Influence of Heat Load on Selection of Laboratory Design Parameters and Dynamic Performance of Laboratory Environment

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ABSTRACT

The laboratory heat load is a significant contributor of internal heat gains. The heat load is usually generated by instrumentation, experimental equipment, furnaces, etc. The sensible to total heat ratio in laboratories is high. The magnitude of heat load varies from laboratory to laboratory. In addition, the rate of heat generation may be rapid and extensive. For example, in a matter of seconds the heat load may jump from 5 W/ft^2 to 60 W/ft^2.

The objective of a laboratory control system is to maintain both temperature and pressure at desired settings. Both temperature and pressure are interdependent and they also depend on the other control variables such as air flow rates and discharge air temperature. Therefore, an understanding of physical laws and how they couple laboratory temperature and pressure with the internal load is important. The coupling exists for both steady-state and transient conditions. The steady-state coupling is significant in selecting laboratory design parameters, while the transient aspect of internal load will influence the dynamic performance of laboratory temperature and pressure.

This paper describes the mathematical models related to the laws of conservation to predict laboratory temperature and pressure. Steady state results are presented and discussed to demonstrate the relationship between the control parameters and the heat load. Finally, the simulated results are included to predict the transient effect of heat load on the dynamics of the laboratory environment.

INTRODUCTION

Laboratory internal heat generation is an important variable with respect to the design and operation of a laboratory heating ventilating and air-conditioning (HVAC) system. The magnitude and dynamics of heat generation in a laboratory can be significantly different than in a commercial space. Besides occupant and lighting loads, the heat can be generated due to exothermic reactions and use of laboratory equipment (i.e., autoclave, furnace, etc.). Neuman (1989) mentions that laboratory heat generation varies from 5 W/ft^2 to as high as 60 W/ft^2. Moreover, the heat load can change drastically within a few seconds. Traditionally, research has focused on identifying laboratory heat-generating equipment and the amount of heat the equipment generates (ASHRAE 1991). Such information has value in designing and selecting laboratory HVAC systems and components. However, from a control point of view, it is important to know the impact of laboratory internal load on space design variables such as room pressure differential and temperature. Most laboratory control strategies either use the direct pressure differential across the room or the differential volumetric flow rate between total laboratory exhaust and supply air as control inputs to maintain desired laboratory pressure. For temperature control, reheat is usually provided to the supply air. Published literature has discussed control strategies for various types of laboratory HVAC systems including variable-air-volume (VAV) and constant-air-volume (CAV) systems. However, the literature does not address correlating the differential pressure, differential flow, and supply air temperature to the internal load. This correlation is needed to select the correct design variable setpoints.

The transient effect of internal load dynamics on control variables is equally important. This is especially true in a VAV laboratory where the control variables are modulated in order to maintain design variable setpoints. For laboratory pressure, the supply air flow rate is a common control variable whereas the supply air temperature is modulated to maintain room temperature. In a VAV system, the general exhaust flow rate is also used as a control variable for room temperature. No matter what control variable is used, the dynamic effect of a disturbance on each control variable is important. Use of such information will aid in the proper selection of control components (i.e., sensors, dampers, and valves) in order for the control system to respond to a disturbance such as a sudden jump in internal load.

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Based on these factors, two observations can be summarized. The correlation between the magnitude of internal load and space variables (i.e., temperature and differential pressure) is to be carefully analyzed for a well-designed lab HVAC system. The transient information is useful for proper selection and operation of an HVAC control system.

This paper uses previously published models (Ahmed ate al. 1993) to correlate design variables with the internal load. The models are then simulated to investigate the transient effects on laboratory space variables due to sudden changes in internal load. Finally, the importance of the control system on handling a transient internal load is discussed by choosing different trajectories of control variables. The method and analysis are valid for all such spaces in which the pressure is kept higher than the adjacent space (i.e., a clean room.)

MODEL DEVELOPMENT

The details of the model development are described in Ahmed et al. (1993) and summarized below. An additional simple term to compute heat transfer through the laboratory envelope is added for completeness of the room model. The envelope model is based on considering a single equivalent thermal mass to the four walls, ceiling, and floor. Instead of treating a complicated model, a single simple envelope model serves the purpose of illustrating the influence of heat load on room dynamics.

The models are developed based on the following assumptions:

- air is an ideal gas,
- laboratory temperature and pressure are spatially uniform,
- adjacent space temperature and pressure are constant,
- supply duct pressure is constant,
- internal heat generation is purely sensible, and
- wall conduction is one dimensional.

The control volume is the air in the laboratory space. The conservation of mass principle applied to the air yields

$$\frac{dm}{dt} = \dot{m}_i - \dot{m}_e . \tag{1}$$

Also, the mass flow rate can be related to the volume flow rate as $\dot{m} = \dot{v}\rho$.

