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COMPUTER SIMULATION OF ENERGY USE AND THERMAL CLIMATE IN GLAZED SPACES

Kjell Kolsaker
Division of Heating and Ventilation
Norwegian Institute of Technology

Hans Martin Mathisen
SINTEF Applied Thermodynamics

SUMMARY

In buildings with stratified room air temperature, improved accuracy in calculated annual energy consumption and air temperatures should be obtained by including a two zone or linear temperature stratification model in the simulation programs.

This paper reviews the principles of a two zone model and a model with linear temperature stratification and shows how the latter can be implemented in existing simulation programs with one air node in each zone. The model is implemented in the Norwegian simulation program FRES (Flexible Room Climate and Energy Simulator).

The model is applied on an existing atrium, and the results are compared to measurements. Two models are tested, constant air stratification and a model using a separate heat balance for the air volume near the floor. The results are promising, and some further work should be carried out to evaluate the model for a wider range of conditions.

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BACKGROUND

The use of glazed atria has become more common during the last years. One typical characteristic of these type of premises is that the air stratifies with a temperature increasing with the height. The displacement ventilation system, which have the same quality has also become common in use. It is therefore a demand for simulation programs for calculation of the annual energy use and peak loads in such situations.

Rooms with displacement

Displacement ventilation systems have proved to be efficient means of removing contaminants and excess heat. In this system, air is supplied to the room with low velocity through openings close to the floor at a temperature lower than the room air temperature. Usually the buoyancy forces will influence the flow.

Displacement ventilation, Fig. 1, is secured by supplying the ventilation air at a temperature that is always lower than the air temperature in the zone of occupation. The necessary heating of the room is usually provided by the use of panel heaters under the windows. Close to the heat sources in the room the air will rise upwards due to buoyancy. Often contaminants are released from these heat sources, for instance people. Then the contaminants will be transported towards the ceiling, where the exhaust opening is placed. More air means that the rising warm air can be fed with fresh air to a higher level before it recirculates and feeds itself. In this way the air in the room will be stratified into a lower zone with fresh air, and a upper zone with contaminated air. Downward convection flows moves contaminants from the upper part to the lower part of the room. With a given heat load the height of the lower zone depends on how much air is supplied.

Another consistency of the displacement system is that the supply air temperature required for cooling is higher than for complete mixing because it is supplied directly to the occupied space i.e. free cooling could be used for longer periods of the year. However, in some types of rooms relatively large airflow rates are needed to obtain this improved efficiency. This is because the lower zone has to be "lifted" above the height of the people in the room, see Fig. 1.

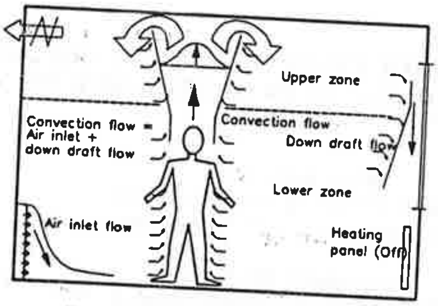


Fig. 1. The principle of displacement ventilation.

The concept of displacement also means that the air near the floor is driven by the buoyancy forces acting on the supply air due to the low velocity. Parameters among others important to thermal comfort are room air temperatures, temperature gradients and air velocities. If the ventilation airflow rate is kept constant and the heat load is increased the vertical temperature differences and the velocity along the floor will increase. This means that the vertical temperature gradient and the velocity close to the floor limits the cooling ability of the displacement system.

Some parts of this paper are discussed in more detail in [4], [5] and [6].

Glazed atria

In glazed atria the ventilation airflow rate is often zero. Then the temperature stratification is maintained only by the convection flows, i.e. flows from heat sources like windows and other surfaces heated by the sun and flows directed downwards due to surfaces with a temperature lower than the room air temperature.

If the atrium is ventilated by air blown in with an impulse strong enough to cause mixing of the air, a uniform temperature will be the result.

