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ACCURACY—A PROGRAM FOR COMBINED PROBLEMS OF ENERGY ANALYSIS, INDOOR AIRFLOW, AND AIR QUALITY

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ABSTRACT

A new user-friendly, personal computer program, ACCURACY, has been constructed for energy analysis, room air temperature and contamination field predictions. The program is a coupling of a cooling load program and an airflow program. It is based on the room energy balance method and uses Z-transfer functions and window energy balance equations for heat transfer through enclosures. The computer program also involves the solution, in finite-domain form, of three-dimensional, transient equations for the conservation of energy and contamination concentration using airflow patterns, which are precalculated from dedicated computer programs, such as PHOENICS (3D) and CHAMPION SGE (2D). The influence of room air supply system on the temperature distribution and indoor air quality and subsequently on the energy consumption can be calculated by ACCURACY. The agreement between measured and simulated results by ACCURACY is good.

INTRODUCTION

Most available normal cooling load computer programs are based on the one-air-point model, which means that the values of the whole temperature field in a room are assumed to be uniform. In many air-conditioned rooms with a high ventilation rate, this kind of treatment can be accepted because the room air temperatures are the same. However, if this one-air-point model is applied to an air-conditioned room with a low ventilation rate or with natural convection or to big industrial halls or theaters, a large error in the energy consumption will result. This is due to the presence of temperature gradients in room air ($\partial T/\partial x$, $\partial T/\partial y$ and $\partial T/\partial z$). These gradients, especially the one that is generated from buoyancy in the vertical direction ($\partial T/\partial z$), play an important role in the computations of an air-conditioning load.

In order to study the influence of the temperature gradients, van der Kooi et al. (1983, 1985, 1987) developed three improved methods for the computation of air-conditioning loads: (1) adjusting the convective heat-exchange coefficient near the inside wall surface, (2) introducing constant temperature differences between the middle point of the room and the air layers adjacent to ceiling and floor, and (3) using more air points, each representing a part of the room and mutually connected by adjustable mass flows. The first method gives unrealistic physical results, since the adjusted heat exchange coefficients have to be negative sometimes in order to calculate the right heat flux into the ceiling, floor, and wall surfaces. The second method requires that temperature differences are obtained from measurements, and they are not always available. Van der Kooi and Chen (1987) demonstrated in a previous paper that, for situations with a varying cooling load, the constant-temperature method gave results different from the measured ones, because the temperature differences change with time. In the third method, the results are noted to depend critically on the mass flow pattern used. These mass flows are usually obtained from measurements. The more-air-points model with 9 or 16 points cannot be applied to describe the complicated airflow structure in a room because more points are needed. These methods, therefore, cannot account precisely for the influence of room

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air movement and temperature distribution on room energy consumption.

Since the early 1970s, many computer programs have been written for the calculation of heat transfer and fluid flow. A much greater reliability on the calculation of fully turbulent flow was obtained by introducing the k-ε model (Harlow and Nakayama 1968; Launder and Spalding 1974). However, this turbulent model cannot properly describe the airflow phenomenon in an air-conditioned room when the turbulent Reynolds number ($Re_t = k^2/\nu\varepsilon$) is very low (Spalding 1981). The results predicted using the 3D computer program PHOENICS (Rosten and Spalding 1981) and the 2D computer program CHAMPION SGE (Chen and van der Kooi 1987), both of which are based on the k-ε model, are not in perfect agreement with measured ones. However, flow predictions from these two programs are good enough for energy consumption simulations in buildings. The inputs required by an airflow program are enclosure surface temperatures, inlet and outlet locations, mass flow rate, temperature, etc., which can be obtained from the outputs of a cooling load program. However, the outputs of the airflow program, such as room air velocity, temperature and contamination distributions, and convective heat exchange coefficients, are a part of the inputs for the cooling load program. Therefore, the combination of a normal cooling load program with an airflow program can be used to study the influence of the indoor airflow and temperature distribution on room energy consumption. An approach to this subject is reported in this paper.

THE FUNDAMENTALS OF THE COMPUTER PROGRAM ACCURACY

With Airflow Patterns

Based on the idea of combining a cooling load program with an airflow program, a new nonproprietary program, ACCURACY (which is the acronym for A Cooling-load Code Using Room Air Currents; the final Y is added for euphony), has been constructed (Chen 1987a, 1987b) in order to find a better agreement between computed and measured results of room air-conditioning load and temperatures. The concentration of contamination, air velocity, and temperature distributions can be also predicted with this program for evaluating comfort. The program uses the room energy balance method (Kimura 1977) and the equations used are:

$$[M] \cdot [T] = [q] + [\alpha\ \Delta T] \qquad (1)$$

A more detailed definition of the equations is given in Appendix A.

The main difference between ACCURACY and other cooling-load programs with regard to the equations is that the former considers transient temperature differences between the air near the inside surfaces of an enclosure and the middle point of the room [ΔT]. This term can be obtained from the temperature distributions that are updated hourly from an airflow computer program, such as PHOENICS or CHAMPION SGE. However, according to Chen and van der Kooi (1987), it is too expensive to get the time-dependent airflow and temperature distributions of a room from an airflow computer program based on the k-ε turbulence model. This is due to the properties of the fluid and turbulence governing equations as well as the numerical method, which required a large amount of grid numbers to give a good prediction. For three-dimensional computations, a grid number at least more than 2000 is required for an air conditioned room with nonsymmetrical boundary conditions. This is equivalent to approximately 16 minutes CPU time in a mainframe computer for a steady situation. For these reasons, therefore, a direct combination between a cooling-load program and the airflow program will be unacceptable for practical use. It has been demonstrated by Chen and van der Kooi (1987) that under a forced convection situation, the airflow pattern seems to be independent of time, although there may be a great change in the air temperature of the room. This is explained by examining the governing equations of the airflow programs PHOENICS and CHAMPION SGE:

$$\frac{\partial}{\partial t}(\rho\phi)+\mathrm{div}(\rho\vec{V}\phi-\Gamma_\phi \mathrm{grad}\ \phi)=S_\phi+S_{Buoyancy} \qquad (2)$$

The detailed information about the equation is given in Table 1.

It is clear that temperature and concentration equations have no direct influence on the transient, convective, and diffusive terms of velocity components. However, if the thermal boundary conditions can be estimated somehow, the buoyancy contribution on the source terms of the velocities and the turbulent parameters, such as Γ_ϕ, can be determined for typical situations. When a simulation of annual energy consumption is required, the computer program searches for one of the airflow patterns that were precalculated as the airflow distribution at the moment according to the corresponding thermal boundary conditions. An hourly computation of room air temperature distribution is necessary because the thermal boundary conditions may vary significantly, and this may be obtained from the corresponding flow pattern based on the energy equation:

$$\frac{\partial}{\partial t}(\rho H)+div(\rho\vec{V}H-\Gamma_H grad\ H)=0 \tag{3}$$

<u>where</u>

$$H = C_p T \tag{4}$$

In this case, the temperature is considered to be an auxiliary variable and has no direct influence on flow fields, since the convergence is very fast. When the simulation of contamination concentration is required, it can be obtained using the concentration equation:

$$\frac{\partial}{\partial t}(\rho C)+div(\rho\vec{V}C-\Gamma_C grad\ C)=0 \tag{5}$$

This equation can be solved with the same procedure as the energy equation 3, but because the normal cooling-load calculation is based on a one-hour time step, the transient term can be ignored. For some contaminations, such as tobacco smoke, the time step can be as small as one minute and the transient term must then be taken into consideration. Furthermore, they are solved in two different time steps in ACCURACY. The "staggered grid" and the upwind scheme, which are employed in the airflow programs PHOENICS and CHAMPION SGE, are used here for Equations 3 and 5 in ACCURACY. The finite domain form of Equations 3 and 5 is given in Appendix B.

The solution of Equation 1 requires the temperature differences between the air near the inside enclosure surfaces and the middle point of the room. However, the predictions of the room air distributions require thermal boundary conditions, which are provided in the room energy balance equations 1. Iterations between Equations 1 and 3 are necessary for getting convergent results due to the nonlinear property. A diagram for the computer program ACCURACY is shown in Figure 2.

