TIME-STEPPED ENTHALPY METHOD TO QUANTIFY HVAC AIRFLOW VOLUME

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ABSTRACT

Current commissioning and retro-commissioning processes of Heating Ventilation and Air-Conditioning (HVAC) systems are very limited allowing most systems to operate substantially from their intended design. The consequences are excessive waste of building energy contributing to poor indoor air quality. Current methods measure airflow using Pitot tubes or tracer-gas but these methods are often inaccurate or cost prohibitive, respectively. The hypothesis is the use of time-stepped enthalpy recordings to accurately determine HVAC airflow repeatability as system conditions change. Two series of measurements with associated calculations were obtained and compared. The recorded data from this study identifies that time stepped enthalpy readings yield results within ±10% when belts were slipping and then within ±4% under normal operating conditions. This field study provides a major step towards a reliable eight minute testing time line and an accurate validation of the actual HVAC performance when compared to the intended design. An eight minute test is quick compared to pitot tubes that typically takes an hour per duct reading and tracer gas that often takes more than eight hours. Furthemore, the time-stepped enthalpy method assesses the heat transfer efficiency and air-volume flow, which adds substantial value to the measurement procedure. The end results will be significant reductions in building energy use and improved indoor air quality through proper ventilation, thermal comfort and latent heat transfer management [5-10].

Index Terms— HVAC, commissioning, retro-commissioning, building energy, benchmark, energy audit, indoor air quality, IAQ, energy efficiency, airflow volume, airflow

1. INTRODUCTION

According to the U.S. Energy Information Administration (2009) buildings consume 49% of all energy made available to the market in the United States of America where 39% is used in the Heating, Ventilation and Air-Conditioning systems [11]. Commissioning or retro-commissioning processes of HVAC systems are currently very limited allowing for a majority of the systems to operate substantially deficient when compared to their intended design [12-13]. Due to the amounts of energy consumed by HVAC systems the consequences are enormous amounts of wasted building energy in conjunction with contributing to poor indoor air quality [14]. Through the last decade the HVAC industry has been focusing on methods related to getting more accurate energy performance data [14-15]. Virtually every building has electric, gas or other fuel revenue meters monitoring the buildings energy consumption [16-17], and calculating whole building energy use for a year identified through an energy use index (EUI). EUI is the basis for the U.S. Environmental Protection Agency's (EPA) Energy Star portfolio manager [18] to calculate the energy score benchmark for similar like buildings. Furthermore the U.S. Green Building Council (USGBC) additionally relies on the Energy Star Score to issue LEED credits [19]. However, the EPA energy score benchmark falls short of providing any information about why a building operates poorly. There are currently no reliable or repeatable methods available to properly diagnose what underlying HVAC problems exist. Therefore the HVAC control systems that monitor and operate the energy transfer components will continue to run inefficiently. Problems limiting the energy efficiency of an HVAC system may be related to outdated subsystems, fluid flow balancing, duct leakage, dirty systems and their components, etc [20-23]. For example just obtaining accurate airflow measurements is challenging. Flow-hoods are usually only accurate to within three percent not counting the common operating errors whereas airflow measurements using Pitot tubes can also be inaccurate or in the case of using tracer-gas which is often found to be very accurate but cost prohibitive. Another approach is psychrometrics where delta enthalpies are obtained and then applied using HVAC psychrometric formulas to calculate systems airflow. However, the quickly varying temperature and air velocity environment upstream makes it impossible to obtain reliable results. To overcome these problems measuring system enthalpy's a new time-stepped enthalpy method has been developed [24-25].

2. HYPOTHESIS

One central question in regards to the time-stepped enthalpy method is to determine how airflow volume accuracy is impacted by variations in measurement parameters. The formula for calculating the airflow volume, using delta enthalpy and total heat (Q), is an inverse multivariable function thus linear behavior cannot be expected. The hypothesis for this investigation is if the use of time-stepped enthalpy recordings can accurately determine HVAC airflows volumes repeatedly as system conditions change. Furthermore, the study aims to quantify the airflow volume accuracy as certain measurement variances exists in enthalpy readings. An important factor for this study is the prior knowledge of approximate airflow volume. This allows the measured data with associated calculations to be benchmarked to the previously assumed airflow volumes. Subsequently observed errors can be classified as random or bias to support the objective with a time-stepped enthalpy method to converge to a field recorded accuracy of approximately one percent.

3. METHOD

The time-stepped enthalpy method is the underlying technique for this study [12-13]. It is based on psychrometrics, thermodynamics and utilizes various measurements to quantify the energy released into or extracted from the system (Equ. 1). This value is then substituted into the total heat formula (Equ. 2) to calculate the airflow volume. To show the versatility of the applied technique a water based heat recovery system was selected for the investigation. A HVAC heat recovery system is used to transfer energy from the exhaust air to allow preheating of cooler incoming outside ventilation air [1].

