THERMAL COMFORT AND OPTIMAL ENERGY USE

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ABSTRACT

The most important objective for a building is to provide the occupants with a comfortable indoor climate, and we want to achieve this as economically as possible.

Fanger (1970) deduced a "thermal index" that could express a subject's thermal sensation in a climate deviating from the optimum. Fanger assumed that thermal sensation is a function of the thermal load of the body. He 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 humidity were carefully controlled so that a thermal load could be calculated.

This work shows the results obtained when replacing dry-bulb temperatures with a predicted mean vote (PMV) calculation. The results also show that (1) the fluctuations of the PMV are (within limits) acceptable to the occupant or occupants and (2) under which circumstances these fluctuations will occur. Simulation techniques are used to determine building loads and control strategies. These control strategies are then optimized to minimize the operating cost (i.e., energy and demand charges) while controlling the HVAC system to meet comfort criteria in the occupied zone.

INTRODUCTION

An average modern office has an internal heat gain between 30 and 45 W/m². For an average working space of 10 m² per person, that is the equivalent of 400 watts. Heating is not always necessary. With a well-insulated building envelope, the internal heat production is enough to heat the office even when the outside air temperature is −5°C. Heating will only be necessary when the outside air temperature is less than −5°C and when the office is unoccupied. The mechanical ventilation system is still necessary since people working in offices with a high internal heat production still require fresh air.

Research was carried out to investigate the resulting comfort conditions that were simulated for four different building configurations (see Table 1). These simulation results proved to be successful in providing (theoretical) comfort conditions throughout the year. The control strategies for conditioning these comfortable spaces are analyzed.

Braun (1990) showed that an optimal heating, ventilating, and air-conditioning (HVAC) system operating strategy requires the best combination of temperatures and flow quantities so that the overall cost is minimized. The efficiency of the interaction between the building and its HVAC system while maintaining comfort conditions can be improved to enhance the dynamic performance and reduce peak loads.

As an example, consider a room in which the comfort conditions are to be controlled (House et al. 1991). The state equation is derived from the conservation of energy for the transient response of the room comfort conditions. The room comfort conditions (within limits PMV ±0.5) are the state variable, and the energy required to achieve the devised comfort conditions is the controlled variable.

The cost function would include the cost associated with the energy (Braun 1990; House et al. 1991). The thermal comfort of the occupants of the room is the primary objective, and the penalty for the room not being at the required conditions could be included in the cost function (Braun 1990). The objective of the optimal control scheme would be determined by the optimal trajectory of energy that yields minimum costs while maintaining thermal comfort conditions.

COMPLEXITIES OF THERMAL COMFORT CALCULATIONS

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

- dry-bulb air temperature,
- radiant temperature,
- air speed, and
- vapor pressure.

Such an analysis of the thermal environment requires a complete solution to the equations representing air movement and the thermal response of the building fabric under transient conditions. The benefit of using a dynamic

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## TABLE 1
Building Configurations Used for the Comparison

<table>
<thead>
<tr>
<th>ROOM DIMENSIONS</th>
<th>CBM</th>
<th>FGDP</th>
<th>Schiphol P4</th>
<th>Chipshol</th>
</tr>
</thead>
<tbody>
<tr>
<td>total floor area</td>
<td>25000 m²</td>
<td>2 x 8000 m²</td>
<td>65000 m²</td>
<td>10000 m²</td>
</tr>
<tr>
<td>length</td>
<td>5.4</td>
<td>5.4</td>
<td>5.4</td>
<td>5.4</td>
</tr>
<tr>
<td>width</td>
<td>3.6</td>
<td>3.6</td>
<td>3.6</td>
<td>3.6</td>
</tr>
<tr>
<td>height</td>
<td>3.6</td>
<td>3.6</td>
<td>3.6</td>
<td>3.6</td>
</tr>
<tr>
<td>lowered ceiling height</td>
<td>2.7</td>
<td>2.7</td>
<td>2.7</td>
<td>2.7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ORIENTATION</th>
<th>window orientation</th>
<th>south</th>
<th>south</th>
<th>south</th>
<th>south</th>
</tr>
</thead>
</table>

