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Modelling and Performance Analysis of a Sub-Dew Point Chilled Beam in Mixed Mode Buildings

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A Doctoral Thesis submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University

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Abstract

Prompted by the energy crisis in the 1970's, European level fiscal and financial measures encourage energy efficient building design. "Mixed mode" strategies can be employed in buildings with moderate thermal loads, this approach can reduce building energy usage, and by inference, reduce CO$_2$ emissions. A mixed mode approach might employ chilled surfaces with displacement ventilation. This thesis investigates the performance and integration of one form of chilled surface design, (a sub-dew point chilled beam), within mixed mode strategies.

Sub-dew point chilled beams have a surface temperature that is at or below the zone saturation temperature, this increases the cooling capacity of the chilled beam and consequently produces a latent heat transfer addition due to condensation mass transfer. This thesis describes the sensible and latent modelling approach which models the zone, sub-dew point chilled beam and mixed mode strategy thermal plant response to external disturbances. The thesis describes the use of an Enclosure Comfort Performance Indicator that acts as the objective function for the optimization of the mixed mode strategies with and without the integration of the sub-dew point chilled beam. The implementation of the Complex method for finding the operational optimums of the mixed mode strategy is described, and its effectiveness at finding the optimum solution evaluated.

Normalised energy, cost and comfort performance indicators are used to assess the overall performance and integration of the sub-dew point chilled beam for different mixed mode strategies, for the ambient test conditions and for different thermal weights of building construction.

Keywords sub-dew point chilled beams, lumped parameter thermal network model, moisture modelling, Complex method optimization, enclosure comfort performance, building thermal weight, normalised performance indication.
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I would like to thank my supervisor Dr. J. A. Wright for his guidance, and continued hard work during the work leading to this thesis. I gratefully acknowledge the financial support from Loughborough University and the EPSRC.
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Chapter 1

Introduction

The energy crisis of the 1970's prompted interest in the design and construction of energy efficient buildings. Restrictions on the emission of environmental pollutants is now promoting low energy building design strategies that utilise renewable energy sources. Buildings account for 46 % of the total energy consumption in Europe [1], this has led to increased interest in energy efficiency, and low energy building design, at a European level. The Kyoto Protocol [2] has set targets for CO₂ emission reductions, which have prompted fiscal and financial measures [3] to improve building regulations to include active and passive low energy design.

The principle of low energy and energy efficient building design is to maximize long term energy savings by utilizing natural renewable energy sources and the thermal properties of the building structure in order to minimize building operating costs. Passive technologies for internal climate control that require little or no primary energy input, such as natural ventilation and solar energy, are fundamental to low energy design. Control of the internal climate is not possible for all buildings. Where the peak thermal loads on the building are too high to be met by a purely passive response of the building, "mixed mode" strategies can be employed in which the low thermal loads can be met by the passive response of the building, and the peak loads by mechanical air conditioning strategies. For moderate thermal loads a mixed mode approach might employ chilled surfaces with displacement ventilation to produce a comfortable environment. This thesis investigates the performance and integration of one form of chilled surface design within a mixed mode strategy.
1.1 Low Energy and Energy Efficient Buildings

Careful application of energy efficient design principles can reduce energy use in buildings considerably. Approaches that do not significantly alter the initial cost of a structure (orientation of the building to the prevailing wind to encourage natural ventilation and providing solar shading to reduce summer overheating) can provide substantial long-term energy savings and a more comfortable living environment. Further, long-term energy savings can be realised by incorporating heavy weight thermal elements into the building structure which can reduce the amplitude of the internal temperature swing. Such measures will result in reduced operating costs and mechanical plant size, but this is at the expense of an increased initial running cost of the building. Passive solar design, which uses the building’s structure to capture solar energy and store heat, can alone save at least 50% of the energy used in buildings [1].

According to Abel [4], there are two fundamentally different approaches to the reduction of building energy use:

**Low-energy buildings** The ultimate goal of research and development is a building that requires no energy or external energy supply of energy or, at least, no supply of “purchased” energy.

**Energy-efficient buildings** The ultimate goal of research and development is a building that, in every detail, has the lowest possible energy requirements that can be achieved by sacrificing only a reasonable amount of resources.

It is clear that buildings which have high thermal loads and require mechanical air conditioning, should be referred to as “energy-efficient” even when the building was designed to be a “low energy” building.

1.1.1 Contemporary Low Energy Buildings

Current prediction of the trends in low energy building design and use of renewable energy sources in Europe show a 9% increase per annum and, with an annual growth of 4.3 BECU [5], low energy building design is reducing primary energy
use in Europe. Previous low energy design case studies [6, 7, 8] have all shown long term energy savings. Using openable windows and narrow plan building form, the BRE Low-energy office in Garston UK has shown a 30 % reduction in energy use in comparison to current best practice [8]. Passive ventilation strategies have also been used on the John Cabot City Technology College in Bristol, UK [7]. A two storey building with the ground floor rooms being cross ventilated using exhaust chimneys and the first floor using the pitch of the roof to give the necessary height differential for stack ventilation.

The Learning Resources Centre at Anglia Polytechnic University, Chelmsford UK, has a 6000 m² atrium that uses stack driven natural ventilation in a library/computer building [7]. Additional passive low energy building design strategies include the exposure of heavyweight thermal mass and extended daylighting using lightshelves. Classrooms at the Cable and Wireless Building, Coventry UK, use building form to provide natural ventilation when no cross-ventilating wind pressure is available [7]. The building has a “wave-form” roof to provide the necessary height differential between inlet and outlet for stack ventilation.

Having been refitted to incorporate a combined low and solar energy design principles, the “Skotteparken” housing project, Ballerup Denmark, has shown 50 - 60 % reduction in energy consumption [9] in comparison to energy use before the re-fit. The main low energy design features include: extra insulation in the building envelope; low energy glass in the windows; heat recovery from the ventilation air; solar panels to provide electrical energy and an energy management system which monitors and regulates energy usage for the district heating system.

The C.K Choi Building at the University of British Columbia, Canada, minimizes embodied energy, (the energy used in producing construction materials), by careful selection of building material[10]. The exterior cladding is 100 % recycled brick and the structural system is comprised of 50 % re-used wood. Further, the building uses on-site greywater recycling reducing water pumping and treatment costs.

Landscaping the exterior of a building is an important consideration for low energy design. Shading and evapotranspiration (the process by which a plant actively moves and releases water vapour) from trees can reduce surrounding air temperatures by as much as 5 °C [11]. Studies by the Lawrence Berkeley Laboratory[11], found summer daytime air temperatures to be 2 to 3 °C cooler in tree - shaded neighbourhoods than in treeless areas. A study in South Dakota [11], found that
windbreaks to the north, west, and east of houses cut fuel consumption by an average of 40%.

Where passive approaches fail, simple active methods can be implemented to meet higher thermal loads. A fan, when combined with a passive system, is a common method to pull hot air to other locations in an enclosure. The IFZ Building, Giessen Germany[12], uses a fan situated at the top of an atrium to assist natural stack ventilation when ventilation rates fall. Warm air can also be brought to a low point in the building to create natural assisted convective flow, the GVB Building, Bern-Ittigen Germany[12], has low level diffusers in its atrium which help to circulate air throughout the building.

1.2 Energy Efficient Building Design and Operation

The focus of this research is on buildings having thermal loads that dictate some use of mechanical comfort conditioning, and as such are "energy efficient" rather than "low energy" buildings. The goal of energy efficient design is to match the internal thermal loads on the building to the "mode" of operation of the proposed air conditioning/ventilation system. Passive operation (natural ventilation, solar design) has greatest effect within low load conditions, whereas mechanical operation, (displacement ventilation, chilled surfaces, full air conditioning), having greatest effect at high loads. Mixed mode buildings, a mode of operation utilizing both passive and mechanical systems, "bridges the gap" between low load passive operation and high load mechanically conditioned buildings. Mixed mode design can be used in a "low energy" building but the concept is one of "energy efficiency".

1.2.1 Passive Modes of Operation

The Building Research Establishment, (BRE)[6], provides design guidelines for low energy, "passive mode" building design. These state that the upper limit for effective natural ventilation is at an internal heat load of 40W/m². A building
that could operate in passive mode would ideally have:

- a shallow plan building layout;
- shading (preferably external) to minimize summer overheating;
- an airtight building envelope to minimize unwanted air infiltration;
- trickle ventilators proving controllable background ventilation;
- openable windows for day and night cooling;
- occupant control of cooling and ventilation;
- use of low energy lighting and IT equipment to reduce internal heat gains.

In addition, the BRE states that for higher internal loads, consideration of the building form, orientation and finish and the exposure of internal thermal mass to dampen the internal diurnal temperature swing, has a large impact upon the efficacy of passive design approaches. The Union of Concerned Scientists, (UCS), [13] also give guidelines for passive low energy design that show that building orientation and window shading and location can reduce summer overheating and decrease winter heating cost.

Further, passive cooling can be enhanced by ensuring windows are placed to capture prevailing winds, narrow plan building design improving the efficacy of cross flow ventilation, while the inclusion of high openings or thermal chimneys will encourage buoyant "stack" ventilation. Consideration is also given to external landscaping which can be employed to "funnel" prevailing winds and to the placement of deciduous trees in southerly locations to provide shade in summer and allow penetration of light and heat in winter.

Natural ventilation systems, passive modes of operation, are controlled using both occupant control and automatic control systems. During occupied periods, occupants, by opening and closing windows as desired, control both temperature and ventilation, automatic control only takes over to increase the ventilation rate in the event of excessive build-up of CO₂. Outside of the occupied periods, automatic control systems control the temperature and ventilation rates:
1.2. Energy Efficient Building Design and Operation

User control: desire for fresh air or change in temperature, prompts users to open and close windows as desired.

Automatic control: CO₂ and temperature sensors, act to prevent inadequate ventilation or overheating by opening trickle vents or windows.

In addition, automatic control can be used over night to pre-cool the building; flushing the enclosure with cool night air during the unoccupied periods removes heat from the building fabric which promotes radiant cooling from the room surfaces and extends the period of natural ventilation during the next day.

1.2.2 Mechanical Modes of Operation

For high internal heat loads, mechanical modes of ventilation and cooling are necessary to meet comfort requirements. Mechanical air conditioning is either by all-air or an air-and-water system. The all-air system supplies both cooling and ventilation, whereas the air and water system has separate systems such as a chilled ceiling and displacement ventilation.

Displacement ventilation systems, supply air at low velocities at floor level in the enclosure. The mass flow rates are typically that for the fresh air requirement of the enclosure, (for an office space 8 l/s per person), and the supply air temperature is maintained at approximately 18 °C. Buoyant plumes are created around sources of heat, focusing the cooling effect, and rises to a high level where it is extracted. The primary energy demand for displacement ventilation is low compared to traditional air conditioning plant as there is no humidity control.

Chilled surfaces have emerged as an air-conditioning alternative, with this cooling technique now found in many office buildings [15], particularly in Europe [14]. All chilled surfaces operate in the same way, a cooling medium, usually water, is passed through a heat exchanger to cool the surface which then cools the enclosure by a combination of convective and radiant effects.

Chilled surfaces can be categorized into two main types, chilled panel or ceilings and chilled beams. Chilled beams can be classified as either static, in which the heat exchange between the beam and room air is provided by means of natu-
1.2. Energy Efficient Building Design and Operation

1.2.1 Natural Convection

Natural convection, or active, where fans regulate the flow rate of air forced across the beam. Chilled surfaces consume less energy than conventional air systems because they maintain acceptable thermal comfort levels at higher drybulb temperatures - a consequence of their radiant cooling element[16]. In addition, the use of water rather than mechanically cooled air gives a more energy efficient method of cooling[16].

Typically, full air-conditioning systems supply conditioned air at high level in the space. The supply air mixes with the enclosure air and is extracted again at high level. The air mass flow rate for “mixing” air-conditioning systems are set by the ventilation requirement for the space and the enclosure thermal load. Primary energy demand upon the mechanical plant associated with mixing systems is high, (in comparison to displacement ventilation systems) as the energy requirement has an additional latent load associated with the need to control the humidity by means of humidification and de-humidification.

1.2.3 Mixed Mode Strategies

Mixed mode strategies implement a holistic, whole building, approach to combine the best features of both natural (passive) and mechanical systems[17]. Whenever possible such an integrated system is intended to operate in the natural passive mode to reduce energy use, with the mechanical system use being restricted to the peak thermal loads. At the point when natural systems fail, the internal temperature rising above a predetermined limit, mechanical systems are used to provide sufficient cooling. The mixed mode philosophy allows internal temperatures to rise above conventional limits during the occupied periods, as this encourages window use and reduces the operation of the mechanical plant. Mixed mode buildings require careful control strategies that minimize energy usage while maintaining acceptable thermal comfort levels.

Little research has been performed upon the integration of natural ventilation and chilled surfaces, whereas much research has been performed upon mixed mode systems which include displacement ventilation, with many buildings already using the technology[17].

A three storey building with a central atrium, the PowerGen headquarters, Covent-
try UK, implements a mixed mode strategy which promotes cross, stack and mechanical displacement ventilation[18] with building form and orientation. The office spaces are shallow plan to promote cross-flow ventilation and surround the central atrium, which provides the height differential for stack ventilation to pull air through the office space in low wind conditions.

The axis of the building is east to west and houses the mechanically ventilated dependant offices and computer suites, at the eastern and western ends where overheating due to solar gain is most likely. This use of building form and function exemplifies the energy efficient concept utilizing natural modes of operation only the where internal thermal loads can be met. In addition, the PowerGen building uses controlled night-time ventilation, a high thermal insulation building fabric, window shading, light-shelves and double-glazed low emissivity glazing.

1.3 Chilled Surfaces

For the intermediate thermal loads on a building (including mixed mode buildings), cooling can be provided by ventilation, chilled ceilings or beams. Chilled ceilings and beams, (chilled surfaces), provide an intermediate cooling strategy to offset increasing thermal loads before the implementation of full air conditioning. By using low grade cooling, (the cooling medium is only a few degrees below ambient room temperature), chilled surfaces have a low plant primary energy use in comparison to full air conditioning systems.

Research into chilled surfaces has concentrated on thermal comfort, the control of condensation and the energy performance and integration with conventional mechanical ventilation systems. Little work has been performed on chilled beam surface integration with natural ventilation.

Chilled surface thermal comfort issues focus on the radiant coupling and downdraught effects upon occupants. Heiselberg [19], investigated draught risk from cold vertical surfaces and developed expressions for cold natural convective air flows along the floor beneath a chilled surface. Heiselberg showed that the velocity of the cold convective flow and therefore, that the thermal comfort measured in terms of the percentage of persons dissatisfied decreased within the first two metres
of the chilled surface. The percentage of persons dissatisfied was nearly constant in the rest of the room.

Kümpmann [20], investigated the thermal comfort for chilled ceilings integrated with an upward displacement ventilation system. Using thermal mannequins a full-scale experiment showed, for different internal thermal loads, chilled ceilings guarantee high level thermal comfort. Nui [21], investigated the thermal comfort and ventilation effectiveness of chilled ceilings integrated with an upward displacement ventilation system and a ceiling air supply system. Nui showed through simulation that chilled ceilings in combination with an upward displacement ventilation system gives good performance in both thermal comfort and ventilation effectiveness, and that the performance at a cooling load of 50 W/m² is equivalent to that of a displacement system alone at 25 W/m². Further investigation showed that an air ceiling system, (ventilation air supplied at ceiling level), in combination with a chilled ceiling produced enhanced downward convective flows that tend to increase the draught risk along the floor, and also reduce the ventilation effectiveness.

Livtchak et al [22], compared the cooling performance of static and active chilled beams and showed that the active chilled beam design had the greater cooling capacity. Livtchak, like Nui, stressed the importance of the thermal environment for chilled beam selection; the location of air distribution devices and heat sources in a room having a large effect upon performance and thermal comfort. Wyatt [23], in an empirically validated CFD study of chilled beams integration with a displacement ventilation showed ventilation effectiveness increased due to the combination of the convective air flows which lifted air pollutants higher than a displacement system alone. In addition, it was shown that chilled beams are capable of a degree of self-balancing; in situations with no heat gain, the convective output, which was around 85 %, dropped to zero, whereas the radiant output remained the same.

Brunk [24], compared the energy use of different air conditioning strategies in real buildings. Brunk stated that if an adequately selected overall cooling system is utilized, for example, ice storage making favourable use of off-peak electrical energy, then total energy costs of a chilled ceiling system can be reduced by 50 % compared to a variable air volume system.

Martin and Alamdari[16] have examined the extent to which condensation occurs
1.3. Chilled Surfaces

<table>
<thead>
<tr>
<th>Time (mins)</th>
<th>Condensation Formation.</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Room reaches dew point of 15.5°C</td>
</tr>
<tr>
<td>90</td>
<td>Pipe ‘perspiring’ with 1mm diameter ‘pimples’.</td>
</tr>
<tr>
<td>170</td>
<td>2mm-diameter pimples start to coalesce to form water film. 3mm-diameter droplets form on underneath of pipe.</td>
</tr>
<tr>
<td>240</td>
<td>Droplets run together and fall.</td>
</tr>
</tbody>
</table>

Table 1.1: Observations of Condensation Formation

on chilled surfaces, and has reviewed the available control strategies. Table 1.1 shows the typical rate of condensation formation on copper pipe-work, (which forms the chilled ceiling), with an inlet chilled water temperature 14°C, when the room dew point temperature changes from 14 to 15.5°C. It can be seen that it is several hours before droplets start to fall from the chilled surface. Further, Martin and Alamdari shows the additional thermal resistance of the chilled surface panel increases the surface temperature and thus the time for droplets to fall.

Martin and Alamdari[16] states for conventional chilled surfaces the selection of the optimum control strategy (and appropriate set-points) should not allow the development of condensation. Equally, it should not unduly limit the cooling output from the surface. Comparison of control strategies demonstrated that control of supply (room) air relative humidity to maintain a low saturation temperature is the most effective method of avoiding condensation. However, as latent cooling of the air stream is energy intensive, maintaining a constant temperature differential between the chilled water and the room saturation temperature is the energy efficient alternative. However, the control strategy and the energy requirement to maintain the saturation temperature differential would be un-necessary if condensation were allowed to form.

Reducing the surface temperature can increase the cooling capacity of chilled surfaces, yet, the surface temperature of conventional chilled beams and ceilings must be kept above the dew-point temperature of the enclosure in order to avoid condensate forming on the surface. Sub-dew point chilled beams utilise the conventional chilled surface mechanisms of convection and radiation, but can have a surface temperature at or below the dew-point which would provide increased cooling capacity to further offset the implementation of the mechanical plant associated with full air conditioning.
1.4 Sub-Dew Point Chilled Beam

In a sub-dew point chilled beam system chilled water is passed through the chilled beam at or below the enclosure saturation temperature. Condensation is allowed to form on the chilled surface which produces a latent heat addition to the total heat flux, (Figure 1.1). Positioned high on internal walls, sub-dew point chilled beam design is such that the condensate is allowed to fall from the chilled surface onto a drip tray.

To date, little research has been directed towards the modelling of sub-dew point chilled beams either within mixed mode strategies or as standalone systems. Hirayama (1998), has investigated the heat and mass transfer regimes of chilled water radiators, (conventional room radiators purposely brought below the dew point temperature of the room). Mathematical models describing the regimes were empirically validated using climate chamber tests. Hirayama states that the chilled radiator can improve both thermal comfort and energy efficiency and, by partially or fully matching the sensible and latent component of the radiator to the load requirements of a building, it may offer a definite alternative to conventional air conditioning systems.

Due to the low thermal capacity of the chilled beam, the system is able to provide a rapid response to the zone thermal loads and therefore an effective control strategy is required to ensure the correct response of the system to the zone loads. Given that the sub-dew point chilled beam has a higher cooling capacity than conventional chilled surfaces it is anticipated that a sub-dew point chilled beam will off-set the introduction of mechanical plant until higher internal thermal loads are encountered. In order to maximise enclosure thermal comfort, optimal control of the chilled beam is necessary so that beam cooling output and chiller operation is scheduled to meet the thermal loads in the enclosure. This thesis investigates the optimum operation of a sub-dew point chilled beam for a range of mixed-mode building operation strategies.

1.5 Research Objectives

The aim of this research is to investigate the performance and integration of
1.5. Research Objectives

The approach to the research is to investigate the performance and integration of a sub-dew point chilled beam by using a numerical optimization algorithm and a zone thermal and mass flux model to identify the optimum operating characteristics of the beam for a range mixed-mode strategies.

1.5.1 Strategic Objectives

The strategic objectives of this research are as follows:

- to develop a zone thermal model that is capable of predicting the zone thermal response to a given mixed mode strategy; to develop a steady state model of a sub-dew point chilled beam; combine the zone and sub-dew point chilled beam model; the robustness of the model must be examined within
1.5. Research Objectives

the range of operating conditions for each mixed mode approach (Chapter 2);

- to develop an zone moisture model that is capable of predicting the enclosure moisture response to a given mixed mode strategy; the robustness of the model must be examined within the range of operating conditions for each mixed mode approach (Chapter 3);

- to develop steady state models of the heating ventilation and cooling (HVAC) plant, natural ventilation, infiltration and sub-dew point chilled beam chiller plant; the robustness of the model must be examined within the range of operating conditions for each mixed mode approach (Chapter 4);

- to examine the model parameters and testing procedures; to introduce an optimization routine and an energy and comfort performance indicator (Chapter 5);

- to apply an optimization algorithm to the simulation, and investigate the performance of the optimization algorithm in finding an optimum solution; to investigate the performance and integration of the sub-dew point chilled beam in a optimized passive mixed mode strategy for a range of climatic conditions (Chapter 6);

- to investigate the performance and integration of the sub-dew point chilled beam in an optimized active mixed mode strategy for a range of climatic conditions (Chapter 7);

- to draw conclusions and suggest possible areas where there is the potential to conduct further research (Chapter 8).
Chapter 2

Zone Dynamic Thermal Model

Introduction

The building thermal model is the central element of the sub-dew point chilled beam performance analysis, since it is the building thermal response from which the comfort performance is derived. The thermal model has to be able to simulate the building thermal response to a variety of heat disturbances, since it is the main aim of the research to model the performance and integration of a sub-dew point chilled beam in mixed mode buildings. To this end it is necessary to be able to model the thermal response of a building to a range of modes of operation.

The thermal model has to simulate the dynamic response of the zone air and building fabric elements to external diurnal temperature changes. Performance modelling of the sub-dew point chilled beam includes simulation of the moisture response of the enclosure, easy integration of moisture and thermal response of both the chilled beam and the enclosure is therefore also essential.

Simulation of thermal storage in the building fabric must also be simulated. Therefore the thermal model must include treatment of real material properties. The model must also be flexible enough to model three different ventilation strategies, natural, forced fan and mechanical (displacement and mixing) ventilation. The sub-dew point chilled beam model must describe both the sensible and latent heat transfer phenomena. The sub-dew point chilled beam is modelled in thermal
network form using a partially wet heat exchanger analogy.

This chapter describes a dynamic thermal network model of an enclosure and a sub-dew point chilled beam. The model addresses the heat exchange between building fabric elements and the internal air conditions, and the heat storage within the building fabric. The enclosure moisture model is described in Chapter 3.

2.1 Building Thermal Models

Building thermal modelling has been the subject of much research. Clarke [26] states that building simulation models should include:

- the transient conduction of heat through the enclosure;
- casual gains from occupancy, lighting and equipment;
- infiltration and ventilation through the zone;
- the effects of shortwave solar radiation on exposed and internal surfaces;
- longwave radiation exchange between exposed external surfaces and the sky and surroundings;
- window shading effect and the shading from surrounding buildings;
- the varying convective effect;
- plant characteristics and controller strategy;
- effects of moisture.

Models that address these requirements fall into two main categories [27]; the heat balance method and the weighting factor method. In the heat balance method, transient thermodynamic heat balance equations are solved simultaneously. The weighting factor method applies a weighting factor to each heat disturbance in an enclosure and sums the effect of each. Each weighting factor, which for a dynamic simulation would change every time step, is calculated in advance based upon
2.1. Building Thermal Models

the relative contributions of each heat disturbance derived from a heat balance approach. As this is computational intensive, the weighting factor method which is essentially a solution strategy, is considered inappropriate for this project.

Heat balance methods, according to Clarke [26], are thermal events described by differential and partial differential equations and can be described by five strategies; steady state, simple dynamic, response function, numerical methods and electrical analogue. The simplest method, steady state, omits transient response treating enclosure steady state heat flow only. Steady state modelling cannot simulate the dynamic response of the enclosure air and building fabric due to the changes in diurnal activity that is associated with typical buildings, plant operation, heat disturbances, occupancy patterns. Simple dynamic methods address dynamic performance using regression, (or other statistical strategies), to assess performance.

Response function methods model transient conduction and inter-zone energy balance relationships using either time-domain response or frequency-domain response functions to solve partial differential equations. Time-domain methods pre-process the response of a system to unit excitation and use these 'response factors' to calculate the heating or cooling load. Frequency-domain methods represent climatological data as a series of periodical functions and calculate the influence of any heat flux as a harmonic of the cyclic contribution. This method, the admittance method, is used as a standard design approach adopted by the Chartered Institute of Building Services Engineers (CIBSE) [28]. The admittance method regards the diurnal 24 hour cycle as the most important. The external temperature variations are treated as periodic with a single cycle of 24. Cycles of other frequencies are neglected, this simplification invalidates the use of the admittance method for this project due to the necessity for accurate input weather data, which will have more complicated external temperature variations.

Numerical methods approximate the derivatives of heat balance differential and partial differential equations. The numerical approximation technique depends upon the accuracy required by the simulation. Numerical methods such as the finite element and the finite volume method involve setting up a mesh of discrete nodes [26] at pre-selected points of interest. Then for each of these elements a heat flow simulation equation is derived from the governing partial differential equation which links the separate elements. Then the nodal equation sets have to be solved
2.1. Building Thermal Models

simultaneously. For an enclosure, the number of nodes to accurately evaluate the internal response to diurnal temperature changes would be large and the resultant computational time immense, thus invalidating the use of finite volume or finite element method for this project.

Electrical analogue methods analogize heat flow as an electrical flow by representing room thermal conditions as electrical circuits. Heat flux sources are represented as either current or voltage sources, considerable thermal storage capacity is represented as capacitance. Heat flow paths, conduction, radiation and convection, are represented as resistances. These methods are sometimes referred to as time constant methods as the response of the capacitance to thermal disturbance is represented by a $RC$ time constant which simulates the dynamic thermal response of the building fabric. Nodes represent the considered temperatures, which typically are, building fabric mass and surface temperatures and the enclosure air temperature. Usually one node is used to represent each building fabric element, some simple electrical analogue methods assign all the building fabric elements to one node [31]. This method of "lumping" building components together lends the electrical analogy its other name, the Lumped Parameter Method. Lumped parameter assignment prevents any analysis of the temperature profiles through a building fabric element as it is represented by one uniform temperature.

Ozişik [30], showed by application of the Biot number that the lumped parameter method can be used with reasonable confidence only when the Biot number is less than 0.1, where the Biot number is given by:

$$\text{Bi} = \frac{h \cdot L_c}{\lambda} \quad (2.1)$$

where $h$ is the surface heat transfer coefficient (W/m²K), $L_c$ is the characteristic length (m), and $\lambda$ is thermal conductivity (W/mK). For a uniform brickwork wall of 0.22 m thick the Biot number is approximately 0.4. In reality, a wall construction is not infinite and usually not of a uniform property, and due to the difference in air temperature between the external and internal surfaces of the wall, the temperature difference inside the wall is inevitable. However, the ambient boundary air temperature changes in a similar way to a periodic wave, and rarely with a sudden change of large amplitude. In addition, the temperature difference between the fabric and boundary air is not substantially large. The Biot number does not
impose a strong restriction on the application of the lumped parameter method for modelling building fabric components, since the temperature distribution within the component can be approximately neglected due to the relatively mild boundary condition. The lumped parameter model is useful for building energy analysis and design purposes where the influence of major heat disturbances on the building environment can be effectively assessed, without modelling the building in great detail (such as the mass temperature of each layer of a component).

A lumped parameter thermal network approach, based on Mathews et al. [31], has been chosen in this research to model the thermal performance of the enclosure and sub-dew point chilled beam. The approach is attractive for three reasons; (a), model parameters are easily derived from the thermal properties of the building materials, (b), the approach simulates dynamic response, and (c), the approach is computationally fast, the optimization study requires that computational time be kept to a minimum. Further, the approach is easily interpretable, an advantage when different mixed mode systems are to be simulated, and is easily integrated with the moisture model, the thermal nodes corresponding to certain moisture transfer processes. In addition, this approach has been shown to perform well in the simulation of fabric thermal storage [32].