The equation of state for an ideal gas relates the temperature, pressure, and mass as

$$Pv = mRT \tag{2}$$

and the internal energy to temperature as

$$du = c_v dt \text{ and } dh = c_p dt . \tag{3}$$

Using the equation of state, Equation 1 becomes

$$\frac{d(PV/RT)}{dt} = \frac{P_s \dot{v}_s}{RT_s} + \frac{P_{ad}\dot{v}_{ad}}{RT_{ad}} - \frac{P\dot{v}_e}{RT} . \tag{4}$$

Differentiating with respect to P and T,

$$V\left[\frac{1}{T}\frac{dP}{dt} - \frac{P}{T^2}\frac{dT}{dt}\right] = \frac{P_s \dot{v}_s}{RT_s} + \frac{P_{ad}\dot{v}_{ad}}{RT_{ad}} - \frac{P\dot{v}_e}{RT} . \tag{5}$$

The conservation of energy principle applied to the air in the room is

$$\frac{dU}{dt} = h_i\dot{m}_i - h_o\dot{m}_o + \dot{q}_{gen} + \dot{q}_{tr} . \tag{6}$$

Equation 6 can be further expanded by writing the internal energy as the product of mass and specific energy and differentiating by parts:

$$m\frac{du}{dt} + u\frac{dm}{dt} = h_i\dot{m}_i - h_o\dot{m}_o + \dot{q}_{gen} + \dot{q}_{tr} . \tag{7}$$

Introducing the equation of state, the inflows and outflows, and simplifying yields

$$c_v V\frac{dP}{dt} = \frac{P_S \dot{v}_s}{RT_s}h_s + \frac{P_{ad}\dot{v}_{ad}}{RT_{ad}}h_{ad} - \frac{P\dot{v}_e}{RT}h + \frac{\dot{q}_{gen}}{cf_1} + \frac{\dot{q}_{tr}}{cf_2} \tag{8}$$

where heat transfer across the wall is expressed in terms of the difference in temperature between the wall surface and air temperatures and a combined heat transfer coefficient:

$$\dot{q}_{tr} = A_w hc_{r,w}(T_w - T) . \tag{9}$$

Infiltration

The infiltration model is based on the assumption of orifice flow. The infiltration through envelope holes and cracks is represented using Equation 10. An equation of this type has been used in previous studies (Shah 1980; ASHRAE 1989).

$$\dot{v}_{ad} = K_{ad}(\Delta P)^n . \tag{10}$$

SINGLE EQUIVALENT WALL MODEL

Instead of treating each of the envelope components separately, an approximate equivalent thermal mass can be used. The heat exchanges between the room air and those from the ceiling, walls, and floor are combined. The basic method uses a thermal network as shown in Figure 1:

From an energy balance on the wall capacitance node, the wall surface temperature, T_w, can be expressed as,

$$C_w\frac{dT_w}{dt} = hc_{r,w}A_w(T - T_w) + R_{w,ad}(T_{ad} - T_w) \tag{11}$$

where the wall resistance, $R_{w,ad}$, is given by

$$R_{w,ad} = \frac{1}{U_{w,ad}} - \frac{1}{hc_{r,w}} . \tag{12}$$

In order to show the effect of changes of supply and exhaust air

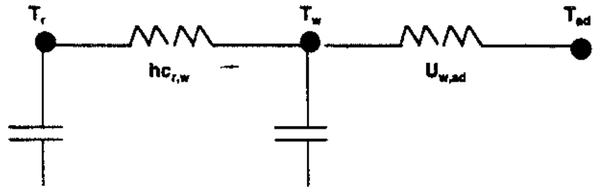

Figure 1

flow rates to sudden changes in heat load, the response is modeled as a simple first-order system shown below.

$$tau_d \frac{d\dot{v}}{dt} + \dot{v} = \Delta\dot{v}_{csp} \qquad t \le d \tag{13}$$

$$tau \frac{d\dot{v}}{dt} + \dot{v} = \Delta\dot{v}_{csp} \qquad t \ge d \tag{14}$$

The response is split in two time domains: before and after the dead time (*d*). The tau_d and *tau* are the system time constants for before and after dead time (*d*) respectively. By choosing *d* and suitable values for tau_d and *tau* for supply and exhaust flow rate, it is possible to simulate different responses of both supply and exhaust air systems.

Model Parameters

The following is a summary of the model parameter assumed for the simulation and for the design process.

$$K = 1000\frac{\text{cfm}}{(\text{w.c.})^n}\left(1165.789\frac{\text{L/s}}{(\text{kPa})^n}\right)$$

$n = 0.65$
$P_s = 408.00$ w.c. (101.526 kPa)
$P_{ad} = 408.00$ w.c. (101.526 kPa)
$T_{ad} = 70°\text{F}$ (21°C)

$$hc_{r,w} = 1.46\frac{\text{Btu}}{\text{ft}^2 \cdot \text{h} \cdot °\text{F}}\left(8.289\frac{\text{W}}{\text{m}^2 \cdot \text{K}}\right)$$

$$U_{w,ad} = 0.30\frac{\text{Btu}}{\text{ft}^2 \cdot \text{h} \cdot °\text{F}}\left(1.7034\frac{\text{W}}{\text{m}^2 \cdot \text{K}}\right)$$

$$Tau_w = 15{,}000\frac{\text{Btu}}{°\text{F}}\left(28{,}467\frac{\text{kJ}}{\text{K}}\right)$$

Base laboratory module: 30 ft x 25ft x10ft (9.15 m x 7.62 m x 3.048 m)
Maximum fume hood flow rate: 1,000 cfm (471.9 L/s)
Minimum fume hood flow rate: 225 cfm (106.2 L/s)
Number of fume hoods: Two

The basis of selecting the model parameters is cited in Ahmed et al. (1993). Both the overall equivalent wall heat transfer coefficient, U_{ad} and thermal capacitance, C_w, are approximated assuming that the wall is made of eight inches of lightweight concrete.

