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THE UTILIZATION OF A BUILDING'S THERMAL INERTIA

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ABSTRACT

The purpose of this paper is to show how the heating and ventilating plant of a lease building was designed. An optimal design was achieved by utilizing the building's thermal inertia for both winter and summer operations.
Building environmental performance was critically assessed and the capital costs of the services necessary to provide environmental conditions came under careful scrutiny.

The heat loss or gain through the room envelope and the losses by infiltration and ventilation are calculated. Nonetheless, problems have arisen with installations calculated in this manner. In certain circumstances these problems have had to do with insufficient capacity . Two factors play a role:

- First, the increased degree of insulation and air tightness of a building due to rising energy costs. This results in a lower specific heat capacity of the installation, which means that the sensitivity to the accuracy of the calculation methods and its correction factors are considerably increased.
  Dynamic phenomena such as the heating up or cooling down of the building are more critical since the overcapacity in an absolute sense has decreased considerably.

- The second factor involved is the occupant's demand for a high quality indoor environment. The thermal comfort in a room must be adhered to at all times.

  This is demonstrated with the aid of computer simulation programs to show how the energy demands are kept at a minimum, resulting in a lower investment and a lower energy consumption.
  The ultimate result was the achievement of a balanced thermal response for the building.

Keywords

architecture	heat transfer
building systems	load calculation
energy conservation	mass transfer
energy calculation	office building
	thermal response
	ventilation

1 INTRODUCTION

Against the background of an increasingly competitive and technologically advancing building industry which serves an ever more demanding client base, the design function is becoming more complex. Building environmental performance is now critically assessed, and the capital and running costs of the services necessary to provide the required environmental conditions come under careful scrutiny.

The design analysis tools in common use in the building industry have not kept abreast of the needs of today's design professional. Although more comprehensive and appropriate analysis methods have been in existence for many years in the academic and research communities, they have been slow to emerge as commonly used design tools in commercial offices.
The purpose of most of today's calculations is to provide the correct dimensions of the heating and possible cooling capacities of a room, so that the required room temperature can be realised at a given outdoor climate (design condition). The calculation methods are based on a steady state model. The heat loss or gain through the room envelope and the losses by infiltration and ventilation are calculated in this manner.

However, in recent years problems have arisen with installations calculated in this way. These problems had to do with insufficient capacity in certain circumstances. Two factors play a role.

First, the increased degree of insulation used as a result of the energy crisis and subsequently higher prices; this also leads to an increased air tightness of buildings.
It results in a lower specific heat capacity of the installation, which means that the accuracy of the calculation method and its correction factors increases considerably.
Dynamic phenomena such as the warming up of the building after an interruption in the heating are much more critical because of overcapacity. In an absolute sense - has decreased considerably - this necessitates an accurate simulation of the heat exchange process.

All physical phenomena have to be calculated precisely to obtain the required accuracy.
The second factor involved is the occupant's demand for a high quality indoor climate.
The thermal comfort of a room must be considered at all times.
Mans thermal sensation is mainly related to the thermal balance of his body as a whole. This balance is influenced by his physical activity and clothing, as well as by environmental parameters: air temperature, mean radiant temperature, air velocity and air humidity.
When these factors have been estimated or measured the thermal sensation for the body as a whole can be predicted by calculating the predicted mean vote (PMV).[16]

At the present, load calculations are usually carried out with the aid of computers. The main advantage of this is that complex calculations can be carried out quickly and more parameters can be included in the assessment calculations.

In general, the thermal load for a July day is chosen as the maximum summer design condition, as shown in fig. 1. Needless to say, the cooling plant increases the mechanical installation costs.
The peak amplitude is, of course, the peak cooling duty which must be met by the plant. The area inside the graph equates to cooling consumption over the design day.

The peak cooling duty with allowance added for system gains, such as from fans and through ducts, determines the size of the cooling plant as a consequence directly relates to the capital cost of the mechanical installations.

The other prime calculation for system design is the thermal load for a January day, chosen as the maximum winter design condition also shown in figure 1. This is a heating duty and is predominantly needed for preheating, to raise the internal conditions to 20°C from the maintained night temperature of 12°C. During the day, the warmth given off by occupants, lighting and user equipment rapidly reduces the building's heating duty.
This, with allowance added for system losses determines the size of the heating plant and its capital cost. Again, the area inside the curve equates to the heating consumption over the design day.

These two calculations, often made only for peak hour, are customarily the sole basis for selecting a certain type of mechanical plant. Scant consideration is ever given to how that plant will operate during the majority of the time.

Using dynamic simulation computer programmes, hourly conditions can be calculated and an annual thermal load can be created (see fig. 2). The area inside the curve equates to the annual thermal consumption and directly relates to the annual running costs of thermal energy.
A further benefit of simulation is a "what if" check which shows the annual worth of the free cooling facility, also illustrated in figure 2. This shows the difference between the cooling curves with and without free cooling.
What is clear from the load diagrams is how rarely the system's cooling and heating plant is used at capacity to its capacity and also the short period of for which it is used at all. It stands idle and when run has little to do. The result is that it runs inefficiently and unhappily.

The annual cooling load through the cooling season May to September equates to about 50% of the plants actual duty when meeting the load for the July peak. Yet, even during the days of peak load in July, the cooling plant would be operated for less than 6 of the 24 hours.

This paper reports on a building designed primarily for the winter situation; secondly, it reports on the results of the building when tested for the summer situation; and thirdly, it tells of the final design stage. During this stage extensive use of computer simulation proved very useful: one of the major results of this simulation was the omission of a chiller installation, by making use of night ventilation (free cooling).

2 A DESCRIPTION OF THE PROJECT

The Hoendiep office building, situated in the town of Groningen in the Netherlands, consists of four wings situated around a patio. The wings have four floors. The technical rooms are on the third floor of the north wing, and the plant room on the roof of the south wing. (see fig. 12)
The building is leased to the Dutch Telecommunication Company.

2.1 Basis for design

The building is fairly conventional in its design. The offices can be built from modules 1,8 m' wide and 5,4 m' deep; there is one window every 1,8 m' and every other window can be opened.

There was however one factor which was to play an important role for the design. Internal or external sun shading would not be tolerated. This meant that during periods of high outside temperatures, a solution had to be found to lower the resulting inside air temperatures to maintain a comfortable inside climate.

The installations for this office building had to adhere to the following guidelines:

- The installations must be simple.
- The operation and maintenance of installations must be kept at a minimum.
- The installations must be automatically controlled in as far as this is possible in a simple manner.
- The cost of mechanical installations must not exceed £ 55/m^2

Because the building was to be leased there were some basic rules for the installations:

Client demands required the air ventilation system to provide maximum letting flexibility with respect to partitioning and floor letting.

Building dimensions and data are given in Table 1.

2.2 Building fabric details

The architect determined how the building fabric would be constructed; the fabric was then analyzed for thermal resistance (U-value), mass and response time, and found to be adequate at this stage. However if modification would have to be made during the simulation the architect was prepared to change the construction (if the budget allowed).
The construction of the outside fabric is shown in figure 11.

