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# A Methodology for Indoor Airflow Computations and Energy Analysis for a Displacement Ventilation System

CHEN QINGYAN

*Energy Systems Laboratory, Swiss Federal Institute of Technology, ETH-Zentrum, CH-8092 Zurich (Switzerland)*

JAN VAN DER KOOI

*Laboratory for Refrigeration and Indoor Climate Technology, Delft University of Technology, Mekelweg 2, 2628 CD Delft (The Netherlands)*

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ABSTRACT

*This article presents a methodology for the computation of indoor airflow, air quality, space load and energy consumption of a room with a displacement ventilation system. Since airflow and transient heat transfer in the room are interrelated, the indoor airflow and the space load of the room must be predicted simultaneously. In order to reduce the computing costs, a simplified method has been introduced for the predictions.*

*According to the present state of the art, the k–ε turbulence model is suggested for the indoor airflow computations within a room. The air temperature distributions of the room are used in a cooling load program for space load calculations. This is very important for a room with a displacement ventilation system. An optimized algorithm is applied for the estimation of the energy consumption of the room.*

*Finally, an application example is presented. The results indicate that, in a room with the displacement system, the indoor air quality is much better than with a well-mixed system, and the energy savings are significant.*

Keywords: displacement ventilation, air distribution, indoor air quality, cooling load, energy conservation, computation.

INTRODUCTION

The insulation of buildings has been improved in order to reduce heat loss in winter, heat gain in summer and infiltration of outdoor air since the oil embargo of 1973. As a consequence, the heat extracted from, or supplied to a room for maintaining a comfortable air temperature is smaller. Because the heat extracted or heat supplied is related to the air supply and the temperature difference between the air inlet and the outlet of a room, the amount of air supplied or the air temperature difference can be reduced. From the viewpoint of energy saving, it is more economical to decrease the air supply and this became the trend in design after the 1970s.

Such a reduction of air supply causes an increase of the concentration of indoor pollutants and sometimes generates a non-uniform distribution of air temperature. In order to remove the contaminants more efficiently, a displacement ventilation system as shown in Fig. 1 has been developed which is widely used in Scandinavian countries [1]. In the displacement ventilation system, the 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 0.5 m/s) and with an air temperature at least 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 because of gravity, resulting in a vertical temperature stratification with the temperature in the upper part of the room 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 the well-mixed situation which exists in usual air supply systems. This means that, for displacement ventilation,

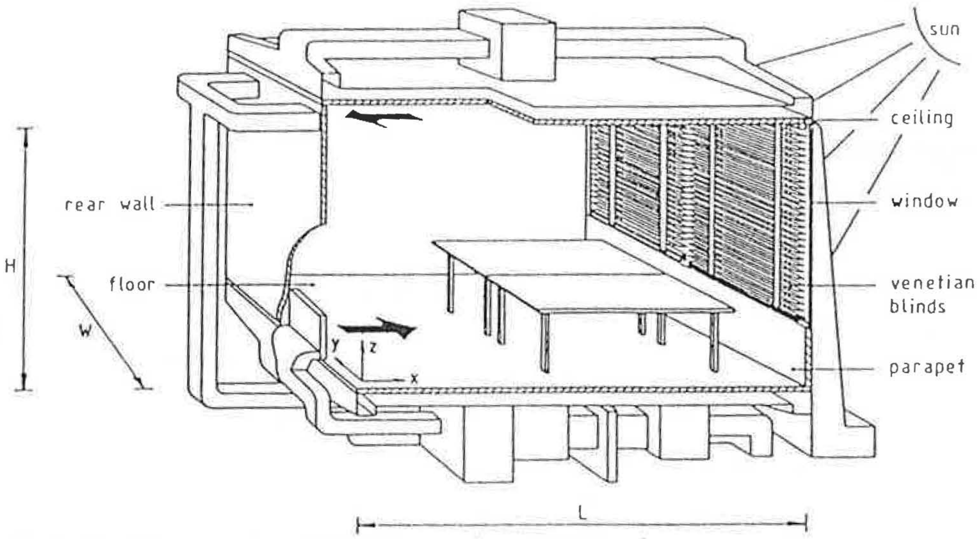



Fig. 1. The displacement ventilation system in a test room.

the temperature difference between the inlet and outlet air is larger for cooling situations. To extract the same amount of heat from the room, the amount of air supply can thus be smaller. This, in fact, is an additional reason why the air supply can be reduced. The indoor air temperature distribution and the inlet and outlet locations in a room have a large influence on energy consumption, because the air supply is directly related to the energy consumption.

In order to estimate the influence of the inlet and outlet locations and the air supply rate on human comfort, the air velocity, the temperature and the contaminant distributions in the room must be predicted. For studying the influence of the inlet and outlet locations and the amount of air supplied on energy consumption, the heat supplied to and heat extracted from the room and energy requirement of the primary equipment such as the ventilator, chiller and boiler of the system should be calculated. With these data, the designer will be able to give an overall assessment of a particular displacement system.

