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ROOM: A METHOD TO PREDICT THERMAL COMFORT AT ANY POINT IN A SPACE

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The paper presents the theoretical background to an analytical tool, called ROOM, that is used by a large design practice to predict comfort conditions, and plot detailed comfort contours within a given space. ROOM is based on a thorough analysis of the radiant heat transfers that occur within a complex space linked to an explicit finite difference treatment of elements that store thermal energy. Airflow modelling is, at present, limited to simple single zone or two zone (stratified) space, with buoyancy driven ventilation as an option. The output can, however, be used as input to a computational fluid dynamics program, and an example is given in the paper.

INTRODUCTION

The general objectives of commercial building and HVAC system designers are to provide spaces in which the occupants can work. This means that, within limits, these spaces must be comfortable. The definition of comfort has been addressed by many workers, (notably Fanger (1) for example) and is not the subject of this paper, except that it needs to be recognised that the use of a space is an important parameter. For example, a transitory space can have a different specification from a sedentary area. Further if occupants are free to move around the space and so avoid cold or hot radiation sources different criteria may be applied (2). Designers within the Ove Arup Partnership recognised the need for a means of calculating comfort throughout a space. The ROOM calculations were known to be complex and best done by computer. A specification for the program (ROOM) was drawn up:

The main requirements were, that the program should:

Model a single space;
The shape should not be restricted to a six sided rectangular prism;
The model should not be steady state;
The main parameters associated with Fanger's (1) comfort equation should be predicted;
The program should be capable of development through a number of levels.
The program should run on a standard IBM PC (or compatible).

The concept of different levels was a very important feature in the development of ROOM. It allowed a rational approach to handling many of the complex issues concerned with heat transfers within spaces, and assisted in quality control and validation. That is as the code is developed the effect of enhancements can easily be checked against early more simplistic versions.

Essentially the levels can be described as:

Level 1 Simple room shapes (of up to 20 surfaces), prediction of bulk comfort parameters such as dry resultant temperature (3) at averaged conditions. Fully mixed space.

Level 2 Simple room shapes (of up to 30 surfaces), air temperature and velocity constant throughout the occupied area. Simple mixed or stratified air flow model. Surface temperatures and direct radiation considered in the prediction of comfort. Comfort indices calculated at discrete points in the occupied space. Fixed surface convection coefficients.

Level 3 Complex room shapes (number of surfaces limited by memory size). Detailed

representation of air flow patterns, and prediction of convection coefficients.

Current development is towards a level 3 program. Algorithms to calculate radiation form (shape or configuration) factors for spaces where not all surfaces can see each other are in the validation stage. The next phase will involve the introduction of variable surface convection coefficients. Simple airflow models will then be introduced. These models will probably make use of prescribed air flow patterns and be the first step towards the combination of ROOM and a computational fluid dynamic code (CFD). It is, of course, most unlikely that a detailed air flow model can be incorporated in a program designed to run on an IBM PC. The final version of ROOM will therefore find its home on one of the newer forms of computer workstations, and a simple 2D version coupled to a computational fluid dynamics code has already been demonstrated (4). At present the link between ROOM and CFD models is limited to ROOM providing the boundary conditions. Figure 1 gives an example of the results of a design study using this technique.

The paper first looks at the history of ROOM, and then presents the basis of the algorithms incorporated within the present version.

HISTORY

The program specification indicated the specific features that would eventually be required. The most important being that the computer code should be capable of continuous development. Further it should preferably make use of the author's experience and programs, or algorithms, already developed in the practice. These two considerations led directly to an explicit finite difference approach to fabric conduction. The requirements for comfort prediction meant that the following would need to be calculated:

Dry bulb temperature;
Surface temperature;
Direct radiation falling on occupants;
Air speed.

Prediction of surface temperatures requires a good representation of the heat transfer processes at the surface. Holmes (5) has demonstrated the effect of a number of long-wave radiation models on predicted surface temperature. Although it is concluded in a more recent publication (6), that relatively simple radiation models are probably sufficiently accurate to model overall energy requirements (design heat loads in the case of 5), this is not so for surface temperatures. Further neither of these investigated anything more than simple room shapes so the conclusions may not be valid. An in-house program, VAULT(7), contained a full longwave radiation model and two other useful features:

Calculation of natural ventilation due to stack effect and effect of mechanical systems on that ventilation rate.