In a recent review of simulation methods performed by the UK Department of the Environment [33, 34], lumped parameter simulation methods were found to be the most suitable for the analysis of energy performance. Lombard [35] showed that a first order lumped parameter model can work to high levels of accuracy in comparison with empirical results. For a wide range of building types a lumped parameter model worked to an accuracy of 2°C for 80% of measurements. A further study by Mathews [36] showed for 70 validation studies in 30 buildings a lumped parameter model worked to an accuracy of 3°C for 95% of measurements.

2.1.1 Electrical Analogue Thermal Network Models

Thermal time constant electrical analogue thermal network models describe the behaviour of the building fabric and the enclosure air to thermal disturbances. These models have been the subject of much research [31, 35, 37, 38, 39]. The importance of thermal time constants in assessing the thermal characteristics has been studied extensively [40, 41, 42, 43]. Bruckmayer [40] first postulated the
concept of a thermal time constant, a simple ratio of steady state terms, for determining transient heat flow phenomenon through individual building elements. Pratt and Ball [41], showed, using exact analytical comparisons, that for both homogeneous and multilayer building elements Bruckmeyer's thermal time constants explain the transient heat flow.

Raychaudhuri [42], extended the thermal time constant approach, most of the previous work concentrating upon homogeneous wall construction, to include structures of different construction and radiative heat exchange between surfaces producing an integrated thermal time constant. This integrated time constant encompassed the entire enclosure, enabling the investigation of thermal mass, ventilation and inter-surface radiant exchange. However, once derived this time constant is only applicable for buildings of similar construction. Givoni [43], extended this work by producing time constants for each building element then producing an averaged area weighted time constant for the whole enclosure. Again, this time constant is only applicable for buildings of similar construction, and too complicated for design purposes.

In its simplest form a building can be modelled as in Figure 2.1, this represents a one node model of the heat flux through a wall, the flux direction being inside to outside. $T_o$ is the outside air temperature, °C, $T_a$ the inside air temperature, $C$ the thermal capacitance of the wall, kJ/K, and $R$ the thermal resistance of the wall, W/m²K. The one node model can be solved by applying a heat balance on the node associated with the capacitance:

$$C \frac{dT}{dt} = q - \frac{(T_a - T_o)}{R} \quad (2.2)$$

This gives a single time constant or first order response of the building to disturbances $q$ and $T_o$. The building in this simple model is represented by only two parameters, the capacitance, $C$, and the resistance $R$. An expansion of the one node model, by Mathews et al [31], lumps the distributed thermal resistance and capacitance of the whole building shell into four parameters. This simplified thermal network for a one node (first order) building zone is shown in Figure 2.2.

In Figure 2.2, $T_{sa}$ represents the sol-air temperature acting upon the exterior surfaces, $Q_{r(t)}$ is the shortwave radiative heat source acting upon the internal
2.1. Building Thermal Models

![One Node Thermal Network](image)

Figure 2.1: One Node Thermal Network

![First Order Thermal Network Model](image)

Figure 2.2: First Order Thermal Network Model
2.1. Building Thermal Models

structure, \( Q_{ct(t)} \) is the convective heat input from ventilation, lighting, occupants and equipment. \( T_{e(t)} \) is the external air temperature. Three resistance paths are shown in the simplified thermal network, \( R_o \) is the area weighted total resistance of all surfaces \( k \) and is given by:

\[
R_o = \frac{\sum_k A_k}{\sum_k \left( \frac{A_k}{R_k} \right)}
\]  

(2.3)

\( A_k \) represents the area of building element \( k \). \( R_{e(t)} \) is the infiltration resistance given by \( \frac{1}{V} \), \( V \) is the enclosure volume, \( C_p \), is the specific heat capacity of the air, \( N \) is the air change rate per second. \( R_a \) is the resistance between the indoor air and the interior surfaces, representing the resistance of the radiative and convective heat transfer coefficients. The thermal storage of the massive elements is given by \( C \), the capacitance, this gives the effective thermal capacity of the building zone and is expressed for the total shell area. This method of representing the resistances of the massive elements only allows the consideration of a disturbance from one direction.

A more advanced model, which considers the thermal response of building elements in both directions, was developed by Hassid [39]. Whilst producing a linear model for passive solar evaluations, Hassid developed the total thermal time constant method to represent the building element as an electrical analogue, introducing the concept of lumped thermal resistances. Consider the multi-layer construction, Figure 2.3, and the thermal network representation, Figure 2.4. \( R_o \) and \( R_i \) represent the thermal resistance of the construction to disturbances from outside and inside respectively. The total heat capacity of the construction is given by \( C \). The time constants which represent the response of the building fabric, are given by:

\[
\tau_i = \sum_{k=1}^{n} \left( C_k \left( R_{si} + \sum_{j=1}^{k-1} R_j + \frac{R_k}{2} \right) \right)
\]  

(2.4)

\[
\tau_o = \sum_{k=1}^{n} \left( C_k \left( R_{so} + \sum_{j=k+1}^{n} R_j + \frac{R_k}{2} \right) \right)
\]  

(2.5)

\( R_j \) and \( R_k \) are the actual resistances of layers \( j \) and \( k \) in the building fabric, \( R_{si} \) and \( R_{so} \) are the resistances of the internal and external surfaces. Given the time
2.1. Building Thermal Models

Figure 2.3: Multi-Layer Construction

Figure 2.4: Multi-Layer Construction Lumped Parameter Model
2.2 Zone Thermal Model

The zone thermal model expands the simplified first order model by treating each massive element separately. Separate treatment of each massive element is necessary in this approach for two reasons; firstly, the radiative exchange between the chilled beam and each of the massive elements will be different due to the orientation and physical geometry, and secondly, to allow investigation of thermal storage in different thermal weights of enclosure.

Figure 2.5, shows the lumped parameter network. $T_a$ is the inside air temperature, $T_o$ the external drybulb temperature. $T_{sa}$ is the sol-air temperature. $T_{ac}$ and $T_{bf}$ represent the temperature above the ceiling and below the floor respectively. $T_{hi}$

constants and the total thermal capacitance of the construction, \( C = \sum_{k=1}^{n} C_k \), the lumped resistances can be found from $R_i = \frac{C}{T}$ and $R_o = \frac{C}{G}$.

Tindale [34], showed that first order model suffered from inadequacies. The identical treatment of radiative and convective gains produced erroneous treatment of ventilation. The model also had poor dynamic performance for heavyweight spaces and only approximate modelling of interzone conduction. Tindale proposed modifications which significantly improved the accuracy of the lumped parameter model without compromising the simplicity or speed. The modifications are; the separate treatment of radiative and convective heat transfer and additions of further time constants to include partition and solid floor thermal capacity. The model performance showed close agreement with a detailed finite difference model.

A lumped parameter model approach based on both Mathews et al [36], and Tindale [34], which utilize Hassid's lumped resistance algorithm, has been chosen in this research to model the thermal performance of the enclosure and the sub-dew point chilled beam. The approach is attractive in that the model parameters are easily derived from actual material properties, and that the model output is easily interpreted. Separate treatment of the convective and radiant heat exchange is also attractive as the sub-dew point chilled beam heat exchange with the enclosure can be analysed. The approach has also been successful in modelling energy efficient buildings incorporating ventilated floor slabs for thermal storage [32].
Figure 2.5: Enclosure Thermal Network Model
2.2. Zone Thermal Model

represents the temperature of the adjacent internal zones. The temperatures of the surfaces in the enclosure are \( T_{sc}, T_{sw}, T_{sf} \) and \( T_{swp} \). The temperatures of the mass nodes within the building fabric are given as \( T_{mc}, T_{mw}, T_{mf} \) and \( T_{mw} \). The subscripts represent the ceiling, walls, floor and partition walls. \( T_{rs} \) is the radiant star index temperature.

\( Q_c \) is a convective gain to the room from occupants, equipment and the ventilation system. \( Q_r \) is short wave radiative gain to the enclosure. \( C_c, C_w, C_f, C_{wp} \) are the effective thermal capacities of the mass of the ceiling, walls, floor and partition walls. \( C_a \) is the thermal storage capacity of the air within the enclosure, typically this thermal capacity is considered negligible in comparison to the thermal capacity of the fabric elements [32], it is considered because of the emphasis placed upon performance within different mixed mode ventilation strategies.

\( R_{rfj} \) gives the radiative resistances for all the surfaces. \( R_{cj} \) gives the convective resistance between all surfaces and the internal temperature node. For resistance through the massive elements the nomenclature is \( R_{o} \) and \( R_{ij} \). This represents the external and internal resistance for all the surfaces. \( R_{e} \) is the resistance for the infiltration loss.

2.2.1 Long-Wave Radiation

Heat flux incident upon the building fabric includes both convective and radiative flux. An equivalent temperature which includes the effects of the convective and radiative flux is the sol-air temperature [28]. The long-wave radiation and convective flux incident on the external wall and the roof are modelled using the sol air temperature, \( T_{sa} \). \( T_{sa} \) is calculated according to [28]:

\[
T_{sa} = (\alpha I_t - \epsilon I_I) R_{so} + T_o
\]

(2.6)

Where \( I_t \) is the total irradiance upon the external surface, W/m². \( R_{so} \) is the external surface resistance. \( \alpha \) and \( \epsilon \) are the absorptance and the emissivity of the external surface. \( T_o \) is the outside air temperature. \( I_I \) is the net long wave radiation loss to the ground, sky or the environment dependant upon cloud clover, external dry-bulb temperature and orientation and is given for values of cloudiness, \( C_1 \), as
2.2. Zone Thermal Model

Vertical Surface.

$$I_t = 93 - 79C_i$$ \hfill (2.7)

Horizontal Surface.

$$I_t = 21 - 17C_i$$ \hfill (2.8)

$C_i$ is taken as being between 0 and 1, clear and completely cloudy respectively, and has been taken as 0.5 for this project. The total irradiance upon the external surface is given as:

$$I_t = I_D + 0.5I_d + 0.5g_rI_{TH}$$ \hfill (2.9)

$I_{TH}$ is the global horizontal radiation which is corrected by the ground reflectance, $g_r$. $I_d$ is the diffuse radiation from the sky, $I_D$ is the direct beam radiation calculated for the external surface from the direct normal beam radiation, $I_{D,N}$, as:

$$I_D = I_{D,N} \cos i$$ \hfill (2.10)

$$\cos i = \cos \theta \sin \phi + \sin \theta \cos \phi \cos \gamma,$$ \hfill (2.11)

where: $i$ is the angle of solar incidence on the external surface. $\phi$ is solar altitude and $\gamma$, the wall-solar azimuth, $\theta$ is the surface tilt, for a vertical surface $\theta = 90^\circ$. The wall-solar azimuth and the solar altitude are calculated from surface orientation, solar time, daily solar declination and location latitude (CIBSE, [28]).

Direct normal beam radiation $I_{D,N}$, diffuse horizontal radiation from the sky $I_d$, and global horizontal radiation $I_{TH}$ are all found from weather data.
2.2.2 Short-Wave Radiant Heat Gain

The short-wave radiant heat gain through the glazing to the enclosure is given by $Q_r$. Using the weather data the value of the short-wave gain is calculated hour by hour. The total gain absorbed by each surface is given by the product of the total short-wave gain $Q_r$, the proportion of radiation incident on each surface $P$ and the short-wave radiant coefficient for each surface, $\beta$. For example, the shortwave gain to the floor is given as:

$$Q_{r,f} = \beta_f P_f Q_r$$  \hspace{1cm} (2.12)

The proportion of the shortwave radiation incident upon a surface is calculated from relative surface areas; geometry, particularly view factor configuration, is not considered in this model.

2.2.3 Radiant Heat Exchange

The radiant heat exchange between surfaces is modelled using the radiant star index temperature [44], a fictitious construct which acts as an intermediate temperature node. Widely used in previous thermal models [32], it was chosen for this thermal network model. The validity of the radiant star index temperature has been assessed by comparing it with a radiation resistance network model (Appendix A). It was concluded that the radiant star temperature node is a valid approach for this thermal network model.

The radiant star index temperature is given for the network as:

$$T_r = \frac{R_{rc}T_{sc} + R_{rw}T_{sw} + R_{rf}T_{sf}}{R_{rc} + R_{rw} + R_{rf} + R_{rpm}}$$  \hspace{1cm} (2.13)

The values of $R$, the radiant resistance for the ceiling, wall, floor and chilled beam depend upon the emissivity, radiant heat transfer coefficient and the surface area [44] and are given by:
2.2. Zone Thermal Model

\[ R_{ri} = \frac{1}{A_i E_i h_r} \]  \hspace{1cm} (2.14)

\( A_i \) is the surface area of each element, \( h_r \) is the linearised radiant heat transfer coefficient, (Appendix A). \( E_i \) is a function of the emissivity of the surface and is weighted by the area of the building element. \( E_i \) can be found from [44]:

\[ \frac{1}{E_i} = \frac{1 - e_i}{e_i} + Y_i \]  \hspace{1cm} (2.15)

\[ Y_i = 1 - \frac{A_i}{A_{total}} \]  \hspace{1cm} (2.16)

e_i is the surface emissivity, \( A_{total} \) is the sum of the surface areas of the walls, ceiling, floor and chilled beam.

2.2.4 Convective Resistances

The convective resistances \( R_{cc}, R_{cw}, R_{cf} \) and \( R_{cwp} \) are given by [30]:

\[ R_{ci} = \frac{1}{h_{ci} * A_{ci}} \]  \hspace{1cm} (2.17)

\( A_i \) is the surface area of each building element, \( h_{ci} \) is the convective heat transfer coefficient. The value of this for walls, floor and ceiling is given by CIBSE [28]. The value of the convective heat transfer for the chilled beam is derived in the thermal plant model, Chapter 4.

\( R_v \) the thermal resistance for infiltration loss is given by [28]:

\[ R_v = \frac{1}{\frac{1}{3}NV} \]  \hspace{1cm} (2.18)

Where \( N \) is the number of air changes per hour and \( V \) is the volume of the enclosure. \( N \) is calculated from the effective air leakage equation[28], described in the
2.2.5 Thermal Capacitance

The thermal storage capacitance is given for the ceiling, walls, partition walls, floor and air as $C_c$, $C_w$, $C_{wp}$, $C_f$ and $C_a$ and is found from [44]:

$$C_i = m_iC_{pi} \quad \text{(2.19)}$$

where $m_i$ is the mass of the building fabric element and $C_{pi}$ is the specific thermal heat capacity. The capacitance of all the elements of the composite building structure are summed to give a total value for the capacitance. For the external wall the thermal capacity of the glazing is ignored [32].

2.2.6 Structure Resistances

The massive element resistances, $R_{oj}$ and $R_{ij}$ representing the external and internal resistance for all surfaces $j$, are calculated using the lumped resistance algorithm given by Hassid, (section 2.1.1). Different elements within each massive element, such as the glazing within the external wall, are integrated in parallel.

2.3 Model Validation

Validation of results given by any simulation method is essential. A four level test for model validation is proposed by Wortman et al [47] and Wiltshire [48]:

Theoretical: examining and testing the physical, mathematical and software components of the model.

Analytical: input chosen so output can be predicted and compared with a known analytical output.
Comparative: inter-model comparison, for when analytic comparison is not applicable.

Empirical: matching test building data to model simulation output/results.

The theoretical level of model performance testing has been performed through the literature review which cites the appropriateness of the models selected. Analytical validation is applicable for the performance testing of the separate models; dynamic thermal, latent and the steady state plant models, ruling out a comparative testing procedure here. Empirical validation is not applicable as there are no test buildings available which utilises sub-dew point chilled beam technology.

Analytical validation of the simulations includes steady state testing, a model initialization test and known response testing. Steady state testing executes the simulation at constant external conditions to determine whether steady state conditions are predicted by the model. Model initialization testing executes the simulation for a number of days of identical weather conditions, to establish that the model will return identical diurnal profiles irrespective of the initial estimate. Known response testing executes the simulation and introduces a heat disturbance which has a predetermined effect. The simulation response is compared to the expected result.

An important aspect of the validation is that the model must produce the "characteristic" behaviour of the zone, absolute accuracy of the model is less critical for this study.

2.3.1 Constructions Validated

Three main zone fabric types have been included in the model validation, light, medium, and heavy-weight buildings. The constructions were taken from a quantitative comparison of cooling load calculation procedures [46]. Tables 5.1, 5.2 and 5.3, (Chapter 5), summarize the general characteristics of each of the building fabric types. Air gaps have been given a constant resistance of 0.18 $m^2.K/W$.

The sub-dew point chilled beam in the validation study is constructed from aluminium, chosen for its conductivity and corrosion resistance. The thermal storage
of aluminium is very small when compared to the storage capacity of the massive elements of the building fabric, and is ignored in the thermal network treatment of the beam. The thermal conductivity of the chilled beam aluminium construction is $160 \text{ W/m.K}$.

### 2.3.2 Thermal Model Validation

Figure 2.6 shows the steady state profiles for the ceiling mass, enclosure air node and internal zone air node. The steady state simulation ran with a constant external air temperature of $30.8 \, ^\circ\text{C}$ and a constant internal zone air temperature of $26 \, ^\circ\text{C}$. With a time step of 600 seconds all the temperature nodes reached steady state within 120 hours. The enclosure air node, having a lower thermal capacity, reaches steady state before the ceiling mass node. The steady state profiles of the ceiling and air temperature match the expected result, in that they lie between the internal and external driving temperature, with the ceiling node being physically closer to outside having a temperature closest to the external temperature, the air temperature being closer to the surrounding internal zone temperature.

Figures 2.7 and 2.8 show profiles for the model initialization tests. The initialization tests were run for 10 identical days for nodal initial conditions of 10, 15, 20 and $25 \, ^\circ\text{C}$ for both heavy and light weight building fabrics. All tests produced identical diurnal profiles within 100 hours (4 days) of simulation, figure 2.7 shows the diurnal profiles for initial conditions of 20 and $25 \, ^\circ\text{C}$ and the temperature comparison between each test. Figure 2.8 shows the diurnal profiles for initial conditions of $20 \, ^\circ\text{C}$ for both light and heavy weight building fabric. The temperature difference comparison (light weight minus heavy weight) shows that identical diurnal profiles are achieved after 100 hours and that the two building weights do not predict the same temperature profiles. This result is expected, the heavy weight building having greater thermal capacity will have a lower diurnal amplitude than the light weight enclosure. The profiles show a phase shift which is indicative of the difference in response of the thermal weights.

Figures 2.9 and 2.10 show expected response diurnal profiles of the enclosure air, ceiling mass and ceiling surface node for light weight and heavy weight enclosures respectively. The profiles are for the "hot" summer day with the period of occupancy being 08:00 to 18:00 hours. At the start of occupancy the thermal heat load
2.3. Model Validation

Figure 2.6: Steady State Profiles

Figure 2.7: Model Initialization Test
is increased due to both occupant and equipment heat loads. In both the heavy and light weight cases the response of the temperature nodes is the same and is a result of the nodal thermal capacity; the air node reacts quickest to the thermal load, followed by the ceiling surface node and lastly the ceiling mass node. The mass node in both cases does not reach a maximum until approximately 12 hours after the introduction of the heat load. Comparison between the heavy and light weight profiles for the mass node, Figure 2.11, show a reduction in the amplitude and the time to reach the peak temperature for the heavy weight nodal response, again a result of its greater thermal capacity, which is expected.

Figure 2.12, shows the effect of occupant density upon the enclosure air temperature. The solid line represents an occupant density of one person for every 3.7 m$^2$, the legislative maximum for an office space, equivalent to approximately 40 people, the dash dot line represents an office with one person. Each simulation ran under identical conditions for ten days, the occupant density changing only on the last day. The effect of the occupant heat load is clear, for all the test days during the occupied period the enclosure air temperature increases due to the extra heat load.
2.3. Model Validation

Figure 2.9: Light Weight Expected Response Profiles

Figure 2.10: Heavy Weight Expected Response Profiles
2.3. Model Validation

2.3.3 Summary

A building thermal model has been described and rigorously tested. The building model expands upon an established lumped parameter method of thermal modelling by treating all massive elements and convective and radiative heat transfer fluxes separately, enabling the investigation of thermal storage and heat transfer. A four part model validation technique has been introduced and the thermal model has been rigorously tested. Steady state testing, model initialization testing and known response testing all produced expected analytical results. An important aspect of the validation is that the model must produce the "characteristic" behaviour of the zone, absolute accuracy of the model is less critical for this study, this has been shown throughout the validation. The next section describes the sub-dew point chilled beam modelling approach.
Figure 2.12: Expected response: Occupancy Variation
2.4 Sub-Dew Point Chilled Beam Model

The sub-dew point chilled beam model predicts the heat transfer rates from the enclosure to the sub-dew point chilled beam. The mechanisms that contribute to the total heat flux are radiation and convection heat transfer plus condensation mass transfer which provides a latent heat addition to the total heat transfer. To facilitate easy integration with the enclosure dynamic thermal response model the sub-dew point chilled beam is modelled as an electrical analogue.

2.4.1 Electrical Analogue Model

The heat transfer through the beam is simulated using a heat exchanger analogy which is easily integrated into the lumped parameter approach. An average chilled water temperature is found using an exponential expression of the heat transfer across the beam. The combined heat and mass transfer is modelled by the approximate "sensible heat ratio" method, and the combined effects of convection and radiation "lumped" in a rad-air formulation. The heat transfer path from the enclosure "rad-air" temperature to the average chilled water temperature is described by three resistances, the air side resistance, the water side resistance, and the chilled beam resistance. Detailed descriptions of these approaches are outlined below.

The air side convective and radiant heat transfer from the surface of the beam has been modelled to take place between the average surface temperature of the beam ($T_{b,av}$), and the rad-air temperature ($T_{ra}$). This is shown in thermal network form, Figure 2.13, $R_a$, represents the air side thermal resistance, $R_w$, the water side resistance and $R_b$ the thermal resistance of the beam walls.

The water side convective heat transfer from the inside surface of the beam has been modelled to take place between the average chilled water temperature, ($T_{aw}$), and the average inside surface temperature of the beam, ($T_{bi}$). Total heat transfer to the beam, $Q_{tot}$, is given by:

$$Q_{tot} = \frac{T_{ra} - T_{aw}}{R_{tot}}$$  \hspace{1cm} (2.20)
2.4. Sub-Dew Point Chilled Beam Model

where: $R_{tot}$ is the thermal resistance between the rad-air node and the average chilled water node.

Rad-Air Temperature

The rad-air node is a construct used to combine the two modes of heat transfer to the chilled beam. Represented in Figure 2.14, the rad air node is given as the resistance weighted combination of the radiant star index temperature ($T_{rs}$), and the air temperature node, ($T_a$) [44]:

$$T_{ra} = \frac{1}{\frac{1}{R_{rb}} + \frac{1}{R_{cb}}} \cdot \frac{1}{R_{rb}} T_a + \frac{1}{\frac{1}{R_{cb}} + \frac{1}{R_{rb}}} \cdot \frac{1}{R_{cb}} T_{rs}$$

(2.21)

where: $R_{rb}$ is the radiative resistance, $R_{cb}$ is the convective thermal resistance.
Average Chilled Water Temperature

Since the chilled beam exchanges heat between the chilled water and the averaged rad-air temperature, it is in effect acting as a heat exchanger with one fluid condensing or evaporating. This form of heat exchanger can be integrated into lumped parameter thermal network models using the average temperature of the fluid in the heat exchanger [32]. In this instance, the average water temperature ($T_{aw}$), is given by:

$$T_{aw} = T_{ra} + \frac{T_{win} - T_{ra}}{\gamma L} \left(1 - e^{\gamma L}\right)$$  \hspace{1cm} (2.22)

where: $T_{win}$ is the water inlet temperature and $\gamma$ is the number of transfer units per unit length of heat exchanger; $L$ being the length of the beam.

The number of transfer units per unit length, is a function of the total resistance of the beam $R_{total}$, the water mass flow rate $\dot{m}$ and the water specific heat capacity $C_p$: 
The combined heat and mass transfer has been modelled by the approximate “sensible heat ratio” method in which the air side heat transfer coefficient is corrected to allow for the increase in total heat flux associated with the mass transfer [72]:

\[ R_{\text{total}} = SHR \times R_a + R_b + R_w \]  \hspace{1cm} (2.24)

\( SHR \) is the ratio of sensible heat transfer to the total heat transfer. \( R_a \) is given by:

\[ R_a = \frac{1}{\left( \frac{1}{R_{a_s}} + \frac{1}{R_{a_t}} \right)} \]  \hspace{1cm} (2.25)

\subsection*{2.4.2 Thermal Resistances}

Three thermal resistances combine to produce the total thermal resistance across the sub dew point chilled beam, air side, beam fabric and water side resistance. Air side thermal resistance includes both the radiant and free convective heat flux path to the chilled beam surface. Beam fabric thermal resistance take into account the thermal conductivity of the beam. Water side resistance accounts for both the forced and free convection between the cooling medium and the inside surface of the chilled beam. All the thermal resistance are highly dependant upon geometry of the beam, for this project the beam is approximated as a horizontal pipe.

\section*{Radiative Resistances}

The radiative resistance depends upon the emissivity, radiant heat transfer coefficient and the surface area[44] and is given by:

\[ R_{rb} = \frac{1}{A_b E_b h_r} \]  \hspace{1cm} (2.26)
2.4. Sub-Dew Point Chilled Beam Model

$A_b$ is the external surface area of the chilled beam, $h_r$ is the linearised radiant heat transfer coefficient, Appendix A. $E_b$ is a function of the emissivity of the surface and is weighted by the total area of the contributing surfaces in the enclosure. $E_b$ is found from [44]:

$$\frac{1}{E_b} = \frac{1 - e_b}{e_b} + Y_b \quad (2.27)$$

$$Y_b = 1 - \frac{A_b}{A_{total}} \quad (2.28)$$

e$_b$ is the chilled beam surface emissivity, dimensionless, $A_{total}$ is the sum of the surface areas of the walls, ceiling, floor and the chilled beam.

Convective Resistances - Air Side

Air side convective resistance derivation depends upon the flow regime, laminar or turbulent, and is highly dependant upon the beam geometry and orientation, and is found from:

$$R_{cb} = \frac{1}{h_m A_b} \quad (2.29)$$

where: $h_m$ is the average convective heat transfer coefficient, $W/m^2K$, found from Nusselt equations. The Nusselt equation for a horizontal pipe with external free convection is given as:

$$Nu_m = \frac{h_m D}{k} = c (Ra_D)^n \quad (2.30)$$

where: $D$, the pipe diameter, is the characteristic dimension of the free convection equation, $m$. $h_m$ is the average heat transfer coefficient over the entire length of the plate, $W/m^2K$, $k$ is the thermal conductivity of the chilled beam. The magnitude of the Rayleigh Number, $Ra_L$, determines the mode of free convection, laminar or turbulent, and is found from:
2.4. Sub-Dew Point Chilled Beam Model

\[ Ra_L = Gr_L Pr \]  \hspace{1cm} (2.31)

\( Pr \) is the Prandtl Number, the ratio of molecular diffusivity of momentum to the molecular diffusivity of heat. \( Gr_L \) is the Grashoff Number, as for most gases \( Pr \leq 0.7 \), it is this which determines the magnitude of the Rayleigh Number and the flow regime. The Grashoff Number is given by:

\[ Gr_L = \frac{g\beta (T_w - T_\infty) L^3}{\nu^2} \]  \hspace{1cm} (2.32)

where \( g \) is the gravitational acceleration, \( \beta \) is the volume coefficient of thermal expansion, \( \nu \) is the kinematic viscosity of the fluid. In the case of ideal gases \( \beta \) is taken to be:

\[ \beta = \left\{ \frac{T_w - T_\infty}{2} \right\}^{-1} \]  \hspace{1cm} (2.33)

The Rayleigh Number for the flat plate is in the turbulent range \( 10^9 < Ra_L < 10^{13} \), which gives the values of the coefficients \( c \) and \( n \) as 0.125 and \( \frac{1}{3} \).

Convective Resistances - Water Side

The chilled beams mode of operation means that the water side convective resistance derivation depends not only on the flow regime, laminar or turbulent, but whether the heat transfer is free or forced. Water side convective resistance is found from:

\[ R_w = \frac{1}{A_i h_m} \]  \hspace{1cm} (2.34)

where: \( A_i \) is the internal surface area of the beam.
2.4. Sub-Dew Point Chilled Beam Model

Forced Convection

When the beam is operational forced convection heat transfer coefficients are found from the Dittus-Boelter equation[30]. For smooth pipes and for small to moderate temperature differences:

\[ Nu = \frac{h_m D}{k} = 0.023 Re Pr^n \]  \hspace{1cm} (2.35)

where: \( Re \) is the Reynolds number and is given by \( \frac{u_m D}{\nu} \). \( \nu \) is the kinematic viscosity of the chilled water, \( u_m \) is the velocity of the chilled water flow. \( Pr \) is the Prandtl number taken at the average chilled water temperature.