During recent years, the predictions of indoor airflow distributions have made considerable progress, but most of the research interest was aimed at studying the influence of indoor airflow on comfort. Predictions of building energy consumption have been based on the assumption of a uniform indoor air temperature distribution, and the influence of the air supply and air exhaust system culd therefore not be calculated.

could

In order to solve this problem, the nature of the indoor airflow and the transient heat transfer to the walls must be predicted simultaneously, especially under low air supply conditions. This is because the airflow and transient heat transfer in a room are interrelated.

STATE OF THE ART

*Numerical calculation of indoor airflow*

The study of airflows in rooms using numerical calculation techniques has continued for nearly twenty uears and has achieved remarkable successes. The range of airflow simulations, which originally comprised laminar, two-dimensional, steady and isothermal situations, has been enlarged to include turbulent, three-dimensional, transient and buoyancy-affected flows. Nielsen [2] was one of the earliest researchers to use numerical predictions of indoor airflow. His work mainly involved two-dimensional, steady and isothermal flows. Although his two-dimensional results are not very useful for engineering applications, the methods he used showed a very strong potential for solving practical airflow problems in a room. Later, he and his co-workers extended the numerical calculations for the indoor airflow problems to non-isothermal situations [3] and pollutant concentrations [4]. Since the end of the 1970s, three-dimensional computations of turbulent airflow in a room have become increasingly

year

popular as more powerful computers became available. Many researchers validated their computational results with experiments [5, 6]. Later, publications concerning indoor airflow dealt with more specific practical problems. Examples are the assessment of the safety and quality of flow in industrial buildings [7], the evaluation of ventilation systems in a room [8], the exploration of the possibilities of applying the computational results to engineering design [9], and the study of the flow characters of air supply terminals, for instance, a radial air distributor [10] and an air-conditioner unit [11]. More applications for airflow patterns in and around buildings can be found in refs. 12, 13. The results indicate that numerical calculation with the $k-\epsilon$ turbulence model can be used to study problems of indoor airflows.

In addition, almost all of the computations are carried out with the assumption that boundary conditions such as wall surface temperatures and the convective heat of solar radiation through windows, which cannot always be prescribed, are known.

From this previous work, we may therefore conclude that numerical calculation techniques can be used to predict the air velocity, temperature and concentration distributions in a room.

*Numerical calculation of transient heat transfer in a room*

The methods for computing the complex heat transfer in walls have evolved from one-dimensional with periodic boundary conditions to one- or two-dimensional with random boundary conditions (e.g., climatic variations). The hourly heat exchange among walls and the annual energy consumption of a building can be accurately determined. Because of the introduction of computers in this field in the late 1960s, the transient heat transfer in a room with random boundary conditions can be calculated by the finite difference method or by the finite element method.

Stephenson and Mitalas [14, 15] presented the methods of response factors and ~~the~~ $z$-transfer factors for calculating transient heat conduction. These methods are generally accepted as the cornerstone of building thermal analysis. They have made it possible to numerically invert the difficult Laplace transform functions for multilayer heat conduction problems. Thermal response factors and $z$-transfer factors are the results of an exact solution of the heat conduction equations, unlike the approximate solutions provided by the finite difference method or the finite element method. The response factors and $z$-transfer factors, once computed for a given multi-layer wall, can be used repeatedly to evaluate the heat flux and surface temperature of that wall.

The response factors method and $z$-transfer factors method have since been successfully combined with the calculation of heat transfer among the walls of a room and of the heat and mass transfer of air-conditioning systems. This allows an hour-by-hour computation of heat supply and heat extraction of a room [16]. As a result, one may obtain an estimation of the annual heat supply and heat extraction of a building based on the hourly approach. However, most cooling load programs, which are used to calculate the heat supply or heat extraction of a room, assume uniform indoor air temperatures. Hence, they are inappropriate to study the influence of air supply and air exhaust systems on building energy consumption and on human comfort.

The preceding results show that the existing methods are adequate for calculating the hour-by-hour heat supply or heat extraction and annual energy consumption of a room. However, modification of the existing approach is necessary in order to consider the influence of indoor air temeprature distributions. temper

The present state of the art allows us to combine an airflow program and a cooling load program for solving the problem mentioned in the former Section. The inputs required by an airflow program are enclosure surface temperatures, inlet and outlet locations, inlet mass flow rate, and inlet air temperature. These can be obtained from the outputs of a cooling load program. On the other hand, 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. Both programs can, therefore, be coupled, in order to obtain more accurate results and to reveal the overall physics phenomena. This implies that the combination of an airflow program with a cooling load program can be used to estimate more accurately boundary conditions for airflow simulations and to study the influence of the indoor airflow and temperature distribution on room energy consumption and comfort. Hence, our work will concern the combined problem.

aim Our air is to present a methodology for the computation of air velocity, temperature and contaminant concentration distributions in a room with a displacement system and for and the prediction of the heat extracted of the energy consumption of the system.

## THE THEORETICAL FUNDAMENTALS

As the basic point of developing the displacement system is to obtain better indoor air quality while saving energy, detailed information of indoor airflow patterns and the energy consumption of primary equipment are very important factors for evaluating the ventilation system. The corresponding computational procedure can be divided into three parts:

(1) the simulation of the indoor airflow patterns;

(2) the prediction of the space loads (heat extraction or heat supply), and

(3) the calculation of the energy consumption of the primary equipment for the displacement ventilation system.

