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Transient Heat Transfer through the Enclosures of a Room with Mixed Convection

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INTRODUCTION

The insulation of buildings has been improved since the 1970's in order to reduce heat loss in winter, heat gain in summer and infiltration of outdoor air. As a consequence, the heat extracted from, or supplied to a room for maintaining a comfortable air temperature is smaller. From the view point of energy saving, the air supply has been decreased.

Such a reduction of air supply makes the indoor air flow change from forced convection into mixed convection and sometimes generates a non-uniform distribution of air temperature. The temperature in the upper part of the room will then be higher than that in the lower part. Does the indoor air temperature distribution have a large influence in the transient heat transfer through the enclosures of the room and consequently on the energy consumption of the air-conditioning system of the room? In order to estimate the influence, the temperature distribution of indoor air and the transient heat transfer through the enclosures must be predicted simultaneously because they are inter-related.

During recent years, the predictions of indoor air flow distributions have made a considerable progress, but most of the research interest was aimed at studying the influence of indoor air flow on comfort. On the other hand, the simulations of transient heat transfer through the enclosures of a room were based on the assumption of a uniform distribution of indoor air temperature, and the influence of the indoor air temperature distributions on the transient heat transfer and on the building energy consumption could therefore not be calculated. Hence, this paper will present a methodology for studying the problem.

MODEL FOR PREDICTING INDOOR AIR FLOW

The study of air flow in rooms using numerical calculation techniques has continued for nearly twenty years and has achieved remarkable successes. The range of air flow simulations, which originally comprised laminar, two-dimensional, steady and isothermal situations, has been enlarged to include turbulent, three-dimensional, transient and buoyancy-affected flows. The k-ε model of turbulence[1] is still the most popular model for practical flow applications in a room. The set of model equations is suitable for high Reynolds number flow. For wall flow, where local Reynolds numbers are considerably low, the equations are normally used in conjunction with empirical wall function formulas. The success of this method depends on the "universality" of the turbulent structure near the wall. When disagreements are found between measurements and predictions, it is difficult to judge whether the weakness of the method lies in the basic model equations or in the wall function formulas. Furthermore, the air flow in the centre of a room may not be turbulence so that the high Reynolds number model is invalid. Therefore, it is better to apply an appropriate low Reynolds number model of turbulence in the computation of indoor air flow patterns. Previous studies[2,3] have concluded that the model of Lam and Bremhorst[4] is one of the best low Reynolds number k-ε model. This model is therefore selected in the present research for predicting air flows in a room.

In the model of Lam and Bremhorst, the transport equations of the kinetic energy (k) and the dissipation rate of turbulence energy (ε) are determined from:

$$\frac{Dk}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_k} + \nu_l\right)\frac{\partial k}{\partial x_j}\right] + \nu_t\left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right)\frac{\partial V_i}{\partial x_j} - \varepsilon \tag{1}$$

$$\frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_\varepsilon} + \nu_l\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1 f_1 \nu_t \frac{\varepsilon}{k}\left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right)\frac{\partial V_i}{\partial x_j} - C_2 f_2 \frac{\varepsilon^2}{k} \tag{2}$$

The time-averaged flow field can be calculated through the eddy viscosity given by:

$$\nu_t = C_\mu f_\mu k^2/\varepsilon \tag{3}$$

where $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $C_1 = 1.44$, $C_2 = 1.92$ and $C_\mu = 0.09$. The functions f_1, f_2 and f_μ are given by the following equations:

$$f_\mu = (1 - e^{-A_\mu R_k})^2\left[1 + \frac{A_t}{R_t}\right] \tag{4}$$

$$f_1 = 1 + \left[\frac{A_{C1}}{f_\mu}\right]^3 \tag{5}$$

$$f_2 = 1 - e^{R_t^2} \tag{6}$$

where the turbulence model constants are $A_\mu = 0.0165$, $A_t = 20.5$ and $A_{C1} = 0.05$, and R_k and R_t are turbulence Reynolds numbers.

In addition to the buoyancy term in the momentum equation, the buoyancy production terms[5] in the k and ε equations should be included[2]. The buoyancy production term for the k equation is:

$$S_k = \beta\frac{\nu_t}{\sigma_H}\frac{\partial(T - T_o)}{\partial x_i} g_i \tag{7}$$

and the term for the ε equation is:

$$S_\varepsilon = C_3 \frac{\varepsilon}{k} S_k \tag{8}$$

where $C_3 = 1.44$. S_k and S_ε are additional source terms on the right side of eqns. 1 and 2.