Design Considerations

The design process for a laboratory HVAC system involves finding the minimum and maximum values of the physical variables that define the practical operating range of the HVAC system.The HVAC equipment is then selected and installed based on the design conditions and the information is passed to the control system to ensure that the operating conditions are achieved within the bounded maximum and minimum values. It is critical to the design process to define the bounds considering all factors that may influence the two extremes. For example, design information typically tells the maximum and minimum supply flow rates that can be expected. Accordingly, the sensors and the control components (i.e., damper/ actuator) are selected and the duct is sized to achieve an appropriate velocity for the given flow.

Figure 2 shows a schematic of a laboratory HVAC system showing all the physical variables. Providing safety and comfort are the two basic functions of a laboratory HVAC system. The safety constraint is satisfied by exhausting effluents through the fume hood exhaust ($\dot{v}_{ex}$) and by preventing any leakage from the laboratory. Keeping the laboratory pressure (*P*) lower compared to the adjacent space pressure (P_{ad}) will ensure that nothing leaks from the laboratory. In certain laboratories, however, the interior pressure is kept higher than the adjacent space ($P > P_{ad}$) to prevent the flow of any foreign particles from the adjacent space. The objective here is to keep the interior space free of impurities as much as possible, a requirement for a clean room.

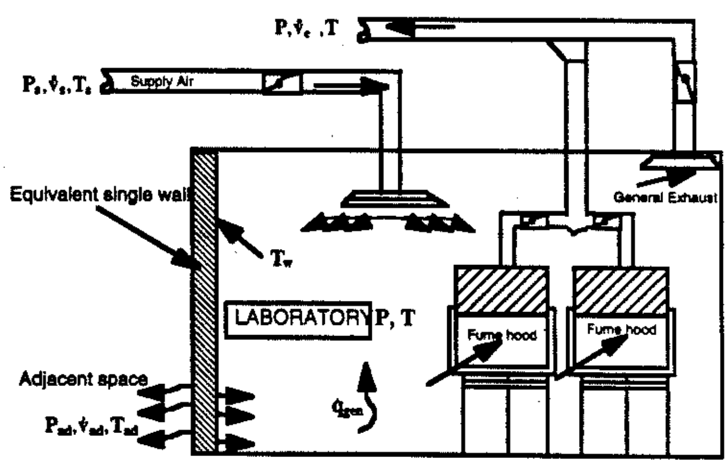

Figure 2 *Laboratory schematics showing physical variables.*

The pressure differential between the laboratory and the adjacent space is achieved by maintaining a difference between the supply flow rate and the total laboratory exhaust. The supply air ($\dot{v}_s$), usually discharged at 55 °F (T_s), is also meant to offset the cooling load. Since supply flow tracks the laboratory exhaust, when fume hood exhaust increases substantially, the supply flow rate may also increase more than what is needed for cooling. As a consequence, the supply air is tempered with the auxiliary heating typically provided by the duct heating coil. When fume hood exhaust is at a minimum, certain cooling loads may demand supply air more than the fume hood exhaust. However, supply flow cannot be increased without increasing the exhaust from the laboratory to maintain the negative pressure in the laboratory. Additional exhaust ($\dot{v}_{ex}$) from the laboratory takes place by opening the general exhaust damper under such a condition.The sequence of operation of the laboratory VAV HVAC system described here illustrates the intricacies in maintaining both safety and comfort constraints.

Internal heat generation is a significant factor in the determination of the value of supply flow rate for a design internal load and other laboratory physical variables. To illustrate the point, consider the process of designing a laboratory HVAC system. Referring to Figure 2, there are 11 independent physical variables that define the operating laboratory environment. Nine physical variables are readily identifiable as a set of independent variables: pressure, temperature, and volumetric flow rates associated with the supply, exhaust, and infiltrating air. In addition, the wall temperature and internal rate of heat generation are also physical variables. Seven more variables need to be assumed. Fortunately, it is not difficult to fix the values of seven variables within reasonable confidence.