A representation of internal temperature gradients by dividing the space into an upper and lower region.

Further VAULT had been used in the design of a number of buildings, including the large tent like structure that was the main building at 1984 Liverpool Festival Garden Exhibition (8) and so had credibility within the practice. It was therefore decided that VAULT could be developed to provide a radiation and simple air flow model for ROOM.

A further important aspect of the model is the representation of the storage of heat within the building fabric. Again previous work was examined. Eventually the method used in a program written to assist in the design of a novel building by Arup Associates (9,10) was adopted.

In addition to radiant heat transfer, ventilation and representation of building fabric it was also necessary to incorporate models of simple space temperature control plant. It would have been relatively simple to specify constant air temperatures (say) during occupation, but this course was not taken. It was decided to introduce some realism and make use of a few very simple system simulations:

Air heating and cooling plant under proportional control;
Heated or cooled floors;
(With proportional control or external dry bulb compensation);

Fixed temperature surfaces;
Profiled supply air temperatures.

Again the development made use of previous research experience (11).

In addition to the technical requirements of the methods used in the program it was also necessary to ensure that ROOM could be used by the design engineer. Much development time therefore, was associated with minimising instabilities in the numerical methods, data vetting, and trying to prevent the user doing silly things (this is almost impossible).

Finally attention had to be given to the presentation of output, and samples are given in Figures 2 to 4, showing predicted space conditions, such as:
Air temperature;
Surface temperatures;
Percentage person dissatisfied (1).

While it was recognised that graphical presentation of results is a useful way to get ideas over, the program is intended to be used for engineering design so all predictions are also available in tabular format.

THEORY

The basic ROOM algorithm makes use of what is known as an explicit method (mathematically-forward finite differences). Essentially this means that conditions in the future are predicted from the current state. It is therefore unnecessary to make assumptions such as that of cyclic conditions used in the CIBSE admittance method. However, in many cases the designer is interested in performance under limit conditions. In that case repeated climatic sequences are often appropriate. Because of this, and the very practical need to have a simple representation of climatic data, at present ROOM runs under steady cyclic conditions. This does not invalidate the mathematical approach because most gains, and control actions do not follow the sine wave with a period of 24 hours that is the basis of the CIBSE admittance method.

This section is divided into five parts:

Heat transfer processes;
Ventilation;
Plant;
Internal gains;
Comfort.

It will be seen that in many cases standard values are quoted. These have been chosen for design purposes. It is easy to modify the code to include algorithms to produce non-standard values should that be required.

Heat transfer processes

Heat is transferred by conduction, convection and radiation. In a building the physical processes are:

Conduction - passage of heat through the building elements;
Convection - interchange of heat between the building element and the surrounding air;
Radiation - interchange of heat between building surfaces, or the transfer of heat from high temperature sources such as the sun.

All heat transfer associated with glazing is calculated using standard optical theory (12) (not solar gain factors or shading coefficients) and the external heat transfer temperature is the sol-air temperature as defined in CIBSE Guide Book A. This section is therefore limited to a discussion of the heat transfers that occur within a space and through its surfaces.

ROOM is a dynamic thermal model, this means that storage of heat within building elements is taken into consideration. The way this is done is discussed first followed by surface convection then radiation, and finally the overall mathematical model (ROOM matrix).

Conduction processes

The model employed in ROOM assumes that a wall can be represented by a number of small 'elements' in series. Each element can both store and conduct heat. The elements are assumed to be sufficiently small that each is at a uniform temperature. In this case:

Heat required to raise the temperature of the element by a small amount is equal to the heat flow into the element in a short time.

That is:

$$CdT = Qdt \qquad (1)$$

where:

C is the thermal capacity (mass x specific heat)

dT is the change in temperature

Q is the heat flow (usually equal to the difference in temperature between the current and adjacent elements divided by the thermal resistance)

dt a small time step

The simplest forward difference method makes use of the following algorithm to calculate the temperature of an element at time t:

$$T_t = T_{t-\delta t} + \frac{dT}{dt}\,\delta t \qquad (2)$$

where:

δt is a small time step

$\frac{dT}{dt}$ is calculated from equation (1) at time $t-\delta t$

Unfortunately numerical stability, which ensures sensible answers, usually requires a small time step or rather large elements. To minimise problems ROOM uses a method called 'fourth order Runge-Kutta' (13). Essentially several estimates are made and averaged in a particular way. Much larger time steps can then be used, this is important if run times are of concern. The problem of stability still exists and it is necessary to make a good estimate of the time step.