MODEL FOR STUDYING THE INFLUENCE OF NON-UNIFORM INDOOR AIR TEMPERATURE DISTRIBUTIONS ON BUILDING TRANSIENT HEAT TRANSFER

With the above air flow model, the air temperature distributions and convective heat transfer coefficients can be computed. They can be used in an improved cooling load program for the calculation of the transient heat transfer through the enclosures with non-uniform distributions of indoor air temperature. Cooling load is the heat extracted from a room in cooling conditions to maintain a constant air temperature in the controlled point of the room. For heat transfer through the enclosures of a room, the air temperatures near the enclosure surface are important. These temperatures can be obtained from the field values of air temperature distributions. Since the outputs of the cooling load program, such as enclosure surface temperatures, are the boundary conditions for the air flow simulations. Iteration between an air flow program and the cooling load program is necessary for adjusting the thermal boundary conditions. This method can be extremely expensive if it is applied for an hour-by-hour computation for a long period (for instance, one reference year).

As an alternative, two approximated methods will be introduced for reducing the computing cost. The first one is to compute the air temperature distributions from the air flow patterns which are pre-calculated by an air flow program and the second is to establish room air temperature difference functions. The two methods will be discussed in the following subsections.

Method Using Air Flow Patterns

We have found[7] that, when the thermal boundary conditions vary in a small range, the air flow pattern is relatively stable while the temperature distribution may have a significant change. Therefore, a number of specific air flow patterns can be pre-calculated by an air flow program and the air temperature distributions are then re-computed from the corresponding air flow pattern with current thermal boundary conditions. The temperature distributions are almost the same as those computed directly by the air flow program but less computing time is required. The "specific" indicates that the boundary conditions used in the flow computations should be realistic and with reasonable combination of heat gain or loss and air supply such that most indoor air flow situations are included. With this method, the hour-by-hour flow information is also available and the indoor air quality can also be assessed quantitatively[7] by the cooling load program. This method needs a large memory and the computing costs are still too high since indoor air temperature distributions are calculated in each time step. However, it can be preformed in a mainframe.

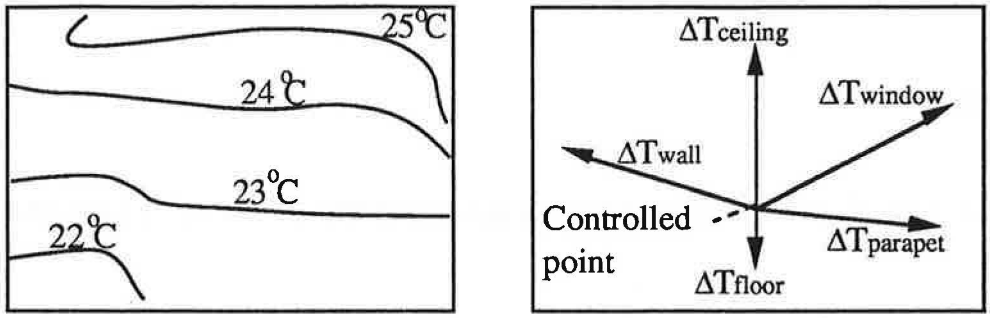

FIGURE 1. ΔT are calculated from air temperature distributions as the functions of Q and Vent.

Method with Air Temperature Difference Functions

Despite the fact that room air temperature distributions can be calculated from air flow patterns as discussed above, it is too expensive to be used for an hour-by-hour cooling load calculation for a long period. Detailed hourly indoor air flow information is not always required for annual energy analysis. In the calculation of the cooling load, the field values of an air temperature distribution can also be replaced by the air temperature at the controlled point and the air temperature differences between the controlled point and the air near the enclosure surfaces (ΔT) as shown in Fig. 1.

The ΔT can be determined through a number of temperature distributions, which are pre-calculated by an air flow program, under specific thermal conditions, such as under different kinds of room ventilation rates (Vent) and space loads (Q):

$$\Delta T = f(Q, Vent) \qquad (9)$$

The ventilation rates are related to the control strategies and comfort requirement, etc. The space loads are the results of the heat gain or loss through windows and walls and the heat gain from occupants, lighting and appliances, etc. 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 air flow 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 air flow patterns. Since no additional computation for the flow or air temperature distribution is required in the cooling load program, the computing cost is small.