For a water based system as investigated in this study only three measurements are required to obtain how much energy is extracted from the buildings exhaust and released into the incoming airflow: water fluid flow (V) and the differential temperatures (T) in and out of the heat transfer device. These measurements are then substituted into the water system equation (Equ. 1) that would yield an accuracy of approximately 1%.

$$Q_M\left(\frac{kJ}{hr}\right) = 250.8\left(\frac{kJ \cdot \min}{hr \cdot l \cdot {}^\circ C}\right) \cdot V_{water}\left(\frac{l}{\min}\right) \cdot \Delta T_M\left({}^\circ C\right) \qquad (1)$$

The constant provided are derived for systems running at sea level and standard air temperatures thus are normally adjusted for systems running at higher elevations or temperatures not consitent with standard air. Nevertheless, the total heat (Q) is found by converting water flow in liters per minutes to liters per hour multiplied with the weight of one standard liter of water, which is a constant to be multiplied again with the heat content constant, and the delta temperature of the water flow that makes up the cofficient. The total heat, here in kJ's per hour, transferred from the up-stream entering heat ladened airflow is equal to the total heat transferred in the water fluid flow. Should the water be changed to a glycol mixture the specific gravity of the fluid shall be determined and the flow and heat content capacity corrected to the specific gravity value. The measured energy will then be substituted into the total heat formula used for actual HVAC system heat transferred into the system (Equ. 2) along with the delta enthalpy reading obtained by running the system at full system capacity and then with no heat transfer present.

$$Q\left(\frac{kJ}{hr}\right) = V_{airflow}\left(\frac{m^3}{\min}\right) \cdot 72.09\left(\frac{\min \cdot kg}{hr \cdot m^3}\right) \cdot \Delta h\left(\frac{kJ}{kg}\right) \qquad (2)$$

Instruments used to perform this study were one ultrasonic transient time liquid flow meter (GE Panametrics pt878, accuracy ±1%), two digital temperature meters (Fluke 5000A-RH/T, accuracy ±0.1%), three digital enthalpy meters (Vaisala gm70, accuracy ±1%) and one electric energy meter (Fluke 435, accuracy ±0.1%). Furthermore, the static pressures were monitored throughout the study using a Shortridge 870, accuracy ±2%). The field experiment was set up by measuring the liquid fluid flow through the exhaust heat recovery loop in tandem with the liquid delta temperature of the same. Ambient enthalpy, relative humidity, and dry bulb temperatures were monitored along with the exhaust air and the outside heat recovered airflow to ensure only moderate changes in test environment occurred during investigation. Fig. 1 is a simplified illustration of a generic test setup where the energy into the system is measured at the heat transfer coil and the enthalpy at the fan outlet. The fan adds the same heat of compression and bearing heat into the airstream when running the heat transfer at full capacity and when shut off, thus automatically neatralizing the enthaphy effects of the fan as can be seen from the following equation e.g. $\Delta h=[h_1+h_{fan}-(h_2+h_{fan})] = h_1-h_2$.

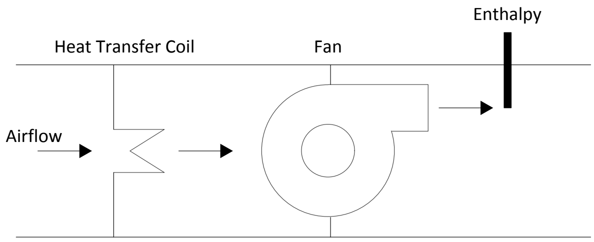

Fig. 1. A simplified field test setup – The exhausted airflow heat gets extracted and this energy is then transferred to the systems supply fan incoming outside air needed for proper design ventilation.

All the instruments were configured to data log and the time stamps were aligned at one minute time steps. Two identical heat transfer tests six weeks apart were executed with five and seven repeated tests respectively. Enthalpy data recorded for one of the test series is shown in Fig. 2.