Free Convection

When the beam is non-operational free convection heat transfer coefficients are found from the mean of four equations that approximate the inside of a pipe; two vertical flat plate coefficients, an upwards facing horizontal flat plate coefficient and a downwards facing flat plate coefficient [74]. The Nusselt equations for each mode are identical in form:

\[ Nu = \frac{h_m X}{k} = c (Gr X Pr)^n \]  \hspace{1cm} (2.36)

\( X \) is the characteristic dimension of each mode. The coefficients \( c \) and \( n \) are given as 0.4 and \( \frac{1}{3} \) for horizontal upwards facing, 0.27 and \( \frac{1}{4} \) for horizontal downwards facing, and 0.1 and \( \frac{1}{3} \) for both vertical plates.

Conductive Resistances

Conductive resistance depends upon geometry, for a cylinder \( R_b \) is given as:

\[ R_b = \frac{\ln \frac{r_o}{r_i}}{2\pi k_{beam} L} \]  \hspace{1cm} (2.37)
2.4. Sub-Dew Point Chilled Beam Model

\( r_o \) and \( r_i \) are external and internal radii of the chilled beam, \( k_{beam} \) is the chilled beam material thermal conductivity, \( L \) is the chilled beam length.

For the flat plate \( R_b \) is given as:

\[
R_b = \frac{l}{A_b k_{beam}} \tag{2.38}
\]

where: \( l \) is the plate thickness and \( A_b \) is the surface area of the plate.

2.4.3 Beam Model Validation

The sub-dew point chilled beam model was validated using the strategy outlined by Wiltshire [48] and Wortman [47]. As with the enclosure dynamic thermal model, the beam model can only be validated using the analytical validation techniques of steady state testing, model initialization testing and expected response testing. Steady state and model initialization tests have already been performed upon the thermal network, thus the sub-dew point model will be validated using known response testing.

The expected response testing was carried out using the simplest of the chilled beam configurations, the horizontal pipe, with one beam, having 0.5 m external diameter, with an internal diameter of 0.495 m. The model was run for ten days, for the hot day weather condition, to ensure that the diurnal profiles are independent of the initial conditions. The expected response tests are; modulating the valve between fully closed and fully open, increasing the physical size of the chilled beam, and increasing the supply temperature of the chilled water.

Figure 2.15 shows the diurnal profiles of enclosure air and radiant temperature plus the predicted mean comfort vote (PMV) for the chilled beam valve closed (dotted line), one third (dot dash line), two thirds (dashed line) and fully open (solid line). The expected profiles for these cases are that when the valve is fully open the cooling effect of the chilled beam is greatest and when the valve is closed the cooling effect at a minimum. The profiles mirror this effect, as the valve opens the cooling effect increases and the enclosure cools, this is shown by both the air and radiant temperature. Also the PMV becomes less positive throughout
the profile indicating a more comfortable, cooler, environment. The difference in temperature between beam on and beam off is greater on the radiant temperature profile than the air temperature profile, which would indicate that the beam has greater radiant the convective heat exchange. Further, the difference between the profiles is less marked as the valve modulates open, the small physical size of the beam means the effective change in heat transfer is small within the range of operation, which may lead to insensitivity of optimized solutions.

Figure 2.16 shows the diurnal profiles of enclosure air and radiant temperature plus the predicted mean comfort vote (PMV) for two sizes of chilled beam, with
external diameters of 0.1 (dot-dash line) and 0.5 (dashed line) m and internal diameters of 0.095 and 0.495 m respectively. It is expected that the beam with the greater physical size will exhibit the greater heat transfer, and cool the room the most. This is shown in the diurnal profiles, the air and radiant temperature profiles all show reductions, indicating cooling. The PMV profile is less positive throughout the day indicating cooler, more comfortable, conditions. Again, the radiant temperature profile is reduced more than the air temperature.

Figure 2.17 shows the diurnal profiles of enclosure saturation (dew-point) temperature, chilled beam surface temperature, and the chilled beam condensation
2.4. Sub-Dew Point Chilled Beam Model

Figure 2.17: Expected Response: Saturation Temperature

production, for a chilled water supply temperature of 6.4 and 13.4 °C. The aim of this test is to show that the condensation production does not occur above the saturation temperature of the room. Both sets of graphs show that whenever the saturation temperature is higher than the beam surface temperature condensation occurs, and that when the beam surface temperature and the saturation temperature coincide, or when the beam surface temperature is higher, condensation production ceases, proving the concept.
2.4.4 Summary

An electrical analogue model of a sub-dew point chilled beam has been developed. The model uses a simple heat exchanger analogy to model the heat exchange and a sensible heat ratio algorithm to model the combined sensible and latent heat flux. The model has been vigorously tested using an expected response validation test procedure, and has been shown to produce the expected response. The test procedure highlighted the need to evaluate the relative proportions of the main heat transfer mechanisms, convection, radiation and condensation, when the integration and performance of the chilled beam in the different mixed mode strategies is tested.

2.5 Zone and Sub-Dew Point Chilled Beam Model Integration

The sub-dew point chilled beam model is an electrical analogue, and is therefore easily integrated with the enclosure electrical analogue through the convective and radiative heat transfer paths. Figure 2.18, the composite electrical analogue, describes the integration of the beam network model with the enclosure network model. The beam external surface temperature, $T_{boe}$, convectively couples with the enclosure air node, $T_e$, the radiant heat transfer path couples the beam external surface to the radiant star temperature node, $T_{rs}$. Short wave radiation is also incident upon the beam, represented by $Q_r$.

2.6 Thermal Model Implementation

Based on the network model, Figure 2.18, 13 equations can be established to describe the relationships of the thermal variables.

For the external walls:
Figure 2.18: Combined Thermal Network
For the partition walls:

\[ C_p \frac{\partial T_{mp}}{\partial t} = \frac{T_{si} - T_{mp}}{R_{cp}} - \frac{T_{mp} - T_{sp}}{R_{ip}} \]  

(2.40)

For the floor:

\[ C_f \frac{\partial T_{mf}}{\partial t} = \frac{T_{sf} - T_{mf}}{R_{af}} - \frac{T_{mf} - T_{sf}}{R_{if}} \]  

(2.41)

For the ceiling:

\[ C_c \frac{\partial T_{mc}}{\partial t} = \frac{T_{ac} - T_{mc}}{R_{oc}} - \frac{T_{mc} - T_{sc}}{R_{ic}} \]  

(2.42)

For the air node:

\[ C_a \frac{\partial T_a}{\partial t} = Q_c + \frac{T_o - T_a}{R_o} - \frac{T_a - T_{sc}}{R_{cc}} - \frac{T_a - T_{sw}}{R_{cw}} - \frac{T_a - T_s}{R_{cf}} - \frac{T_a - T_{sp}}{R_{cp}} - \frac{T_a - T_{sbo}}{R_{cb}} \]  

(2.43)

The heat balance in the room can be represented by the following equations:

\[ T_{sc} = \frac{Q_{rc} + \frac{1}{R_{ic}} T_{mc} + \frac{1}{R_{ec}} T_{rs} + \frac{1}{R_{ce}} T_a}{\frac{1}{R_{ic}} + \frac{1}{R_{ec}} + \frac{1}{R_{ce}}} \]  

(2.44)

\[ T_{sw} = \frac{Q_{rw} + \frac{1}{R_{iw}} T_{mw} + \frac{1}{R_{ew}} T_{rs} + \frac{1}{R_{cw}} T_a}{\frac{1}{R_{iw}} + \frac{1}{R_{ew}} + \frac{1}{R_{cw}}} \]  

(2.45)

\[ T_{sf} = \frac{Q_{rf} + \frac{1}{R_{if}} T_{mf} + \frac{1}{R_{ef}} T_{rs} + \frac{1}{R_{ef}} T_a}{\frac{1}{R_{if}} + \frac{1}{R_{ef}} + \frac{1}{R_{ef}}} \]  

(2.46)
2.6. Thermal Model Implementation

\[ T_{sp} = \frac{Q_{rp} + \frac{1}{R_{rp}}T_{mp} + \frac{1}{R_{rp}}T_{rs} + \frac{1}{R_{cp}}T_{a}}{\frac{1}{R_{rp}} + \frac{1}{R_{rp}} + \frac{1}{R_{cp}}} \]  \hspace{1cm} (2.47)

\[ T_{sbo} = \frac{Q_{rb} + \frac{1}{SHR_{a}}T_{ra} + \frac{1}{R_a + R_w}T_{aw}}{\frac{1}{SHR_{a}} + \frac{1}{R_a + R_w}} \]  \hspace{1cm} (2.48)

\[ T_{ra} = \frac{\frac{1}{R_{cb}}T_{a} + \frac{1}{R_{cb}}T_{rs}}{\frac{1}{R_{cb}} + \frac{1}{R_{cb}}} \]  \hspace{1cm} (2.49)

\[ T_{aw} = T_{ra} + \frac{T_{win} - T_{ra}}{\gamma L} (1 - e^{\gamma L}) \]  \hspace{1cm} (2.50)

\[ T_{rs} = \frac{Q_{lr} + \frac{1}{R_{ec}}T_{sc} + \frac{1}{R_{ec}}T_{sw} + \frac{1}{R_{ef}}T_{sf} + \frac{1}{R_{rp}}T_{sp} + \frac{1}{R_{cb}}T_{sbo}}{\frac{1}{R_{ec}} + \frac{1}{R_{ec}} + \frac{1}{R_{ef}} + \frac{1}{R_{rp}} + \frac{1}{R_{cb}}} \]  \hspace{1cm} (2.51)

Equations 2.44 to 2.51 can be rearranged to a linear equation set:

\[ XT_{\text{surface}} = YT_{sm} \]  \hspace{1cm} (2.52)

where, \( T_{\text{surface}} \) is the surface temperature vector,

\[ T_{\text{surface}} = (T_{sw}, T_{sc}, T_{sf}, T_{sp}, T_{sbo})^{-1} \]  \hspace{1cm} (2.53)

\( T_{sm} \) is treated as the known independent variable vector in Equation 2.52,

\[ T_{sm} = (T_{mw}, T_{mc}, T_{mf}, T_{mp}, T_{win}, Q_{r}, Q_{c}, Q_{lr})^{-1} \]  \hspace{1cm} (2.54)

\( X \) is a 5 \times 5 matrix, and \( Y \) is a 5 \times 8 matrix, consisting of the resistances and coefficients in the equations.

A matrix can be used to represent Equations 2.39 to 2.43:
2.6. Thermal Model Implementation

\[
A_m \ddot{T}_m = ST_m + KT_u
\]  
(2.55)

where, \(\ddot{T}_m\) is the differential vector of the air and mass node temperatures,

\[
\ddot{T}_m = (\dot{T_{mw}}, \dot{T_{mc}}, \dot{T_{mf}}, \dot{T_{mp}}, \dot{T_a})^{-1}
\]  
(2.56)

\(T_m\) is the air and mass node temperature vector. \(T_u\) is the heat disturbance vector in the state space equation,

\[
T_u = (T_{ac}, T_{sc}, T_{sa}, T_{sw}, T_{bf}, T_{sf}, T_{zi}, T_{sp})^{-1}
\]  
(2.57)

\(A_m\) is a \(5 \times 5\) diagonal matrix containing the time constants (RC) for each mass node. \(S\) is a \(5 \times 5\) matrix, and \(K\) a \(5 \times 8\) matrix.

If the \(T_{surface}\) vector (Equation 2.52) is substituted in Equation 2.55, the state space equation becomes:

\[
\dot{T}_m = WT_m + VU
\]  
(2.58)

where, \(U\) is the heat disturbance vector consisting of all the driving variables considered in the model,

\[
U = (T_{sa}, T_{ac}, T_{bf}, T_{zi}, T_0, Q_r, Q_{ir}, Q_c)^{-1}
\]  
(2.59)

and \(W\) and \(V\) are \(5 \times 5\) and \(5 \times 8\) matrices, containing the known coefficients and resistances. Equation 2.58 is a typical state-space equation, the analytical solution for which is:

\[
T_{m,t} = \exp^{Wt} T_{m,0} + \int_0^t \exp^{W(t-\varphi)} VU(\varphi) d\varphi
\]  
(2.60)
where, ϕ is an integral time variable. \(T_{(m),t}\) and \(T_{(m),0}\) are the temperature vectors at time \(t\) and for the initial state. The computational implementation of this analytical solution can be difficult, particularly in the choice of the period over which the integration is performed. Additionally, \(W\) and \(U\) are time dependant, that is the thermal resistance on the water side is flow dependant, and the water flow rate changes with time, (as the valve opens and closes). The time dependency of \(W\) and \(U\) makes equation 2.60 non linear and therefore difficult to solve. Therefore the purely numerical, Runge Kutta method is used to solve Equation 2.60.

### 2.7 Summary

The development of the enclosure dynamic thermal building model in this chapter includes both the background theory and model validation. The building model expands upon an established lumped parameter method of thermal modelling by treating all massive elements and convective and radiative heat transfer fluxes separately, enabling the investigation of thermal storage and heat transfer. A four part model validation technique has been introduced and the thermal model has been rigorously tested. Steady state testing, model initialization testing and known response testing all produced expected analytical results.

The sub-dew point chilled beam model has been developed which incorporates a heat exchanger analogy and a sensible heat ratio algorithm, allowing the assessment of the mechanisms of heat transfer, namely, convection, radiation and condensation. Analytical validation of the sub-dew point chilled beam model produced expected responses, and highlighted the necessity of an investigation into the relative proportions of the sensible and latent heat transfer mechanisms.
Chapter 3

Zone Latent Model

Introduction

Modelling the performance and integration of a sub-dew point chilled beam within mixed mode strategies requires a whole building approach which must include the prediction of both thermal and latent response to disturbances. The latent response of a zone consists of separate moisture transfer processes, moisture absorption/desorption, (chilled beam surface) condensation, infiltration, exfiltration, natural and mechanical ventilation, evaporation and generation.

In this research the latent model must be able to simulate the response of all "typical" moisture transport processes in an enclosure, to show the enclosure air response to disturbances. The latent model must also be easily integrated within the zone thermal model and give an easily interpretable response for input to the sub-dew point chilled beam and HVAC model. Additionally, the model has to be flexible to allow the investigation of the different mixed mode strategies.

The prediction of indoor moisture content, particularly if humidity is to be controlled, and the change in moisture content due to moisture transfer processes, is important when assessing "real" building performance and the primary energy usage of mechanical plant. Further, for this project the prediction of condensation moisture transfer rate enables the evaluation of the additional (latent) cooling capacity of a sub-dew point chilled beam. According to Kusada [49], evaluation of
indoor air humidity level is extremely important for a number of reasons:

- the occurrence of condensation over the windows and other cold surfaces, which becomes a major cause of material deterioration, should be minimized;
- levels of some indoor contaminants are influenced by the humidity level, e.g., formaldehyde;
- comfort of occupancy is influenced by the level of humidity, especially when it is extremely high or low;
- durability of interior furnishings will be affected by room humidity, e.g., mildew growth at a high humidity level and cracking and flaking at low humidity level.

Further to this an accurate dynamic model for simulating indoor air humidity, which includes all moisture transport processes, is essential. This chapter describes a linear differential equation model which, using discrete time steps enables the linearization of the non-linear moisture transport processes. The model describes
the interaction of the moisture transport phenomena and the resultant response of the indoor air humidity. The linearized moisture transport properties considered within the latent model are: moisture absorption/desorption, chilled beam surface condensation, infiltration, evaporation, generation, and natural and mechanical ventilation, Figure 3.1.

3.1 Moisture Transport Models

Until recently[50], building simulation research has been almost entirely concerned with the transport of sensible heat flux in buildings. Humidity calculation procedures existed, but not with the sophistication of the heat transfer simulation models[49]. Also humidity models tended to ignore one important moisture transport process, that of absorption and desorption of moisture in building fabric materials and furnishings. This section shows the selection of an enclosure moisture content response model and the background to the research into the moisture transport processes of condensation mass transfer and absorption/desorption. Models of moisture generation, (Chapter 3), natural and mechanical ventilation and air infiltration, (Chapter 4), are taken from standard text and are described in detail.

Given that it has been reported[51], that approximately one third of the moisture generated in a room is absorbed by room surfaces. Early moisture models focused on a mass balance between the room indoor moisture generation rate and the humidity dilution by air leakage. Consequently, by neglecting important moisture transport processes, the predicted moisture levels in the modelled enclosure were often markedly different to measured results. For example, in a study of thermal environmental conditions of survival shelters by Flanigan and Morrison[52] a large difference was found between predicted levels evaluated using an early mass balance moisture model and measured moisture levels.

Kusada[49], using a concept by Tsuchiya [51], which included moisture absorption and desorption prediction rates improved air humidity predictions and showed close agreement between predicted moisture levels and experimental data from a test house. However, coefficients of moisture transport were found from empirical data and are not commonly available, or easily applied to different enclosures. A
more recent study by Martin and Verschoor[53] clearly showed the effect of absorption and desorption in a typical office. From empirical data they showed that an air-conditioning system would have to be operated for an additional 1.67 hours to remove moisture stored in building elements during a night ventilation period. Therefore, inclusion of the absorption / desorption moisture transfer process will provide better prediction of the sub-dew point chilled beam performance in the different mixed mode strategies.

Combined thermal and moisture modelling has been studied. Early attempts at this considered the thermal and moisture fluxes separately, having the enclosure moisture prediction as either a pre- or post-processor to the thermal model. In a three dimensional finite element model which extended the original mass balance models, Fairey et al [54] coupled a pre-processing absorption / desorption model to an established thermal building model. Barringer [50] continued this work by adding moisture transport through basement walls, window condensation and evaporation and heat recovery ventilators to the finite element model. However, as with thermal finite difference schemes, the number of nodes to effectively model the moisture transfer processes is large and therefore, computationally intensive. Millar [55] used a lumped parameter method to incorporate a moisture transport model as a post processor for a hourly sensible load program. Computationally quicker than the finite element methods the simulation technique included an electrical analogue which enabled the moisture flow to be characterized simply with a time constant and a hydroscopic storage constant. Simultaneous treatment of thermal and moisture fluxes has also been considered, Fairey et al [54] used sophisticated analytical models to predict the dynamic response of various cooling strategies. Again, as with the finite element method, the simultaneous solution of thermal and moisture flux is computationally intensive and therefore considered inappropriate for this project.

In this research the latent model has to be integrated with the thermal model, and it is important that the model results can be easily integrated into, and interpreted by, both the sub-dew point chilled beam model and the HVAC system model. Also the latent model has to be computationally fast and give an accurate dynamic moisture response to disturbances. An approximate analytical approach based on that by Diasty et al [56, 57], has been chosen for this project. This easily interpretable approach uses a computationally fast numerical solution strategy which uses discrete time steps to linearize the highly non-linear moisture transport
3.1. Moisture Transport Models

phenomena, and includes all the moisture transfer processes that occur in real buildings.

3.1.1 Moisture Absorption and Desorption

All materials that bound the room, wall surfaces, furnishings etc, have the capacity to store and release moisture. Inclusion of this phenomena, absorption and desorption, is necessary for a complete building model. Moisture absorption and desorption by building materials has been investigated by both experimental and theoretical method [50, 54, 57, 58]. It has been found that changes in ambient indoor relative humidity elicit different responses from different building materials, although most materials follow an exponential function for absorption and desorption. A lumped parameter analysis has been performed on the absorption/desorption process [58] by matching the convective mass transfer coefficient and the effective moisture capacity in theoretical treatments with empirical data. Absorption/desorption has also been studied using electrical analogy [55], with capacitance representing material moisture capacity and voltage representing vapour pressure.

Many research papers agree [50, 54, 57, 58] that a method of ascertaining the moisture behaviour of a construction is by the application of the Biot number. The value of the Biot number indicates the relative contribution of diffusive and convective processes and hence the contribution of the inner regions of the material to moisture exchange. The Biot Number is the ratio of the material resistance to moisture transfer to the convective mass transfer resistance:

$$Bi = \frac{(V_m/A_e)/D_v}{1/h_m}$$

(3.1)

where: $V_m$ is the volume of the absorbing material, $A_e$ is the exposed area, $D_v$ is the material vapour diffusion coefficient and $h_m$ is the surface mass transfer coefficient. A high value of $Bi$ indicates the moisture transfer process is controlled by the diffusive resistance, and instant equilibrium can be achieved. For small values of $Bi$ diffusive moisture transport dominates, with moisture conditions across the material thickness tending to be constant. Most building materials have moderate Biot numbers.
Kusada [49], determined the existence of a boundary layer in a solid surface that is affected by the diurnal changes in room air conditions. A lumped parameter analysis has been performed to determine the depth of this boundary layer [58], and the interaction with the room air. The moisture interaction depth, $L_m$, is given as the ratio of material volume to exposed area. If long term storage, in the order of days, occurs and the inner regions of a material participate in the mass exchange, boundary layer lumped parameter analysis can fail [57]. Then approximate analytic methods must be used. One such method, (Diasty [56]), addresses moderate Biot number materials by producing a model which relies only on material moisture diffusivity and the absorption/desorption isotherm slope, and it is this approach which has been chosen in this project to simulate the moisture absorption/desorption mass transfer, (section 3.2.1).

3.1.2 Condensation Mass Transfer

Condensation mass transfer occurs when a vapour impinges upon a colder surface whose temperature is below the dew-point temperature of the vapour. It has been established that two modes of condensation exist. If, when the condensate is formed, the liquid wets the surface (hydrophilic), filmwise condensation occurs. If the condensate does not wet the surface (hydrophobic) then dropwise condensation occurs.

Dropwise condensation has by far the greatest condensation rate and much research has been performed to promote hydrophobic condensation[59]. Chemical dropwise promoter performance and physical chemistry has been also been studied [59]. Dropwise and filmwise condensation have been examined theoretically by Nusselt and Schmidt [30]. Despite several simplifying assumptions the derived equations still form the basis of condensation prediction. Combining the laws of laminar flow of a liquid with conduction heat transfer, by Grant [60], Nusselt's assumptions are:

- heat given up by the vapour is latent heat;
- heat is transferred through the liquid by conduction;
- temperature gradients through the liquid are linear;
3.2 Room Humidity Moisture Balance

- solid surface is smooth and clean;
- solid surface temperature is constant;
- the only vapour flow is towards the condensing surface;
- the liquid drains from the surface in laminar flow;

Industrial application of condensation heat transfer has been studied. Dehumidification of air flow for air conditioning systems is of particular interest in the building services field. Work has been carried out to investigate the performance of heat exchangers [61, 62], and dehumidification on finned heat exchangers [63, 64, 65].

Spalding [66] showed that mass transfer processes are capable of description in terms of (a) a relationship, the same for all types of mass transfer, relating transfer rate to the Reynolds number; and (b) a driving force, called the transfer number, which is different for each transfer process. The transfer number for condensation and evaporation in air-vapour mixtures was given in terms of mass concentration or enthalpy. This approach, which utilizes a computationally fast algorithm, produces easily interpretable results and, utilizing the identical treatment of both evaporation and condensation, is easily portable, and has been selected for the simulation of mass transfer in this project.

3.2 Room Humidity Moisture Balance

The latent model simulates the moisture transport processes within an enclosure. The model, using discrete time steps to enable the linearization of the non-linear moisture transport processes, evaluates the change in room air moisture content due to changes in moisture transport processes within the enclosure. With reference to Figure 3.1, the moisture transport processes are; natural and mechanical ventilation, infiltration, condensation, evaporation, moisture generation, absorption and desorption.

The discretized moisture balance for a change in room air moisture content for the moisture transport processes[56], is given by:
3.2. Room Humidity Moisture Balance

\[
\frac{\Delta g_r}{\Delta t} = \frac{1}{\rho_a V} \left[ \sum_{i=1}^{n_a} \frac{\Delta \dot{m}_{s,i}}{\Delta t} + \sum_{i=1}^{n_{a,n}} \frac{\Delta \dot{m}_{a,i}}{\Delta t} + \sum_{i=1}^{n_{a,n}} \frac{\Delta \dot{m}_{c,i}}{\Delta t} + \sum_{i=1}^{n_{a,n}} \frac{\Delta \dot{m}_{e,i}}{\Delta t} + \sum_{i=1}^{n_{a,n}} \dot{m}_{g,i} \right] \tag{3.2}
\]

where, \(\frac{\Delta g_r}{\Delta t}\) is the rate of change in indoor air moisture content, kg/kg s. \(\frac{\Delta \dot{m}_{s}}{\Delta t}\) is the rate of moisture absorption or desorption by interior materials, kg/s. \(\frac{\Delta \dot{m}_{a}}{\Delta t}\) is the rate of moisture added or removed due to air movement across room boundaries, kg/s. \(\frac{\Delta \dot{m}_{c}}{\Delta t}\) is the rate of moisture removed by condensation, kg/s. \(\frac{\Delta \dot{m}_{e}}{\Delta t}\) is the rate of moisture added by evaporation, kg/s. \(\dot{m}_{g}\) is the rate of moisture generation from indoor sources, kg/s. \(\rho_a\) is the air density kg/m³, \(V\) is the zone air volume.

Each moisture transfer process is calculated separately, independant of the other processes, and the algebraic sum gives the enclosure moisture content change over the prescribed time step. The derivation of each of the moisture transfer processes is described below, with the exception of natural and mechanical ventilation which are described in Chapter 4.

3.2.1 Absorption and Desorption

The change in room air moisture content due to absorption or desorption over a discrete time interval, is given by (Diasty et al [56]):

\[
\frac{\Delta \dot{m}_s(t)}{\Delta t} = \frac{A_m h_{m_s}}{\rho_a V} \left[ g_{m_s}(t) - g_{m_s} \right] \tag{3.3}
\]

\(A_m\) is the surface area of the absorbing/desorbing material, \(\rho_a\) is the air density kg/m³, \(V\) is the room volume. \(h_{m_s}\) is a surface mass transfer coefficient, (Kusada [49]), as an average for materials in an enclosure as \(5 \times 10^{-9} kg/ Pa m^2 s\). \(g_{m_s}\) is the initial material moisture content, kg/kg, \(g_{m_s}(t)\) is the material moisture content at the end of the discrete time interval, kg/kg, found from the governing equation for moisture diffusion:

\[
\frac{\partial g_m}{\partial t} = \alpha_m \frac{\partial^2 g_m}{\partial x^2} \tag{3.4}
\]
3.2. Room Humidity Moisture Balance

where: \( g_m \) is the material moisture content, \( kg/kg \), \( \alpha_m \) is the material moisture diffusivity, \( m^2/s \), \( t \) is the time interval and \( x \) is the position within the material, \( m \). This is solved for boundary and initial conditions. Material moisture content change with material thickness is zero at the characteristic maximum moisture interaction depth, \( Lm \):

\[
\frac{\partial g_m}{\partial x} = 0 \quad \text{at } x = L_m \tag{3.5}
\]

At the surface, material moisture content change is controlled by the relative contribution of diffusive and convective mass transfer:

\[
-D_v \frac{\partial g_m}{\partial x} = h_{ms} (g_r - g_m) \quad \text{at } x = 0 \tag{3.6}
\]

Initially the material moisture content is a function of the depth of material interaction:

\[
g_m = F(x) \quad \text{at } t = 0 \tag{3.7}
\]

The governing equation can be solved using the approximate analytical technique, the moment method \([67]\). Assuming a linear behaviour of the surface moisture conditions during the discrete time interval, the surface moisture content of the given absorbing/desorbing surface \( g_{ms} \), at the end of a discrete time step \( t \), is given by \([57]\):

\[
g_{ms}(t) = A/Z \tag{3.8}
\]

where: \( A \) and \( Z \) are given as:

\[
A = \left[ (-217b_1 + 1302b_2 - 1302b_3) \exp(-42\tau) \right]
\]

\[
+ \left[ (168b_1 - 1008b_2 + 1008b_3) \exp(-52\tau) \right]
\]
3.2. Room Humidity Moisture Balance

\[ + \left[ \left( \frac{217}{2520\tau} - \frac{1519}{420} \right) \exp\left( -42\tau \right) \right] \frac{h_m L_m}{D_v} g_{mo} \]

\[ + \left[ \left( \frac{14}{5} + \frac{1}{15\tau} \right) \exp\left( -52\tau \right) \right] \frac{h_m L_m}{D_v} g_{mo} \]

\[ + \left[ \left( \frac{1}{24\tau} + \frac{13}{12} \right) \exp\left( -10\tau \right) \right] \frac{h_m L_m}{D_v} g_{mo} \]

\[ - \left[ \left( \frac{\tau}{2} + \frac{1}{45\tau} \right) \right] \frac{h_m L_m}{D_v} g_{mo} \]

\[ + \left[ \frac{1519}{420} \exp\left( -42\tau \right) - \frac{14}{5} \exp\left( -52\tau \right) \right] \frac{h_m L_m}{D_v} g_r \]

\[ - \left[ \frac{13}{12} \exp\left( -10\tau \right) + \frac{\tau}{3} \right] \frac{h_m L_m}{D_v} g_r \]

\[ (3.9) \]

\[ Z = 1 + \frac{h_m L_m}{D_v} \left( \frac{\tau}{2} - \frac{1}{45\tau} + \frac{1}{3} \right) + \frac{h_m L_m}{D_v \tau} \]

\[ * \left[ \frac{1}{15} \exp\left( -52\tau \right) + \frac{1}{24} \exp\left( -10\tau \right) - \frac{217}{2520} \exp\left( -42\tau \right) \right] \]

\[ (3.10) \]

\[ \tau = \frac{\alpha_m t}{L_m^2} \]

\[ (3.11) \]

$L_m$ is the moisture interaction depth [57] and is given by the ratio of the material volume to its exposed area. $\alpha_m$ is the absorbing/desorbing material moisture diffusivity, m$^2$/s. Material moisture diffusivity is the ratio between vapour permeability and volumetric moisture capacity and is analogous to thermal diffusivity given by:

\[ \alpha_m = \frac{D_v}{\rho_m C_m} \]

\[ (3.12) \]

$D_v$ is the material vapour diffusion coefficient, kg/mPas. $C_m$ is the material
3.2. Room Humidity Moisture Balance

Moisture capacity, kg/kgPa, and is defined as the increase in the mass of moisture in unit mass of the material that follows unit increase in vapour pressure or suction. Material moisture capacity is given by:

\[
C_m = \frac{\xi}{P_{ss}}
\]  

(3.13)

\(\xi\) is the material moisture sorption slope, kg/kg. \(P_{ss}\) is the saturated vapour pressure in the enclosure. \(\xi\) is represented graphically in Figure 3.2, it shows that the moisture content in the material can be found for any given zone relative humidity. The slope for adsorption and desorption for any porous material is different, (Figure 3.3). For the purposes of the moisture model such hysteresis is ignored, \(\xi, D_e\) are material moisture properties and are unique to each material [68].

