sealing Enables Adequate Comfort, IAQ and Energy Efficiency," International Journal of Advanced Robotic Systems, Robotics in Buildings and Infrastructure, Accepted for publication July 2013.

□ B. Askildsen, C. Tolle, et al., "Time-Stepped Enthalpy Software Application," International Society for Computers and Their Applications (ISCA) Journal of Computers and Applications, Accepted for publication June 2013.

□ B. Askildsen, C. Tolle, et al., "Time-Stepped Enthalpy Software Application," 27th International Conference on Computers and Their Applications (CATA-2013), Honolulu, Hawaii, March 4-6, 2013.

□ B. Askildsen, C. Tolle, et al., "Time-stepped enthalpy method to quantify HVAC airflow volume," AIRAH Achieving the Green Dream Conference, Sydney, 2012.

□ Zhang, N., Askildsen, B. A., and Hemmelman, B., "Investigation of Adaptive Filtering for Noisy ECG Signals" IEEE Mountain Workshop on Adaptive and Learning Systems, SMCals/06, Logan, Utah, July 24-26, 2006.

□ Weiss, J.W., Askildsen, B. A., Thompson, S. R., and Gervasi, A., "Advances In Human Target Detection Using Opaque Material Penetrating Radar Data", 20th International Conference on Computers and Their Applications (CATA-2005), New Orleans, Louisiana, Mar 16-18, 2005.

□ Whites, K, Askildsen, B.A., et al., "Economical Resistive Tapering of Bowtie Antennas", IEEE Antennas and Propagation, ISIU RSM, Monterey, California, June 20-25, 2004.

□ Weiss, J.W., Askildsen, B.A., Harstel, C., and Thompson, S. R., "Artificial Neural Network Based Human Target Detection Using RF Sensor Data", 19th International Conference on Computers and Their Applications CATA-2004, Seattle, Washington, March 18-20, 2004.