For example, the adjacent temperature and pressure (T_{ad} and P_{ad}) are typically known, as are the supply air pressure and temperature, P_s and T_s. A good estimate of T_{ad} and P_{ad} will be 70°F and a pressure of 408 w.c., respectively P_s and T_s are typically maintained a bit higher than the room pressure and 55°F, respectively. The laboratory exhaust based on fume hood numbers and sizes is known. Hence, two more additional variables need to be estimated—the desired room temperature and pressure. The room pressure often is defined as a gauge pressure,ΔP, relative to the adjacent pressure, P_{ad}. A typical value of ΔP ranges from .01 to .05 w.c. (Hitchings 1994; Lacey 1994). Room temperature, T, of 70°F is a good choice. By approximating seven variables, four unknowns can be solved by considering three steady-state versions of Equations 5, 8, and 11 and infiltration Equation 10. The steady-state equations are presented here for clarity.

$$\frac{P_S\dot{v}_s}{RT_s}+\frac{P_{ad}\dot{v}_{ad}}{RT_{ad}}-\frac{P\dot{v}_e}{RT}=0 \qquad (15)$$

$$\frac{P_S\dot{v}_s}{RT_s}h_s+\frac{P_{ad}\dot{v}_{ad}}{RT_{ad}}h_{ad}-\frac{P\dot{v}_e}{RT}h+\frac{\dot{q}_{gen}}{cf_1}+\frac{\dot{q}_{tr}}{cf_2}=0 \qquad (16)$$

$$hc_{r,w}A_w(T-T_w)+R_{w,ad}(T_{ad}-T_w)=0 \qquad (17)$$

Table 1 shows the solution of Equations 15 through 17 and Equation 10 for design conditions for different values of exhaust flows. On the left-hand side the specified variables are noted, while on the right-hand side the calculated variables are listed. The laboratory exhaust, $\dot{v}_{ex}$, is based on approximate desired values of room air change rate per hour (ACH). In a design process the ACH requirements often appear as a safety constraint to satisfy ventilation standards. The results in Table 1 show that it is possible to calculate the maximum internal load for different values of ACH to maintain the design conditions of ΔP and T for specified conditions. The range of values of ACH is chosen to represent different laboratory operating conditions (Davis and Benjamin 1987; McDiarmid 1990). Once the internal load is known, the laboratory planner can be consulted to properly choose the laboratory equipment so that the maximum limit is not exceeded.

TABLE 1 Selection of Maximum Internal Load Based on ACH

Fixed Specified Variables

T = 70 °F (21 °c) P_{as} = 408.00 w.c. (101.526 kPa)
ΔP = .05w.c. (.01244 kPa) T_{ad} = 70 °F (21 °c)
P_s = 408.00 w.c. (101.526 kPa)

Assumed Variables			Calculated variables				
ACH	$\dot{v}_{ex}$ (cfm)	$\dot{m}_{ex}$ (lbm/min)	$\dot{q}_{gen}$ (W/ft²)	$\dot{v}_s$ (cfm)	$\Delta\dot{v}$ (cfm)	m_s (lbm/min)	$\Delta\dot{m}$ (lbm/min)
3.6	450*	34	2	307	153	24	10
5	625	46	3.047	468	157	36	10
16	2000**	149	11.735	1804	196	139	10
48	6000	447	37.008	5690	310	437	10

*Minimum total fume hood flow.
**Maximum total fume hood flow.

The differential flow rate, $\Delta\dot{v}$, between exhaust and supply increases as the heat load increases. The increase in heat generation can be viewed as additional supply flow rate, which requires increased laboratory exhaust to maintain the same pressure differential. However, the differential mass flow rate, $\Delta\dot{m}$, remains constant following the law of mass conservation. The difference between the total laboratory exhaust and maximum hood exhaust also sets the design flow rate for the laboratory general exhaust. Instead of using ACH and subsequently the total laboratory exhaust, the laboratory design conditions can be determined by setting the internal heat generation first and then the supply and exhaust flow rates to maintain the desired laboratory temperature and differential pressure. Of course, in this situation the calculated laboratory supply flow rate should be used to verify the constraint of laboratory ACH.

A volume-tracking control strategy is often used in the lab industry in order to maintain laboratory pressure at a prescribed value with reference to the reference pressure. The essence of this strategy is to maintain a fixed difference between the total laboratory exhaust and the laboratory supply flow rates. The difference in flow rates is equal to the infiltrating air (in case the laboratory pressure is kept lower than the reference pressure), which is a direct result of the pressure differential. Often a value of 200 cfm is chosen to be the differential flow rate. In reality, however, the laboratory pressure is coupled with the other physical variables and the leakage characteristics of the laboratory indicated as K. The effect of the rate of internal heat generation on fixed differential flow rate can be seen by examining the results in Table 2. Instead of fixing ΔP, the differential flow rate ($\Delta \dot{v}$) of 200 cfm is kept constant and the ΔP value is computed. All other fixed variables remain the same as shown in Table 1.

TABLE 2 Effect of Constant Differential Flow Rate on Design Conditions

Assumed variables			Calculated variables					
$\Delta \dot{v}$ (cfm)	$\dot{v}_{ex}$ (cfm)	$\dot{m}_{ex}$ (lbm/min)	$\dot{q}_{gen}$ (W/ft²)	$\dot{v}_s$ (cfm)	ΔP (w.c.)	$\dot{m}_s$ (lbm/min)	$\Delta \dot{m}_{in}$ (lbm/min)	$\dot{v}_{ad}$ (cfm)
200	450	34	1.626	250	.0793	19	15	192
200	625	47	2.764	425	.0761	33	14	187
200	2000	149	11.705	1800	.0525	138	11	147
200	6000	449	37.718	5800	.0477	447	2	31