<u>Without Airflow Patterns</u>

When no airflow pattern is available, constant temperature differences between the air near the enclosure surfaces and the middle point of the room can be introduced directly into ACCURACY. This is very useful for initial computations, since ACCURACY runs as fast as normal cooling load programs that were constructed from the room energy balance method. This method can also be applied for a room with natural convection in which the vertical temperature gradient does not change very much. Therefore, it is unnecessary to use airflow patterns for calculating room air temperature distributions.

Another improvement in ACCURACY is that the convective and radiative heat exchange coefficients in the [M] matrix of Equation 1 can be modified. The convective heat exchange coefficients can be calculated from wall functions, as given in Appendix A. The program resets the coefficients if the change is large. In cases where radiative heat exchange coefficients vary rapidly, such as when the radiant panels for heating are switched on and off, these values are recalculated in the program according to the surface temperatures, view factors, and multiple reflections.

In many modern buildings, there is often a cavity above the false ceiling of a room and the predictions of air-conditioning loads or air temperatures of the room and the cavity are related and must be solved together. The computer program ACCURACY is built for this purpose. The cavity is considered as another room, which can be located above, under, or next to the room. The program can be used to predict heat extraction or air temperature in the room or the cavity.

THE STRUCTURE OF THE COMPUTER PROGRAM ACCURACY

A computer simulation of heat transfer in buildings should be a general-purpose program that can be easily adapted for a solution of the particular problem. The common core of the program must be a highly efficient and constantly maintained item of software with which the user can communicate only in carefully controlled ways. At the same time, the program must provide a subroutine in which the program user who needs other facilities that are not provided within the common core can form the required coding structure. The computer program ACCURACY has been constructed to meet those requirements. The present program was developed for personal computers and was written in FORTRAN 77.

The user-friendly computer program ACCURACY has three elements, as shown in Figure 3: (1) STATIC, which provides problem-defining data, (2) DYNAMIC, which is attached to the main part of the program, SOLVER, assisting the latter to complete the simulation tasks, and (3) SOLVER, which is the central equation solver. STATIC is used for static input data and works with the help of group data input subroutines. All variables in the program are equipped with default values that are set in BLOCK DATA. A user requires only minor changes for his need. It is convenient to use the data input subroutines by simply CALL-ing them instead of modifying them. STATIC finishes its work before SOLVER takes off. The information it transmits is therefore one of "once-for-all" characters. For many problems, this is not adequate, such as when the user may want to add into or to extract from the SOLVER more dynamic information that cannot be defined in STATIC, or to modify any variable or parameter in a particular time step, or to offer printout frequency during some especially interesting parts of the process. Therefore, there is often a need for the simulator of a special process to provide coding that interacts with SOLVER during its operations. This function is performed by the DYNAMIC work station.

The SOLVER contains approximately 7000 FORTRAN statements. It is equipped with all equations given in Appendices A and B. The SOLVER first receives the static input data from subroutine STATIC and performs the job associated with DYNAMIC. The user has no direct contact with SOLVER. As far as he is concerned, it is a mathematical apparatus embodying the relevant laws of physics and providing solutions of variables that have been activated by the commands supplied by STATIC.

At the present, a complete listing of the nonproprietary program and the corresponding manual are available from the authors.

VALIDATIONS ON THE PROGRAM ACCURACY

Experiments

A rectangular climate room, 18.37 ft (5.6 m) long, 10.49 ft (3.2 m) wide and 9.84 ft (3.0 m) high, with heavy concrete ceiling and floor, was used for these experiments (Figure 4). A table was placed in the middle of the room near the window. The ventilating inlet units for the supply of cold air were located on the floor near the window. A heating unit with 3412.1 Btu/h (1000 W) heat capacity was on the rear wall at two-thirds the height of the room. A concentrated helium source was set on the right side of the table ($x/L=0.75$, $y/W=0.75$, $z/H=0.45$) for simulating tobacco smoke. The outlets were placed on the rear wall and can be placed either near the ceiling or near the floor. These experiments were used for the measurements of cooling load, enclosure surface temperatures, room air temperature fields, and helium concentration fields. The cooling load was updated from the supplying mass inflow and the temperature difference between the inlets and the outlets. The temperatures were measured by thermocouples, which were connected to a data-logging system. The data were then converted into printed results by a microcomputer. The helium distributions were measured by a helium meter.

The capacities of ACCURACY will be demonstrated by applying it to two different situations: cooling conditions and heating conditions.

Cooling Situation

In the first case, the two ventilating inlet units near the window, 3.28 ft x 0.03 ft (1.00 m x 0.01 m) each, are used. The inlet-mass-flow for the room is 2.65 ft^3/s (0.075 m^3/s) (a ventilation rate of five times per hour), which corresponds to an inlet airflow Reynolds number of 2400. The Reynolds number is based on the bulk velocity and the equivalent diameter of the inlet. The small inlet Reynolds number indicates that the influence on airflow from forced convection is on the same order as that from natural convection. A 3241 Btu/h (950 W) heat (a step function) is uniformly distributed on the venetian blinds. The initial temperature and the temperatures in the point near the floor are controlled at 69.8 F (21.0 °C). A concentrated helium source, 0.018 ft^3/s (5.0×10^{-4} m^3/s), in a step function form, is introduced into the room after the heat is supplied for 16 hours. The concentration source is maintained for one hour, and the outlets located on the rear wall near the ceiling are used for this case. The heating unit on the rear wall and the outlets on the rear wall near the floor are not used.

Cooling Load Computations. The cooling load is calculated by ACCURACY in four ways: (1) without considering air temperature gradients in the room, (2) by introducing constant air temperature gradients in the room, (3) by assuming that the room air temperature gradients are proportional to cooling load, and (4) by using an airflow pattern for computing the transient air temperature gradients in the room.

As illustrated by Figure 5, the cooling load showed a very significant discrepancy between the measured and the computed results using the above methods (1), (2), and (3) for the dynamic period. However, the agreement was very good when the fourth method was employed. The airflow distribution, as shown in Figure 6, was used for calculating room air temperature and concentration distributions.

When the heat was introduced on the venetian blinds, a vertical room air temperature gradient was generated, and this resulted in excessive heat being transferred into the ceiling. In the first method, where the normal one-air-point model was used, this thermal behavior cannot be simulated. The extra heat was considered to be a part of the cooling load, and it gave an exceedingly high value for the computed results. An exceedingly low computed cooling load was obtained from the second method, because the vertical temperature gradient used was obtained from the steady situation (i.e., t=16 hour), and this value was too high to be used for the initial hours (see Figure 7). The exceedingly large temperature gradient set at the beginning for the computation resulted in excessive heat being transferred into the ceiling. As a consequence, a smaller computed cooling load was obtained. Therefore, the temperature difference set in the cooling load program must be a transient one in order to obtain reliable results.

The third method is based on Nielson's (1982) assumption that the vertical room air temperature gradient is proportional to the air-conditioning load of the room. However, the relationship between the cooling load and the room air temperature gradient for this case was not a linear one. A constant proportional factor used for this case still gave a temperature gradient which was too big. It nevertheless still gave an exceedingly low value as shown in Figure 5, although the computed result by this method was better than that obtained from the second method. This method seems to have the same deviation as the first method, which is due to the heat supply on the venetian blinds being a step function. If real weather data are used, this method will give better results than the first one because the real weather data are a random function. The proportional factor can be either measured directly from the room or calculated once from an airflow program. In the fourth method, the transient temperature gradients were calculated from an airflow pattern (Figure 6). The predicted cooling load is good because the computed transient temperature gradients of the room agree with the measured one (Figure 7).

The thermal conditions and the airflow of the rooms above and below were controlled to be the same as those of the climate room. At steady state, the heat obtained from the floor was the same as that transferred into the ceiling in all four methods. As the result, the cooling loads computed were almost the same.