For a uniform initial moisture distribution, \(b_1, b_2\) and \(b_3\) are initial condition parameters, and are given by:

\[
b_1 = g_{mi}, b_2 = \frac{g_{mi}}{2}, b_3 = \frac{g_{mi}}{3}
\]  

(3.14)

In reality all porous building materials in an enclosure will absorb and desorb simultaneously at different rates, but for simplification in the moisture model only three materials will be considered to be absorbing in this research. These materials are pine, carpet and spruce, representing typical office materials. The change in room moisture content is summed for each porous material.

Data for material moisture properties is only just being compiled and found empirically [68], data for many typical office materials has yet to be compiled.

3.2.2 Moisture Transport across Enclosure Boundaries

Air movement, and subsequently moisture transport, across enclosure boundaries is the fastest, and as a result the most important moisture process. Moisture transport across enclosure boundaries includes (natural) ventilation, infiltration and exfiltration. A function of air flow rates and the moisture content of the air
3.2. Room Humidity Moisture Balance

Depending on the material, capillary condensation may start anyway in the range.

Figure 3.2: Sorption curve of a porous building material
3.2. Room Humidity Moisture Balance

Figure 3.3: Absorption/desorption Hysteresis
movement, the change in indoor room moisture content due to air entering the zone across enclosure boundaries is given by [56]:

\[
\frac{\Delta m_a(t)}{\Delta t} = \frac{Q_a}{V} [g_o - g_r(t)]
\]  

(3.15)

\(Q_a\) is the air flow rate of the entering air, \(m_3/s\), \(g_o\) is the external moisture content, \(kg/kg\). \(g_r(t)\), the room moisture content due to moisture transport at the end of a discrete time step is given as:

\[
g_r(t) = g_o + (g_{ro} - g_o) \exp\left\{ -\frac{Q_a}{V} t \right\}
\]  

(3.16)

where: \(g_{ro}\) is the last time step room moisture content.

\(Q_a\) is evaluated for each moisture transport flow path, (natural and mechanical ventilation and infiltration), and the contribution to the change in moisture content of the enclosure is found. The air flow rate for mechanical ventilation is discussed along with natural ventilation and infiltration flow rates in Chapter 4.

### 3.2.3 Condensation and Evaporation

Condensation in the enclosure is taken to be caused solely by the sub-dew point chilled beam. Prediction of the condensation rate and the change in room air moisture content uses the mass transfer theory outlined below. Evaporation in the enclosure is taken to be caused from the condensate deposited upon a drip tray beneath the chilled beam. Evaporation rate prediction utilizes the same mass transfer theory as condensation, the film temperatures and characteristic lengths being the only difference.

#### Convective Mass Transfer

Convective mass transfer flux can be derived using a direct comparison with convective heat transfer. In convective heat transfer with a surface, momentum
3.2. Room Humidity Moisture Balance

exchange and convective heat exchange in similar flows are assumed equal; the Reynolds Analogy. Using the same premise for convective mass transfer, exchange of mass in a convective flow can be assumed to be equal to the rate of mass transfer from the surface over which the flow passes. The Reynolds Analogy/Hypothesis can be summarised as:

- convective transport of the mass flux is independent of direction;
- the magnitude of mass transfer conductance, \( T \), is not dependant upon the concentration gradients or chemical reaction;
- mass transfer rates are low.

All three conditions of the Reynolds Hypothesis are met by the chilled beam and the methods of mass transfer, condensation and evaporation. From Spalding [66], and using the Reynolds Analogy, ((Appendix B), the mass transfer flux in the absence of chemical reaction, \( \dot{m}'' \), \( kg/m^2s \), is given by:

\[
\dot{m}'' = \frac{\Delta \dot{m}_{c,d}}{\Delta t} = TB
\]  

(3.17)

where: \( B \) is the dimensionless Transfer Number. \( T \) is the convective mass transfer coefficient evaluated under the Reynolds Analogy, \( kg/m^2s \), and is given by:

\[
T = \frac{h_c}{C_p} Le^\frac{3}{2}
\]

(3.18)

where: \( h_c \) is the convective heat transfer coefficient, \( W/m^2K \), derived for each chilled beam type (Chapter 4). \( C_p \) is the specific heat capacity of the air, \( J/kg K \). \( Le \) is the Lewis number, a dimensionless ratio of the Prandtl number and the Schmidt number. Under the conditions of the Renolds Analogy both the quotient of the Prandtl and Schmidt number are unity, hence:

\[
T = \frac{h_c}{C_p}
\]

(3.19)
3.2. Room Humidity Moisture Balance

The Transfer Number, \( B \), is a potential gradient of conserved properties derived by either conservation of mass or enthalpy. If the Reynolds Hypothesis conditions are met, either derivation method is valid and produce equivalent values of the Transfer Number. Setting up a control volume around the mass transfer region, (Figure 3.4), mass conservation produces:

\[
(m_{H_2O,G} \ast \Upsilon) + (m_{H_2O,S} \ast \dot{m}''') = (m_{H_2O,S} \ast \Upsilon) + (\dot{m}'' \ast 1) \tag{3.20}
\]

where: \( m_{H_2O,G} \ast \Upsilon \) is the influx of water vapour across the gaseous region, \( G \). \( m_{H_2O,S} \ast \dot{m}''' \) is the influx of condensate on the film surface, \( S \). \( m_{H_2O,G} \ast \Upsilon \) is the efflux of water vapour across the gaseous region, \( G \). \( \dot{m}'' \ast 1 \) is the bulk flow efflux of condensate onto the film surface, \( S \). \( m_{H_2O} \) is the mass fraction of water vapour.

Rearranging gives:

\[
\frac{\dot{m}''}{\Upsilon} = \frac{m_{H_2O,G} - m_{H_2O,S}}{m_{H_2O,S} - 1} \tag{3.21}
\]
Therefore from equation 3.17 and 3.21, the Transfer Number is given by:

\[
B = \frac{\dot{m}''}{\dot{m}} = \frac{m_{H_2O,G} - m_{H_2O,S}}{m_{H_2O,S} - 1}
\]  
(3.22)

For evaporation processes \(m_{H_2O,S} < m_{H_2O,G}\), hence \(B\) is positive. For condensation processes \(m_{H_2O,G} > m_{H_2O,S}\), therefore \(B\) is negative. The mass fraction of water vapour for either the G or S states is given by:

\[
m_{H_2O,G/S} = \frac{g_{G/S}}{1 + g_{G/S}}
\]  
(3.23)

where:

\[
g_G = \frac{0.622 p_{svpG}}{P_{atm} - p_{svpG}}
\]  
(3.24)

\[
g_S = \frac{0.622 p_{svpG} \phi}{P_{atm} - p_{svpG} \phi}
\]  
(3.25)

\(g_{G/S}\) is the moisture content \(kg/kg\) in the gaseous and surface states respectively. \(P_{atm}\) is atmospheric pressure, \(Pa\). \(p_{svpG/S}\) is the saturated water vapour pressure in the surface and gaseous states. \(\phi\) is the relative humidity of the room air.

**Dew Point**

If the temperature of a surface in a zone falls beneath the dew point or saturation temperature of the air in the zone, condensation will form upon its surface. The latent heat transfer to the chilled beam only occurs when the chilled beam surface is at or below the saturation temperature. Below the dew point temperature the sensible heat ratio correction to the air side resistance has to be implemented in the simulation, above the dew point temperature no correction is necessary. Dew point temperature is found by [70]:

\[\text{Dew Point} = \ldots\]
3.2. Room Humidity Moisture Balance

\[
T_{\text{dew}} = \frac{237.3 \log_{10} p_{\text{vap}} - 661.1}{10.29 - \log_{10} p_{\text{vap}}} \tag{3.26}
\]

where:

\[
p_{\text{vap}} = \frac{\phi \cdot p_{\text{sv}}}{100} \tag{3.27}
\]

and:

\[
p_{\text{sv}} = 610.5 e^{0.024 T} \tag{3.28}
\]

\(\phi\) and \(T\) are the room relative humidity and air temperature, °C respectively.

3.2.4 Moisture Generation

Moisture generation occurs as a result of occupants and certain equipment. Generation is independent of the indoor air moisture content and depends only upon occupancy and equipment levels and the rate of production. Discretized diurnal occupancy profiles have been used in conjunction with moisture generation rates for sedentary occupants to simulate the change in room air moisture content, which is found by:

\[
\sum_{i=1}^{n_g} \frac{m_{g,i,t}}{\rho_a V} \tag{3.29}
\]

\(m_{g,i}\) is the moisture generation rate, kgs⁻¹, for each generation source. Typical moisture generation rates for a human adult are given in British Standard code of practice for natural ventilation [69], and are reproduced in Table 3.1. Moisture generation rates for occupants includes both respiration moisture production and perspiration moisture production.
### 3.3 Moisture Balance

Taking all the moisture transport processes the differential moisture balance equation becomes:

\[
\Delta g_r = \sum_{i=1}^{A_i} \left[ \frac{A_m}{h_m} \rho_a V (g_{ms,i}^*(t) - g_r) + \frac{1}{\rho_a} \sum_{i=1}^{A_i} [\rho Q_{xi,i} (g_{oi,i}^* - g_r) - \dot{m}_{c,i}^* + \dot{m}_{e,i}^* + m_{g,i}] \right]
\]

(3.30)

Using fourth order runge kutta integration of the moisture balance differential equation, the next time step value of room air moisture content is found. To provide stability [54], the room air moisture content is then averaged over the time step and used as initial conditions for the next time step using:

\[
g_r = \frac{g_r^t + g_r^{t+1}}{2}
\]

(3.31)

### 3.4 Latent Model Validation

Validation of the latent model uses the testing procedure given by Wortman [47] and Wiltshire [48] which is described in Chapter 2. Again, the analytical validation method is the approach chosen which utilizes steady state testing, model initialization testing and known response testing.

Figure 3.5 shows the steady state profile of the enclosure moisture content. The steady state simulation was run with a constant external air moisture content of 0.01532 kg/kg. With a time step of 600 seconds the enclosure air moisture
3.4. Latent Model Validation

content reached steady state within 12 hours. The initialization tests, Figure 3.6 where run for 10 identical days with initial conditions of 40%, 50%, 60% and 80 % relative humidity. All tests produced identical diurnal profiles within 12 hours of simulation time. Figure 3.6 shows the diurnal profiles of moisture content for enclosure initial conditions of 50% and 60 % relative humidity and the difference between the predicted profiles.

Figures 3.7 and 3.8 show known response diurnal profiles of enclosure moisture content for a hot summer day. Figure 3.7 shows a comparison of absorption/desorption profiles between absorbing surfaces of different surface areas. The solid lines represent enclosure moisture content, material moisture content and enclosure moisture content change for an absorbing surface (pine) with a surface area of two square metres. The dashed lines represent a pine surface with a surface area of thirty square metres. The material with the greater surface area absorbs more moisture than the surface with less surface area, which can be seen in the diurnal profile of the material moisture contents. Also, the rate of absorption/desorption is higher for the material with the greater surface area. This can be seen on the profile of the enclosure moisture content change. The increased absorption/desorption
effect can also be seen on the diurnal profile of the enclosure moisture content where the peak at 3:00 hours is reduced, and at 10:00 hours where the peak is higher, corresponding to the peak occurrences of moisture change in the material.

Figure 3.8 shows the enclosure and a moisture absorbing materials (pine) response to 10 days of constant moisture generation in the enclosure. The enclosure moisture content is much higher than usual, (Figure 3.7), due to the generated moisture. The diurnal profile for the moisture content of the absorbing material shows saturation (of the moisture interaction depth) for the first ten hours, then desorption until 18:00 with re-absorption occurring until saturation. Moisture absorption/desorption rates are controlled by partial pressure gradients and are therefore a function of the relative humidity of the enclosure, the desorption period is coincidental with the afternoon temperature rise due to occupants, equipment and external conditions which lowers the relative humidity in the space, causing desorption.
3.4 Latent Model Validation

Figure 3.7: Expected Response Test: Absorbing Material Surface Area

Figure 3.8: Expected Response Test: Constant Internal Moisture Generation
3.5 Latent Model Implementation

One equation has been established that describes the relationship of the moisture transfer variables, the moisture balance equation, and is given as:

\[
\Delta g_r = \sum_{i=1} A_m \frac{h_{ms}}{h_{ms}} \rho_a V (g_{ms,i}(t) - g_r) + \frac{1}{\rho_a V} \sum_{i=1} \left[ \rho Q_{a,i} (g_{a,i} - g_r) - m_{c,i}'' + m_{e,i}'' + m_{g,i} \right]
\]  
(3.32)

A state space equation can be used to represent Equation 3.32

\[
\dot{g}_r = Mg_r + NP
\]  
(3.33)

where, \( P \) is the moisture disturbance vector consisting of all the driving variables considered in the model,

\[
P = f(g_{ms,i}(t), g_{a,i}, m_{c,i}''', m_{e,i}''', m_{g,i})
\]  
(3.34)

and \( M \) and \( N \) are matrices containing the known coefficients and constants. The analytical solution for Equation 3.33 has the same form as Equation 2.60, and is represented by:

\[
g_{(r),t} = \exp^{Mt} g_{(r),0} + \int_0^t \exp^{M(t-\varphi)} NP(\varphi) d\varphi
\]  
(3.35)

\( g_{(r),t} \) and \( g_{(r),0} \) are the room moisture contents at time \( t \) and for the initial state. As for the thermal treatment, the Runge Kutta method is used to solve Equation 3.35.
3.6 Latent and Thermal Model Integration

The latent model acts as a post-processor to the thermal model. Enclosure nodal temperature solutions are used as inputs for the evaluation of the following moisture transfer processes in the latent model:

**Air infiltration:** $T_a$ and the external temperature, $T_e$, provide the temperature potential for the air flow.

**Natural Ventilation:** $T_a$ and the external temperature, $T_e$, provide the temperature potential for the air flow.

**Absorption/desorption:** $T_a$ is used in the evaluation of the relative humidity which calculates the sorption isotherm, and, used in a regression equation, evaluates both the material moisture diffusivity, $\alpha_m$, and the material vapour diffusion coefficient, $D_v$.

**Condensation:** The condensation film temperature, a numerical mean of $T_a$ and the chilled beam surface temperature, $T_{cbo}$, is used in regression equations to evaluate the air/liquid film specific heat capacity, $C_p$, the chilled beam convective heat transfer coefficient, the air/film Prandtl number, kinematic viscosity and thermal conductivity.

**Evaporation:** The evaporation film temperature, a numerical mean of $T_a$ and the drip tray surface temperature, taken as being numerically equal to the wall temperature, is used in regression equations to evaluate the air/liquid film specific heat capacity, $C_p$, the chilled beam convective heat transfer coefficient, the air/film Prandtl number, kinematic viscosity and thermal conductivity.

3.7 Summary

This chapter developed and validated a linear differential equation which, using discrete time steps to enable the linearization of non-linear moisture transport processes, describes the interaction of the moisture transport phenomena and the resultant response of the indoor air humidity. The linearized moisture
transport properties considered within the latent model are: moisture absorption/desorption, chilled beam surface condensation, infiltration, (natural) ventilation, evaporation and generation. Steady state, model initialization and known response testing procedures have validated the moisture mass balance model. The latent model implementation has been discussed along with the integration of the latent model within the thermal model.
Chapter 4

Ventilation and Thermal Plant Models

Introduction

Modelling the performance and integration of a sub-dew point chilled beam within mixed mode strategies requires the simulation of mechanical plant response and energy usage of the chiller and HVAC plant. The thermal plant models must be easily interpreted by and integrated with the dynamic thermal building model and the moisture balance model to produce a composite "real" building model. Modelling the mechanical plant, chilled beam and associated chiller must enable the evaluation of the primary energy usage and the performance of the sub-dew point chilled beam in the different mixed mode strategies. The mechanical plant model must be able to respond to the thermal and latent disturbances and give interpretable evaluations of energy performance.

This chapter describes the models of natural ventilation, air infiltration and models of the thermal plant associated with the enclosure; a chiller plant model that describes the energy consumption of the chilled beam, and a model of a HVAC mechanical plant, plus the control strategies for the HVAC mechanical plant. Using a typical HVAC scheme, the mechanical plant simulation describe steady state numerical models of a mixing box, heating coil, cooling coil, steam humidifier, fan and associated ductwork. The mechanical plant models are solved using a subse-
4.1 Natural Ventilation

According to the British Standard [69], natural ventilation is the movement of air through openings in the building fabric, due to wind or to static pressure created by differences in temperature between the interior and exterior of the building, or to a combination of these acting together. Natural ventilation is dependant upon wind speed, wind direction and air temperature and it is these which dictate what role any opening has upon air flow into or out of a building. Natural ventilation is categorized into two mechanisms associated with the orientation of the contributing openings; single sided or cross flow ventilation.

Prediction of natural ventilation rates through openings due to wind and temperature differences is difficult, localized turbulence, internal obstructions, external geography, location and orientation contribute greatly to air flow patterns which dictate air flow rate. However the general characteristics of natural ventilation can be evaluated by utilizing a simple two dimensional representation. Figure 4.1, shows the arrangement of openings for cross flow ventilation, single sided ventilation is represented in Figure 4.2.

4.1.1 Cross-flow

Cross flow ventilation is driven by wind effects and temperature difference. Pressure differentials on the windward and leeward side of a building cause wind dependant cross flow ventilation. Static pressure differentials produced by temperature gradients cause temperature dependant cross flow ventilation. Wind and temperature effects however, occur simultaneously. When temperature gradients are low, cross flow ventilation will closely resemble that of wind driven only[69]. When temperature gradients are high, it is these which dominate the ventilation. These phenomena lead to a simple mathematical comparison to evaluate which mechanism dominates the ventilation, the largest air flow prediction becomes the
mechanism for the cross flow ventilation. With reference to Figure 4.1, wind only cross flow ventilation is given by\[69\]:

\[ Q_w = C_d A_w U_r (\Delta C_p)^{\frac{1}{2}} \]  

(4.1)

\[ \frac{1}{A_p^2} = \frac{1}{(A_1 + A_2)^2} + \frac{1}{(A_3 + A_4)^2} \]  

(4.2)

\( C_p \) is the applied differential mean pressure coefficient and has a range of values between 0.1 and 1.0. 1.0 is typical of an exposed building with no external obstructions, 0.1 is representative of a sheltered building, enclosed by other buildings or obstructions. \( U_r \) is the external wind velocity, m/s, taken from weather data. \( C_d \) is a coefficient of discharge for the opening. \( C_p \) and \( C_d \) are taken as 0.69 and 0.1 respectively. \( A_{1,2,3,4} \) is the window opening area, m\(^2\). Temperature only cross flow ventilation flow rate is given by\[69\]:
\[ Q_b = C_d A_b \left( \frac{2\Delta T g H_1}{T} \right)^{\frac{1}{2}} \] (4.3)

\[ \frac{1}{A_b^2} = \frac{1}{(A_1 + A_3)^2} + \frac{1}{(A_2 + A_4)^2} \] (4.4)

\( \Delta T \) is given by the difference between the interior and exterior temperature, \( H_1 \) is the vertical height between the openable windows, \( \bar{T} \) is the mean temperature of the external and internal temperatures. A direct numerical comparison can be used to determine the dominating flow rate for cross flow ventilation, the evaluation for the cross flow rate, \( Q_c \), is thus:

\[ Q_c = Q_b \]

For \[ \frac{U_r}{\sqrt{\Delta T}} < 0.26 \left( \frac{A_b}{A_w} \right)^{\frac{1}{2}} \left( \frac{H_1}{\Delta C_p} \right)^{\frac{1}{2}} \]

\[ Q_c = Q_w \]

For \[ \frac{U_r}{\sqrt{\Delta T}} > 0.26 \left( \frac{A_b}{A_w} \right)^{\frac{1}{2}} \left( \frac{H_1}{\Delta C_p} \right)^{\frac{1}{2}} \]

4.1.2 Single Sided

Single sided ventilation is also driven by wind and temperature gradients. The mechanisms for temperature driven ventilation are the same as for cross flow ventilation. Air flow exchange at a single opening due to wind effects occurs because of the localized turbulent nature of the air flow around the opening. Again direct numerical comparison dictates the dominant air flow rate. For wind only single sided ventilation the air flow rate is given as[69], Figure 4.2:

\[ Q_w = 0.025 A U_r \] (4.7)

For temperature only single sided ventilation the flow rate is given as[69], Figure 4.2:
4.1. Natural Ventilation

Figure 4.2: Single Sided Natural Ventilation

\[ Q_b = C_d A \left[ \frac{\epsilon \sqrt{2}}{(1 + \epsilon)(1 + \epsilon^2)^{\frac{1}{2}}} \right] \left( \frac{\Delta T g H_1}{T} \right)^{\frac{1}{2}} \]

\[ \epsilon = \frac{A_1}{A_2} \]

\[ A = A_1 + A_2 \]

(4.8)

For combined wind and temperature difference the ventilation flow rate is given by direct numerical comparison, the greater of the flow rates gives the value for the single sided ventilation rate.

4.1.3 Infiltration

Infiltration is the air movement across enclosure boundaries associated with unavoidable gaps in building construction. The rate of air movement for infiltration
is dependant upon pressure difference across the building caused by wind velocity and direction and the exterior/interior temperature difference. Most building construction aims to keep infiltration to a minimum. The effective leakage area for infiltration can be found for any building type \([28]\). \(Q_i\) for infiltration is taken as \([28]\):

\[
Q_i = L(A\Delta t + Bv^2)^{0.5}
\]  

(4.9)

where \(Q_i\) is the infiltration air flow rate, l/s. \(L\) is the effective leakage area taken for a one storey office as \(690 \text{ cm}^2\). \(A\) is the stack coefficient, \(0.000145 \text{ l/s}^2\text{cm}^{-4}\text{K}^{-1}\), for a one storey building. \(\Delta t\) is the average indoor/outdoor temperature difference for the time interval, Kelvin. \(B\) is the wind coefficient, with a value of \(0.000174 \text{ l/s}^2\text{cm}^{-4} \text{(m/s)}^{-2}\) for local wind shielding (Class 3). \(v\) is the average wind speed, m/s.

### 4.2 HVAC Model

The mechanical plant model simulates two ventilation systems; a conventional air conditioning system and a displacement ventilation system comprising a mixing box, heating coil, cooling coil, steam humidifier, fan and associated ductwork. The solution algorithm utilized is successive approximation. Established steady state models of HVAC plant are described, the dynamic response of the HVAC plant is not required for this project. Both ventilation systems are controlled using sequential control. The full air conditioning system utilises temperature and humidity control, whereas the displacement ventilation system only uses temperature control, ie has no humidifier. The schematic for the mechanical plant is shown in Figure 4.3.

#### 4.2.1 Mixing Box

The mixing box, using control signals, modulates the position of dampers in the recirculated and external supply air duct to change the fresh air fraction of the conditioned air. The mixing box damper position ensures that a minimum fresh air
fraction of 10% is maintained. The mixing box is described using mass, moisture and enthalpy conservation. Ten variables form the mixing box model, Figure 4.4; the external air conditions, temperature and moisture content, the mass flow rate in the recirculation, external, and supply duct and the control signal for damper position. Performing a mass balance upon the mixing box produces the following equation:

\[ \dot{m}_s = (1 - x)\dot{m}_r + (x)\dot{m}_o \]  

(4.10)

where \( \dot{m}_s \) is the supply air mass flow rate, \( \dot{m}_r \) is the return air mass flow rate and \( \dot{m}_o \) is the external air mass flow rate. \( x \) is the control signal for damper position. To maintain a 10% fresh air fraction the value of the control signal \( x \) is never less than 0.1. Performing an enthalpy and moisture balance upon the mixing box gives the following equations:

\[ \dot{m}_s t_s = (1 - x)\dot{m}_r t_r + (x)\dot{m}_o t_o \]  

(4.11)
4.2. HVAC Model

\[ m_s g_s = (1 - x)m_r g_r + (x)m_o g_o \]  
(4.12)

where \( t_s, t_r \) and \( t_o \) and \( g_s, g_r \) and \( g_o \) are the temperature and moisture content of the supply, recirculated and external air ducts.

4.2.2 Fan Model: Component Modelling Methodology

Component modelling methodology has been investigated and established for use in HVAC optimization [75], and is to be used to model fan performance. The component model format utilised had four parts; an energy model to assess the system's energy consumption; a cost model, for both first and maintenance cost; a performance model, which reproduces the operating characteristics, and a constraint model, which defines the dimensional, operating and configuration limits of the component. Component cost modelling is not considered in the performance testing of the chilled beam and HVAC plant. Normalised fan curves were produced to simulate a variety of fan diameters and speeds with similar geometries to re-
4.2. HVAC Model

roduce the amount of characteristic curves needed for solution. The non-dimensional normalized curve fit method was found to be 95% accurate with the original manufacturers data.

The fan component model comprises nine variables; size, fan blade angle and speed of the fan, the mass flow rate, total pressure, temperature and moisture content of the inlet and the resultant total pressure and temperature at the outlet, Figure 4.5.

Performance Model

The performance model calculates, for a given fan speed, mass flow rate and fan blade angle the air temperature and total pressure rise across the fan. Constant speed curves of normalised pressure versus normalised volume flow rate form a pressure-volume flow characteristic. Normalised coefficients for volume, $\phi$, and total pressure, $\psi$, are given by:
4.2. HVAC Model

\[ \phi = \frac{\dot{m}}{\rho d^2 d^3 n^4} \quad (4.13) \]

\[ \psi = \frac{2P_{tf}}{\rho (\pi dn)^2} \quad (4.14) \]

where \( \dot{m} \) is the mass flow rate of the air stream, kg/s, \( \rho \) is the density of the air stream, kg/m\(^3\). \( d \) and \( n \) are the diameter and the speed of the fan respectively. \( P_{tf} \) is the fan total pressure.

The actual pressure rise across the fan is found by taking the air mass flow rate at inlet and converting it to the normalised volume flow rate. Using the characteristic curves the normalised total pressure rise is found and this is converted to the pressure rise across the fan. The temperature rise across the fan is found from [75]:

\[ \Delta t = \frac{q}{\dot{m} C_p} \quad (4.15) \]

where \( q \) is the fan power, W, which if all the changes in kinetic energy are neglected will appear as a rise in air temperature across the fan.

