#3719

ENERGY ANALYSIS OF BUILDINGS WITH DIFFERENT AIR SUPPLY AND EXHAUST SYSTEMS

Q. Chen, Ph.D.
Associate Member ASHRAE

T. G. Hoornstra

J. van der Kooi, Ph.D.

ABSTRACT

Research was carried out in order to study the influence on energy consumption of different kinds of air supply and exhaust systems in a room. Based on the air temperature distribution of the room with a displacement ventilation system and a well-mixed system, which were calculated by an airflow program, the cooling load program ACCURACY and the energy analysis program ENERK were used for the calculation of the space load and the annual energy consumption of the room.

The results showed that the air temperature distributions in the room are very important in the prediction of room energy consumption. For a variable-air-volume system, the energy required by the chiller and the ventilator with the displacement ventilation system is 26% smaller than that with the well-mixed system. The air displacement system is recommended for practical applications for saving energy and obtaining better indoor air quality.

INTRODUCTION

Since the energy crisis of the 1970s, the insulation of buildings has been improved in order to reduce heat loss in winter, heat gain in summer, and infiltration of outdoor air. As a consequence, there is less heat extracted from or supplied to a room for maintaining a comfortable air temperature. Because the heat extracted or supplied is related to the air supply and temperature difference between the air inlet and outlet of a room, both the amount of air supplied and the air temperature difference can be reduced.

A reduction of air supply causes an increase in the concentration of indoor pollutants. In order to remove indoor pollutants effectively, a displacement ventilation system as shown in Figure 1a has been used during recent years in Scandinavia and western European countries (Danielsson 1987). In the displacement ventilation system, air is supplied into a room in such a way that it fills the occupied zone with clean air. This can be done if it is supplied with low velocity (mostly less than 1.6 ft/s [0.5 m/s]) and at an air temperature at least 2°F (1°C) lower than that in the occupied zone. As the supply air is colder than the air in the occupied zone, it will fill the lower part of the room due to the buoyancy effect. This buoyancy effect results in a vertical temperature stratification in the room and the temperature in the upper part of the room will then be higher than that in the lower part. Because the air exhaust outlets of the room are near the ceiling, the temperature of the extracted air is higher than that in a well-mixed situation as created by a system shown in Figure 1b. This means that the air temperature difference between the inlet and outlet in the displacement system is larger than that in the well-mixed system for cooling situations. To extract the same amount of heat from the room, the amount of air supply can thus be much smaller or the air temperature supplied can be higher. The air inlet and outlet locations of a room and indoor air temperature distribution have a large influence

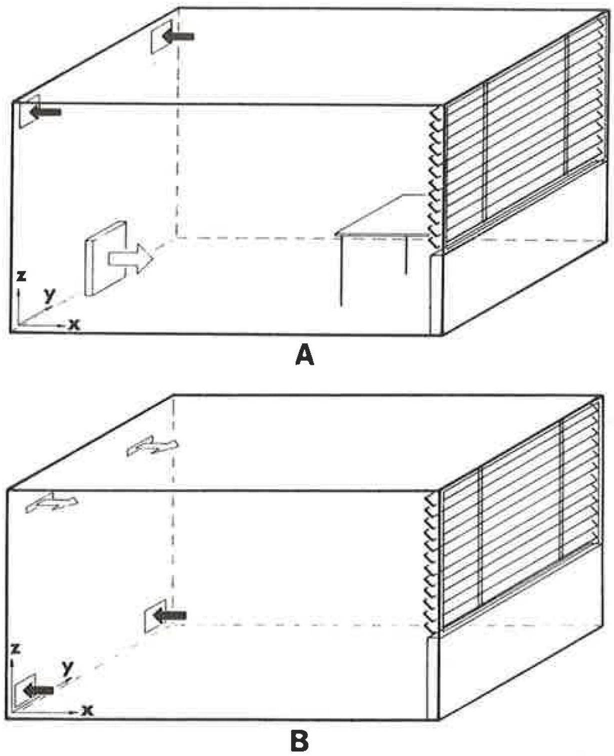


Figure 1 The room with a displacement ventilation system (A) and a well-mixed ventilation system (B)

Qingyan Chen was a Ph.D. student, **Tom G. Hoornstra**, a graduate student, and **Jan van der Kooi** is a senior staff member, Laboratory for Refrigeration and Indoor Climate Technology, Delft University of Technology, The Netherlands. Chen is currently with Energy Systems Laboratory, ETH-Zentrum, Zurich, Switzerland.

on energy consumption because the air supply is directly related to the energy consumption.

In order to estimate the influence of the air inlet and outlet locations on human comfort and energy consumption of the room, the nature of the indoor air flow and the transient heat transfer in the walls must be predicted simultaneously, especially under small air supply conditions. This is because the air flow and transient heat transfer in a room are interrelated.

Chen et al. (1988) revealed the influence of the air inlet and outlet locations on human comfort. In that paper, the air velocity, temperature, and contaminant distributions in a room were predicted with several kinds of air supply and exhaust systems. The displacement ventilation system and the well-mixed system were included. They also constructed a new cooling load program, "ACCURACY," by which the influence of the inlet and outlet locations on the space load of a room can be studied (Chen and Van der Kooi 1988). This is done by introducing temperature distributions of room air in the computation of the heat transfer through enclosures. With these previous results, it is now possible to investigate the influence of the air inlet and outlet locations on the energy consumption of a room. This is the aim of the present paper. The annual energy consumption in a room with the two kinds of air supply and exhaust systems will be calculated and compared.

Numerical calculations of air flow and heat transfer in a room have been used extensively in recent years. Success has been achieved in calculating indoor air velocity, temperature, and pollutant concentration distributions, and in computing the hourly space load and energy consumption of a room. This approach involves the numerical solution of a set of partial differential equations for the turbulent flow and heat transfer. Therefore, this numerical approach will be used for the present research.

THE THEORETICAL FUNDAMENTALS

In order to study the influence of different air supply and exhaust systems on room energy consumption, it is necessary to calculate (1) indoor air temperature distributions, (2) the space load of the room, and (3) the energy consumption of the air-conditioning system in the room. This is because different kinds of air supply and exhaust systems result in different kinds of indoor air temperature distributions. These temperature distributions are very important in the computation of heat transfer through the enclosures of the room and, consequently, in the calculation of space load of the room (Chen and Van der Kooi 1988). The space load is related to the amount of air supply and the air temperature difference between the inlet and outlet of the room. The amount of air supply and the air temperatures of the inlet and outlet are directly connected to the energy consumption of the air-conditioning system.

The Algorithm Outline

The algorithm used in the present study can be divided into three major parts.

(1) An airflow program is used to calculate the air velocity and temperature distributions for a room under a number of specific situations, such as different kinds of room ventilation rates (*Vent*) and space loads (*Q*). The ventilation rates are related to the control strategies and comfort requirement, and the space loads are the results of the heat gain or heat loss through windows and the heat gain from occupants, lighting, and appliances, etc. The choice of the ventilation rates and space loads should be realistic. For instance, for a two-person office, the ventilation rates can be from 0 times per hour up to 7 times per hour and the space loads can be 0 to 3000 Btu/h (0 to 1000 W) for summer conditions. Of course, the locations of the air supply and exhaust unit, the room geometry, and the positions of the heat sources are extremely important in airflow computations. However, for a certain air supply system in a room, most of the above parameters are known. Hence, the ventilation rates and space loads are the dominant factors concerning indoor airflow patterns. In order to simulate air velocity and temperature fields that may be encountered in the room under various conditions, about 10 flow fields should be computed with reasonable combinations of *Vent* and *Q*.

(2) From the computed temperature fields, the following function can be determined:

$$\Delta T = f(Q, Vent) \qquad (1)$$

where ΔT = the temperature differences between the controlled point (i.e., in the middle of the occupied zone) and the air at points near the enclosure surfaces. The ΔT will then be used in the cooling load program ACCURACY to study the influence of different kinds of air supply and exhaust systems on space loads and extracted air temperatures of the room. The hour-by-hour annual space loads and extracted air temperatures calculated by ACCURACY with the ΔT are different from those obtained with the assumption of a uniform indoor air temperature distribution (Chen and Van der Kooi 1988). In the space load computations, the surface temperatures of the room enclosure are also available. They are compared with those used in Part 1. If the difference is too large, iterations between Parts 1 and 2 are usually necessary.

(3) The hour-by-hour annual space loads and extracted air temperatures of the room are used as the input for an energy analysis program, ENERK (Van Paassen 1986), to calculate the annual energy consumption of the room.

The interrelationship among these three parts is illustrated in Figure 2. In the following sections, the theoretical fundamentals of the computation of indoor air temperature distributions, space load, and annual energy consumption of a room will be briefly discussed.