The computing time used by these four methods in a mainframe computer and a personal computer for one-month hourly simulation is given in Table 3. Clearly, the method using flow patterns is much more expensive. The predicted cooling loads are the same between grid numbers 6x11x17 and 9x18x27, because the average temperature differences predicted by the two grid numbers are the same, and in ACCURACY only the average temperature differences are employed for cooling load computations.

The storage required by ACCURACY for the computation with the grid number 6x11x17 is less than 340 kilobytes on a personal computer.

The Calculations of Room Air Temperature and Contamination Distributions. The room air temperature distribution in the middle section (y/W=0.5) at t=16 hours is shown in Figure 8. The agreement between the measured data and the calculated values by ACCURACY is good. This result is nearly the same as that obtained by calculating it directly from the airflow program PHOENICS. ACCURACY gives hourly room air temperature distributions and cooling load for a 24-hour period, and it only costs 7.25 seconds in the mainframe computer. However, when this is done using the 3D airflow program, it will entail 3.5 minutes CPU time in the mainframe computer for calculating the temperature distributions at each time step.

The computed and measured concentration field after the helium source is bled in for one hour (step function) is shown in Figure 9. The agreement between the computation and the measurement is fairly good. Because the room time constant for the helium concentration is very small, the transient concentration of helium in the middle point of the room has to be simulated with a time step of one minute. The comparison between the computed and the measured results for the transient response is shown in Figure 10. The discrepancy between these results is notable, and this is probably due to the influence of the helium density, which was not considered in the calculation.

Temperature Efficiency and Ventilation Efficiency. Although cooling loads calculated by the four methods are the same under the steady situation (Figure 5), they still have different physical meanings in terms of energy consumption. There are many ways to define temperature efficiency. In ACCURACY, temperature efficiency (η_T) is defined as

$$\eta_T = \frac{T_{out} - T_{in}}{T_{occu} - T_{in}} \qquad (6)$$

where

T_{out} = outlet air temperature

T_{in} = inlet air temperature

T_{occu} = air temperature in occupied zone.

The temperature efficiency at t=16 hour for this case is 1.57 (T_{out} = 75.7 F (24.3°C), T_{in} = 59.5 F (15.3°C), and T_{occu} = 69.8 F (21.0°C)). In other words, to remove 2798 Btu/h (820 W) cooling load, the required inlet air temperature has to be 59.5 F (15.3°C) if the inlet mass flow is kept as constant. If this were calculated by the normal one-air-point model in which the temperature efficiency is always assumed to be 1.0, the required inlet air temperature would have to be 53.6 F (12.0°C). The 5.9 F (3.3°C) temperature difference is very significant with respect to energy saving. The higher the inlet air temperature, the lower the amount of energy is used. The temperature gradient in the occupied zone is acceptable for comfort.

If a similar definition is applied for the ventilation efficiency (η_V), we have

$$\eta_V = \frac{C_{out} - C_{in}}{C_{occu} - C_{in}} \qquad (7)$$

The ventilation efficiency for the case at t=17 hours (after the helium source was bled in for one hour) is 5.87 (C_{out} = 8.8%, C_{in} = 0.0%, C_{occu} = 1.5%). When the outlets located on the rear wall were near the floor, recent measurements indicated that, not only is the ventilation efficiency poor (≤1.0), but also the concentration gradient in the room is large. The contamination remains under the ceiling and is very difficult to remove. Of course, this depends on the ventilation rate and airflow patterns. A commonly used air-conditioning system is one where the inlets are on the rear wall near the ceiling and the outlets are on the rear wall near the floor. This system generally has a very uniform room air temperature and contamination distributions that require a lower inlet air temperature or larger air supply in summer.

The computations of cooling load with sampling time intervals of 15 minutes and one hour gave nearly the same results. When the room air mixing is good and there is no temperature gradient, a normal one-air-point model gave good predictions.

Heating Situation

The second case concerned a mixture of natural convection with a cold window surface and forced convection with a hot air supply on the rear wall at two-thirds the height of the room (Figure 4) and the outlet at the same height. The ventilating units were closed. The outside air temperature of the window was 37.4 F (3.0°C), and initial room temperature was 57.9 F (14.4°C). The heater capacity was 3412 Btu/h (1000 W) (also a step function). No external ventilation was introduced for the system.

We have noted that there is no difference in energy predictions between the heating case and the cooling case. In order to get better computational results, the fourth method used above should be employed for this case. However, Figure 11 shows that the temperature difference between the air points near the ceiling and the floor for this case rises very quickly to its maximum value (about 15 minutes) in contrast to this very long time required to reach the maximum for the first case (Figure 7). In the room air temperature prediction, the error caused by the first 15-minute change of the temperature gradient can be ignored because it only causes 0.1°C difference. The air temperature computed for the middle point of the room by introducing constant temperature gradients gives a very good agreement with the measured one (Figure 12). The other calculation based on the normal one-air-point model is used for comparison in Figure 12. With such a heating system, there is always a very large vertical temperature gradient in the room, which makes it uneconomical. Especially when the floor and the ceiling of a room are constructed of heavy concrete, the normal one-air-point model does not give good results, since the heat obtained from the floor is not the same as that transferred into the ceiling. The constant temperature gradients should be recalculated from the corresponding airflow pattern, and this is used only once for the computation. The CPU time in the mainframe computer for the system is 2.88 seconds (Table 3) for one-month hourly predictions by introducing constant room temperature gradients. It takes seven minutes to do this computation in a personal computer.

FURTHER REMARKS

From the results given above, it can be concluded that when there is no air temperature gradient in a room, normal cooling-load computer programs are sufficient for obtaining accurate results. However, if the room air temperature gradients are large, normal cooling-load computer programs have to be improved with respect to the following points:

-- If the room air temperature gradients are not functions of time, the one-air-point model with constant temperature gradients will give good predictions. The constant temperature gradients should either be calculated from an airflow program once or directly measured from the room.

-- If the room air temperature gradients are transient, they can be approximated to be proportional to the room cooling load. However, if the gradients are not linear to the cooling load, the predicted results will deviate from the measured results. The best way is to calculate the room air temperature gradients from room airflow patterns, but this is very expensive.

Room air temperature and contamination distributions calculated from airflow patterns are much cheaper than those computed directly from a flow program. ACCURACY gives detailed information of room air temperature, contamination, and air distribution. When airflow patterns are required, it has to work in cooperation with an airflow computer program. We have developed a user-friendly airflow program, CHAMPION SGE, for personal computers, but it has some significant limitations because the program is a two-dimensional one. The three dimensional computer program PHOENICS is too expensive to run, and a complete listing is not available. Furthermore, an earlier study (van der Kooi and Chen 1986) has shown that it is very difficult to get good predictions by the three-dimensional airflow program under natural convection. Computations using ACCURACY with airflow patterns are expensive; it will not be economical to use this method for hour-by-hour air-conditioning load calculations. However, room response factors may be determined from ACCURACY under different kinds of airflow patterns. These room response factors can be employed for the calculations of annual energy requirement. The other possibility is to use a short reference year (van Paassen and Liem 1985).

When wall functions given in Appendix A are used for calculating convective heat exchange coefficients, the values seem to be 0.18-0.35 Btu/ft²·h·F (1-2 W/m²·°C) lower than measurements under mixed convections. A new type of wall function for convective heat exchange coefficients is under development in our laboratory. The convective heat exchange coefficients used here are obtained from the measurements.

CONCLUSIONS

ACCURACY is a new computer program for energy analysis, load calculation, the determination of indoor air distribution, and the computation of room air temperature and contamination distributions. It uses an airflow computer program for precalculating airflow patterns for typical situations.

Cooling load programs by using normal one-air-point model seem to be suitable for a room without temperature gradients. When the room temperature gradients are constant, this can be used directly in ACCURACY for cooling load computations. However, when the temperature gradients are transient functions, one much improved method is to assume that the room air temperature gradients are proportional to the cooling load and the best way to calculate the transient temperature gradients is from airflow patterns. ACCURACY considers the influence of room air temperature distributions on its cooling load or room air temperature predictions and also calculates contamination distribution. This improvement is necessary for the investigation of the energy saving and comfort of the indoor environment in different air supply systems.