Each fabric type (wall) is subjected to a large change in air temperature (+50°C at one side and -50°C at the other). The response is calculated using a 'sixth order Runge-Kutta' (13) algorithm. The special feature of this algorithm is that it can calculate the numerical error and adjust the time step to ensure an acceptable level of accuracy (in practice so that the error does not grow). The integration time used in ROOM is determined by this method.

Convection processes

Convection is a complex process. Accurate values of surface heat transfer coefficients are not known. Values are given in the CIBSE Guide. Some work (14) has been done to look at internal surface coefficients in naturally ventilated rooms, which suggest that the CIBSE Guide values are reasonable.

Under natural convection the heat transfer coefficient is a function of the air-surface temperature difference and the direction of heat flow. This introduces non-linearities into the system that would require the heat transfer equations to be re-assembled at every time step. This would mean a significant increase in run-time. So ROOM uses a fixed surface heat transfer coefficient, for natural convection, of $3W/m^2K$. This is an average of values in the CIBSE Guide and also the figure used in the calculation of admittance and it associated parameters (15).

There is even less information available for inside convection coefficients with forced

convection, and so a figure of $3W/m^2K$ is used again.

External convection coefficients depend upon the local micro-climate, surface roughness and the position of the surface on the facade. Studies have been carried out (16) to determine the effect of wind speed on the external surface convection coefficient, but more work needs to be done. A research project is underway at Loughborough University of Technology (17) to examine the effects of mullions and similar irregularities on the convection coefficient. The results will not be available for some time, so a fixed external heat transfer coefficient of $20W/m^2$ (an average value from the CIBSE Guide) is used in ROOM.

The limitations of these assumptions are recognised and are being addressed by current development work.

Radiation and the ROOM matrix

Heat loss and gain to rooms is closely linked to the radiation exchanges that occur within the space. The effect of solar radiation and lights is fairly obvious, the significance of radiant exchange between room surfaces is often not realised. However, it only requires a cursory glance at CIBSE Guide Book A to see that the radiation heat transfer coefficient for walls is about $5W/m^2K$, compared with the typical value of $3W/m^2K$ for convection. The physics of radiant heat transfer are well understood but their application to buildings is often confused and incorrect. Indeed it is this aspect of heat transfer that causes most arguments when the calculation of steady state heat loss is discussed. ROOM avoids the problem by going back to the fundamental equations. Assumptions are made, however they are limited, the main ones being:

Grey body radiation - a single emissivity regardless of the wavelength of the radiation.

Linearised radiant heat transfer.

The first of these is a generally accepted approximation, however it is usual to separate radiation into two groups, long and shortwave. Longwave radiation occurs between room surfaces, and is temperature dependent. Shortwave radiation comes from high temperature sources such as the sun and lights. The transfer of shortwave radiation within a room does not depend upon the surface temperature. Shortwave radiation bounces around the space either being absorbed at the surface or transmitted by any glazed surfaces. It is a common mistake to assume that all the solar radiation entering a space is trapped within that space. Glazing is indeed opaque to longwave radiation, however shortwave radiation is only converted to longwave by the increase in surface temperature on absorption at the surface. The emissivities appropriate to long and shortwave are often very different. For example a white wall may have a shortwave emissivity of 0.4, while that for longwave will be nearer to 0.9. At present ROOM uses the same emissivity for both long and shortwave radiation. The reflection from surfaces is however taken into consideration as is the direct transmission of solar radiation should it fall on the inside of a glazed surface.

The linearised radiation heat transfer coefficient is again a common assumption. The reason for doing this in ROOM is to speed up calculations. ROOM uses a radiation coefficient of about $5W/m^2K$.

This section discusses how ROOM handles:

Short wave radiation
Longwave radiation
The interaction between the three modes of heat transfer.

Short wave radiation

This needs to be divided into the radiant input from lights and the sun. Radiation from lights is in the present version, assumed to fall on the floor.