Energy Model

The fan's energy consumption over the simulation period is represented by the energy model. The energy model is presented in normalised form and is given as:

\[ \lambda = \frac{8q}{\pi^4 d^6 n^3 \rho} \quad (4.16) \]

\( \lambda \) is the normalised fan absorbed power, \( q \) the fan absorbed power incorporates all energy losses associated with the drive components.
4.2. HVAC Model

Constraint Model

The constraint model places limits upon the normalised volume flow to ensure any solution lies within the modelled performance of the fan. Modelling outside the fan performance data is not recommended [75]. Firstly extrapolation beyond the performance data gives no assurance of accuracy, the fan is likely to stall, and secondly, the high order of the least-squares curves fit can lead to unreliable behaviour in extrapolated regions.

Simple boundary conditions are applied to the speed and blade angle. These inequality constraints dictate the maximum and minimum values of normalised volume flow as a function of the fan blade angle. Normalised volume flow is an independent variable within the performance model curve fit which ensures that the simulation will not extrapolate outside of the original data.

4.2.3 Humidifier

Humidification is simulated using an enthalpy balance model of a steam humidifier. The component model comprises seven variables, inlet air conditions, temperature and moisture content, mass flow rate of the inlet air, mass flow rate of the injected steam, the control signal and the outlet conditions, Figure 4.6.

During humidification there is an increase in both moisture content and temperature due to the latent heat addition. Assuming no pressure drop across the humidifier, and all the latent heat is converted into the air stream, then the component equations are given by:

\[ g_o = g_i + \frac{\dot{m}_s}{\dot{m}_a} \]  \hspace{1cm} (4.17)

\[ h_o = h_i + \frac{\dot{m}_s}{\dot{m}_a} h_g \]  \hspace{1cm} (4.18)

\( h_g \) is the enthalpy of dry saturated steam, 2675.8 kJ/kg. The second component equation for enthalphy can be rearranged for the inlet air conditions of temperature
4.2. HVAC Model

4.2.4 Duct Model

The duct model simulates the pressure loss characteristics within the duct air stream. Friction loss within a duct accounts for the loss of pressure along the duct system. The duct component model consists of six variables; the inlet air temperature, moisture content, and mass flow rate, the inlet pressure, the duct diameter and the resultant outlet pressure, Figure 4.7.

The inlet temperature and moisture content are used to calculate the specific volume of the air stream. Between any two points in a duct system the outlet pressure, \( p_{o} \), is given by:

\[
\frac{[1.007t_{i} - 0.026 + g_{i}(2501.0 + 1.84t_{i})] + \frac{\dot{m}_{a}h_{g}}{m_{a}} - (2501.0g_{o} - 0.026)}{1.007 + 1.84g_{o}} = t_{o} \tag{4.19}
\]

4.2.4 Duct Model

The duct model simulates the pressure loss characteristics within the duct air stream. Friction loss within a duct accounts for the loss of pressure along the duct system. The duct component model consists of six variables; the inlet air temperature, moisture content, and mass flow rate, the inlet pressure, the duct diameter and the resultant outlet pressure, Figure 4.7.

The inlet temperature and moisture content are used to calculate the specific volume of the air stream. Between any two points in a duct system the outlet pressure, \( p_{o} \), is given by:
4.2. HVAC Model

where \( \Delta p \), the pressure change in the duct due to friction losses, is given by:

\[
\Delta p = \frac{1}{2} v k \dot{m}_v^2
\]  

\( v \) is the specific volume of the air stream \( m^3/kg \), \( k \) is a friction factor and \( \dot{m}_v \) is the mass velocity, a quotient of the mass flow rate and the face area of the duct.

4.2.5 Heating Coil

The heating and cooling coil utilize the effectiveness-number of transfer units method, \( \varepsilon - NTU \) [29], to evaluate temperature change. Nine parameters form the component model of the heating coil; inlet temperature and moisture content,
4.2. HVAC Model

Outlet temperature

Outlet temperature, $t_{oa}$, is found simply from:

$$t_{oa} = t_{ai} + \frac{Q_{th}}{m_a C_{pa}} \tag{4.22}$$

$Q_{th}$, the total duty of the heating coil, $W$, is found from the following equation:

$$Q_{th} = \varepsilon m_a C_{pa} (t_{wi} - t_{ai}) \tag{4.23}$$

$\varepsilon$, the effectiveness of the heating coil, is given as the ratio of the actual heat
4.2. HVAC Model

The air pressure fall across the heating coil is dependant upon the coil geometry and the face velocity of the air stream. The outlet pressure, $p_{to}$, is given as:

$$p_{to} = p_{ti} - \Delta p$$  \hspace{1cm} (4.27)

$\Delta p$, the pressure loss across the coil is given as:

$$\Delta p = kv_f^{1.7}$$  \hspace{1cm} (4.28)
4.2. HVAC Model

4.2.6 Cooling Coil

The cooling coil component model comprises three typical modes of operation; dry, partially wet and wet. The component model contains nine variables; inlet temperature, moisture content, pressure and mass flow rate, cooling medium control signal, the physical dimensions of the coil, and outlet conditions; temperature, moisture content and pressure, Figure 4.9.

Inlet and outlet conditions are used to determine the enthalpy, these when compared to the enthalpy at dew point of the air stream dictate the mode of operation of the coil. An enthalpy above the dew point at the outlet of the coil ensures dry operation. An enthalpy below the dew point at inlet determines wet operation, an enthalpy between these extremes gives partially wet operation.
4.2. HVAC Model

Dry Coil Operation

Dry coil operation, as with the heating coil, uses the \( \varepsilon - NTU \) method for calculating the total duty, \( Q_{tc} \), and the outlet temperature, \( t_{ao} \):

\[
Q_{tc} = \varepsilon m_a C_{pa} (t_{ai} - t_{wi}) \quad (4.29)
\]

\[
t_{ao} = t_{ai} - \frac{Q_{tc}}{m_a C_{pa}} \quad (4.30)
\]

Wet Coil Operation

Wet coil operation uses a logarithmic mean enthalpy to calculate the total duty of the coil, this, in combination with the bypass factor and apparatus dew point temperature and moisture content, is used to find the outlet state variables [75]. Wet coil total duty, \( Q_{tcw} \), is found from:

\[
Q_{tcw} = \frac{A_t h_m}{C_{pa} R_{wet}} \quad (4.31)
\]

where \( R_{wet} \) is the wet surface resistance, \( m_2 K/W \), and \( h_m \) is the logarithmic mean enthalpy given by:

\[
h_m = \frac{(h_{ai} - h_{si}) - (h_{ao} - h_{so})}{\log \left( \frac{h_{ai} - h_{si}}{h_{ao} - h_{so}} \right)} \quad (4.32)
\]

\( h_{so} \) and \( h_{si} \) are coil surface enthalpies, outlet and inlet respectively. \( h_{ao} \) and \( h_{ai} \) are outlet and inlet air stream enthalpies, \( kJ/kg \). The wet coil total duty is then used to find the apparatus dew point enthalpy, \( h_{adp} \):

\[
h_{adp} = h_{ai} - \frac{Q_{tcw}}{m_a (1 - B_f)} \quad (4.33)
\]
4.2. HVAC Model

$B_f$ is the coil bypass factor, dimensionless. The apparatus dew point enthalpy is used to calculate the apparatus dew point temperature and moisture content. Outlet state conditions of temperature and moisture content are then found from:

\[ t_{ao} = t_{adp} + (t_{ai} - t_{adp}) B_f \]  
\[ g_{ao} = g_{adp} + (g_{ai} - g_{adp}) B_f \]  

$t_{adp}$ and $g_{adp}$ are, respectively, the apparatus dew point temperature and moisture content. The bypass factor, a measure of the effectiveness of the cooling coil, is found from:

\[ B_f = e^{\frac{-A_f}{\epsilon \rho_{a} h_{dry} m_a}} \]  

**Partially Wet Coil Operation**

Partially wet coil operation evaluates the duty from both wet and dry surfaces to find the total duty. A numerical check between the calculated wet and dry areas and the total surface area is performed [75]. Duty from the dry surface, $Q_{dry}$, and the wet surface, $Q_{wet}$, is given by:

\[ Q_{dry} = m_a (h_{ai} - h_{bound}) \]  
\[ Q_{wet} = m_a (h_{bound} - h_{ao}) \]  

$h_{bound}$ is the enthalpy at the boundary between the wet and dry surfaces. A function of the characteristics of the coil, $h_{bound}$ is given by:

\[ h_{bound} = \frac{[t_{adp} - t_{wo} + y_{a} h_{ai} + \omega h_{sbound}]}{y_{ao} \omega} \]
where:

\[ y_a = \frac{m_a}{m_w C_{pw}} \]  
\[ \psi = \frac{R_w}{C_{pa} R_{wet}} \]

\( h_{\text{bound}} \) is the surface enthalpy at the partially wet boundary, \( m_w \) is the mass flow rate of the cooling water, kg/s, \( R_{\text{wet}} \) and \( R_w \) are the thermal resistances of the wet surface and the cooling medium respectively. To establish the accuracy of the partially wet total duty the surface areas of wet and dry operation are summed and compared to the physical total surface area of the coil. Wet surface area, \( A_{\text{wet}} \), and dry surface area, \( A_{\text{dry}} \), are evaluated thus:

\[ A_{\text{wet}} = \frac{C_{pa} Q_{\text{wet}} R_{\text{wet}}}{h_{\text{mean}}} \]  
\[ A_{\text{dry}} = \frac{Q_{\text{dry}} (R_{\text{dry}} + R_w)}{t_{\text{mean}}} \]

\( h_{\text{mean}} \) and \( t_{\text{mean}} \) are the logarithmic mean enthalpy and temperature respectively. \( h_{\text{mean}} \) is evaluated for the wet surface area to include the latent heat flux. \( t_{\text{mean}} \), purely sensible heat flux, is evaluated for the dry surface area.

**Outlet Pressure**

The outlet pressure after the cooling coil is found using the same method as for the heating coil, with the exception that the pressure drop, \( \Delta p \), is a function of the ratio of the wet coil surface area, \( A_{\text{wet}} \), to total coil surface area, \( A_t \):

\[ \Delta p = k v_f^{1.7} \left[ 1.0 + \frac{A_{\text{wet}}}{4A_t} \right] \]
4.2. HVAC Model

4.2.7 Control Procedures

The HVAC system has a sequential control strategy. Fourteen variables and parameters form the full air conditioning control strategy, Figure 4.10. Zone temperature and relative humidity are the state input variables. Supply air temperature, moisture content and mass flow rate form the output state variables. Damper position in the mixing box, (FAF) heat exchange fluid mass flow rate in the heating and cooling coil, \( Q_h \) and \( Q_c \) and hence the coil duties, and the steam mass flow rate in the humidifier \( m_s \) are the controlled variables. The control strategy operates a four degree Kelvin dead band around the room set-point, and by assuming linear relationships, neglects valve opening (and closing) characteristics.

The full air conditioning sequential control is split into temperature and humidity control, with displacement ventilation controlled only using temperature. The mixing box, heating and cooling coil are temperature controlled. The cooling coil and the humidifier are humidity controlled. The highest demand from temperature or humidity control dictates the cooling coil output. The sequential control algorithms are summarised below, and shown in graphical form in Figures 4.11,
4.2. HVAC Model

The displacement ventilation system set points are the supply air temperature and the supply air mass flow rate. Typical displacement systems supply air at 18 °C ±1°C, supply mass flow rates are low, usually at or above the minimum fresh air requirement, allowing buoyancy driven flows to occur around heat source in the zone. The full air conditioning system takes the enclosure air conditions as set points, supplying air at conditions driven by the heat disturbances in the enclosure. The control algorithms presented below are dependant upon the controlling temperature, \( t_r \), the supply temperature for displacement ventilation and room temperature for full air conditioning, and the controlling air relative humidity \( RH_r \), the room relative humidity, (for full air conditioning only).

**Heating**

The control algorithms for heating fix the mass flow rate of the heating medium hence the heater duty, \( Q_h \):
4.2. HVAC Model

Figure 4.12: Temperature Control Strategy

Figure 4.13: Humidity Control Strategy
4.2. HVAC Model

\[ If \left( t_r \leq \left( SP_h - \frac{PB_h}{2} \right) \right) Q_h = Q_{h_{max}} \]

else if \( \left( t_r < (SP_h + \frac{PB_h}{2}) \right) \) and \( t_r > (SP_c - \frac{PB_c}{2}) \)

\[ Q_h = Q_{h_{max}} \left( \frac{Q_{h_{max}}}{PB_h} \right) \left( SP_h + \frac{PB_h}{2} - t_r \right) \]

\[ else Q_h = 0 \] \hspace{1cm} (4.45)

\( SP_h \) is the set-point for the heating coil, \( PB_h \) is the proportional band, (for heating), around the set-point.

Mixing Box

The control algorithms for the mixing box alter the position of dampers to modulate the amount of fresh air entering the zone. The minimum amount of fresh air is maintained at 10 %. The fresh air fraction, \( FAF \), refers to the percentage opening of the external air supply duct damper.

\[ If \left( t_r \leq \left( SP_h + \frac{PB_h}{2} \right) \right) FAF = 0.1 \]

else if \( \left( t_r > (SP_h + \frac{PB_h}{2}) \right) \) and \( t_r < (SP_c - \frac{PB_c}{2}) \)

\[ FAF = \text{linear increase across dead band.} \]

else if \( (h_0 < h_r) \) \( FAF = 1.0 \)
4.2. HVAC Model

\[ \text{else } FAF = 0.1 \quad (4.46) \]

\( SP_c \) is the temperature set-point for the cooling coil, \( PB_c \) is the proportional band, (for cooling), around the set-point.

Cooling

The control algorithms for cooling fix the mass flow rate of the cooling medium hence the cooling coil duty, \( Q_c \).

\[ \text{If } \left( t_r \leq \left( SP_c - \frac{PB_c}{2} \right) \right) \quad Q_c = 0 \]

\[ \text{else if } \left( t_r > \left( SP_c - \frac{PB_c}{2} \right) \text{ and } t_r < \left( SP_c + \frac{PB_c}{2} \right) \right) \]

\[ Q_c = Q_{cmax} \left( \frac{Q_{cmax}}{PB_c} \right) \left( t_r - \left( SP_c - \frac{PB_c}{2} \right) \right) \]

\[ \text{else } Q_c = Q_{cmax} \quad (4.47) \]

Humidification

Humidification control algorithms fix the mass flow rate of injected steam into the air stream.

\[ \text{If } \left( RH_r \leq \left( SP_s - \frac{PB_s}{2} \right) \right) \quad m_s = m_{smax} \]

\[ \text{else if } \left( RH_r < \left( SP_s + \frac{PB_s}{2} \right) \text{ and } RH_r > \left( SP_s - \frac{PB_s}{2} \right) \right) \]
4.2. HVAC Model

\[ m_s = m_{smax} \times \left( 1.0 - \frac{(RH_s - (SP_s - \frac{PB_s}{2}))}{(SP_s + \frac{PB_s}{2} - (SP_s - \frac{PB_s}{2}))} \right) \]

else \( m_s = 0.0 \)  (4.48)

\( SP_s \) is the relative humidity set-point for the humidifier, \( PB_s \) is the proportional band, (for humidification), around the set-point.

Dehumidification

Using room relative humidity as the controlling variable, dehumidification is achieved using the cooling coil. The control algorithm fixes the mass flow rate of the cooling medium.

\[ If \left( RH_r \leq \left( SP_d - \frac{PB_d}{2} \right) \right) \quad Q_c = 0.0 \]

else if \( \left( RH_r < \left( SP_d + \frac{PB_d}{2} \right) \text{ and } RH_r > \left( SP_d - \frac{PB_d}{2} \right) \right) \)

\[ Q_c = Q_{cmax} \times \frac{(RH_r - (SP_d - \frac{PB_d}{2}))}{((SP_d + \frac{PB_d}{2} - (SP_d - \frac{PB_d}{2}))} \]

else \( Q_c = Q_{cmax} \)  (4.49)

4.2.8 Simulation Technique

A successive substitution algorithm is implemented to solve the HVAC component model. Subsequent substitution links the individual component models. The output from one component model is used as input for the next component model.
in the HVAC system. The state input variables for the HVAC model are zone air temperature and moisture content, the state output variables are supply air temperature, moisture content and mass flow rate.

The successive approximation algorithm operates by using two parameters to evaluate the current guess of the search variable. One parameter is the desired output of the simulation, the other the calculated output evaluated using the current guess. A comparison is performed upon the two parameters and if they fall within an acceptable error limit the simulation ends. If the parameters lie outside of an acceptable error the search parameter is changed to improve the approximation. An essential pre-requisite of a successive approximation algorithm is a knowledge of the effect of a change in the search parameter.

The HVAC system model uses air mass flow rate as the search parameter. The output parameter is the system outlet pressure. The desired output pressure is 0.0Pa. The first approximation of air mass flow rate is taken as the halfway point of the limits of the chosen fan performance data. The outlet pressure is then calculated and compared to the desired value. If the calculated outlet pressure is too high, fan pressure is too high which can be reduced by increasing the mass flow rate. If the calculated outlet pressure is too low the reverse applies.

To augment the necessary change to the search parameter a direction vector is employed. The direction vector sets the direction of the search parameter away from the initial guess. The next approximation is set by the desired direction and the step size. Step size, an even number, is reduced by half for each iteration until the error check is satisfied. If the HVAC system is not properly sized the successive approximation algorithm can fail as the desired output can lay outside of the boundary conditions dictated by the performance data. The successive approximation algorithm is reproduced in flow chart form, Figure 4.14 [75].

The iteration on the air mass flow rate solves the fan-duct equations. Each time the coil model is called, for each air flow rate iteration, there is an internal iteration in the cooling coil to solve the coil heat and mass transfer equations. The iteration variable here is the outlet water temperature. Therefore there are two main iteration loops, one on the air mass flow rate to solve the fan-duct equations, and one on the coil water outlet temperature to solve the coil heat and mass transfer equations. No iteration is required for the humidifier, it can be solved in sequence with the other components. The HVAC plant has no control loop
iteration. Current room conditions simply set the plant output conditions based on a "reset schedule".

Figure 4.14: Successive Approximation Flow Chart
4.2. HVAC Model

4.2.9 HVAC Model Validation

The HVAC model was validated using the strategy outlined by Wiltshire [48] and Wortman [47], reproduced in Chapter 2, since the component models of the HVAC mechanical plant are well established [32], the validation of the HVAC plant will concentrate on testing control algorithm response to various external and internal conditions. The displacement ventilation system is tested; the control set points are the supply air temperature, (18°C), and the supply mass flow rate, (0.62 kg/s), equivalent to 12 l/s per person. The displacement ventilation system has no humidity control, control centres on the mixing box, the heating and the cooling coil.

Figure 4.15 shows the supply temperature and air mass flow rate for a transitional day. The dotted lines show the set points for the system, it is clear that the control system provides the correct response, providing the supply set points with a tolerance of ±2 %. This response is mirrored across all the test days, average, hot and high diurnal. The variation in air mass flow rate is due to the changes in air density throughout the day. Figure 4.16 shows a psychrometric representation of the control response of the displacement system and a bar chart showing the control signals of the mixing box, cooling and heating coil for an instantaneous response in transitional day conditions. The dash-dot line represents the mixing box, the solid line represents the cooling and dehumidifying in the cooling coil and the subsequent reheating in the heating coil and the heat gain from the fan. Control response was tested for; summer, hot summer, transitional and winter external conditions with low, medium, high and very high internal loading, all conditions tested produced expected results supplying the set points with a tolerance of ±2 %.

4.2.10 Summary

Established steady state models of HVAC mechanical plant components have been described and, using an expected response technique, validated. Two mechanical systems have been described, displacement and full air conditioning systems, each system comprising a mixing box, heating coil, cooling coil, fan and associated ductwork, with the full air conditioning system also comprising a steam humidifier.
4.2. HVAC Model

Figure 4.15: Expected Response: HVAC Set Point Test

Figure 4.16: Expected Response: HVAC Psychrometric and Control Response
Control strategies have been discussed and the solution algorithm described. The control algorithms were shown to provide the correct control measures and keep the controlled variables within a tolerance of ±2 %.

4.3 Chiller Model

In order to predict the energy performance of the sub-dew point chilled beam the associated plant, a chiller unit, has to be simulated within the model. The chiller plant is simulated as a black box model, performance equations are produced from manufacturers data.

4.3.1 Plant Model

The plant model consists of a chiller, a motorised three port diverting valve, a proportional controller, a pump and the sub-dew point chilled beam, Figure 4.17. The controller operates the diverting valve using a comparison of the room ambient air temperature and a user defined set point. The diverting valve would be on full divert, closed, when no cooling is required and open when cooling is required, controlled by the proportional controller.

The chiller model is a curve fit polynomial regression of manufacturers data, based upon the temperature difference between ambient air and the chilled water supply, and a ratio of the actual load to the design load conditions[80]. The general equation for chiller performance, the coefficient of performance, is represented below:

\[
COP = P_0 + P_1 \Delta T + P_2 \Delta T^2 + P_3 \text{PLR} \\
+ P_4 \text{PLR}\Delta T + P_5 \text{PLR}\Delta T^2 + P_6 \text{PLR}^2 \\
+ P_7 \text{PLR}^2 \Delta T + P_8 \text{PLR}^2 \Delta T^2
\]  

(4.50)

where COP is the coefficient of performance for the chiller, \(\Delta T\) is the temperature difference between the ambient air and the chilled water supply, and PLR is the
part load ratio, the ratio of the part load conditions to the design load value. The coefficients $P_0 - P_8$ are part of the general polynomial regression equation and are taken from manufacturers data[32].

### 4.3.2 Integration

Integrated into the building thermal, the chiller COP is found by direct input of the room ambient air temperature produced at the end of each time step. Using the COP the chiller compressor energy usage is evaluated. From this the primary energy consumption and efficiencies can be calculated. The heat transfer along the beam can be found from:

$$Q_{beam} = \dot{m} \, s \, C_p \, (T_{wout} - T_{win})$$

(4.51)

$s$ is the fraction of chilled water flow through the chilled beam dictated by the diverting valve position, $\dot{m}$ is the mass flow rate of the chilled water, $C_p$, the
specific heat capacity of water, $T_{win}$ is the inlet temperature and $T_{wout}$ the outlet temperature of the chilled water. There are two unknowns in this equation, the overall heat transfer of the beam and the outlet temperature, however from the chilled beam model we know:

$$Q_{total} = \frac{T_{ra} - T_{aw}}{R_{cor}}$$

(4.52)

$T_{ra}$ being the rad-air temperature, $T_{aw}$ the average chilled water temperature found from logarithmic treatment of the heat transfer along the beam, and $R_{cor}$ the corrected resistance evaluated to include the latent heat transfer effect of condensate formation. Assuming no heat losses, combining the equations gives:

$$T_{wout} = T_{win} + \frac{T_{ra} - T_{aw}}{R_{cor} \rho C_p}$$

(4.53)

The outlet chilled water temperature is essential for the calculation of the evaporator energy consumption, which can be found from:

$$Q_{evap} = \dot{m} C_p (T_{ret} - T_{flow})$$

(4.54)

$T_{ret}$, the return chilled water temperature to the evaporator is not always the same as $T_{wout}$, as when the diverting valve is half open, for example, half the chilled water flow will be off the chilled beam and half from the divert. This mixing will, through heat transfer, change the flow temperature. To evaluate the effect on temperature of the mixing the following equation, an energy balance on the mixing streams, will be utilised:

$$T_{ret} = \frac{T_{div} \dot{m}_{div} + T_{wout} \dot{m}_{wout}}{\dot{m}_{wout} + \dot{m}_{div}}$$

(4.55)

$\dot{m}_{div}$ is the mass flow rate of the diverted flow, $\dot{m}_{wout}$ is the mass flow rate of the flow across the beam. The fraction of flow through the divert is controlled by the valve position which is dictated by the proportional controller. The set point, upper limit and lower limit of the controller are user defined. Having evaluated $T_{ret}$ the compressor energy consumption can be found:
4.4 Model Integration

The HVAC plant takes as input the enclosure air temperature and humidity, these conditions are used as the set-points and return air conditions for the full air conditioning system and as the return air conditions for the displacement ventilation system. The displacement ventilation control system uses the supply air conditions as set-points. The supply air temperature and mass flow rate are used to calculate the convective heat input to the thermal network description of the zone, \( Q_c \), for each of the mixed mode ventilation strategies, natural ventilation, forced fan ventilation, displacement and full air conditioning system:

\[
Q_c = m_i C_p (T_{\text{supply}} - T_a)
\]  

(4.57)

where, \( T_{\text{supply}} \) represents the supply temperature of the ventilation strategy, \( T_a \) is the enclosure air temperature, \( m_i \) is the supply air mass flow rate for the ventilation strategy, \( C_p \) is the specific heat capacity of air.

4.5 Summary

Two thermal plant models have been described, a steady state HVAC component model, and a black box regression equation treatment of a chiller. Using estab-
lished steady state HVAC component models two mechanical plant systems have been simulated, displacement ventilation and a full air conditioning system. HVAC system control algorithms have been validated for use, producing expected control responses to within a ±2 % tolerance. An established black box chiller model has been described and its integration with the sub-dew point chilled beam outlined.

A steady state natural ventilation model and air infiltration model have been described. The natural ventilation model uses temperature and pressure driven potentials to evaluate air flow rate and uses a simple numerical comparison to determine the predominant driving force. Integration of the steady state model with the dynamic thermal model has been discussed.
Chapter 5

Performance Analysis
Methodology

Introduction

This chapter describes the composite model, (the integration of the dynamic zone thermal model, the steady state latent model, the thermal plant models and the ventilation models), its implementation and the testing methodology. The optimization routine, the Complex method is described along with the comfort performance indicator, (the enclosure comfort performance).

5.1 Model Integration: Composite Model

The composite model simulates the enclosure response to external and internal sensible and latent heat disturbances. The composite model incorporates, lumped parameter treatment of the enclosure (Chapter 2) and the sub-dew point chilled beam (Chapter 2), discretized linear differential equation treatment of the moisture response of the enclosure (Chapter 3), and steady state models of mixed mode ventilation strategies (Chapter 4).

Post-processor models of enclosure thermal comfort and chiller plant are used to
5.2. Composite Model Implementation

investigate the thermal comfort and energy performance. The lumped parameter treatment of enclosure thermal response forms the basis for the composite model. Moisture response, thermal plant performance and the sub-dew point thermal network are integrated with the enclosure thermal network at appropriate nodal points.

The post-processed enclosure thermal comfort model provides the objective function for the optimization routine. The optimization strategy is as such; that the enclosure thermal comfort is maximized during occupied periods for the given search parameters. Optimized strategies for natural ventilation take window opening area as the search parameter, forced fan ventilation take the fan mass flow rate and sub-dew point chilled beam operation, the valve stem position. Control algorithms dictate the mechanical plant response. Where simulation of the chilled beam performance is to be evaluated, optimized profiles of the ventilation strategy are used as input.

5.2 Composite Model Implementation

The simulation program of the enclosure dynamic response has been written in three parts, the pre-processor, the simulator and the post-processor. The pre-processor reads the input files which contain the weather data and time invariant parameters associated with the simulation. The simulator runs the simulation according to the pre-processed information and the mixed mode operational strategy. The output from the simulator predicts the mass, surface and air nodal temperatures and enclosure moisture content. The post-processor evaluates the enclosure comfort performance as the objective function for the optimization routine, the Complex method.

5.3 Model Parameters

The composite model simulation has been written to evaluate the integration of building form and fabric with a sub-dew point chilled beam in optimized mixed mode strategies. To augment this, the computational implementation requires
5.4. Simulation Zone and Boundary Conditions

that account is taken of the following:

Building Fabric: investigate the dynamic response of light, medium and heavy thermal “weights”;

Seasonal Performance: investigate the performance of the sub-dew point chilled beam for an average summer day, a high diurnal temperature swing day, a hot summer day and a transitional day;

Mixed Mode Strategy: investigate chilled beam performance with natural ventilation, forced fan ventilation and displacement ventilation.

Tables 5.1, 5.2, and 5.3 show the light, medium and heavy weight constructions, used as model parameters in this study.

