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Displacement ventilation in glazed atria matrix modelling technique for energy, temperature and peak load

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DISPLACEMENT VENTILATION IN GLAZED ATRIA

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SUMMARY

The displacement ventilation concept was gradually introduced in Norway in the early 70's. Adequate working environment was now possible to reach economically in heavy industry. Then in the early 80's displacement ventilation was introduced as a low draught and low noise concept for commercial buildings and ventilation installations.

For glazed atria and in halls of considerable height the concept leads to significant reductions in cooling load and cooling energy consumption. The concept was first used for design of the glazed atria in Royal Garden hotel in 1982. Later it has been developed into a powerful all-purpose matrix based simulation concept (Royal ELISE) for hour by hour simulations for a design day or a design year.

It is used as an advanced engineering design tool. The paper will demonstrate the basic outlines of the concept, and how it takes in account the displacement ventilation effects.

Hour-by-hour simulations in a glazed atrium placed in Washington DC is shown. The displacement ventilation aspects of cooling and heating loads, energy consumption and temperature variations are discussed.

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Introduction

The displacement ventilation concept is based on moving the room air in a specific direction. In large halls, glazed atria or industrial plants the overall air flow is vertical. I.e. fresh and cool low turbulent air is supplied at the floor level. The warm and often polluted air is extracted at the ceiling level.

Normal air conditioning design has earlier been based on the concept of mixing ventilation. I.e. the fresh air is thoroughly mixed into the whole room. In principle the temperature and pollution in the exhaust will be the same for displacement ventilation and mixing ventilation. I.e. if we use the same air flow rate and inlet conditions. The principal feature of displacement ventilation is that it improves the conditions in the upstream part of the room.

Very advanced simulation programs treat temperature and pollution as uniform throughout the room. Todays research is refining details on wall layer buildup, conduction and local heat transfer. This paper, however, stresses the importance of modifying the room air flow model so that the possible effects of displacement ventilation can be seen and taken into account during design.

We will demonstrate a matrix concept of a very flexible room model where also displacement ventilation effects are taken into account. We have denoted it the RE-concept (Royal ELISE "Energy and Loads In Separate Elements" and it is based on dividing the room into air volumes zones. Fixed air flow rates are set between the different zones. By the use of a matrix set-up of the interconnection between ZONES, WINDOWS, SURFACES and thermal MASSES, the model becomes both simple flexible and powerful.

Finally we will show typical results obtained in a 2,000 m2 (21,500 sqft) glazed atrium placed in a 85,000 m2 (915,000 sqft) shopping mall. The mall is simulated both in Oslo/Norway and in Washington DC. Temperature stratification, cooling load, heating loads, yearly running hours and typical effects of using normal versus reflected glazing and displacement versus mixing ventilation are demonstrated. All alternatives are based on hour-by-hour simulations over a year.

The RE matrix modelling concept

The history of the concept applied for large space air conditioning dates back to Norwegian Institute of Technology in 1982. The aim was to simulate the effects of displacement ventilation in the three glazed atria in the Royal Garden hotel being built in Trondheim. The particular room model has since then been refined. Up to now its typical use has been for glazed atria design. To give a very brief model introduction, we have included the following:

In general the energy equations are linearised for the room model and put into a matrix system as shown in figure 1.

In short term we can write the equation system as follows:

$$A * O + P * u = 0$$

The equations are solved with Gauss-Jordan matrix method:

$$O = A^{-1} * P * u$$

and

$$O = R * u$$

where R is the result matrix.

Room modelling by direct input in the matrix

The room model is divided into four variable types to handle the air flows, the convective and radiant heat transfer and the thermal capacity in the building structure. The four types are:

- ZONE (air-zone)
- SURFACE (a surface area with no thermal capacity. It is connected to the zones through convective heat transfer, connected to other surfaces through radiation and to masses through conduction)
- WINDOW (a transparent surface, i.e. it is connected with the outputs from the solar simulation procedures).
- MASS (thermal capacities)

A key to flexibility of the concept is that each type may have as many elements as the user wishes. I.e. as many zones, surfaces, windows and masses as required to describe the system. In addition to this the user specifies which elements are linked and how they are linked together.

Example 1.

An example could be a room model with two zones (nz = 2), four surfaces (ns= 4), three windows (nw = 3) and four masses (nm = 4). An increasing number of elements can mean more accurate calculations, however, the number of inputs increase and also the simulation time. The matrix model increases size with the square of sum of elements, i.e. $(nz + nm + ns + nw)^2$. A typical layout of the system matrix is shown in figure 2.

The first quadrant to the upper left of figure 2 takes into account the rate of flow (m3/s) between the different air-zones. The next quadrant handles area (m2) ties between surfaces and air-zones. the third handles area (m2) ties between inside windows and air-zones. The fourth quadrant in the middle handles the radiation between all surfaces. Angle factors are accounted for. To ease the input work, a preprosessor has been developed. The user works with the submatrixes, and the preprosessor builds the system matrixes.

Why use the matrix concept?

The matrix concept is of course just describing the ordinary energy balance equations for a room model. Why we find it so useful is, however, that it gives the model set up an enormous flexibility:

a) by allowing the types (ZONE, SURFACE, WINDOW, MASS) have as few or as many individual elements as practical possible.

b) by allowing them to interconnect in just the way that gives the best model description of the physical system.