OUTLINE OF THE COMPUTER PROGRAMS

The program PHOENICS[6] has been extended to include the formulations of the turbulence model described above. The computational method involves the solution, in finite-volume form, of three-dimensional conservation equations. The calculations were performed using an up-wind differencing scheme and staggered grids.

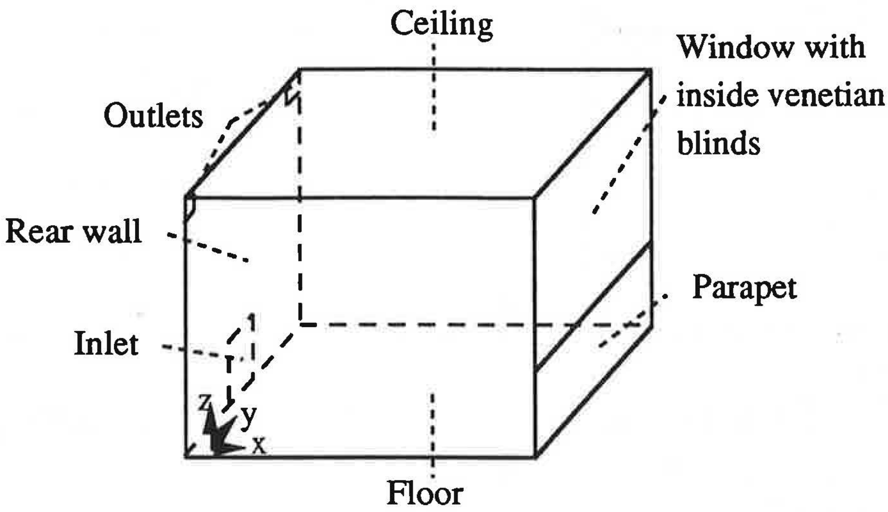

Figure 2. The sketch of the test room.

The cooling load program ACCURACY[7,9] has been developed with the transient heat transfer model discussed above. It is based on the room energy balance method and uses the Z-transfer function[8] and window energy balance equations for the transient heat transfer through enclosures. Since the inner enclosure surfaces are assumed to be grey bodies, the multiple reflections among the surfaces can also be studied in the program.

AIR FLOW AND TRANSIENT HEAT TRANSFER OF A ROOM

The model will be demonstrated by applying it to a room which is 5.6 m long, 3.0 m wide and 3.2 m high as shown in Fig. 2. The ceiling and floor of the room were constructed with heavy concrete and the sum of the heat transfer through the side walls and rear wall was controlled to be zero. The spaces above and below the room were designed to be the same thermal conditions as those of the room. An displacement air supply unit with the dimension of 0.68 m in height and 0.50 m in width was on the floor near the rear wall. Two outlets were in the rear wall near the ceiling. A table, 1.75 m long, 1.45 m wide and 0.85 m high, was placed near the window. The inlet-mass-flow was 0.126 kg/s (a ventilation rate of 7 ach). A 950 W of heat (a step function) was uniformly distributed on the venetian blinds of the window for the simulation of a summer cooling situation. The initial temperature and the temperature in the centre of the occupied zone (the height is 0.9 m) were controlled at 23.0^{0}C. A concentrated helium source, 0.5% of the air supply, was supplied into the room at point x=3.7 m, y=0.7 m and z=1.15 m to simulate a smoking person. The Reynolds number (Re) is 1.5x10^{4} which is based on the bulk velocity and the equivalent diameter of the inlet. The Rayleigh number (Ra) based on the window height is 1.6x10^{10}. This is a mixed convection case with strong buoyancy since $Re^2/Ra = 0.014$.

Fig.3 shows the air velocity, temperature, helium concentration and turbulence energy distributions obtained from the computations and the measurements. This air flow distribution is the one under steady state (after the heat was introduced in the venetian blinds for 16 hours). The agreement between the computations and the measurements is rather good although there are some discrepancies. The