Solar radiation is handled in a fairly conventional way. Incident radiation is calculated using methods described in the CIBSE Guide, then the amount absorbed and transmitted at glazed surfaces is calculated using the fundamental equations for glazing (12). The transmitted radiation is separated into direct and diffuse components. The diffuse radiation is distributed within the room in proportion to the radiation form factor

between the window and each room surface. The treatment of direct solar radiation is a little more complex. In this case the internal distribution of radiation is calculated by determining where the solar radiation falls within the space. This is done by projecting appropriate glazed surfaces onto the other room surfaces. The direct radiation is then averaged over each surface as appropriate and added to the diffuse. It is then assumed that non-specular reflections occur, so that the radiant load can be distributed over the room surfaces in accordance with the theory of multiple reflections represented by the radiosity for each surface (18). The method used is equally applicable to indirect lighting calculations.

The amount of solar radiation entering a room not only depends upon the properties of the glazing but also of any shading devices. External shade and internal blinds are modelled.

Surfaces are not shaded from direct sunlight by devices mounted in the proximity of the fenestration, but by other surfaces, structure and fittings within the space. The effect of other surfaces is taken into consideration in the geometrical manipulations carried out in the sunlight distribution routines. Other shading devices are accounted for by two additional factors: the high and low level shade factors.

The high level factor operates on all direct radiation entering the space converting a specified proportion into convective heat. Furnishing, plants and other items will prevent solar radiation reaching the floor. The fraction of the floor so obscured is represented by the low level shade factor. In this case a proportion of the direct solar radiation entering the space (after application of the high level factor) will be converted to convective heat.

Surface Radiation

The simple radiation transfer models used in design such as the CIBSE Guide were not considered appropriate for ROOM. The method used in ROOM is based on considerations that take into account the infinite number of reflections that occur between surfaces - as used in indirect lighting calculations. However, it has already been stated that longwave radiation depends on surface temperature, so how is it that the radiant exchange be decoupled from the conduction and convection calculations? The reason is simple, the explicit finite difference formulation makes use of the current state to predict the future, so all that is necessary is to obtain current values of longwave radiation incident on each surface and use these in the prediction of temperatures within the fabric at future time. (Things are a little more complicated where low mass surfaces are concerned and this is discussed later. The distribution of longwave radiation (and non-specular shortwave) is calculated as follows:

The rate at which radiation leaves a surface (radiosity) is equal to the sum of reflected and emitted radiation, that is:

RAD(N) = RHO(N) * RINC(N) + EMISS(N) * EB(N) (3)

where

RAD(N) = radiosity of surface N W/m^2

RHO(N) = reflectivity

RINC(N) = incident radiation W/m^2

EMISS(N) = emissivity

= (1-RHO) if the surface does not transmit radiation

EB(N) = black body emissive power W/m^2

The black body emission is equal to the Stephan Boltzman constant (5.678×10^{-8} $W/m^2(K^4)$) multiplied by the absolute temperature raised to the fourth power. This non-linearity is avoided in ROOM by linearisation, that is:

EB = a + bt (4)

and a good fit over the range -10°C to + 35°C is

a = 311.46

b = 5.26

The incident radiation comprises that from other surfaces and other internal and external sources. The amount of radiation received from other surfaces on surface N is:

$$A(N) * RINC(N) = \sum_{J=I,N} RAD(J) * A(J) * F(J,N) - Q(N) \qquad (5)$$

where A(N) = surface area (m^2)

F(J,N) = the radiation form factor (or configuration factor, or shape factor) for radiant heat flow from surface J to surface N.

-Q(N) = any other radiation incident on N.

The form factor between two small elements of a surface is:

$$f_{i,j} = \frac{\cos\phi_i \cos\phi_j \, \delta A_i \, \delta A_j}{r^2} \qquad (6)$$

where ϕ is the angle between the normal to surface and a line of length r joining them.

δA = the surface area.

Equation 6 has to be integrated over the whole of each surface, and although there are some standard solutions available (19 for example), general solutions are impossible. A numerical method is used in ROOM. The simplest method is to divide the surfaces into a large number of elements and apply 6 directly and sum the results. This is not a good way to do the calculation. The r^2 term can give rise to very large errors when r is small. A much better way is to convert the surface integral into a line integral and tackle that numerically. This is what is done in ROOM. The calculation of radiation form factors is complicated when the view between two surfaces may be partially obstructed by another of the room's surfaces, several stages are required to evaluate the form factors:

1. A simple geometrical test to find which surfaces are potential view obstructors.

2. Taking a pair of the surfaces:

   (a) Apply tests to determine the geometrical relationship between the surface pair.