For instance:

a) <u>Wall with different materials</u>: let the different masses of each material connect to each other with their individual thickness and thermal conductivity coefficient.

b) <u>Building with energy storage in the ground</u>: let the zone with the exhaust air connect through a surface to one mass for the ground - or rather many inter connecting masses see figure 3. For instance one mass (m3) for the Ø 100 mm of earth closest to the pipe; the next mass (m4) could demote the Ø 500 mm outside of the first mass. The other mass (m5) could be a Ø 2,000 mm thick earth layer around the second mass.

Notice how powerful the concept becomes also for solar system applications. The ordinary part of the building and air conditioning system simulation will include

- <u>weather data</u> for simulations of specific days of a whole year
- <u>air handler data,</u> i.e. heating coils, cooling coils, air-to-air heat recovery units etc.
- <u>building controls,</u> i.e. temperature and humidity controls, control strategies and running schedules. Thereby it will be fairly

simple to analyse how and ordinary building will operate together with an active or passive solar system which also can load a heat storage in the ground.

Greenhouse simulations

We have earlier used Royal ELISE for simulating solar shading and solar energy systems in Dutch greenhouses. The greenhouse temperatures were simulated hour-by-hour over several years. Excess heat was transferred into a thermal ground storage during periods of overheating and utilised in the heating season.

We underline that the simulation program Royal ELISE has 4 matrix models which allows us, to some extent, to take changing conditions into account. This could, for instance, be changes in air flow day and night and major changes in air flow pattern summer versus winter.

Glazed atrium in shopping mall

To illustrate the effects of displacement ventilation a simplified glazed atrium with 2,000 square meter (21,500 sqft) ground area is analysed. The atrium is actually a part of a 85,000 m2 (915,000 sqft) shopping mall "Oslo City" in Oslo/Norway.

Figure 4 shows the atrium model containing a total of 6 zones, 6 windows and 8 surfaces. Each surface is connected to a mass. Notice that surface s6 and s7 are on each side of mass m6. The masses m1 through m5 have a thermal connection to the surrounding office building, which is year-around held at 24°C (75.2°F). Table 1 shows the window and wall surface areas.

The total internal heat transfer between the atrium and the surrounding office building is 10,300 W/K, (19,500 Btu/h°F). The total heat transfer to the outside air is mainly through the windows and amounts to 10,400 W/K (19,700 Btu/h°F). In addition comes an air infiltration rate of 2.5 m3/s which approximates 0.2 air changes per hour.

The window U-values are 3.0 W/m2K (0.53 Btu/h sqft°F). A partly reflecting glazing is used giving the windows a shading coefficient, Sc = 0.55 (relative to a single pane clear glass). During direct sun shine an internal shading is used; which gives a total of Sc = 0.36. A special case using normal clear double pane glazing is also simulated. The total shading coefficient without vs. with the internal shading in use is then 0.90 and 0.58.

The building is simulated to be in use 24 hours a day, 7 days per week the whole year. The supply air temperature is kept constant at 14.0°C (57.2°F). During the heating season the minimum zone air temperatures are kept at 18°C (64.4°F).

Figure 5 shows the air handler and the air distribution and exhaust ducting. The heat recovery wheel is controlled so as to minimize the energy consumption in the air handling unit. The recovery effectiveness is 50% for both temperature and humidity, i.e. also for enthalpy. The figure gives the basis for our displacement ventilation. The internal air flows between the zones are marked. Fully mixing is assumed within each zone. The recirculation, or "down-mix", between zones is significant. However, the sizes given here will only to a small degree change the results from an ideal displacement flow from zone 1 through 5.

Figure 6 indicates how we have simulated a mixing ventilation concept. This has been done by increasing the recirculation flows between the zones 10 times.

3-day summer load simulation

Design summer conditions with a clear sky are simulated as follows on date June 15:

Oslo : outdoor temp. max. 25.2°C (77.4°F)
" " min. 13.2°C (55.8°F)
humidity (dew point 14°C) 0.010 kg/kg (lb/lb)

Wash, DC : outdoor temp. max. 33.0°C (91.4°F)
" " min. 23.0°C (73.4°F)
humidity (dew point 18°C) 0.013 kg/kg (lb/lb)

The maximum zone air temperatures that were reached are given in table 1. There are several interesting aspects to observe. We will analyse the differences between the simulated cases, rather than the specific temperatures:

a) <u>Mixing ventilation</u>, case IV. The down-flow rate is as much as 5-10 times the net up-flow ventilation rate. Notice that there is still a quite significant difference in temperature between the zones.

b) <u>Displacement versus mixing ventilation</u>, case II and IV. We notice that the air temperatures of zone 1 through 3, i.e. floor 1 through 5 are kept significantly lower by use of displacement ventilation. This gives a significant improvements in cases where the temperature on the upper floors are less critical.