<table>
<thead>
<tr>
<th>Layer Material</th>
<th>Thickness (mm)</th>
<th>( \rho ) Kg/m³</th>
<th>( C_p ) kJ/kg.K</th>
<th>( k ) W/m.K</th>
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Table 5.1: Fabric Construction Types - Lightweight

5.4 Simulation Zone and Boundary Conditions

A typical office is taken as the simulated zone. The room has dimensions 15 * 10 * 3 m, with one external wall 15 * 10 and three internal partition walls. The wall constructions for light, medium and heavy weight thermal mass are shown in tables 5.1, 5.2, and 5.3. The external wall is considered to have a single glazed
5.4. Simulation Zone and Boundary Conditions

<table>
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<tr>
<th>Layer Material</th>
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<th>(\rho) (Kg/m^3)</th>
<th>(C_p) (kJ/kg.K)</th>
<th>(k) (W/m.K)</th>
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<td>0.9</td>
<td>1.73</td>
</tr>
<tr>
<td>screed</td>
<td>70</td>
<td>1920</td>
<td>0.88</td>
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</tr>
<tr>
<td>vinyl tiles</td>
<td>6</td>
<td>800</td>
<td>1.25</td>
<td>0.8</td>
</tr>
<tr>
<td>CEILING: CONCRETE SLAB INSULATED</td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>stone chippings</td>
<td>13</td>
<td>881</td>
<td>1.67</td>
<td>1.436</td>
</tr>
<tr>
<td>felt and membrane</td>
<td>10</td>
<td>1121</td>
<td>1.67</td>
<td>0.19</td>
</tr>
<tr>
<td>insulation</td>
<td>50</td>
<td>40</td>
<td>0.92</td>
<td>0.025</td>
</tr>
<tr>
<td>cast concrete</td>
<td>150</td>
<td>2300</td>
<td>0.9</td>
<td>1.73</td>
</tr>
</tbody>
</table>

Table 5.2: Fabric Construction Types - Mediumweight

<table>
<thead>
<tr>
<th>Layer Material</th>
<th>Thickness (mm)</th>
<th>(\rho) (Kg/m^3)</th>
<th>(C_p) (kJ/kg.K)</th>
<th>(k) (W/m.K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>EXTERIOR WALL: HEAVYWEIGHT BLOCKWORK AND CAVITY INSULATION</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>facing brick</td>
<td>100</td>
<td>1600</td>
<td>0.79</td>
<td>0.84</td>
</tr>
<tr>
<td>air gap</td>
<td>100</td>
<td>1.3</td>
<td>1.005</td>
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</tr>
<tr>
<td>insulation</td>
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<td>32</td>
<td>0.71</td>
<td>0.04</td>
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<tr>
<td>concrete block</td>
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<td>2100</td>
<td>0.92</td>
<td>1.63</td>
</tr>
<tr>
<td>plaster</td>
<td>15</td>
<td>720</td>
<td>0.84</td>
<td>0.16</td>
</tr>
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<td>PARTITION WALL: BLOCKWORK</td>
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<td></td>
</tr>
<tr>
<td>plaster</td>
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<td>720</td>
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<tr>
<td>concrete block</td>
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<td>2100</td>
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<td>plaster</td>
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<td>720</td>
<td>0.84</td>
<td>0.16</td>
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<td>FLOOR: IN-SITU CONCRETE SLAB AND FINISH</td>
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<td>2300</td>
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<td>screed</td>
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<tr>
<td>CEILING: CONCRETE SLAB INSULATED</td>
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<td>150</td>
<td>2300</td>
<td>0.9</td>
<td>1.73</td>
</tr>
</tbody>
</table>

Table 5.3: Fabric Construction Types - Heavyweight

window of dimensions 2 * 2 m. The office is considered to have an occupancy level of one person per 3.7 m². The office is to be fully occupied between the hours of 8:00 and 18:00. To establish equipment gains in the space each occupant is to have a personal computer and the lighting heat gain is taken as 15 W/m².

The chilled beam is simulated as a cylindrical aluminium beam with an external diameter of 0.5 m. The chilled beams design includes the capture of condensate in a drip tray, condensate being used as a source of nutrition for vegetation, and as such is to be fixed to a wall at high level. The chilled beam is considered to
be fixed to all four walls in the zone. The chilled beam is approximated as a cylinder at high level upon the wall for simulation of the convective heat transfer and as a flat plate at high level for radiative heat transfer. The flat plate approach simulates the aspect of the chilled beam 'seen' by the other radiative surfaces in the space. The chilled beam surface is taken to be the only condensing surface, (neglecting glazing), and the drip tray is taken to be the only evaporating surface.

Absorbing and desorbing surfaces in the simulated zone are taken to be pine, spruce, and carpet. This is because material moisture data for office type materials is limited. Pine surfaces represent the approximate amount of desks in the office, based upon the number of occupants and a typical desk size for a personal computer workstation. Spruce surfaces represent an approximate number of cabinets and similar office storage units. The amount of carpet is taken as the floor area. For the modelling exercise the pine, spruce and carpet absorbing surfaces are approximated as boxes appropriate to the actual physical dimensions.

5.5 Optimization Variables

The optimization routine uses the enclosure thermal comfort as its objective function to provide an indication of performance. Performance is evaluated for all the mixed mode strategies, natural ventilation, forced ventilation, and the performance of the chilled beam with each of the mixed mode strategies. The variables for the evaluation of the optimum are dependant upon the particular optimization. For natural ventilation the optimization variables are the window opening position for each hour of the considered day, the window opening position can only be altered once an hour, thus forming 24 optimization variables.

The optimization variable for forced ventilation is the forced fan mass flow rate, again the variable can only be changed hourly. For the optimization of the chilled beam in the different mixed mode strategies the optimization variable is the valve stem position. The valve stem position sets the position of the bypass valve in the chilled water circuit, effectively fixing the mass flow rate of the chilled water to the chilled beam. Again, the valve stem position can only be changed hourly.
5.6 Optimization Flow of Control

To find the optimum performance, (the lowest value of the objective function), the optimization routine runs the composite model a number of times. Using the Complex method, the optimization variables are changed within their boundary limits, in the case of natural ventilation windows opening position is changed between fully closed and fully open. This continues until the optimum is found. For this project the optimum is considered to be found when the enclosure thermal comfort is at a minimum value for the occupied period.

5.7 Enclosure Thermal Comfort Performance Prediction

The prediction of the enclosure thermal comfort performance forms the objective function of the optimization routines. An indication of the comfort response of occupants to an enclosure's thermal environment can be evaluated by application of the comfort requirement standard, ISO 7730-1984[76]. The standard specifies methods of measuring and evaluating moderate thermal environments for human thermal comfort. Thermal comfort can be predicted using a 7 point bipolar index, the Predicted Mean Vote index, (PMV). The standard also specifies a method of assessing dissatisfaction with the thermal environment, Predicted Percentage Dissatisfied, (PPD).

Evaluation of the Comfort Performance of an enclosure utilises both the PPD and PMV. Rather than an instantaneous evaluation, the comfort performance of an enclosure is evaluated as an integral of a function of the PPD over a period of a year[77]. Integrating thermal comfort response over a time interval allows transient discomfort conditions (cool early mornings, slight over-heating late afternoon) to occur, given that the comfort performance over the time interval stays below a predefined limit.
5.7.1 Human Thermal Comfort

An established quantitative model for the assessment of human thermal comfort is the Fanger model[78]. The model, calibrated empirically with a large number of subjects, performs a heat balance upon the human body. The resultant expression, Predicted Mean Vote (PMV), provides occupant thermal sensation response to the local environment. The PMV is determined by parameters relating to both occupants, metabolism, activity level and clothing level, and the environment, air temperature, mean radiant temperature, humidity and air velocity. The PMV expression is found thus:

\[ PMV = (0.0303e^{-0.036M} + 0.028) \left\{ (M - W) - 3.05 \times 10^{-3} \right. \\
\times \left[ 5733 - 6.99(M - W) - p_a \right] - 0.42 \\
\times \left[ (M - W) - 58.15 \right] - 1.7 \times 10^{-8}M \left(5867 - p_a \right) \\
- 0.0014M(34 - t_a) - 3.96 \times 10^{-8}f_{cl} \\
\times \left[ (t_{cl} + 273.0)^4 - (t_r + 273.0)^4 \right] - f_{cl}h_{c}(t_{cl} - t_a) \right\} \]

(5.1)

where \(M\) is the metabolic rate, in watts per square metre of body surface area, \(W\) is the external work, \(W/m^2\), equal to zero for most activities, \(I_{cl}\) is the thermal resistance of clothing, \(m^2K/W\), \(f_{cl}\) is the ratio of man's surface area while clothed to man's surface area while nude, \(t_a\) is the air temperature, \(t_r\) is the mean radiant temperature, \(v_{ar}\) is the relative air velocity, \(ms^{-1}\), \(p_a\) is the partial water vapour pressure, \(Pa\), \(h_c\) is the convective heat transfer coefficient, \(W/m^2K\) and \(t_{cl}\) is the surface temperature of clothing. The surface temperature of clothing, the convective heat transfer coefficient and the ratio of clothed surface area to naked are found from:

\[ t_{cl} = 35.7 - 0.28(M - W) - I_{cl} \left\{ 3.96 \times 10^{-8}f_{cl} \\
\times \left( t_{cl} + 273 \right)^4 - (t_r + 273)^4 \right\} + f_{cl}h_{c}(t_{cl} - t_a) \]  

(5.2)

\[ h_{c} = \begin{cases} 2.38(t_{cl} - t_a)^{0.25} \text{ for } 2.38(t_{cl} - t_a)^{0.25} > 12.1v_{ar}^{0.25} \\ 12.1v_{ar}^{0.25} \text{ for } 2.38(t_{cl} - t_a)^{0.25} \leq 12.1v_{ar}^{0.25} \end{cases} \]

(5.3)
Iteration is required to calculate $t_d$ and $h_c$ due to the mutual dependence of the expressions. The PMV index has a seven point bipolar thermal sensation scale, which corresponds to the following:

- $+3$ hot
- $+2$ warm
- $+1$ slightly warm
- $0$ neutral
- $-1$ slightly cool
- $-2$ cool
- $-3$ cold

Thermally neutral conditions, (thermal balance), exist when the human internal heat production is equal to the heat loss to the environment.

The predicted percentage dissatisfied, PPD index, establishes the proportion of thermally dissatisfied persons. PPD is given as:

$$PPD = 100 - 95e^{0.0333PMV^4 + 0.2179PMV^2} \quad (5.4)$$

Inspection of the PPD equation shows that for PMV votes of zero the value of predicted percentage dissatisfied is 5%. Hence, even at optimum values of PMV, not all occupants will be satisfied with the thermal environment.

### 5.7.2 Enclosure Comfort Performance

PMV evaluation provides an instantaneous thermal comfort response. To measure the comfort performance of an enclosure it has greater emphasis if the thermal comfort response over time is evaluated. Mathematically this requires the integration of thermal comfort response, the integrated variable becoming the comfort performance of the enclosure.
The Government Building Agency in The Netherlands\cite{ref77} proposed a state comfort performance function. Based on Annex D of the ISO 7730 standard, which recommends that PPD should be below 10% for acceptable thermal conditions, the function depends not only on whether the PPD exceeds 10% but also the interval over which discomfort occurs. The GBA standard state comfort performance function is shown thus; Figure 5.1.

Integration of the state comfort performance yields the building comfort performance. Integration over a year, and comparison with the recommended minimum -150 hours, establishes the enclosure comfort performance $U_B$, thus:

$$
U_B = \int_T U_S \delta t
$$

(5.5)

where $U_S$ is the state comfort performance and $T$ is the prescribed time interval over which the building comfort performance is evaluated.
5.8 Composite Model Optimization: the Complex method

Analysis of chilled beam performance in mixed mode buildings requires comparison between mixed mode strategies. This is achieved by creating a minimum load condition for the chilled beam using an optimisation tool. The optimized parameters, window position, valve position and fan speed are obtained using the Complex search method [79], an established algorithm that has proved effective in optimizing zone temperature setpoints in buildings that utilize fabric thermal storage [80].

The Complex method, adapted from the Simplex direct search method by Box (1965)[79], is an empirical optimization procedure used to solve constrained minimization problems. The optimisation of, the valve position of the chilled beam for comfort performance for example, can be described as a multi-variable constrained optimisation problem. The problem is to minimise the objective function:

$$f (x_1, x_2, ..., x_n)$$

subject to the explicit contraints:

$$g_i \leq x_i \leq h_i, \quad i = 1, ..., n$$

and the implicit contraints:

$$g_i \leq y_i \leq h_i, \quad i = 1 + n, ..., m$$

The implicit variables $y_{n+1}, ..., y_m$ are dependent functions of $x_1, ..., x_n$. The constraints $g_i$ and $h_i$ are either constants or functions of $x_1, ..., x_n$.

The Complex method generates a geometric figure with an initial feasible guess and randomly constituted feasible vertices. The worst vertex of the geometric figure is reflected through the centroid of the remaining vertices. If this generated
point provides a lower value of the objective function a new geometric figure is formed, the new point replacing the old "worst" point. In this way geometric figures roll over each other and collapse to converge upon the "best" objective function.

If the nonlinear problem in Equation 5.6 is to be solved, it is necessary to use \( k \) points for a geometric figure, where \( k \geq n + 1 \) (according to Box should \( k = 2n \)). The starting points are randomly generated so that both the implicit and explicit conditions in Equations 5.7 and 5.8 are satisfied. Let the points \( x^h \) and \( x^i \) be defined by:

\[
f(x^h) = \max f(x^1), f(x^2), \ldots, f(x^k) \quad (5.9)
\]

\[
f(x^i) = \min f(x^1), f(x^2), \ldots, f(x^k) \quad (5.10)
\]

Calculate the centroid \( \bar{x} \) of these points except \( x^h \) by:

\[
\bar{x} = \frac{1}{k-1} \sum_{i=1}^{k} x^i, \quad x^i \neq x^h \quad (5.11)
\]

The algorithm then replaces the worst point \((x^h)\) by a new and better point. The new point \( x^r \) is calculated as a reflection of the worst point through the centroid:

\[
x^r = \bar{x} + \alpha (\bar{x} - x^h) \quad (5.12)
\]

where the reflection coefficient is chosen as \( \alpha = 1.3 \) (according to Box).

The point \( x^r \) is examined with regard to explicit and implicit constraints and if it is feasible \( x^h \) is replaced with \( x^r \) unless \( f(x^r) \geq f(x^h) \). In that case, it is moved halfway towards the centroid of the remaining points:

\[
x^{(\text{new})} = \frac{1}{2} (\bar{x} + x^r) \quad (5.13)
\]
This is repeated until it stops repeating as the worst (highest) value. However, as pointed out by Guin[81] (1968), this cannot handle the situation where there is a local maximum located at the centroid. The method used here is to gradually move the point towards the minimum value if it continues to be the worst value. This will, however, mean that two points can come very close to each other compared to other points, with a risk of collapsing the complex. Therefore, a random value is also added to the new point. In this way, the algorithm will take some extra effort to search for a point with a better value, but in the neighbourhood of the point of the minimum value. It is consequently guaranteed that a point better than the worst of the remaining points will be found.

Convergence of the complex is tested according to:

- The complex shrinks to a specified size.
- The standard deviation of the function value becomes sufficiently small.

5.9 Performance Analysis - Weather Data

Three different summer ambient conditions and one transitional day were chosen for the performance analysis (Figure 5.2): an average UK summer day taken from August of a “standard” weather year, (Kew, UK, 1964); a hot summer day; and a day with a large diurnal temperature variation. The hot day, transitional day and day with a large diurnal temperature variation were taken from data measured in Garston, UK, 1994. The hot summer day was chosen to investigate the performance for a potentially overheated building, whereas the day having a large diurnal temperature variation was selected to investigate the impact of ventilation night cooling. The transitional day was chosen to investigate the influence of chilled beam use in moderate climatic conditions.

5.10 Energy Cost- Tariff Structure

The tariff structure provides an indication of the relative cost of the primary energy use, (both gas and electricity), of each of the mixed mode strategies. One
5.11. Summary

Two pricing levels are adopted for a tariff on the electrical demand upon the primary plant. From 0:00 to 7:00 the tariff is rated at 2.79 pence per kWh, and 4.87 pence per kWh at all other times \[82\]. The electrical demand is evaluated for the compressor in the chiller plant, (sub-dew point chilled beam), and the fan, humidifier and cooling coil in the mechanical ventilation systems.

5.11 Summary

Integration of the enclosure dynamic thermal model, the moisture mass balance model, the sub-dew point chilled beam model and the thermal plant models and the solution approach has been described. An optimization routine, the Complex method, has been chosen and the objective function of the optimization, the en-
5.11. Summary

closure comfort performance, and the optimisation variables, have been described. For this project a minimum value of the enclosure thermal comfort integral over the occupied period of the zone is considered the optimum performance of the particular mixed mode strategy.
Chapter 6

Performance Analysis for Passive Building Control

Introduction

Analysis of simulation results is divided into passive and active mixed mode strategy simulations. Passive mode simulation analysis investigates the simulation performance, enclosure thermal weight, external heat disturbances within a natural ventilation mixed mode strategy. Additionally, the passive chapter will include an investigation of chilled beam performance at and below the saturation temperature of the zone. Active simulation analysis includes the investigation of the performance of a sub-dew point chilled beam within a forced fan, (Chapter 7), and a displacement ventilation strategy, (Chapter 8). Full HVAC ventilation strategies and comparison of active and passive mixed mode strategies and the role of sub-dew point chilled beam will be investigated in the performance comparison chapter, (Chapter 9).

6.1 Natural Ventilation

For a naturally ventilated building, the zone thermal comfort conditions are dependent on the extent to which the ventilation control strategy responds to the
prevailing climatic conditions. In general, control strategies for naturally ventilated buildings are heuristic, in this investigation optimization of window position dictates the control strategy. The optimized ventilation strategy will maximize enclosure thermal comfort to make full use of free cooling.

The effect of window opening position upon internal conditions and building fabric storage, for different thermal weights of building, will be investigated and the natural ventilation driving forces identified, and the optimized ventilation strategy, which maximizes thermal comfort during occupied periods, analysed. An optimized ventilation strategy which has the minimum ventilation requirement as an optimization constraint will also be investigated. Chilled beam performance will be evaluated for an optimized window opening position strategy, a low load condition for the beam. The beam output is then to be optimized using the same objective function, namely to minimize occupant discomfort in occupied periods, as the ventilation strategy. The effect of fabric thermal storage upon beam operation and the optimized ventilation strategy will also be investigated.

6.1.1 Natural Ventilation: Enclosure Response

Before beginning analysis of the natural ventilation strategy it is important that the natural ventilation model prediction, and the enclosure thermal and latent model response, matches an expected response. Using the expected response method of validation given by Wortman[47] and Wiltshire[48], a single sided natural ventilation strategy is tested by running the simulation keeping all the parameters constant apart from changing the opening position of the window. The simulations were performed at opening areas of 0, 2, 3, 4 and 6 m², (the maximum opening area), for the four considered weather days, (section 5.6), with a heavy weight construction (section 5.3). Dashed lines represent fully open windows and dot-dash lines represent fully closed windows. Optimized profiles of natural ventilation strategies are to be found for all the considered weather days, testing over the full range of operation will reinforce the validity of any optimum and further, the reason for the optimum can more easily be determined.

Figure 6.1, shows the enclosure air temperature and moisture content response to the natural ventilation strategy for the different window opening areas, for the considered weather days. Dashed lines represent fully open windows and dot-dash
6.1. Natural Ventilation

lines represent fully closed windows. The dotted lines represent the external air conditions. It is clear that for all the test days the response of the enclosure air to the increased window opening is the same, in that, the greater the window opening area the closer the internal conditions tend toward external conditions, with the air temperature reducing by 3 to 4 Kelvin. The amplitude of the diurnal profiles also closely approximates the external conditions. This result is expected, the greater window opening area provides a greater ventilation area, which, typically, increases the ventilation air mass flow rate, meaning the external conditions will have greater influence upon the internal air composition.

Figure 6.2, shows the ceiling mass and surface temperature response, for the considered weather days, with the window opening position changing as before. Dashed lines represent fully open windows and dot-dash lines represent fully closed windows. The upper profiles show the ceiling mass temperature response, the lower profiles show the ceiling internal surface temperature response. The ceiling surface temperatures show a response which closely approximates the zone air temperature response, with the surface temperature decreasing with increasing window opening area. The reduction of surface temperature with increasing window opening area is approximately 1 Kelvin over the full range of opening area, less than the air response, due to the heat transfer flux being "shared" over all the incident surfaces in the enclosure. The surface temperature also shows a similar diurnal profile with the peak and trough temperatures coinciding with the air temperature response, but with a smaller amplitude. This phenomena, mirrored in all the building element surfaces, exists due to the fast convective and radiative coupling which exists between the air and surface nodes, the lower amplitude exists due to the slower thermal response of the surface temperature.

The ceiling mass temperatures show a response which is markedly different to the surface profile, the peak and troughs of the profile do not coincide with the air temperature, lagging by 1 to 2 hours, the profile amplitude is much lower (2 to 3 Kelvin), however, the change in mass temperature for increasing window opening area is reduced like the surface temperature, but at a much lower amount, less than half a degree Kelvin. All these phenomena are due to the increased thermal capacity of the ceiling fabric which reacts much slower than both the air and surface temperature to heat disturbances, this can be seen as the mass temperature only starts to increase, for all profiles, at approximately 10:00 hours, where the air temperature begins to rise on occupancy at 8:00 hours. The slow
Figure 6.1: Window Opening Position: Air Temperature and Moisture Content
reaction time of the mass leads to the reduced amplitude, on occupancy the mass reacts to a positive heat disturbance, stops cooling from the previous night and begins to warm. As the occupancy period ends, (the mass is still reacting to the earlier disturbance), the air temperature, external and internal, fall which slows the mass temperature increase until, always lagging the surface temperature, the mass temperature falls. The change in mass temperature for increasing window opening area is lower than the surface temperature response, this is due to the increased thermal capacity of the building fabric.

Figure 6.3, shows the diurnal heat flux profiles for heat transfer through the enclosure floor, for the four considered weather days, with the window opening position changing as before. Dashed lines represent fully open windows and dot-dash lines represent fully closed windows. The upper profiles show the heat flux from the external temperature node to the floor mass temperature node, the lower profile shows the heat flux profile from the floor mass temperature node to the internal surface node. For all the considered weather days the heat flux of the top profiles is negative overnight, indicating the direction of the heat flux is to outside, and positive in the day indicating the direction of the heat flux is toward the mass node. For the lower profiles the heat flux overnight is positive, indicating the direction of the heat flux is into the enclosure, and negative in the day indicating the heat flux is toward the mass node. This trend, of heat flux away from the mass node overnight and toward the mass node in the day, represented in Figure 6.4, indicates the nature of the thermal storage in the massive elements in summer conditions.

Figure 6.5, shows the change in thermal storage diurnal profiles for the floor, overnight the profile is negative, indicating release of heat (flux away from the mass node), in the day the thermal storage profile is positive, indicating heat storage (flux toward the mass node). This phenomena is typical of summertime thermal mass response, in that daytime temperatures, both inside and out, are likely to be greater than the mass temperature which, having a greater thermal capacity, takes longer to “warm-up”. The building fabric lags the fast response of the air temperatures and overnight the fabric is warmer than the air and “releases” the heat to the enclosure and the external environment.

Increasing the window opening area for all the considered days reduces the internal
6.1. Natural Ventilation

![Diagram of temperature changes over time for different window opening positions: Average, Hot, High Diurnal, Transitional.](image)

Figure 6.2: Window Opening Position: Mass and Surface Temperatures
Figure 6.3: Window Opening Positions: Building Fabric Heat Flux
6.1. *Natural Ventilation*

Figure 6.4: Heat Flux Paths: Building Fabric

...
6.1. Natural Ventilation

Figure 6.5: Window Opening Positions: Floor Fabric Storage
Figure 6.6, shows the driving force of natural ventilation, the external/internal temperature difference, the ventilation heat flux associated with the driving force, and the PMV index of the enclosure for the increasing window opening area. Dashed lines represent fully open windows and dot-dash lines represent fully closed windows. The negative temperature difference and the negative heat flux due to natural ventilation reduce as the opening area increases, this is because as the opening area increases the internal air temperature approaches the external temperature, reducing the temperature difference which is the potential for the heat flux. The PMV index profiles for all the weather days decrease as the window opening area increases. For the hot day this decrease improves the comfort throughout the whole day. For the average and high diurnal day, the reduction of comfort in the occupied period is an improvement, but out of the occupied period the comfort is reduced, showing the potential for night cooling. For the transitional day the reduction in the PMV index leads to increased discomfort. The PMV index profiles promote the premise that, if the PMV value is positive and a cooling potential exists the window should be open, and, if the PMV value is negative and a cooling potential exists the window should be closed, and vice versa.

The trend of the internal air node, the resultant changes in the surface and mass temperatures and the PMV response of the enclosure can easily be described by the heat flux due to ventilation or the internal/external temperature difference. Potential for night cooling has been observed, and inspection of the PMV profiles has lead to intuitive descriptions of the optimized profiles, the next section will describe the optimized ventilation strategy.

6.2 Optimized Window Position

An optimized summer natural ventilation strategy would maximize the use of night free cooling while ensuring that the building was not excessively cool at the start of occupancy. Therefore, during periods where free cooling is available, the ventilation openings will tend to be open, and at other periods, tend to be closed. The results for the “average” summer day clearly illustrate this characteristic (Figure 6.7). During the early part of the night, there is no potential for free cooling and the ventilation openings tend to be closed. However, a few hours before the start of occupancy, free cooling became available and the ventilation
6.2. Optimized Window Position

Figure 6.6: Window Opening Position: Driving Force
open area was increased. Since the zone was too cool (PMV < 0.0), just prior to occupancy, the ventilation openings were closed so that as soon as the gains due to the occupants and equipment occurred, the PMV was brought back towards “neutral” (0.0 PMV). Once inside the comfort band however, the internal gains rapidly increase the potential and need for free cooling, with the result that the ventilation open area was once again increased.

The ventilation characteristics are repeated for the “hot” summer day, the day with a “large diurnal variation” in ambient temperature, and the transitional day. However, for the “hot” day, the closure of the ventilation openings just prior to the start of occupancy is not due to the need to move the zone towards a “neutral” PMV, but rather that the ambient conditions are such that any ventilation would tend to increase the zone temperature and move the PMV further from “neutral”. The benefit of night cooling is clear for the day with a large diurnal variation in ambient temperature, since the windows are fully open during the night with only a brief period of closure at the start of occupancy (in order to move the PMV towards “neutral”). The closure of the windows to prevent “over-cooling” is clear for the transitional day, since the windows have negligible opening area, a brief period of complete closure exists at the start of occupancy, (in order to move the PMV toward neutral), with a slight opening late afternoon, off-setting the internal loads (and an increase in the PMV).

For all the test cases, the optimized natural ventilation strategy for light weight building (Figure 6.8) show close agreement with the heavy weight strategies. Increasing the thermal mass reduces the amplitude of the diurnal temperature swing for the building mass, surface and air temperatures. The window opening strategies are similar however, since the general profile of both the PMV and the ambient to zone temperature difference remain the same. The thermal storage of the heavier mass building also reduces the amplitude of the PMV profiles. For the “average”, “large diurnal variation”, and “transitional” days, the reduction in amplitude produces PMV values which closer to thermal neutrality. The “hot day” reduction in PMV amplitude results in a lower positive PMV during the occupied period and higher positive PMV during the unoccupied period. The effect of thermal mass upon the diurnal temperature profiles, is shown, for the “hot day”, in Figure 6.9, the heavy and light weight profiles represented by solid and dotted line respectively.
6.2. Optimized Window Position

Figure 6.7: Optimized Ventilation Strategy - Heavy Weight Construction
Figure 6.8: Optimized Ventilation Strategy - Light Weight Construction
6.2. Optimized Window Position

Figure 6.9: Ceiling Thermal Mass Comparison: Hot Day Window Optimized Profiles
6.3. Optimized Window Position: Minimum Ventilation Requirement

6.2.1 Sensitivity of the Optimization

It is clear that in this study, the optimized ventilation strategy could be largely replicated by simple rule based approach (apart from the period immediately prior to occupancy, if \( T_o < T_a \) then the windows should be open, else the windows should be closed). This raises the question as to why the optimization algorithm did not always yield fully open or fully closed window opening positions? The answer to this lies in the extent to which the comfort integral is sensitive to a given window opening, the sensitivity increasing with the difference in the ambient and zone air temperatures (not only does a large temperature difference increase the potential for cooling (or heating), but the larger the temperature difference, the higher the buoyancy driven ventilation rate). This effect can be seen by comparing the window open areas of the “hot” day with the areas for the “average” and “large diurnal variation” days (Figure 6.7). The temperature difference during the occupied period is smaller for the “hot” day than for the other two test days. The reduced sensitivity of the “hot” day led to the optimization converging on less than fully open window openings, whereas the larger temperature difference for the other two test days resulted in greater sensitivity and convergence on the more correct, fully open windows. This is replicated in the “transitional day”, where the closed characteristics of the light weight profile are more pronounced than the heavy weight profile, as the light weight response returns a higher temperature difference.

It can be concluded, that the optimization method clearly identifies the optimum ventilation characteristic but for some insensitive periods of ventilation, the solutions may only be near optimal. The next section will investigate the window opening position when the constraint of minimum ventilation requirement for occupants is introduced.