   (b) Apply geometrical tests to find if any potential view obstructors (found in 1) can be discounted as view obstructors for the current surface pair.

   (c) Project the remaining view obstructors in order to define the unobstructed view polygons, between the current surface pair.

   (d) Calculate the form factor between the surface pair from their unobstructed view polygons, using a method which converts the surface integral into a line integral.

3. Repeat stages 2(a) to 2(e) for all the surface pair combinations.

It is more useful to write equation 5 in terms of the form factor from surface N to surface J. This is done using the simple heat balance:

$$A(N) * F(N,J) = A(J) * F(J,N) \qquad (7)$$

(Similar considerations allow easy calculation of form factors for windows contained within other surfaces).

So:

$$A(N) * RAD(N) = \sum_{J=I,N} A(N) * F(N,J) * RAD(J) - Q(N) \qquad (8)$$

where - Q(N) is any other radiation incident on (N)

It is now necessary to carry out a surface heat balance.

If the amount of heat leaving a surface is QS(N),

then:

QS(N) = A(N) * (RAD(N) - RINC(N)) (9)

substituting 3 in 9 gives

QS(N) = A(N) * ALP(N) * (EB(N) - RAD(N)) (10)

where ALP(N) = EMISS(N)/(I-EMISS(N))

Equation 8 can be substituted into 9 which is then equated to 10 to give the equation set:

RAD(1) * (1 + ALP(1)) - RAD(2) * F(1,2)....RAD(N) * F(1,N)
= ALP(1) * EB(1) - Q(1)/A(1) etc (11)

The equation set 11 can be solved to give the radiosity (RAD) for each surface. This could then be used to obtain the heat flows into the surface. The surface temperature being contained in the black body emissive power (EB(N)). ROOM does not do this, but combines 11 with equations representing heat flows to produce a single matrix representing the thermal balance on the room.

The ROOM matrix

Three types of surfaces are defined within ROOM:

High mass surface - stores & conducts heat
Low mass surface - a pure resistance
Glass (or translucent) - a pure resistance that also transmits shortwave radiation.

The high mass surface requires the rate of heat input at a particular time. That is the surface heat flow as defined by equation 9. Low mass and glazed surfaces are assumed to respond instantaneously so it is necessary to incorporate equations relating their temperature to heat flow through them. So what has to be done is to re-write equations 11 so that the solution yields:

Surface temperatures of low mass elements.
Surface heat flux for high mass elements, (this flux can then be used to predict the temperature of the high mass element at the end of the current time step).

(a) Low mass elements

The heat flow into a surface that does not store heat can be written in the form:

q = ALPHA + (BETA * TAI) - (GAMMA * T) (12)

where
ALPHA = KT_{eo}

BETA = the internal convection coefficient

GAMMA = K + internal convection coefficient

K = thermal conductance from the room surface to the external environmental temperature Teo

Glazed surfaces can be represented using an identical expression to 12 (different constants). The absorbed radiation being contained in the ALPHA term.

(b) High mass surfaces

In this case the surface heat flux is required. This can be obtained from equation 10.

The ROOM matrix is then generated as follows:

(a) Linearise the black body emissive power - equation 4.
(b) For high mass surfaces replace the radiosity term (RAD) in 11 by substituting 10 (linearised EB(N)) into that equation.
(c) In the case of low mass surfaces the heat flow is replaced by equation 12.

What results is an equation set that when solved produces either surface heat flows or temperatures.

Ventilation models

Ventilation models represent the way air moves into a room and within the space. Both depend upon the thermal load in the space. That is the internal convective heat gains. In common with all thermal models convective gains can be independent of temperature, be fixed inputs or vary. The variable gains in ROOM are from occupants and surfaces.

Internal models

The mixed model assumes that all air within the space is at the same temperature. Thus, for example, representing conventional ceiling mounted air distribution systems.

The stratified model assumes that the space is divided into two parts:

A lower region influenced by heat gains from the floor, occupants, low level shading devices, and equipment within the space. Air supplied via mechanical or natural means is assumed to first flow through this region.

An upper region influenced by the room surfaces (other than the floor) the convective gain from lights and any high level shading devices.

