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AN EQUILIBRIUM CALORIMETRY METHOD TO MEASURE VENTILATION RATE IN AN AIRSPACE

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SUMMARY

Air quality within an airspace largely depends on the ventilation rate of the airspace. In many cases, air flow rates of microenvironments (e.g., zones within a room) are unknowns and vary considerably from mean room (macroenvironmental) conditions. Yet, appropriate ventilation of the microenvironment is more critical because the occupants are directly affected by the microenvironmental conditions in the occupied zone.

An equilibrium calorimetry method to estimate the ventilation rate in an airspace is described. The validity of the theory was verified by controlled laboratory chamber experiments and is described in this paper. The method is harmless to the occupants of the airspace and suitable for small airspaces that are not bounded by solid boundaries.

The equilibrium calorimetry method can not only be applied to enclosed mechanically ventilated airspaces, but also to those less enclosed airspaces such as a room with a door open or a zone within a room. The method is more appropriate for small airspaces (e.g., zones within a room) where thermal capacitance effects are of little influence.

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INTRODUCTION

Air quality within an airspace largely depends on the ventilation rate of the airspace. As part of routine trouble shooting procedures, it is important to know whether or not an unacceptable environment is due to an inadequate or excessive ventilation rate. It is also necessary in many applied research projects to measure or estimate the air flow rate to calculate energy and mass balances.

In many cases, air flow rates of microenvironments (e.g., zones within a room) are unknowns and vary considerably from mean room (macroenvironmental) conditions. Designs of ventilation systems are nearly always based on the macroenvironment with the assumption that room air is well mixed. Measurements verify that the well mixed assumption is frequently not valid. Yet, appropriate ventilation of the microenvironment is more critical because the occupants are directly affected by the microenvironmental conditions in the occupied zone.

Many methods have been developed to measure or estimate air flow rate through an enclosed airspace. The most common method is to use a flow measuring meter attached either upstream or downstream of each supply or exhaust fan [1, 2]. Use of hot-wire anemometers and tracer gases to measure air flow in ducts downstream of exhaust fans is another alternative [3]. A third method is to monitor the ventilation rate by measuring the rotation speed of exhaust fans and the static pressure difference across the fans and then use fan data to estimate the air flow rate [4]. All these methods require that the airspace be enclosed so that ducts and instrumentation can be installed to conduct the measurement. It is impossible to use these methods to measure the air flow rate of a less confined airspace (e.g., a naturally ventilated building) because it is difficult to locate the sensing points.

Another well developed method used to measure and estimate the building air exchange rates is to perform tracer gas experiments. Hitchin and Wilson [5] successfully used this method in industrial and residential buildings to estimate ventilation rate. Leonard et al. [6] used the tracer gas method for estimating ventilation rates in livestock buildings. Ventilation rate can also be estimated by measuring the difference of carbon dioxide concentration assuming that the average carbon dioxide production rate inside the airspace is known [7]. The reported tracer gas methods require complicated instrumentation and an expensive gas analyzer. Moreover, use of the tracer gas method is limited or prohibited in airspaces (e.g., laboratory animal rooms) where occupants can not be disturbed.

Similar to the rate-of-decay tracer gas method, ventilation rate can be estimated by monitoring temperature changes during the heating and cooling periods of a heating/cooling cycle [8]. The method requires simple measurement instrumentation and is harmless to the occupants of the airspace. However, the estimated ventilation rate is sensitive to the time constant of the heating source used for the step change of heat input. The estimated ventilation rate is also influenced by the time constants of the building shell and materials within the airspace.

The objective of this paper is to describe an equilibrium calorimetry method to estimate the ventilation rate in an airspace. The method is harmless to the occupants of the airspace and suitable for small airspaces that are not bounded by solid boundaries. The theory is described and the validity of the method is verified by laboratory experimentation.