The dominant heat loss in an atrium is due to transmission losses through glazing and infiltration losses.

Accordingly:

Under conditions with complete mixing (heating in the lower part and/or significant down draft, no solar radiation) simulations with programs using one node to represent the air temperature, should give adequate results for air temperature and energy demand for potential heating. During the hours of the year when there is some solar radiation and a heating demand, (i.e. some stratification), we will under-estimate the heating demand. We need to heat the lower part in spite of that the upper part is thermally comfortable.

Under conditions with poor mixing, the simulated air temperature will represent the temperature in the upper part of the atrium (more so in a linear and a core atrium than in an attached and envelope atrium). The calculated thermal climate gives us little information about the climate at floor level.

MODELS

Two-zone model

A two zone mixing model could be used both to describe the concept of ventilation and to define its effectiveness. This simple stratified model has been experimentally verified by earlier work, Mathisen [4].

However, the two-zone model does not always adequately simulate the thermal conditions in the room, but there are other methods that can be used. A model that uses the general equations for heat and mass transport (including a proper turbulence model) can give good results. A main drawback with such models is the excessive CPU-time that makes them unsuitable for annual calculations.

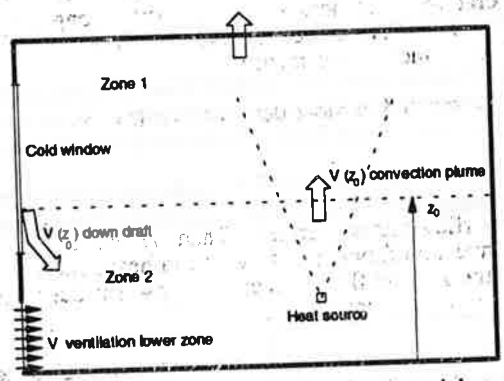


Fig. 2. Sketch of the principles for the two-zone model.

The two zone model would be close to reality in rooms with rather clear temperature stratification, as in many plants in the metallurgical industry - with well-defined convection sources and displacement ventilation. In rooms with displacement ventilation, with less well-defined convection sources there is no defined step in the vertical temperature gradient in the room. It is rather a more continuous increasing temperature corresponding to the height above the floor.

The height of the lower zone in the two-zone model is governed by the mass-balance for the air feeding the thermal convection plumes in the lower zone. This is calculated from:

$$\dot{V}(z_0)_{convection\ plumes} = \dot{V}_{ventilation\ lower\ zone} + \dot{V}(z_0)_{down\ draft} \tag{1}$$

z_0 is the critical height above floor level, and $\dot{V}$ is a volume flow rate.

The equations for the flow rate in the convection plumes depend on the kind of convection source and the shape of the source of the flow. However, it is only possible to use simple algebraic equations for flows in space with no vertical temperature gradient or with prefixed gradients. In general simple algebraic solutions are not possible.

The air exchange between the zones is represented by:

$$\dot{V}_{ventilation} + \dot{V}(z_0)_{downdraft} + general \quad turbulence \tag{2}$$

In the kind of rooms that are treated here, large/tall rooms with considerable heating loads, the most viable ventilation system will be displacement ventilation. Here the ventilation air which is somewhat cooler than the room air, is let into the room at low speed. In an atrium case, there is no mechanical ventilation system. The use of infiltration in cold outdoor periods and venting with hatches in warm outdoor periods will enable the system to function as a displacement ventilation system, most of the time.

Linear temperature stratification model

As mentioned, simple algebraic calculation of convection flows in rooms with a changing vertical gradient is difficult. Experiments and field measurements have shown that the profile in many cases becomes more or less linear, as shown in principle in Fig. 3.

The reason is that the heat sources are distributed by radiation to the surfaces in the room. Thermal transmission through walls and windows may also influence on the thermal stratification.

