Optimal Control of Fabric Thermal Storage Systems

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Abstract

Hollow core ventilated slab systems provide an effective means of utilizing the building structure as a thermal store to reduce energy cost while maintaining the thermal comfort of the occupants. The optimum control strategy for the system would be one that minimizes energy use without prejudicing the thermal comfort requirement. This paper describes the characteristics of the optimal control problem and its potential for reducing energy cost.

The optimization study was undertaken by a numerical optimization method known as Genetic Algorithm (GA) applied to a dynamic building model and static plant model. The formulation of the control problem is discussed and the results illustrate the need for optimum control and that the GA is a viable method for investigating the characteristics of the control problem. A simplified control strategy is developed that features the optimal control characteristics.

1 Introduction

Thermal Energy Storage systems provide a promising approach to reducing building energy costs and to restricting the emission of environmental pollutants. The potential for storing thermal energy within the structure has been exploited by building engineers since the 1970's. A simple approach to utilizing the building fabric as a thermal store is to expose as much of the internal surface of the structure as possible, by for instance having no false ceiling. This can be effective due to the increased heat transfer between the building fabric and the room environment. However, the thermal capacity of the mass is under utilized due to the weak thermal coupling with the room air. Hollow core ventilated slab systems are a fabric thermal storage system that actively utilizes the thermal capacity of the building mass. This paper investigates the optimal control of a hollow core ventilated slab system.

In effect, ventilated hollow core slabs act as regenerative heat exchangers. During the summer, cool air is used to lower the temperature of the slab at night, so that the slab can absorb the heat from the higher temperature outside air during the day. In the winter, the slab can be used to store heat supplied from off-peak tariff heating, to be released during peak periods. The effectiveness of the systems has been previously investigated by monitoring real buildings and by computer simulation [1,2,3,4].

1.1 Conventional Control of Ventilated Slab Systems

In the conventional control of hollow core ventilated slab systems, it is usual to monitor the slab mass and or the room air temperature and to use these to dictate the time switching of the plant operation. A building at the University of East Anglia (UEA), UK, has been installed with a hollow core ventilated slab system. The supervisory control strategy for this building during the summer is described by the following rules [5]:

- night operation:

1. if during the day $T_o > 15\ ^oC$ or at 10:00pm $T_{az} > 23\ ^oC$ and $3\ ^oC < T_{az} - T_o < 8\ ^oC$, then the ventilation fans are switched ON;
2. if the fans are ON and $T_s < 21\ ^oC$ or $T_o < 8\ ^oC$, then the fans are switched OFF.

- operation during occupancy:
  the fans are ON to provide full fresh air ventilation at a constant rate in a range 20-40 l/s (depending on the zone type).

T_o is the ambient air temperature, T_{az} is the zone air temperature and T_s the slab mass temperature.

These rules may not fully utilize the potential of structural mass capacity and sometimes may waste energy, for instance, it may not be necessary to start free night cooling as early as 10:00pm to relieve the cooling load the following day. The minimization of fan operating cost can only be achieved by the optimum control of fan operating times and ventilation rates. In the presence of an electricity tariff structure, the optimal control is especially preferable since it can provide the optimum ventilation for both the night and day operation by taking account of the varying cost of electricity.

1.2 Optimal Control of Building Thermal Systems

A significant amount of research has been conducted on the optimal control of building thermal plant [6,7,8,9,10,11,12,13,14,15,16,17,18], including the optimal control of ice thermal storage systems [19]. Less research has been focused on ptimizing plant operation with building fabric thermal storage. Braun [20] presented the potential of energy cost savings through the optimal control of fabric thermal storage.Though the study was of a conventional building, the thermal capacity of the building mass was utilized for relieving the daytime cooling load. The approach adopted was to formulate the zone temperature setpoints over a 24 hour period as optimization variables and to use these to determine the setpoints that minimized the total energy cost while maintaining a pre-defined level of thermal comfort. The Complex search method was used for solving the optimization problem. The approach adopted in this paper can be generalized for the applications of the optimal control of fabric thermal storage systems. Morris, et.al [21] have also tested the approach of dynamic optimal control of the building thermal storage and compared it with night setback control.

Little work has been done on the optimal control of hollow core ventilated slab systems. Due to its special air supply path, the control of this system differs from that of conventional buildings. This paper investigates the optimal control of the hollow core ventilated slab systems for summer operation. The control characteristics is presented and the potential for reducing energy costs is revealed under various operational conditions and in comparison with the conventional control method. Also presented is a simplified control strategy of the system (`time-stage control'), which has been derived from the optimum control characteristics.

2 Control Hierarchy and Variables

The system considered in this paper is shown in Figure 1. It contains three components: building with installed ventilated slabs, the primary plant and an air-handling unit (AHU). The primary plant consists of a chiller. The AHU includes a heat recovery device (HRD), a cooling coil, an electric duct heater, supply fan and exhaust fans.

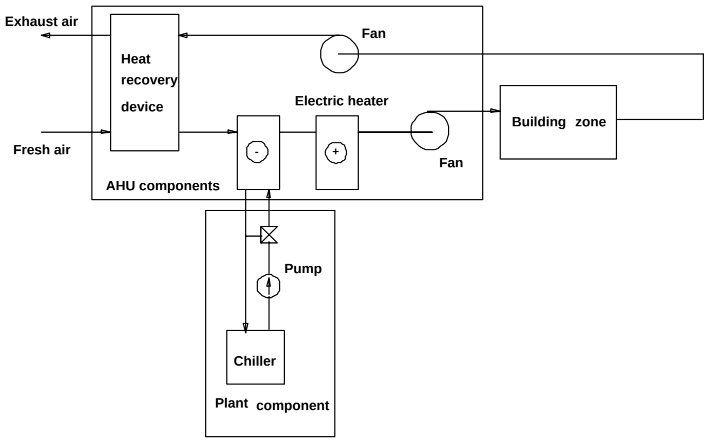

Figure 1, Plant Configuration

The optimum control of this building system involves minimizing the total energy cost from the chiller and AHU over a specified period (24 hours), while maintaining the zone thermal comfort requirements. During the summer, the components to be controlled are the fans, chiller, cooling coil and heat recovery device (HRD). The optimum solution to the problem is a schedule of control operation throughout the 24 hours optimization period. Such an optimization problem is complicated due to the fact that there are many independent control variables and discontinuities in the search space associated with the different possible operating modes of the system. However, the decoupling of the different levels of control can significantly reduce the complexity of the problem.