2.3 Thermal insulation regulations

In the Netherlands the thermal insulation regulations (NEN 1068)[21] must be applied to all buildings.

These regulations show how the thermal insulation index is formulated from the following:
- minimal values for the thermal resistance for the fabrics,
- minimal values of the thermal insulation index,
- a combination of these two possibilities.

According to the NEN 1068[21], this thermal transmittance value is a function of the relationship between the encasing area and the gross volume (the Ao/v relationship) and the thermal insulation index IT value which is calculated as follows:

$$IT = \frac{80.(Ao/v).(1-U) + 30}{4.(Ao/v) + 1}$$

In this manner, the defined IT index can be used to show the quality of a building's thermal insulation. The thermal insulation index for a well-insulated building must be at least 12. Energy saving buildings must attain a value of at least 14.

The thermal insulation calculation for this building and the minimum IT values for different countries are shown in figure 3.

2.4 Winter temperatures

The heat loss calculations were made in accordance with the ISSO publication no. 4.[22]
The minimum outside temperature for the calculations is -10°C, and the maximal wind velocity is 10 m/s.

The following minimum temperatures were used for the calculations

offices	20°C
lavatories and utility cupboards	15°C
halls and stairways	15°C

Outside of working hours the temperature in the building was not allowed to drop below 12°C.

Table 4 shows the heat loss calculations for the reference room.

2.5 Summer temperature

General

The design must be based on the need for a comfortable inside climate for the occupants.
In the summer season the occupant will be much happier if he can personally influence the inside climate, for example, by opening a window.
However, when the windows are opened the noise level inside can increase and the chance of drafts will increase, therefore it will be necessary to design the installation with the windows closed.

Summer temperature regulations

The inside climate is judged in the summer situation according to the Dutch Government Building Services department regulations 3, 4, 5 and 6. These regulations state the following:

1. An inside air temperature of 25°C may not be exceeded for more than 5% of the yearly occupancy period (125 hours).

2. An inside temperature of 28°C may not be exceeded for more than 1% of the yearly occupancy period (25 hours).

Summer design conditions

Outside 28°C/60% RH
Inside 25°C *)

*) These temperatures may not be exceeded for longer than the Dutch Building Services Department regulations 3, 4, 5 and 6 stipulate.

For the various thermal analysis calculations the total internal loads were 37 W/m²
- which were distributed in

	occupants	12W/m²
	machines	10W/m²
	lighting	15W/m²

2.6 Ventilation requirements

- The minimum fresh air per person must be 15 m³/h.

Ventilation amounts

office ventilation rate **8 m³/h/m²**

2.7 Thermal comfort

Recently, the prediction of human comfort has been an area in which extensive research has been carried out. Essential to the prediction of human thermal comfort is the concept of thermal neutrality. Thermal neutrality can be predicted with the comfort equation, which considers six parameters:
- activity level
- clothing
- space air temperature
- mean radiant temperature
- air velocity
- humidity.

For comfort to be achieved, there must also be an absence of local body thermal discomfort caused by local convective cooling (draft), vertical air temperature gradients or asymmetric thermal radiation. Design standards using thermal comfort have been set forth in ASHRAE Standard 55-81, NKB guidelines Nordic Committee on Building regulations) and ISO 7730[16].

Predicted mean vote (PMV) 16

Fanger[11] deduced a "thermal index" which could express a subject's thermal sensation in a climate deviating from the optimum.
Fanger[11] assumed that thermal sensation is a function of the thermal load of the body. The thermal load is defined as the difference between the internal heat production, and the heat loss to the actual environment for a person with (theoretical) mean skin temperature and sweat secretion at the actual activity level.
Fanger[11] quantified the relationship from the results of experiments in which people were asked to cast a "thermal sensation" vote. Clothing, activity, air temperature, mean radiant temperature, relative air velocity, and air humidity were carefully controlled so that the thermal load could be calculated.

The PMV scale is perhaps a little difficult to interpret. People are not identical, so in reality a group of people would report a varying range of thermal sensations. The PMV indicates the most probable sensation from the thermal load conditions. Thus, the PMV gives a general indication of the level of comfort (i.e. thermal sensation), but contains no indication of the range of comfort actually experienced (because people are not identical).

Predicted percentage of dissatisfied (PPD) 16

The PPD scale does provide an indication of the range of comfort experienced in reality, due to individual differences. The relationship between PPD and PMV was deduced by Fanger. The PPD indicates the percentage of people, who when asked the question "How comfortable are you ?", would say "I feel too warm" or "I feel too cold".

It is impossible to satisfy all persons in a large group in the same climate. Even with a perfect environmental system a PPD less than 5% is unattainable (a point often overlooked in practice where any complaints, however few, are taken as an indication that the system is defective or badly operated).
If the thermal field is uniform, the PMV will be the same for all of the occupied zone and by changing the temperature level a PMV of zero can be obtained. This is the only way the minimum PPD of 5% can be achieved for a whole zone. If the thermal field is not uniform in the occupied zone, changing the temperature level will make it still possible to achieve an average of zero PMV, but the PPD will be higher than its minimum value (of 5%).

Mean radiant temperature (MRT)16

The mean radiant temperature has a considerable influence on a person's heat loss and thus his state of comfort. MRT is dependent on geometry and will generally be different at each point in the occupied zone. So the MRT is dependent on occupant position, and occupant orientation relative to the surfaces of the space.

3 THERMAL CALCULATIONS

3.1 Heat loss calculations

General

The heat losses of a building generally take place at the borders between the inside and outside climates, that is to say the building shell, which consists of the outside walls and glass areas, the roof and the floor.

To keep these resulting heat losses at a minimum it is necessary to keep the borders to a minimum; this requires a unique building form. For this reason the Ao/v relationship is important, that is to say the relationship between the heat exchanging encasing area Ao (the shell) and the enclosed volume v (the gross volume). This building, due to its compact size, has a low Ao/v relationship (A o/v = 0,34 m^2/m^3).
The thermal insulation index (It-value) for the office building is shown in table 2.

Next to minimizing the outside area and form of the building, thermal insulation is one of the most important factors for keeping the energy usage at a minimum.

The heat loss calculations for the reference room of the Hoendiep office building are shown in table 4.

Table 4 shows how the heat losses for the reference room were calculated. It can be seen that the basis transmission was 2083W while the total transmission was 4046W, nearly twice as much; this was due to the oversize ratio of the system resulting from intermittent heating. When intermittent heating is used, a period of preheat is necessary before the building is occupied, and this preheating period varies with the oversize ratio of the system.
The total thermal heat loss for the building was calculated to be 505kW which can also be shown as 60.8 W/m^2 of the building floor area of 19 W/m^3 of the building's volume.

3.2 Summer temperatures

As I stated before, during warm sunny periods buildings are subjected to daily cyclical heat gains from solar radiation; in addition, further gains arise from artificial lighting, occupants and other sources.