For initial computations, approximate room temperature gradients can be introduced directly in the computer program. In this case, ACCURACY runs as fast as normal cooling load computer programs. The method, which uses a proportional factor for obtaining temperature gradients from the cooling load, is 2.3 to 2.5 times more expensive than the normal one-air-point model. The computations using room air currents, when the 6x11x17 grid point is used, appear to be 10.0 to 12.4 times more expensive than the normal one-air-point model.

ACCURACY can also be used for evaluating temperature efficiency and ventilation efficiency in order to select the most economical air terminal devices and locations. It also calculates indoor convective and radiative heat exchange coefficients. The coefficients will be reset in the room energy balance equations when they have a large variation.

NOMENCLATURE

A	= grid surface or enclosure surface area, ft² (m²)
$a_{subscript}$	= coefficient of the finite-domain equation, (-)
C	= concentration of contamination, ft³/ft³·air (m³/m³·air)
$C_{subscript}$	= convective term of the flow equations, lb/s (kg/s)
C_p	= constant-pressure specific heat, Btu/lb·F (J/kg·K)
C_1, C_2, C_3, C_D	= coefficients in the k-ε turbulent model, (-)
$D_{subscript}$	= diffusive term of the flow equations, lb/s (kg/s)
G	= generation due to turbulence or buoyancy, (-)
g	= gravitational acceleration, ft/s² (m/s²)
H	= specific enthalpy, Btu/lb (J/kg)
k	= kinetic energy of turbulence, Btu/lb (J/kg)
M	= matrix, (-)
N	= total enclosure surface number of the room, (-)
p	= pressure, psi (Pa)
q	= heat flux, Btu/h·ft² (W/m²)
q	= excitation matrix, (-)
Re	= airflow Reynolds number, (-)
S	= source term of general fluid property, (-)
T	= temperature, F (°C) or R (K)
T	= temperature matrix, (-)
t	= general time, s (s)
u	= velocity component in x-direction, ft/s (m/s)
V	= volume of the room or a flow cell, ft³ (m³)
V_i	= u, v, w; flow velocities in x, y and z direction respectively, ft/s (m/s)
V_i	= ventilation or infiltration through surface i, lb/s (kg/s)
v	= velocity component in y-direction, ft/s (m/s)

w	= velocity component in z-direction, ft/s (m/s)
x	= general coordinate, ft (m)
x,y,z	= coordinate, ft (m)
y	= first grid distance to the boundary, ft (m)

Symbols in Greek

α_{ic}	= convective heat exchange coefficient on an inside surface, Btu/h·ft²·F (W/m²·°C)
$\alpha_{r_{i,j}}$	= radiative heat exchange coefficient of inside surface i to surface j, Btu/h·ft²·F (W/m²·°C)
β	= volume coefficient of thermal expansion, 1/F (1/K)
Δ	= difference between values, (-)
δ	= time interval, s (s)
Γ	= diffusive coefficient, lb/ft·h (Pa·s)
ε	= dissipation rate of turbulence energy, Btu/lb·h (J/Kg·s)
η	= efficiency, (-)
μ	= dynamic viscosity, lb/ft·h (Pa·s)
ν	= fluid kinetic viscosity, ft²/h (m²/s)
ρ	= fluid density, lb³/ft³ (kg/m³)
σ	= Schmidt or Prandtl number, (-)
τ	= Reynold's stress, psi (Pa)
ϕ	= general fluid property, (-)

Subscripts

B	= buoyancy
c	= convection
eff	= effective coefficient
ex	= extraction
H	= specific enthalpy
i	= inside
i,j,k	= subscripts denoting coordinate directions
in	= inlet
k	= kinetic turbulence energy
o	= outlet
o	= reference point
occu	= occupied zone
P,N,S,E,W,H,L,T	= subscripts denoting directions
p,n,s,e,w,h,l,t	= subscripts denoting directions
l	= laminar
R	= room
r	= radiation
T	= temperature
t	= turbulent
V	= ventilation
w	= window or wall
ε	= dissipation rate of turbulence energy
ϕ	= general fluid property

Superscripts

$\rightarrow$	= vector
'	= previous time step

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APPENDIX A

Room Energy Balance Equations

Simultaneous room energy balance equations for predicting air temperature in the middle point of the room may be expressed in matrix form as shown below:

$$[M] \cdot [T] = [q] + [\alpha\ \Delta T] \qquad \text{(A-1)}$$

where

$$[M] = \begin{bmatrix} \alpha_{1c}+\sum_{k=1}^{N}\alpha_{r_{1,k}}, & -\alpha_{r_{1,2}}, & -\alpha_{r_{1,3}}, & \cdots, & -\alpha_{r_{1,N}}, & -\alpha_{1c} \\ -\alpha_{r_{2,1}}, & \alpha_{2c}+\sum_{k=1}^{N}\alpha_{r_{2,k}}, & -\alpha_{r_{2,3}}, & \cdots, & -\alpha_{r_{2,N}}, & -\alpha_{2c} \\ \cdots & \cdots & \cdots & & & \\ -\alpha_{r_{N,1}}, & -\alpha_{r_{N,2}}, & \cdots, & -\alpha_{r_{N,N-1}}, & \alpha_{Nc}+\sum_{k=1}^{N}\alpha_{r_{N,k}}, & -\alpha_{Nc} \\ \alpha_{1c}A_1, & \alpha_{2c}A_2, & \cdots, & \alpha_{N-1c}A_{N-1}, & \alpha_{Nc}A_N, & -\mathrm{term1} \end{bmatrix}$$

$$[T] = \begin{bmatrix} T_1 \\ T_2 \\ \cdot \\ \cdot \\ \cdot \\ T_N \\ T_R \end{bmatrix}; \qquad [q] = \begin{bmatrix} q_1 \\ q_2 \\ \cdot \\ \cdot \\ \cdot \\ q_N \\ \mathrm{term2} \end{bmatrix}; \qquad [\alpha\ \Delta T] = \begin{bmatrix} \alpha_{1c}\ \Delta T_{r1} \\ \alpha_{2c}\ \Delta T_{r2} \\ \cdot \\ \cdot \\ \cdot \\ \alpha_{Nc}\ \Delta T_{rN} \\ 0 \end{bmatrix}$$

α_{ic} = convective heat exchange coefficient in surface i, which can be calculated in ACCURACY by wall functions (see below) ($i=1, 2, \cdots, N$).

$\alpha_{r_{i,j}}$ = radiative heat exchange coefficient between surface i and j, which can be calculated in ACCURACY by considering multiple reflection ($i=1, 2, \cdots, N$; $j= 1, 2, \cdots, N$).

T_i = temperature of inside surface i ($i=1, 2, \cdots, N$).

T_R = air temperature in the middle point of the room.

q_i = incoming heat through inside surface i; for a window, it is updated using the window energy balance equations, and for the rest, it is calculated using Z-transfer function (Stepenson and Mitalas 1971) ($i=1, 2, \cdots, N$).

ΔT_{ri} = temperature difference between the air near inside surface i and middle point of the room ($i=1, 2, \cdots, N$).

$$\mathrm{term1} = \sum_{i=1}^{N}(\alpha_{ic}A_i) + \frac{\rho V_R C_p}{\Delta t} + \sum_{i=1}^{N} V_i$$

$$\mathrm{term2} = H_{ex} - H_{other} + \sum_{i=1}^{N}(\alpha_{ic}\Delta T_{ri}A_i) + \sum_{i=1}^{N}(V_i\Delta T_{ri} - V_i T_{wi}) - \frac{\rho V_R C_p}{\Delta t}\, T'_R$$

V_i = ventilation or infiltration mass flow through the surface i.

T_{wi} = air temperature of window gap or ventilation inlet.

T'_R = air temperature in the middle point of the room at previous time step.