Figure 2 illustrates a possible control system structure. The controller uses a thermal model of the building and plant as the basis for predicting the schedule of the plant setpoints and the building response for the next 24 hours. This paper examines the characteristics of the setpoint schedule under the perfect prediction of the thermal disturbances upon the building; a prediction algorithm and the effect of prediction errors is not addressed here.

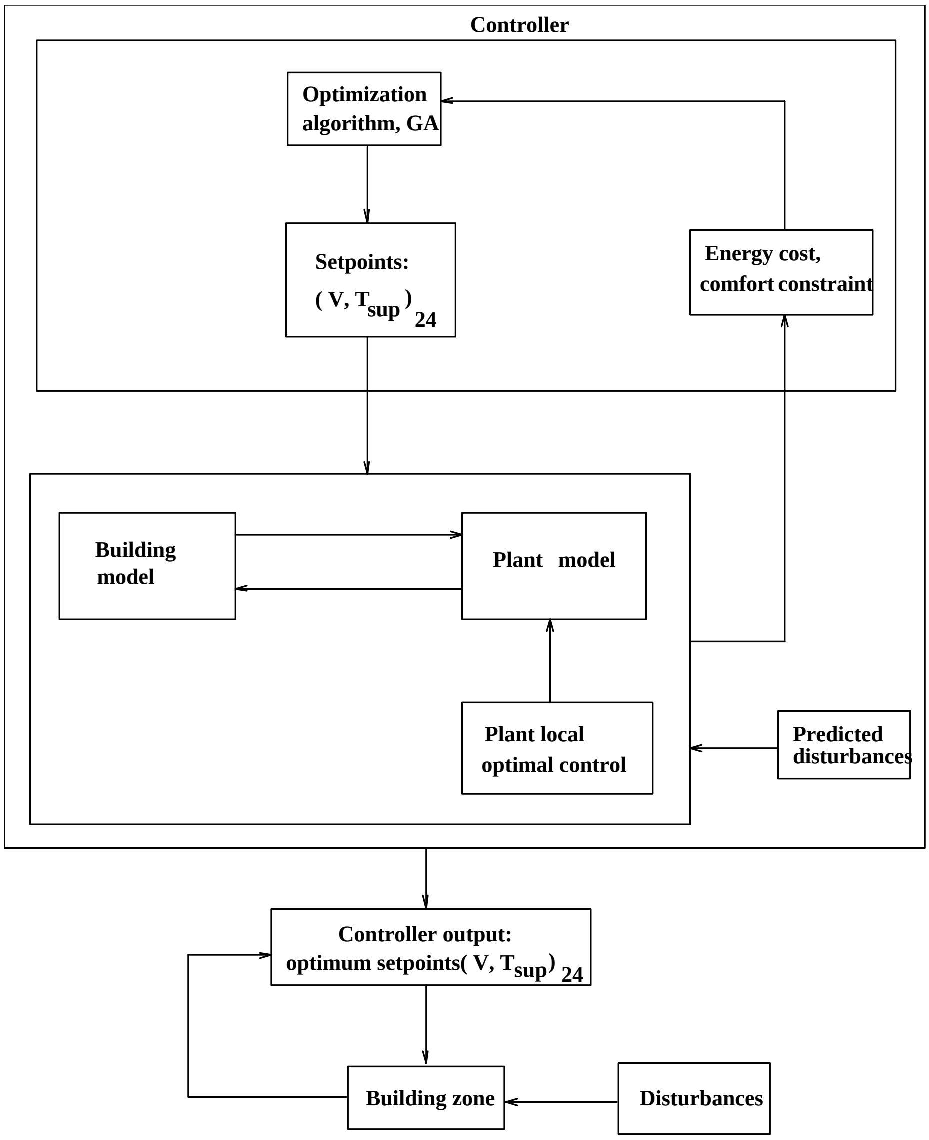

Figure 2, Control System Structure

The controller is considered at two levels. The higher level is concerned with optimizing the control variables that directly influence the thermal storage over a planning period of 24 hours. This allows the daily variation of the ambient and room thermal environment, as well as the cost of electricity, to be taken into account. The high level control variables are the setpoints for the supply air condition to the ventilated slab. They are optimized in order to minimize the energy costs without violating predefined comfort levels in the occupied space. The energy cost and comfort levels are simulated from the building and plant models. In order to ensure that the energy cost is minimized, it is necessary to optimize the plant operation in each hour at the lower level; the problem being constrained by plant design capacity and the setpoints passed from the high level.

The control hierarchy illustrated in Figure 2 reflects the nature of the supervisory control problem for buildings with thermal storage. The high level control is concerned with optimizing the operation of thermal storage over 24 hours, whereas the low level control ensures the optimum plant operation in each of 24 hours to meet the high level control requirements. The decoupling of the controls also has the advantage that it simplifies the optimization problem.

2.1 Thermal Storage Control Variables

There are two independent variables in the supervisory control of the thermal storage in ventilated slab systems, supply air flow rate and supply air temperature. The room thermal states are uniquely determined by these two variables together with ambient boundary conditions and room thermal disturbances. The supervisory setpoints for supply air flow rate (V) and temperature (T_{sup}) over the entire 24 hours planning period are optimized subject to the time-varying room and ambient conditions.