*The interface between the airflow, space load and energy consumption calculations*

(1) The air velocity, temperature and contaminant concentration distributions can be computed by an airflow program which is based on the conservation equations of mass (continuity), momentum, energy, concentration, and turbulence parameters. The inputs required by an airflow program are enclosure surface temperatures and mass flow rate and the temperature of air supply, etc. These can be obtained from the outputs of a cooling load program. 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.

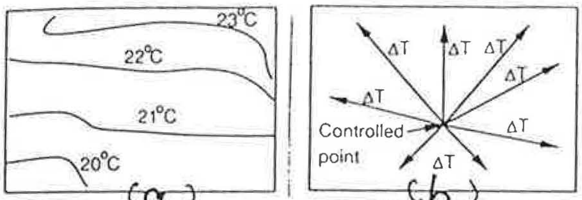
The computations of airflow for a displacement system are very expensive [17]. With present computer capacity, a direct combination between the two programs is not realistic for an hour-by-hour computation for a reference year. Chen *et al.* [17] proposed a simplified method to avoid the direct combination. They suggested to calculate a number of temperature distributions by the airflow program for a room under 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, etc., and 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. window The choice of the ventilation rates and space loads should be realistic. For instance, for a two-person office room, the ventilation rates can be 0 - 7 air changes per hour (ach) and the space loads can be 0 - 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, temperature and contaminant concentration fields which may be encountered in the room under various kinds of conditions, about 10 flow fields should be computed with reasonable combination of Vent and $Q$.

(2) From the computed temperature fields such as the one shown in Fig. 2(a), the following function can be determined:

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

where $\Delta T$ are the temperature differences between the controlled point (i.e., in the middle of the occupied zone) and the air near the enclosure surfaces as shown in Fig. 2(b).

There are some methods for determining the air temperature in a near-wall point. For wall-jet flow, Alamdari and Hammond [18] recommenced use of the wall-jet maximum temperature. This may not be suitable in many near-wall places since the airflow is not of the wall-jet such that no 'maximum' point is available. In addition, the determination of




a + (b) X

Fig. 2. Cooling load program ACCURACY uses air temperature distributions. (a) The temperature distributions are calculated from an airflow program. (b) The temperature differences are used in the cooling load program.

the 'maximum' location can be difficult in many flow programs with the $k-\epsilon$ turbulence model and the wall functions [19]. Another method proposed by Altmayer *et al.* [20] is to calculate from the wall surface temperatures. The method seems rather complicated in three-dimensional cases. Hence we use the air temperatures at the points 0.1 m from the walls, because the inner region thickness of the boundary layer is 0.01 - 0.1 m in a room in most cases. The selection of 0.1 m will ensure the near-wall temperature can be calculated from the temperature fields which are computed by the $k-\epsilon$ turbulence model with the wall functions.

For space load computations, the enclosure surface of a room is usually divided into less than 30 areas. However, the flow grid number normally exceeds 3000. Therefore, an area-averaged method is used for finding $\Delta T$ from the temperature fields.

The $\Delta T$ functions will then be used as an input to a cooling load program, such as ACCURACY [21], which can study the influence of temperature distributions on the space load. In this way the influence of different kinds of ventilation systems on space loads and extracted air temperatures of the room can be studied. The hour-by-hour annual space loads and extracted air temperatures calculated by ACCURACY with the temperature gradients are different from those obtained with the assumption of a uniform indoor air temperature distribution [21].

(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 to calculate the annual energy consumption of the room.

The interrelationship among these three parts is illustrated in Fig. 3. In the following subsections, the theoretical fundamentals of the computations of indoor airflow, space

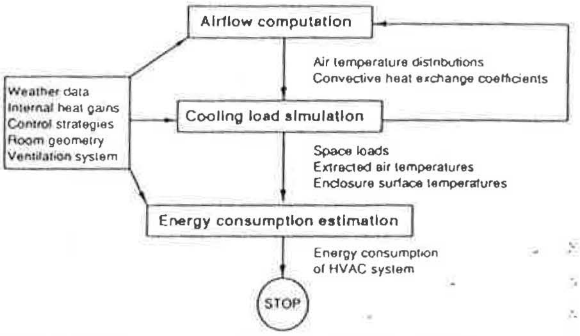



Fig. 3. The diagram of the simulation model.

load and annual energy consumption of a room will be discussed.

*The computation of indoor air temperature distributions*

In most cases, the airflows encountered in an air-conditioned room are turbulent. The mathematical modelling of turbulent flow is now within the capabilities of modern mathematical and numerical methods. Among the turbulent models, the two-equation high Reynolds number $k-\epsilon$ model of turbulent transport [19] seems most widely used for airflows in a room. The $k$ stands for the kinetic energy of turbulence and $\epsilon$ for the dissipation rate of turbulence energy.