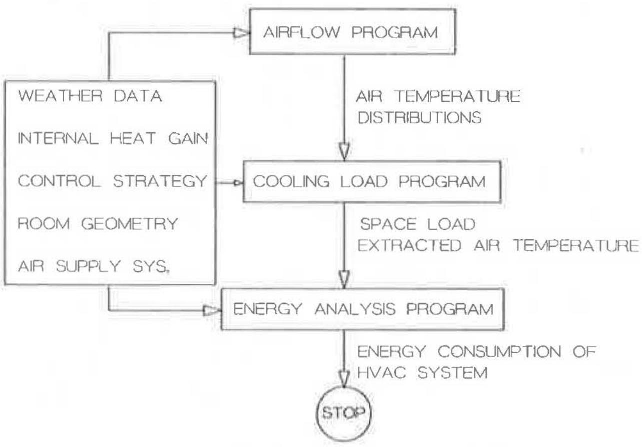


Figure 2 ***Diagram of the simulation model***

Computation of Indoor Air Temperature Distributions

In most cases, the air flows encountered in an air-conditioned room are turbulent. The mathematical modeling of turbulent flow is now within the capabilities of modern mathematical and numerical methods. Among the turbulent models, the two-equation k-ϵ turbulent transport model (Launder and Spalding 1974) seems most suitable for air flow in a room. The k stands for the kinetic energy of turbulence and ϵ is the dissipation rate of turbulence energy. An airflow program based on this model (Gunton et al. 1983) has been used for the present study. The governing equations with the k-ϵ model consist of the conservation equations of mass (continuity), momentum (u, v, w), energy (H), concentration (C), turbulence energy (k), and dissipation rate of turbulence energy (ϵ). A complete description of the theoretical basis and solving procedure can be found in Rosten and Spalding (1981).

Computation of Space Load of a Room

Normal cooling load programs that assume room air temperature to be uniform are not suitable for energy analysis with different air supply and exhaust systems. This is because they will present the same results, such as inlet and outlet air temperatures and space loads, for all kinds of systems. For heat transfer through the enclosures of a room, the air temperatures near the enclosure surface are important. Hence, in the computation of space load, the field values of air temperature distributions can be replaced by the air temperature at the controlled point and by the air temperature differences between the controlled point and the air points near the surfaces (ΔT). The air temperature at the controlled point (i.e., in the middle of the occupied zone) depends on control strategy and is known. The ΔT can be determined as the function of space loads (Q) and room ventilation rates (*Vent*) from the precalculated fields of the room air temperature, as discussed above.

In the cooling load program ACCURACY, Equation 1 is combined into the calculation of heat transfer through the enclosures of the room. Therefore, the influence of different kinds of air supply and exhaust systems on space load can be studied. This is the main difference between the ACCURACY program and other cooling load programs based on the room energy balance equations. A detailed description of the program was presented by Chen and Van der Kooi (1988).

ACCURACY uses the Z-transfer function method (Stephenson and Mitalas 1971) for the calculation of heat conduction through walls. The heat exchange in an external wall is shown in Figure 3. The inner enclosure surfaces are assumed to be gray bodies and the multiple reflections among the surfaces are also studied. The radiative heat exchange (Q) between surfaces i and j is calculated from:

$$Q = \epsilon_i \varphi_{i,j} (E_{b,i} - E_{b,j}) A_i \text{ (W)} \qquad (2)$$

where

ϵ_i = emissivity of surface i
$\varphi_{i,j}$ = radiative heat exchange factor between surfaces i and j
$E_{b,i}$ = emissive power of black body for surface i
$E_{b,j}$ = emissive power of black body for surface j
A_i = area of surface i

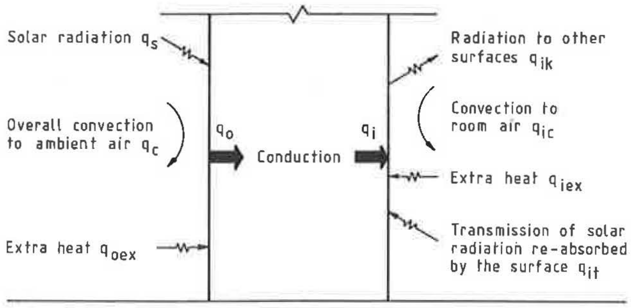


***Figure 3** Heat exchange in an exterior wall*

The $\varphi_{i,j}$ in Equation 2 is determined from:

$$[\varphi] = \{[I] - [F]\cdot diag(1 - \epsilon)]\}^{-1}\cdot[F]\cdot diag(\epsilon) \qquad (3)$$

where

$[I]$ = unit matrix
$[F]$ = view factor matrix

Since a considerable part of the space load is caused by solar radiation through the window, ACCURACY calculates the heat transfer in a window, as shown in Figure 4 by the energy balance method. However, the heat capacity of the window is neglected. The transmission of solar radiation in the room is assumed to be reabsorbed uniformly by each inner surface. The absorptivities, transmissivities, and reflectivities of the glass panes and venetian blinds are calculated based on the method described in Kimura (1977).

Computation of Annual Energy Consumption of Buildings

There are many methods available for building energy analysis. They can be classified as (1) single-measure methods, (2) simplified multiple-measure methods, and (3) detailed simulation methods (ASHRAE 1985). Methods 1 and 2 are not good enough for accurate energy analysis and, therefore, they are not used for the present study. There are many computer programs based on method 3, as reviewed and validated by Irving (1988) and Judkoff (1988). Among these programs, most are based on the method of hour-by-hour prediction. Therefore, the predic-

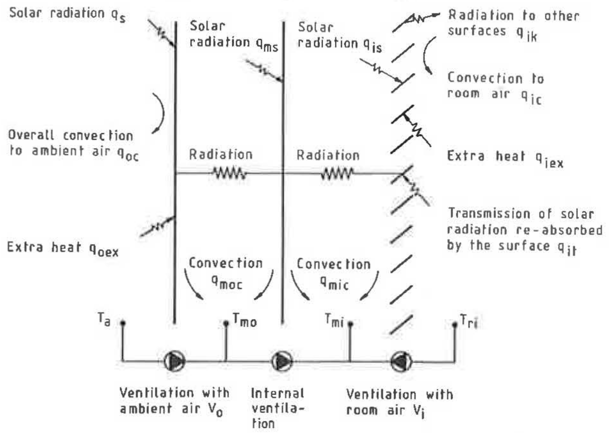


***Figure 4** Heat exchange in an exterior window*

tions are expensive, if the weather data of a full reference year are employed. When the prediction is applied to optimize a system for the best energy-saving solution, many computations are required for comparing various alternative designs of buildings and installation. In order to reduce the computational time, a new program, ENERK, which was developed by Van Paassen (1986), is used to optimize the computer algorithm used in the secondary and primary equipment. The optimized algorithm will be introduced in the present study.

Based on an hour-by-hour calculation of the indoor temperature and the space loads, which have been obtained from ACCURACY, the energy analysis program ENERK calculates the annual energy consumption in the following steps:

(1) calculates the probability of the joint occurrence of specific values of space load (Q), outdoor temperature (T), and humidity (X);

(2) determines the energy consumption of the plant for all values of the temperatures and humidities of the outside air and the space loads of the room; and

(3) gives the annual energy consumption by the combination of the values of the load probability matrix and the energy matrix of the plant.

The space load of a room can be characterized by the probability distribution matrix of multi-variables, $P(Q,T,X)$. It gives the probability of the joint occurrence of Q to $Q + \Delta Q$, T to $T + \Delta T$ and X to $X + \Delta X$. In the computations presented here, the space load is only divided into two parts, one for heating and the other for cooling. Therefore, only the probabilities $P+$ and $P-$ are calculated for each of the seven sections in the psychrometric chart shown in Figure 5. Then the temperatures and humidities of outdoor air, the extracted air temperatures, and the heating and cooling loads are averaged per section.

The sections are classified in such a way that the control strategy in each section is the same. The more sections are used, the higher the accuracy will be. Van Mierlo (1986) demonstrated that a partition of the psychrometric chart as given in Figure 5 results in calculations with a difference of less than 5%, as compared to those obtained with an hour-by-hour simulation approach. The accuracy is acceptable for practical applications and, therefore, it is used for the present study.