Previous research [20] has focused on the control of an air conditioning system installed in a conventional building. For such a building, the thermal storage in the building fabric is controlled by the zone setpoint since it is the zone thermal condition that governs the heat flow to and from the building envelope. The control requirement of the building space upon the plant is determined from the zone setpoint. This is not the case for a ventilated slab system where the air entering the slab has a significant influence on the thermal storage and thus the entire zone thermal condition. Further, most ventilated slab control systems are open-loop to the zone condition. This is typical for highly insulated heavy weight low energy buildings which are relatively insensitive to the diurnal changes in ambient thermal conditions.

2.2 Plant Control Variables

The optimization of plant operation in each hour is concerned with minimizing energy consumption while meeting the supply air setpoints. The setpoint of supply air flow rate can be met directly by controlling the fan speed. However, the control of the supply air temperature can be optimized by making a choice between free and mechanical cooling and whether the heat recovery device is operated or not.

Energy costs for mechanical cooling can be minimized by optimizing chiller operation. In terms of the chiller control variables, the water flow rate in the evaporator and condenser has less influence on the chiller performance than chilled water leaving temperature and condenser water leaving temperature [8,9]. For an air-cooled chiller, the chilled water leaving temperature is the most strongly correlated with the chiller performance [8,9]. However, the fabric storage and the use of high effective heat recovery device greatly reduces the required capacity of the chiller, an air-cooled package chiller usually being of sufficient capacity. It is common practice to fix the setpoint of chilled water leaving temperature in small package chillers. This together with the low mechanical cooling duty has led to the chiller control being excluded from the optimization in this study.

In this study, it is assumed that the plant is controlled to meet the supply air setpoints, even if heating is required during the summer. For a given supply air temperature setpoint, it may be necessary to heat or cool the ambient air to meet the setpoint. The operation of the heat recovery device may, or may not, benefit this process. The plant supervisory control variables to be optimized in each hour are therefore concerned with the mode of operation of the plant. If the supervisory control is limited to switching the plant either ON or OFF, and the simultaneous heating and mechanical cooling is not an option, the six plant modes exit for summer operation (Table 1).

Mode	Heat Recovery Device	Electric Heater	Chiller
1	ON	OFF	OFF
2	OFF	OFF	OFF
3	ON	ON	OFF
4	ON	OFF	ON
5	OFF	ON	OFF
6	OFF	OFF	ON

Table 1, Plant Operating Modes

3 Control Optimization

An optimization problem is formulated from three key parts, problem variables, objective function and constraint functions. The choice of algorithm to solve the optimization problem is dependent upon the characteristics of the problem.

3.1 Optimization of the Fabric Thermal Storage

The problem variables for the fabric thermal storage optimization are the setpoints for the flow rate and temperature of the air supplied to the ventilated slab. Two setpoints are to be optimized over the 24 hours planning horizon, giving 48 problem variables.

The objective function is the total energy cost of the system over 24 hours. It is a function of the power consumption of the supply and exhaust fans, the chiller, the electric heater and electricity tariff structure. The objective function can be discontinuous where a change in the supply air temperature setpoint causes a change in the operating mode of the plant, bringing a sudden change in energy cost.

The constraint on minimizing the energy cost is that thermal comfort in the occupied space must be maintained. The Predicted Percentage of Dissatisfied (PPD) index [23,24], has been used to define the comfort constraint with a limit of 10% PPD set for the occupied period. This is a generally recommended limit and corresponds to a Predicted Mean Vote in the range -0.5 to +0.5 [22,23]. No comfort constraint needs to be set for the unoccupied period, however, in order to reduce the risk of over cooling the building, a limit of 20% PPD has been set during the unoccupied period.

A Genetic Algorithm (GA) search method has been selected to solve the optimization problem. GA's are particularly suited to solving problems with a high number of problem variables and a discontinuous objective function [27,28]. GA's have also been used successfully to solve highly constrained optimization problems [27]. GA's operate with a population of solutions. Each ‘individual’ in the population represents one solution and is defined by a coding of the problem variables with a ‘fitness’ derived from the objective function and any constraint violations. A set of probabilistic transition rules based on `survival of the fitness' strategies are used to generate successively fitter populations of individuals (having lower objective function values and fewer, if any, constraint violations). The simple GA implemented in this research has proved to be a robust technique for solving the thermal storage optimization problem.

3.2 Optimization of the Plant Operating Mode

A single discrete problem variable can be used to represent the plant operating modes. The mode is optimized to minimize the plant energy costs while being constrained to meet the setpoints for the temperature of the air being supplied to the ventilated slab. In practice, when the plant load is very low the plant may not be operated and therefore the setpoint may not be met. Similarly, the setpoint will not be met for loads over the design capacity of the plant.

The feasible number of operating modes can be reduced by an analysis of the zone exhaust air temperature, ambient air temperature and the supply air setpoint. Since the number of feasible modes is small, the optimum mode is selected by an exhaustive search.

4 Building and Plant Models

The building and plant models are an integral part of the controller. The building model is a dynamic model. The plant performance is simulated by simple steady-state models. The building and plant models are decoupled, although simple iterations between them are necessary when the plant is unable to meet the setpoints of the air supplied to the ventilated slab.

4.1 Building Model

The zone model is derived from the first-order thermal network model [24,25,26]. The building model has been extended to include ventilated floor and ceiling slabs [29]. The new model can address the heat exchange between the slab cores and the ventilation air, the thermal storage in the building fabric, and the effect of the heat disturbances on the room.