By fixing $\Delta \dot{v}$, the ΔP value varied from 5% to 60% of the desired value of .05 w.c. in order to conserve the mass balance. The rate of infiltrating air, $\dot{v}_{ad}$, driven by ΔP also deviates from the fixed value of $\Delta \dot{v}$ of 200 cfm. The results demonstrate the flaw in assuming the differential flow rate, as actual ΔP and $\dot{v}_{ad}$ may vary significantly from the expected values. The significance of such departure depends on the critical nature of laboratory safety. For critical applications, where a precise ΔP value is to be maintained, any departure may jeopardize the safety constraint. A comparison between Tables 1 and 2 reveals more useful information. Comparing the values of $\dot{q}_{gen}$ and $\dot{v}_s$ between Tables 1 and 2, it is clear that an increase in $\dot{v}_s$ also increases $\dot{q}_{gen}$ and vice versa. This is obvious since less cooling will be available with a lower supply air flow rate to maintain a room temperature of 70°F and vice versa. In addition, the supply air flow rate is indirectly proportional to infiltration. A higher value of $\dot{v}_s$ means a higher mass flow rate, thereby reducing infiltration mass flow rate, $\Delta \dot{m}$. This is evident by comparing the $\Delta \dot{m}$ value of 2 lbm/min in Table 2 with the to 10 lbm/min in Table 1 for an ACH of 48.

A simple yet useful relationship between supply flow rate (or ACH) and internal heat generation can be determined by assuming that the room temperature equals the adjacent temperature, i.e., $T_{ad} = T$. With this assumption, Equation 16 can be rewritten as in Equation 18, setting $\dot{q}_{lr} = 0$ and using the ideal gas law for enthalpy.

$$\frac{P_S\dot{v}_s}{RT_s}C_pT_s + \frac{P_{ad}\dot{v}_{ad}}{RT_{ad}}C_pT_{ad} - \frac{P_S\dot{v}_s}{RT_s}C_pT - \frac{P_{ad}\dot{v}_{ad}}{RT_{ad}}C_pT + \frac{\dot{q}_{gen}}{cf_1} = 0 \quad (18)$$

Canceling similar temperatures in numerator and denominator, deleting terms with Pad and rearranging, Equation 19 can be expressed as,

$$P_S\dot{v}_sC_p - \frac{P_S\dot{v}_sC_pT}{T_s} + \frac{R\dot{q}_{gen}}{cf_1} = 0. \quad (19)$$

Writing Equation 19 in terms of two independent conditions, I and II, of the variables, dividing one by another and noting that R and Cf1 are constants, the following equation can be obtained.

$$\frac{\left[C_pP_s\dot{v}_s\left(\frac{T}{T_s}-1\right)\right]_I}{\left[C_pP_s\dot{v}_s\left(\frac{T}{T_s}-1\right)\right]_{II}} = \frac{[\dot{q}_{gen}]_I}{[\dot{q}_{gen}]_{II}} \quad (20)$$

Noting that values of all the variables remain constant for both conditions except for $\dot{q}_{gen}$ and $\dot{v}_s$, the relationship between heat generated and supply flow rate can be determined as linear or

$$\frac{\dot{v}_s|_I}{\dot{v}_s|_{II}} = \frac{\dot{q}_{gen}|_I}{\dot{q}_{gen}|_{II}} \quad (21)$$

or

$$\frac{ach|_I}{ach|_{II}} = \frac{\dot{q}_{gen}|_I}{\dot{q}_{gen}|_{II}}. \quad (22)$$

This relationship can be easily used in a design process to correlate the supply flow rate or room air changes per hour to internal heat generation if the values of such variables at some reference conditions are known. The proportionality relationship is clear from Table 1. For example, for the supply flow rate of 307 cfm, the value is 2 W/ft² Now, for $\dot{v}_s$ of 468 cfm or a 52% increase, the $\dot{q}_{gen}$ should also increase by same margin as per equation 21 or to 3.047 W/ft², as indicated in Table 1. Equation 21 is not applicable for values in Table 2 since the ΔP is not held constant.

Dynamic Simulation

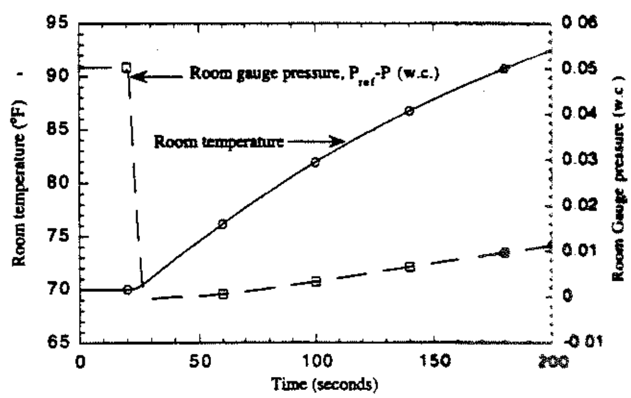

Figure 3 Room response due to sudden increase in internal load.