| OCCUPATION | | | | |
| occupants per room | 2 | 2 | 2 | 2 |
| occupancy period | 8:00 to 18:00 | 8:00 to 18:00 | 8:00 to 18:00 | 8:00 to 18:00 |

| INTERNAL LOADS | | | | |
| lighting (W/m²) | 8 | 15 | 9 | 15 |
| occupants (W/m²) | 8 | 10 | 8 | 10 |
| equipment (W/m²) | 10 | 7 | 18 | 20 |
| total (W/m²) | 26 | 32 | 35 | 45 |

| CONSTRUCTION | | | | |
| exterior wall inside | concrete 200 mm | concrete 200 mm | concrete 200 mm | concrete 230 mm |
| min. wool | 50 mm | insulation | 100 mm | min. wool | 70 mm | min. wool | 50 mm |
| cavity | 50 mm | concrete | 100 mm | cavity | 250 mm | cavity | 250 mm |
| concrete | 100 mm | steel plate | 2 mm | concrete | 150 mm |
| K-value (W/m²K) | 0.5 | 0.35 | 0.44 | 0.5 |
| roof | screed | 20 mm | screed | 20 mm | screed | 20 mm | screed | 20 mm |
| insulation | 80 mm | insulation | 80 mm | insulation | 80 mm | insulation | 80 mm |
| concrete | 220 mm | concrete | 250 mm | concrete | 220 mm | concrete | 200 mm |
| K-value (W/m²K) | 0.4 | 0.4 | 0.4 | 0.4 |
| floor/lowered ceiling | carpet | 5 mm | carpet | 5 mm | carpet | 5 mm | carpet | 5 mm |
| concrete | 220 mm | concrete | 220 mm | concrete | 220 mm | concrete | 200 mm |
| cavity | 660 mm | cavity | 660 mm | cavity | 660 mm | cavity | 700 mm |
| tiles | 10 mm | tiles | 10 mm | tiles | 10 mm | tiles | 10 mm |
| K-value (W/m²K) | 1.71 | 1.71 | 1.71 | 1.71 |

| GLAZING | | | | |
| glazing percentage | 32% | 28% | 43% | 36% |
| type | climate window | Reversal silver | climate window | IPN |
| K-value with vent. | 1.54 | 1.65 | 1.7 |
| K-value without vent. | 2.02 | 3.3 | 1.9 | 2.0 |
| solar transmission | 19% | 37% | 15% | 20% |
| light transmission | 32% | 27% | 32% | 25% |

| HEATING/VENTILATION | | | | |
| infiltration (a.c./h) | 0.5 | 0.5 | 0.5 | 0.5 |
| mechanical vent. (a.c./h) | 2 | 2 | 2.5 | 2.5 |
| supply air temp. summer | 17 | 17 | 17 | 17 |
| supply air temp. winter | 20 | 20 | 20 | 20 |
| type of heating | air | radiator | air | radiator |
| secondary cooling | VHV unit | VHV unit | VHV unit | VHV unit |
| IT value | 17 | 16 | 18 | 17 |
simulation tool is that complex peak conditions can be modeled that would not be possible with traditional design methods. In order to give the designer a complete picture of a building's thermal characteristics, the simulation tool must sufficiently model the dynamic effects of the building and HVAC systems. Heat is transferred by conduction through the building envelope and partition walls; by convection, causing heat exchange between surfaces and air; and by radiation in the form of solar transmission through transparent parts of the building envelope and in the form of long-wave radiation exchange between surfaces, each of which must be considered separately. The calculation of radiant form factors is essential to the assignment of radiant heat flow. Conduction through the enclosure surfaces is calculated dynamically with time steps adjusted by the model. The program takes account of the spatial arrangement of the surface and the way each surface affects others, which makes it possible for the sunlight falling on each surface to be treated separately. The program also takes into account the effects of humidity and room temperature on the heat given off by the occupants, as well as the level of activity.

For comfort analysis, the model carries out an analysis of the radiant field at each point of analysis in the occupied zone. A person is assumed to exist at each point, and a heat balance can be established at the surface of the person. Angle factors are used to evaluate the surface/subjects radiant exchange.