The wall function method is the one that has been most widely used and is, indeed, still to be preferred for airflow near solid boundary walls (Launder and Spalding 1974). There are many kinds of wall functions available at present (Piece, McAllister, and Tennant 1982). The one used here was developed by van de Leur (1983) from the Couette-flow equations in the boundary layer (Patankar and Spalding 1970). The wall function for heat transfer through solid boundaries is

-- when the subboundary layer is laminar ($y+ \leq 11.5$)

$$q = \frac{(\tau_w/\rho)^{1/2}\rho\, C_p(T_p - T_w)}{\sigma_1\, y+} \tag{A-2}$$

-- when the subboundary layer is turbulent (y+ > 11.5)

$$q = \frac{(\tau_w/\rho)^{1/2}\rho\, C_p(T_p - T_w)}{\sigma_1 11.5 + (y^+ - 11.5)\sigma_t \frac{\mu_1}{\mu_t}} \tag{A-3}$$

<u>where</u>

$$y+ = C_D^{1/4}\rho k^{1/2} y/\mu_1 \tag{A-4}$$

APPENDIX B

The Finite Domain Form of Airflow Governing Equations

The "staggered grid" is used for Equations 3 and 5. Figure 1 shows the points at which u, v, and w are stored in the flow codes PHOENICS and CHAMPION SGE. Temperature and concentration are stored at point P. The cross-stream velocity u, v, and w are stored just at the points at which they are needed for the calculation of the convective contribution to the balance of energy and concentration.

Let ϕ stand for T and C; the finite-domain form of Equations 3 and 5 can be expressed as:

$$\phi_P = \frac{a_E\phi_E + a_W\phi_W + a_N\phi_N + a_S\phi_S + a_H\phi_H + a_L\phi_L + a_T\phi_-}{a_E + a_W + a_N + a_S + a_H + a_L + a_T} \tag{B-1}$$

The formulae for coefficients in Equation B-1 are given in Table 2 in which the upwind scheme is used. The S is an additional source term for special purposes. The air temperature near enclosure surface i, T_{ri}, is calculated from the T_P's, because enclosure surface i may connect with many flow grid points.

TABLE 1
Values of ϕ, Γ_ϕ, S_ϕ and $S_{Buoyancy}$ Terms

ϕ	Γ_ϕ	S_ϕ	$S_{Buoyancy}$
1	0	0 (continuity)	0
V_i	μ_{eff}	$-\partial p/\partial x_i + \partial(\mu_{eff}(\partial V_j/\partial x_i)\partial x_j$	$-\rho\beta g_i\theta$
H	μ_{eff}/σ_H	0	0
k	μ_{eff}/σ_k	$G-\rho\varepsilon$	G_B
ε	$\mu_{eff}/\sigma_\varepsilon$	$\varepsilon(C_1G-C_2\rho\varepsilon)/k$	$C_3\varepsilon G_B/k$
C	μ_{eff}/σ_C	0	0

$\mu_{eff} = \rho(\nu_1 + \nu_t)$

$\mu_t = C_D k^2/\varepsilon$

$\Theta = T - T_0$

$G = \mu_t(\partial V_i/\partial x_j + \partial V_j/\partial x_i)\partial V_i/\partial x_j$

$G_B = \rho\beta g_i \frac{\nu_t}{\sigma_H} \frac{\partial\theta}{\partial x_i}$

$C_1 = 1.44$, $C_2 = 1.92$, $C_D = 0.09$, $C_3 = 1.44$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $\sigma_H = 0.9$, $\sigma_C = 1.0$

TABLE 2
Formulae for Coefficients in Equation B1

Coeff.	Values
a_E	$\max(D_e, D_e - C_e)$
a_W	$\max(D_w, D_w + C_w)$
a_N	$\max(D_n, D_n - C_n)$
a_S	$\max(D_s, D_s + C_s)$
a_H	$\max(D_h, D_h - C_h)$
a_L	$\max(D_l, D_l + C_l)$
a_T	C_t

Cell	C's	D's
e	$\rho\ u_e\ A_e$	$[2/(1/\Gamma_P + 1/\Gamma_E)][A_e/(X_e - X_P)]$
w	$\rho\ u_w\ A_w$	$[2/(1/\Gamma_P + 1/\Gamma_W)][A_w/(X_P - X_w)]$
n	$\rho\ v_n\ A_n$	$[2/(1/\Gamma_P + 1/\Gamma_N)][A_n/(Y_n - Y_P)]$
s	$\rho\ v_s\ A_s$	$[2/(1/\Gamma_P + 1/\Gamma_S)][A_s/(Y_P - Y_s)]$
h	$\rho\ w_h\ A_h$	$[2/(1/\Gamma_P + 1/\Gamma_H)][A_h/(Z_h - Z_P)]$
l	$\rho\ w_l\ A_l$	$[2/(1/\Gamma_P + 1/\Gamma_L)][A_l/(Z_P - Z_l)]$
t	$\rho\ V_P/\delta_t$	-

TABLE 3
Computing Time

Cases	Methods	Mainframe (second)	Personal computer (minute)
Cooling	(1) Without gradient	2.46	6
	(2) With constant gradients	2.46	6
	(3) Proportional factor	6.20	14
	(4) With airflow pattern grid number 6x11x17	30.43	60
	(4) With airflow pattern grid number 9x18x27	129.97	-
Heating	(1) Without gradient	2.88	7
	(2) With constant gradients	2.88	7