The combined thermal network model for the building zone and slab is illustrated in Figure 3. There are five thermal capacitances: C_b (walls), C_{uc} (upper part of ceiling), C_{lc} (lower part of ceiling), C_{uf} (upper part of floor) and C_{lf} (lower part of floor). Since the slab construction may not be symmetrical about the ventilation air path (due to the use of a floor screed and finishes), and since there may be a difference in temperature from one zone to another, the mass temperature may not be distributed symmetrically about the center of the slab. This effect is accounted for by modelling the slab in two halves, one half coupled to the environment above the ceiling (or floor) slab and the second to the environment below (the adjacent zone temperatures being represented by T_{ac} and T_{af}).

The thermal resistances of the building elements (R_{ob}, R_{ib}, R_{auc}, R_{cuc}, R_{alc}, R_{clc}, R_{auf}, R_{cuf}, R_{alf} and R_{clf}), are obtained by an established algorithm [24,26]. The internal surface temperatures for the wall, ceiling and floor are represented by T_{sw}, T_{sc} and T_{sf} respectively. $\overline{T}_{avc}$ and $\overline{T}_{avf}$ are the mean ventilation air temperatures for the ceiling and floor slabs. T_z is the air temperature of the modelled zone which is coupled to the wall, ceiling and floor surfaces by the convective surface resistances R_{sb}, R_{sc} and R_{sf}. The long wave radiant heat exchange between surfaces is modelled by the exchange with the radiant star index temperature T_{rs}. This temperature is coupled to the room surfaces by the radiant resistance R_{rw} for the wall, R_{rc} for the ceiling and R_{rf} for the floor. The short wave heat gain Q_r is distributed to each surface by weighting factors. The convective heat gain Q_c, includes the heat gain from the ventilation system supplied via the ceiling slab. T_{sa} is the sol-air temperature and T_o the outside ambient air temperature.

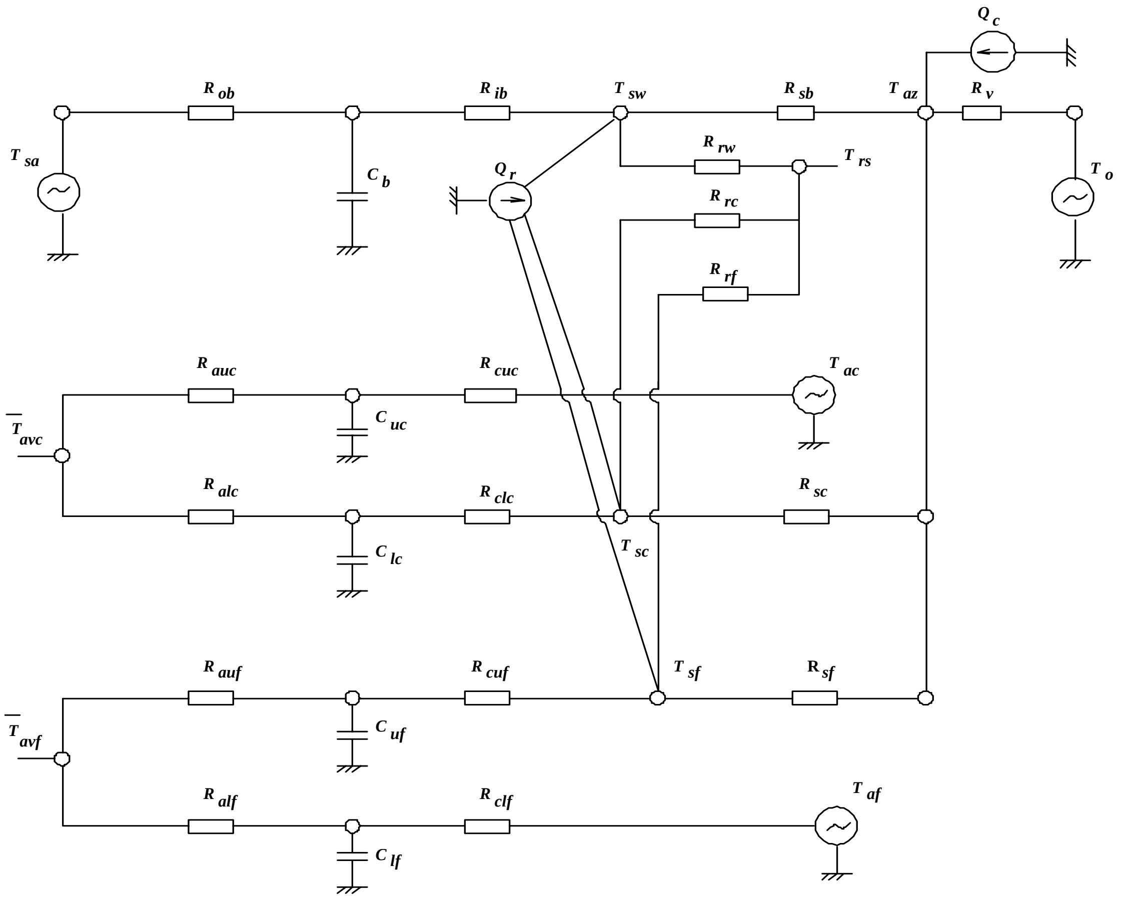

Figure 3, Building Simulation Model

The model has been validated against measurements taken from an experimental chamber at the Building Research Establishment in the UK. The model prediction agreed closely with the test measurements [29].

4.2 Plant Model

The AHU in the system shown in Figure 1 includes five elements: two fans, a heat recovery device, a cooling coil and an electric duct heater. A variable-speed fan has been selected in this study. For variable speed control, the fan power consumption varies with the cube of the flow rate:

$$P_{fan} = P_{fan,des} (V_{air} / V_{air,des})^3$$

where P_{fan} and V_{air} are the actual fan power and air volume flow rate. $P_{fan,des}$ and $V_{air,des}$ are design fan power and design air volume flow rate. At a very low fan speed and flow rate, the fan power no longer obeys this relationship as a result of a degradation in fan motor efficiency. However, the fan is assumed to be OFF for a fan part load (volume flow) ratio below 0.2.