Displacement ventilation, floor supply systems and natural ventilation are examples where the stratified model might be appropriate.

The incorporation of heat storage in the air can be a source of instability in the numerical calculation. This is because the thermal storage capacity of the air in a space is usually rather less than that of the enclosing fabric. The obvious way to avoid difficulties in this area is to either use a very small time step or ignore heat stored in the air. Instead ROOM assumes that the surface temperature of the space and all heat inputs are constant over a time step. It is then possible to obtain a simple analytical solution to the room air response.

Things are a little more complex in ROOM when a stratified model is used. In this case it is assumed that the above approach is valid for the upper region, and an instantaneous heat balance is applicable to the lower region.

External models

External air flow models are representations of how the air enters a space. In the case of ROOM there are three considerations:

Infiltration;
Natural ventilation;
Mechanical ventilation.

(a) Infiltration

Infiltration is unwanted air flow through the building. It is represented in ROOM by a default level set by the user at data entry (this is valid because ROOM is a design program not a simulation tool and so we know the infiltration rate). Both natural and mechanical ventilation default to this value if either are not set or are predicted to be

below it.

(b) Natural ventilation

This is the air that passes through the building due to the action of pressures generated by the wind and temperature differences. It is assumed in ROOM that critical design conditions occur when the wind speed is zero so only buoyancy driven ventilation is considered. In addition only two openings are modelled. One at low and one at high level. The flow through these is calculated using the 'stack pressure' difference as defined in the CIBSE Guide, based on the current mean internal temperature (average of low and high regions when a stratified model is used) and external dry bulb. Following the philosophy of the explicit method the air flow calculated at present time is used in the prediction of the future air temperature.

(c) Mechanical ventilation

On the face of it the simplest ventilation model. Provided the fan has been correctly sized the ventilation rate is known. However, things may be more complicated. First the heat gain from the fan (unless only extract is used) will raise the supply air temperature, and secondly it is possible for both mechanical and natural ventilation to occur at the same time. Four possible situations are represented in the program:

Pressurised room, natural ventilation flow from low to high region;
Pressurised room, natural ventilation flow from high to low region;
Extract fan, natural ventilation flow from low to high region.
Extract fan, natural ventilation flow from high to low region.

It is assumed that the mechanical supply or extract remains constant. The pressure loss through the ventilation openings due to the specified mechanical ventilation flow is combined with that generated by buoyancy using a simultaneous solution of the flow equations. The resultant combined flow (usually rather less than the sum of mechanical plus natural ventilation) is used in the analysis.

Plant

The following plant items are simulated in ROOM:

Mechanical ventilation;
Air heating or cooling (no latent transfer);
Chilled or warmed floors;
Air heating or cooling using:

Constant specified supply air temperature
Thermostatically controlled heater or cooler battery.

The former is simply a modification of the mechanical ventilation system, the latter assumes proportional control. The plant simulations are very simple, this is because the intention is to concentrate on what happens in the space, not to size plant.

Internal Gains

Three sources of internal heat are taken into consideration:

Occupants

Lights

Machines/equipment

Occupants

Both sensible and latent gains are considered. The calculated gains are based on data contained in the CIBSE Guide, Ashrae Applications Handbook and Humphreys Henschel and Lee (20). Sensible heat gains are taken as constant when the room temperature is below 20°C and proportional to (37.78 - dry bulb) above that temperature.

The latent gain is the difference between the sensible gain and the heat emitted by the occupants at the specified activity level. There is a limit to the latent heat that can be rejected from the body. This depends upon the difference between the vapour pressure of saturated air at the skin temperature, and the local vapour pressure. If the sensible heat emission is less than the metabolic input and the space humidity is such that the latent emission cannot make up the difference, a state of thermal stress exists. ROOM does not indicate this condition.

Lighting Gains

The input from the lights is a combination of convection and radiation. The exact split being selected by the user when entering data. At present the radiant gain is assumed to fall upon the floor where it is added to the solar gain and distributed following the theory for non-specular reflections. In theory all the convective output from lighting will find its way into the upper zone (when a stratified model is specified). In practice some of this heat will enter the lower region due to mixing between the two zones, and of course task lights may be used. It is therefore assumed that 25% of the convective gain from lighting enters the occupied portion of the room.

