Since the plant is a fully automated station, routine functions are performed by previously employed building service employee firemen. Additional service is provided on a call basis and is factored into the cost.

With severed interconnection to Con Edison, it was necessary for the plant to function efficiently and continuously from the moment it started. Normally utilities start up gradually over a long period of time to eliminate system defects. Big Six did not have that luxury. In addition, from the moment the plant began operating, it moved immediately into the hottest summer on record, incurring a high electrical demand, mostly for air conditioning, on an untried system. There were a few initial bugs in the system that caused problems but they were cleared up and the plant performed efficiently.

With twelve months of operation, the cogeneration concept is no longer an abstraction. The plant was built because the cost of energy had risen from approximately 5% of the operating budget when Big Six was first built in 1963, to approximately 30 % and still increasing, making it the single largest inflationary factor in the operating budget. The saving achieved is substantial. In these twelve months:

Consumption. 6,693,144
Cogeneration with heat recovery . . . .$306,858.
Utility Cost for comparable comsumption . . . . .$765,520.
Savings, exclusive of debt service on the plant. . .$458,662.

The saving was achieved despite the fact that, for the first 8½ months of plant operation, more expensive #2 oil was used because the less expensive natural gas system was not yet completed. For the entire 12 month period, only a fraction of the heat recovery available was recovered, because the engine jacket water recovery system had not been completed. In the future it will practically eliminate the fuel previously purchased to make domestic hot water.

Based on recent 15.5% utility electrical rate increase and completion of all the saving features of the plant, expected savings for the coming year are expected to be in the area of $520,000.

The are possibilities that may further improve the benefits derived from the cogeneration plant, according to builder-manager Richard Stone. For example, implementation of the federal law will allow sale of excess capacity to the utility at a profit. There is still the possiblity of an impending tax benefit and there is the possibility of legislation to provide lower natural gas fuel costs through incentive rates to cogenerators.

Effects of Air Supply Rates on Space Energy Consumption

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Abstract

IN THIS PAPER, an analytical model has been applied to study the effects of the air supply* rates on the dynamics of an occupied space as related to energy consumption and occupant's comfort. The concept of Air Diffusion Performance Index (ADPI) is reviewed and used to derive air movement control strategies to reduce energy consumption in buildings. The relationship between the air movement control strategies and passive thermal control strategies for energy efficient buildings is discussed. Procedures to calculate the terminal air velocity for a proposed diffuser and to optimize the selection of diffusers are described. Derivation of the air movement control strategies has shown that a design criterion based upon the dynamic conditions can easily be coordinated with comfort criteria, load criteria and ADPI criteria.

Introduction

Energy required to operate and maintain a building system depends directly on the desired level of indoor air quality. Air quality considerations of indoor climates must include thermal and mass air quality, lighting, and noise levels. Air supply rates can have a significant influence on the mass air quality, thermal air quality, and noise levels in an occupied space.

Appropriate mass air quality is necessary for safety and health of the occupants. Thermal air quality determines the thermal comfort of the occupants and is dependent upon many factors including building envelope characteristics, occupants, internal thermal

*Air supply includes recirculated and ventilated air, see Figure 1 on next page.

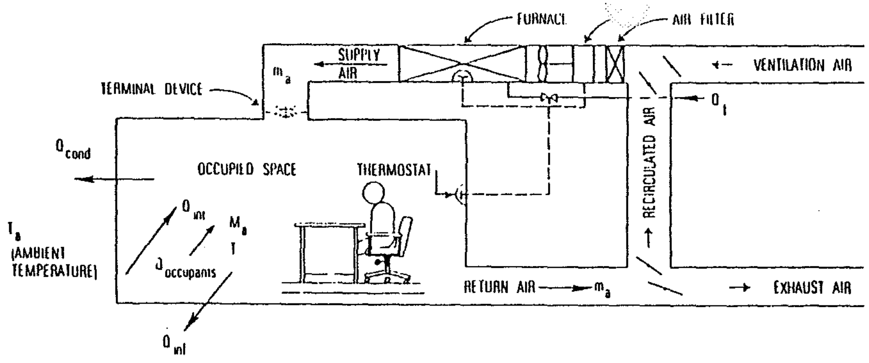

Fig. 1. Schematic representation of a forced air heating system.

processes, weather, air supply rates, conditioned air velocity, and location of the supply air terminal device(s). These factors interact with each other on a dynamic basis and result in a net energy transfer between the interior and exterior environments. To reduce energy consumption in buildings, it is very important to understand the dynamics of these interactions. The intent of this paper is to report an analytical model which describes the dynamic interactions among the factors having influence on thermal air quality and to apply the model to determine the effects of air supply rates on the thermal air quality as related to thermal comfort and energy consumption.