A regenerative heat recovery device (HRD) is commonly used in ventilated slab systems. A typical ventilated slab system HRD is constructed with two cell packs. When in operation, one cell pack is recharged for 30 seconds with the warm exhaust air in the winter or the cool exhaust air in the summer, during which time the fresh air is allowed to discharge the other cell pack to be heated or cooled. The cycle is repeated unless the operation strategy decides that the HRD should be OFF.

The dynamic characteristics of the HRD can be neglected when compared with the dynamic response of the building. The HRD is simulated in steady-state by means of its overall effectiveness. The average effectiveness of the HRD is 87.5% according to the manufactures data. The air temperature leaving the HRD (T_{sup}), is given by:

$$T_{sup} = T_{fresh} - 87.5\,\% \,(T_{fresh} - T_{room,ex})$$

The performance of the chiller is closely correlated to the chilled water leaving temperature, ambient air temperature and the chiller part load ratio (PLR), [8,9]. The chiller performance can therefore be modelled by steady-state curve fit of manufactures data, COP = f((T_{amb} - T_{chw}), PLR). The chiller power consumption is then calculated from the COP and the load requirement.

The output of the cooling coil has been simulated using a standard, number of transfer units and effectiveness model. Perfect control of the coil output air temperature has been assumed, up to a maximum output as dictated by the limit of the coil's effectiveness. The electric duct heater is also assumed to be operated under perfect control with the limit of its capacity and 90% efficiency.

5 Characteristics of the Optimal Control Problem

The control optimization has been applied to a ‘typical’ office zone, and therefore to some degree, the characteristics of the results can be considered as ‘generic’. The investigation in this paper is based on the assumption that the next day’s weather conditions can be predicted perfectly. A set of real weather data monitored in Garston, UK, in 1994, has been used here as an example year for which to study the control characteristics. A hollow core ventilated slab building is usually well-insulated. Table 2 gives the typical building construction adopted in this study. The construction is for a ventilated slab building located at the University of East Anglia, UK [5].

Type	Construction
External Wall	200mm Rockwool, 25mm air gap 200mm heavy weight concrete block, 10mm plasterboard

Internal Wall	10mm plaster, 100mm heavy weight concrete block, 25mm air gap, 100 heavy weight concrete block, 10mm plaster
Floor/Ceiling	250mm hollow core concrete slab, 100mm screed
Window	triple-glazed, low-emissivity coated

Table 2, Building Construction

The U value of external walls is only 0.2 W/m^2 K and that of the window is 1.3 W/m^2 K. The floor area of the zone studied is 4.0 m × 6.0 m with a ceiling height of 2.84 m. The ventilation air is introduced to the space by ceiling diffusers, which are connected to the ventilated slab air outlet. The ceiling has five ventilated slabs, each of which is 4.0 m long, × 1.2 m wide, × 0.25 m thick.

Five identical zones in the building have been modelled. The zones have one south facing external wall with a 4.2 m^2 window. Ventilation from the plant is equally divided and supplied to each zone. The design minimum ventilation during the occupancy period is 2 air changes per hour for each zone. Occupancy is scheduled from 8:00am to 4:00pm. Three electricity tariff structures have been studied, 3:1, 2:1 and 1:1 (the ratio of the cost of on-peak electricity to the cost of off-peak electricity).

5.1 The Characteristics of the Optimal Schedule of the Control Variables

Figure 4 displays an optimal solution for a day in the summer, 13th July 1994. Three obvious stages can be seen in the ventilation rate schedule. They result from the thermal comfort requirement during occupancy and the applied electricity tariff structure (3:1). During the night, a low ambient temperature and the cheaper cost of electricity encourages higher ventilation rates, shifting the daytime cooling load to the night. When the occupancy starts in the morning, only the minimum ventilation is required. As soon as the cooling load increases near midday, higher ventilation is required to extract more ‘coolth’ from the slab thermal store. The ventilation is switched OFF after the occupancy period.

Figure 4 also illustrates the setpoint temperature for the air supplied to the ventilated slab. Where the setpoint matches the ambient air temperature, then the plant is operating in a free cooling mode. Where the supply air setpoint is lower than the ambient temperature, then either the heat recovery device, and or, the chiller is in operation. In this instance, the chiller is operated for 5 hours at night and one hour during occupancy period. The heat recovery device is however in operation throughout the occupied period. Note that when the ventilation is OFF, the supply air temperature setpoint is set to zero.