The zone 5 temperature of displacement ventilation is 1.4 K higher than for the mixing ventilation (case II vs. IV). The mixing ventilation mixes the heat into the lower zones. It will thereby directly increase the cooling loads in the surrounding offices and shops by the heat conduction through the walls. With ideally insulated walls (and windows) between the atrium and the surrounding building, the top zone temperature of case II and IV would be the same.

c) <u>Normal clear glazing:</u> Case II compared with case III shows that the choice of glazing will have a significant impact on the zone temperatures. Increased zone temperatures will increase the loads on the surrounding building.

d) <u>Climatic variations:</u> We notice that with the same supply air temperature the zone temperatures with the building placed in Oslo/Norway (60 degrees North) will be kept about 1.5 K (2.7°F) lower than in Washington DC (38 degrees North).

Simulations over a year

Simulations have been done to show the aspects of temperature, loads and energy consumption over a year both for cooling and heating. These are shown in table 3. For Washington DC the WYEC-year is used, and for Oslo the meteorological year 1964 is used.

Air supply temperature is kept at 14°C (57.2°F) year around. No local cooling is provided. Local heating ensures that zone air temperatures never fall below 18°C (64.4°F). Typical simulation time on a 80386-computer with 80387 math-processor (25 MHz) is 10 minutes with a model of this size.

The table 3 presents the results in three parts. The upper part shows the maximum simulated zone air temperatures and how many hours per year the temperature has exceeded 26°C (78.8°F).

The table also summarises the cooling and heating load conditions, i.e. the maximum load size, how many hours per year there has been a need for cooling or heating - and finally the yearly energy consumption for heating and cooling. The Royal ELISE simulations have used local heaters in each zone so that the air temperatures never go below 18°C (64.4°F).

Discussion

First of all we notice that there is a very good agreement with the results from the three day summer load simulations in table 2 and the maximum temperatures in table 3. This indicates that for this case the chosen conditions for the three day simulations coincide remarkably well with the chosen reference years.

<u>Glazing has a significant impact</u>

Comparing the cases II and III, i.e. reflective glazing versus clear glazing shows that the reflective glazing significantly lowers the zone temperatures during the cooling season. The reflective glazing gives the same maximum heating load and only gives a marginal increase in heating energy consumption.

User-friendly zones

The Washington DC simulations show that only displacement ventilation gives acceptable temperature conditions on the lower floors. The temperature conditions in the upper zones, i.e. 3 through 5, seem to be too high for normal use.

If, however, the upper zones are primarily used for shorter stays, like bridge crossings over the atrium from one part of the building to another, the higher temperatures could be acceptable. We also stress the fact that due to fire and thereby smoke hazards the upper parts of an atrium will often be closed.

Midsummer outdoor air cooling

As indicated for case III, the maximum temperature will be above the outdoor temperature. This actually means that outdoor air can be used directly for cooling the upper zone(s). Be careful, however, so that there is no mixing of the humid outdoor air into the lower zones.

Climatic variations

Case I versus II show that moving the mall from Oslo to Washington DC more than doubles the cooling load. Still the high temperatures in the upper zones in the Washington case might be questionable. The necessary running hours for the cooling will be more than doubled.

Notice also that while the cooling load increases 2-2.5 times, between Oslo and Washington DC, the cooling energy increases close to 10 times.

The heating season has the opposite effect. While in Oslo the heating load is close to 40% higher than in Washington; the heating energy consumption is approximately 3 times as large.

Atrium heating temperature

The chosen minimum air temperature of 18°C in the atrium means that a significant amount of heat is transferred from the surrounding building to the atrium. The surrounding building is assumed to have a constant 24°C temperature.

By increasing the minimum atrium temperature to 20°C; the atrium does not only increase its loss to the outside - but it also reduces its gain from the surrounding buildings. Simulations which we have done with 20°C minimum indoor temperature give for heating:

Case I (Oslo)	721 kW	1201 MWh/yr
Case II (Washington)	547 "	494 "

This indicates that the heating loads are moderately increased (11% respectively 17%) by the 2K increase in temperature. However, the increase in energy consumption are significant, i.e. 45% respectively 84%.

Floor cooling possible

Floor heating is often installed for maintaining a high degree of comfort. Especially for glazed atria the comfort aspect is interesting - in addition to the fact that water based floor heating allows us to utilise low temperature heat. In Norway we have not only used the floor piping for heating purposes - but also for cooling of direct or indirect solar heated floors. In quite a number of the new atria, floor cooling is installed

Some of our additional computer simulations, which not shown here, indicate that by utilising 1,100 m2 (11,800 sqft) for floor cooling the zone air temperatures could be held at the levels indicated in table 2, even with a 35% reduction in air flow rate. However, note that floor surface temperatures should not be lower than 20°C (78°F)

The most significant comfort effect of floor cooling is, however, not that it lowers air temperatures; but it lowers the mean radiant temperature. It thereby has an interesting effect on the operative temperature. This effect we will try to show in a later paper.

Conclusions

Computer programs that can simulate the effects of displacement ventilation are a must when it comes to system design and cooling load design in halls of considerable heights like glazed atria.

It has been demonstrated how a building simulation program with displacement ventilation may be set up. The program flexibility using the RE-concept is emphasised.

Glazed atria represent spaces where displacement ventilation are of considerable interest. The paper shows how displacement ventilation can significantly reduce cooling load and thereby installation costs when increased temperatures are accepted in the upper zones.

The possibility of introducing floor cooling is pointed out as being another interesting aspect. This is also a feature which can be taken into account when using the RE-concept.