For each section in the psychrometric chart, the average values of the heating load, cooling load, outdoor temperature, and humidity are calculated. This results in a combination of Q, T, and X for each section. Then, the energy consumption is calculated for each section, which is required by the air-conditioning system to compensate the space load Q. This energy is expressed by the matrices E_{gas} (T,X) and $E_{electricity}$ (T,X). For instance, for the air-handling process shown in Figure 5, the amounts of energy for heating, cooling, and air transport can be calculated as:

$$E_{gas} = \Delta h^{+} \dot{m}/\eta_b \text{ (kWh)} \quad (4)$$

$$E_{electricity} = \Delta h^{-} \dot{m}/\eta_{ch} + \Delta p_{fan} \dot{m}/(\eta_{fan}\,\rho) \text{ (kWh)} \quad (5)$$

where

$\Delta h^{+}\ \dot{m}$ = heat supplied by hot water
$\Delta h^{-}\ \dot{m}$ = heat removed by cold water
Δp_{fan} = fan pressure
η_b = boiler efficiency
η_{ch} = chiller efficiency
η_{fan} = fan efficiency

These efficiencies depend on various variables, such as load, and can be determined in many ways (Van Paassen 1986).

Multiplication of the elements of the energy matrix with corresponding elements of the probability matrix and with the total hours of the considered period gives the energy consumption of each possible combination of Q, T, and X. Summing all the products gives the total cost of energy consumption.

By this method, only the non-identical situations are calculated in contrast to the hour-by-hour approach, where identical situations encountered at different moments of time are recalculated all over again. A more detailed description of the ENERK program is given in Van Paassen (1981, 1986).

In the present study, the weather data of the Dutch short reference year (Liem and Van Paassen 1986) are used to reduce the computing time.

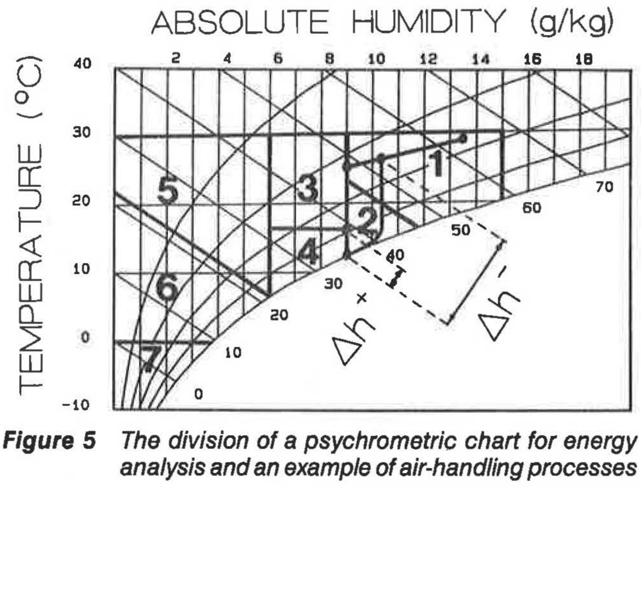


***Figure 5** The division of a psychrometric chart for energy analysis and an example of air-handling processes*

COMPUTATIONAL RESULTS

In this section, a room that is 18.4 ft (5.6 m) long, 9.8 ft (3.0 m) wide, and 10.5 ft (3.2 m) high, as shown in Figure 1, is used to study the influence of the two kinds of air supply and exhaust systems on annual energy consumption. The wall material data of the room are presented in Table 1.

Indoor Airflow Distributions of the Room

In order to determine the function in Equation 1, it is necessary to precalculate the air temperature distributions in a room in specific situations. However, in this subsection, the calculation will be demonstrated only under two specific situations: a cooling situation and a heating situation. These computed results have been validated by measurements (Chen et al. 1988). More detailed information about other specific situations can be found in Chen (1988) and Chen et al. (1988).

Cooling Situation A 2047 Btu/h (600 W) electricity supply is introduced on the inside venetian blinds to simulate solar radiation. The air temperature of the outside window is controlled at 73.4°F (23.0°C). The radiator under the window, which is the same width as the room and is

TABLE 1
Room Enclosure Materials

Enclosure	Thickness	Density	Specific Heat	Thermal Conductivity
(I-P units)				
	in.	lb/ft^3	Btu/lb·°F	Btu·in./h·ft^2·°F
Ceiling	6.89	143.75	0.2	13.19
Floor	6.89	143.75	0.2	13.19
Rear wall	5.51	43.75	0.2	1.60
Side walls	5.51	43.75	0.2	1.60
Parapet	3.94	1.88	0.4	0.24
(SI Units)				
	m	kg/m^3	J/kg·k	W/m·K
Ceiling	0.175	2300	840	1.9
Floor	0.175	2300	840	1.9
Rear wall	0.140	700	840	0.23
Side walls	0.140	700	840	0.23
Parapet	0.100	30	1470	0.035

2.4 ft (0.75 m) high, is switched off for this situation. The inlet air supply is 0.20 lb/s (0.09 kg/s), corresponding with a mechanical ventilation rate of 5 times per hour. The room air temperature in the middle of the occupied zone is controlled at 73.4°F (23.0°C).

Heating Situation There is no electricity supply on the venetian blinds and the air temperature of the outside window is controlled at 37.4°F (3.0°C). The radiator is switched on and its surface temperature is 65.0°C. A minimum amount of fresh air, 0.02 lb/s (0.009 kg/s), corresponding with a mechanical ventilation rate of 0.5 times per hour, is supplied from the inlet to meet comfort requirements.

Figure 6 shows the computational air velocities and temperature distributions for the displacement ventilation system in the cooling and heating situations. There is a large vertical temperature difference in room air in the cooling situation but almost no temperature difference in the heating situation.

The computational air velocities and temperature distributions of the well-mixed system are illustrated in Figure 7. The temperature distributions of the room air are very uniform, both in the cooling and the heating situations, because the mixture of the room air is very good. Therefore, there is almost no vertical temperature difference in the room air.

Space Loads of the Room

As mentioned in the preceding section, the influence of indoor air distribution on annual energy consumption is simulated by the temperature differences of room air. Because there is no temperature difference in the well-mixed system, the ΔT in Equation 1 can then be written as:

$$\Delta T = 0 \tag{6}$$

Equation 6 is also valid for heating with the displacement ventilation system, because the temperature gradients are small. However, ΔT is not zero for cooling.

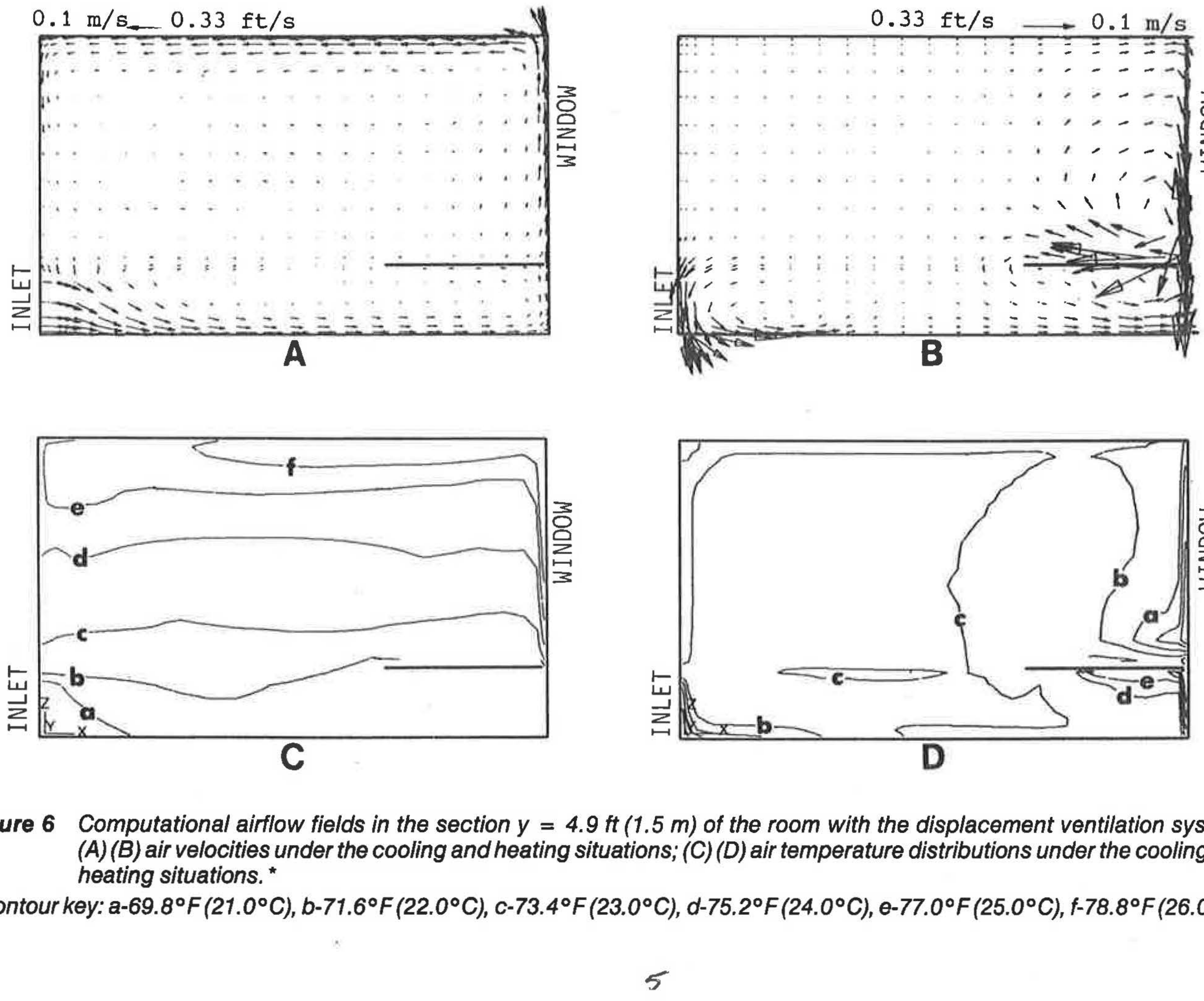


*Figure 6 Computational airflow fields in the section y = 4.9 ft (1.5 m) of the room with the displacement ventilation system. (A) (B) air velocities under the cooling and heating situations; (C) (D) air temperature distributions under the cooling and heating situations.**