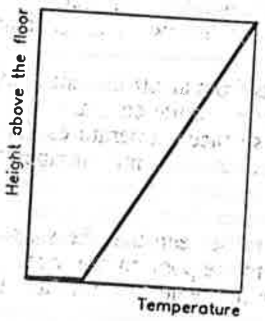


Fig. 3. Temperature profile in rooms with weak and distributed heat sources and little mixing of the air.

An air temperature distribution dependent on the level above floor level and some characteristics about the room is also an interesting alternative, and may be more simple to implement than a two or multi-zone model.

In Fig. 4a) examples from measurements in a test room with displacement ventilation is shown. The dimensionless temperature $\frac{T_{actual}-T_{supply}}{T_{exhaust}-T_{supply}}$ is plotted against the height above the floor. The temperatures were measured from the surface of the floor to a point close to the ceiling.

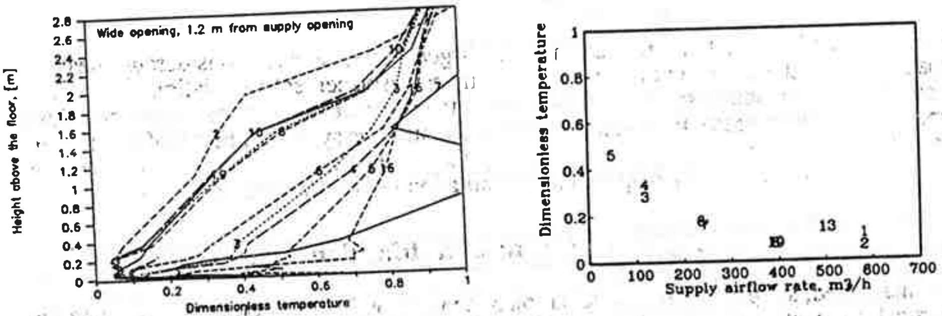


Fig. 4. a) Dimensionless temperatures in a room with displacement ventilation plotted against the height above the floor. b) Dimensionless temperatures plotted against the supply flow rate. The numbers in a) refer to the numbers in b)

It can be clearly seen from Fig. 4a) that the shape of the profile varies from test to test. If the curves are related to the airflow rates as shown in Fig. 4b) it is obvious that the shape of the curves depends on the airflow rate. What actually happens is that when the supply airflow rate is reduced the height of the lower zone decreases when the entrainment to the convection flow is constant.
The profiles in fig. 5 are measurements from an atrium under different solar and thermal conditions. In this case the air flow rate is quite constant. The gradient is small during the night with small heat load and low surface temperatures. During the day, the gradient can be conciderable, with temperature near the inlet temperature near the floor and a high temperature (45 C) at the top.

From experimental data [4] it can also be seen that the shape of the profiles is almost the same in all positions. This is due to the poor entrainment of ambient air in the flow. However, the quality and the position of the heat sources and the ceiling height plays an important role for the shape.

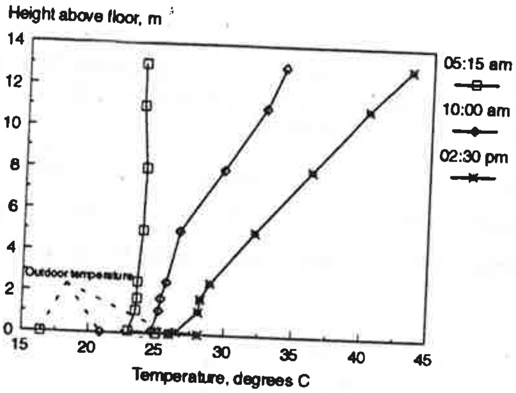


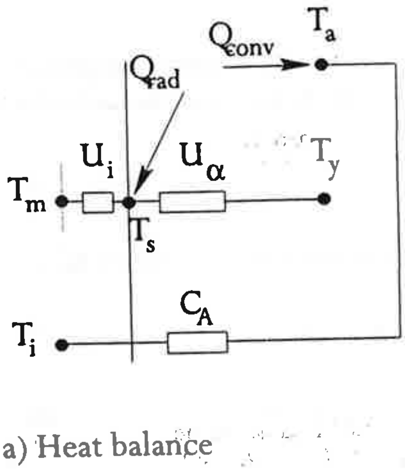
Fig. 5. Temperatures in an atrium (see fig.7) plotted against the height above the floor under different conditions the same day (18-july, see fig.8).