Climatic variations between Oslo and Washington DC are shown. Of particular interest are the possible energy savings (30-40%) by reducing the minimum atrium temperatures from 20°C to 18°C.

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3) Mathiesen, H.M. "Case studies of displacement ventilation in public halls" ASHRAE Trans. Vol.95 Part II, 1989.

4) Skaret, E., Mathiesen, H.M., "Ventilation efficiency - a guide to efficient ventilation", ASHRAE Trans., Vol.89, Part II, 1983.

SURFACES AREAS			WINDOW AREAS		
s 1	1,544 m2	(16,500 Sqft)	w 1	113 m2	(1,210 sqft)
s 2	2,677 "	(28,600 ")	w 2	195 "	(2,090 ")
s 3	2,032 "	(21,700 ")	w 3	165 "	(1,770 ")
s 4	1,679 "	(18,000 ")	w 4	165 "	(1,770 ")
s 5	1,755 "	(18,800 ")	w 5	172 "	(1,840 ")
s 6	807 "	(8,600 ")	w 6	2,828 "	(30,300 ")
s 7	3,270 "	(35,000 ")			
s 8	795 "	(8,500 ")			

Table 1 Window and surface area. Surfaces include floors, inner walls and inner ceiling. The total floor area, i.e. ground floors and upper floors, is 4,900 m2 (52,400 sqft).

CITY:	OSLO	WASHINGTON DC		
CASE	I	II	III	IV
GLAZING:	REFL.	REFL.	CLEAR	REFL.
VENTILATION:	DISPLACE	DISPLACE	DISPLACE	MIX.
ZONE 5	29.8	32.5	36.1	31.1
" 4	28.3	30.4	34.0	30.5
" 3	27.0	28.7	31.9	29.7
" 2	25.6	27.0	29.7	28.8
" 1	24.6	26.2	28.3	28.2
ZONE 0	24.5	25.8	27.6	26.3

Table 2. The maximum zone air temperatures (°C) for summer load conditions.

CITY:	OSLO	WASHINGTON DC		
CASE	I	II	III	IV
GLAZING:	REFL.	REFL.	CLEAR	REFL.
VENTILATION:	DISPLACE	DISPLACE	DISPLACE	MIX.
<u>Temperature</u>				
Zone 5, max.				
temp. (°C)	29.5	32.5	36.0	31.0
exceed 26°C (h/yr)	282	1422	2313	1126
Zone 3, max.				
temp. (°C)	26.5	28.5	32.0	29.5
exceed 26°C (h/yr)	7	513	1635	777
Zone 1, max.				
temp. (°C)	24.5	26.0	28.5	28.0
exceed 26°C (h/yr)	0	0	388	342
Zone 0, max				
temp. (°C)	24.5	26.0	28.0	26.5
exceed 26°C (h/yr)	0	0	465	16
<u>Cooling</u>, load (kW)	513	1133	1150	1120
running time (h/yr)	1963	4644	4679	4617
energy (MWh/yr)	200	1909	1948	1887
<u>Heating</u>, load (kW)	646	467	454	471
running time (h/yr)	4266	2207	2059	2193
energy (MWh/yr)	829	268	223	266

Table 3. Results for hour-by-hour simulations of a 2.000 m2 glazed atrium. Displacement ventilation versus mixing ventilation is used. Minimum indoor air temperature is 18°C.