**Contour key: a-69.8°F (21.0°C), b-71.6°F (22.0°C), c-73.4°F (23.0°C), d-75.2°F (24.0°C), e-77.0°F (25.0°C), f-78.8°F (26.0°C).*

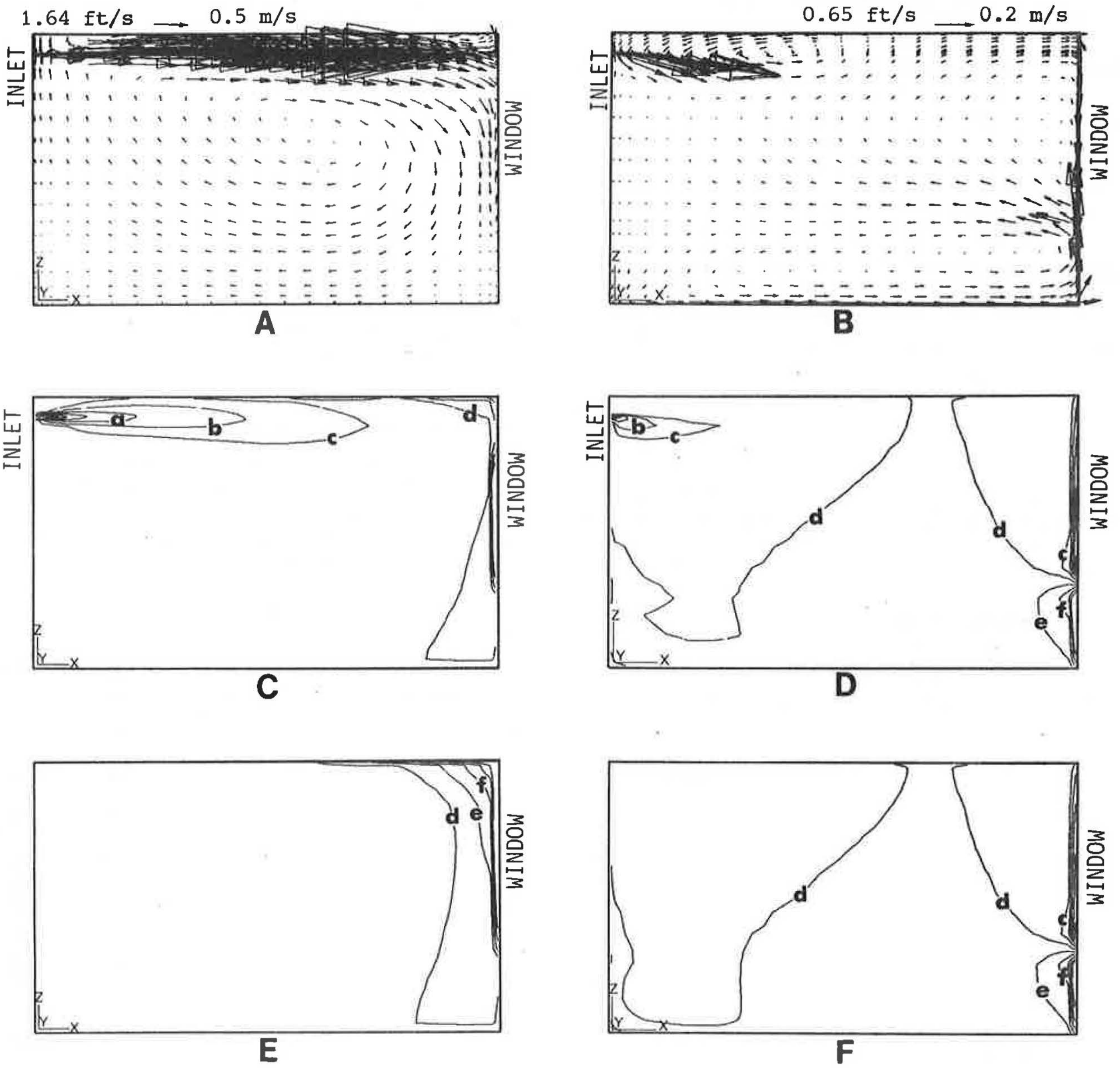


Figure 7 *Computational airflow fields in the room with the well-mixed system. (A) (B) air velocities in section y = 2.5 ft (0.75 m) (via inlet) under the cooling and heating situations; (C) (D) air temperature distributions in section y = 2.5 ft (0.75 m) under the cooling and heating situations*; (E) (F) air temperature distributions in section y = 4.9 ft (1.5 m) (middle section) under the cooling and heating situations.**

**Contour key: a-68.0°F (20.0°C), b-69.8°F (21.0°C), c-71.6°F (22.0°C), d-73.4°F (23.0°C), e-75.2°F (24.0°C), f-77.0°F (25.0°C).*

When the supplied air temperature is fixed, such as in a variable-air-volume air-handling system, space load is related to ventilation rates of the room. Hence, the function in Equation 1 can be simplified as:

$$\Delta T = f\,(Vent) \tag{7}$$

From the air temperature distributions that were calculated from the airflow program (Chen et al. 1988), the following equations have been found by curve fitting:

$$\Delta T_{ceiling} = 1.8\,(0.088 + 0.59 \cdot Vent + 0.06 \cdot Vent^2)\ (°F) \tag{8}$$
(I-P unit)

or

$$\Delta T_{ceiling} = (0.088 + 0.59 \cdot Vent + 0.06 \cdot Vent^2)\ (°C) \tag{8}$$
(SI unit)

and

$$\Delta T_{floor} = 1.8\,(-0.005 - 0.175 \cdot Vent + 0.015\ Vent^2)\ (°F) \tag{9}$$
(I-P unit)

or

$$\Delta T_{floor} = (-0.005 - 0.175 \cdot Vent + 0.015 \cdot Vent^2)\ (°C) \tag{9}$$
(SI unit)

TABLE 2
Total Heat Extraction and Heat Supply in the Displacement Ventilation System and the Well-Mixed System (kWh)

Seasons	Heat extraction		Heat supply	
	Displacement	Well-mixed	Displacement	Well-mixed
Winter	0.0	0.0	57.5	57.4
Spring	23.2	25.2	3.9	4.0
Summer	50.1	53.7	0.0	0.0
Autumn	3.5	14.5	4.3	4.4
Total	86.8	93.4	65.7	65.8

With Equations 8 and 9 and the weather data of the short reference year, ACCURACY calculated the annual heat extraction and heat supply of the room with the two air supply and exhaust systems. The results are presented in Table 2. Due to the existence of a vertical temperature difference, the room air temperature near the window in the displacement ventilation system is higher than that in the well-mixed system. On the other hand, the displacement system uses the ceiling heat capacity more effectively in this case. Therefore, the heat gain from the window for the displacement ventilation system is smaller in cooling conditions. As a result, the heat extracted is less, but the difference between these two systems is not very evident (about 8%). However, Table 3 indicates that the total air mass supplied for cooling in the displacement ventilation system is about 37% smaller than that in the well-mixed system.

Since the outlets in the displacement ventilation system are near the ceiling, the extracted air temperature in cooling conditions is higher than that in the well-mixed system (with an average value of 2.83°F [1.57°C] of the temperature difference between the inlet and outlet air) and inlet air temperature is fixed at 60.8°F (16.0°C). This is why the required mass flow for the displacement ventilation system is considerably smaller.