At this stage, only long-wave radiation from normal low-temperature surroundings are 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 surroundings, due to the high radiant source temperature).

The effects are considered using the theory developed by Fanger (form factors). 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 point of analysis is not shaded.

In order to study the effect of weather on dynamic building comfort conditions in a systematic manner, a weather generator was used to produce diurnal variations in ambient dry-bulb, wet-bulb, and solar radiation based upon daily statistics. The advantages of this approach over the use of a standard reference year is that it allows for systematic parametric studies of the effects of weather on dynamic comfort control in terms of simple statistical parameters. The temperature, humidity, and solar radiation are the daily average temperature and solar turbidity. It is also necessary to evaluate the incident radiation on the exterior building surfaces. This requires knowledge of components of orientation and ground reflectance properties.

Limitations of Comfort Analysis

Fanger's 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 that 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 wear a light suit if he or she was at work).

These characteristics mean that the level of comfort expected in a working environment is probably tougher than the level of comfort expected in a leisure environment. So, when analyzing a space that is not a workspace, 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.

Complexities of Building Load Calculations

As mentioned before, there are several factors that influence the results. Until recently, it was thought that ambient conditions and solar radiation largely influenced the baseline loads. The corresponding architectural design of a building can influence building loads considerably. Temperature effects (swing) can be influenced by building constructional design. The configuration of perimeter walls and the percentage of glazing in the external and internal walls are very important. Usually the glazing represents a weak point for both heating and cooling loads, and special attention must be paid to their constructional design. The U-value, the transparency, and the reflectance are features that have considerable effects on the cooling load. It must not be forgotten that the qualities that are desirable for summer conditions (low transparency and low reflectance) lessen the heating load during the winter.

Dynamic Optimization

Braun (1990) provided a methodology for dynamic optimization that was adapted for this research. The optimal control of a system takes advantage of the harmony of the building, its architecture, its materials, and its systems, which involves minimizing the integral of operating costs over a specified period of time (e.g., a day) while satisfying the required comfort conditions (PMV ±0.5). The operating costs at any given point are equal to the product of the power consumption of its systems (heating and cooling) and the costs of fuel (gas or electric).

The constraints include both required comfort conditions for the building (PMV ±0.5) and limits on the operation of equipment (e.g., capacity). The optimal solution is a trajectory of control throughout the specified optimization period. The control variables include some setpoints derived from PMV conditions, VHV coil dis-
charge air temperatures, and all plant control (e.g., chilled-water temperature and pump speeds).

The approach used in this study is practically identical to that originated by Braun (1990) and involves discretizing the cost function in time and applying a nonsmooth optimization algorithm to determine the set of controls that minimize the sum of costs over the specified time. The computer requirements necessary to solve this problem are quite complicated; this study differs from that described by Braun (1990) in that the plant and building are not decoupled during analysis.

Mathematically, the optimization problem is stated as follows:

\[ J = \sum_{k=1}^{K} \left( R(k) + P^* \left( \text{PMV}_z(k), f(k) \right) \right) \]

for all \((k)\) with respect to \(\text{PMV}_z(k)\) \((k = 1, K)\) subject to \(\text{PMV}_{z_{\text{min}}}(K) \leq \text{PMV}_z(K) \leq \text{PMV}_{z_{\text{max}}}(K)\).

System Characteristics

In this study, a low rate of primary ventilation air (two air changes per hour [ach]) is incorporated with VHV units (Figure 1). Fresh air is supplied into the space via slot diffusers mounted in the lowered ceiling. Any occupants and their accompanying equipment displace unwanted heat clear of the occupied zone of this space. Mechanical cooling is based on a thermal storage system, with perimeter heating based on LTHW obtained from recovered sources. Where possible, this study assumed that additional cooling, needed to counteract internal loads, was gained from using VHV units running on tepid water.