THE MODEL IMPLEMENTED IN FRES

FRES (Flexible Room climate and Energy Simulator) is a dynamic simulation program for multi-zone buildings developed at SINTEF Division of Heating and Ventilation. The program is a tool for HVAC consultants and building designers, widely used in Norway. The objectives are to implement a simple and still reliable model that can improve the existing single-temperature zone model and make it a better tool for atrium simulation.

The proposed linear stratification model is implemented in FRES as described in the previous sections. For calculation of heat transfer between room air and room surfaces, the temperature difference between surface and room air at the mean height of each surface is used.

The convective heat flow into each surface is calculated for the stratified case, ensuring the correct heat balance for the whole building. The stratification will for example make floor and ceiling "feel" different air temperatures. To take care of this, the equation for convective heat flow is modified, taking into account the linear stratification model. This is quite simple, as will be shown here.



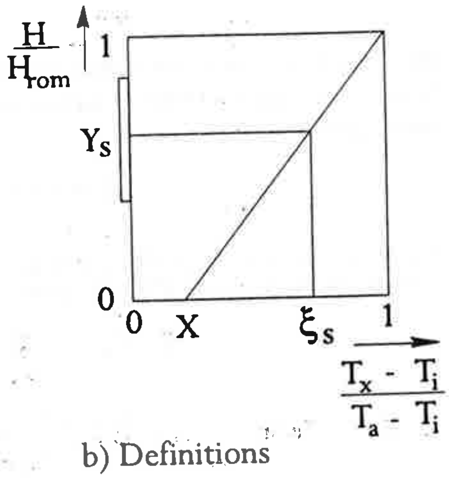


Fig. 6. The model implemented in FRES.

The temperature T_x near the floor is given by the equation

$$T_x = X T_a + (1 - X) T_i \tag{3}$$

where X = Dimensionless temperature near the floor; $X = \frac{T_x - T_i}{T_a - T_i}$
T_a = Temperature in air outlet
T_i = Temperature in air inlet

By geometry, mean dimensionless height of a rectangular surface can be expressed by

$$Y_s = \frac{H_1 + H_2}{2H} \tag{4}$$

where H_1 = Height of lower side of rectangle
H_2 = Height of upper side of rectangle
H = Height of room

Mean temperature for the air T_y "felt" by the surface can be expressed by the inlet temperature and the air outlet temperature:

$$T_y = \xi_s \cdot T_a + (1 - \xi_s) \cdot T_i \tag{5}$$

The value ξ_s is a local stratification number for the surface s. This number is expressed by the stratification number X for the room and the mean height Y_s for the surface by simple geometry:

$$\xi_s = X + Y_s - XY_s \tag{6}$$

The energy balance for an air volume with one single surface s can be expressed for the surface and the air volume by the equations

$$U_i(T_m - T_s) + U_a(T_y - T_s) + F_r Q = 0 \tag{7}$$

$$C_a(T_i - T_a) + U_a(T_s - T_y) + (1 - F_R)Q = 0 \tag{8}$$

where

T_m = The neigbour temperature inside the wall
T_s = The surface temperature
T_i = Air inlet temperature
T_y = Air temperature at level $Y = Y_s$ (mean surface height)
T_a = Air temperature at level $Y = 1$ (Air outlet)
Q = Room heat load
U_i = Heat conductance from surface to nearest wall node
U_a = Convective heat conductance for the surface
C_a = Heat capacity rate of inlet air
F_r = Fraction of radiation for room heat load

This model uses the temperature difference $(T_y - T_s)$ instead of $(T_a - T_s)$ as the driving force for the convective heat transfer between room air and the surface. If for example a room faces to the upper part of an atrium and another room faces to the lower part of the same atrium, the model will catch the different conditions of theese two rooms.