In designing the building it is important to ensure that it will not become uncomfortably hot during hot sunny periods i.e. that the maximum peak temperature should not frequently exceed 25°C and 28°C, as stipulated by the Dutch Government Building Services Department regulations 3, 4, 5 and 6.

The internal temperatures for this building were simulated with the aid of the TOVER[19] computer program. The first results calculated with a ventilation rate of 3 air changes per hour during the occupancy period exceeded the maximum limits of the regulations. The supply air temperature was then set at 19°C (top cooling) with 3 air changes per hour during the occupancy period. The result of this simulation were within the regulation limits. The costs for a chiller installation exceeded the financial budget.

A ventilation system with night ventilation was also simulated; that is a system in which 3 air changes of outside air are supplied during the occupation period at 18°C, unless the outside air temperature is higher than 18°C. In that case the supply temperature is the same as the outside air temperature plus the heat gained from the supply fan.
If, outside of the occupation period, the inside air temperature is predicted to be higher than 22°C, the ventilation system is switched on (without any heating) and 100% outside air is continually supplied until the inside air temperature reaches 18°C. This can only take place by a temperature difference of 3 degrees or more.
It can been seen in figure 4 that the results of the simulation were lower than the maximum regulations, therefore acceptable.

The effect of this night ventilation on the buildings mass can be described as follows. The heat capacity of the buildings mass is lowered by reducing the thermal capacitance of the building's fabric. It is a widely accepted principle that steady state heat flow through a slab can be modelled as an electric current flowing through a resistor; unfortunately, this analogy only holds true if the outside and inside climates are constant. Kreith[13] shows that the effect

of time on the heat flow through the slab can be modelled by introducing the concept of "thermal capacitance". This thermal capacitance adds a time delay to the temperature and heat flows that are predicted in the slab. What happens in practical terms is that during occupation periods, heat is supplied from outside, loading the thermal capacitance. If no cooling or night ventilation is used then this capacitance will unload, especially after periods of occupation transferring heat energy back into the room.
By utilizing night ventilation, both sides of the building fabric are subjected to lower temperatures. Heat is rejected from the building mass in both directions, outside and inside; it is then absorbed by the ventilation air. The thermal capacitance of the building mass is therefore lowered.
Another advantage of using night ventilation is that heat from the building's mass is extracted from the inside. The building fabric was essentially designed for the winter period, where the heat energy flow is from the inside to the outside. Therefore the response is slower. When this heat flow is reversed or split into two directions, the response time is then shorter.

3.3 Thermal comfort

The thermal comfort simulations were made with the aid of the ROOM[20] program.

The prediction of comfort level requires calculation of the following factors at the required location the occupied zone:

- Dry bulb air temperature
- Radiant temperature
- Air speed
- Vapour pressure

Such analysis of the thermal environment requires a complete solution to the equations representing air movement and thermal response of the building fabric under non-steady state conditions.

Conventional thermal analysis methods, such as the environmental temperature and admittance methods used in other programs, make simplified assumptions about heat transfer.

Usually these assumptions are about the uniformity, homogeneity and direction of heat flow and temperature gradients. As spaces get larger, the validity of these assumptions reduces, and more rigorous calculation methods are appropriate. A consequence of the assumptions in the simpler calculations is that a space is uniformly comfortable (or uncomfortable). This is not always true and ROOM[20] calculates the dry resultant temperature and the dry bulb temperature in the occupied zone.

Heat is transferred by conduction, convection and radiation, each of which is considered separately. The calculation of radiation form factors is essential to the assessment of radiant heat flow; these are calculated for the space. Conduction through the enclosure surfaces is calculated dynamically, with time steps adjusted by the program.

The program takes account of the spatial arrangement of the surfaces and the way each surface affects others. This makes it possible for the sunlight falling on each surface to be treated separately, for example.

The program also takes into account the effects of humidity and room temperature on the heat given off by occupants, as well as level of activity.

Figure 5 shows the effects of the temperatures for the reference room.

Explanation of comfort parameters

ROOM[20] uses Fanger's[11] comfort equation to express thermal comfort in a space in terms of PMV (Predicted Mean Vote) and PPD (Predicted Percentage of Dissatisfied). In addition, the true mean radiant temperature experienced by people is calculated.

Limitations

Fanger's[11] comfort equation is empirical, and is based upon statistical data gathered in working environments. Consequently, care is necessary in interpreting comfort results in a space which is not essentially a workplace (for example a shopping mall or leisure complex). The main characteristics of a work environment are:

- Occupants are in the space for continuous long periods.
- People are at work, not at leisure.
- The minimum clothing level is generally higher in a working environment (i.e. on the hottest day someone at leisure wearing shorts and a T-shirt would probably tougher than the level of comfort expected in a leisure environment. So, when analyzing a space which is not a workplace the comfort results will probably be pessimistic. That is, people will feel less satisfied with an environment if it is their workplace rather than a place of leisure, because they will tend to judge the environment more severely.

For a global analysis ROOM[20] carries out an analysis of the radiant field at each analysis point in the occupied zone. A person is assumed to exist at each analysis point and a heat balance can be established at the surface of the person. Angle factors are used to evaluate the surface/subject radiant exchange.

At this stage only, long wave radiation from normal low temperature surroundings has been considered (where radiant exchange surroundings are dependent on the temperature of the surroundings). However, short wave solar radiation can have a significant effect upon the MRT (as radiant exchange with the surroundings is independent of the temperature of the surroundings, due to the high radiant source temperature). This effect is considered using the theory developed by Fanger. The path of sunlight is tracked in the space and modifies the MRT (at each hour) to include the effect of direct solar radiation if the analysis point is not shaded.

The problem faced having lowered the thermal capacitance of the buildings mass was the fact that mean radiant temperatures were also lowered. To see exactly what the effect of this was the ROOM[20] program was used to simulate the effect. Figure 5 and 6 show the results of the computer simulation. It can be seen that the inside air temperature at 8 a.m., the beginning of the occupation period, is 16,6°C and that the mean radiation temperature is 20,3°C. The result is a cold sensation producing a PMV of -2,21. As soon as the ventilation system is started conditions improve. By 11 a.m. the comfort conditions are acceptable until the end of the occupation period.

4 SYSTEM CHOICE

4.1 Heating

The building is heated by means of radiators placed under the windows, which also compensates for the negative radiation effect of the windows. The ventilation air is not used to heat the building.
The offices are kept at a 20°C minimum; outside of working hours the temperature in the building is not allowed to fall below 12°C.
When intermittent heating is used a period of preheating is necessary before the building is occupied; this preheating period varies with the oversize ratio of the system.

In practice, the oversize ratio varies from day to day based on outside conditions. For example, a heating system which was designed to provide 20°C with a constant outside temperature of -12°C would be under half load conditions at a constant 4°C outside temperature, or, alternatively, it would have an oversize capacity of 100%. The time when the heating system should start therefore varies from day to day, if the building is always to achieve its desired temperature or comfort condition at 9 A.M. for example.

If a heating system is oversized, the preheat period will be shorter and the system will operate for a reduced period of time, resulting in lower running costs. It follows, however, that if too large of an oversize ratio or margin is used, any saving would be more than offset by the reduced part load efficiency of the plant. In addition, a larger system will have more heating surface and an increased boiler capacity resulting in a higher capital cost.