Background

Mass Air Quality: Historically, interest in the effects of building air supply on safety, health and comfort of the occupants can be traced back to as early as 1500 when Leonardo da Vinci designed and built a water driven fan to ventilate the bedrooms of his patron (1).* This was probably the first mechanical ventilation system.

An account of the use of natural ventilation is given when Charles the First of England, in 1600, ordered that no house could be erected whose rooms were less than 10 ft in height (2). He also specified that window height must exceed the width. Subsequent understanding

* Numbers in parentheses refer to references.

of the relation between "bad" air, ventilation, safety, and health was based on the discoveries of O_2 and CO_2, Dalton's law of partial pressures, description of the composition of air, and description of the respiratory function. Emphasis on the effects of organic impurities in indoor climates on the health of the occupants started in 1872 when Billings, a physician, recommended 30-40 cfm of outdoor air per person (3). The American Society of Heating and Ventilating Engineers (ASH&VE), one of the predecessors of American Society of Heating, Ventilating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), organized in 1895, adopted the view that engineers were ready to accept the idea of hygienists and physiologists (4). The hygenic and physiological communities had adopted the 30 cfm minimum as being required for adequate ventilation. This necessitated mechanical means. Therefore, ventilation to control mass air quality was essentially an engineering problem. Consequently, it was the engineers who were held responsible in future years for wasting large sums of money in over-designed ventilating systems.

Engineers initiated work to save money by correcting over-designed ventilation systems as early as 1930 when a major change in ventilation standards resulted from the experimental work reported by Yaglou, et al. (5). Conducted under controlled experimental conditions, the Yaglou studies have served as the primary reference in codes and standards for the last forty-five years.

In response to demands for energy efficient buildings, attempts have been made to prescribe lower and lower ventilation rates (6, 7). For example, ASHRAE Standards 90-75 "Energy Conservation in New Buildings" states that the minimum recommended values of ventilation rates prescribed in ASHRAE Standard 62-73 "Standards for Natural and Mechanical Ventilation" shall be used for design purposes for each type of occupancy. The implication of all such past methods to control mass air quality is that ventilation air quantity, by itself has been used as a reliable gauge of indoor mass air quality. In practice, many factors can influence mass air quality and if sufficient ventilation rates are not prescribed to meet foreseeable variabilities, the resultant indoor air could be unhealthy. Conversely, if sufficient ventilation rates are prescribed to meet all contingencies, the energy requirements for thermal control will be excessive.

Therefore, to provide acceptable indoor mass air quality at minimum rates of energy consumption under various occupancy conditions will require environmental control systems that can dynamically respond to changes in occupancy load and to outdoor air quality. However, reliable analytical models which can predict the dynamic conditions in an occupied space and their effects on the occupants, and which can be used to design and analyze the control systems have not yet been reported.

Thermal Air Quality: Thermal air quality of controlled environments can be assessed in terms of human thermal comfort. Thermal comfort effects must be evaluated from two standpoints. First, the thermal environment must be such that the occupant will be thermally neutral for a given condition of dress and activity. This condition can be termed "whole-body" comfort. Second, the thermal environment surrounding the individual should be completely uniform so that conditions of local discomfort do not exist.