A combinition of the previous equations results in the following equation system, which can be easily extended to multiroom models with a variable number of walls and a free air flow pattern:

$$\begin{bmatrix} -U_i + U_a & U_a \xi_s \\ U_a & -\xi_s U_a + C_a \end{bmatrix} \begin{bmatrix} T_s \\ T_a \end{bmatrix} + \begin{bmatrix} U_i & U_a(1 - \xi_s) \\ 0 & C_a - (1 - \xi_s) U_a \end{bmatrix} \begin{bmatrix} T_m \\ T_i \end{bmatrix} + \begin{bmatrix} F_r \\ 1 - F_r \end{bmatrix} Q = 0 \tag{9}$$

The local stratification number ξ_s must be calculated for every surface in the room for a given X. You can easily see that $\xi_s = 1$ for the ceiling for all values of X. For the floor, $\xi_s = X$. Further, the case $X = 1$ (no stratification) result in $\xi_s = 1$ for all surface positions. This case reduces the problem to a normal single zone model.

As discussed in the previous section, X is a function of both the airflow rate and the heat load. At the moment, a constant value of X is used. A model for correlation to the floor temperature is implemented as an option. The model is proposed by Mundt [12], based on a simple energy balance for the air volume close to the floor, neglecting induction of room air into inlet air:

$$C_a(T_i - T_x) + U_a(T_{floor} - T_x) = 0 \tag{10}$$

where T_{floor} is the floor surface temperature. This equation is solved for the air temperature T_x near the floor using a mixed air inlet temperature for all air inlets and the floor temperature calculated by FRES. The calculated air temperature T_x is used when calculating $X = \frac{T_x - T_i}{T_a - T_i}$

SIMULATIONS AND DISCUSSION

An atrium, the ELA building at the Norwegian Institute of Technology in Trondheim (Fig. 7), has been simulated over a period and compared to measurements.

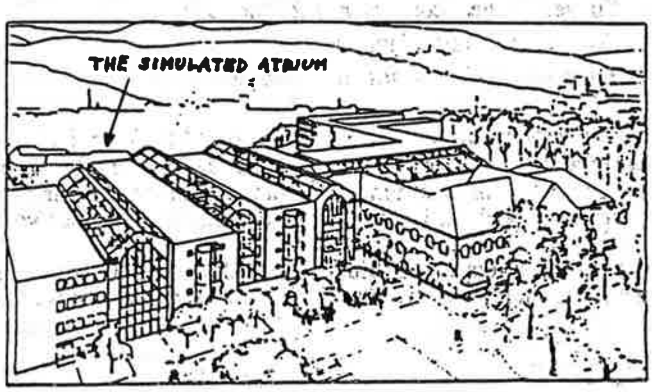


Fig. 7. The ELA building.

One single atrium was modelled. Solar insolation and climatic data are measured over a 3 day period with quite warm weather and clear sky conditions. Three simulations are presented:

- Ordinary one zone model, $X = 1.0$
- Constant air stratification, $X = 0.2$
- Variable air stratification, $X = f(T_i, T_{floor}, C_a)$

The simulations uses measured outdoor temperature over a 3 day period as input. Solar data are calculated by FRES. A Cloud Cover Factor is chosen so the calculated total radiation on a horizontal surface during a day is close to the measured value.

The results are presented in the figures 8a, b, c and d. The simulated period is a quite warm period with day temperatures over 20 C, preceeded by a colder period. There was no heating demand exept by the first night. The controller setpoint in the atrium is 15 C.