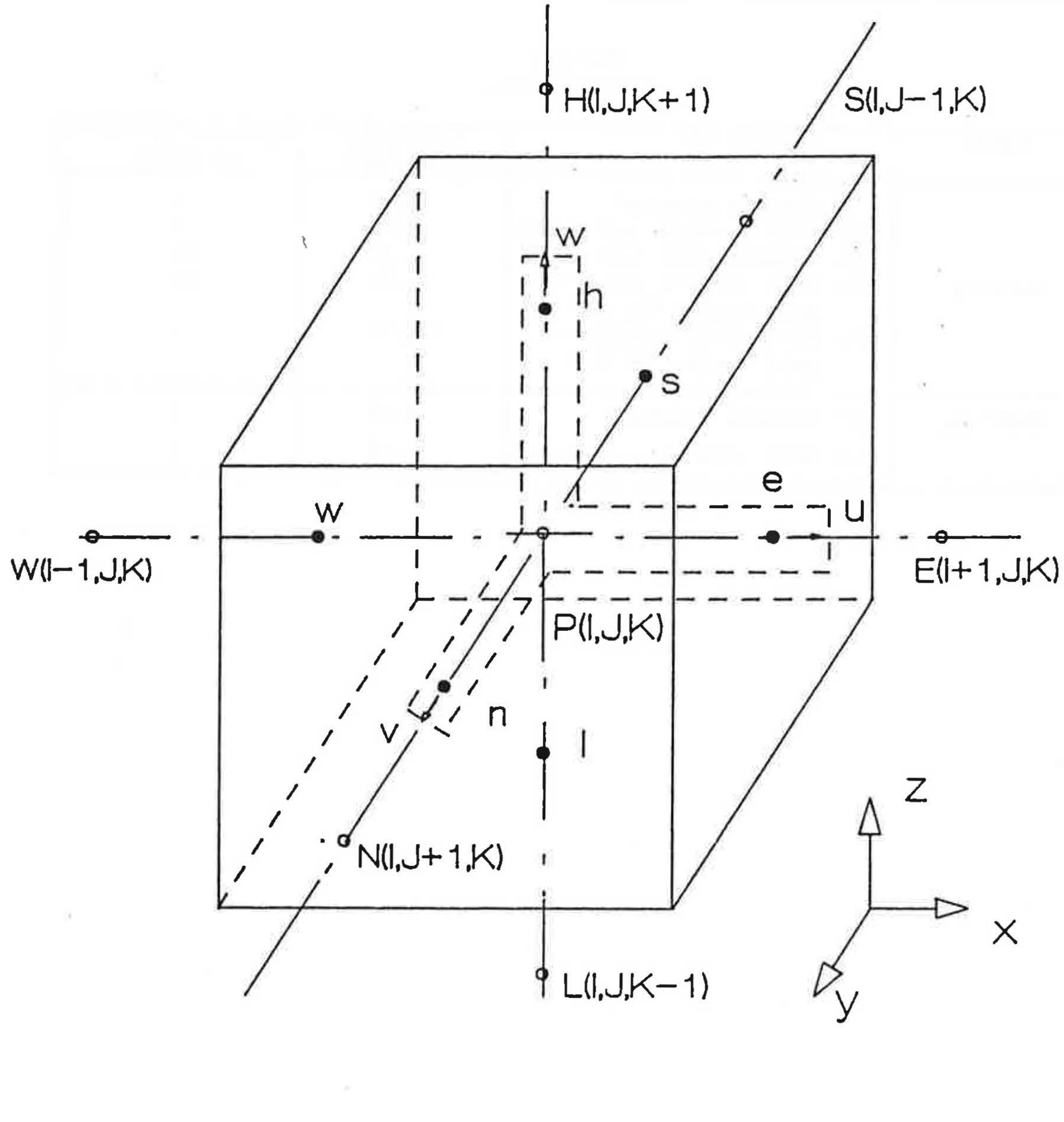

Figure 1 Grid information

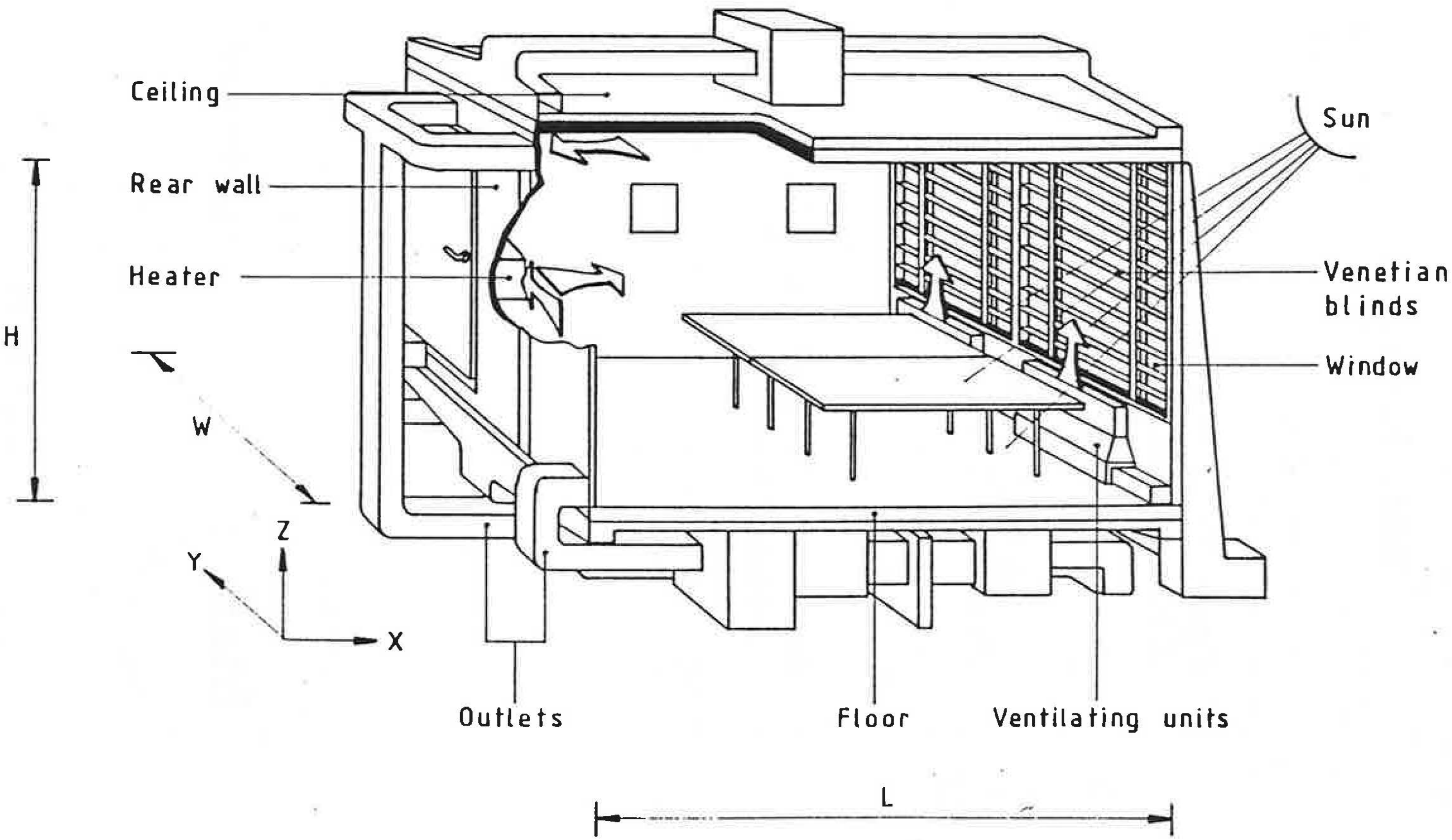

Figure 4 The climate room used for measurements

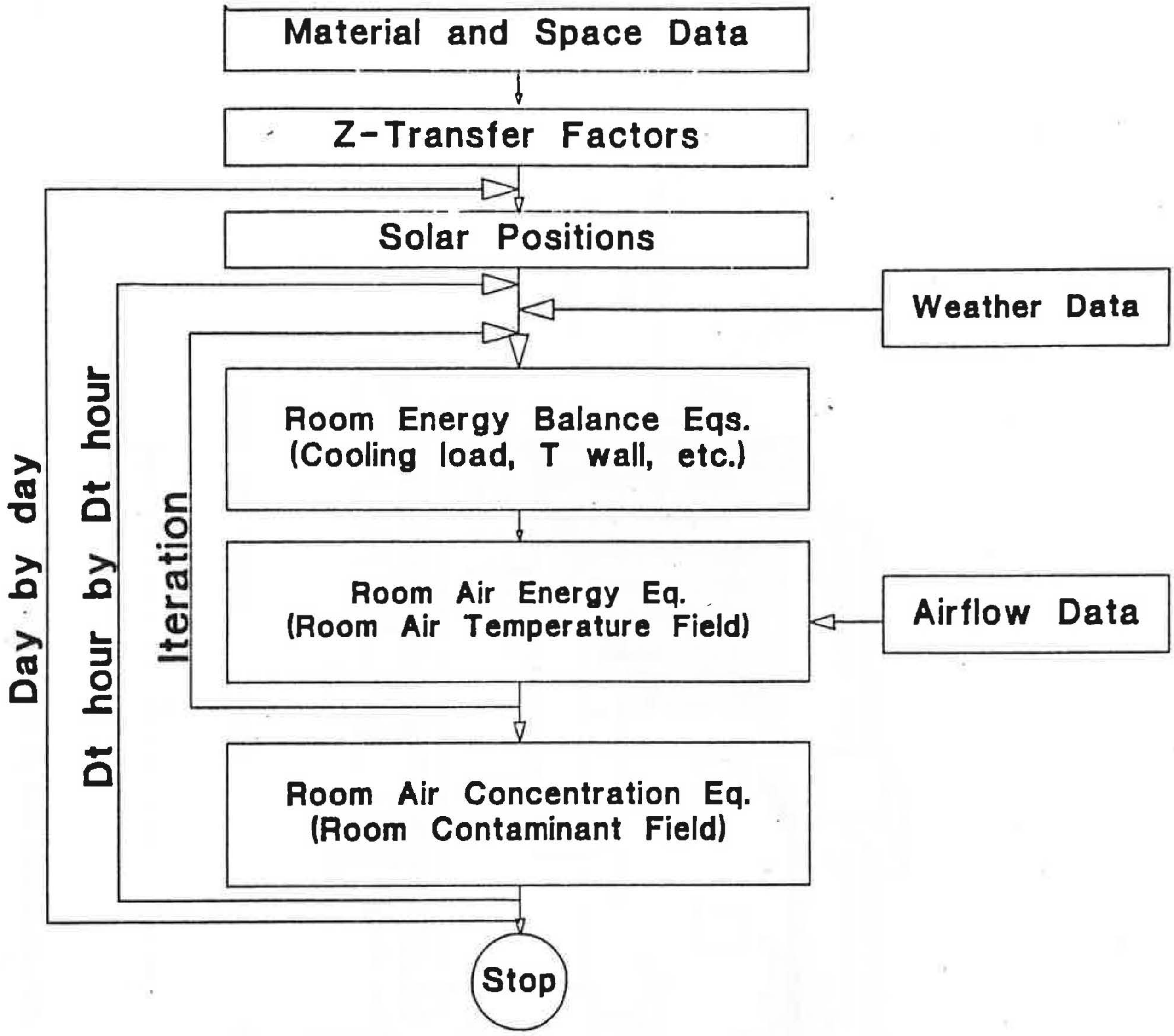

Figure 2 The diagram for ACCURACY

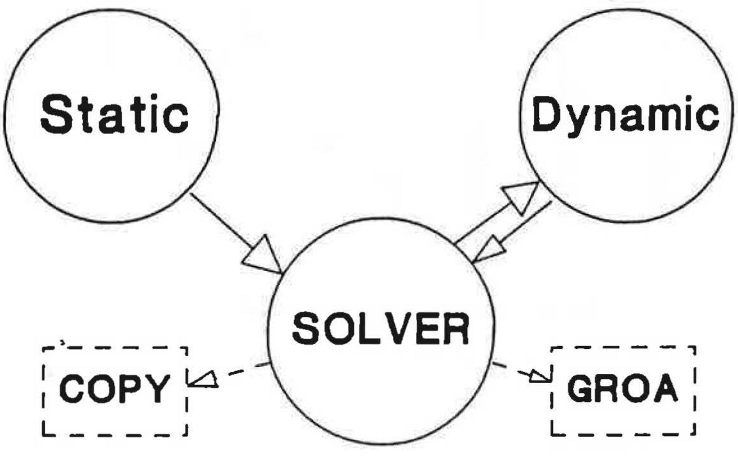

Figure 3 The structure for ACCURACY. COPY stores the numerical results as it appears in screen and GROA is the Graphical Representation Of Accuracy