Annual Energy Consumption of the Room

In the room shown in Figure 1, the rooms above, below, and next to it are assumed to be at the same condition as the room considered. The wall with the window is the only exterior wall. There are 55 office hours each week, i.e., from 7:00 a.m. to 18:00 p.m. Monday through Friday. During these office hours, the air temperature in the middle of the occupied zone is maintained between 71.6°F ±3.6 R (22.0°C ±2.0 K) and the absolute humidity is between 42.0 gr/lb and 63.0 gr/lb (6.0 g/kg and 9.0 g/kg). If the air temperature is in the range between 68.0° and 75.2°F (20.0° and 24.0°C), there is no heat extraction in the room. However, ventilation is always necessary for fresh air during office hours. It is supposed that there is always 1024 Btu/h (300 W) convective heat released into the room air during office hours. This amount of heat is considered to be the convective heat from occupants, lights, and appliances in the room. For weekends and nights, there is no internal heat gain in the room and the air-conditioning system is switched off. Therefore, there is no energy consumption during weekends and nights. From April to September, an external venetian blind is used to reduce heat gain through the window.

The efficiencies of the primary equipment (boiler, ventilator, and chiller) are assumed to be constant. The pressure of the ventilator is 0.2 lb/in^2 (1400 Pa) and its efficiency is 0.6. The efficiency of the boiler is 0.75 and the COP of the chiller is 3.5.

TABLE 3
Total Amount of Air Supply Required for Cooling in the Displacement Ventilation System and the Well-Mixed System

Seasons	Displacement	Well-mixed
(I-P units) (10^3 lb)		
Winter	0.0	0.0
Spring	17.42	24.76
Summer	38.46	52.73
Autumn	10.73	14.23
Total	66.65	91.72
(SI units) (10^3 kg)		
Winter	0.0	0.0
Spring	7.91	11.24
Summer	17.46	23.94
Autumn	4.87	6.46
Total	30.24	41.64

The energy analysis is performed for two types of air-conditioning systems, i.e., a variable-air-volume system and a constant-air-volume system. The variable-air-volume system and the constant-air-volume system here mean that the air-handling processes are the same as those for normal variable-air-volume systems and constant-air-volume systems, but the inlet and outlet locations are different.

Variable-Air-Volume System In the variable-air-volume system, the two kinds of air supply and exhaust systems shown in Figure 1 are studied. When the displacement ventilation system is used, the supply air temperature must be higher than 60.8°F (16°C). Thus, the air temperature distribution in the occupied zone satisfies comfort requirements. But, the supply air temperature for the well-mixed system can be lower because the fresh air is supplied outside of the occupied zone. In order to compare the energy consumption with the two air supply and exhaust systems, the computations are carried out in two groups: with the same supply air temperature and with different supply air temperatures.

With the Same Supply Air Temperature In this subsection, we will discuss the results for the two air supply and exhaust systems with the same lowest supply air temperature (60.8°F [16.0°C]).

From the view of energy saving, it is better to use different kinds of control strategies for different indoor and outdoor conditions. Van Paassen (1981) developed a set of optimal control strategies based on a psychrometric chart, and they are used for the present research.

It is very possible that the room sometimes needs heating and sometimes requires cooling although the outdoor climate may be in the same section of the psychrometric chart. Because the control strategies for cooling and heating are different, they must be described separately in each section. The control strategies of the variable-air-volume system for cooling and heating with the displacement ventilation system are presented in a psychrometric chart as shown in Figures 8 and 9.

Although the heat extraction in sections 2 and 3 for heating is zero, ventilation is still required for office hours (i.e., $P+$ is not equal to zero). Therefore, the control strate-

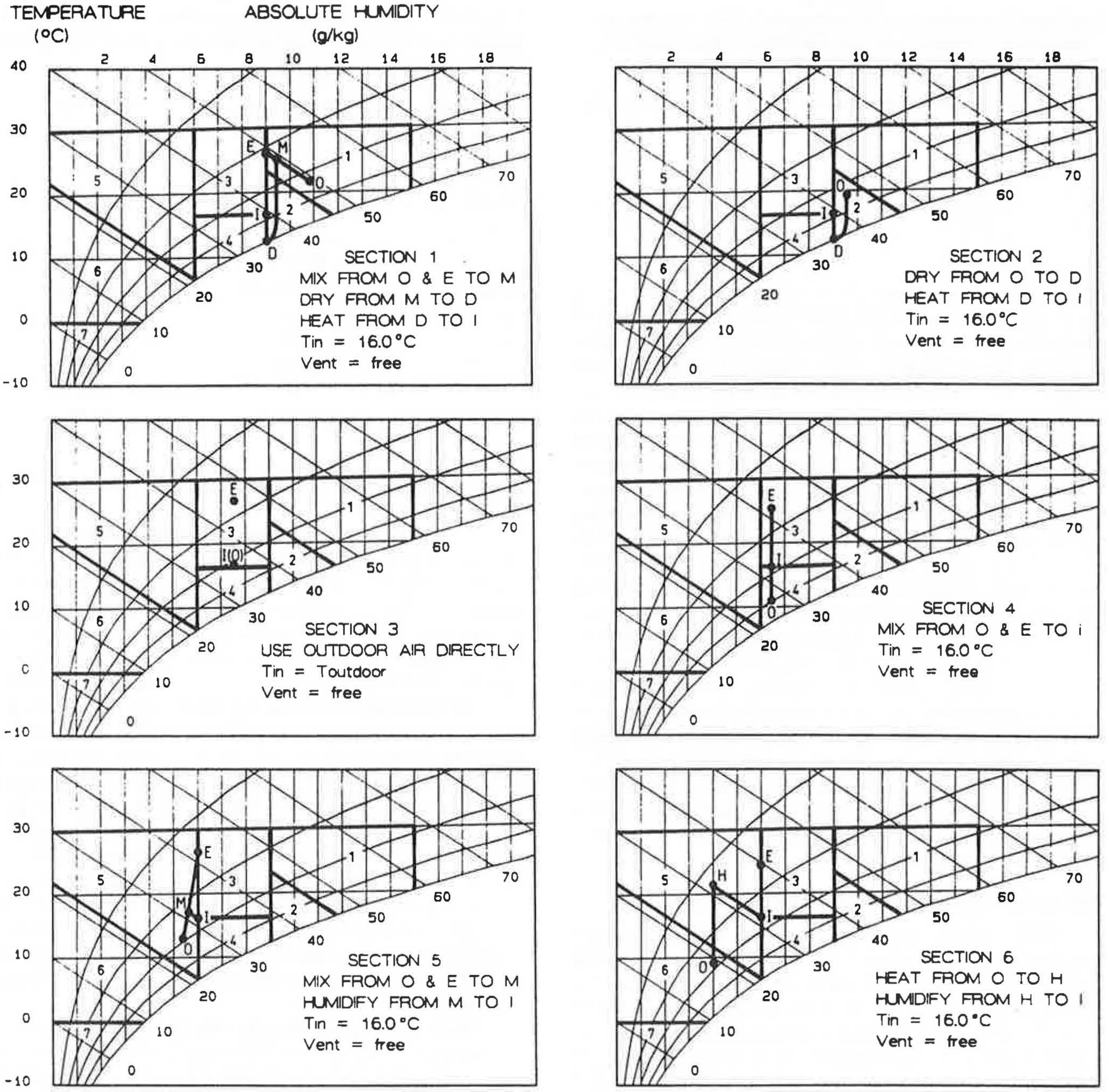


Figure 8 *The control strategies for cooling in the variable-air-volume system with the displacement ventilation system (lowest air supply temperature = 60.8°F [16.0°C]) (O–outdoor air, E–air extracted, D–dew point, I–air supplied by inlet, M–mixing point, H–heating point)*

gies for these two sections are also presented. However, according to the results computed by ACCURACY, there is no heating in section 1 and no cooling in section 7 (i.e., $P+$ in section 1 and $P-$ in section 7 are zero). Thus, the control strategies for these two sections are unnecessary. The control strategies for the displacement ventilation system are the same as those for the well-mixed system.

Table 4 presents the heat removed by cold water and the heat supplied by hot water (via the heater and the radiator) in different sections of the psychrometric chart for the displacement ventilation system (with an average vertical air temperature difference of 3.51°F [1.95°C] for cooling). Table 5 shows the results for the well-mixed system (without a vertical air temperature difference for cooling).