The conditions for "whole-body" comfort are determined from four physical factors: dry-bulb temperature; water vapor pressure; air motion (relative air velocity) and mean radiation temperature and two reciprocative factors which are associated with each occupant, namely activity (metabolic heat production) and clothing (thermal insulation). Indoor design temperatures, as specified by building codes, standards, or design criteria, provide conditions which normally satisfy "whole-body" comfort for the occupants of a given space. This, of course, assumes that adjustments have been made for the type of activity and type of clothing to be worn in the space. The works of Rohles and Nevins (8) and McNall et al. (9) provide one source of data from which these criteria can be formulated. Fanger's comfort equation (10) provides another excellent means to predict the comfort response of occupants or to determine the preferred environmental conditions for a given set of variables. Fanger's equation is based on the assumption that there is a unique relationship between skin temperature and metabolic rate and sweat rates and metabolic rate for thermal comfort. A simple model of human physiological response of Gagge et al. (11) can also be used to predict thermal comfort and to provide detailed physiological data such as wetted area and skin temperature.

Of particular interest for this paper is the influence of air motion on thermal comfort. Subjective reactions to velocities from 0.1 to 1.0 m/s have been studied in the laboratory and thermal sensations of "comfortable" or "thermally neutral" have been achieved at velocities up to 1.0 m/s (12). In practical applications, air motion or air velocity will range from 0.1 to 0.2 m/s to a probable maximum of

0.5 or 0.8 m/s, except in those cases using "spot" ventilation. Based on the work of several investigators, Table 1 shows the preferred or comfort temperatures for the range of air velocities given above. Ostergaard et al. (13) have found that the direction of the air motion appears to have no effect on the comfort sensation, at least for velocities up to 0.8 m/s. It must be remembered that these data are for air velocities which envelop the occupant uniformly, and they do not apply to the localized or jet cooling of small areas of the body.

Table 1. Preferred Ambient Temperature (°C) for Comfort
Sedentary activity, MRT=DBT, I_{cl}=0.6, RH=50%

STUDY	*VELOCITY, m/s*				
	0.2	0.2	0.4	0.8	1.0
Fanger: Comfort Eqn. (10)	25.7	26.7	27.2	27.8	—
Ostergaard et al. (13)	25.1	—	—	27.4	—
Houghten, Yaglou (14)	—	26.7	27.2	28.1	—
Olesen, Fanger, Bassing (17)	24.5	—	—	26.5	—
Rohles, Woods, Nevins (18)	25.9	26.6	27.0	27.9	—
Fanger, Nevins, McNall (19)	—	—	—	—	30.0

Even though the requirements for "whole-body" thermal comfort are satisfied by the air supply system, there may exist local drafts which can cause discomfort within an otherwise comfortable environment. Houghten et al. (14) defined a draft as:

> Any local sense of cooling of a portion of the body caused either by excessive movement of air of normal temperature, by air having a normal velocity but a lower temperature, by excessive radiation to cold surface, or any combination of these three effects.

It is obvious from the work of Houghten and others that sensitivity to draft varies with different parts of the body. The effective draft temperature can be used to analyze the effect of velocity and temperature variations on the local feeling of comfort or discomfort. The effective draft temperature was based on the work of Rydberg and Norbach (15) and is defined as:

$$\theta = (T - T_c) - 7.66(V_x - 0.15)^* \quad (1)$$

From Houghten's relationship for the values of local velocity and local draft temperature (temperature of the moving air stream minus the average room temperature) at which 80% of the subjects were comfortable, the limits of θ are -1.67 and $+1.11$, for V_x between 0 and 0.36 m/s. Figure 2 illustrates the comfort criteria resulting from Eqn. 1 and the limits developed from Houghten's data. Also shown are the conditions at which 90% of the subjects reported comfort when exposed to drafts on the ankle and on the neck, and the conditions at which 80% of the subjects reporting comfort when exposed to drafts on the neck.