Machinery and equipment

Only sensible gains are dealt with. It is assumed that 35% of the gain is radiant. The radiant gain is distributed evenly over all room surfaces other than the floor. All the machinery load enters the lower region when the stratified air flow model is used.

Comfort

The primary objective of the ROOM programme is to predict comfort at any point in a space. There are many definitions of comfort, and at present ROOM uses the term to mean "thermal comfort" as defined by P.O. Fanger (1). Comfort is defined as a state of thermal equilibrium. Heat gains to the body indicate "hot" and "cold" feelings. The human body acts as a heat exchanger with both sensible and latent heat transfers so it is necessary to take the following into consideration:

Dry bulb temperature;
Wet bulb temperature;
Mean radiant temperatures;
Direct radiation;
Air speed.

In general comfort terms humidity is not too important and so the wet bulb temperature can be ignored. At present the prediction of air speed is beyond the scope of programmes for general engineering design. ROOM uses a single velocity input by the user for calculating the convective heat transfer coefficient for the human body.

Similar considerations apply to dry bulb temperature, although a user input is not required as this is calculated during the thermal analysis and assumed constant throughout the occupied zone. Mean radiant and direct radiation effects require additional analysis.

In addition to the thermodynamic parameters, it is necessary to consider the rate of generation of body heat, the level of insulation surrounding the body and the dimensions of the body. The first by selection of occupant activity level, the second by selection of a clothing level and the third by the use of average body dimensions used by Fanger.

The main comfort parameters considered are:

PMV

This index represents how a large population would "vote" the comfort in a space. It has been determined by experiment and then related to the heat balance equation. A PMV of zero means neutrally/comfortable. Positive values indicate warmth (+3 means Hot).

PPD
This statistical parameter is derived from the PMV. It shows that is never possible to achieve perfect conditions. The value of PPD at a PMV of zero is 5%. The ISO comfort

standard (21) suggests 10% as a maximum value of PPD.

While the above are considered good measures of comfort, the engineer should use experience in their interpretation. For example the average PPD for a whole space might be 30%, however close examination of the predictions might show that this was because there were areas near glazing where the PPD was 100%. If the space is one where people choose where they sit or stand, they would, avoid the sunlit areas and so "make themselves comfortable". Similarly, clothing levels are often "adjustable", jackets and ties may be removed and again satisfaction achieved.

Prediction of thermal comfort requires the solution of Fanger's comfort equation (1). This is easy to do once all the inputs are known. In addition to dry bulb temperature and air speed it is necessary to predict:

The mean radiant temperature at the point where the occupant is positioned;
The amount of direct solar radiation falling on the occupant;
The amount of diffuse solar radiation incident on the occupant from reflections within the space.

The first of these requires calculation of the radiation interchange between the human body and the room surfaces. This means calculation of the form factor between the body and room walls. To do this Fanger introduces the concept of 'angle factor' which he obtained experimentally. This factor is related to the relative positions of the body and surface. To ensure a heat balance it is essential that the angle factors for any specified occupant location add to unity. To achieve this in practice it was found necessary to divide each surface into a large number of elements. This meant that fairly long run times were required to obtain comfort plots similar to that shown in figure 4. ROOM makes use of what is thought to be a reasonable approximation. The human body is represented as a small sphere. Calculation of the angle factor to a complete surface is then a fairly simple matter. The main objection to this method might be that the body reacts to radiation from different directions with varying levels of sensitivity. This is accepted, but in general the designer has no knowledge of which way occupants will face, so occupant orientation needs to be averaged. The sphere method goes someway to achieve this. This simplification does not remove all difficulties in the calculation of radiant temperatures. It is still necessary to consider the effect of surfaces that obscure the occupant from direct solar radiation or other surfaces. This must be dealt with in a similar way to the calculation of the intersurface radiant exchanges.

CONCLUSIONS

The calculation of comfort at any point in a space requires the use of a relatively complex computer program. ROOM goes much of the way to achieving the objective, the main limitation is hardware. It is planned to adapt the program to run on a workstation, in which case variable convection coefficients, and possibly a detailed air-flow model will be added. It has however, to be remembered that ROOM is a design tool and as such needs to meet the requirements of the engineer, not the research worker.

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D SHALLOE - Conversion from VAULT to ROOM Level 1

C COOMBER - Addition of the comfort equations.

G DAVIES - Sphere to surface radiation model.

F COUSINS - Validation of obstructed surface form factors

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