TABLE 4
Heat Extraction and Heat Supply of the Room, Heat Removed by Cold Water, and Heat Supplied by Hot Water in the Variable-Air-Volume System with the Displacement Ventilation System (kWh)

Sections in psychro. chart	1	2	3	4	5	6	7	Total
Heat extraction (cooling)	114.3	121.7	239.2	17.9	69.8	2.8	—	565.7
Heat removed by cold water	154.0	104.7					—	258.7
Heat supplied by hot water	37.2	41.6				5.9	—	84.7
Heat supply (heating)	—	0.0	0.0	26.6	25.3	199.0	177.1	428.0
Heat removed by cold water	—	1.8						1.8
Heat supplied by hot water	—	5.0	6.2	51.6	51.8	347.2	244.5	706.3

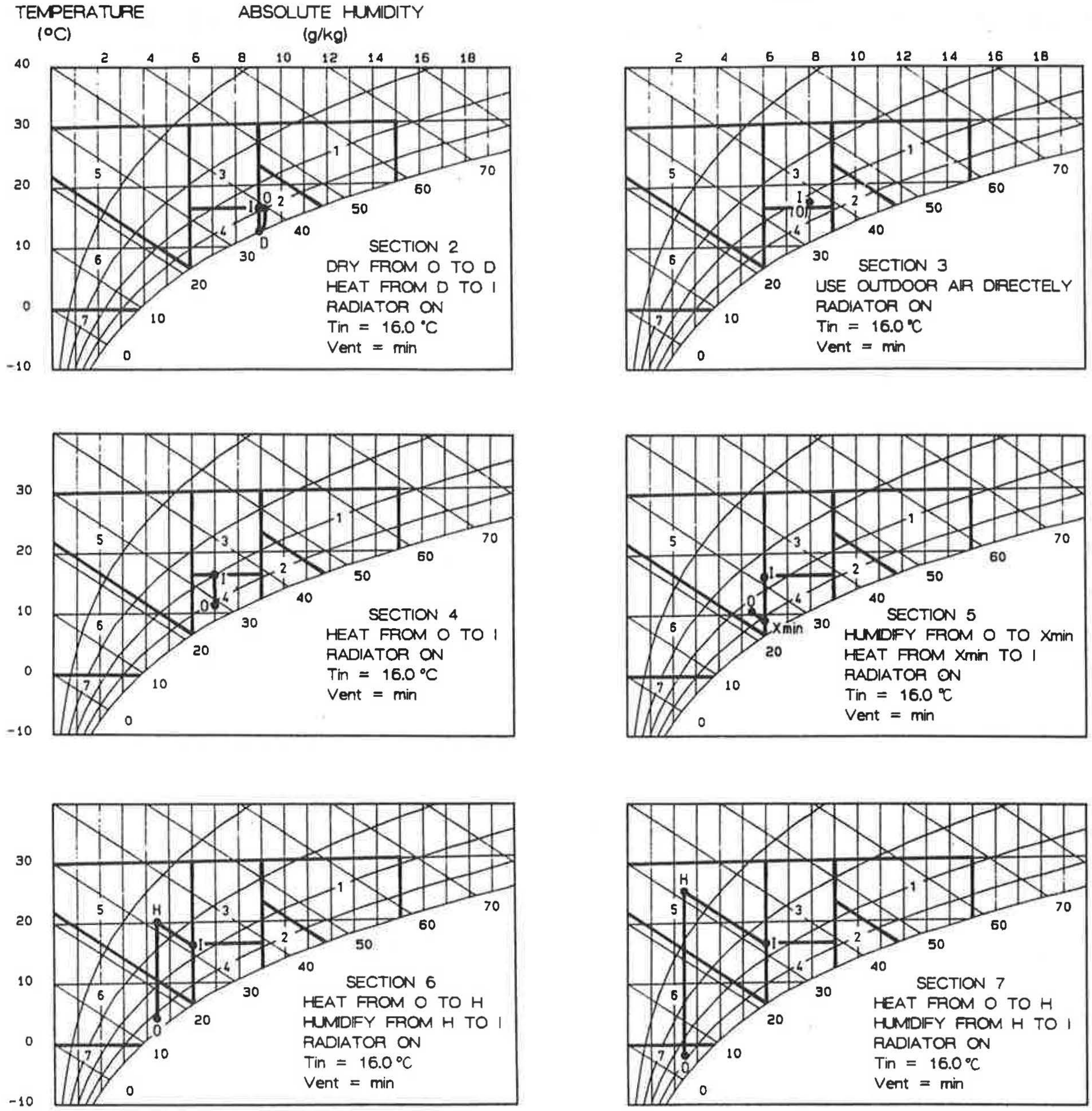


Figure 9 *The control strategies for heating in the variable-air-volume system with the displacement ventilation system (lowest air supply temperature = 60.8° F [16.0° C]) (O–outdoor air, E–air extracted, D–dew point, I–air supplied by inlet, M–mixing point, H–heating point)*

Although the computations are done with the weather data of the short reference year, the results have been extended for a normal reference year by multiplying by a factor of 2868/440, where 2868 and 440 are the numbers of office hours for the normal reference year and the short reference year, respectively.

From these two tables, we can see that the relationship between the heat extraction, heat supply, heat removed by cold water, and heat supplied by hot water is very complicated. In section 1, as shown in Figure 8, for example, the enthalpy difference between the outdoor air and the dew point is higher than that between the extracted air and the air supplied. Therefore, the heat removed by cold water is higher than the heat extraction. However, in section 3, free

TABLE 5
Heat Extraction and Heat Supply of the Room, Heat Removed by Cold Water, and Heat Supplied by Hot Water in the Variable-Air-Volume System with the Well-Mixed Ventilation System (kWh)

Sections in psychro. chart	1	2	3	4	5	6	7	Total
Heat extraction (cooling)	124.6	132.4	254.2	18.4	76.6	2.9	—	609.1
Heat removed by cold water	182.7	139.3					—	322.0
Heat supplied by hot water	51.6	54.9				5.9	—	112.4
Heat supply (heating)	—	0.0	0.0	26.6	25.3	200.1	177.0	429.0
Heat removed by cold water	—	1.8						1.8
Heat supplied by hot water	—	5.6	9.4	51.6	51.8	347.2	244.5	710.1

TABLE 6
Energy Consumption in the Variable-Air-Volume System with the Displacement Ventilation System (kWh)*

Sections in psychro. chart	1	2	3	4	5	6	7	Total
(During Cooling Period)								
Chiller	44.0	29.9					—	73.9
Ventilator	21.0	23.5	52.7	3.8	13.5	0.8	—	115.3
Boiler	49.6	55.4				7.8	—	112.8
(During Heating Period)								
Chiller	—	0.5						0.5
Ventilator	—	0.9	1.3	4.8	4.1	13.5	4.1	28.7
Boiler	—	6.6	8.3	68.8	69.1	462.9	326.0	941.7

*The lowest supply air temperature is 60.8°F (16.0°C).

TABLE 7
Energy Consumption in the Variable-Air-Volume System with the Well-Mixed Ventilation System (kWh)*

Sections in psychro. chart	1	2	3	4	5	6	7	Total
(During Cooling Period)								
Chiller	52.2	39.8					—	92.0
Ventilator	29.2	31.0	70.6	4.3	17.9	0.8	—	153.8
Boiler	68.8	73.2				7.2	—	149.2
(During Heating Period)								
Chiller	—	0.5						0.5
Ventilator	—	1.0	2.0	4.8	4.1	13.5	4.1	29.5
Boiler	—	7.5	12.5	68.6	60.1	462.9	326.0	946.6

*The lowest supply air temperature is 60.8°F (16.0°C).

cooling can be used so that heat removed by cold water is zero.

Tables 4 and 5 also show that the difference of the heat removed by cold water between the two cases is larger than that of the heat extraction. The reason is that the difference of the amount of air mass supplied in the two cases is large, as shown in Table 3.

Table 6 presents the computational results of energy consumption during the cooling and heating period for the displacement ventilation system (with an average vertical air temperature difference of 3.51°F [1.95°C] for cooling). Table 7 gives the results for the well-mixed system (without a vertical air temperature difference for cooling). The comparison of energy consumption between the two cases is illustrated in Figure 10. The annual energy consumption by the chiller and the ventilator for the displacement ventilation system is 26% smaller than that for the well-mixed system and by the boiler it is 3% smaller.

According to the results computed by ACCURACY, as shown in Table 4, the heat extraction for cooling is higher than the heat supply for heating. But the total energy consumption of the boiler seems to be 10 times higher than that of the chiller and the ventilator, as indicated in Tables 6 and 7. This is because the efficiency of the boiler is low (0.75), the COP of the chiller is high (3.5), and a part of the energy is required for reheating during the cooling period. However, the prices for gas, which is used by the boiler, and for electricity, which is used by the chiller and the ventilator, are different. According to an electricity company in Amsterdam, one kWh of electricity costs Dfl. 0.272 (Dutch Guilders) and one m^3 of gas costs Dfl. 0.447. Because one m^3 of gas delivers 9.72 kWh of heat, one kWh of gas energy costs Dfl. 0.046. With these prices, the costs of annual energy consumption are shown in Table 8. It is clear that the cost for the chiller and the ventilator is higher than that for the boiler. The total cost of energy consumption for the displacement ventilation system (with an average vertical air temperature difference of 3.51°F [1.95°C] for cooling) is about $(125.6 = 108.0)/108.0 \times 100\% = 16\%$ less than that for the well-mixed system (without a vertical air temperature difference for cooling).