VHV units together with low-ach primary ventilation can combine to produce an alternative to conventional forms of air conditioning, with the emphasis on getting the highest air quality in the workspace. Investigations have concluded that supplying air at low velocity will create a pool of fresh air that will pour evenly through the space. When this air comes into contact with heat sources, such as equipment and people, it will rise up, ventilating and removing heat. This air then collects at a high level, is then cooled by the VHV unit, and enters into the room where its physical properties circulate the air further through the room. The amount of secondary circulation is dependent upon the temperature differential between the air in the lowered ceiling and the air in the VHV unit. With this system, the circulation rate increases as the heat load in the space increases.

The variable function of the VHV unit can be expressed as follows.

The room temperature, \(Y_1\), is a function of \(Y_1 = 0.019x + 19.2\).

\[ \text{Figure 1 } \quad \text{RTB Van Heugten - Climate control for offices VHV - system.} \]
Therefore, when the dry-bulb temperature in the room is either measured or predicted, \( x \) is substituted into

\[
Y_2 = 0.026x + 19.1
\]

and

\[
Y_3 = 0.0075x + 18.875.
\]

The air movement provided by the VHV unit can be shown as

\[
Q_a = m_4 \cdot \rho \cdot c_p \cdot (Y_2 - Y_3),
\]

where \( Q_a \) is the sensible cooling load of the VHV unit, and \( m_4 \) can then be substituted into

\[
(m_1 \cdot t_1) + (m_2 \cdot y_2) = (m_3 \cdot t_3).
\]

Using these simple equations, a “quasi”-variable-volume strategy is included in the simulation program.

For each thermal zone, the following form is utilized for estimating the sensible cooling load of the zone to maintain a specified comfort condition at any stage (k):

\[
Q_{z,k} = \sum_{K=1}^{\infty} \left[ T_{z,k-1} + PMV_{z,k-1} + Q_{g,s,k-1} + Q_{s,k-1} - Q_{z,k-1} \right] + \sum_{K=1}^{\infty} Q_{z,k-1}
\]

where

\[
Q_{z,k} = \text{sensible cooling requirement for stage } k,
T_{z,k} = \text{ambient temperature for stage } k,
PMV_{z,k} = \text{zone comfort condition setpoint for stage } k,
Q_{g,s,k} = \text{total sensible internal gains (e.g., lights, people, equipment) for stage } k,
Q_{s,k} = \text{total incident solar radiation on all exterior zone surfaces for stage } k,
N = \text{number of stages in the day (e.g., 24 for a one-hour interval}).
\]

Figure 2 shows a schematic of the system, derived from previous research, which can be explained as follows. A central constant-volume air-handling unit has a thermal wheel for energy recuperation purposes, filters, a combined heating/cooling coil, a ventilator, and, in most cases, an electrically driven steam humidifier. Two of the variants have reheat coils for zone heating, while the other variants have perimeter heating by means of radiators (see Table 1).

**Heating**

The offices are kept at a 20°C minimum; outside working hours, the temperature in the building is not allowed to fall below 18°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 that 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 conditions 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 an oversize ratio or margin is used, any saving would be more than offset by the reduced part of the plant. In addition, a larger system will have more heating surface and an increased boiler capacity, resulting in a higher capital cost. The discussion now returns to the optimization condition usually sought in the intermittent heating of buildings.

The optimal strategy consists of two elements. The first element is an optimal preheating strategy, which brings the building temperature up to the required target just as the period of occupation begins. At this time, the heating system is likely to be at a higher temperature than that required for steady state.

The second element is an optimal strategy that brings the system down to its steady condition. This latter strategy consists of neutralization at zero power where the building temperature is maintained by the surplus heat in the heating system.

A varying external temperature does not affect the principle that periods of full power and zero power are the only admissible strategies. However, the optimal strategy is now only defined uniquely if the temperature variation is known in advance. This means in practice that the control has to undertake a strategy that may be non-optimal. For example, one may suppose that the external temperature will usually rise during a preheating period, following overnight shutdown. Consequently, a control set the preheat schedule in motion on the assumption that the instantaneous-ly recorded external temperature would remain constant right up to the beginning of the occupied period would be adopting a “maximum” strategy.