6.3 Optimized Window Position: Minimum Ventilation Requirement

The premise behind this optimization investigation is to analyse the effect upon the window optimized profiles if a further constraint is added to the optimization
6.3. Optimized Window Position: Minimum Ventilation Requirement

routine, that of the minimum ventilation requirement of the zone being met by the natural ventilation strategy alone. The objective function for the ventilation profiles was once again the comfort integral during the occupied period. CIBSE [28], states that the minimum ventilation requirement for an office space is 8 l/s per person, in legislative occupancy, (one person for each 3.7 m² of floor area), this equates to an enclosure ventilation rate of \(0.324 \text{ m}^3/\text{s}\), 2.6 air changes per hour. The minimum ventilation requirement constraint only applies during the occupied period, 0800 hours to 1800 hours.

Figure 6.10 illustrates the diurnal profiles for the optimized natural ventilation strategy with the minimum ventilation constraint, for the four test days and a heavy weight construction, lightweight profiles again being very similar. The upper graphs compare the volume flow rate through the window openings, (solid line), with the minimum ventilation volume flow rate, (horizontal solid line). The middle graphs compare the predicted mean vote for the optimized natural ventilation strategy with, (solid line), and without, (dotted line), the constraint of the minimum ventilation requirement. The lower graphs illustrate the opening position of the constrained natural ventilation strategy.

It is clear from the graphs that the added constraint of the minimum ventilation rate is met in each of the four test days, the ventilation rate given by the optimized ventilation strategy is always greater than the minimum ventilation rate in the occupied period. Inspection of the window opening profiles indicates that this is achieved in the “average”, “hot” and “large diurnal variation” day by fully opening the windows, the minimum ventilation rate is achieved in the “transitional” day with a “two thirds open” window position. The transitional day has the highest internal/external temperature difference which increases the flow rate of the temperature driven flow, leading to the minimum ventilation rate being met with a smaller window opening area.

The minimum ventilation rate constraint decreases the positive PMV profiles, beneficial cooling, during the occupied period for the “average”, “hot” and “large diurnal variation” days, with similar PMV profiles outside of the occupied period. The PMV profiles for the “transitional” day all have greater negative values, producing too cool internal conditions. With the exception of the “transitional” day the ventilation profiles are very similar to the non-constrained profiles outside of occupancy, this explains the similar PMV prediction at these times. The
ventilation strategy for the "transitional" day is markedly different to the non-constrained strategy. The non-constrained strategy closed the window to prevent "over-cooling", the minimum ventilation constraint "forces" the opening of the windows to meet the required flow rate, this over-cools the space.

Again for the "average", "hot" and "large diurnal variation" day the minimum ventilation constraint "forces" the opening of the windows to meet the required flow rate, this provides additional beneficial cooling which is not given by the non-constrained strategy, undermining this strategy as a true optimum, the constrained strategy has the better performance. The reason for the increased performance of the constrained ventilation strategy is that during the occupied periods, where the constrained strategy indicates the need for fully opened windows, the windows were not opened fully in the non-constrained strategy due to the insensitivity of the solution close to thermal neutrality.

It can be concluded, with the exception of the "transitional" day, that the introduction of a minimum ventilation rate constraint improved the comfort performance of the natural ventilation strategy during periods of insensitivity. The next section investigates the performance of the chilled beam within the optimized ventilation strategies.

6.4 Optimized Chilled Beam Operation

The optimized window openings obtained from the constrained ventilation optimization for the "average", "hot" and "large diurnal variation" day and the non-constrained ventilation optimization for the "transitional" day were used during the optimization of the beam operation, the resultant zone conditions then represent a minimized load on the chilled beam. The optimization criterion for beam operation was once again the comfort integral during the occupied period. Figures 6.11 and 6.12 illustrate the optimized beam operation for both lightweight and heavyweight constructions respectively; the horizontal axis of the graphs is the time of day with the two vertical dotted lines indicating the occupied period. The beam operation is indicated by the fractional position of the valve controlling the water flow to the beam. The PMV graphs illustrate two results, the solid lines indicating the comfort conditions for natural ventilation alone, and the dashed
Figure 6.10: Optimized Ventilation Strategy: Minimum Ventilation Constraint
6.4. Optimized Chilled Beam Operation

line the effect of the chilled beam operation.

All four test days have a period during occupancy when cooling is required and the beam is expected to be fully ON, and both lightweight and heavyweight constructions have similar profiles of valve opening position. This is clear for the "hot" day where the beam was continuously operated with the control valve fully open. Similarly, for the day having a "large diurnal variation" in ambient temperature, there is a need for cooling during occupancy. However, unlike the "hot" day, the zone was too cool at the start of occupancy, which resulted in the closure of the control valve, and the PMV moving back towards "neutral" due to the internal gains.

The results for the "average" and "transitional" day are perhaps less clear. This is due to the comfort conditions during occupancy being closer to neutral, which reduces the need for cooling and consequently, the sensitivity of the optimization. The results are clear however, in that some night cooling from the beam has been utilized, (the "transitional" day having the greater potential), followed by closure of the valve at the start of occupancy (in order to allow the internal gains to bring the PMV towards "neutral"). Once the PMV begins to rise, the valve was opened once again, only to be closed, for the "average day", once the PMV was in the region close to 0.0, (the optimization problem being least sensitive during this period).

It is interesting to note that, for the "average", "large diurnal variation" and "transitional" days, in order to suppress the afternoon overheating, the period of "cool" comfort conditions during early occupancy was increased. This characteristic would not result from a more conventional control strategy, in which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling). It is also likely that a different operating characteristic would result from an energy optimization (with comfort being an optimization constraint rather than the optimization criterion).
Figure 6.11: Optimized Beam Operation - Light Weight Construction
Figure 6.12: Optimized Beam Operation - Heavy Weight Construction
6.5 Chilled Beam Heat Transfer

Chilled beam heat transfer is analysed using both the optimized ventilation strategies and the optimized valve position strategies. The chilled beam has three heat transfer mechanisms which contribute to the cooling capacity, radiation, convection and condensation. Figures 6.13 and 6.14, illustrates the chilled beam heat transfer for all the test days with a light weight construction, figures 6.15 and 6.16 illustrate chilled beam heat transfer for heavy weight construction for two 100.0mm diameter cylinder plain aluminium beams. It is clear that the greatest heat transfer mechanism is the convective heat transfer followed by the latent and then radiant heat transfer. In comparison to the convective and latent heat transfer, the radiant heat transfer is relatively constant throughout the day. The buoyancy driven convective heat transfer increases with the zone temperature during the high gain occupied period. The latent gain to the zone during occupancy also promotes the latent heat transfer to the beam.

Comparison of heat transfer between the test days shows that higher internal temperatures increase the heat transfer to the beam, ie that the hot day has the greatest heat transfer to the beam. The proportion of the heat transfer mechanisms does not change in any of the test days, buoyancy driven convective heat transfer remains the dominant transfer mechanism contributing approximately 75\% of the total heat transfer, radiant fraction contributes approximately 10\%, and the latent fraction of the total heat transfer is approximately 15\%.

6.6 Chilled Beam Performance

The energy performance of the sub-dew point chilled beam is evaluated as a coefficient of performance, (COP), of the chiller plant. This performance is closely correlated to the chilled water leaving temperature, ambient (external) air temperature and the chiller part load ratio [80]. Figure 6.17 shows the part load characteristics of the chiller model within the range of operation of the chiller. Clearly, a lower differential of ambient and chiller leaving temperature plus a part load close to the design maximum load gives the best chiller performance. These profiles are indicative of a chiller with a 'poor' part load characteristic [80].
6.6. Chilled Beam Performance

Figure 6.13: Heat Transfer Fractions: Lightweight
6.6. Chilled Beam Performance

Figure 6.14: Heat Transfer Fractions: Lightweight
6.6. Chilled Beam Performance

Figure 6.15: Heat Transfer Fractions: Heavyweight
Figure 6.16: Heat Transfer Fractions: Heavyweight
Figures 6.18, 6.19, 6.20 and 6.21 show the coefficient of performance (COP, dimensionless), work done, (Watts), by the compressor and the cumulative cost, (Pounds Sterling), of the sub-dew point chilled beam chiller plant for all the test days for light and heavy weight constructions respectively. For the “average”, “large diurnal variation” and “transitional” day the COP returns an average value of 2, increasing during the occupied period where the part load ratio is closer to the design maximum. For the “hot” day the COP falls during occupancy, this is because the “hot” day has a larger temperature differential between the ambient and chiller leaving temperature. The COP falls outside of occupancy due to the lower loads encountered and the poor part load performance of the chiller. The effect of the poor COP value and the increased heat transfer of the “hot” day is evident in the comparison of the cost and energy usage, with the hot day having by far the largest value of both compressor work and cumulative cost.

The “large diurnal variation” and the “transitional” day have similar values of COP, and hence the energy usage and cumulative cost are similar. The reason for the similar COP values is a function of the part load characteristics of the chiller; the “large diurnal variation” day has a larger heat transfer rate, (higher part load
6.6. *Chilled Beam Performance*

An interesting point to note is that the running cost of the chiller is consistently less for heavy weight construction than for light weight, indicating that the dampening of the diurnal temperature swing by using increased thermal mass can reduce...
6.6. Chilled Beam Performance

Figure 6.19: Energy Performance: Lightweight
Figure 6.20: Energy Performance: Heavyweight
6.6. Chilled Beam Performance

Figure 6.21: Energy Performance: Heavyweight
thermal plant running costs.

6.7 Sub-Dew Point Analysis

Traditional chilled beam systems operate above the saturation temperature of the zone to avoid the formation of condensation. Sub-dew point chilled beams can operate above and below the saturation temperature. At lower chilled beam surface temperatures convective and radiant heat transfer will increase, and the condensation mass transfer will produce a latent heat addition. This section investigates the energy and comfort performance of the chilled beam at and below the saturation temperature of the zone. This is achieved in the simulation by maintaining the inlet chilled water temperature at the saturation temperature and 6.4 °C respectively. The investigation concentrates on the “hot” test day conditions which has the greatest cooling demand and so indicates the maximum load performance of the beam.

Figure 6.22 illustrates the total heat transfer, (Watts), and the fractions of the radiant, convective and latent heat transfer for the chilled beam at, (dotted line), and below, (solid line), the saturation temperature, for the “hot” test day and heavyweight construction. It is clear that the total heat transfer of the “sub-saturation” temperature simulation is far greater than the “at-saturation”, and that the “at-saturation” simulation has no latent heat transfer fraction. The relative proportions of the convective and radiant heat transfer remain similar for each case, with the convective heat transfer mechanism dominating.

Comfort performance of the chilled beam at and below the saturation temperature of the zone is provided by the enclosure comfort performance. Figure 6.23 illustrates the predicted mean vote, (PMV), and the cumulative performance of the beam at, (dotted line), and below, (solid line), the saturation temperature for the hot test day and heavy weight construction. Throughout the test day the “sub-saturation” case has a lower positive PMV, (cooler more comfortable conditions), and has a better enclosure comfort performance value. The “sub-saturation” case has the better comfort performance.

Energy performance of the chilled beam at and below the saturation temperature
6.7. Sub-Dew Point Analysis

Figure 6.22: Heat Transfer Fractions: At and Below Saturation Temperature
of the zone is provided by the coefficient of performance, (COP). The COP of a chiller is a function of its design maximum load. Comparison of COP values for different chillers is only valid when the chiller is sized using representative design maximum load. The total heat transfer for the chilled beam beam at and below the saturation temperature of the zone is different, which means the maximum design load for the chiller in each case must be different.

Figure 6.24 show the coefficient of performance, compressor work, (Watts), and the cumulative cost of the chilled beam at and below the saturation temperature of the zone with a chiller with a maximum design load of 10000 and 15000 Watts respectively. Outside of occupancy the two cases show similar values of COP, during occupancy the performance of the “sub-saturation” temperature case is lower, (≈ 2), than that of the “at-saturation” temperature case, (≈ 2.5). The compressor work and the cumulative cost of the “sub-saturation” case is greater than the “at-saturation” case.

The comparison of above and below the saturation temperature of the zone showed that the “sub-saturation” case gives a higher cooling capacity that improves the
6.7. Sub-Dew Point Analysis

Figure 6.24: Energy Performance: At and Below Saturation Temperature
comfort performance of the zone, however, this leads to an increase in both the energy usage and cost and a reduction in the energy performance.

6.8 Summary

The passive simulation testing has been performed. The effect of enclosure thermal weight, external heat disturbances and ventilation opening area upon a zone with a natural ventilation strategy has been analysed. Using numerical optimization procedures optimized ventilation strategies, which provide a minimum load condition for the chilled beam, have been found and evaluated. It can be concluded, that the optimization method clearly identifies the optimum ventilation characteristic but for some insensitive periods of ventilation, the solutions may only be near optimal.

The role of a minimum ventilation requirement as a constraint in the natural ventilation optimization has been evaluated and discussed. It was shown, with the exception of the "transitional" day, that the introduction of a minimum ventilation rate constraint improved the comfort performance of the natural ventilation strategy during periods of insensitivity.

Chilled beam operation strategies have been discussed and a minimum load condition optimization has been performed. It was noted that, for certain test days, in order to suppress the afternoon overheating, the period of "cool" comfort conditions during early occupancy was increased. Additionally, a more conventional control strategy, in which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling), would produce a different optimum. It should be noted that, as the zone comfort performance is based upon a single temperature node it is not possible to analyse location specific thermal comfort. In a zone with a fully opened window an occupant positioned centrally in the zone may experience thermally neutral conditions, whereas an occupant beside the window may experience "too cool" conditions based on the localised air speed. A one node approximation can not show this.

Energy and comfort performance of the chilled beam has been evaluated for "at and below" the room saturation temperature. It was shown that the "sub-
saturation" case gives a higher cooling capacity that improves the comfort performance of the zone, however, this leads to an increase in both the energy usage and cost and a reduction in the energy performance.
Chapter 7

Performance Analysis for Forced Ventilation

Introduction

For a forced fan ventilated building, the zone thermal comfort conditions are dependant on the extent to which the ventilation control strategy responds to the prevailing climatic conditions. Forced fan ventilation provides a controlled constant supply air mass flow rate at external conditions. The ventilation mass flow rate, (unlike natural ventilation), is not dependant upon temperature or pressure gradients, but the controlled fan speed. In this investigation optimization of fan mass flow rate dictates the control strategy. The optimized ventilation strategy will maximize enclosure thermal comfort during occupied periods.

The effect of forced fan mass flow rate upon internal conditions and building fabric storage, for different thermal weights of building, will be investigated and the optimized ventilation strategy, which maximizes thermal comfort during occupied periods, analysed. Chilled beam performance will be evaluated for an optimized forced fan mass flow rate strategy, a low load condition for the beam. The beam output is then to be optimized using the same objective function as the ventilation strategy, namely to minimize occupant discomfort in occupied periods.
7.1 Forced Fan Ventilation: Enclosure Response

Before beginning analysis of the forced fan ventilation strategy it is important that the forced fan ventilation prediction, and the enclosure thermal and latent model response, matches an expected response. Using the known response method of validation given by Wortman[47] and Wiltshire[48], the forced fan ventilation strategy is tested by running the simulation keeping all the parameters constant apart from changing the ventilation supply air mass flow rate. The simulations were performed at mass flow rates of 0.0, 0.25, 0.5, 0.75 and 1.0 kgs\(^{-1}\), (equating to a maximum air change rate of 6.67 ach), for the four considered weather days, with a heavy weight construction. Dashed lines represent the full speed and dot-dash lines represent the off position of the fan. Optimized profiles of forced fan ventilation strategies are to be found for all the considered weather days, testing over the full range of operation will reinforce the validity of any optimum and further, the reason for the optimum can more easily be determined.

Figure 7.1, shows the enclosure air temperature and moisture content response to the forced fan ventilation strategy for the different supply air mass flow rates, for the considered weather days. The dotted lines represent the external air conditions. Dashed lines represent the full speed and dot-dash lines represent the off position of the fan. It is clear that for all the test days the response of the enclosure air to the increased fan speed is the same, in that, the greater the fan speed the closer the internal conditions tend toward external conditions, with the exception of the hot day where the temperature increases. The amplitude of the diurnal profiles also closely approximates the external conditions. This result is expected, the greater ventilation air mass flow rate means the external conditions will have greater influence upon the internal air composition.

Figure 7.2, shows the ceiling mass and surface temperature response, for the considered weather days, with the forced fan ventilation rate changing as before. Dashed lines represent the full speed and dot-dash lines represent the off position of the fan. The upper profiles show the ceiling mass temperature response, the lower profiles show the ceiling internal surface temperature response. As with the natural ventilation strategy, the ceiling surface temperatures show a response which closely approximates the zone air temperature response, with the surface temperature decreasing with increasing air supply mass flow rate. The reduction of surface temperature with increasing air supply mass flow rate is \(\approx 0.5\) Kelvin over the full range of operation, less than the air response, due to the heat transfer
Figure 7.1: Forced Fan Mass Flow Rate: Air Temperature and Moisture Content
7.1. Forced Fan Ventilation: Enclosure Response

Flux being "shared" over all the incident surfaces in the enclosure. The surface temperature also shows a similar diurnal profile with the peak and trough temperatures coinciding with the air temperature response, but with a smaller amplitude. This phenomena, mirrored in all the building element surfaces, exists due to the fast convective and radiative coupling which exists between the air and surface nodes, the lower amplitude exists due to the slower thermal response of the surface temperature.

Again as for the natural ventilation strategy, the response of the ceiling mass temperatures to the forced fan supply mass flow rates show a response which is markedly different to the surface temperature profile, the peak and troughs of the profile do not coincide with the air temperature, lagging by 1 to 2 hours, the profile amplitude is much lower, (2 to 3 Kelvin), however, the change in mass temperature for increasing air supply mass flow rate is reduced like the surface temperature, but at a much lower amount, much less than half a degree Kelvin. All these phenomena are due to the increased thermal capacity of the ceiling fabric which reacts much slower than both the air and surface temperature to heat disturbances, this effect is explained in more detail in the passive performance analysis chapter, (Chapter 6).

Figure 7.31 shows the diurnal heat flux profiles for heat transfer through the enclosure floor, for the four considered weather days, with the air supply mass flow rate changing as before. Dashed lines represent the full speed and dot-dash lines represent the off position of the fan. The upper profiles show the heat flux from the external temperature node to the floor mass temperature node, the lower profile shows the heat flux profile from the floor mass temperature node to the internal surface node. For all the considered weather days the heat flux of the top profiles is negative overnight, indicating the direction of the heat flux is to outside, and positive in the day indicating the direction of the heat flux is toward the mass node. For the lower profiles the heat flux overnight is positive, indicating the direction of the heat flux is into the enclosure, and negative in the day indicating the heat flux is toward the mass node. This trend, represented in Figure 7.4 and discussed in detail in Chapter 6, indicates that the nature of thermal storage in massive elements in summer conditions is typified by daytime storage and night-time release of heat energy.

Figure 7.5, shows the external/internal temperature difference, the ventilation heat flux, and the PMV index of the enclosure for the increasing fan mass flow rate. Dashed lines represent the full speed and dot-dash lines represent the off position
7.1. Forced Fan Ventilation: Enclosure Response

Figure 7.2: Forced Fan Mass Flow Rate: Mass and Surface Temperatures
Figure 7.3: Forced Fan Mass Flow Rate: Building Fabric Heat Flux
Figure 7.4: Forced Fan Mass Flow Rate: Building Fabric Storage
7.2 Optimized Forced Fan Ventilation Strategy

of the fan. Both the positive and negative temperature difference and heat flux reduce as the air supply rate increases, this is because the increased mass flow rate brings the internal air temperature closer to the external temperature, reducing the temperature difference, which is the potential for the heat flux. The PMV profile response is different for each of the weather days. For the "average" day there is a positive heat flux value over night which, for increasing air supply rate, decreases the negative PMV which improves the comfort. For the "large diurnal variation" day there is a negative heat flux which, for increasing air supply rate, increases the negative PMV value which increases discomfort. For both days in the occupied period, there is a negative heat flux which, for increasing air supply rate, decreases the positive PMV which improves the comfort. As the air supply rate increases the heat flux decreases which leads to a diminishing return for the improvement in comfort.

For the "hot" day the positive ventilation heat flux reduces as the air supply rate increases. This produces a very slight reduction in the positive PMV profiles throughout the day, improving the comfort. For the "transitional" day the ventilation heat flux is always negative, and decreases as the air supply rate increases. This would normally reduce the negative PMV values, in this case it does not. This is because the increasing air supply rate also increases the negative PMV value which therefore increases the thermal discomfort. The PMV index profiles promote the premise that, if the PMV value is positive and a cooling potential exists the fan flow rate should be at a maximum, and, if the PMV value is negative and a cooling potential exists the fan flow rate should be minimized, and vice versa.

The trend of the internal air node, the resultant changes in the surface and mass temperatures and the PMV response of the enclosure can easily be described by the heat flux due to ventilation and by the air supply rate. Potential for night cooling has been observed, and inspection of the PMV profiles has lead to intuitive descriptions of the optimized profiles, the next section will describe the optimized forced fan ventilation strategy.

7.2 Optimized Forced Fan Ventilation Strategy

As discussed in the passive performance chapter, (Chapter 6), an optimized summer natural ventilation strategy would maximize the use of night free cooling while
7.2. Optimized Forced Fan Ventilation Strategy

Figure 7.5: Forced Fan Mass Flow: Driving Force
7.2. Optimized Forced Fan Ventilation Strategy

ensuring that the building was not excessively cool at the start of occupancy. A forced fan ventilation strategy would extensively be the same, with the air supply rate minimising the thermal discomfort in the occupied period. However, the air supply mass flow rate for a forced fan strategy is dependant only on the fan speed, and is not dependant upon temperature or pressure gradients between ambient and zone air, therefore closer control of the zone conditions should be expected. Like natural ventilation, forced fan zone comfort performance will still depend upon the ambient conditions, but with higher air change rates, forced fan comfort performance will also be a function of the relative air speed.

The results for the “average” summer day clearly illustrate this characteristic (Figure 7.6). During the early part of the night, there is a potential for “heating”, the PMV value is negative and therefore air supply rate is high to bring the zone closer to thermal neutrality. At occupancy the PMV rises sharply, the cooling potential increases, hence the air supply rate decreases to prevent “over-cooling”. Since the zone was too cool, (PMV < 0.0), just prior to occupancy the ventilation mass flow rate was reduced so that as soon as the gains due to the occupants and equipment occurred, the PMV was brought back towards “neutral” (0.0 PMV). Once inside the comfort band however, the internal gains rapidly increase the potential and need for cooling, with the result that the air supply rate was once again increased. The results for the “large diurnal variation” day replicate the “average” day except for over night. Here the temperature difference is negative and air supply rates are high to reduce the negative PMV value, providing pre-cooling.

The ventilation characteristics for the “hot” summer day are not immediately obvious. They show high air supply rates during the night where positive heat flux would tend to raise the positive, (too hot), PMV value, increasing the discomfort and, during occupancy, where a negative heat flux reduces the positive PMV value. In both instances, over night and during occupancy, the air mass flow rates are high to reduce the thermal discomfort. High air speeds increase the sensation of draught, a “too cold” response, which, for the “hot” day increases the comfort. The reduction of the air supply mass flow rate to prevent “over-cooling” is clear for the “transitional” day. Over night the air supply rates are low, stepping up during the occupied period to avoid late afternoon overheating.

For all the test cases, the optimized forced fan ventilation strategy for light weight building (Figure 7.7) show close agreement with the heavy weight strategies. Increasing the thermal mass reduces the amplitude of the diurnal temperature swing
7.2. Optimized Forced Fan Ventilation Strategy

Figure 7.6: Forced Fan Ventilation Strategy - Heavy Weight Construction
Figure 7.7: Forced Fan Ventilation Strategy - Light Weight Construction
7.3. Sensitivity of the Optimization

for the building mass, surface and air temperatures. The air supply strategies are similar however, (as with the natural ventilation optimization), since the general profile of both the PMV and the ambient to zone temperature difference remain the same. This phenomena is explained in detail in the passive performance analysis chapter, (Chapter 6).

7.3 Sensitivity of the Optimization

It is clear that in this study, (as for the optimized natural ventilation strategy), the forced fan optimized strategy could be largely replicated by simple rule based approach (apart from the period immediately prior to occupancy, if \((T_o < T_a)\) and \(PMV > 0.0\) or \((T_o > T_a)\) and \(PMV < 0.0\) then the air supply rate should be at a maximum, else the air supply rate should be at a minimum). This raises the question as to why the optimization algorithm did not always yield a maximum or minimum air supply rate and why does the optimization algorithm yield maximum air supply rates when the temperature difference is negligible, ("hot" day)? The answer to this lies in the extent to which the comfort integral is sensitive to both the heat flux and the air supply rate, the sensitivity increasing with the difference in the ambient and zone air temperatures and the increasing air mass flow rate, comfort being a function of relative air speeds, and the value of the comfort integral itself. This effect can be seen by comparing the air supply rate overnight for the "average" day with the air supply rate overnight for the "large diurnal variation" day, (Figure 7.6). The temperature difference during the over night is smaller for the "average" day than for the "large diurnal variation" day. The reduced sensitivity of the "average" day led to the optimization converging on a less definite medium air supply rate, whereas the larger temperature difference for the "large diurnal variation" day resulted in greater sensitivity and convergence on the more correct, maximum air supply rate. This is replicated in the "average day", where the air supply rates of the light weight profile are more pronounced than the heavy weight profile, as the light weight response returns a higher temperature difference.

A further point of interest is the "hot" day where during the occupied period the air supply rates are close to the maximum, but the temperature difference is negligible. This is a result of the PMV algorithm being sensitive to relative air speed. High air supply rates, an increased sensation of draught, decrease the PMV value increasing "too cool" thermal discomfort. This increased sensitivity at higher air speeds led to the high air supply rates for the whole of the "hot" day,
and to the low, (but not zero), air supply rates over night for the "transitional" day, (the high flow rates and negative temperature difference both increase the negative PMV value).

It can be concluded, that the optimization method clearly identifies the optimum ventilation characteristic but for some insensitive periods of ventilation, the solutions may only be near optimal. The next section will investigate the optimization of the chilled beam within the optimized forced fan strategies.

7.4 Optimized Chilled Beam Operation

The optimized air supply rates obtained from the forced fan ventilation optimization were used during the optimization of the beam operation, the resultant zone conditions then represent a minimized load on the chilled beam. The optimization criterion for beam operation was once again the comfort integral during the occupied period. Figures 7.8 and 7.9 illustrate the optimized beam operation for both lightweight and heavyweight constructions respectively; the horizontal axis of the graphs is the time of day with the two vertical dotted lines indicating the occupied period. The beam operation is indicated by the fractional position of the valve controlling the water flow to the beam. The PMV graphs illustrate two results, the solid lines indicating the comfort conditions for forced fan ventilation alone, and the dashed line the effect of the chilled beam operation.

With the exception of the "transitional" day, each test day has a period during occupancy when cooling is required and the beam is expected to be fully ON, and both lightweight and heavyweight constructions have similar profiles of valve opening position. This is clear for the "hot" day where the beam was continuously operated with the control valve close to fully open. The results for the "average" and "transitional" day are also clear. Very slight values of positive PMV exist in the afternoon, indicating a need for cooling, this is achieved by the valve opening at occupancy, gradually stepping open as the demand increases. The "transitional" day has no periods where cooling is required, this is mirrored in the valve opening positions, the valve is completely closed, and the beam provides no cooling.

It is interesting to note that, (unlike natural ventilation), only the light weight "large diurnal variation" days increases the period of "cool" comfort conditions during early occupancy in order to suppress the afternoon overheating. This characteristic, which would not result from a more conventional control strategy, in
which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling), only occurs when the heat gains in the afternoon threaten the thermal comfort. The heavyweight "large diurnal variation" day only has a short period of "too warm" PMV, and as a consequence does not instigate night time pre-cooling. As with the natural ventilation beam operation, it is likely that a different operating characteristic would result from an energy optimization (with comfort being an optimization constraint rather than the optimization criterion).
7.4. Optimized Chilled Beam Operation

Figure 7.9: Optimized Beam Operation - Heavy Weight Construction
7.5 Chilled Beam Heat Transfer

Chilled beam heat transfer is analysed using both the optimized ventilation strategies and the optimized valve position strategies. The chilled beam has three heat transfer mechanisms which contribute to the cooling capacity, radiation, convection and condensation. Figures 7.10 and 7.11 illustrate the chilled beam heat transfer for all the test days with a light weight construction, figures 7.12 and 7.13 illustrate chilled beam heat transfer for heavy weight construction for two 100.0mm diameter cylinder plain aluminium beams. It is clear that the greatest heat transfer mechanism is the convective heat transfer followed by the latent and then radiant heat transfer. In comparison to the convective and latent heat transfer, the radiant heat transfer is relatively constant throughout the day. The buoyancy driven convective heat transfer increases with the zone temperature during the high gain occupied period. The latent gain to the zone during occupancy also promotes the latent heat transfer to the beam.