With Different Supply Air Temperature This subsection will demonstrate the results of the displacement ventilation system and the well-mixed system with different supply air temperatures. The lowest supply air temperature for the displacement ventilation system is assumed to be 60.8°F (16.0°C) and for the well-mixed system to be 54.5°F (12.5°C). Because the supply air temperature for the well-mixed system is lower, the air mass flow supplied can be smaller in sections 1, 2, 4, 5, and 6 for cooling, as shown in Figure 8. Besides, reheating becomes unnecessary in sections 1 and 2 during the cooling period.

For the well-mixed system, there is no vertical air temperature difference in the room. Thus, the heat extraction for cooling and the heat supply for heating with the lowest air supply temperature (54.5°F [12.5°C]) is the same as those in the case with the lowest supply air temperature (60.8°F [16.0°C]) as shown in Table 5.

Since reheating is unnecessary for the well-mixed system (supply air temperature of 54.5°F [12.5°C]) and the supply mass flow is smaller in some sections during the cooling period, the energy consumption is smaller in these

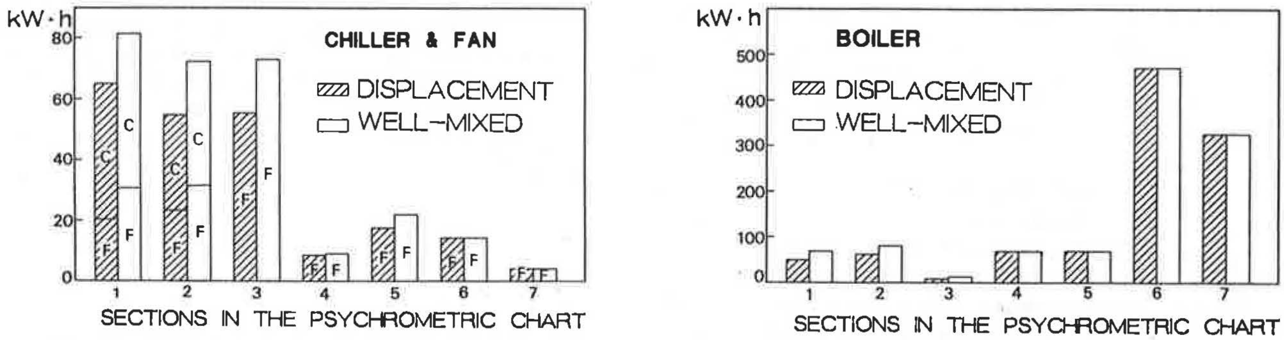


Figure 10 *The annual energy consumption of the variable-air-volume system with the displacement ventilation system and the well-mixed system (C–chiller, F–fan)*

TABLE 8
Annual Cost of Room Energy Consumption (Dutch Guilders)

Air handling	Air supply sys.	Inlet temperature	Chiller	Ventilator	Boiler	Total
Variable-air-volume	Displacement	60.8°F (16.0°C)	20.27	39.22	48.51	108.0
	Well-mixed	60.8°F (16.0°C)	25.20	49.93	50.44	125.6
	Well-mixed	54.5°F (12.5°C)	17.57	42.52	43.67	103.8
Constant-air-volume	Displacement	Variable	34.32	136.96	69.80	241.1
	Well-mixed	Variable	31.41	132.05	58.94	222.4

TABLE 9
Energy Consumption in the Variable-Air-Volume System with the Well-Mixed Ventilation System (kWh)*

(During Cooling Period)								
Sections in psychro. chart	1	2	3	4	5	6	7	Total
Chiller	36.3	27.7					—	64.0
Ventilator	20.3	21.6	70.6	1.6	12.0	0.5	—	126.6
Boiler						2.8	—	2.8

*The lowest supply air temperature is 54.5°F (12.5°C).

sections. The computed results are presented in Table 9. But, the energy consumption for the well-mixed system during the heating period is the same as that given in Table 7. Compared with the results of the displacement ventilation system (supply air temperature 60.8°F [16.0°C]) shown in Table 6, the annual energy consumption of the chiller and the ventilator of the well-mixed system (supply air temperature 54.5°F [12.5°C]) is the same and that of the boiler is 10% smaller. The cost of annual energy consumption in the well-mixed system with a supply air temperature of 54.5°F (12.5°C) is given in Table 8. This is 4% less than or nearly the same as that in the displacement ventilation system with a supply air temperature of 60.8°F (16.0°C).

Of course, the cost of annual energy consumption is not the only standard for evaluating a system. It should be based on the same comfort standard and the same cost of the air-conditioning equipment, etc. Indoor air velocity, air temperature, and contaminant distributions are very different between the displacement ventilation system and the well-mixed system. Some people may prefer the displacement ventilation system for better air quality and some may need the well-mixed system for a uniform indoor air temperature distribution. The indoor environment of each system has been discussed by Chen et al. (1988).

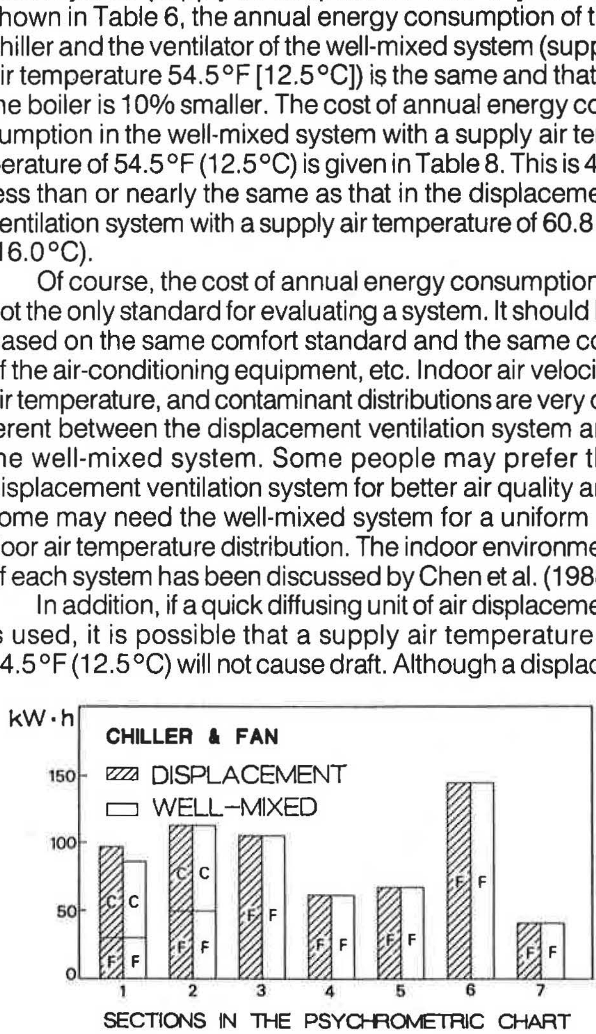
In addition, if a quick diffusing unit of air displacement is used, it is possible that a supply air temperature of 54.5°F (12.5°C) will not cause draft. Although a displacement system with a supply air temperature of 54.5°F (12.5°C) has not been calculated, the lowest energy consumption may be expected (for the same reasons as discussed in the previous subsection). The development of such an air displacement unit is possible. Designers of air-conditioning systems are advised to use this kind of system for energy saving and better indoor air quality.

Constant-Air-Volume System In the constant-air-volume system, the supplied mass flow is maintained at 0.20 lb/s (0.09 kg/s) (a ventilation rate of 5 times per hour) for all office hours. The energy consumption is also calculated for the displacement ventilation system and the well-mixed system. Because no radiator is used for heating in the displacement ventilation system (normal all-air system), there is always a vertical temperature difference in the room air, both for cooling and heating. Since the room air is mixed very well in the well-mixed system, both for heating and for cooling, no vertical temperature exists.

The computational results of the energy consumption for the two cases are presented in Tables 10 and 11. The comparison of the energy consumption between the two cases is illustrated in Figure 11. The annual energy consumption of the chiller and the ventilator in the displacement ventilation system (with a vertical air temperature difference) is 2% larger than that in the well-mixed system (without a vertical air temperature difference) and that of the boiler is 16% larger.

As mentioned above, there is always a vertical temperature difference of room air in the displacement ventilation system. Because the mass flow supplied is fixed and the air temperature extracted from the room is higher, the supply air temperature of the room must be higher too. This will use more energy to reheat the air from the dew point to the supply point.

The cost of energy consumption for the constant-air-volume system is also shown in Table 8. The total cost for the displacement ventilation system is (241.1 − 222.4)/ 222.4 × 100% = 8% higher than that for the well-mixed system.