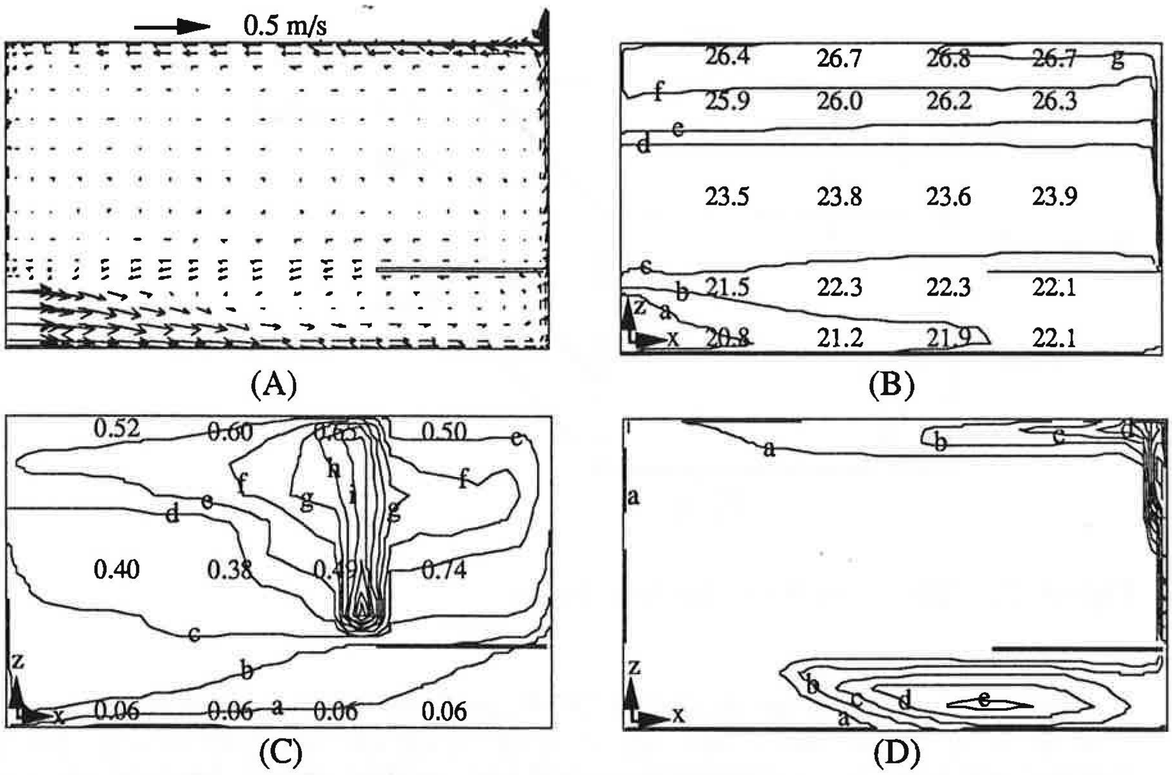

FIGURE 3. The computed and measured air flow distribution of the test room. (A). Computed air velocity distribution in section y=1.5 m; (B). computed (contours) and measured (numerical values) temperature distribution in section y=1.5 m (°C)#; (C). computed (contours) and measured (numerical values) helium concentration distribution in section y=0.7 m (%)*; (D). computed turbulence energy distribution in section y=1.5 m (10^{-3}J/kg)+.
Contour key for B: a-21, b-22, c-23, d-24, e-25, f-26, g-27.
* Contour key for C: a-0.1, b-0.2, c-0.3, d-0.4, e-0.5, f-0.6, g-0.7, h-0.8.
+ Contour key for D: a-0.5, b-1.0, c-1.5, d-2.0, e-2.5.

difference in most cases occurs where the variation in gradient is large. This means that a slight difference in measuring location can cause a significant error. The computed results with the low Reynolds number k-ε model are much better than those with the high Reynolds number model[1] as detailed in the thesis[9]. There is a vertical temperature gradient and the air temperature in the outlets is higher than than in the centre of the occupied zone. Besides, this ventilation system yields a very clean area in the zone of the occupation because the overall helium concentration in the zone is less than 0.5%. From the distribution of the turbulence energy, it is clear that, in most part of the room, the flow is laminar. Therefore, it is necessary to apply a low Reynolds number model.

The cooling load was calculated by the program ACCURACY. As illustrated by Fig.4, the agreement is very good between the experimental data and the computed results by the two methods presented above. Very big discrepancy has been found if the computation is carried out with the assumption of a uniform distribution of indoor air temperature.