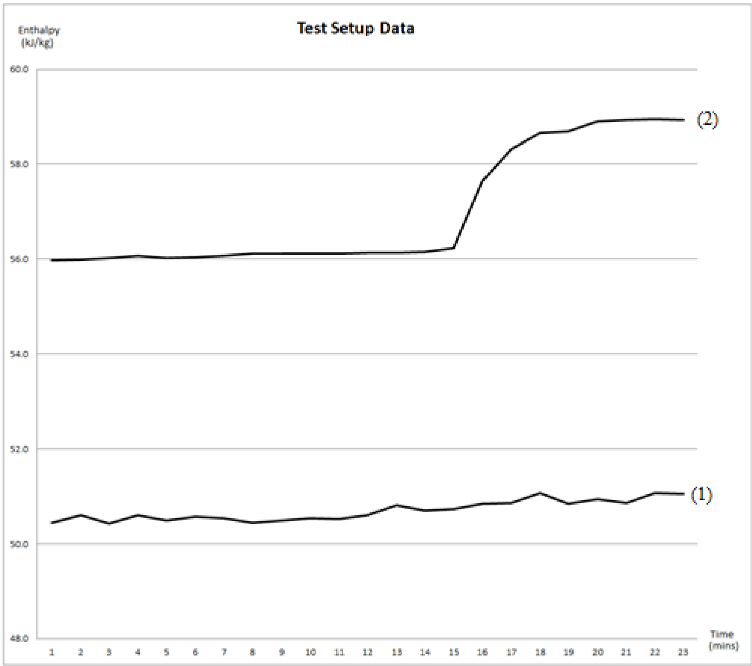

Fig. 2. The test setup data identifies the increasing outdoor enthalpy over time in the graph labeled one and the time-stepped enthalpy method in graph labeled two.

Outdoor enthalpy typically increases at a slow phase throughout the day as the solar energy delivered by the sun accumulates. Graph one in the test setup data figure shows where the enthalpy increases from approximately 50.56 kJ/kg to almost 51.14 kJ/kg in 23 minutes. Graph two has captured the time-stepped enthalpy method where the heat recovery system runs at full capacity from the first minute to minute 14 where the heat recovery energy is shut off. The time response of the system is seen as the energy is slowly removed from the system. It is the time difference between the energized and non-energized heat transfer state of the system that yields the pertinent delta enthalpy, thus given the name time-stepped enthalpy method. During the process it is very important that steady system state is reached to ensure that transient conditions that exit do not skew calculated results. As seen in Fig. 2, the delta enthalpy in this case would be calculated using time value 12 and 23.

4. FINDINGS

The initial objective was to perform only one series consisting of five independent measurements that would be recorded over eight hours. However, during the first test series the airflow volume errors observed were substantially higher than what was expected where the time stepped enthalpy readings were only within ±10% of actual value. This prompted an investigation and it was found that the fan motor belts had a substantial slip factor which resulted in supplying a varying airflow volume.

After the belt slippage issues were resolved by the manufacturer a new measurement series was performed. One of seven repetitive tests (Fig. 3) depicts the calculated airflow volume over time. The initial values from minute one through six are the earlier discussed transients before system steady state had been reached. Steady state occurs after the six minute interval where the airflow converges towards 2265 cubic meters per minute (CMM). It is also important to notice that if the time frame of the test is expanded, less accuracy will result due to larger variances in measurement parameters such as the outdoor enthalpy, heat tarnsfer in the system and building heat load.

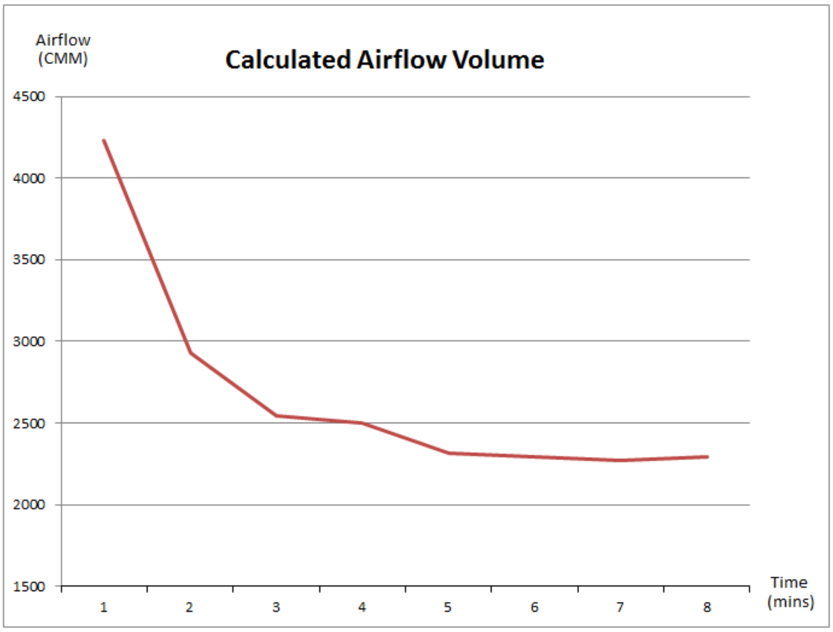

Fig. 3. Using the time stepped enthalpy method the airflow volume is calculated and plotted. Identified, the system reaches steady state where the airflow converges at 2265 CMM.