4.2 Mechanical ventilation

The air supply for the offices will be changed 3 times every hour. The offices have a ceiling height of 2700 mm which provides approximately 8,1 $m^3/h/m^2$ of floor area or when occupied by 1 person per 10 m^2, then this equates to 80 m^3/h per person.

The minimum amount of fresh air per person is 15 m^3/h; therefore, it can be deduced that the total supply air may consist of 18% fresh air and 82% recirculation air.

The air supply temperature is controlled at a constant 18°C (unless the outside air temperature is higher than 18°C).

Night time ventilation is used to cool the building down during the night period. When the air temperature measured in the reference room is higher than 22°C, the ventilation system is switched on (without any heating) and 100% outside air is continually supplied until the air temperature in the reference room is lower than 18°C. (This is only by a temperature difference of at least 3K between the air temperature in the reference room and the outside air temperature).

Due to the fact that night ventilation is used to cool the building during periods of high outside air temperatures, a Chiller machine is not necessary.
If it was necessary to utilize top cooling, that is to say lowering the air supply temperature to about 19°C which would create an inside air temperature of about 24°C (PMV $\pm$ 0,5).

Which then needs 270 kW of cool energy (thermal). This is approximately 60 kW of electrical energy (COP Chiller machine = 3).

Therefore the annual electrical consumption is about 36000 kWh. The extra electrical consumption resulting from the use of fans for free cooling (night ventilation) is roughly 6000 kWh. The difference is 30.000 kWh per year of Dfl 6600,= (£ 2200,00).

The annual electrical energy savings are quite small really but if one does include the costs of a Chiller installation and the maintenance of the Chiller, installation costs would be about £ 2800.00 per annum.

5 OPERATIONS

It can be seen from the design calculations and simulations what should happen in heating and cooling the building. But what about the actual situation? The building has been in use now for about 30 months and is exceeding the expectations of both the owner and the user.

During the winter period, however, the building has the inclination to slightly overheat to about 22°C (the design temperature was 20°C). This can be the result of the following factors:

- The U-values are slightly better than the design U-values.
- The radiators are slightly oversized due to the fact that the same size radiator is used for a 1,8 m' module, with or without the possibility of opening the window.
- The thermostatic radiator valves respond slightly slower than the manufacturer states.

During the summer of 1989, the inside climate functioned according to the simulations. Inside temperatures were measured in various offices throughout the summer period and temperatures of 25°C were rarely exceeded.

CONCLUSIONS

- Investigations into a thicker concrete layer to increase the mass of the slab did not greatly improve the performance of the thermal capacitance of the multiple layer slab but the increased weight of the building would increase the buildings costs and also provided problems for the structural engineers. Reducing the thickness of the concrete layer and increasing the insulation layer did not drastically reduce the U value; however, the thermal capacitance of the multiple slab was reduced so that the building reacted very badly during simulation.

- Radiators provide compensation for the cold windows during the winter operation. Simulation with an all air heating system proved disappointing, mostly due to the fact that higher air supply temperatures and colder walls would require a greater air volume and use more energy.

- The radiators were provided with thermostatic valves which allowed for individual room control during the winter period.

- Before night ventilation was fully simulated, the choice of glass was an important factor in predicting the inside air temperatures. It was quite clear that the architect had definite ideas about the colour of the glass. After testing several types of glass reversol silver 17/19 was chosen. This glass proved a good alternative for double glass with outside sun shading. The amount of solar energy which was admitted into the room together with the internal heat production by lighting, people and machines (37 W/m^2) determined the inside temperatures. Because the ventilation air supply is 100% outside air, heat can either be absorbed into the air producing high inside air temperatures, or it will be absorbed into the available cold sinks the outside walls. When the saturation point of the outside wall is reached then the inside air temperature will rapidly rise.However, reserve sections for cooling coils were incorporated in the air handling units, but even during the hot summer of 1989, when outside temperatures of 30°C and 32°C were recorded, it still was not necessary to install a Chiller plant.

- When the building was being simulated for thermal effects there were several apprehensions mainly due to the fact that the comfort analysis was not a parameter included in the original design phase. The dynamic phenomena of warming up and cooling down the building was included in the design, but no account was taken of how a person or persons would feel under these conditions.

- Because a maximal inside air temperature of 28°C or higher would be tolerated for 1% of the occupation period, a comfort analysis was simulated with the ROOM[20] program. Results of this simulation show that the average PPD would be around 10% for most of the day, which under the circumstances was acceptable. (See Figure 7 and 8).

- Air temperatures of 25°C or higher were permitted to occur for 125 hours of the occupation period; this was also simulated with the ROOM[20] program. The results shown in figure 10 give a more acceptable result.

- Another problem arising from thermal comfort analysis is that the PMV and PPD values tend to be the average for the room being simulated. In developing the ROOM[20] program, special attention was paid to calculating and displaying the conditions of comfort at various places in the room.
  This can clearly be seen in figure 9 Because of this detailing it will be possible to indicate which positions should be defined as comfortable zones.

  Due to the fact that temperatures of 25°C and 28°C were only moderately exceeded it was quite clear that a Chiller would not be necessary. Analysis of the simulations showed that after about 2 P.M. the inside air temperatures rose quite rapidly. Experiments into lowering the temperature of the slab, so that the thermal capacitance of the slab would be increased resulted in uncomfortable climates during the morning periods.

- The omission of a chiller plant was very attractive to the architect and the consulting engineers. Firstly, due to the fact that the initial plant room size was decreased. Secondly, the mechanical installation costs were lowered.

- The first simulations with night ventilation proved very interesting. The inside air temperatures rarely exceeded the 25°C and 28°C limits. Investigations into the circumstances under which the night ventilation should be switched on and off proved very laborious. Eventually, the night ventilation could be effectively (under simulation) be switched on and off by means of the inside and outside temperature differences. The room temperature must be higher than 22°C and there must be a temperature difference of at least 3K between the outside and inside air temperature. This protocol is very easily integrated into the simple controls of the building and the air handling units, avoiding complicated control systems. The result of this night ventilation cooling can be seen in fig. 6.

- During the comfort analysis for the winter period, one of the most startling findings was made; under the steady-state calculation method nearly 100% of extra capacity was included to cope with such problems as orientation, wind pressure and the warming-up of the building.

  After night set back, the installation was automatically switched on so that an inside air temperature of 20°C was achieved by 8 P.M., the beginning of the occupation period. It was however not until about 10 or 11 P.M. that the comfort results were acceptable. On Monday the comfort settings were not judged to be acceptable until nearly 2 P.M. in the afternoon. To compensate for this, the installation was switched on earlier to achieve a comfortable inside climate on monday morning at 8 A.M. The plant had to be switched on at about 23.00 on Saturday evening. Investigations were then carried out to try to determine under which conditions and with what plant capacity a reasonably comfortable climate could be achieved. The results indicated that steady state calculations using an outside temperature of -7°C in place of -10°C and continually heating the building instead of using night or weekend setback would lower the heating capacity and therefore decrease the costs of the heating plant and would increase the internal comfort. At the moment the Dutch Government is going to change its regulations in accordance with this method.