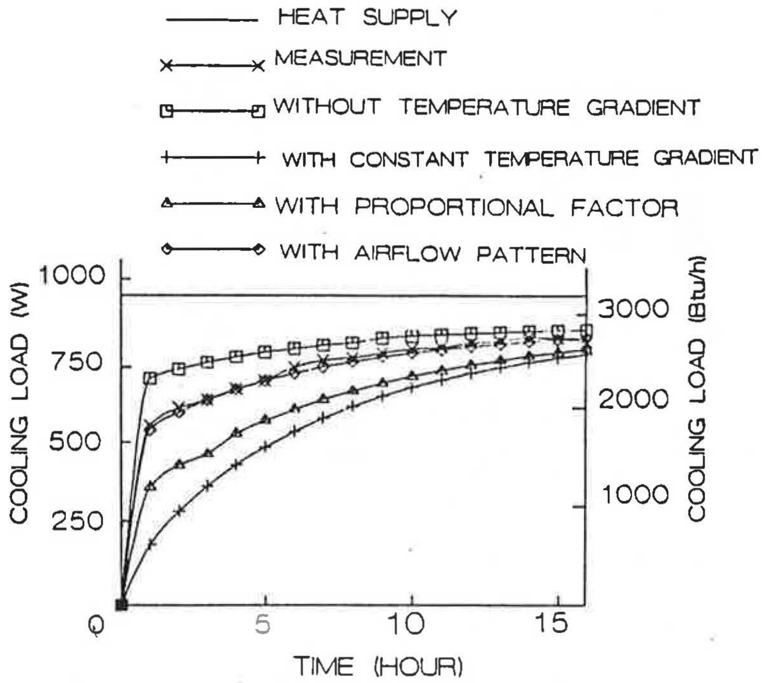

Figure 5 Cooling load measured and computed by ACCURACY

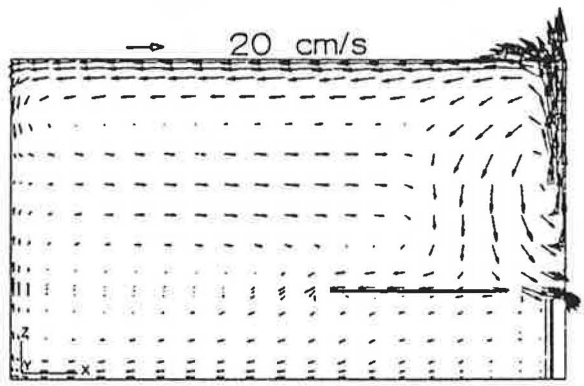

Figure 6 Airflow field of the room in section y/W=0.5 computed by PHOENICS

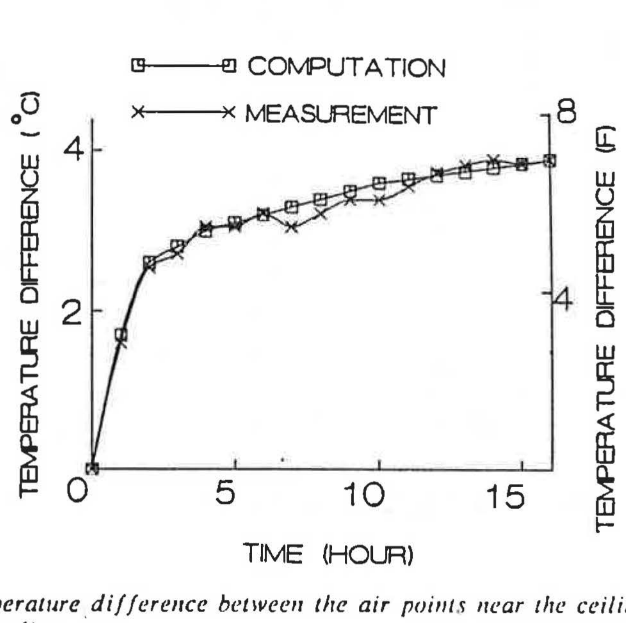

Figure 7 Temperature difference between the air points near the ceiling and the floor in the cooling case