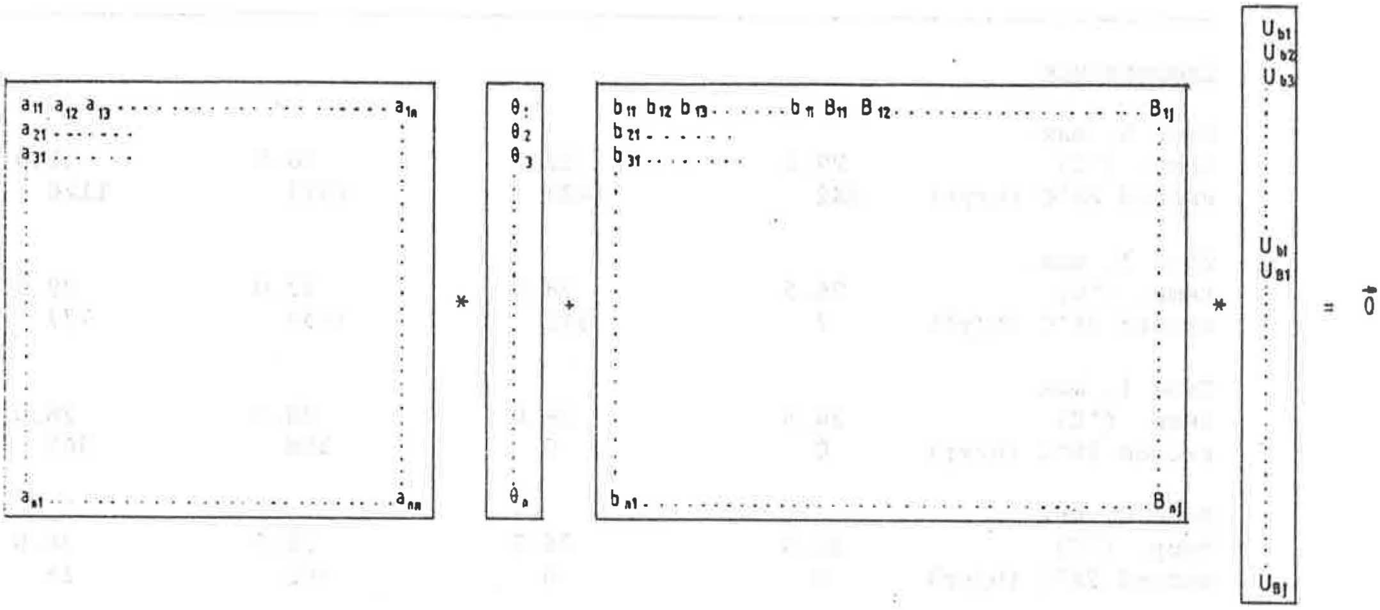

Figure 1. Room model matrix system where "a" describes the room model and the connection between the different elements (W/K).

"θ" are the calculated temperatures in the room model elements (°C).

"b" describes how the different loads aspect in the elements (dimensionless)

"B" are the connection between temperatures outside, other connected buildings, supply air from different air handlers, infiltration rate etc. (W/K)

"ub" are loads from the sun, indoor lightning and other sources (W)

"UB" are different temperatures loadings from outdoor air, connected buildings, supply air temperatures etc. (°C)