FUTURE WORK

Because most of today's buildings are designed with the aid of computer simulation, the results of these simulations often lead to actual design specifications, and that's as far as it goes. What I am doing at the moment is extending the simulation program so that it can be integrated into the building's control system. Therefore, the available measured information (outside air temperature, inside air temperature, supply and return air temperature, time etc.) simulation of the installed plant's behaviour can first be carried out. Then the results can be analyzed and if required stored before adjustments are made to the controls of the system. Providing the simulation program is good, it will be possible to control a building within the comfort index parameter. With the readily available temperature thermostats special control equipment all that will be necessary is a good computer; this would obviously have to the base of a knowledge-based system, because it can store its simulated recorded results, it will be able to teach itself and become an expert system.

APPENDIX A

The specific working mass (SWM)[5] of a room is defined as the total of the working mass of the rooms surfaces divided by the total internal surfaces of the room.

$$SWM = \frac{\Sigma\ (f\ m \cdot Mw)}{Av}$$

Where: e is the working mass of a layer in kg
w = pw.dwz.Aw.0

ew = the density of the material kg/m³
dwz = the effective thermal mass of the layer note 60 mm is the maximal thickness which is taken into consideration.

w.0 = in the total internal area of the room
wm = a reduction factor for lowered ceilings.

Table A appendix

CALCULATION OF THE SWM FOR THE REFERENCE ROOM

	Aw.0 [m²]	dw.z [m]	Pw [kg/m³]	Mw [kg]	Mw.fwm [kg]	Aw.t [m²]
Outside wall						
concrete	24,72	0,2	2500	12360		33,48
insulation	24,72	0,04	50	49,44		
wood	24,7	0,02	700	346		
			=	12755	12755	
Inside walls						
fibre panel	33,48	0,001	1200	40.176		33,48
wood	33,48	0,006	700	140		
insulation	33,48	0,03	50	50		
			=	230	230	
Corridor wall						
fibre panel	33,48	0,001	1200	40.176		33,48
wood	33,48	0,006	700	140		
insulation	33,48	0,03	50	50		
			=	230	230	
Ceiling						
ceiling tile	58,32	0,06	500	174		58.32
cavity	58,32	-	-	-		
concrete	58,32	0,2	2500	29160		
			=	29339	20533	
Floor						
concrete	58,32	0,2	2500	29160		58,32
			=	29160	29160	

total mass Σ (fwm.M10) = 62908
total internal area Av = Σ Aw.t = 217

Specific working mass $SWM = \frac{62908}{217} = 289\ kg/m^3$

Table B Appendix

OVERALL THERMAL PERFORMANCE

A conveniant method of expressing the overall performance of a room is given by the ratio known as the response factor given by:

$$fr = \frac{\sum(AY) + 1/3\ NV}{\sum(AU) + 1/3\ NV}$$

Where : N = number of air changes (3)
V = room volume (10,5 x 5,5 x 3,1) = 179,025

Note

The Areas (A) required for the response factor calculation are calculated as follows:

outside wall	= 10,5 x 3,1	= 32,55 m^2	A = 32,55 m^2
inside walls	= 5,5 x 3,1	= 10,85 m^2]	
	5,5 x 3,1	= 10,85 m^2]	A = 54,25 m^2
	10,5 x 3,1	= 32,55 m^2]	
floor	= 10,5 x 5,5	= 57,75 m^2	A = 57,75 m^2
ceiling	= 10,5 x 5,5	= 57,75 m^2	A = 57,75 m^2

Response factor calculations

	A	Y	(AY)	A	U	(AU)
outside walls	32,55	6,26	203,703	32,55	0,59	19,20
inside walls	54,25	4,68	253,89	-	-	
floor	57,75	2,36	136,29	-	-	
ceiling/roof	57,75	1,11	64,102	57,75	0,23	13,28
			$\sum$(AY) = 658,0455		$\sum$(AU) =	32,487

$$fr = \frac{658.045 + 161}{32.487 + 161}$$

$$fr = 4.2$$

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	floor area		volume (gross)	
	office	halls/toilets	office	hall/toilets
Ground floor	1527	507	4123	1217
1st floor	1545	470	4172	1128
2nd floor	1555	460	4199	1104
3rd floor	1005	414	2713	994
rooftop plant room	218			
total = 8300 m²			26500 m³	

gross volume		= 26.500 m³
percentage of glass in the facade		= 24%
type of glazing		= reversol silver 17/16
insulation index (It value)		= 17,85
occupation period		= 08.00-18.00
design conditions	- winter	= -10°C/90% R.H.
	- summer	= 28°C/60% R.H.

TABLE 1

DIMENSIONS AND DATA FOR THE HOENDIEP OFFICE BUILDING

Fabric	Area Ao(m²)	W.Factor a	U-value (W/m².k)	Ao.a.k (W/k)	%
Floor incl. 6 m'	220	0,68	0,46	69,3	1,4
Floor excl. 6m'	1842	0,20	0,46	169,5	3,4
Roof	2168,6	1,00	0,50	1084,3	21,7
Glazing	798	1,00	1,80	1436,4	28,7
Doors	16	1,00	3,50	56	1,1
Facade I	1710	1,00	0,60	1026	20,5
Facade II	1552	1,00	0,46	713,9	14,3
3d Floor	642	1,00	0,55	353,1	7,1
overhang	195	1,00	0,46	89,7	1,8
total =	9143,6			4.998,2	100

The building volume = 26540 m³
The mean U value = 0,55
Ao/v m²/m³ = 0,34
Thermal insulation index = 17,87

TABLE 2

THE THERMAL INSULATION INDEX CALCULATION (IT-VALUE) FOR THE OFFICE BUILDING

Temperature = 22°C
Orientation = south

Outside air temp. = -10°C
Wind velocity = 10 m/s

The office will be occupied for only 9 of the 24 hours.