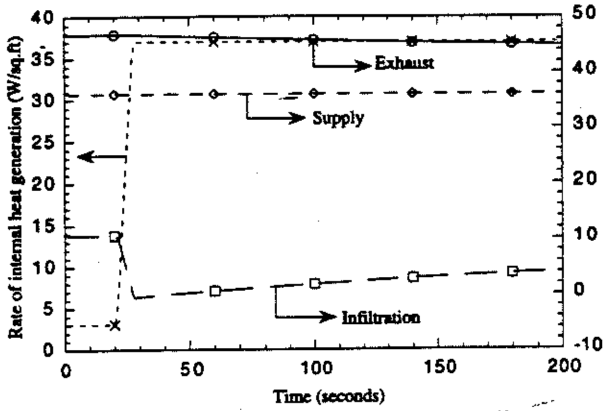

Figure 4 *Response of mass flow rates due to increase in internal load.*

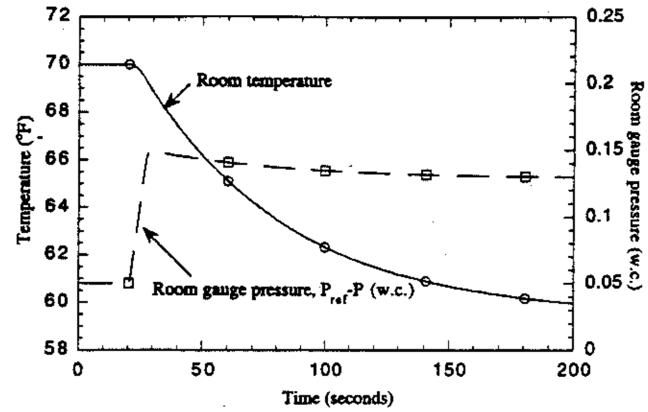

Figure 5 *Room response due to sudden decrease in internal load.*

Besides the influence on design, the sudden change in internal heat generation will also affect the room temperature and pressure. The control system needs to act fast enough to compensate for the increase or decrease in the rate of heat generation in order to maintain room temperature and pressure setpoints. Transient equations 5,8, and 11 and infiltration equation 10 are used to simulate the room temperature and room gauge pressure responses, which are shown in Figures 3 through 6. Figures 3 and 4 show the room temperature and pressure responses due to a sudden jump in the rate of internal heat generation, $\dot{q}_{gen}$, while Figures 5 and 6 show the responses when the rate of internal heat generation is decreased. The simulation is performed using an engineering equation solver (EES) package (Klein and Alvarado 1994).

The sudden jumps in the rate of internal heat generation may be viewed as additional supply flow rate at a constant temperature. As a consequence, the infiltration mass flow rate drops, causing the room pressure to increase in transient. Note that as room pressure increases, the gauge value of room pressure or ΔP (i.e., $P_{ad} - P$) decreases. As the rate of internal heat generation reaches a steady value, the room temperature rises steadily, the exhaust mass flow rate decreases slightly, and infiltration mass flow rate increases to balance the mass influx to the mass efflux. The effect of internal heat generation can be compensated by noting the room pressure deviation from the setpoint and decreasing the supply flow rate. However, the increase in room temperature calls for more cooling or more supply flow. The conflicting nature of supply flow requirements can be resolved by opening the general exhaust and then increasing supply flow to satisfy the room temperature and pressure requirements.

Figures 5 and 6 show the identical trend but in reverse when the rate of internal heat generation drops suddenly. Room pressure increases momentarily but then gradually decreases to a steady-state value. The room temperature declines gradually to a steady-state value. From a safety point of view, both increases and decreases in room pressure from its setpoint are not desirable.

When the room pressure is substantially lower than the adjacent pressure ($P << P_{ad}$), the high pressure gradient between the adjacent space and the room may cause leakage from the room. Such observation was made in earlier experiments (Ahmed and Bradley 1990; Knutson 1987). In addition, lower room pressure means the fume hood and general exhaust fans need to work hard to exhaust air at a low pressure.

Similar observations may be made when the room pressure is higher compared to the adjacent space ($P >> P_{ad}$), as a high pressure gradient between the room and the fume hood interior may facilitate leakage from the lab and around the fume hood. However, experimental work is needed to confirm the effect of high room pressure on lab and fume hood leakage. Also, the supply fan needs to compensate for increases in the room pressure. To sum up, the room pressure needs to be maintained at its desired value in order to avoid possible causes of leakage from the laboratory and the fume hood.

The trends of mass flow rates in Figure 6 are again just the opposite to those observed in Figure 4. The infiltration mass flow rate increases momentarily and then becomes steady as the heat generation becomes steady. Exhaust mass flow rate increases slightly with the gradual decline of the room temperature.

The analysis of the physical system response as observed in Figures 3 through 6 is important to probe the control system requirements. Given the observed response, a control system can be chosen such that the room temperature and pressure requirements are satisfied within the tolerance. The choice of a fast control system will avoid any significant excursion of room temperature and pressure, whereas, a slowly responding control system will be more tolerant of such excursions.

The effects of slow and fast control systems are shown in Figures 7 through 10. Figures 7 and 8 show the response of laboratory exhaust and supply flow rates. The control time constants tau_d and tau in Equations 10 and 11 are chosen as follows to create the response profiles.