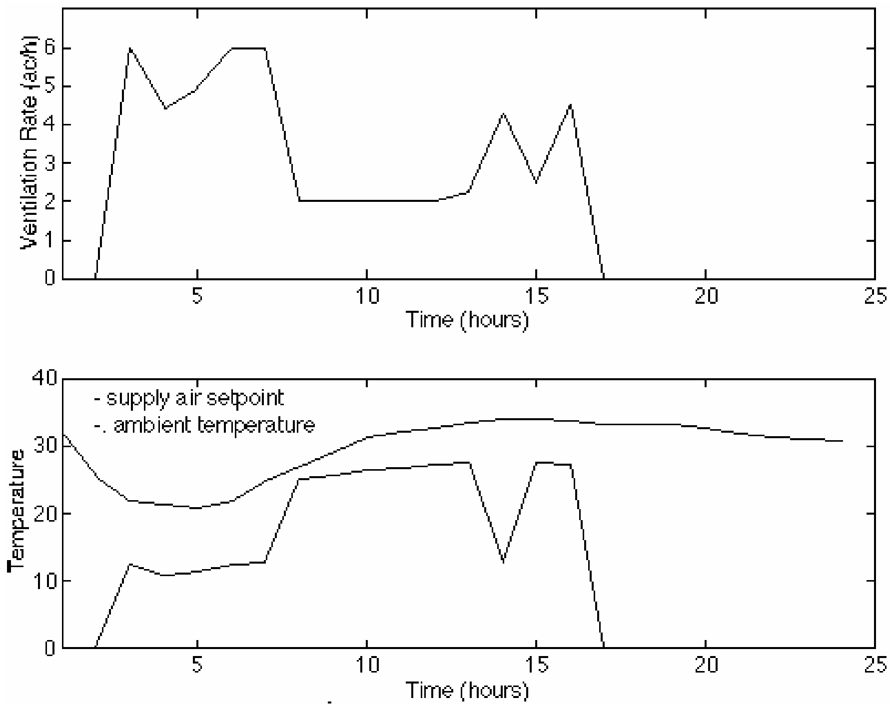

Figure 4, Optimum Scheduling of Control Variable under a 3:1 Electricity Tariff Structure

In general, sufficient pre-cooling of the ventilated slab can be achieved by beginning cooling after midnight, since this is when the lowest ambient temperature occur. Therefore, there is a period between the end of occupancy and the start of pre-cooling the next day, when the plant is not operated. This is an important characteristics in that the plant operation is decoupled from one day to the next day, with the control optimization only needing to be considered over a 24 hours period.

Figure 5 illustrates the optimum setpoints for the week of the 9th to the 15th July 1994. During the first three days, the cooling loads are relatively low leading to no pre-cooling of the ventilated slab, the residual thermal storage being sufficient to offset the cooling loads and minimum ventilation only being necessary during the occupied period. However, the thermal loads are higher during the rest of the week with pre-cooling of the ventilated slab being evident. The ‘spikes’ in the temperature setpoints during occupancy are a result of the chiller being operated to offset the peak loads. In practice, such short periods of chiller operation and frequent varying fan flow rate would be avoided.

Figure 6 shows the optimal setpoints for a 1:1 tariff structure on the 13th July. In comparing Figure 4 with Figure 6, it is seen that the weight of shifting the cooling load to the night is reduced. The maximum ventilation rate at night is similar to the peak during occupancy. The chiller is operated for 4 hours during occupancy period, but not at all at night. This is in contrast to the 5 hours night chiller operation for the 3:1 tariff. There is no incentive to shift more cooling to the night. However, free cooling using the lower night air temperature is still efficient for saving energy due to active utilization of building fabric storage. The total power consumption is slightly lower than that from 3:1 structure. The result from a 2:1 tariff structure shows no significant difference from that of the structure 3:1.

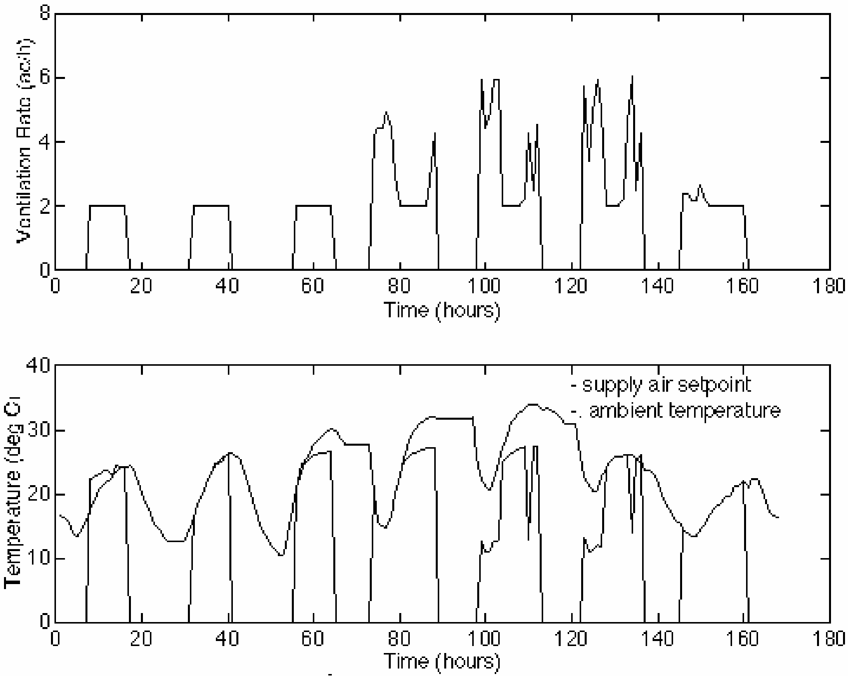

Figure 5, Optimum Scheduling of Control Variables for Seven Day

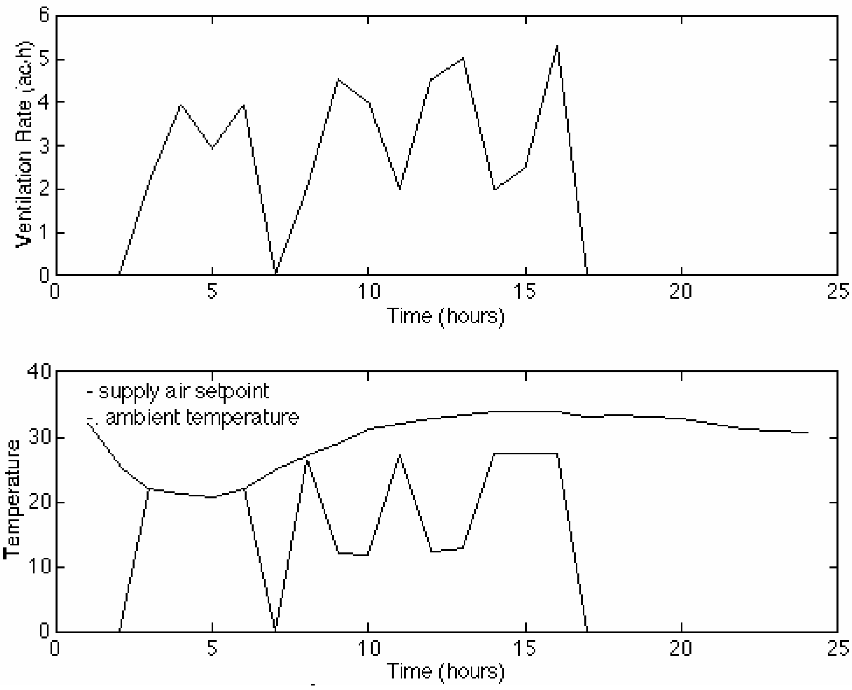

Figure 6, Optimum Scheduling of Control Variables under a 1:1 Electricity Tariff Structure