Fabric	U-value w/m².K	temp. other side	length m	width/ height m	total m²	subtraction	total m²	transm. watt
Double glazing	1,8	-10	0,80	0,80	6	-	3,84	221,2
Facade	0,60	-10	1,8	3,50	6	3,84	33,96	652
evn.	3,50	15	0,80	2,00	1	-	1,60	39,2
bid	2,30	15	1,00	2,00	1	-	2,00	32,2
bim.	0,51	15	12,00	3,50	1	3,6	38,4	137,1
roof	0,5	-10	10,00	6,26	1	-	62,6	1001,6
D factor = 0,276						total =	235,9 =	2083
						additions =	23% =	479
						total transmission losses =		2562 W

Holes and cracks	length	R/D Quality m^3/s m Pa 1 2/3
Crack 1	9,6 m	$c = 0,2 \times 10^{-3}$
Crack glass/frame	18 m	$c = 0,5 \times 10^{-3}$
Crack frame/facade	19,2 m	$c = 0,4 \times 10^{-3}$
Facade/glass area	34 m²	$c = 0,4 \times 10^{-3}$

Housing factor = 5392 JPa 1 2/3 / [m³.k]

Room factor = 0,7

Losses though holes and cracks = 1484 Watt

Totaal heat losses for this office = 4046 Watt
=========

Heat loss per m² = 64,6 W/m²
Heat loss per m³ = 18,5 W/m³

TABLE 3

THE HEAT LOSS CALCULATIONS FOR THE REFERENCE ROOM OF THE OFFICE BUILDING

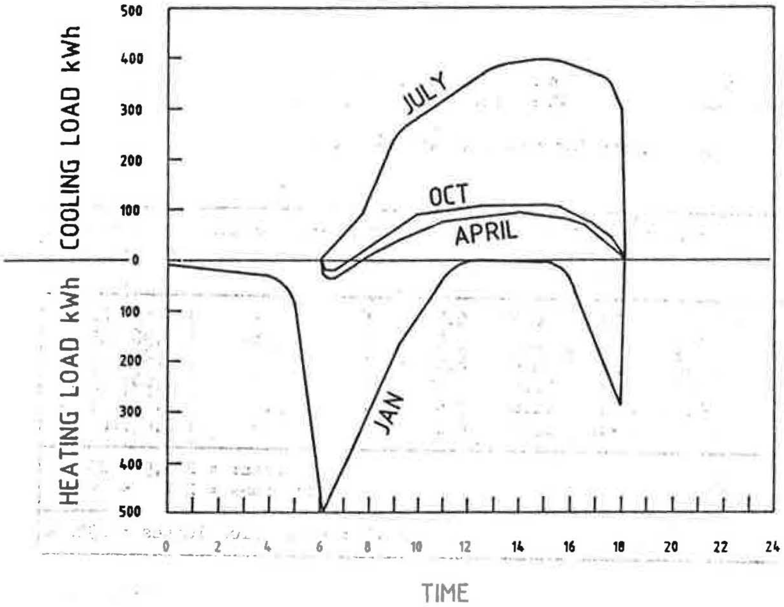


FIGURE 1 : MONTHLY HEATING AND COOLING LOADS

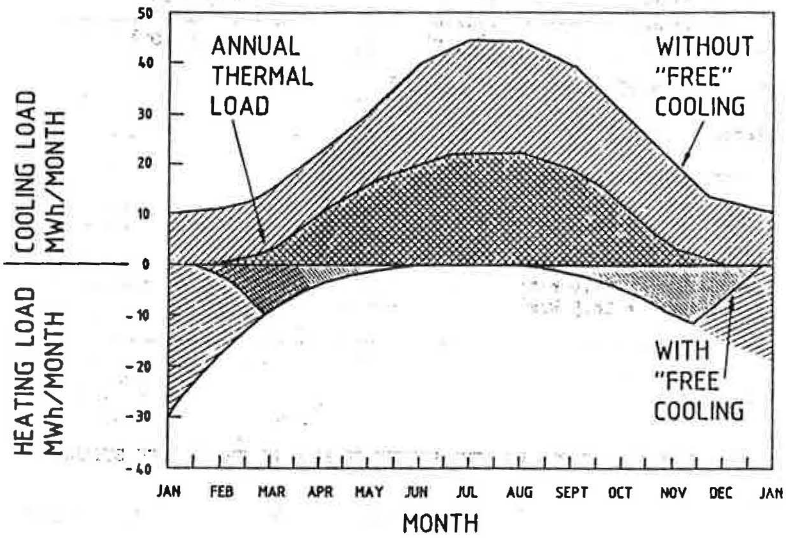
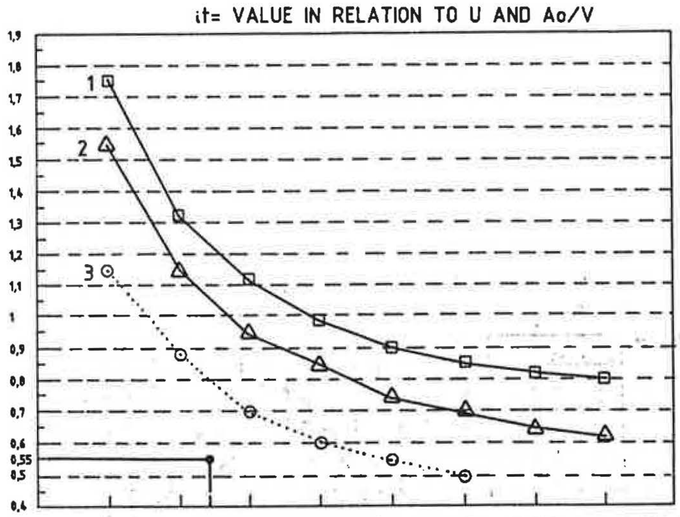


FIGURE 2 : DESIGN DAY THERMAL LOADS



1 = it DUTCH REGULATIONS (it=10)
2 = it RGD REGULATIONS (it=12)
3 = it DANISH REGULATIONS (it=15)

FIGURE 3 : THERMAL INSULATION VALUES

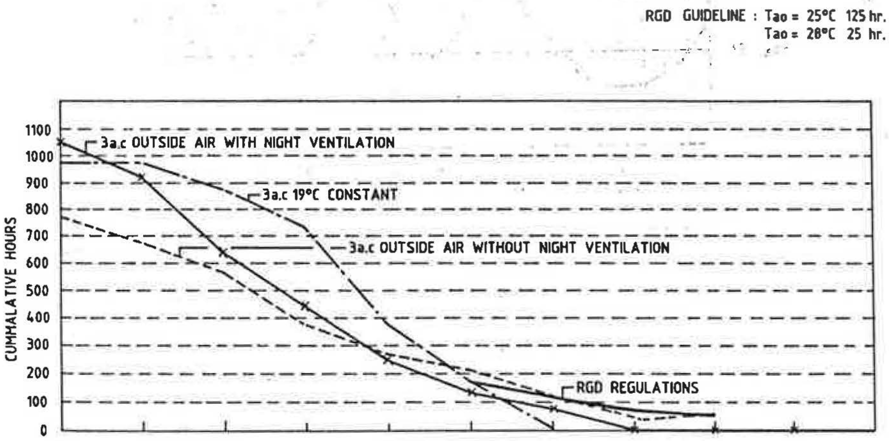


FIGURE 4 : SHOWS THE CUMMALATIVE FREQUENCY OF THE PREDICTED INSIDE AIR TEMPERATURE FOR 3 DIFFERENT SYSTEMS

INSIDE AIR TEMPERATURE (°C)