DEVELOPMENT OF THEORY

An airspace can be treated as a single bounded control volume (Figure 1). The incoming air to the airspace is characterized by a mass flow rate, m_i, and an enthalpy, h_i. Similarly, the outgoing air from the airspace is denoted as a mass flow rate, m_o, and an enthalpy, h_o. The total mass of moist air within the airspace is given by M. Assuming the air within the airspace is uniform in temperature and humidity distribution, then the enthalpy of the airspace is h, and the total energy within the airspace is uniquely defined by hM.

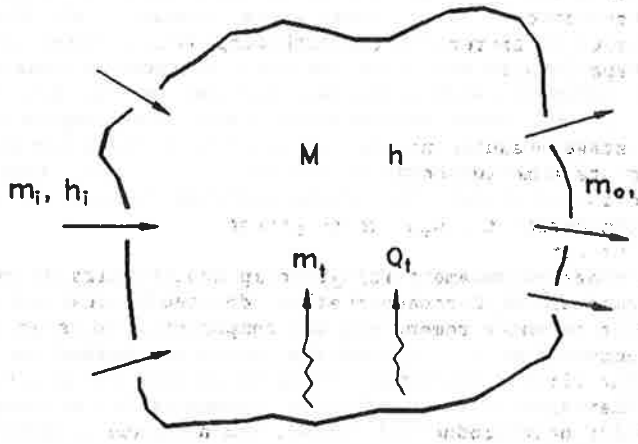

Figure 1. Sketch of mass and energy exchange for an airspace which mass and energy conservation laws apply.

Considering that mass and energy produced within the airspace are at a net rate of m_t and Q_t, respectively, and noting that h and h_o are equal because the air within the airspace is uniform, then applying the principles of mass and energy conservation results in two differential equations:

$$m_i - m_o + m_t = \frac{dM}{dt} \tag{1}$$

$$h_i m_i - h m_o + Q_t = \frac{d(hM)}{dt} \tag{2}$$

where all symbols are defined in table 1.

Table 1. Nomenclature

A	-	Surface area of airspace shell	(m^2)
h	-	Enthalpy of air	(kJ/kg)
M	-	Total mass within air space	(kg)
m	-	Mass air flow rate	(kg/h)
n	-	Total number of components of airspace shell	
Q	-	Heat production rate	(kJ/h)
Q_o	-	Heat production of occupants within airspace	(kJ/h)
q	-	Volumetric air flow rate	(m^3/h)
S_c	-	Heat source or sink within airspace	(kJ/h)
T	-	Dry bulb temperature	(°C)
U	-	Heat conductance	(kJ/hm²C)
v_a	-	Specific volume of air	(m^3/kg)
W	-	Humidity ratio	
ΔT_j	-	Temperature difference across jth component of airspace shell	(°C)
Subscripts:			
1	-	First stage measurement	
2	-	Second stage measurement	
c	-	Conduction	
cj	-	Heat conduction through jth component	
i	-	Incoming air	
j1	-	First stage measurement for jth component of airspace shell	
j	-	jth component of airspace shell as denoted in text	
j2	-	Second stage measurement for jth component of airspace shell	
m	-	Management	
o	-	Outgoing air	
s	-	Supplemental	
t	-	Total net heat production rate within airspace	

At a steady state, there is no change of the mass and energy content within the airspace. Therefore, the differential terms at the right sides of equations 1 and 2 are zero. Assuming that the production rate of mass (e.g., dusts, airborne micro organisms and moisture) in the airspace is negligible compared with the total mass of the airspace air, Equation 1 becomes:

$$m_i = m_o = m \tag{3}$$

Substituting Equation 3 into Equation 2 yields:

$$m(h_i - h) + Q_t = 0 \tag{4}$$

The net energy production rate, Q_t, is composed of several sources within the airspace: heat production rates of equipment such as fans and lights, Q_e, heat loads of occupants within the airspace, Q_o, heat loads of management activities, Q_m, and heat conduction rate through the airspace shell, Q_c.