5.2 Simplified Time-Stage Control

From the results of the optimum scheduling of the supply air setpoints, the optimal control strategy for daily operation can be adequately represented by a ‘time-stage’ control. Figure 7 illustrates the control stages S_1 signifies the start of plant operation with a supply air flow rate of V_1, C_1 and C_2 is a period of chiller operation during which the supply air temperature has a setpoint of T_1. No temperature setpoints are required during the free cooling periods (S_1-C_1, C_2-8:00, 8:00-S_2, S_2-C_3, C_4-16:00), since the supply air temperature is dictated by the ambient temperature and whether the heat recovery device has been selected to be in operation or not. From the start of occupancy (8:00) to S_2 is a period of minimum ventilation rate. S_2 signifies the second stage of high cooling with the ventilation rate set to V_2 and a supply air temperature setpoint of T_2 during chiller operating period C_3-C_4. The plant operation stops at the end of occupancy (16:00).

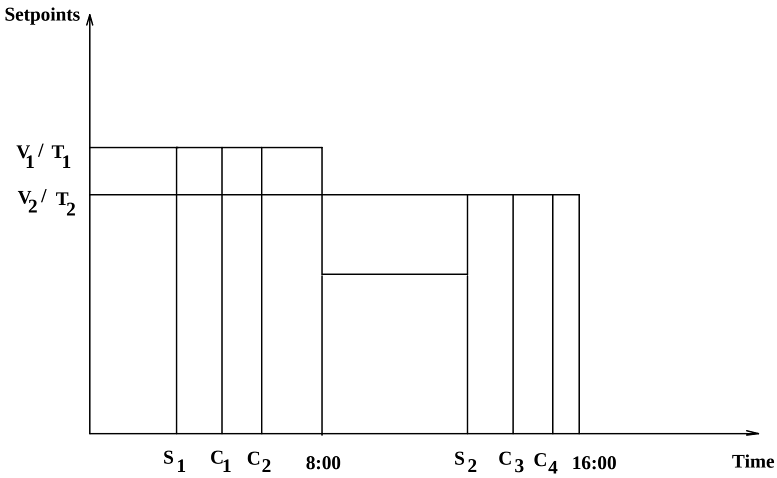

Figure 7, Simplified Time-Stage Control

The optimization problem has therefore been reduced from 48 variables to 15 variables, 6 time stage variables (S_1, C_1, C_2, S_2, C_3, C_4), 4 setpoints (V_1, T_1, V_2 and T_2) and the ON/OFF operation of the heat recovery device in each of the 5 free cooling periods. This framework makes for robust supervisory control which tends to eliminate the short periods of chiller operation and frequent varying fan flow rate operation exhibited in the optimum scheduling of the setpoints.

Figure 8 illustrates the optimal time-stage control strategy for 13th July 1994. The chiller is in operation for 4 hours during the night and 2 hours during the occupancy period. The heat recovery device is in use for the free cooling periods. A comparison of Figure 8 with Figure 4 will indicate that the time-stage control encompasses the characteristics of the optimum scheduling of the supply air flow rate and temperature, the main difference being that one hour night chiller operation is moved to the occupancy period.

The time-stage control optimization problem was solved by Genetic Algorithm used for the optimum setpoint scheduling. The algorithm proved to be efficient and robust in searching through the discontinuous solution space resulting from the mixed nature of the variables.