The most consistent form of this strategy would permit the control to reappraise its decision continuously. Insofar as the assumption of a rising external temperature is true, this control will produce a minimum energy consumption without ever breaking the temperature constraints (but it could exceed the comfort limitations).

The internal loads are sufficient to heat the space when the outside temperature is −5°C or more. But the internal load is only present during the period of occupation and cannot be included in any preheat strategy. Both variations, which have spaces heated by radiators, do overheat at the beginning of the period of occupation. This is due to the overheat capacity (time constant) of the radiator and the nearly instantaneous internal load. A thermostatic terminal heating control will perform better, but the initial overheat-
ing caused by the preheat will result in delaying the heat pulse required under optimal operation.

At the beginning of the occupancy period, when the space is heated by reheat coils in the ventilation system, air to be supplied at about 38°C to provide heat to the space can be rapidly lowered to counteract the internal load. The only overheating that could occur is due to radiative heat from the building mass.

COOLING

Computer simulation and measured results from buildings show that from April to October, cooling will be necessary. Because the prevailing cooling load is largely the result of internal loads, it can be assumed that these loads are reasonably constant. Cooling is provided by means of a thermal storage system, which has a series chiller (see Figure 3). In the event of extreme ambient conditions, an increase in the return water to the cooling system is used to implement the chiller in series. This chiller then lowers the water temperature being supplied to the thermal store, ensuring that the store provides chilled water at the required temperatures.

Because the load is kept nearly constant, the economics of the thermal storage system are improved. There are no part-load characteristics to be dealt with. The only part of the system that runs on part load is the supplementary

Figure 2   Schematic free cooling.

Figure 3   Thermal storage and chiller in series.
chiller, but, as this is only a part of the total load, economizing this system is economically not justifiable. A chiller with a good part-load characteristic must be sought.

Free Cooling Mode

During the periods where the cooling loads (internal and external) are higher than heat losses from the space, the space air temperature will be permitted to rise (within comfort limits). The complete PMV equation is used constantly to ensure the comfort limits are maintained. If cooling is required during this period to maintain the space within comfort limits, this will be provided by the VHV units. Figure 2 shows how the VHV is connected to the mixing pipe of the combined heating/cooling coil. Convective heat is extracted from the room by the VHV units; water running through the VHV units is then heated from 13°C to 17°C. This 17°C water is then mixed with 13°C water returning from the combined coil. The VHV water is then cooled to 13°C; the return water from the combined coil is a constant 13°C. The exiting air temperature from the combined coil is also controlled at a constant 13°C. This primary ventilation air can then be heated to 17°C, by means of a reheat coil, from recovered sources. A three-way control valve is used to control the 13°C supply water temperature to the VHV units.

Chilled Water Production

In series systems, the cool energy is a function of the applied load and the magnitude of the load profile of the anticipated diurnal cooling energy (Figure 4). Based upon this profile, which is calculated by the dynamic simulation, it is possible to size the TES and the chiller. For the purpose of comparing the results, forecasts must be made with known load and weather conditions.

Minimum Energy Charges

The discrete form of the general optimization problem for minimizing energy changes over a day may be expressed as

$$J = \sum_{k=1}^{N} R(k) \cdot P[x(k), u(k), k] \cdot Dt$$

with respect to $u(k)$ for all discrete time stages $k$ subject to

$$x(k) = f[x(k-1), u(k), k];$$
$$x_{min}(k) \leq x(k) \leq x_{max}(k);$$
$$u_{min}(k) \leq u(k) \leq u_{max}(k).$$

Note: The stage interval is one hour, so $n = 24$.

Ice-Priority Control

The aim of ice-priority control is to melt ice as much as possible during the day (Braun 1992; Kirshenbaum 1991) in response to the building’s internal loads. The advantage of ice-priority control is that it maximizes the ability of ice storage systems to fulfill their first operating purpose—to reduce a building’s peak electric demand charges. The disadvantages of ice-priority control are that the control equipment and sequences may be more complicated than chiller-priority control; that the total tank volume may be greater for a given ice discharge rate; and that if controls are not properly designed, the ice can be exhausted too early on a hot day, resulting in an inadequately cooled building.