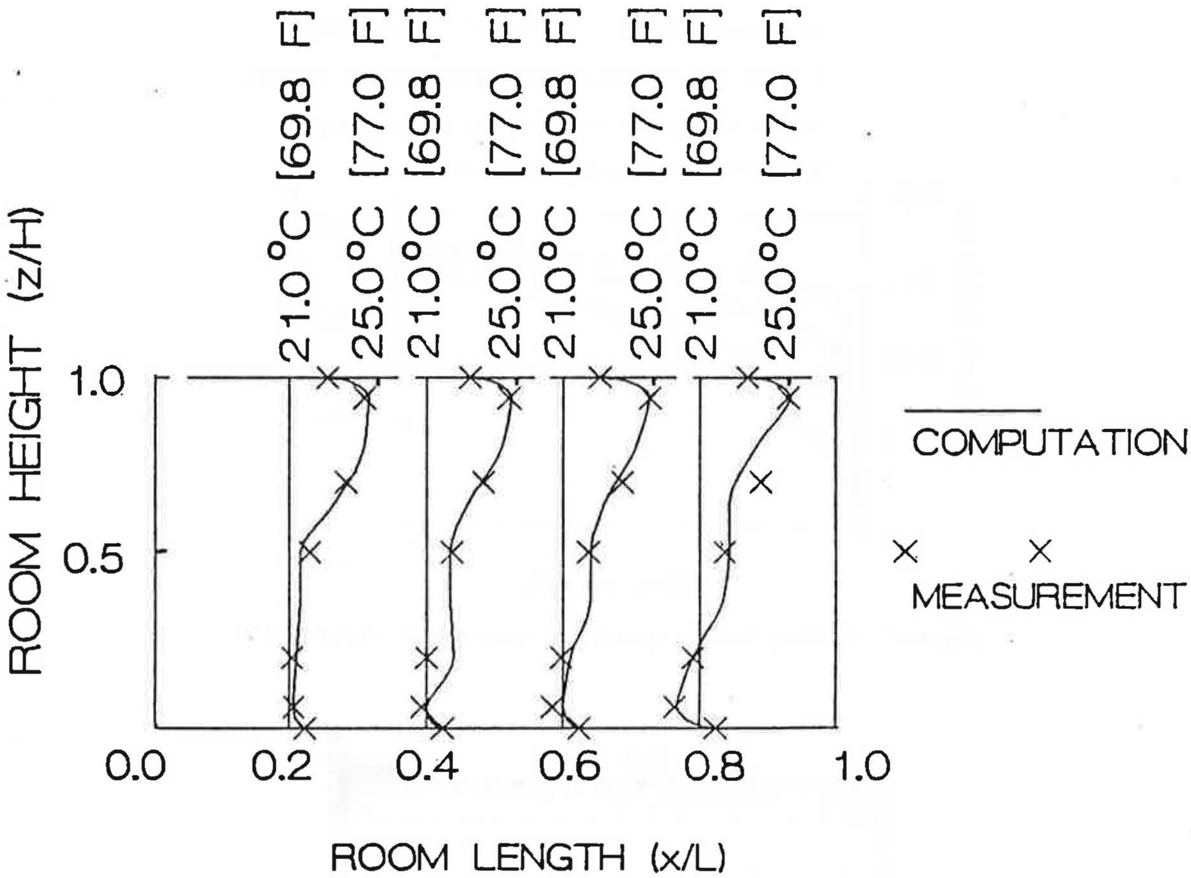

Figure 8 Temperature distribution of the room air in section y/W=0.5

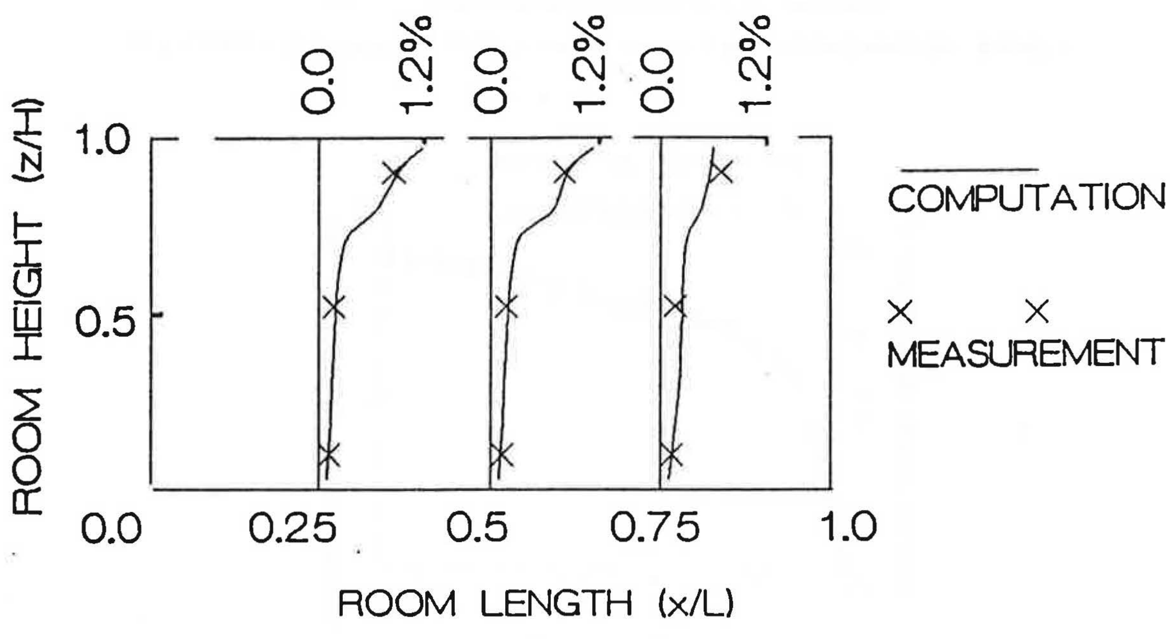

Figure 9 Contamination distribution of the room air in section y/W=0.5

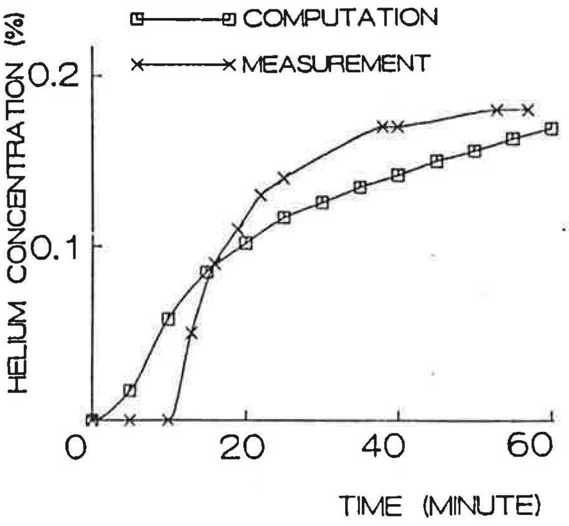

Figure 10 Contamination concentration in the middle point of the room

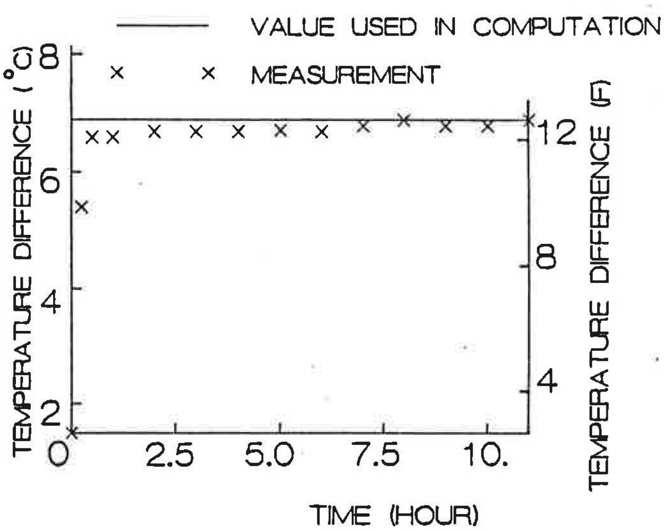

Figure 11 Temperature difference between the air points near the ceiling and the floor in the heating case

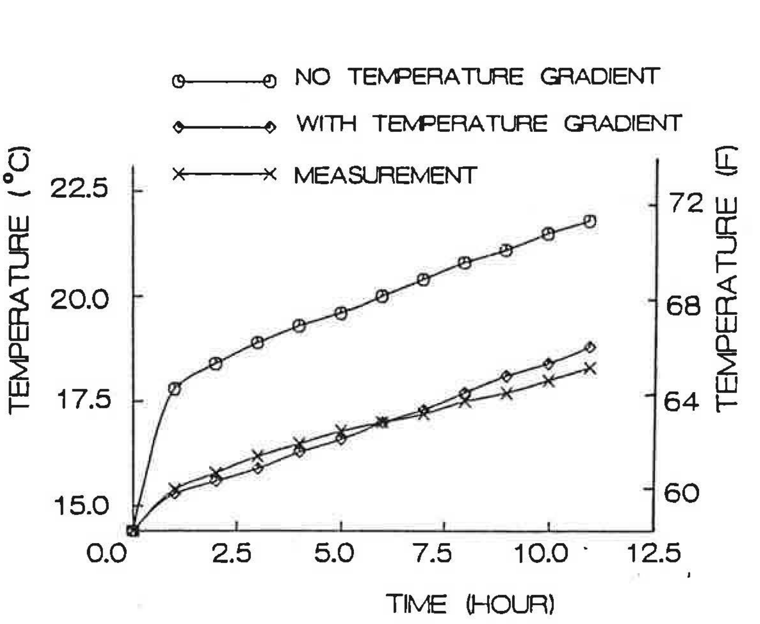

Figure 12 Computed and measured air temperature in the middle point of the room