Since the temperature difference in room air is relatively small compared to the mean Kelvin temperature, it is a common practice to use the Boussinesq approximation. This approximation takes air density as constant and considers buoyancy influence on air movement in the momentum equation. Since the $k$ and $\epsilon$ equations are derived from the momentum equation, the buoyancy term in the momentum equation is then changed into buoyancy production terms in the $k$ and $\epsilon$ equations. The governing equations with the $k-\epsilon$ model consist of the transport conservation equations os mass (continuity), momentum ($V_i = u$, $v$ or $w$), energy ($H$), concentration ($C$), turbulence energy ($k$) and dissipation rate of turbulence energy ($\epsilon$). They are: of

$$\frac{\partial V_i}{\partial x_i} = 0 \quad (2)$$

$$\frac{DV_i}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_t} + \nu_l\right)\frac{\partial V_i}{\partial x_j}\right] - \frac{\partial}{\partial x_i}\left(\frac{p}{\rho} + \frac{2}{3}k\right) + \beta(T_o - T)g_i \quad (3)$$

$$\frac{DH}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_H} + \frac{\lambda}{\rho C_p}\right)\frac{\partial H}{\partial x_j}\right] \quad (4)$$

$$\frac{DC}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_C} + D\right)\frac{\partial C}{\partial x_j}\right] \quad (5)$$

$$\frac{Dk}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_k} + \nu_1\right)\frac{\partial k}{\partial x_j}\right] + \nu_t\left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right) \times \frac{\partial V_i}{\partial x_j} - \epsilon + \beta\,\frac{\nu_t}{\sigma_H}\,\frac{\partial(T - T_o)}{\partial x_i} g_i \quad (6)$$

$$\frac{D\epsilon}{Dt} = \frac{\partial}{\partial x_j}\left[\left(\frac{\nu_t}{\sigma_\epsilon} + \nu_1\right)\frac{\partial \epsilon}{\partial x_j}\right] + C_1\nu_t\,\frac{\epsilon}{k} \times \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right)\frac{\partial V_i}{\partial x_j} - C_2\,\frac{\epsilon^2}{k} + C_3\,\frac{\epsilon}{k}\,\beta\,\frac{\nu_t}{\sigma_H}\,\frac{\partial(T - T_o)}{\partial x_i}\, g_i \quad (7)$$

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

$$\nu_t = C_\mu k^2/\epsilon \quad (8)$$

where $\sigma_k = 1.0$, $\sigma_\epsilon = 1.3$, $\sigma_H = 0.9$, $\sigma_C = 1.0$, $C_1 = 1.44$, $C_2 = 1.92$, $C_3 = 1.44$ and $C_\mu = 0.09$.

For wall flow, where local Reynolds numbers are considerably lower, the equations are normally used in conjunction with empirical wall function formulas [19]. It should be pointed out that 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 the wall function formulas.

A computer code, PHOENICS, developed by Gunton *et al.* [22] was used for the present study. The computational method involves the solution, in finite-volume form, of two- or threedimensional conservation eqns. (2) - (7). The methodology for performing the numerical calculations uses under-relaxation and false time-step factors for obtaining a convergent solution. All calculations were performed using an up-wind differencing scheme and staggered grids. A complete description of the theoretical basis and the numerical solving procedure can be found in ref. 23.

three-dimensional

*The simulation of the space load of a room*

Normal cooling load programs that assume the room air temperature to be uniform are not suitable for energy analysis with different air supply and exhaust systems, such as displacement ventilation systems and well-mixed systems. This is because they will yield the same results such as inlet and outlet air temperatures and space loads, for all kinds of ventilation systems. For heat transfer through the enclosures of a room, the air temperatures near the enclosure surface are important. Hence, in the computation of the space load, the field values of air temperature distributions can be replaced by the air temperature at the controlled point and the air temperature differences between the controlled point and the air points near the surfaces ($\Delta T$) as shown in Fig. 2. The air temperature at the controlled point (i.e., in the middle of the occupied zone) depends on the control strategy and is known. The $\Delta T$ can be determined as the function of space loads ($Q$) and room ventilation rates (Vent) from the pre-calculated fileds of air temperature as discussed above.

In addition, the $\Delta T$ can also be calculated from the airflow patterns which are precomputed by a flow program. Details can be found in ref. 23.

In the cooling load program ACCURACY [21, 23], eqn. (1) is combined with the energy balance equations of the room:

$$\mathrm{M}\cdot\mathrm{T} = \mathrm{q} + \alpha\,\Delta T \quad (9)$$

where

$$\mathrm{M} = \begin{bmatrix} \alpha_{1c} + \sum_{k=1}^{N} \alpha_{r_{1,k}}, & -\alpha_{r_{1,2}}, & -\alpha_{r_{1,3}}, & \cdots, & -\alpha_{r_{1,N}} \\ -\alpha_{r_{2,1}}, & \alpha_{2c} + \sum_{k=1}^{N} \alpha_{r_{2,k}}, & -\alpha_{r_{2,3}}, & \cdots, & -\alpha_{r_{2,N}} \\ \cdots & & \cdots & & \cdots \\ -\alpha_{r_{N,1}}, & -\alpha_{r_{N,2}}, & \cdots, & -\alpha_{r_{N,N-1}}, & \alpha_{Nc} + \sum_{k=1}^{N} \alpha_{r_{N,k}} \end{bmatrix} \quad (10)$$