Another advantage of ice-priority control is that it forces more use of the TES chiller at night in the ice-making mode.

Because the intent of ice-priority control is to melt as much ice as possible and because the series chiller is only sized to meet part of the building’s peak load, care must be

![Figure 4](image-url)  
**Figure 4**  
Anticipated load profile of the diurnal cooling energy for a single room.
taken that the ice is not exhausted prematurely on a hot day, leaving inadequate chilled-water capacity to meet the rest of the load. (This implies that it is necessary to predict the required load.)

The simplest method of controlling a TES system with ice is to have the series chiller operate in response to the temperature of the chilled-water supply returning to the building and, consequently, the chilled-water supply temperature (Figure 3).

The water is chilled at 6°C for supply and 14°C for return. The return water temperature of 14°C is a result of mixing the 12°C return water from the air-handling unit and the return water from the VHV units at 17°C. The chilled-water systems are controlled by two-way valves, and the chilled water is distributed by variable-speed pumps. During part-load conditions, the return water temperature could rise to about 18°C. A three-way mixing valve across the heat exchanger of the TES ensures a constant chilled-water supply temperature of 6°C (the flow over the heat exchanger is also variable). When the three-way valve is fully open, the pump is at full speed, and the chilled-water temperature rises above the 6°C supply temperature, the series chiller is then activated.

This chiller is designed to cool the return water from 14°C down to 10°C. In this manner, the chiller will have a smaller temperature differential, which results in a smaller capacity. Because the chilled-water leaving temperature is 10°C, then the evaporating temperature can be about 7°C, which will improve the coefficient of performance (COP) and the performance of the chiller. The condensing temperature will be about 47°C, which is 15 K above the maximum ambient temperature of 32°C. When the cooling load decreases, the chilled-water temperature will drop below the required 6°C, the series chiller will stop operating, and the TES will continue to supply 6°C chilled water. It is, however, advisable to use a chiller with a good part-load characteristic (a screw compressor, for example). Figure 5 shows the COP values for both TES with a series chiller and a conventional chiller.

The governing rules for the system can be described as follows. If the outdoor air temperature is lower than 16°C and there are no internal loads, then cooling is unnecessary.

**Rule No. 1**

If \( T_{out} < 16°C \) and \( T_{in} < 23°C \), then no cooling is needed (between 18°C and 23°C zero band).

The second rule is governed by the internal loads at 21°C room temperature. The temperature in the ceiling plenum is about 22°C, and the VHV unit mounted in the ceiling will lower the temperature to about 19°C, which is about 200 watts of sensible cooling entering the room. Because the VHV unit has a linear characteristic, the amount of cooling provided is proportional to the load (i.e., temperature in the ceiling plenum) when the maximum room temperature of 25°C (as shown in the simulation results) is attained. When the temperature in the ceiling plenum is 27°C, the VHV unit cools this down to 20°C, which is supplied at 260 m³/h back into the room (sensible cooling of 600 watts).

However, simulation results have shown that during the winter period (maximum PMV in the room), cooling will still be necessary. (The internal heat loads exceed the heat losses of the building.) Due to the configuration of the VHV units and the air-handling unit, as shown in Figure 2, the VHV units can be run in the free cooling mode.

So the first rule can be modified as follows:

If \( T_{out} < 16°C \) and \( Q_{int} > Q_{losses} \) and \( t_{room} > 23°C \), then cooling is needed.

This rule is applicable until the outside air temperature exceeds 16°C. Simulation results show that this rule is applicable for six months of the year (see Figure 4).

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**Figure 5** COP values for a conventional chiller and a TES priority with secondary series chiller.
Rule No. 2

If $T_{in} > 16\degree C$ and $Q_{in} > Q_{tension}$, then $T_{in} = 16\degree C$; otherwise, rule 1 applies.

If $Q_{in} > TES$, then TES needs a chiller.

RESULTS OF THE SIMULATION

Figures 6 and 7 show the simulation results for two periods, one in winter, the other in summer. The results show that the PMV is within acceptable limits. The corresponding indoor air temperature was then used to investigate an optimal control strategy to maintain these conditions.