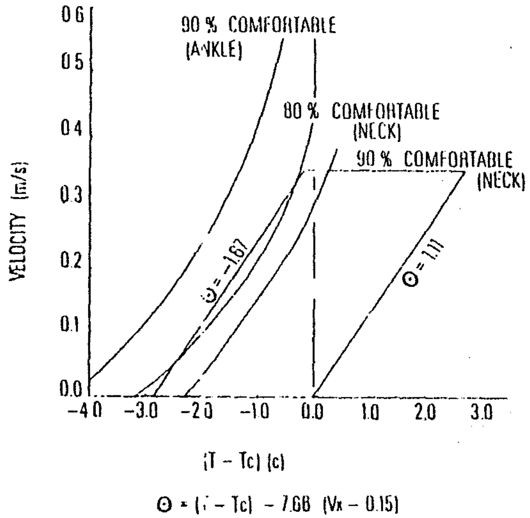

Fig. 2. Subjective response to drafts and criteria for the Air Diffusion Performance Index, ADPI (after reference 12)

Air Diffusion Performance Index, ADPI: The ultimate test of an air distribution system is the response of the occupants to conditions in the occupied space. In an effort to provide a single number rating system for the evaluation of air distribution systems; the Air Diffusion Performance Index, ADPI, was proposed in 1964 (2). Using the comfort criteria described above, ADPI was defined as:

> The number of measuring positions, uniformly distributed in a vertical centerline plane passing through the diffuser or throughout the occupied zone, at which the comfort criteria are satisfied, expressed as a percent of the total number of positions at which measurements of temperature and velocity are made.

ADPI was found to be a valid, sensitive, single number index relating outlet performance, the space, and occupant's comfort.

ADPI was shown to be a function of the type of terminal device, the room load, the supply air flow rate and the room geometry for all systems except the air distribution ceiling systems.

Miller and Nash (16) combined the parameters affecting ADPI for terminal devices and presented ADPI as a function of the ratio of the isothermal jet throw distance *(X)* to a characteristic room dimension *(L)*. Room load was found to be a parameter. The throw of a jet was defined as the distance from the outlet to a point in the isothermal air stream where the maximum velocity occurring in the stream cross-section has been reduced to a selected "terminal" velocity. Throw data cataloged by most diffuser manufacturers, are taken under isothermal conditions.

Throw coefficients vary with the diffuser type. The relationship between throw *(X)* and outlet velocity (V_o) is:

$$\frac{X}{\sqrt{A_e}} = K\frac{V_o}{V_x} = \frac{KQ_o}{V_x A_e} \quad (2)$$

Typical values of K for several diffuser types are given in Table 2 which has been adapted from reference 12.

The characteristic room length *(L)* is, in some installations, an easy dimension to determine and, in other installations more difficult. For example, for a high side wall grille located in the one end of the test room, the characteristic room length would be the length of the room in the direction of the throw. A characteristic dimension for slot diffusers located on the center-line of the ceiling would be half of the width of the room. In some installations the air pattern will be directed towards an adjacent device such that the primary jets will

*See list of nomenclature at the end of this article.

Table 2. Typical Throw Coefficients (adapted from reference 12)

DIFFUSER		K
High Sidewall Grille		
Vane Angle	0.0 deg	4.7 — 8.5
	22.5 deg	3.6 — 6.5
	45.0 deg	2.6 — 4.7
Circular Ceiling Diffuser		0.8 — 1.2
Slot Diffuser		0.9 — 3.1
Light Troffer Diffuser		1.5 — 4.0

interact. This configuration occurs in large spaces served by multiple outlets. In this case, L would be taken as half the distance between units plus the distance from the ceiling to the top of the occupied zone. A terminal velocity of 0.25 m/s is normally used.

Analytical Model for Dynamic Thermal Conditions

The thermal processes to describe the heat transfer rates in an occupied space of a building are shown in Figure 1. The following assumptions are made:

1. The properties of the thermal process involved in heating the occupied space are "lumped" at the location at which the occupants experience the controlled conditions. This assumption permits us to describe the process with ordinary linear differential equations instead of partial differential equations.
2. Changes in the stored energy of space air are assumed to be due only to changes in dry-bulb temperatures; those due to humidity changes are considered negligible.
3. Heat transfer by radiation is not included in this analysis.
4. Infiltration losses are assumed to be a function of air supply flow rate, occupant activity, internal heat sources, room temperature, and construction details in a given occupied space.