Although the ratio of sensible to latent heat produced by the occupants will change when temperature changes, the total heat production of the occupants, Q_o, will remain relatively constant if the temperatures are within the thermoneutral range [9, 10]. Assuming that the heat production of equipment and management activities are constants, then Q_t is given by Equation 5:

$$Q_t = S_c - \sum_{j=1}^{n} A_j U_j \Delta T_j \tag{5}$$

where $S_c = Q_e + Q_o + Q_m$ and subscript j refers to the jth component (e.g., a wall or a ceiling) of the airspace shell. Substituting Equation 5 into Equation 4, gives the equilibrium energy equation of an airspace:

$$m(h_i - h) + S_c - \sum_{j=1}^{n} A_j U_j \Delta T_j = 0 \tag{6}$$

There are three terms on the left side of Equation 6. The first term represents the heat transfer through convection, the second the net heat production within the air space and the third the heat transfer through conduction. Assuming that the heat conductances, U_j, are known and A_j, ΔT_j can be measured, then the heat transfer rate through conduction, Q_c, can be calculated. With incoming air temperature (T_i) and humidity (W_i), and outgoing air temperature (T_o) and humidity (W_o) measured, enthalpies of the incoming and outgoing air, h_i and h_o, respectively, can be calculated using the following Equation [11]:

$$h_i = 1.006T_i + W_i(2501 + 1.775T_i) \ \left(\frac{kJ}{kg}\right) \qquad (-50°C < T_i < 110°C) \tag{7}$$

$$h_o = 1.006T_o + W_o(2501 + 1.775T_o) \ \left(\frac{kJ}{kg}\right) \qquad (-50°C < T_o < 110°C) \tag{8}$$

$$\Delta h = h_o - h_i \tag{9}$$

Therefore, there are only two unknowns, m and S_c, in Equation 6. To solve the unknowns, m and S_c, two independent equations are required. Assuming that the airspace is at an equilibrium state 1 with S_c, all other parameters are measured as T_{I1}, T_1, W_{I1}, W_1 and ΔT_{j1}. Then apply a supplemental heat input, Q_s, with known capacity to the airspace. The airspace will reach a second equilibrium state 2 with T_{I2}, T_2, W_{I2}, W_2 and ΔT_{j2}. Incorporating these parameters into Equations 7, 8 and 9 to calculate h_I, h, Δh, and then substituting Δh into Equation 6, gives a pair of equations:

$$-m\Delta h_1 + S_c - \sum_{j=1}^{n} A_j U_j \Delta T_{j1} = 0 \tag{10}$$

$$-m\Delta h_2 + S_c + Q_s - \sum_{j=1}^{n} A_j U_j \Delta T_{j2} = 0 \tag{11}$$

Subtracting Equation 10 from Equation 11 and solving for the mass flow rate of the air space, m, gives:

$$m = \frac{1}{\Delta h_2 - \Delta h_1} [Q_s - \sum_{j=1}^{n} A_j U_j (\Delta T_{j2} - \Delta T_{j1})] \tag{12}$$

Because the air mass flow rate $m = q/v_s$, where v_s is the specific volume of air within the airspace, Equation 12 can be rewritten in terms of volumetric air flow rate:

$$q = \frac{v_s}{\Delta h_2 - \Delta h_1} [Q_s - \sum_{j=1}^{n} A_j U_j (\Delta T_{j2} - \Delta T_{j1})] \tag{13}$$

The volumetric air flow rate, q, corresponds to the specific volume, v_s (e.g., if incoming air specific volume, v_{si}, is used, q will be incoming volumetric air flow, q_i). From the mass conservation defined by Equation 3, the outgoing volumetric air flow, q_o, can be calculated :

$$q_o = \frac{v_{so}}{v_{si}} q_i \tag{14}$$

EXPERIMENT AND RESULTS

The technique developed in the previous section for estimating the ventilation rate of an airspace was tested in a laboratory chamber. The following test and calculation procedures can also be considered as an example of the application of the equilibrium calorimetry method.