$$
\mathrm{T} = \begin{bmatrix} T_1 \\ T_2 \\ \vdots \\ T_N \end{bmatrix} \quad \mathrm{q} = \begin{bmatrix} q_{1,\mathrm{coming}} + \alpha_{1c} T_{\mathrm{R}} \\ q_{2,\mathrm{coming}} + \alpha_{2c} T_{\mathrm{R}} \\ \vdots \\ q_{N,\mathrm{coming}} + \alpha_{Nc} T_{\mathrm{R}} \end{bmatrix} \quad \alpha\Delta \mathrm{T} = \begin{bmatrix} \alpha_{1c}\Delta T_1 \\ \alpha_{2c}\Delta T_2 \\ \vdots \\ \alpha_{Nc}\Delta T_N \end{bmatrix} \tag{11}
$$

$\alpha_{i_c}$ is the convective heat exchange coefficient for enclosure surface $i$, $\alpha_{r_{i,j}}$ is the radiative heat exchange coefficient between surfaces $i$ and $j$, $N$ is the enclosure surface number of the room, $T_i$ is the temperature of inner surface $i$, and $T_{\mathrm{R}}$ is the air temperature at the controlled point. The $q_{1,\mathrm{coming}}$ in eqn. (11) are the heat flows coming into the room and can be expressed as:

$$
q_{i,\mathrm{coming}} = q_i + q_{it} \qquad \text{(for walls)} \tag{12}
$$

$$
q_{i,\mathrm{coming}} = q_{is} + q_{mir} - q_{imc} + q_{it} \qquad \text{(for venetian blinds inside of windows)} \tag{13}
$$

The terms in eqns. (12) and (13) are illustrated in Figs. 4 and 5. It should be pointed out that the ventilation rate through blind slats in the present work is determined by measurements [24]. A numerical study for computing airflow field in a window with venetian blinds is being carried out. With eqn. (9), the influence of different kinds of air supply and exhaust systems on space load can be studied.

ACCURACY uses the $Z$-transfer function method [15] for the calculation of heat conduction through the walls. For an external wall, as shown in Fig. 4, the heat conduction to the inner surface, $q_i$, is calculated from:

$$
q_{i_n} = \sum_{j=0}^{M} (Z1_j T_{o_{n-j}}) - \sum_{j=0}^{M} (ZA_j T_{i_{n-j}}) - \sum_{j=1}^{M} (ZB_j q_{i_{n-j}}) \tag{14}
$$

where $n$ stands for current time, $Z1$, $ZA$ and $ZB$ are $z$-transfer factors and $T_o$ is the outside surface temperature of the wall.

Since the inner enclosure surfaces are assumed to be grey bodies, the multiple reflections among the surfaces must be accounted for. The radiative heat exchange ($Q$) between surfaces $i$ and $j$ is calculated from:

$$
Q = \epsilon_i \varphi_{i,j} (E_{\mathrm{b},i} - E_{\mathrm{b},j}) A_i \tag{15}
$$

where $\varphi_{i,j}$ is determined from:

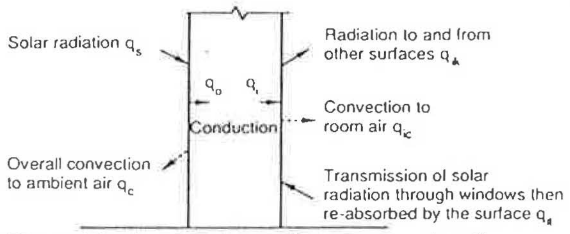



Fig. 4. The heat exchange in an external wall.

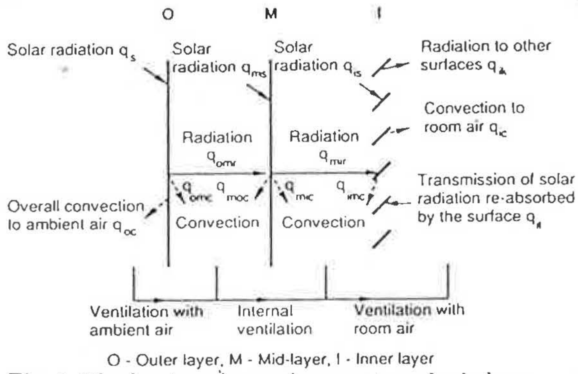



Fig. 5. The heat exchange in an external window.

$$
[\varphi] = \{[I] - [F]\cdot\mathrm{diag}(1-\epsilon)\}^{-1}\cdot[F]\cdot\mathrm{diag}(\epsilon) \tag{16}
$$

leave [ ] out

In eqn. (16), [I] is the unit matrix, [F] is the view factor matrix and $\epsilon$ is the enclosure surface emissivity.

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 Fig. 5 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 re-absorbed uniformly by each inner surface. The overall absorptivities, transmissivities and reflectivities of a window with double glass panes and venetian blinds are calculated based on the method described by Oegema [25]. A more detailed description is reported in ref. 23.