An energy balance on the occupied space of Figure 1 yields: Rate of energy in = Rate of energy out + rate of change in stored energy, or

$$\dot{Q}_f + \dot{m}_a c_{pa} T_i + \dot{Q}_{occ} + \dot{Q}_{int} = \dot{m}_a c_{pa} T + \dot{Q}_{cond} + \dot{Q}_{inf} + M_a c_{pa} \frac{dT}{dt} \tag{3}$$

If we define

$$\dot{Q} = \dot{Q}_f + \dot{Q}_{occ} + \dot{Q}_{int} \tag{4}$$

and

$$\dot{Q}_{cond} = UA\,(T - T_a) \tag{5}$$

then we may rewrite Eqn. 3 as:

$$\frac{dT}{dt} = \frac{\dot{Q}}{M_a c_{pa}} + \frac{\dot{m}_a c_{pa}}{M_a c_{pa}}(T_i - T) - \frac{UA(T - T_a)}{M_a c_{pa}} - \frac{\dot{Q}_{inf}}{M_a c_{pa}} \tag{6}$$

For most applications, relative small fluctuations in net heat gains and space temperature will be experienced within the occupied space. For this reason, only small perturbations about the operating or set point will be considered. The operating point may be found by setting the time derivative contained in Eqn. 6 to zero:

$$\frac{\bar{\dot{Q}}^{*}}{M_a c_{pa}} + \frac{\dot{m}_a c_{pa}}{M_a c_{pa}}(\bar{T}_i - \bar{T}) - \frac{UA(\bar{T} - \bar{T}_a)}{M_a c_{pa}} - \frac{\bar{\dot{Q}}_{inf}}{M_a c_{pa}} = 0 \tag{7}$$

Eqns. 6 and 7 may be combined to yield an equation in terms of perturbations about the operating point:

$$\frac{d(T - \bar{T})}{dt} = \frac{\dot{Q} - \bar{\dot{Q}}}{M_a c_{pa}} + \frac{\dot{m}_a c_{pa}}{M_a c_{pa}}\left[(T_i - \bar{T}_i) - (T - \bar{T})\right] - \frac{UA}{M_a c_{pa}}\left[(T - \bar{T}) - (T_a - \bar{T}_a)\right] - \frac{(\dot{Q}_{inf} - \bar{\dot{Q}}_{inf})}{M_a c_{pa}} \tag{8}$$

*Bar (—) notations indicate values at the operating point.

Eqn. 8 may be rewritten as:

$$\frac{d\Delta T}{dt} = \frac{\Delta \dot{Q}}{M_a c_{pa}} + \frac{\dot{m}_a c_{pa}}{M_a c_{pa}} (\Delta T_i - \Delta T) - \frac{UA}{M_a c_{pa}} (\Delta T - \Delta T_a) - \frac{\Delta \dot{Q}_{inf}}{M_a c_{pa}} \quad (9)$$

From Assumption 4, we may write

$$\dot{Q}_{inf} = f(\dot{Q}, T) \quad (10)$$

Linearizing Eqn. 10 about the operating points yields:

$$\Delta \dot{Q}_{inf} = K_1 \Delta \dot{Q} + K_2 \Delta T \quad (11)$$

The constants, K_1 and K_2, are defined as follows and are evaluated at the operating point:

$$K_1 = \left.\frac{\partial \dot{Q}_{inf}}{\partial \dot{Q}}\right| T = \text{constant} \quad (12)$$

$$K_2 = \left.\frac{\partial \dot{Q}_{inf}}{\partial T}\right| \dot{Q} = \text{constant} \quad (13)$$

Substituting Eqn. 11 into Eqn. 9, taking the Laplace transform, and rearranging, we obtain for step inputs:

$$\Delta T(s) = \frac{K_{\dot{Q}} \Delta \dot{Q}(s)}{S(1 + \tau_{os} S)} + \frac{K_{T_i} \Delta T_i(s)}{S(1 + \tau_{os} S)} + \frac{K_{T_a} \Delta T_a(s)}{S(1 + \tau_{os} S)} \quad (14)$$

where

$$K_{\dot{Q}} = \frac{1 - K_1}{\dot{m}_a c_{pa} + UA + K_2} \quad (15)$$

$$K_{T_i} = \frac{\dot{m}_a c_{pa}}{\dot{m}_a c_{pa} + UA + K_2} \quad (16)$$

$$K_{T_a} = \frac{UA}{\dot{m}_a c_{pa} + UA + K_2} \quad (17)$$

$$\tau_{os} = \frac{M_a c_{pa}}{\dot{m}_a c_{pa} + UA + K_2} \quad (18)$$

A block diagram representation of Eqn. 14 is shown in Figure 3. It may be noted that τ_{os}, the time constant of the occupied space, is a function of the enclosed mass of air, the mass flow rate of air, the envelope characteristics and infiltration.