Laboratory chamber

The chamber used for this experiment (Figure 2) had inside dimensions of 3.05 X 2.44 X 2.40 m (10 x 8 x 7.8 ft), giving an interior surface area A = 41 m^2 (442 ft^2), and a volume 17.8 m^3 (627 ft^3). The chamber is located within the Bioenvironmental and Structural Systems Laboratory (BESS) which is a large airspace with an air conditioning system. The supply air was maintained at a constant temperature of 22°C and a relative humidity of 32%. Air entered the chamber through a slotted air inlet at an end wall. The air was exhausted by a centrifugal fan with a flow measurement section upstream from the fan. The exhaust fan was maintained at a static pressure of 147 Pa (0.58 inch of water column) to ensure accurate measurement of the air flow rate. A small oscillating propeller fan was operated continuously inside the chamber to ensure complete mixing.

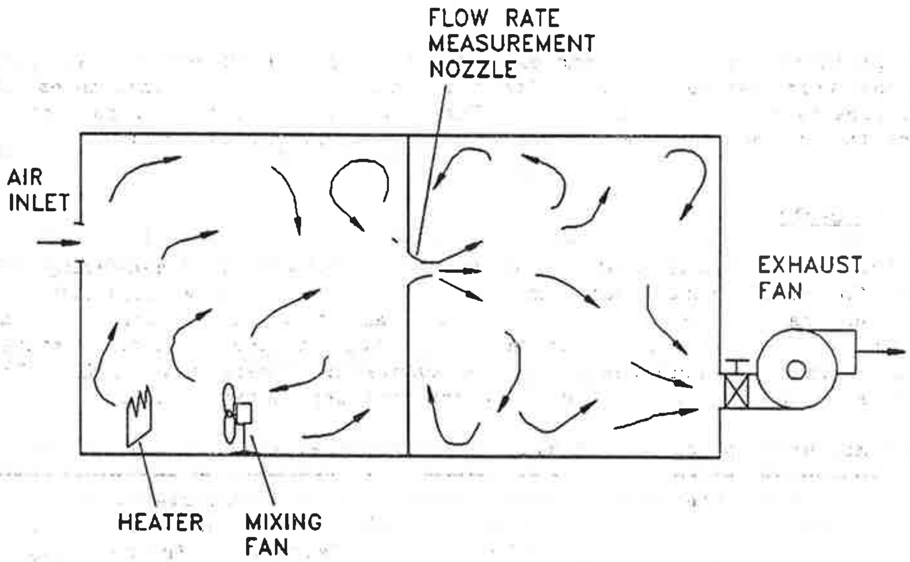

Figure 2. Sketch of laboratory chamber used to evaluate the equilibrium calorimetry method for estimation of zonal air flow rate.

Calibration of heat conductance of the chamber shell

The chamber was made of 2 cm (0.75 in) thick plywood supported with 5 X 10 cm (norminal 2 x 4 in) trusses in 30 cm (1 ft) intervals. Although the heat conductance of the chamber material can be estimated using available information [12], it is appropriate to calibrate the heat conductance of the chamber shell because the exterior surfaces of the chamber are bounded with supporting trusses and equipped with instrumentation. These irregular surfaces will introduce an error source in the convective heat transfer coefficient and, in turn, an error of heat conductance.

The calibration was conducted by putting an electrical heater (Q_s = 1.1 kW) into the sealed chamber airspace until the air temperature within the airspace reached an equilibrium state. Since the airspace was completely enclosed, there was no heat loss through convection. All supplemental heat (1.1 kW) was dissipated by heat conduction through the chamber shell. At the equilibrium state, air temperature (T) within the chamber was 38.6°C and ambient air temperature (T_i) was 22.2°C. There was no moisture production within the air space, hence the humidity ratio $W = W_i = 0.005$. The average heat conductance U for the chamber shell was calculated using

$$U = \frac{Q_s}{A(T - T_i)} = 1.636 \frac{W}{m^2C} \qquad (15)$$

In practice, an air space enclosure consists of n components such as walls and ceilings which may be made of different materials. The heat conductances for these components are usually different. Therefore, heat loss through conduction should be the sum of heat transfer through these different components.