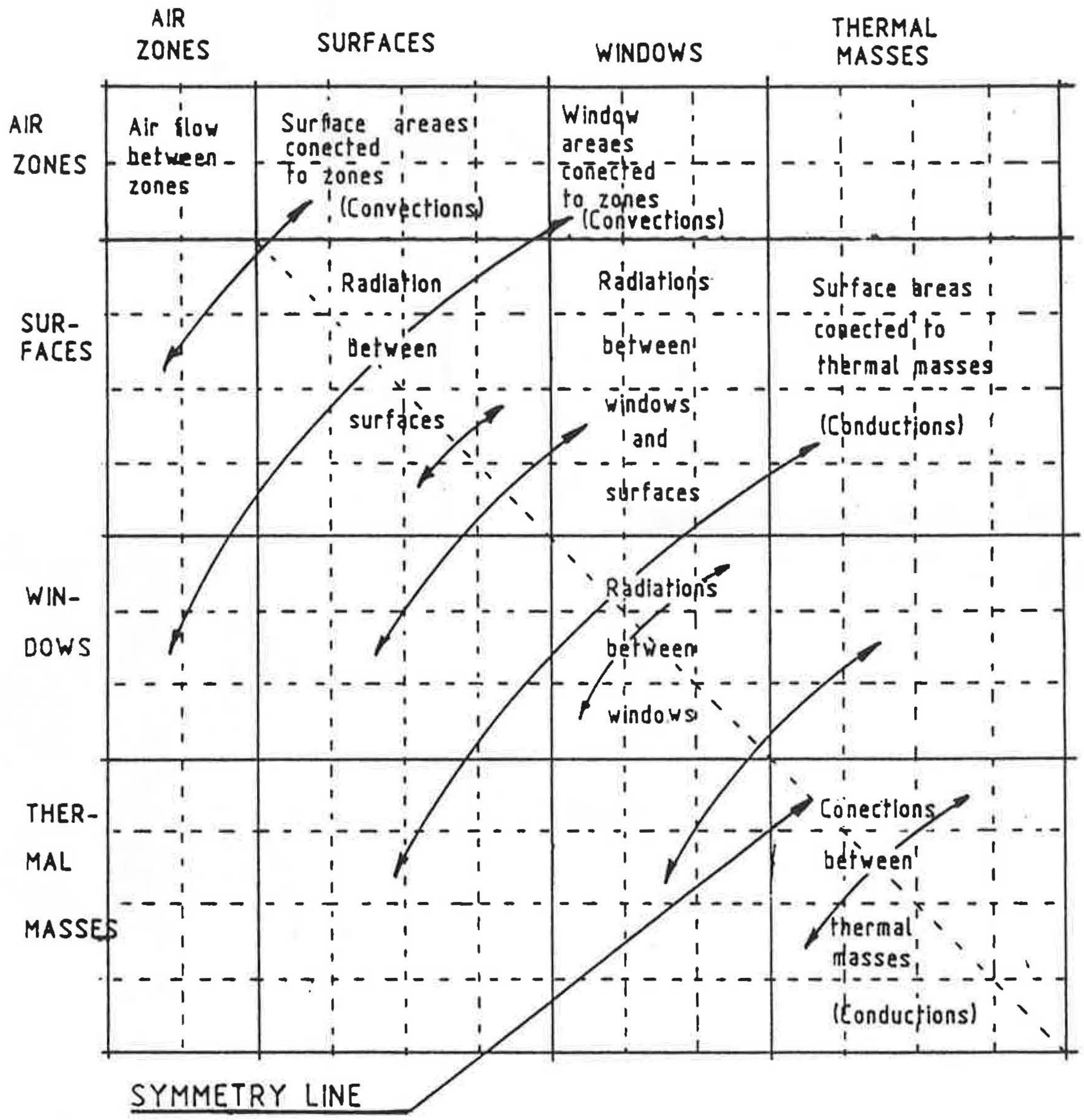

Figure 2. System matrix A

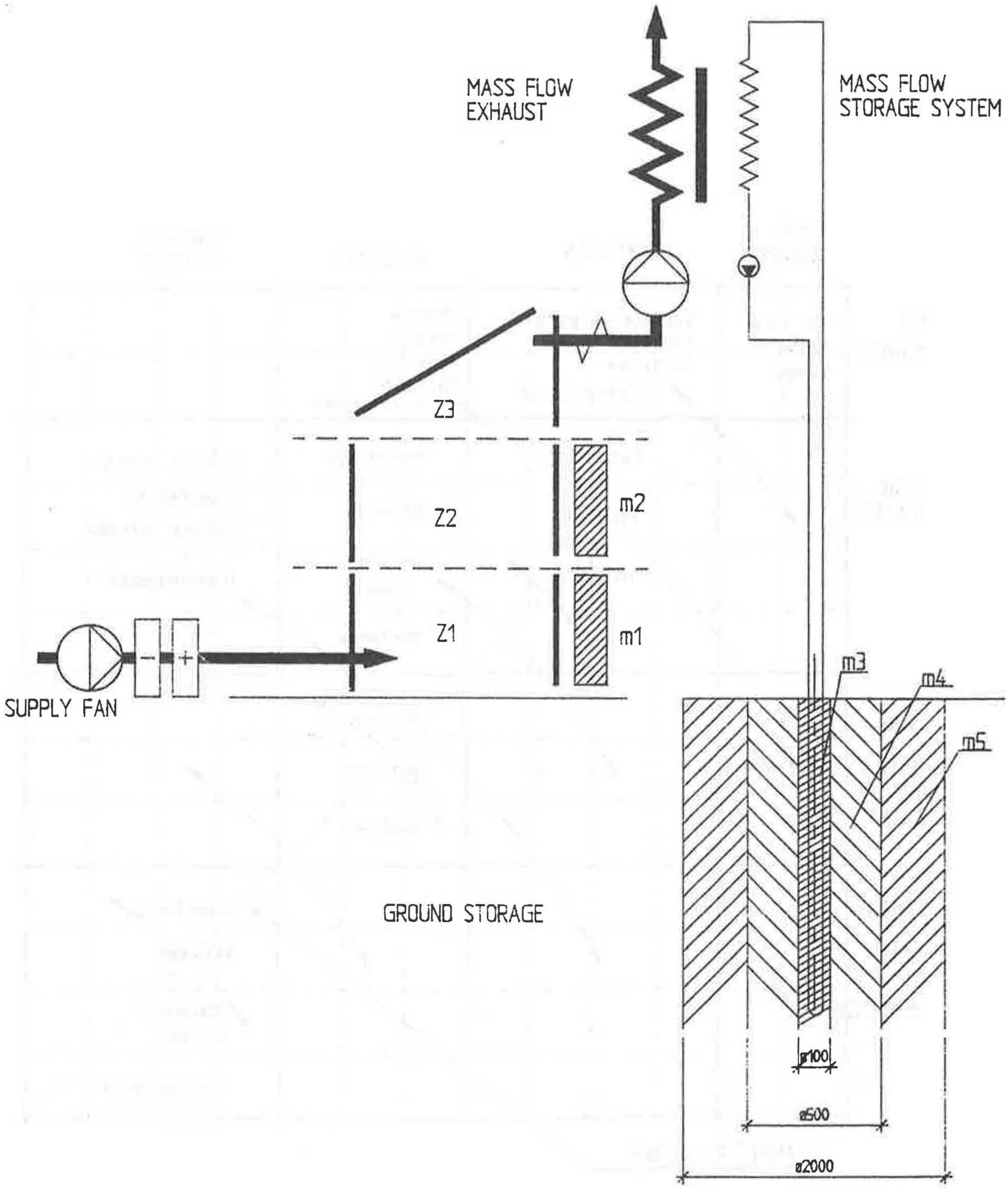

Figure 3, Principle for building with air handler, room with three zones and two building masses. The exhaust air is taken from the upper, warm zone and blown through an air/water heat exchanger. The warm water from the heat exchanger is during the summer season circulated into a ground storage, here simulated with three concentric masses.