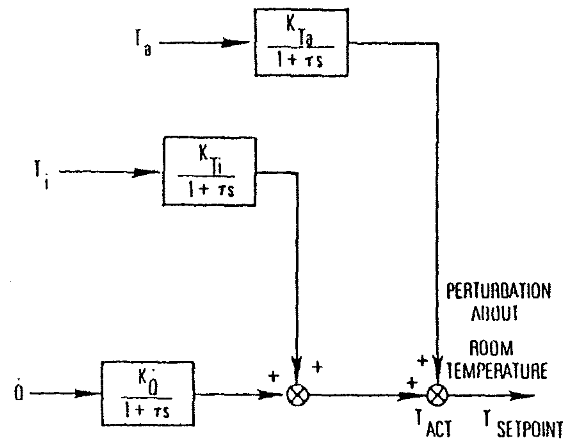

Fig. 3. Block diagram presentation of the mathematical model.

Eqn. 14, represented by the clock diagram of Figure 3, is in a form which can be coupled to the models of other components, including the occupant, in an environmental control system to yield an overall transfer function for the building system. This can be subjected to the techniques of control theory for design purposes.

Passive Energy Efficient Control Strategies

Passive Control Strategies: The parameter K_2 as described by Eqn. 13 can be used to develop criteria for evaluating energy consumption characteristics of envelopes within a given category of buildings. For example, a number of single family residences with equal floor areas can be compared to study the influence of different types of structures, shapes, infiltration characteristics and life styles can be used to derive passive control strategies for energy conservation. To illustrate this application it may be noted that

$$K_2 = \frac{\partial \dot{Q}_{inf}}{\partial T} \bigg| \; \dot{Q} = \text{Const.} \tag{13}$$

$$\text{and } \dot{Q}_{inf} = \dot{m}_{ainf} c_{pa} (T - T_a) \tag{19}$$

Under the conditions when heating or cooling is not required,

$$K_2 = \dot{m}_{ainf} c_{pa} - \dot{m}_{ainf} c_{pa} \frac{(dT_a)}{dT} \tag{20}$$

Therefore, the value of K_2 depends upon the mass flow rate of infiltration air, $\dot{m}_{ainf}$, and the factor $(dT/_a dT)$. This factor depends on the type of construction of the house, heat transfer and infiltration characteristics. Under passive conditions its value may range from +1 to ∞ as shown in Figure 4. A value of (+1) for dT_a/dT means a very loose construction from an infiltration point of view or no resistance to heat transfer across the envelope. Substitution of these two extreme values of (dT_a/dT) in Eqn. 20 reveals that K_2 can vary from 0 (no envelope) to $-\infty$ (perfect envelope for a building with passive control. The numerical value of K_2 is one single parameter which combines the elements of infiltration and heat transfer characteristics of a structure. If the infiltration losses can be defined to include losses due to door openings or window openings, the numerical value of K_2 becomes a parameter which can be used to compare life styles. To use K_2 as a criterion to evaluate energy consumption characteristics of envelopes within a category of buildings, the following procedures are suggested.

Procedures for Existing Buildings:

1. Select buildings within the same category on equal floor area basis.
2. Record hourly indoor temperatures and outdoor air temperatures during spring or fall when heating or cooling is not needed.
3. Measure infiltration rates at the time the temperatures are being measured using tracer gas techniques.
4. Calculate the values of $|K_2|$ using equation 20.
5. Structures within the same category designed to provide equal floor areas and which have an equal number of air changes per