A summary of the time-stepped enthalpy method result is presented in table 1. The airflow volume in this study was approximately 2336 CMM so the deviations in column three (in CMM) and four (in percent) were calculated in relation to the actual value. From the data and independent of the increasing outdoor enthalpy and variations inside the building the time-stepped enthalpy method yields reasonable and accurate airflow volumes as listed in column 2.

Test	Measured Airflow (CMM)	Airflow Deviation (CMM)	Airflow Deviation (%)
1	2293	-43	-2%
2	2424	88	4%
3	2404	68	3%
4	2375	39	2%
5	2337	130	0%
6	2421	85	4%
7	2403	67	3%

Tab.1. The summary of the seven repetitive tests is presented where the measured airflows are listed in column two, the deviation in CMM in column three and the deviation percent from 2336 ±1% CMM in column four.

These results are extremely good even as the delta enthalpy in this heat recovery system is very low, which should be noted is typical for this type of equipment. Test data with low delta enthalpy is very susceptible for producing a very large swing in the calculated airflow volume. Small parameter variance can be seen in figure 2 where the outdoor enthalpy changes 0.58 kJ/kg in only 23 minutes with small ripples throughout the range. The best way to illustrate this relationship is to plot an error function for the total heat formula shown in equation 2. The percentage error for a delta enthalpy of 2.32 kJ/kg due to the inverse multivariable nature of the equation translates into a 10% error with a variation of only 0.23 kJ/kg. From the Fig. 4 it can be seen that the expected accuracy with a 0.23 kJ/kg variance converges to 1% error when the delta enthalpy is 23.24 kJ/kg.

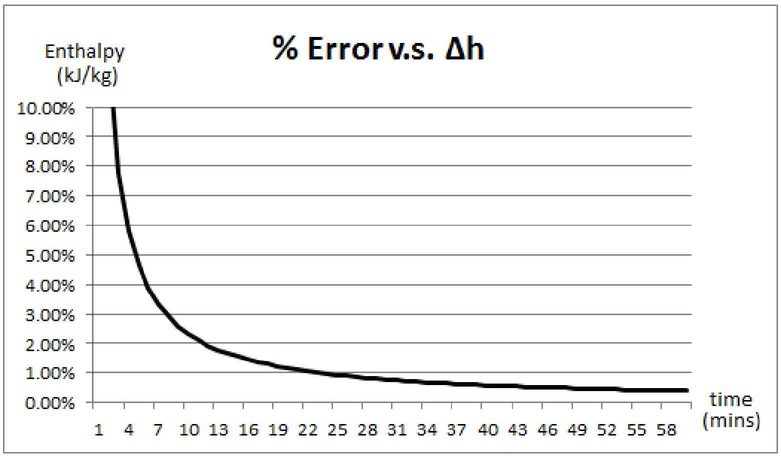

Fig. 4. The airflow error percentage converges to less than one percent for a delta enthalpy above 23.24 kJ/kg when experiencing a 0.23 kJ/kg variation over time.

The delta enthalpy in this study was in the range from 2.32 kJ/kg to 3.25 kJ/kg which makes the airflow volume calculations very susceptible to large volume swings due to these small enthalpy variances. To combat this intrinsic sensitivity of the time-stepped enthalpy method the objective in any assessment of HVAC systems would be to maximize the delta enthalpy to minimize the associated error.

5. CONCLUSIONS

This field study provides a major step towards a quick and accurate measurement of airflow volume that may also be valid for even very small delta enthalpy environments. The airflow volume is one of several important parameters needed to compare actual HVAC performance to the intended design. It was shown that the method accuracy to be around ±4 percent even with delta enthalpy in the range of 0.23 kJ/kg to 2.32 kJ/kg . This is significant as enthalpy variances of 0.23 kJ/kg would alone be the cause of an inaccuracy equal to 10% for a delta enthalpy of 2.32 kJ/kg. Furthermore, the time-stepped enthalpy method converges below 1% accuracy with 0.23 kJ/kg enthalpy variance when the Δh is 23.24 kJ/kg or more. The end result of the time-stepped enthalpy method will be substantial when trying to achieve large reductions in the buildings energy use and along with improving the buildings indoor air quality through proper ventilation, thermal comfort and advanced latent heat transfer management. Although this study focused on an energy reclaim system the time stepped enthalpy method is applicable to any air handler unit of any fuel or energy source and climate control. Furthermore it can also be used on vav boxes if such is equipped with a reheat coil.

6. ACKNOWLEDGMENT

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