*The estimation of annual energy consumption of buildings*

The energy requirements and fuel consumption of HVAC systems should be estimated for short- or long-term operation. The procedures for estimating energy requirements vary considerably in degree of sophistication. However, they have three common elements: (1) space load calculation, (2) secondary equipment load computation, and (3) primary equipment energy requirements computation [26]. The secondary refers to equipment that distributes the heating, cooling or ventilating medium to the conditioned spaces. The primary refers to central plant equipment that converts fuel or electricity to heating or cooling effect.

There are many kinds of methods available for the energy analysis. They can be classified as (1) single measure methods, (2) simplified multiple measure methods and (3) detailed simulation methods [26]. The single measure methods use only one measure, such as annual degree-days. They may be appropriate only for simple systems and applications. In the simplified multiple measure methods, the accuracy is improved by using more information, such as the number of hours anticipated under particular operating conditions. Among these methods, the bin-method is best known [26]. The most elaborate methods perform energy balance calculations hourly over a given analysis period, typically one year. These are called detailed simulation methods.

The single measure methods and simplified multiple measure methods are not good enough for accurate energy analysis and, therefore, they are not used for the present study.

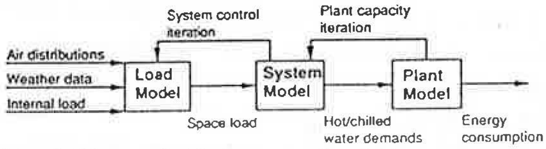



Fig. 6. The model for energy simulation.

Figure 6 describes a simulation model and shows the basic function of each major model element on an input and/or output basis. Based on this model, many computer programs are written for the building energy analysis such BLAST [27], DOE-2 [28] and [illegible] grams can be found in refs. 30 and 31. Among these programs, most are based on the method of hour-by-hour prediction in all the three elements. The hour-by-hour prediction in all the three elements is expensive, if the weather data of a full reference year are employed. When the prediction is applied to optimize a system for best solution of energy saving, many computations are required for comparing various alternative designs of buildings and installation. In order to reduce the computational time, there are two possibilities. The first one is to use short reference year weather data [32]. The second is to optimize the computer algorithm used in the secondary equipment and the primary equipment to reduce the computing cost [29]. 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 [29] calculates the annual energy consumption in the following steps:

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

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

(3) estimate 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 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 Fig. 7. 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 [illegible] the accuracy will be. Van Mierlo [33] demonstrated that a partition of the psychrometric chart as given in Fig. 7 results in a difference

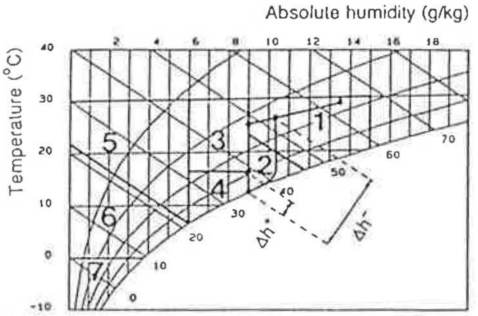



Fig. 7. The division of a psychrometric chart for energy analysis and an example of air handling processes.

less than 5% from those obtained with an hour-by-hour simulation. 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, the cooling load, the 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 $Q^+$ and $Q^-$. 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 Fig. 7, the amounts of energy for heating and for cooling and air transport can be calculated as:

$$E_{gas} = \Delta h^+ \dot{m} / \eta_b \qquad \text{(kWh)} \qquad (17)$$

$$E_{electricity} = \Delta h^- \dot{m} / \eta_{ch} + \Delta p_{fan} \dot{m} / (\eta_{fan} \rho) \qquad \text{(kWh)} \qquad (18)$$

where $\eta_b$, $\eta_{ch}$ and $\eta_{fan}$ are the efficiencies of boilers, chillers and fans, respectively. They depend on various variables, such as load, and can be determined in many ways [29]. Here $\Delta h^- \dot{m}$ and $\Delta h^+ \dot{m}$ are the heat removed by cold water and the heat supplied by hot water, respectively, and $\Delta p_{fan}$ is fan pressure.

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 $X$. The sum of all the products amounts to the total energy consumption. Multiplying with electricity and gas price, it presents the total energy costs.

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 can be referred to [23].

In the present study, the weather data of the Dutch short reference year [32] is used to reduce the computing time.

## AN APPLICATION EXAMPLE

In order to demonstrate the methodology described, an application example for the room shown in Fig. 1 will be briefly presented in this Section.

### *Indoor airflow distributions*

The mechanical ventilation rate of the room is controlled at 5 ach and the heat gain due to the solar radiation through the window is 600 W. In order to simulate a smoking person, a contaminant source, which is 0.5% of the air supply, together with a heat source is introduced on one side of the table.

The air velocity, temperature and contaminant concentration distributions and the measurements are shown in Fig. 8. The agreement between the computations and the measurements is rather good. The concentration of the contaminant in the occupied zone is lower than that in a perfect mixed system where the concentration is 0.5%. It should be noted that, in the displacement system, there is a temperature stratification in the room and a stagnation zone of air. A person feels discomfort to high-speed air movement and to stagnation of air. More detailed information is available in refs. 17 and 23.