Experiment results:

After the calibration of the chamber, a test was performed using the chamber. The supplemental heat source (Q_s) was 1.1 kW. The ventilation rate through the chamber was measured using an mass flow rate nozzle and was controlled at 114 m³/h (67 cfm). Stages 1 and 2 refer to the equilibrium states before and after the application of the supplemental heat source, Q_s. The measured data and calculated values from the test are listed in Table 2.

Table 2. Measured and calculated data from chamber experiment

Stage	Measured						Calculated				
	T_i (°C)	W_i	T (°C)	W	A (m²)	Q_s (kJ/h)	h_i (kJ/kg)	h (kJ/kg)	Δh	Q_c (kJ/h)	v_s (m³/kg)
1	22.8	0.005	22.8	0.005	41	0	35.1	35.1	0	0	0.85
2	22.8	0.005	33.2	0.005	41	3960	35.1	46.4	11.3	2512	0.875
Measured air flow rate through chamber:									114 m³/h (67 cfm)		

Substituting the data in table 2 into Equation 13, gives an outgoing volumetric air flow rate:

$$q_o = \frac{0.875}{11.3 - 0}(3960 - 2513) = 112 \ \frac{m^3}{h} \qquad (16)$$

Compared with the measured outgoing volumetric air flow rate of 114 m³/h (67 cfm), the calculated value is sufficiently accurate for most engineering applications.

APPLICATION AND LIMITATIONS

Estimation of the air flow rate using Equation 13 requires the measurements of the following variables:

- temperature of incoming air;
- humidity ratio of incoming air;
- air temperature within the airspace;
- humidity ratio of air within the airspace;
- area of the airspace shell;
- heat conductance (U) or resistance (R) value of the airspace shell;
- capacity of the supplemental step-change heat input (Q_s).

If the moisture production rates do not change during the two equilibrium states, humidity ratios of the incoming and the inside air will be the same and the enthalpy difference ($\Delta h_2 - \Delta h_1$) can be calculated from the temperature difference. Therefore, the measurements of air humidity can be omitted.

Since measurements are taken after the system reaches equilibrium, estimation of air flow rate is independent of heat capacitances of the building shell and other materials within the airspace. These heat capacitances are error sources when the dynamic calorimetry method is used [8].

The equilibrium calorimetry method can not only be applied to enclosed mechanically ventilated airspaces, but also to those less enclosed airspaces such as a room with a door open or a zone within a room. The method is most accurate for small airspaces (e.g., zones within a room) where thermal capacitance are effects of little influence. Large airspaces usually have large heat capacitances which can result in failure to reach an equilibrium state because of diurnal temperature fluctuation.

The capacity of supplemental step-change heat input, Q_s, is one of the key parameters of the measurements and should be carefully selected. Too small of Q_s increases error due to instrumentation sensitivity. Too large of Q_s can distort air exchange rate measurements because the thermal buoyancy force becomes a significant driving force for the zone air exchange rate. The temperature increase should be estimated using Equation 13 to approximate that which will occur due to occupants.

In case of an open space, Q_s should be small so the temperature difference can be kept at a low value. A large temperature increase will result in a significant buoyancy force and change the flow patterns within the airspace. To reduce the effect of the buoyancy force, it is desired to empty the airspace to do the stage 1 measurements, and then perform the second stage measurements by applying a Q_s which has an approximately the same heat capacity as the heat production rate within the airspace, S_c.

CONCLUSIONS

The equilibrium calorimetry method for estimating the air flow rate through an airspace is derived from the laws of mass and energy conservation. The validity of the theory was verified by a controlled laboratory chamber experiment and the test results are satisfactory. Accuracy of the method depends on the accuracy of temperature and humidity measurements, and calculation of heat transfer rate through the airspace shell.

The supplemental step-change heat input (Q_s) should be selected to avoid an undesired temperature increase during the measurements of stage 2. Generally, Q_s should be small for open airspaces so the temperature difference is small to reduce the effects of buoyancy. For enclosed airspaces, temperature increase should be estimated and Q_s should be reasonably sized so as to avoid unacceptably hot environments for the occupants.

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