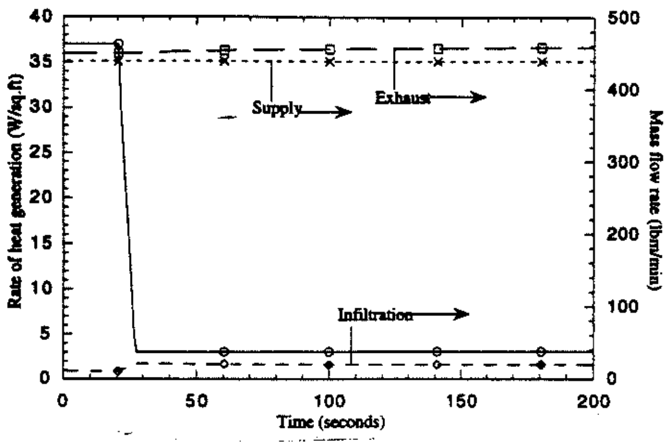

Figure 6 *Response of mass flow rates due to decrease in internal load.*

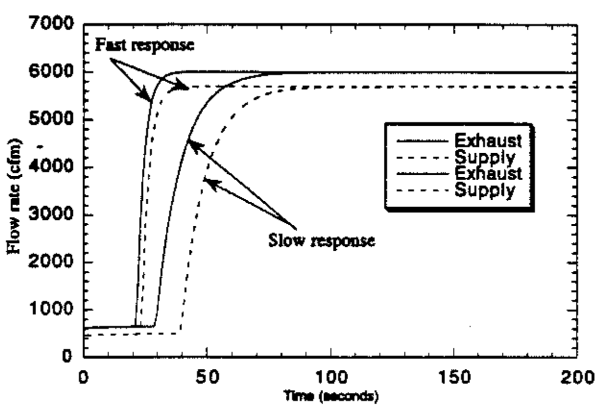

Figure 7 *Profiles of flow rates in response to increasing internal load.*

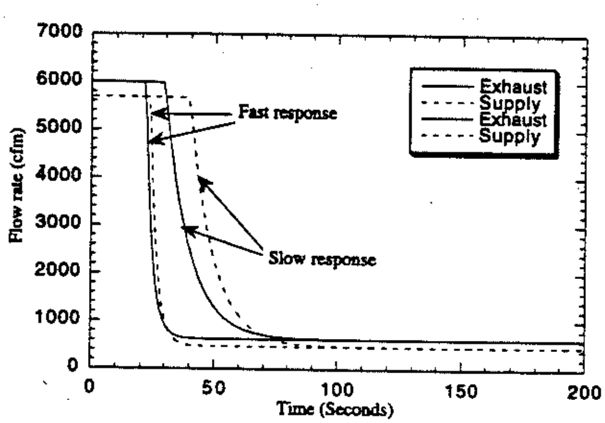

Figure 8 *Profiles of flow rates in response to decreasing internal load.*

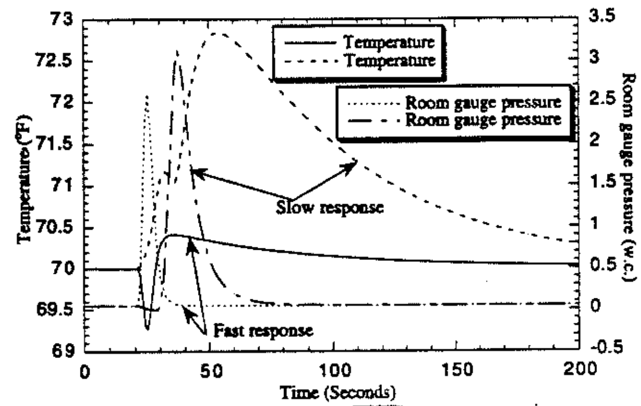

Figure 9 *Room pressure and temperature response due to slow and fast flow control to compensate increase in internal load.*

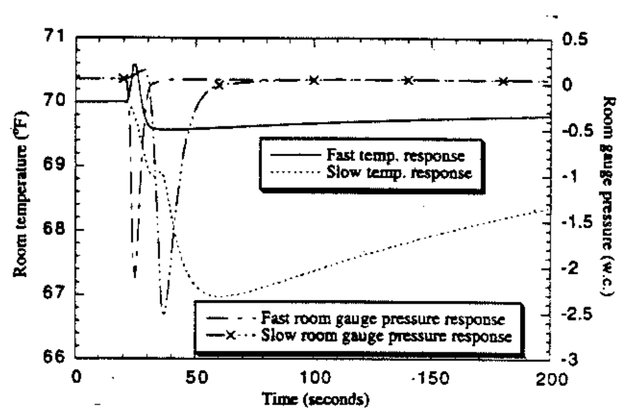

Figure 10 *Room pressure and temperature response due to slow and fast flow control to compensate decrease in internal load.*

	Fast Response		Slow Response	
	Exhaust	**Supply**	**Exhaust**	**Supply**
tau_d	600 s	600 s	600 s	600 s
tau	3 s	3 s	10 s	10s
d	0	3 s	9 s	15 s

The effects of fast and slow responses of flow rates are shown in Figures 9 and 10. Figure 9 represents the responses when the internal load is increased, while the responses due to a decrease in load are shown in Figure 10. The profile of internal load increase is same as that shown in Figure 4, while the decrease in internal load is depicted in Figure 6. The internal load in both cases varied from a minimum of 3.047 W/ft^2 to a maximum of 37.008 W/ft^2. For both these extremes, the corresponding required supply and exhaust flow rates are known from the steady-state solutions as to the sudden increase in internal load from 3.047 W/ft^2 to 37.008 W/ft^2, the exhaust flow rate will increase from 625 cfm to 6,000 cfm while the supply flow rate will increase from 468 cfm to 5,690 cfm.