### *Space loads*

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. If a variable-air-volume air handling system is used, the space load is related to ventilation rates of the room. Hence, the function in eqn. (1) can be signified as: simplified

$$\Delta T = f(\text{Vent}) \qquad (19)$$

and it can be determined from a number of specific air temperature distributions which were calculated from the airflow program. The procedure of determination has been detailed in refs. 23 and 34.

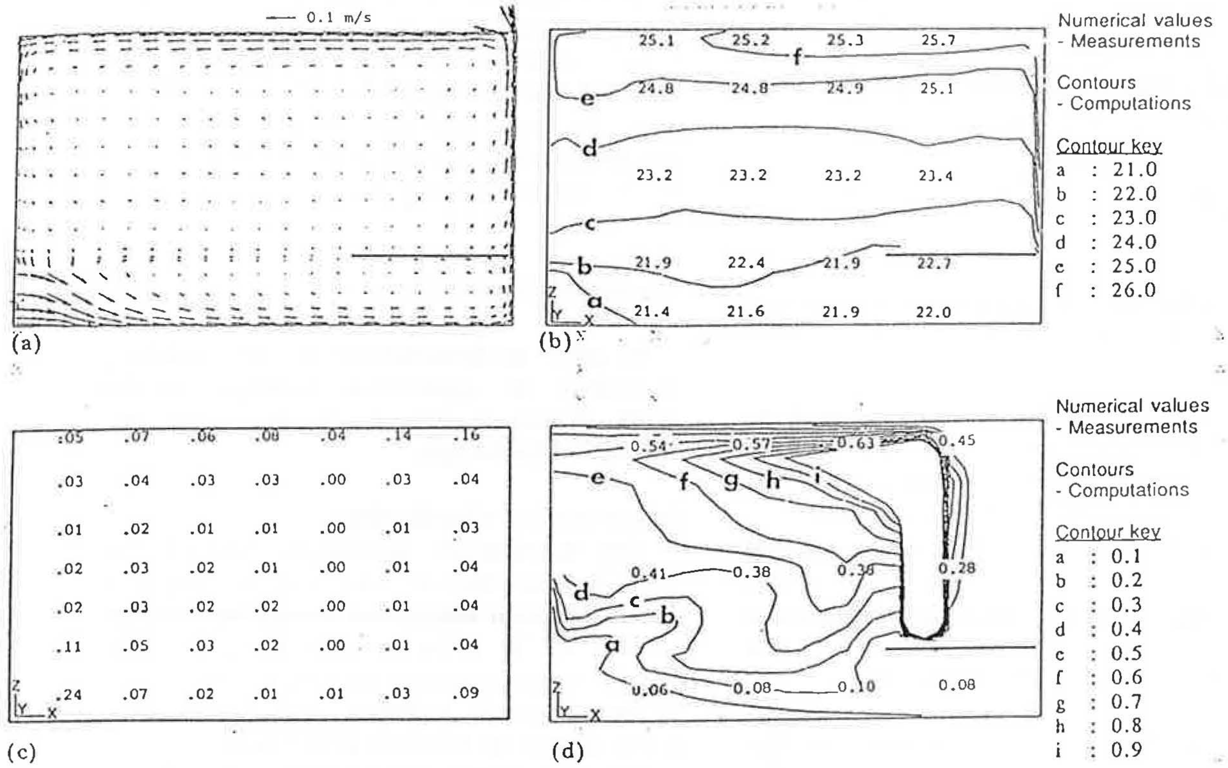



Fig. 8. Computed and measured airflow distributions. (a) Computed velocity distribution in the middle section. (b) Measured velocity in the middle section (m/s). (c) Computed and measured temperature distribution in the middle section (°C). (d) Computed and measured concentration distribution in the section where the contaminant source was introduced (%).

TABLE 1

Total amount of air supply required for cooling in the displacement ventilation system and the well-mixed system ($10^3$ kg)

| Seasons | Displacement system | Well-mixed system |
|---|---|---|
| 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 |

ACCURACY calculates the annual heat extraction and heat supply of the room with the weather data of a short reference year [32]. The total amount of air supply required for cooling is presented in Table 1. The results for a well-mixed system are also given for comparison. Due to the existence of an average vertical temperature difference of 1.95 K with the sidplacement system, the total air mass supplied for cooling is about 37% smaller than that in the well-mixed system (no vertical temperature difference).

splacement

*Energy consumption of primary equipment*

Table 2 presents the computational results of the energy consumption for cooling situations for the displacement ventilation system (with an average vertical air temperature difference of 1.95 K for cooling). Table 3 gives the results for cooling situations for the well-mixed system (without a vertical air temperature difference for cooling). The energy consumption during cooling situations by the chiller and the ventilator for the displacement ventilation system is 30% smaller than that for the well-mixed system and by the boiler it is 32% smaller. The energy consumption between the two systems for heating situations would be the same provided that a radiator is used for heat supply in the displacement system. More detailed results can be referred to [23, 34].