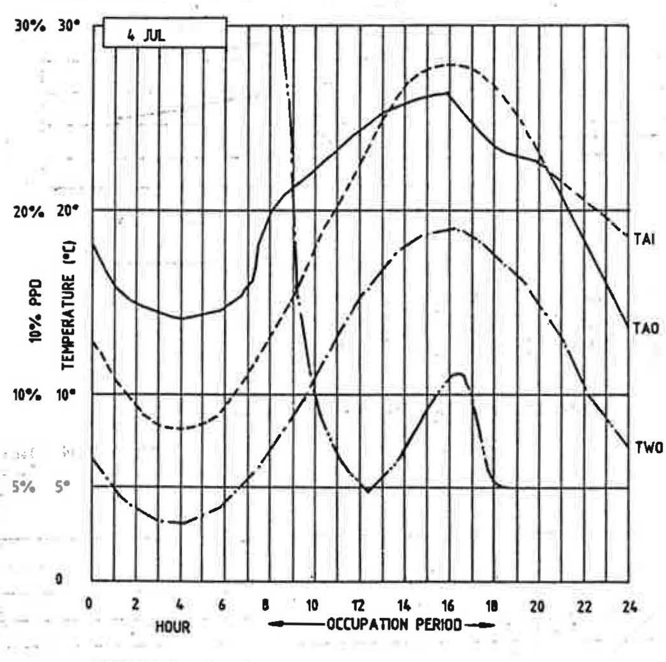


FIGURE 5 : COMFORT PREDICTIONS BY 28°C OUTSIDE AIR TEMPERATURE

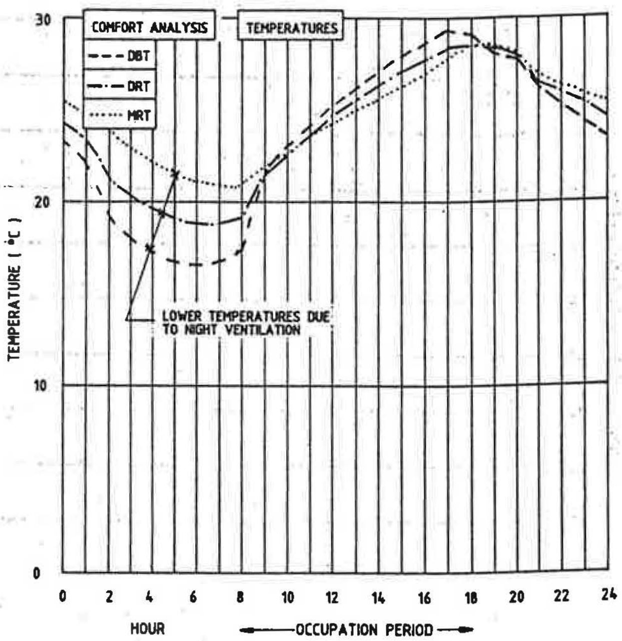


FIGURE 6 : DBT, DRT AND MRT CYCLIC TEMPERATURES SHOWING THE 28°C MAXIMUM ALLOWABLE INSIDE AIR TEMPERATURE

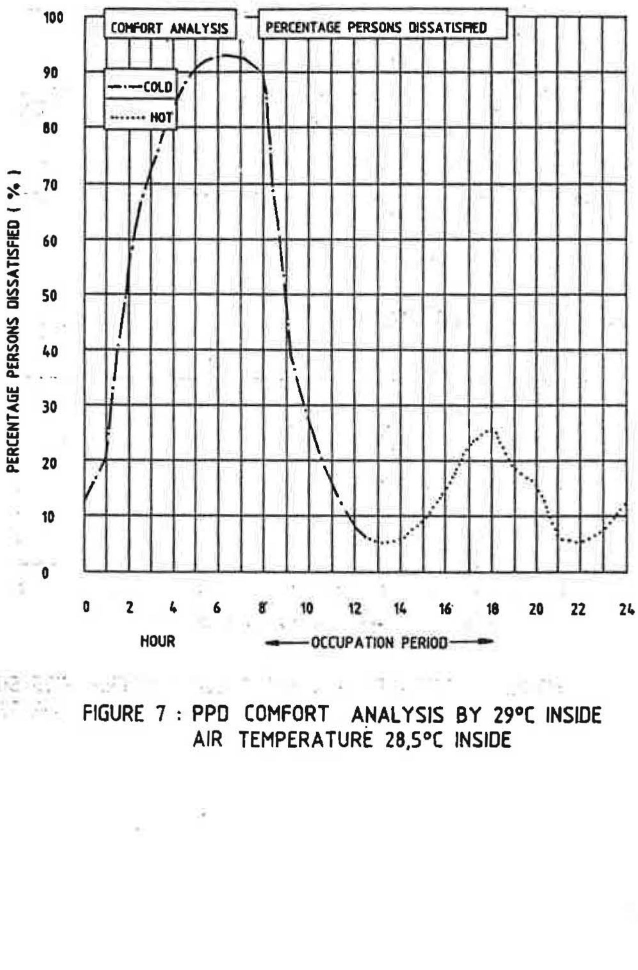


FIGURE 7 : PPD COMFORT ANALYSIS BY 29°C INSIDE AIR TEMPERATURE 28,5°C INSIDE

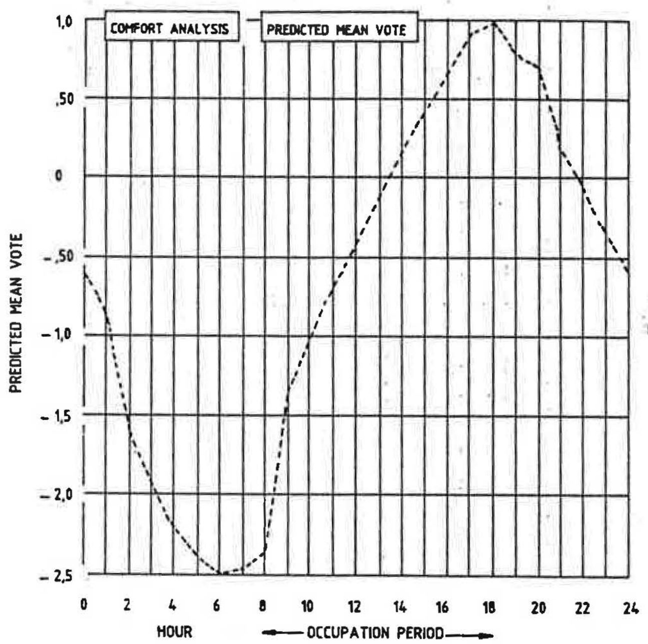