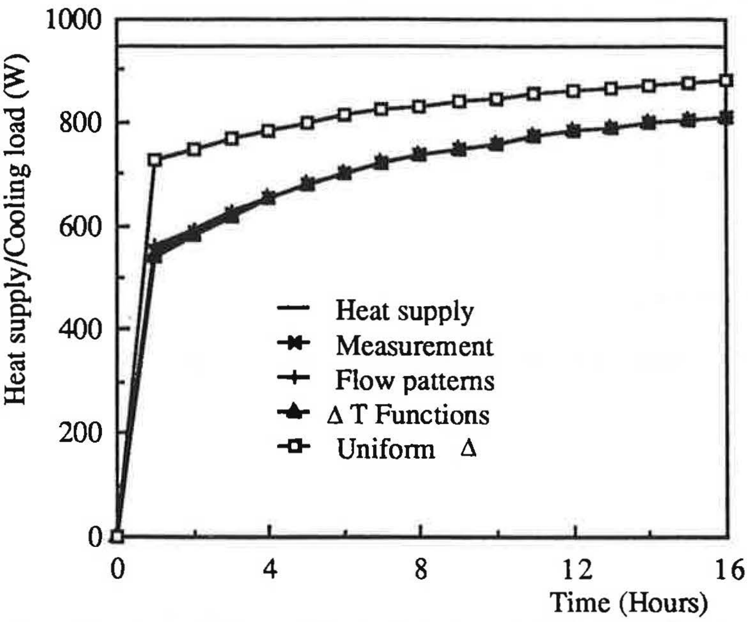

FIGURE 4. The cooling load measured and computed by different methods.

When the venetian blinds were heated, a vertical room air temperature difference was generated as shown in Fig. 5. Because the air temperature in the centre of the occupied zone (at point x=2.8 m, y=1.5 m and z=0.9 m) was controlled to be a constant, the air temperature difference between the air near the ceiling and the controlled point was much bigger than that between the air near the floor and the controlled point. This resulted in more heat being transferred into the ceiling than that being obtained from the floor at the initial hours. In the method with the assumption of a uniform distribution of indoor air temperature, this thermal behaviour cannot be simulated. The extra heat was considered to be a part of the cooling load, and it gave a too high value for the computed results in the initial hours. Hence, the temperature distribution of the room must be used in the cooling load program in order to obtain reliable results.

In the method which re-calculated the transient temperature distributions from the air flow patterns, the air flow patterns used are those with 300 W, 600 W and 900 W convective heat gains from the room. It is necessary to pre-computed the air flow patterns under different convective heat gains because the heat introduced in the window did not totally change into convective heat at the initial hours. A part of it was first absorbed by the other enclosures and then released into the room air with a time delay.

The method with the temperature difference functions determines the functions in eqn. 9 by curve fitting from the computational temperature distributions. Since the air supply is a constant in this case, the functions in eqn. 9 can be written as:

$$\Delta T_{ceiling} = 0.036 + 6.99\times10^{-3}Q - 2.72\times10^{-6}Q^2$$
$$\Delta T_{floor} = 0.026 - 1.83\times10^{-3}Q + 5.55\times10^{-7}Q^2$$
$$\Delta T_{window} = 0.036 + 6.99\times10^{-3}Q - 2.72\times10^{-6}Q^2$$
$$\Delta T_{walls} = 0.047 + 2.10\times10^{-3}Q - 1.07\times10^{-6}Q^2$$

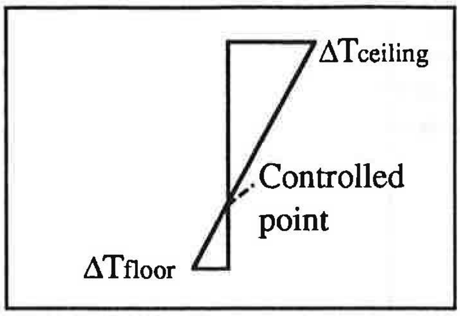

FIGURE 5. The vertical temperature gradient in the room.

$\Delta T_{parapet} = 0$

where Q is cooling load.

Since the method using air flow patterns and the method with the temperature difference functions yield the same air temperature distributions in this case, the computed cooling loads are the same.

At steady state, the cooling load is the same as the heat supply because all the heat supplied has to be removed. However, the air temperature extracted from the outlets is higher than that with the assumption of a uniform distribution of air temperature. As a result, the amount of the air supply can be smaller or the temperature of the air supply can be higher. This has a very large influence in the energy consumption of the air conditioning system. In order to study the influence, the annual energy consumption of the ventilator, chiller and boiler of the system has been estimated with the displacement ventilation system and with variable-air-volume air handling processes[9]. The result indicates that the energy consumption of the chiller and ventilator by the present method is 26% smaller than that by the method with the assumption of a uniform air temperature distribution. The energy requirement by the boiler is nearly the same. This also implies that energy saving with the air displacement system is very significant because many ordinary air supply systems present a uniform distribution of indoor air temperature. More detailed results have been reported in the thesis[9].