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## CONCLUSIONS

This paper concentrates on the development of a methodology for the combined problems of indoor airflow computation,

TABLE 2

The energy consumption of the room with the displacement ventilation system for cooling situations (kWh)

| Sections in psychrometric chart | 1 | 2 | 3 | 4 | 5 | 6 | 7 | Total |
|---|---|---|---|---|---|---|---|---|
| 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 |

TABLE 3

The energy consumption of the room with the well-mixed ventilation system for cooling sutations (kWh)

| Sections in psychrometric chart | 1 | 2 | 3 | 4 | 5 | 6 | 7 | Total |
|---|---|---|---|---|---|---|---|---|
| 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 |

space load calculation and the estimation of energy consumption of a room with a displacement ventilation system.

Since a direct combination between an airflow program and a cooling load program is not possible due to the limited capacity of existing computers, a simplified method has been introduced. A number of specific indoor airflow fields of a room are first computed by an airflow program with the $k$–$\epsilon$ turbulence model. The influence of an air supply system on the space load as well as on the energy consumption of primary equipment can then be studied by computing temperature distributions in the room air. The cooling load program ACCURACY has been developed for this purpose. A special algorithm is also suggested for the estimation of the energy consumption of the room.

The application example shows that, in a room with a displacement system, the indoor air quality is much better than with a well-mixed system and the energy saving is si-nificant. sig

## LIST OF SYMBOLS

| | |
|---|---|
| $A_i$ | area of wall surface $i$ ($m^2$) |
| $C$ | contaminant concentration (%) |
| $C_p$ | constant-pressure specific heat (J/kg K) |
| $C_1$, $C_2$, $C_3$, $C_\mu$ | coefficients in the $k$–$\epsilon$ turbulent model (—) |
| $D$ | diffusion coefficient of concentration ($m^2$/s) |
| $E_{b,i}$, $E_{b,j}$ | emissive power of black body for surfaces $i$ and $j$ (W/$m^2$) |
| $E_{gas}$, $E_{electricity}$ | gas energy and electricity energy (kWh) leave out [ ] (2) |
| $\mathbf{F}$ | view factor matrix (—) |
| $H$ | specific enthalpy (J/kg) |
| $\mathbf{I}$ | unit matrix (—) bold text for the I and not italicized |
| $k$ | kinetic energy of turbulence (J/kg) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $M$ | total term of the $z$-transfer factors |
| $\mathbf{M}$ | coefficient matrix in the energy balance equation (—) |
| $n$ | current time (h) |
| $N$ | total surface number of a room (—) |
| $p$ | pressure (Pa) |
| $P$ | probability (—) |
| $\mathbf{q}$ | heat flux matrix in the energy balance equation (—) |
| $q_i$ | heat flux in the inside surface of room enclosure (W/$m^2$) |
| $Q$ | radiative heat exchange between surfaces $i$ and $j$ (W) |
| $Q$ | space load (W) |
| $T$ | temperature (K or °C) |
| $\mathbf{T}$ | matrix of the inside surface temperatures of enclosure (—) |
| $T_i$ | inside surface temperature of enclosure (°C) |

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$T_o$ — reference temperature or outside surface temperature of enclosure (°C)
$T_R$ — air temperature at the controlled point of a room (K or °C)
Vent — room ventilation rate (ach)
$V_i$ — general velocity (m/s)
$t$ — general time (s)
$x_i$ — general coordinate (m)
$X$ — absolute humidity (kg vapour/kg air)
$Z1_j, ZA_j, ZB_j$ — $z$-transfer factors (—)
$\alpha_{ic}$ — convective heat exchange coefficient on inside surface $i$ (W/m$^2$ K)
$\alpha_{r_{i,j}}$ — radiative heat exchange coefficient from inside surfaces $i$ to $j$ (W/m$^2$ K)
$\beta$ — gas expansion coefficient (1/K)
$\Delta h^+, \Delta h^-$ — enthalpy difference for heating and cooling (J/kg)
$\Delta p_{fan}$ — fan pressure (Pa)
$\Delta T$ — air temperature difference between the controlled point and a near wall point (K)
$\epsilon$ — dissipation rate of turbulence energy (J/kg s)
$[\epsilon]$ — ~~matrix of wall emissivity (—)~~ surface
$\epsilon_i$ — emissivity of wall surface $i$ (—)
$\eta_b, \eta_{ch}, \eta_{fan}$ — efficiencies of boiler, chiller and fan (—)
$\lambda$ — heat conductivity (W/m K)
$\nu_l$ — fluid laminar kinetic viscosity (m$^2$/s)
$\nu_t$ — fluid turbulence viscosity (m$^2$/s)
$\rho$ — air density (kg/m$^3$)
$\sigma_k, \sigma_\epsilon, \sigma_H, \sigma_C$ — Prandtl or Schmidt number (—)
$[\varphi]$ — matrix of the radiative heat exchange factor (—)
$\varphi_{i,j}$ — radiative heat exchange factor between surfaces $i$ and $j$ (—)

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