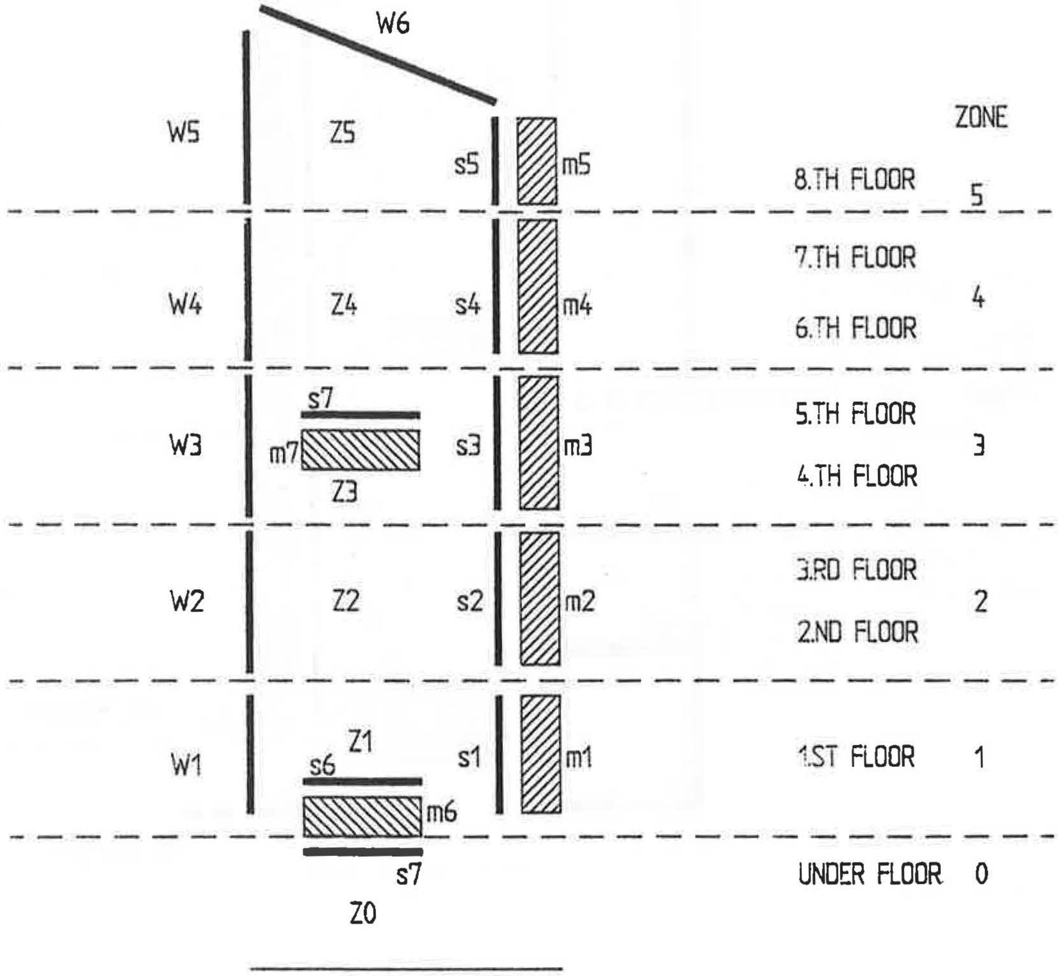

Figure 4. Simplified glazed atrium model, showing the six zones, six window areas, six thermal masses and eight wall surfaces.