CONCLUSIONS

The air flow in a room with a displacement ventilation system is mixed convection. As the flow is turbulent but with relative low Reynolds numbers, a low Reynolds number k-ε model is used for the air flow simulation. The computed air flow is in rather good agreement with experimental data. The air flow program can also provide the distribution of indoor contaminant concentration. The ventilation efficiency of the room can therefore be calculated quantitatively.

For studying the influence of the non-uniform distributions of indoor air temperature on the transient heat transfer through building enclosures, a direct combination of an air flow program and a cooling load program is not possible due to the limited capacity of existing computers. Hence, two simplified methods are introduced in the present study. The first one calculates the temperature distributions from the air flow patterns which are pre-computed by an air flow program and the second one determines indoor air temperature distributions as the functions of cooling loads and ventilation rates. The first method is expensive but with detailed indoor flow information.

The two approximated methods have been employed for the cooling load computation in a room with a step heat supply in a window. It has been found that the computed cooling loads by the two methods are in very good agreement with the measurements. The results with the assumption of a uniform air temperature distribution deviate from the measurements.

The energy consumption has been studied for a room with a displacement ventilation system and with variable-air-volume air handling processes. The results show that the estimated energy consumption of the chiller and ventilator by the present methods is 26% smaller than that by the method with the assumption of a uniform air temperature distribution.

NOMENCLATURE

A_{C1}, A_t, A_μ	coefficients in the k-ε turbulence model (-)
C_p	constant-pressure specific heat (J/kg K)
C_1, C_2, C_3, C_μ	coefficients in the k-ε turbulence model (-)
f_1, f_2, f_μ	functions in the k-ε turbulence model (-)
k	kinetic energy of turbulence (J/kg)
Q	space load (W)
R_k	turbulence Reynolds number $k^{1/2}y/\nu_l$
R_t	turbulence Reynolds number $k^2/(\nu_l\varepsilon)$
T	temperature (K)
Vent	room ventilation rate (air changes/hour)
V_i	general velocity (m/s)
S_k, S_ε	buoyancy production terms in the k-ε model
t	general time (s)
x_i	general coordinate (m)
β	gas expansion coefficient (1/K)
ΔT	air temperature difference between the controlled point and near wall air (K)
ε	dissipation rate of turbulence energy (J/kg s)
ν_l	laminar kinetic viscosity (m^2/s)
ν_t	turbulence viscosity (m^2/s)
ρ	air density (kg/m^3)
σ_H, σ_k, σ_ε	Prandtl number (-)

REFERENCES

1 Launder, B. E., and Spalding, D. B., The Numerical Computation of Turbulent Flows, *Computer Methods in Applied Mechanics and Engineering*. vol. 3, pp. 269-289, 1974.

2. Chen, Q., Moser, A., and Huber, A., Prediction of Buoyant, Turbulent Flow by a Low Reynolds Number k-ε Model, Unpublished paper.

3. Patel, V. C., Rodi, W., and Scheuerer, G., Turbulence Models for Near-Wall and Low Reynolds Number Flows: a Review, *AIAA Journal*, vol. 23, no. 9, pp. 1308-1319, 1985.

4. Lam, C. K. G., and Bremhorst, K., A Modified Form of the k-ε Model for Predicting Wall Turbulence, *Journal of Fluids Engineering,* ASME Transactions, vol. 103, pp. 456-460, 1981.

5. Nielsen, P. V., Restivo, A., and Whitelaw, J. H., Buoyancy-Affected Flows in Ventilated Rooms, *Numerical Heat Transfer*, vol. 2, pp. 115-127, 1979.

6. Rosten, H. I., and Spalding, D. B., The PHOENICE Reference Manual, Version 1.4, Report TR200, CHAM Ltd., London, 1987.

7. Chen, Q., and Van der Kooi, J., ACCURACY - a Computer Program for Combined Problems of Energy Analysis, Indoor Airflow and Air Quality, *ASHRAE Transactions*, vol. 94, part 2, 1988.

8. Stephenson, D. G., and Mitalas, G. P., Calculation of Heat Conduction Transfer Functions for Multilayer Slabs, *ASHRAE Transactions*, vol. 77, part 2, pp.117-126, 1971.

9. Chen, Q., *Indoor Airflow, Air Quality and Energy Consumption of Buildings*, Ph.D. Thesis, Delft University of Technology, 1988.