FIGURE 8 : PMV COMFORT ANALYSIS CYCLIC CONDITION WITH A 29°C INSIDE AIR TEMPERATURE

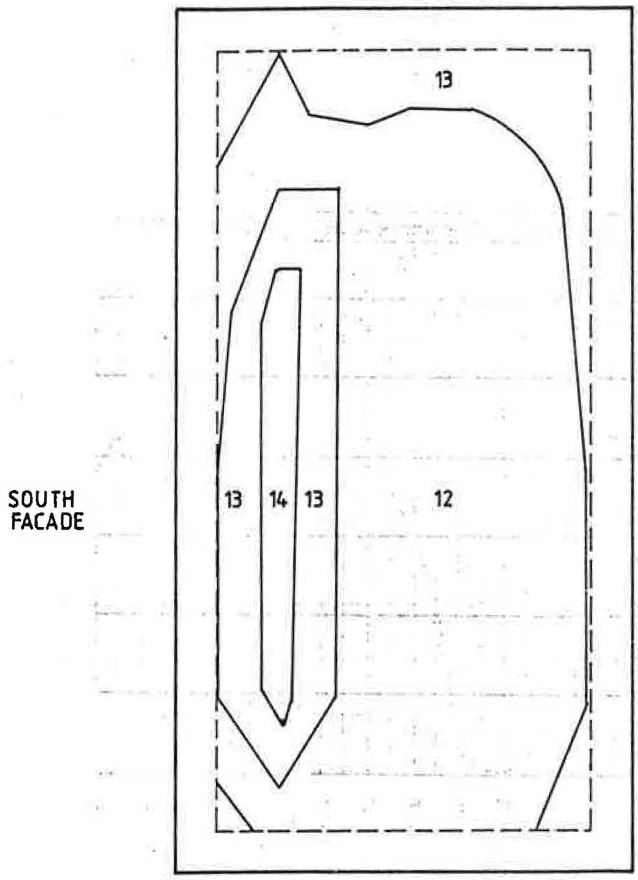


FIGURE 9: PPD COMFORT ANALYSIS
INSIDE AIR TEMPERATURE 26·4° C

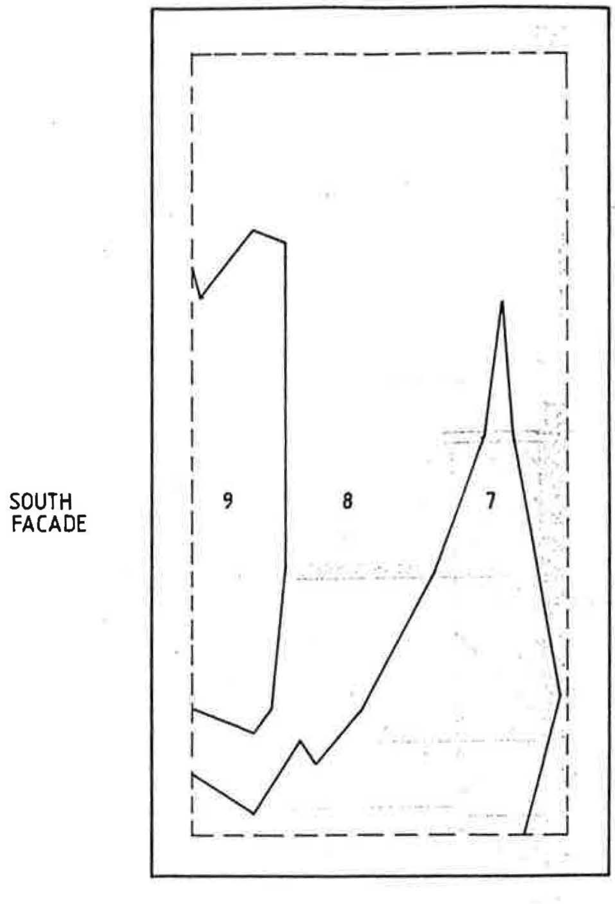


FIGURE 10: PPD COMFORT ANALYSIS
INSIDE AIR TEMPERATURE 25·4°C

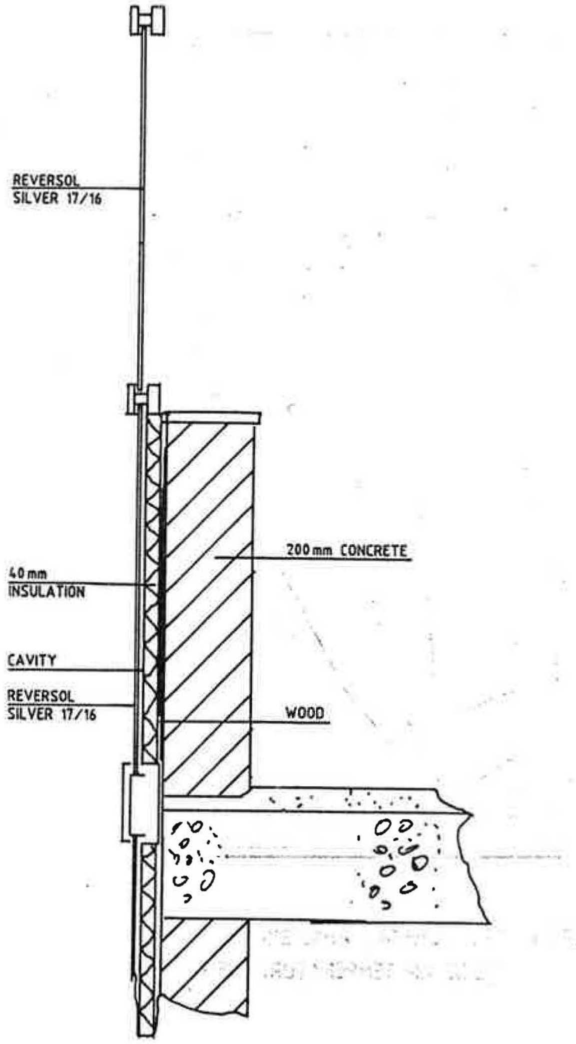


FIGURE 11: FACADE CONSTRUCTION

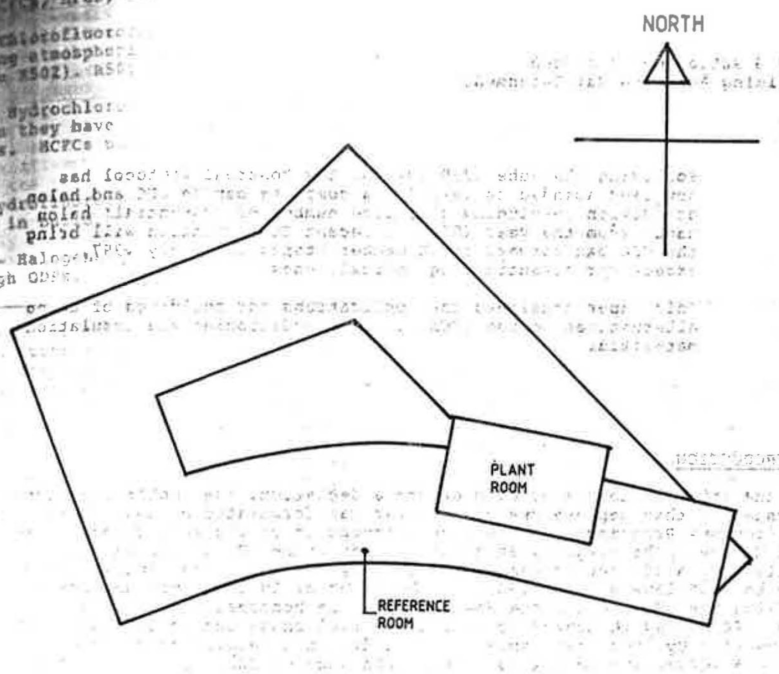


FIGURE 12: BUILDING SITUATION