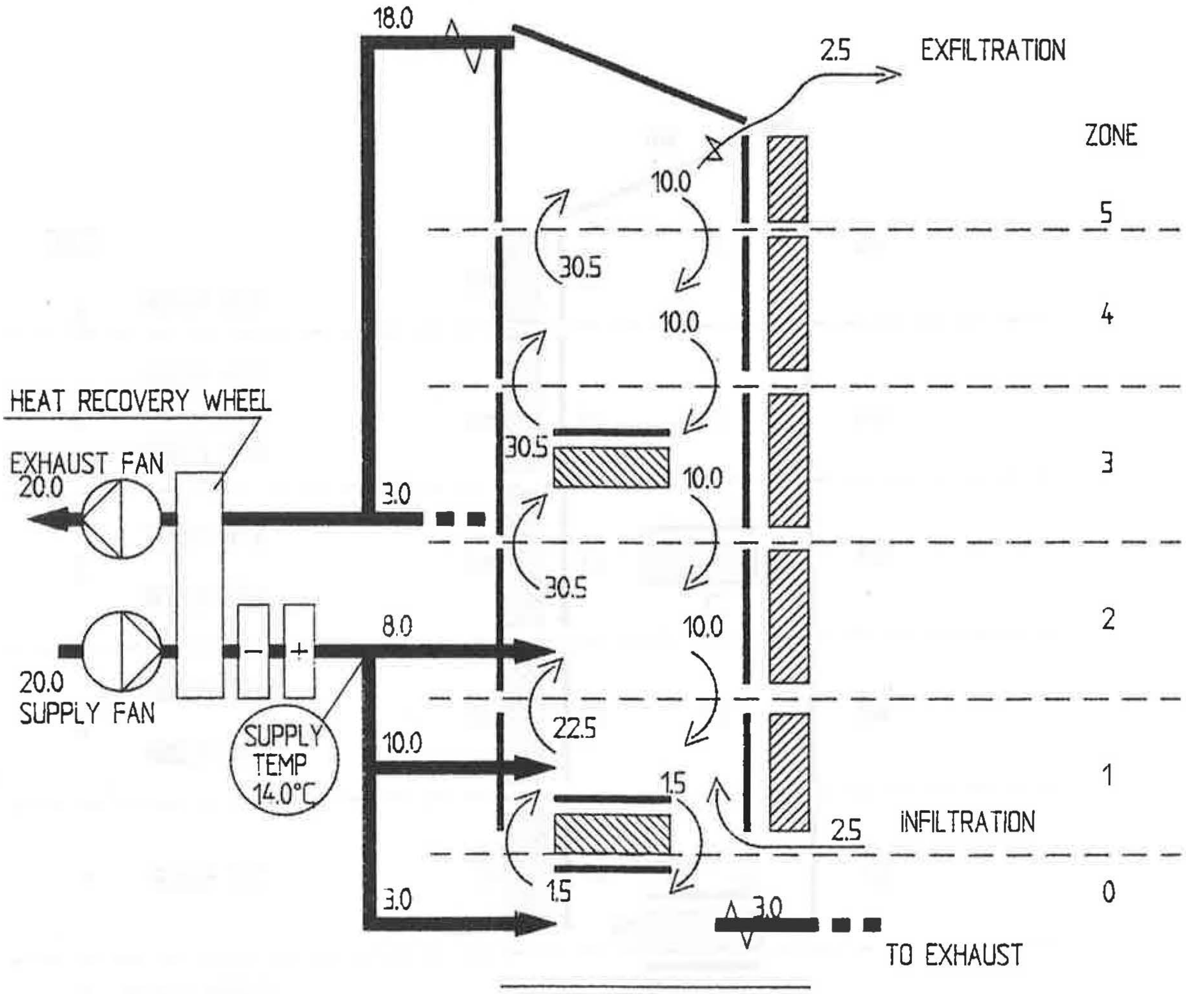

Figure 5. The air handler unit with supply and exhaust fan is shown with the internal displacement ventilation circulation. The air supply exhaust and internal air circulation rates are all given in (m3/s). Multiply the figures by 2,100 to achieve the rates in cfm. Total air handler supply rate 20 (m3/s), i.e. 42,000 (cfm.)

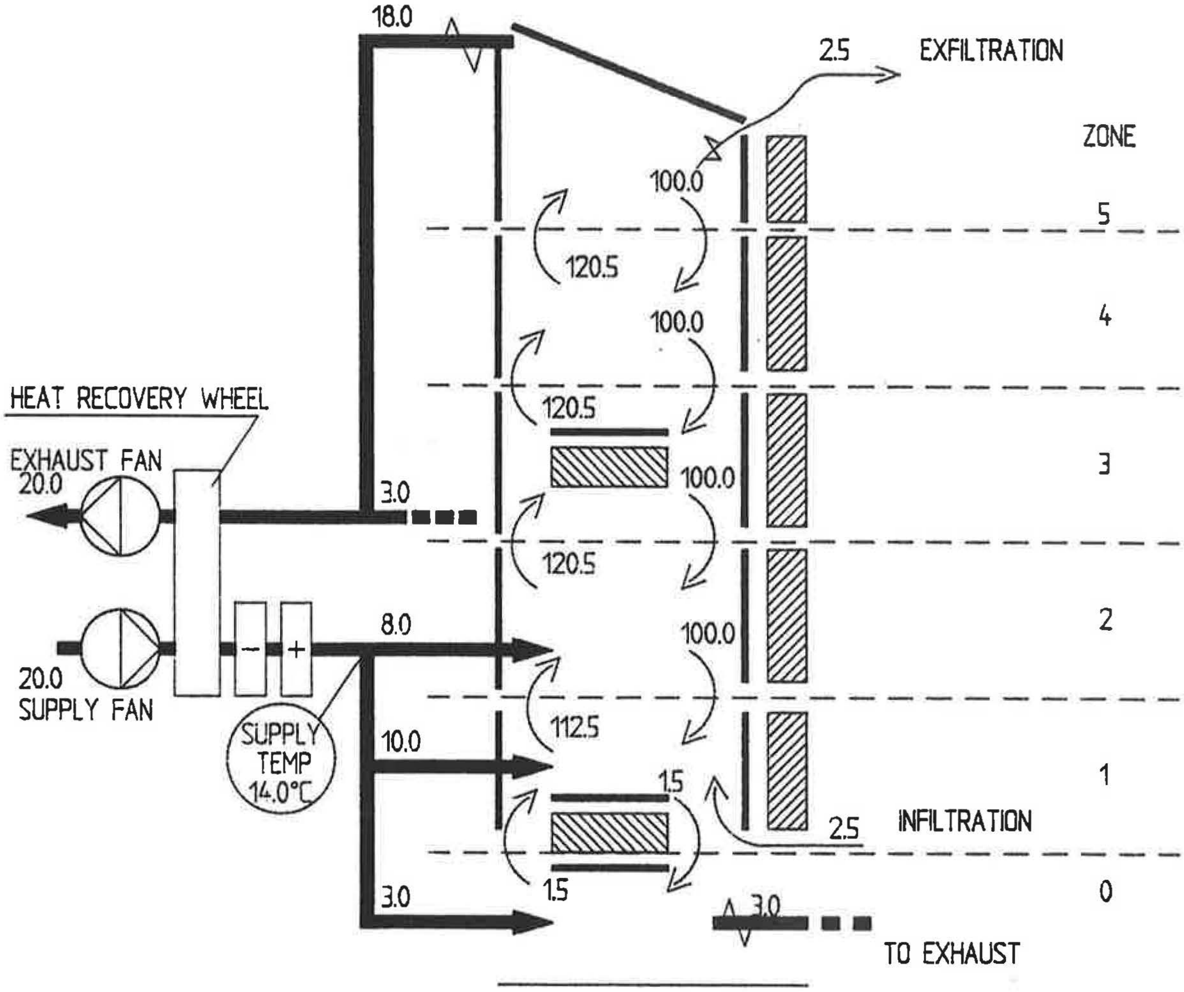

Figure 6. The internal air flow rates simulated for the case of mixing ventilation. All flow rates are given in (m3/s), to achieve (cfm) multiply by 2,100.