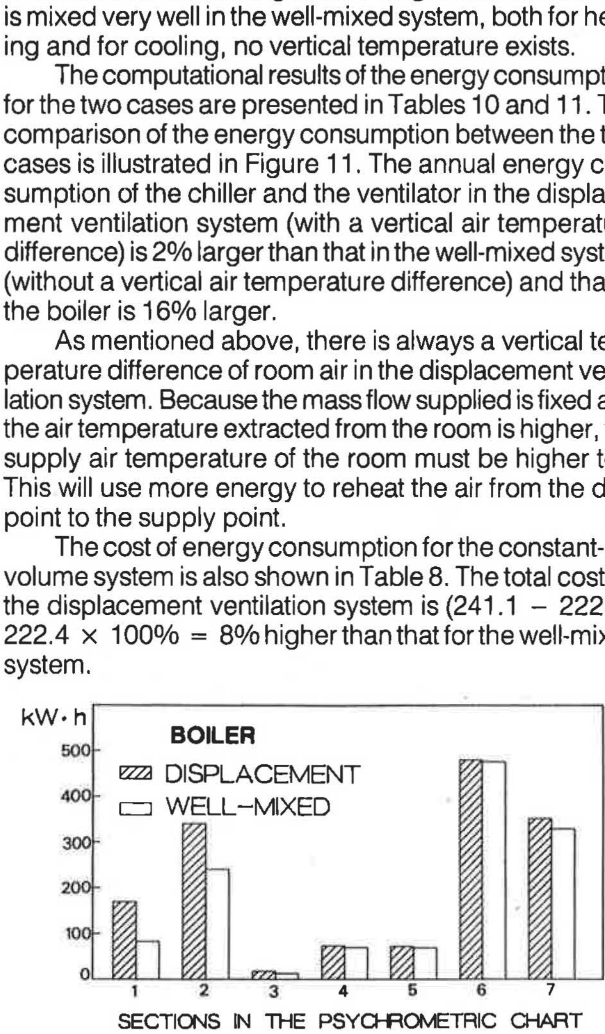


Figure 11 *The annual energy consumption of the constant-air-volume system with the displacement ventilation system and the well-mixed system (C–chiller, F–fan)*

TABLE 10
Energy Consumption in the Constant-Air-Volume System with the Displacement Ventilation System (kWh)

(During Cooling Period)								
Sections in psychro. chart	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Total**
Chiller	65.9	55.4					—	121.3
Ventilator	30.9	43.4	92.6	13.7	26.3	9.2	—	216.1
Boiler	169.1	273.0				9.7	—	451.8
(During Heating Period)								
Sections in psychro. chart	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Total**
Chiller	—	4.7						4.7
Ventilator	—	9.1	12.6	48.0	41.1	134.8	41.1	286.7
Boiler	—	67.5	7.8	74.4	73.4	479.3	363.2	1065.6

TABLE 11
Energy Consumption in the Constant-Air-Volume System with the Well-Mixed Ventilation System (kWh)

(During Cooling Period)								
Sections in psychro. chart	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Total**
Chiller	55.1	54.3					—	109.4
Ventilator	30.9	42.3	84.6	13.7	26.3	8.0	—	205.8
Boiler	82.7	165.3				7.3	—	255.3
(During Heating Period)								
Sections in psychro. chart	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Total**
Chiller	—	5.9						5.9
Ventilator	—	10.2	20.6	48.0	41.1	136.0	41.1	279.0
Boiler	—	75.9	12.7	69.4	69.7	469.2	329.2	1026.1

From the results of the variable-air-volume system and the constant-air-volume system, the energy consumption of the variable-air-volume system is much smaller than that of the constant-air-volume system. It is clear that the constant-air-volume system is not good for energy saving. However, this strongly depends on the supplied airflow rate in the constant-air-volume system.

From the results presented in this paper, we can see that only the gas and electricity cost is considered and not the cost of air-conditioning equipment. On the other hand, the humidity is only discussed with its absolute values instead of its relative values. Since the division of the psychrometric chart will be very complicated if the relative values of humidity are used, the moisture generated from occupants and indoor equipment is not taken into consideration. If all of these are considered, the final results may be different. Further investigation of this problem is necessary.

CONCLUSIONS

The following paragraphs summarize the main points from this paper:

1. The air supply and exhaust systems of a room have a significant influence on building energy consumption. They must be considered in space load calculations and the prediction of energy consumption of buildings.

2. In the variable-air-volume system and with the same lowest supply air temperature (60.8°F [16.0°C]), the displacement ventilation system (with an average vertical air temperature difference of 3.51°F [1.95°C] for cooling) will save 26% energy on the chiller and the ventilator and 3% on the boiler, compared with the well-mixed system (without a vertical air temperature difference for cooling). This is because the mass flow supplied for the displacement ventilation system is lower all the time. The cost of annual energy consumption for the displacement ventilation system is 16% smaller than that for the well-mixed system.

3. However, if the lowest supply air temperature for the displacement ventilation system (with an average vertical air temperature difference of 3.51°F [1.95°C] for cooling) remains at 60.8°F (16.0°C) and that for the well-mixed system (without a vertical air temperature difference for cooling) is controlled at 54.5°F (12.5°C), the cost of annual energy consumption is nearly the same.

4. The development of a quick diffusing unit of air displacement is possible so that a supply air temperature of 54.5°F (12.5°C) will not cause draft. The lowest energy consumption may be expected in this case. Designers of air-conditioning systems are advised to use this kind of air displacement system for energy saving and better indoor air quality.

5. In the constant-air-volume system, the displacement ventilation system (with a vertical temperature difference both for cooling and heating) requires 2% more energy for the chiller and ventilator and 16% more for the boiler, compared with the well-mixed system (without a vertical air temperature difference for both cooling and heating).

6. The energy cost for the constant-air-volume system is twice as much as that for the variable-air-volume system under the circumstances discussed here. Hence, the constant-air-volume system is not good for energy saving. All the conclusions are obtained from the energy consumption of the air-conditioning equipment. The moisture generated from occupants and indoor equipment is not considered in this paper. Further investigation of the cost of the air-conditioning equipment and the influence of moisture on energy costs is suggested.

REFERENCES

ASHRAE. 1985. *ASHRAE handbook—1985 fundamentals*. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Chen, Q. 1988. "Indoor airflow, air quality and energy consumption of buildings." Ph.D. thesis, Delft University of Technology.

Chen, Q., and Van der Kooi, J. 1988. "ACCURACY—a computer program for combined problems of energy analysis, indoor airflow and air quality." *ASHRAE Transactions*, Vol. 94, Part 2.

Chen, Q.; Van der Kooi, J.; and Meyers, A. 1988. "Measurements and computations of ventilation efficiency and temperature efficiency in a ventilated room." *Energy and Buildings*, Vol. 12, No. 2.

Danielsson, P.O. 1987. "Convective flow and temperature in rooms with displacement system." *Proceedings of International Conference on Air Distribution in Ventilated Spaces*, Vol. 4b, Stockholm, Sweden, June.

Gunton, M.C.; Rosten, H.I.; Spalding, D.B.; and Tatchell, D.B. 1983. *PHOENICS: an instruction manual*. Report No. TR/75/. London: CHAM Ltd.

Irving, A.D. 1988. "Validation of dynamic thermal models." *Energy and Buildings*, Vol. 10, pp. 213-220.

Judkoff, R.D. 1988. "Validation of building energy analysis simulation programs at the Solar Energy Research Institute."

Energy and Buildings, Vol. 10, pp. 221-239.
Kimura, K. 1977. *Scientific basis of air conditioning*. London: Applied Science Pub. Ltd.
Launder, B.E., and Spalding, D.B. 1974. "The numerical computation of turbulent flows." *Computer Methods in Applied Mechanics and Energy*, Vol. 3, pp. 269-289.
Liem, S.H., and van Paassen, A.H.C. 1982. "Establishment of short reference years for calculation of annual solar heat gain or energy consumption in residential and commercial buildings (part 1)." Final Report, Contract No. ESF-010-80 N(B), Delft University of Technology.
Rosten, H.I., and Spalding, D.B. 1981. "The mathematical basis of the PHOENICS EARTH computer code." Report No. TR/58b/. London: CHAM Ltd.
Stephenson, D.G., and Mitalas, G.P. 1971. "Calculation of heat conduction transfer functions for multilayer slabs." *ASHRAE Transactions*, Vol. 73, Part 2.
Van Mierlo, W.J.M. 1986. "Onderzoek naar de nauwkeurigheid van het programma 'ENERK.'" Report No. ST-255, Lab. for Refrigeration and Indoor Climate Technology, Delft University of Technology.
Van Paassen, A.H.C. 1981. "Indoor climate, outdoor climate and energy consumption." Ph.D. thesis, Delft University of Technology.
Van Paassen, A.H.C. 1986. "Design of low energy HVAC systems with a computerized psychrometric chart." *Proceedings of the 2nd International Conference on System Simulation in Buildings*, Liege, December.