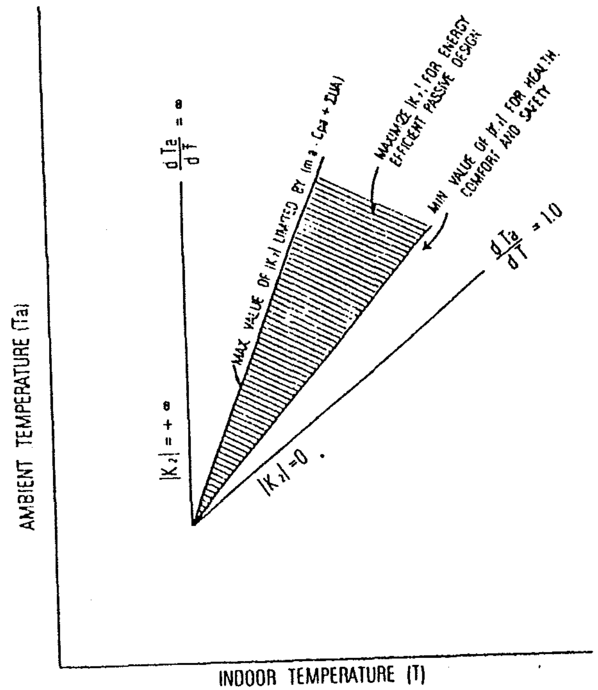

Fig. 4. Relationships between ambient and indoor temperatures for passive design

hour will be more energy intensive as the value of K_2 decreases. Occupants in structures with equal floor areas, identical envelope characteristics and designed for the same number of air changes per hour will be identified as having more energy intensive life styles as the value of $|K_2|$ decreases. In structures with equal floor areas but with different numbers of air changes per hour, the one will be more energy intensive as the value of (dT_a/dT) decreases. Some corrective steps to control energy intensive passive buildings are suggested in the discussion below.

Procedures for Energy Efficient Passive Design: It has been shown above in Step 5 of the procedures for existing buildings that $|K_2|$ should be as large as possible. In the limit $|K_2|$ has to be less than $(\dot{m}_a c_{pa} + \Sigma UA)$ yields an infinite value for the time constant for the occupied space, see Eqn. 18. Physically an infinite time constant of an occupied space in response to inputs from weather, internal sources or from heat exchanger has no meaning and hence a limit on the value of $|K_2|$ is shown in Figure 4. Thus, codes should be developed to specify maximum and minimum value of $|K_2|$ for health, comfort and safety, and energy considerations. The following procedures are suggested to maximize the value of $|K_2|$ within the limits shown in Figure 4.

1. Increase the thermal mass of the structure. The results will be shown by increased values of (dT_a/dT) and of $|K_2|$.
2. Reduce over-capacity of the heating source.
3. Use space temperature reset whenever possible to reduce the temperature difference between indoor and outdoor temperatures. The result will be shown by increased values of (dT_a/dT) and $|K_2|$.
4. Reduce infiltration losses by incorporating self-closing doors and tight fitting windows.
5. Increase air supply rates at constant values of (ΣUA). This results in increased value of $(\dot{m}_a c_{pa})$ and hence an increase in $|K_2|$. However, increased air supply rates should take into consideration the limits imposed by discomfort due to drafts as discussed above.

Air Movement Strategies

Procedure 5 for energy efficient passive design has suggested that increases in air supply rates can improve the energy consumption characteristics of the envelope. It has been shown in Table 1 that an increase in whole-body air movement is accompanied by an increase in preferred ambient temperature for constant comfort levels. Woods has also shown that an increase in whole-body air movement must be accompanied by an increase in SET for constant comfort level (20). Calculations in reference (20) indicate that, for air movement in the range of 0.2 to 1.0 m/sec, a shift of 1C° is required for an increase of 0.38 m/sec to keep the same comfort level. However, when air movement is less than 0.2 m/sec (i.e., still air), the calculated shift is significantly greater: namely 1C° for an increase of 0.04 m/sec. So a very low velocity of air is desired for winter conditions to provide comfort at lower space temperatures to save energy. Alternatively, high air velocities in the summer can permit higher acceptable space temperatures resulting in reduced air conditioning loads and energy savings. Thus, air movement strategies derived from comfort considerations are in agreement with those derived from energy efficient passive design considerations for summer conditions whereas they conflict for winter conditions. Figure 5 has been constructed using equations (16)-(18) and can aid the designer to seek an optimum solution. The following steps are suggested:

1. Select supply air temperature, T_a, for a certain design load and a desired control temperature, T_c, so that $\dot{m}_a$ calculated from the following equation:

$$\dot{Q}_L = \dot{m}_a c_{pa} (T_s - T_c) \tag{21}$$

lies to the right of line XY in Figure 5. This step will prevent the selection of a value of K_{Ta} which is too high and will result in an optimum combination of ΣUA, $\dot{m}_a$ and K_2.