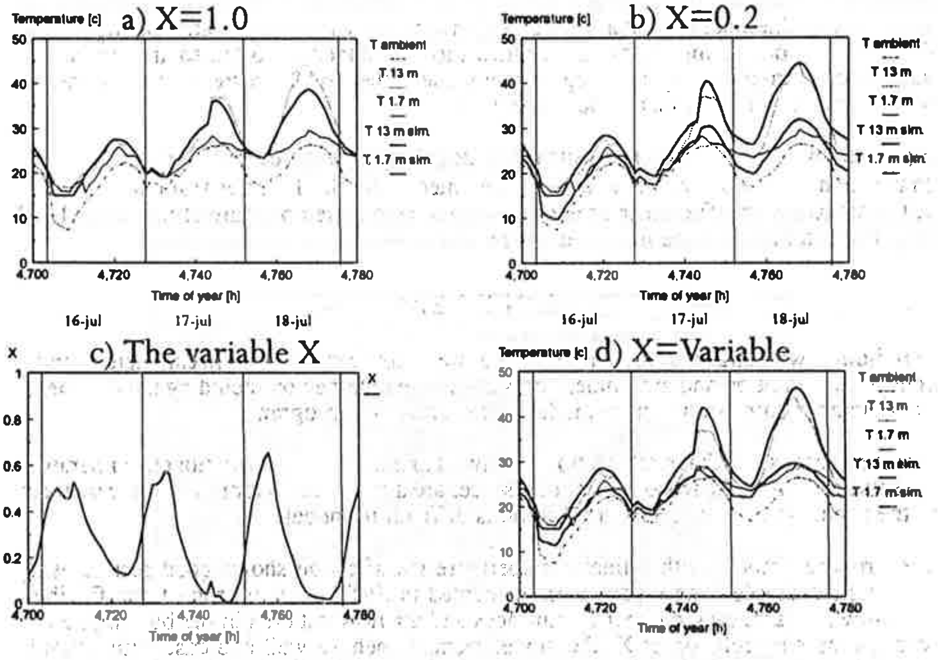


Fig. 8. The simulation results.

Fig 8a shows the temperature using $X = 1.0$. This simulation is identical to a one zone simulation with no air stratification model. The thick curve is the simulated air temperature. You can oberve the effect of heating the first night. The air hatches were fully open the first period, using a measured air exchange rate of about 4 exchanges/h. At the time $t = 4743$ h, and the rest of the period, the hatches were closed, using a measured air exchange of about 0.45 exchanges/h. That results in a temperature rise of 6-7 C which can be found in the graph. In the period with closed hatches, the simulated temperature is slightly lower then the measured value.

Fig 8b shows a simulation with constant $X = 0.2$. This results in two simulated temperatures, one corresponding to the upper level and another corresponding to a level 1.7 m above the floor. The upper level temperatures are higher than the temperatures from the previous simulation with $X = 1.0$, due to the fact that convective heat transfer is connected to the average air temperature outside each surface. Since this temperature is lower then the upper level air temperature, the calculated heat loss is lower. This results in a higher temperature in the latter case.

The calculated temperature at a level of 1.7 m is too low in the night and too high in the day. The reason for this is that the stratification is connected to the solar load, which varies from zero in the night to a significant value in the day. To correct for this, a model which includes the heat load should be applied.

Fig 8d shows a simulation using such a model. The model is described in the previous chapter, and the resulting value of X is presented in fig 8c. It varies from close to zero in the day and about 0.6 during the night. The simulated temperature at the 1.7 m level is now much closer to the measured value.

CONCLUSIONS

In buildings with stratified room air temperature, improved accuracy in calculated annual energy consumption and air temperatures should be obtained by including a two zone or linear temperature stratification model in the simulation programs.

Measurements show that stratification with two separate zones with homogen temperature are seldom found. The reason is that heat sorces are distributed by radiation to the surfaces in the room. In addition, such a situation is difficult to model.

The proposed model with a linear temperature stratification shows good results using one single example. The model as implemented in FRES, is quite robust and flexible, and allows an arbitrary number of rsurfaces and air flow patterns in the building. Even with a simple correlation of X, the model seems to behave well in a case with variable conditions. A few other cases have also been tested, but more testing work remains before the model can be released.

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