Figure 6 shows both indoor air temperatures and PMV values for winter conditions. During the winter periods, both CBM and Schiphol had lower inside air temperatures; the variations were heated by air, as both had climate windows. The other two tended to overheat by about 1 to 2 K. This was due to the response time of the radiators; because of the high internal gains, the optimum start algorithm simulated could not cope with the pending internal load. The reason for this was that the simulation model targeted its heat requirement at $20\degree C \pm 1 K$ at 08.00. At 08.00, the computer simulation model proceeded to "switch" on the internal load. The room thermostat then proceeded to switch the heating off, but the radiator kept on supplying heat for some time after this signal. This phenomenon resulted in an activation of the VHV unit, which
meant that the convective cooler was compensating for this overshoot.

Experiments were carried out at optimal start times for the plant, but these proved to be insecure due to two main reasons.

1. When the system was switched on to provide an indoor air temperature at approximately 07.00, the heating plant was then shut down and left to coast until 08.00, when the internal loads (lighting, occupants, and occupants’ equipment) would provide internal heat. But heat provided by the radiators was absorbed by the building fabrics, which, in turn, increased the mean radiant temperatures and the PMV, decreasing comfort conditions.

2. The second reason was the occupancy profile, bearing in mind that the office being simulated was only one of about 200 in the actual building. In practical terms, this meant that everybody would have to begin work and switch their equipment on at the same time, every day of the week, etc., which, of course, does not occur in an actual office building.

Because the heating was switched off, cooling during the winter was not viewed as optimal use. This did not really worry us because we were running in the free cooling mode.
mode so that the costs were virtually zero. It did, however, prove that individual room control could be provided in a very simple manner.

The rooms that were simulated with climate windows were cooler than the rooms with glazing and radiators due to the convective heating of the air. Because the resulting room temperatures were within design limits, no heating or cooling was required. Heating would be necessary if the internal loads were lower (if the offices were unoccupied, for example). The PMV results for these two variants were lower than the models with radiators because there was no direct radiation compensation available.

The amount of energy required to produce the results shown in Figure 6 is less than a conventional control strategy, mainly due to the lower temperature at the beginning of the occupation period. A conventional control system would be targeted at, say, 20°/21°C at 09:00, whereas these simulation results show temperatures that are 2 or 3 K lower. This obviously consumes less energy while still maintaining the required comfort criteria.

If cooling is needed during the winter, it can be provided by the free cooling between the outside air and the VHV units. The only penalty for this cooling is the electrical energy needed for the pumps. The dip is caused by the reduction of heat output from the radiators, which is replaced by the internal gains (equipment, lighting, etc.). Figure 6 shows that the comfort conditions have PMV values of about -0.2, which indicates that the space will feel cold rather than warm. The cold effect is created by the negative radiation from the windows and because the internal heat sources are more convective than radiative (equipment, lighting, etc.).

Figure 7 shows the summer temperatures and PMV values of the four variants. Both variants simulated with climate windows had a better temperature profile. The PMV values were all within the limits of -0.5 to +0.5. All simulations were carried out with a maximal outside air temperature of 32°C (a very warm period), a total solar intensity of 800 W/m². Direct solar intensity was 650 W/m² with zero cloud cover. The VHV unit proved to be very successful. The self-regulating convective cooling capacity needed no optimal operation algorithm. As the convective heat gain increased, the amount of air moved by the cooling effect increased and subsequently decreased when the load decreased. The combination of variable load and circulation rate resulted in a comfortable indoor environment throughout the day. Temperature fluctuations were minimal. Because both occupancy and ambient conditions cannot be predicted, a certain amount of flexibility must be included in plant dimensions and control strategy.

Figure 7 shows temperatures at around 25°C for three of the simulations. The temperatures for the Schiphol simulation fluctuated around 23°C. The energy required to create these comfort conditions is less than for a conventional strategy, mainly due to the fact that the inside air temperatures are higher than with conventional systems. Because the VHV units are quick to react to any thermal fluctuation in the space, no unnecessary energy is wasted. The VHV units also provide individual comfort control, which again adapts to building requirements. The COP values shown in Figure 5 for the cooling plants show that the configuration of TES with a secondary series chiller uses less energy than a conventional chiller.