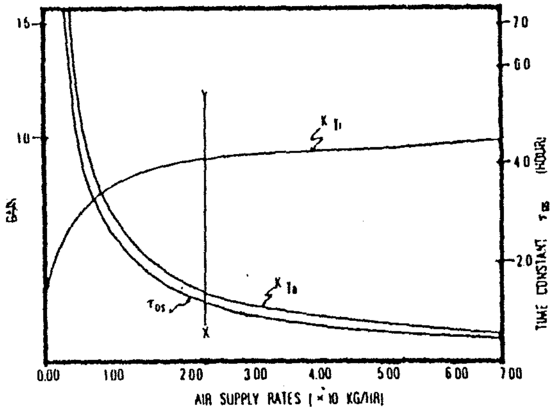

Fig. 5. Calculated effects of air supply rates on the dynamics of occupied space.

2. Determine the outlet air velocity for the proposed diffuser from the equation:

$$\dot{m}_a = A_e V_o \rho \tag{22}$$

3. Calculate V_x, the local velocity at the location of the occupants, for a specified occupied space temperature, T, using Eqn. 2. The values of K to be used in Eqn. 2 can be read from Table 2.
4. Calculate the effective draft temperature, θ, from Eqn. 1.
5. Check that the values of θ lie between (—1.67) to (+1.11) when velocities are m/sec and temperature in $^\circ K$.
6. If the calculated values in step 4 do not fall within the limits stated in step 5, choose a new effective area for the diffuser and repeat steps 1-5.

A design procedure based on steps 1-6 should coordinate load, dynamic performance and air diffusion performance index (ADPI) criteria.

Conclusions

Acceptable indoor thermal and mass air quality can be maintained with minimum energy consumption if environmental control systems respond dynamically to the changes in those variables which have influence on energy consumption in buildings. Analytical models which can predict dynamic conditions in occupied spaces can prove to be useful tools in the design and analysis of such control systems.

A mathematical model which describes the dynamic interactions among the thermal energy characteristics of a building has been presented in this paper. The model has been applied to derive some energy efficient control strategies. Effects of air supply rates on the dynamics of an occupied space as related to energy consumption and occupant's comfort have been discussed.

Acknowledgments

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Nomenclature

i inlet
inf infiltration
int internal; integral
m meter
o outlet; outside
occ occupant
os occupied space
s sensor, supply
x local, at the occupant's position

Greek Symbols
Δ perturbation of a variable
θ effective draft temperature (°C) or (°F)
ρ density: (Kg/m^3) or (lbm/ft^3)
τ time constant: (hr)

3. Calculate V_x, the local velocity at the location of the occupants, for a specified occupied space temperature, T, using Eqn. 2. The values of K to be used in Eqn. 2 can be read from Table 2.
4. Calculate the effective draft temperature, θ, from Eqn. 1.
5. Check that the values of θ lie between (−1.67) to (+1.11) when velocities are m/sec and temperature in °K.
6. If the calculated values in step 4 do not fall within the limits stated in step 5, choose a new effective area for the diffuser and repeat steps 1–5.

A design procedure based on steps 1–6 should coordinate load, dynamic performance and air diffusion performance index (ADPI) criteria.

Conclusions

Acceptable indoor thermal and mass air quality can be maintained with minimum energy consumption if environmental control systems respond dynamically to the changes in those variables which have influence on energy consumption in buildings. Analytical models which can predict dynamic conditions in occupied spaces can prove to be useful tools in the design and analysis of such control systems.

A mathematical model which describes the dynamic interactions among the thermal energy characteristics of a building has been pre-

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