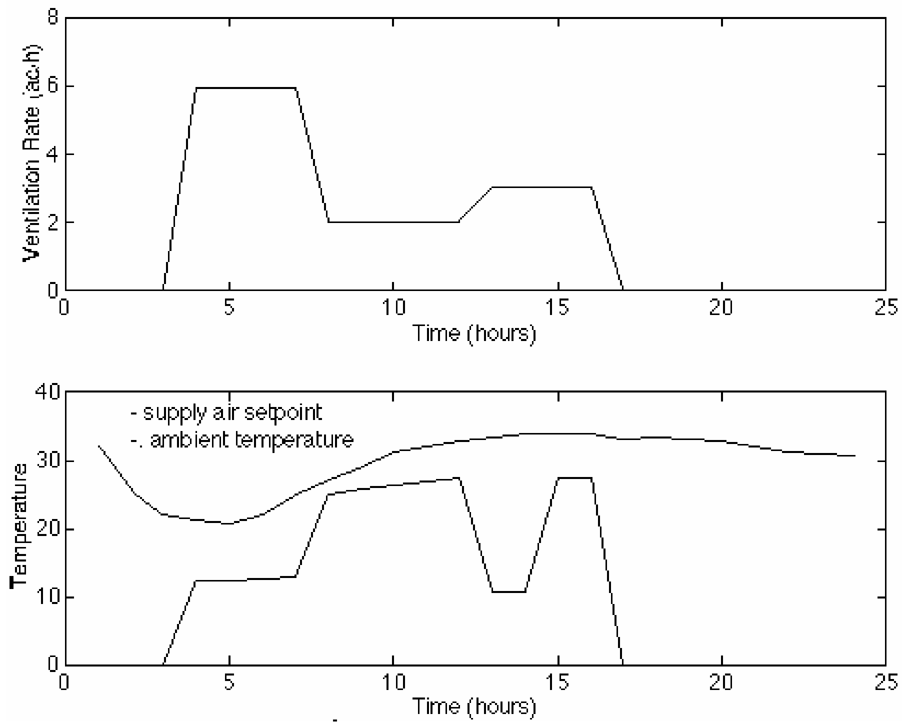

Figure 8, Optimal Scheduling of Control Variables from Time-Stage Control

6 Controller Performance

The performance of the optimum setpoint scheduling control, the time-stage control and the conventional control strategy, has been compared for two weeks in the summer of 1994. During the first week, 24th to 30th June, the cooling load was such that the comfort constraints could be met without the use of mechanical cooling. Mechanical cooling was however necessary during the second week, 9th to 15th July. A nominal cost of £ 0.10/kWh is used for the cost of off-peak electricity.

Table 3 indicates the relative performance of the controllers for when no mechanical cooling is required (24th to 30th June). The thermal comfort in the zone was maintained under all three control strategies, with on average a slightly cool room (PMV below 0) resulting from the conventional controller and a slightly warm room resulting the optimal controllers (PMV over 0). The energy cost from the optimum setpoint scheduling control was £ 0.54, with a power consumption of 1.95 kWh, whereas the conventional control energy cost was £ 6.08 and consumed 28.07 kWh of energy. From the result, it is clear that the optimum setpoint scheduling control strategy leads to significant savings in both the energy use and consumption. This is due to shorter period of pre-cooling operation, and indeed for some days, the plant was controlled with minimum ventilation throughout the occupancy period and without pre-cooling at night. However for the conventional control, a constant ventilation was applied for both the night and the day. Since under the conventional control strategy, the ventilation operation is based on if-then rules, it is impossible to take account of the hourly varying ambient and room thermal conditions. Although the occupant thermal comfort was satisfied during the week, the plant was operated for unnecessary longer hours of pre-cooling, which resulted in a slightly cool zone comfort condition. It was also noted that the zone temperature was more sensitive to changes in ambient conditions than in comparison to the setpoint scheduling control. The optimum time-stage control has a similar performance to the optimum setpoint scheduling control, the difference being due to a slightly different strategy for the operation of the heat recovery device.

Type of Controller	Energy Cost (£)	Energy Use (kWh)	Mean PMV	Maximum PPD (%)
Conventional Control	6.08	28.07	-0.18	9.4
Setpoint Scheduling	0.54	1.95	0.13	10.1
Time-stage Control	0.52	1.74	0.09	9.9

Table 3. Performance Comparison for 24th to 30th June

Table 4 compares the control strategies where mechanical cooling is required if the thermal comfort conditions are to be maintained throughout the occupied period. The energy cost from the conve ntional control strategy was £ 5.98 with an energy consumption of 27.03 kWh. The energy costs under the setpoint scheduling control was £ 6.58 and the energy consumption 42.86 kWh. Although the mean PMV under the conventional control was relatively low, the comfort conditions in the occupied zone were frequently unacceptable with a maximum PPD of 19.8%. This was due to the conventional control not including any chiller operation and the ambient air temperature remaining high throughout the night, which prevented effective pre-cooling of the ventilated slab and resulted in the conventional controller only assigning short periods of night ventilation (typically two hours between 4:00am and 5:00am). This led to a lower energy consumption than for the operation under moderate thermal gains as for the first test period (Table 3). The higher energy consumption and cost from the optimal control was due to the chiller operation which was necessary to maintain thermal comfort. During the period of highest thermal load,

the chiller was in operation for a few hours at night and one hour during the occupancy period. It is also evident that although the plant power consumption was much higher than that for the conventional control, the energy cost was only marginally higher. The optimum time-stage control resulted in a higher energy consumption and cost than for the setpoint scheduling, but had the same basic strategy. This slight degradation in the controller performance was due to the simplification from the setpoints characteristics to the simple stages of plant control. The simplifications make it more difficult to account for hourly changes in thermal load but lead to a more stable plant operation.

Type of Controller	Energy Cost (£)	Energy Use (kWh)	Mean PMV	Maximum PPD (%)
Conventional Control	5.98	27.03	0.21	19.8
Setpoint Scheduling	6.58	42.86	0.26	10.4
Time-stage Control	7.20	45.72	0.24	10.0

Table 4. Performance Comparison for 9th to 15th July

7 Conclusion

This paper investigates the characteristics of an optimum control strategy for fabric thermal storage systems and the strategies potential for reducing energy cost. The performance of the optimum controller has been investigated for two thermal loads, the first for a load where the thermal comfort could be maintained without the use of mechanical cooling, and the second load where mechanical cooling was necessary.

Two optimal control strategies have been investigated, the first optimized the setpoints for the flow rate and temperature of the air supplied to the ventilated slab. The second strategy, a 'time-stage' control of plant operation, was derived from the characteristics of the optimum setpoint schedule. During the summer, the general characteristics are that the low ambient air temperature and off-peak electric tariff leads to high ventilation rates at night, followed by an initial period during occupancy when the ventilation rate is set to a minimum. The ventilation rate may be increased later in the day to compensate for higher thermal loads. Similarly, where mechanical cooling is required, the chiller will be operated during the off-peak period with supplementary operation during peak periods only when necessary. In comparison to a conventional rule based control strategy, the optimal control strategy provides significant energy cost savings and guaranteed thermal comfort.

The optimum control problem was solved by Genetic Algorithm. The result shows that it is a viable method for investigating the characteristics of the optimum supervisory control of thermal plant.

Future work will address the implementation of the optimum control strategy. A weather predictor is required and a predictive control algorithm necessary for the implementation of the optimum controller in real buildings.

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