CONCLUSIONS

The object of this research was to investigate the operation of a comfortable inside climate according to the Fanger thermal index. A technique has been derived for the operation of the building’s plant, based upon a knowledge of the various building interactions. The rule-based control strategy was devised from the dynamic program used for the simulation of the plant interaction. It provides a useful means for the operation of a building and its plant under transient conditions. The results show how the VHV units with low-ach primary ventilation can combine to produce an alternative to conventional forms of air conditioning, with the emphasis on obtaining the highest comfort levels in the workspace.

The results show that supplying air at low velocity into a space will create a pool of fresh air that will circulate evenly throughout the space. When this air comes into contact with heat sources, such as equipment and people, it will rise up, ventilating and removing heat. This air is then collected at a high level, cooled by the VHV unit, and ventilated into the room, where its physical properties circulate the air further through the room. The amount of secondary circulation is dependent upon the temperature differential between the air temperature in the lowered ceiling and the (surface) temperature of the VHV unit.

Another advantage of this system is that individual zone conditions can be created. If a particular office or part of an office had a lower internal load than simulated, the office temperature would be lower and still remain within comfort limits. The convective heat gain would be minimal; therefore, the VHV unit would not be activated and, consequently, no ancillary cool energy would be required. The system is self-regulating, without a complicated control strategy. The primary ventilation system is a constant-volume system supplying only minimal ventilation air into the space. Therefore, the ductwork and air-handling units are smaller and cheaper than for variable-volume air-handling systems. This work also shows that the control strategy for comfort control is simpler and can be easily optimized to provide minimal energy use. Operating a space under comfort conditions, not temperatures, clearly shows that the energy necessary to maintain the required conditions is less than with conventional systems.

When using thermal storage systems, the operation of the total system must be taken into account. Series systems have shown that the physical circuiting of the TES and chiller can have a dramatic effect upon the rate of storage depletion and the complexity of the control strategy required to maintain optimal performance. Unless some care is taken

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in maintaining the proper proportion of heat removal from each component, it is possible to either prematurely deplete the store or not completely utilize it. The series arrangement offers significant advantages over parallel systems because it tends to operate more efficiently and is simple to control. This simple control also enhances the ability to fine-tune the system, adding to the overall efficiency.

NOMENCLATURE

\[ Y_2 = \text{dry-bulb temperature in the ceiling void} \]
\[ Y_3 = \text{dry-bulb temperature under the VHV unit} \]
\[ m_1 = \text{volumetric flow rate of the 2-sch primary ventilation air} \]
\[ t_1 = \text{dry-bulb temperature of the primary ventilation air} \]
\[ m_3 = (m_1 + m_2), \text{the total circulation rate of air in the space} \]
\[ t_2 = \text{dry-bulb temperature of the circulation air in the space} \]
\[ Q_{ek} = \text{sensible cooling requirement for stage k} \]
\[ T_{a,k} = \text{ambient temperature for stage k} \]
\[ PMV_{z,k} = \text{zone comfort condition setpoint for stage k} \]
\[ \dot{Q}_{e,s,k} = \text{total sensible internal gains (e.g., lights, people, equipment) for stage k} \]
\[ Q_{s\text{ol},k} = \text{total incident solar radiation on all exterior zone surfaces for stage k} \]
\[ J = \text{total cost of the operation over N stages in a day} \]
\[ D_k = \text{time step associated with each stage and for each stage k} \]
\[ P^* = \text{average power consumption of the cooling system} \]
\[ C = \text{cost of electricity} \]
\[ \alpha = \text{state of storage at the end of the stage} \]
\[ u = \text{rate of charging for storage} \]
\[ x_{\text{max}} = \text{maximum admissible state of storage} \]
\[ x_{\text{min}} = \text{minimum admissible state of storage} \]

\[ u_{\text{max}} = \text{maximum charging rate} \]
\[ u_{\text{min}} = \text{negative of the maximum discharging rate} \]
(i.e., a negative rate of charging).

REFERENCES


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