By choosing a fast-reacting control system it is possible to maintain the room temperature within ±0.5°F and bring back the ΔP to its setpoint within five seconds. By choosing the slowly responding control system the temperature deviated from the setpoint by more than 2.5°F, but more important it takes more than two minutes for the temperature to reach its setpoint. Similarly, the ΔP setpoint is reached one minute. after the change in the rate of internal heat generation is introduced.

CONCLUSIONS

A simple model is presented for analyzing the effect of internal load on both steady and transient response of laboratory variables such as temperature and gauge pressure. The steady analysis can be incorporated in the design process to select proper HVAC systems and HVAC control systems. Conversely, by knowing the limits of the internal load for a selected laboratory system, the laboratory planner can select the proper laboratory equipment. A simple correlation is found between the room ACH and $\dot{q}_{gen}$, which may be used effectively by knowing the ACH and $\dot{q}_{gen}$ values at some reference condition.

The dynamic simulation is useful in testing a laboratory control system to verify whether it will be able to handle sudden disturbances created by the change in internal load. For a given control system, it is possible to generate the response of supply and exhaust flow rates and then use them in the simulation model. For a critical laboratory application, simulation results can be used to properly select the laboratory control system. The prior simulation is a cost-effective means of determining the response profiles and evaluating system performance. Often, the system performance is detected and corrected by means of costly commissioning or field services.

NOMENCLATURE

A = area, ft^2 (m^2)

ACH = room air changes per hour, $\frac{60\dot{v}}{V}$

Cf$_1$ = conversion factor = 0.09725

Cf$_2$ = conversion factor = 10.2877

C_v = specific heat constant volume, Btu/lbm·°F (kJ/kg·°C)

Cw = wall thermal capacitance, Btu/°F (kJ/K)

h = specific enthalpy of air, Btu/lbm (kJ/kg)

hc = convection heat transfer coefficient, Btu/ft^2·h·°F (W/m^2·h·°C)

m = mass, lbm (kg)

n = flow exponent

P = pressure, in. w.c. (kPa)

$\dot{q}_{gen}$ = rate of heat generation (W/ft^2)

$\dot{q}_{tr}$ = rate of heat transfer, Btu/ft^2·h·°F (W/m^2·h·°C)

R = gas constant for air, ft·lbf/lbm·°R (N.m/kg. °K)

ρ = density of air, lbm/ft$_3$ (kg/m^3)

T = temperature, °F (°C)

Tau = supply air system time constant after dead time, seconds

Tau_d = supply air system time constant before dead time (d), seconds

u = specific internal energy of air, Btu/lbm·°F (kJ/kg·°C)

U = overall heat transfer coefficient, Btu/ft^2·h·°F (W/m^2·h·°F)

$\dot{v}$ = volumetric flow rate, cfm L/s)

V = room volume, ft^3 (m^3)

Subscripts

ad = adjacent space

csp = control setpoint

e = exhaust

gen = generation

i = in

o = out

r,w = between room air and wall surface

rm = room

s = supply

w = wall

w,ad = between wall surface and the adjacent space

$w.c.$ = unit of pressure in inches of water column

REFERENCES

Ahmed, O., and S.A. Bradley. 1990. An approach to determining the required response time for a VAV fume hood control system. *ASHRAE Transactions* 96.

Ahmed, O., J.W. Mitchell, and S.A. Klein. 1993. Dynamics of laboratory pressurization. *ASHRAE Transactions* 99.

ASHRAE. 1989. *1989 ASHRAE handbook—fundamentals*, chap. 23. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, inc.

ASHRAE. 1991. *1991 ASHRAE handbook—HVAC applications*, chap. 14. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Davis, S.J., and R. Benjamin. 1987. VAV with fume hood exhaust systems. *Heating/Piping/Air-Conditioning*, August.

Hitchings, D.A. 1994. Laboratory pressurization *ASHRAE Journal*

Klein, S.A., and F.L. Alvarado. 1991. Engineering equation solver. Middleton, Wis.: F-Chart Software.

Knutson, G.W. 1987. Testing containment of laboratory hoods: A field study. *ASHRAE Transactions*.

Lacey, R. Laboratory design *ASHRAE Journal* 1994.

McDiarmid, M.D. 1990. Variable volume fume hoods: User experience in a chemistry research lab. *ASHRAE Transactions* 96.

Shah, M.M. 1980. Estimated rate of pressurization and depressurization of buildings. *ASHRAE Transactions*, 86(1): 251-257.

Neuman, V.A. 1989. Design considerations for laboratory HVAC system dynamics. *ASHRAE Transactions*, 95 (1).