Comparison of heat transfer between the test days shows that higher internal temperatures increase the heat transfer to the beam, i.e. that the hot day has the greatest heat transfer to the beam. The proportion of the heat transfer mechanisms does not change in any of the test days, buoyancy driven convective heat transfer remains the dominant transfer mechanism contributing between approximately 70% of the total heat transfer, radiant fraction contributes approximately 5%, and the latent fraction of the total heat transfer is approximately 25%. The latent heat transfer fraction is higher for chilled beam operation in a forced fan optimized strategy than for the natural ventilation optimized strategy. This is because the higher air supply rates given by the forced fan produce lower zone air conditions, (closer to ambient), than natural ventilation, (Figures 6.1 and 7.1), which means that the total heat transfer to the beam is lower, but the condensation mass transfer rates are similar, which increases the relative latent heat fraction. This is shown for lightweight construction in Figure 7.14.

7.6 Chilled Beam Performance

Figures 7.15, 7.16, 7.17 and 7.18 show the coefficient of performance (COP, dimensionless), work done, (Watts), by the compressor and the cumulative cost, (Pounds Sterling), of the sub-dew point chilled beam chiller plant for all the test days for
Figure 7.10: Heat Transfer Fractions: Lightweight
7.6. Chilled Beam Performance

Figure 7.11: Heat Transfer Fractions: Lightweight
Figure 7.12: Heat Transfer Fractions: Heavyweight
7.6. Chilled Beam Performance

Figure 7.13: Heat Transfer Fractions: Heavyweight
Figure 7.14: Condensation Rate Comparison
light and heavy weight constructions respectively. For the "average" and "large
diurnal variation" day the COP returns an average value of 2, increasing during
the occupied period where the part load ratio is closer to the design maximum.
For the "hot" day the COP falls during occupancy, as with the natural ventila-
tion chill beam profiles, this is because the "hot" day has a larger temperature
derential between the ambient and chiller leaving temperature. The COP falls
outside of occupancy due to the lower loads encountered and the poor part load
performance of the chiller. The effect of the poor COP value and the increased
heat transfer of the "hot" day is evident in the comparison of the cost and energy
usage, with the hot day having by far the largest value of both compressor work
and cumulative cost.

Again, as for natural ventilation, the running cost of the chiller is less for heavy
weight construction than for light weight, indicating that the dampening of the
diurnal temperature swing by using increased thermal mass can reduce thermal
plant running costs.

7.7 Forced Fan Ventilation Summary

The effect of enclosure thermal weight, external heat disturbances and ventilation
opening area upon a zone with a forced fan ventilation strategy has been analysed.
Using numerical optimization procedures optimized ventilation strategies, which
provide a minimum load condition for the chilled beam, have been found and
evaluated. It can be concluded, that the optimization method clearly identifies
the optimum ventilation characteristic, forced fan ventilation optimizations having
increased sensitivity due to the combined cooling effect of temperature potentials
and higher air supply mass flow rates. It should be noted though, that the inputs
for the comfort algorithm are based on the single zone temperature node. Both
temperature and air flow rates in the zone will change with location, (for example,
beneath the chilled beam it is likely that a cold downward plume of air will exist
whose temperature and flow rate will change in relation to the proximity to the
beam), hence the one node approximation only provides an "averaged" indication
of thermal comfort.

Chilled beam operation strategies have been discussed and a minimum load con-
dition optimization has been performed. It was noted that, for certain test days,
unlike natural ventilation strategies where a period of "cool" comfort conditions
Figure 7.15: Energy Performance: Lightweight
Figure 7.16: Energy Performance: Lightweight
Figure 7.17: Energy Performance: Heavyweight
7.7. **Forced Fan Ventilation Summary**

![Graphs showing energy performance for different conditions](image)

**Figure 7.18: Energy Performance: Heavyweight**
during early occupancy was required to prevent afternoon overheating, the forced
fan ventilation optimums provided "closer to thermal optimum conditions" and
pre-cooling was found to be un-necessary in all but one case.

The forced fan optimization was shown to have, (in comparison to natural ven-
tilation), an increased latent heat transfer contribution. This was shown to be a
function of the high air supply rates which gave zone conditions closer to ambient
and consequently decreased the total heat transfer to the beam. The condensation
mass transfer rates were similar which increased the proportion of the latent heat
transfer.
Chapter 8

Performance Analysis for Displacement Ventilation

Introduction

For a building utilizing displacement ventilation, the zone thermal comfort conditions are dependant on the extent to which the ventilation control strategy responds to the prevailing climatic conditions. Displacement ventilation provides a controlled constant supply air mass flow rate at constant temperature, typically 18 °C. The ventilation mass flow rate, (unlike natural ventilation), is not dependant upon temperature or pressure gradients, but the controlled fan speed, typically 8 l/s per person. In this investigation the ventilation strategy will be dictated by the control algorithms which set the temperature and mass flow rate. No optimization of the displacement ventilation strategy will be performed as it is considered that the control strategy will provide the “correct” controlled conditions.

The effect of the controlled displacement ventilation strategy upon internal conditions and building fabric storage, for different thermal weights of building, will be investigated. Chilled beam performance will be evaluated for the controlled displacement ventilation strategy. The beam output is then to be optimized using the minimization of occupant discomfort in occupied periods as an objective function.
8.1 Displacement Ventilation: Enclosure Response

Unlike the previous mixed mode strategies, (natural ventilation and forced fan ventilation), where an optimized strategy was calculated which provided minimum load conditions for the chilled beam optimization, the displacement ventilation strategy will utilize more conventional control algorithms. The displacement ventilation strategy being to control the supply air temperature and volume flow rate to 18 °C and 8 l/s per person respectively. This is achieved using sequential control of a heating coil, cooling coil and a mixing box, (Chapter 4). The ventilation control strategy is operational throughout each test day to provide minimum load conditions, (for the chilled beam), given by the previous mixed mode strategies.

Figure 8.1 shows that the supply conditions for the displacement ventilation are met in all of the test days. The supply temperature and mass flow rate stay within ≈ 3 % of the desired set-point. Figure 8.2 shows the zone air temperature and moisture content for the controlled displacement ventilation strategy. Comparison between these internal conditions and the ones for the forced fan ventilation, (Figure 7.1), show that except for the “hot” day, (where the internal temperature is lower) the displacement ventilation strategy increases the internal temperature. For the zone moisture content, with the exception of the “hot” day, the displacement ventilation strategy produces moisture contents which are similar to the forced fan strategy. The “hot” day has a much lower internal moisture content, (0.0114 kg/kg vs 0.018 kg/kg). To achieve the supply temperature set point on the “hot” day, some degree of cooling is necessary, this will result in dehumidification upon the cooling coil, and a reduction in supply air moisture content.

Figure 8.3 shows the ceiling mass and surface temperature of the zone within the controlled displacement ventilation strategy. As with the natural and forced fan ventilation strategies, the surface temperature has the fastest response time and matches the diurnal changes in the zone air temperature, with the mass temperature lagging. An interesting point to note is the speed of response of the mass temperature in relation to the maximum internal zone temperature. The “hot” day and the “large diurnal variation” day have the greatest internal zone temperatures, and therefore the greatest heat flux potential. This leads to a faster response from the mass temperature, the “trough” of minimum mass temperature for the “hot” day, (the hottest day), is before the “average” day, (the coolest day).

A further interesting point is that, a comparison between the mass temperature of heavy weight and light weight construction for the displacement ventilation strat-
8.1. Displacement Ventilation: Enclosure Response

Figure 8.1: Displacement Ventilation: Supply Conditions
Figure 8.2: Displacement Ventilation: Zone Conditions
Figure 8.3: Building Fabric Response - Heavy Weight Construction

Figure 8.5, shows the diurnal heat flux profiles for heat transfer through the enclosure floor, for the four considered weather days, for the displacement ventilation
8.1. Displacement Ventilation: Enclosure Response

Figure 8.4: Building Fabric Response - Light Weight Construction
strategy. The upper profiles show the heat flux from the external temperature node to the floor mass temperature node, the lower profile shows the heat flux profile from the floor mass temperature node to the internal surface node. As for the forced fan ventilation strategy, all the considered weather days show that the heat flux of the top profiles is negative overnight, indicating the direction of the heat flux is to outside, and positive in the day indicating the direction of the heat flux is toward the mass node. For the lower profiles the heat flux overnight is positive, indicating the direction of the heat flux is into the enclosure, and negative in the day indicating the heat flux is toward the mass node. This indicates that for summer conditions building fabric elements will store heat during the day and release heat at night, (Figure 8.7).

Comparison of the heat flux between the displacement ventilation strategy, (Figure 8.5), and the forced fan strategy, (Figure 7.3), shows that, with the exception of the “hot” day, both daytime storage and night time release are greater for the displacement ventilation. As the internal zone air temperature is greater the potential for heat flow to the building fabric is higher, increasing the storage, this increases the mass temperature which in turn increases the subsequent release. The “hot” day has a lower zone air temperature in the displacement ventilation strategy than the forced fan ventilation strategy and as a result has lower values of heat storage and release.

Figure 8.8 shows the supply air, zone air temperature differential and the PMV profiles for all the test days with a displacement ventilation strategy. Negative values of temperature differential indicate a potential for cooling, positive values indicate a potential for heating, and that cooling or heating respectively is taking place. It is clear, for all the test days, that the displacement ventilation strategy provides cooling during the occupied periods, and with the exception of the “hot” day, that heating occurs during the night. These diurnal profiles form the driving potentials for the zone. It was noted that, except for the “hot” day, the zone air temperatures given by the displacement ventilation strategy were greater than that for the forced fan ventilation. The reason for that is now clear, overnight heating increases the zone air temperature. The “hot” day has lower internal zone temperatures because cooling occurs throughout the day.

The effect upon the PMV profiles is that the “hot” day has markedly lower positive PMV profiles than the forced fan ventilation, indicating cooler more comfortable conditions, and the other test days have greater negative PMV values, indicating warmer more comfortable conditions. A point to note is that the PMV value is
8.1. Displacement Ventilation: Enclosure Response

Figure 8.5: Displacement Ventilation: Building Fabric Heat Flux
8.1. *Displacement Ventilation: Enclosure Response*

![Graphs showing heat flux for different conditions: Average, Hot, High Diurnal, Transitional. The graphs display heat flux as a function of time (in hours).](image)

**Figure 8.6: Displacement Ventilation: Building Fabric Heat Flux**
8.1. Displacement Ventilation: Enclosure Response

Figure 8.7: Displacement Ventilation: Building Thermal Storage
Figure 8.8: Displacement Ventilation: Driving Potential
a function of both the temperature and the relative air movement. Displacement ventilation produces both lower air supply rates and temperatures which are closer to thermal neutrality, which ensures more comfortable conditions in the zone.

Figure 8.9 shows the effect of thermal mass upon the PMV profiles for a displacement ventilation strategy. The dotted line represents the lightweight building fabric and the solid line the heavyweight fabric. It is clear from the profiles that in all of the test days the increase of thermal weight reduces the amplitude of the predicted mean vote values. As the PMV values are sinusoidal around thermal neutrality, a reduction in the diurnal amplitude produces conditions that are closer to thermal neutrality. Hence, for a displacement ventilation strategy, an increase in thermal weight improves comfort conditions in the zone.

The trend of the internal air node, the resultant changes in the surface and mass temperatures and the PMV response of the enclosure can easily be described by the heat flux due to ventilation and by the air supply rate. The effect of thermal weight has been described, and it has been shown that heavy weight building fabric improves the comfort conditions in the zone with displacement ventilation. The next section addresses the optimization of the chilled beam control valve with the controlled displacement ventilation strategy.

8.2 Optimized Chilled Beam Operation

The air supply rates obtained from the enclosure response testing for displacement ventilation were used during the optimization of the beam operation. The optimization criterion for beam operation was once again the comfort integral during the occupied period. Figures 8.10 and 8.11 illustrate the optimized beam operation for both lightweight and heavyweight constructions respectively; the horizontal axis of the graphs is the time of day with the two vertical dotted lines indicating the occupied period. The beam operation is indicated by the fractional position of the valve controlling the water flow to the beam. The PMV graphs illustrate two results, the solid lines indicating the comfort conditions for displacement ventilation alone, and the dashed line the effect of the chilled beam operation.

All four test days have a period during occupancy when cooling is required and the beam is expected to be fully ON, and both lightweight and heavyweight constructions have similar profiles of valve opening position. This is clear for the "hot" and "large diurnal variation" day where the beam was continuously operated with the
8.2. Optimized Chilled Beam Operation

Figure 8.9: Displacement Ventilation: Thermal Weight Comparison
control valve close to fully open. Similarly, for the "average" and "transitional" day there is a need for cooling during occupancy. However, unlike the "hot" and "large diurnal variation" day, the control valve closes at the start of occupancy where the PMV moves back towards "neutral", indicating no control action, due to the internal gains. Outside of occupancy all four test days have the control valve at approximately 80% opening, which steps to fully open during occupancy, except for the valve closure in the "average" and "large diurnal variation" days.

Again, as for the natural ventilation, it is interesting to note that, for all the test days, in order to suppress the afternoon overheating, the period of "cool" comfort conditions during early occupancy was increased. This characteristic would not result from a more conventional control strategy, in which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling). It is also likely that a different operating characteristic would result from an energy optimization (with comfort being an optimization constraint rather than the optimization criterion).

A further interesting point to note is that the valve closure at occupancy only occurs for the test days when afternoon over-heating is less of a problem, ("average" and "transitional" day). This is indicative of a highly sensitive optimization. Additionally, the valve control response is more definite, (valve position closer to fully open), in the lightweight case. This is because the heavyweight construction dampens the internal temperature swing, which leads to a cooler occupied period, (lower positive PMV improving the comfort), and a warmer unoccupied period, (lower negative PMV improving the comfort), which calls for a less definite, (less-er), response from the chilled beam control valve.

8.3 Chilled Beam Heat Transfer

Chilled beam heat transfer is analysed using the optimized valve position strategies within the controlled displacement ventilation strategy. Figures 8.12 and 8.13, illustrate the chilled beam heat transfer for all the test days with a light weight construction, figures 8.14 and 8.15 illustrate chilled beam heat transfer for heavy weight construction for two 100.0mm diameter cylinder plain aluminium beams. As for all previous tests it is clear that the greatest heat transfer mechanism is the convective heat transfer followed by the latent and then radiant heat transfer. In comparison to the convective and latent heat transfer, the radiant heat transfer
Figure 8.10: Optimized Beam Operation - Light Weight Construction
Figure 8.11: Optimized Beam Operation - Heavy Weight Construction
is relatively constant throughout the day. The buoyancy driven convective heat transfer increases with the zone temperature during the high gain occupied period. The latent gain to the zone during occupancy also promotes the latent heat transfer to the beam.

Comparison of heat transfer between the test days shows that higher internal temperatures increase the heat transfer to the beam, ie that the hot day has the greatest heat transfer to the beam. The proportion of the heat transfer mechanisms does not change in any of the test days, buoyancy driven convective heat transfer remains the dominant transfer mechanism contributing approximately 90% of the total heat transfer, radiant fraction contributes approximately 4%, and the latent fraction of the total heat transfer is approximately 6%. The latent heat transfer fraction is lower for chilled beam operation in a displacement ventilation strategy than for both a natural ventilation and forced fan optimized strategy. This is because the total heat transfer to the beam is greater, (approximately double that of the natural ventilation strategy), due to the greater temperature differential between chilled beam surface and zone air temperature, (displacement air supplied at a constant 18 °C producing higher zone air temperatures) even though the condensation rate and latent heat transfer rates are similar, (Figure 8.16).

8.4 Chilled Beam Performance

Figures 8.17, 8.18, 8.19 and 8.20 show the coefficient of performance (COP, dimensionless), work done, (Watts), by the compressor and the cumulative cost, (Pounds Sterling), of the sub-dew point chilled beam chiller plant for all the test days for light and heavy weight constructions respectively. For all the test days the COP returns an average value of 3, higher than the forced fan and the natural ventilation because the part load ratio is closer to the design maximum. For all the test days the COP falls during occupancy, because the temperature differential between the ambient and chiller leaving temperature increases. The COP falls outside of occupancy due to the lower loads encountered and the poor part load performance of the chiller. As for both the natural ventilation and the forced fan chilled beam optimizations the effect of the poor COP value and the increased heat transfer of the “hot” day is evident in the comparison of the cost and energy usage, with the hot day having by far the largest value of both compressor work and cumulative cost.
Figure 8.12: Heat Transfer Fractions: Lightweight
Figure 8.13: Heat Transfer Fractions: Lightweight
Figure 8.14: Heat Transfer Fractions: Heavyweight
Figure 8.15: Heat Transfer Fractions: Heavyweight
Figure 8.16: Condensation Rate Comparison: Forced Fan vs Displacement Ventilation
8.4. Chilled Beam Performance

Figure 8.17: Energy Performance: Lightweight
8.4. Chilled Beam Performance

Figure 8.18: Energy Performance: Lightweight
Figure 8.19: Energy Performance: Heavyweight
8.4. Chilled Beam Performance

Figure 8.20: Energy Performance: Heavyweight
As for the natural ventilation and the forced fan optimized chilled beam strategy the running cost of the chiller is less for heavy weight construction than for light weight, indicating that the dampening of the diurnal temperature swing by using increased thermal mass can reduce thermal plant running costs.

8.5 Displacement Ventilation Summary

The effect of enclosure thermal weight and external heat disturbances upon a zone with a displacement ventilation strategy has been analysed. Control algorithms have been used to approximate "real" displacement ventilation strategies to provide a realistic internal environment for the assessment of the chilled beam.

Chilled beam operation strategies have been discussed and an optimization has been performed. It was noted that, for all test days, a period of "cool" comfort conditions during early occupancy was required to prevent afternoon overheating, with the chilled beam control valve open even during periods of negative PMV values, a condition that would not occur in traditional control strategies where cooling strategies respond only to a definite need for cooling, (positive PMV). Additionally, the optimized results were shown to be highly sensitive, with control valve closure on occupancy existing only on days where lower values of positive PMV occurred during occupancy, (low risk of afternoon overheating). Again, as for all the mixed mode strategies, it should be noted that the inputs for the comfort algorithm are based on the single zone temperature node and can only provide an "averaged" indication of thermal comfort.

Heavyweight building structure was shown to dampen the internal temperature swing, which dampened the response of the PMV value and, (the thermal comfort within the zone being the objective function of the chilled beam optimization), reduced the valve control action which in turn reduces the running cost of the beam.
Chapter 9

Comparitive Performance

Introduction

Full HVAC simulation analysis investigates both the comfort and energy performance of a traditional air conditioning strategy. Comparison of the performance of active, passive and full HVAC mixed mode strategies and the role of a sub-dew point chilled beam within these strategies will be analysed, with the full HVAC performance acting as a "benchmark" to clearly identify the relative performance of each mixed mode strategy.

9.1 Full HVAC Ventilation Strategy

For a building utilizing full air conditioning, the zone thermal comfort conditions are dependant on the extent to which the ventilation control strategy responds to the prevailing climatic conditions. Full air conditioning maintains a predetermined zone condition setpoint within a band of tolerance dictated by the control strategy. The ventilation mass flow rate, (unlike natural ventilation), is not dependant upon temperature or pressure gradients, but the controlled fan speed, governed by the heat disturbances in the zone. In this investigation the ventilation strategy will be dictated by the control algorithms which set the temperature and moisture content in the zone. No optimization of the full air conditioning ventilation strategy will be performed as it is considered that the control strategy will provide the "correct" controlled conditions.
The effect of the full air conditioning ventilation strategy upon internal conditions and building fabric storage, for different thermal weights of building, will be investigated.

### 9.1.1 Full HVAC Ventilation: Enclosure Response

Like the displacement ventilation strategy, the full air conditioning strategy will utilize conventional control algorithms, (not optimization algorithms), to determine the ventilation strategy. The ventilation control strategy is operational throughout each test day to provide close control that is equivalent to the minimum load conditions, (for the chilled beam), given by the previous mixed mode strategies. The full air conditioning ventilation strategy being to control the zone air temperature and humidity to set-points of 20 °C and 50 % relative humidity within a proportional band of 4 °C and 20 % relative humidity respectively. This is achieved using sequential control of a heating coil, cooling coil, mixing box, and steam humidifier, (Chapter 4).

Figure 9.1 shows that the zone conditions set by the control algorithms are met by the full air conditioning ventilation strategy in all of the test days, (section 5.6). The zone temperature stay within the proportional band of 4 °C around the desired set-point in all but the "hot" day where the afternoon over-heating causes the temperature to rise above the upper limit of the proportional band. The zone relative humidity stays within the proportional band of 20 % relative humidity around the desired set-point. Comparison between these internal conditions and the ones for displacement ventilation, (Figure 8.2), show that for all the test days, the full air conditioning strategy reduces the amplitude of the diurnal temperature swing, (full air conditioning zone temperature having an amplitude of \( \approx 4 °C \), (the proportional band), and displacement ventilation zone temperature having an amplitude of \( \approx 10 °C \)). For the zone moisture content, the full air conditioning strategy produces moisture contents which are lower than the displacement ventilation strategy. To achieve the zone temperature set point some degree of cooling is necessary, this will result in dehumidification upon the cooling coil, and a reduction in zone air moisture content, provided the humidity doesn't fall below the lower limit of the proportional band.

Figure 9.2 shows the ceiling mass and surface temperature of the zone within the controlled full air conditioning strategy. As with the all the previous mixed mode ventilation strategies, the surface temperature has the fastest response time and
9.1. Full HVAC Ventilation Strategy

Figure 9.1: Full Air Conditioning: Zone Conditions
9.1. Full HVAC Ventilation Strategy

matches the diurnal changes in the zone air temperature, with the mass temperature lagging.

An interesting point to note is that, a comparison between the mass and surface temperatures of the full air conditioning and displacement ventilation strategies for heavyweight construction, (Figures 9.2 and 8.3), shows that the mass temperature response for both cases is similar with the full air conditioning strategy having only very slightly lower temperatures. Whereas, the surface temperatures of the full air conditioning strategy have diurnal temperature profiles which closely resemble the zone air response, (i.e. a reduction in the amplitude of the diurnal profile of the surface temperature). This indicates that the slow response time of the heavyweight structure dominates over the heat disturbance caused by the ventilation strategy, which is an expected result. As for the previous studies, a comparison of heavy and light weight structures showed increased thermal mass reduced the amplitude of the zone air, surface and mass temperature.

Figure 9.3, shows the diurnal heat flux profiles for heat transfer through the enclosure floor, for the four considered weather days, for the full air conditioning strategy. Comparison of the heat flux between the full air conditioning and the displacement ventilation strategy, (Figure 8.5), shows that both daytime storage and night time release are greater for the displacement ventilation. The internal zone air temperature in the full air conditioning strategy has a smaller diurnal amplitude, this reduces the potential for heat flow to and away from the building fabric, reducing the daytime storage and night time release. As for the all the previous mixed mode ventilation strategies, all the considered weather days show that in summer conditions building fabric elements will store heat during the day and release heat at night, (Figure 9.4).

Figure 9.5 shows the supply air, zone air temperature differential and the PMV profiles for all the test days with a full air conditioning ventilation strategy. Negative values of temperature differential indicate a potential for cooling, positive values indicate a potential for heating, and that cooling or heating respectively is taking place. It is clear, for all the test days, that the full air conditioning ventilation strategy provides cooling during the occupied periods, a heating strategy exists over night. These diurnal profiles form the driving potentials for the zone.

The effect of the driving potential created by the full air conditioning strategy upon the PMV profiles is twofold. Firstly, the PMV profiles show a marked reduction in diurnal amplitude in comparison to all the previous mixed mode strategies, and
Figure 9.2: Full Air Conditioning: Building Fabric Response
9.1. Full HVAC Ventilation Strategy

Figure 9.3: Full Air Conditioning: Building Fabric Heat Flux
Figure 9.4: Full Air Conditioning: Building Thermal Storage
Figure 9.5: Full Air Conditioning: Driving Potential
secondly, the PMV profiles are much closer to thermal neutrality, \( \text{PMV} \approx 0.0 \), than all the previous mixed mode strategies.

Figure 9.6 shows the effect of thermal mass upon the PMV profiles for the full air conditioning ventilation strategy. The dotted line represents the lightweight building fabric and the solid line the heavyweight fabric, (section 5.2). It is clear from the profiles that unlike all the previous mixed mode strategies, where the increase of thermal weight reduces the amplitude of the predicted mean vote values, the change in thermal weight has very little effect upon the PMV profiles. This is due to the close control of the zone air temperature by the full air conditioning strategy which by reacting faster than the thermal mass to heat disturbances effectively "masks" the effect of thermal mass upon the PMV value.

### 9.1.2 Full Air Conditioning Summary

It has been shown that in all but the highest internal heat load conditions the full air conditioning strategy can control the internal air conditions to a pre-determined set-point within a proportional band. The full air conditioning strategy reduces the diurnal amplitude of the zone conditions, and as a result reduces the amplitude of the diurnal PMV value. Additionally, the full air conditioning strategy produces PMV profiles that are closer to thermal neutrality than any of the previous ventilation strategies throughout the whole day. Also, it was shown that the response time of the full air conditioning strategy "masked" the effects of thermal weight upon the thermal comfort in the zone.

It should be noted that as thermal comfort analysis is performed upon a single temperature node, localised phenomena of air conditioning systems which effect thermal comfort, such as "dumping" beneath supply terminals, or analysis of occupant location in the zone have not been investigated.

### 9.2 Performance Comparison

Performance analysis of each mixed mode strategy is performed by investigating three performance indicators, namely, enclosure comfort performance, energy performance and cost. Enclosure comfort performance is a cumulative sum of the percentage predicted dissatisfied, \( \text{PPD} \), integrated over the occupied period,
9.2. Performance Comparison

Figure 9.6: Full Air Conditioning: Thermal Weight Comparison
9.2. Performance Comparison (08:00 - 18:00), (Chapter 5). Energy performance is a cumulative sum of the total energy use for each mixed mode strategy throughout the considered test day. Cost performance is a cumulative sum of the total cost of primary energy throughout the considered test day for each mixed mode strategy based on the tariff structure described in Chapter 5.

To enable performance comparison the performance indicators are normalised in the range 0 to 1, 0 being the lowest, (worst), performance and 1 being the greatest, (best), performance. The performance indicators are evaluated for all seven mixed mode approaches; natural ventilation, natural ventilation plus chilled beam, forced fan ventilation, forced fan ventilation plus chilled beam, displacement ventilation, displacement ventilation plus chilled beam, and full air conditioning.

The data from which the normalised performance indicators was taken is reproduced in Tables 9.2.1, 9.2.2 and 9.2.3, which represent the enclosure comfort performance, cumulative cost, and cumulative energy use of each mixed mode strategy for each test day for both light and heavy weight building construction. Table 9.2.4 represents the cumulative cost and energy usage of the chilled beam alone in each of the mixed mode strategies for each test day and for both light and heavyweight building construction.

9.2.1 Comfort Performance

Figure 9.7 shows the normalised comfort performance for all seven mixed mode strategies for each test day. It is clear that in all test days that full air conditioning gives the greatest comfort performance. For the “average” and “large diurnal variation” day the displacement ventilation only strategy gives the worst comfort performance. For the “hot” day, natural ventilation gives the worst performance, whereas forced fan ventilation only (and with chilled beam, chilled beam being off in this strategy), gives the worst performance for the “transitional” day.

The poor performance of the displacement ventilation in the “average” and “large diurnal variation” test days is due to the supply air temperature set point. Air is supplied at a constant 18 °C throughout the day, being initially warmer than the zone temperature this acts to increase the temperature in the zone and therefore increase the warm discomfort value of the PPD. For the “hot” and “transitional” day the displacement ventilation has better performance, for the “hot” day this is because the supply air temperature is lower than the zone temperature and cools
9.2. Performance Comparison

Table 9.1: Cumulative Comfort Performance Indicators: Heavy and Lightweight

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<th>Forced Fan</th>
<th>Displacement</th>
<th>Full HVAC</th>
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<td>+ Beam</td>
<td>+ Beam</td>
<td></td>
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<td>314.78</td>
<td>226.24</td>
<td>43.92</td>
</tr>
<tr>
<td>Large Diurnal'</td>
<td>71.34</td>
<td>44.94</td>
<td>115.25</td>
<td>36.36</td>
</tr>
<tr>
<td>Transitional'</td>
<td>60.14</td>
<td>47.69</td>
<td>51.71</td>
<td>38.26</td>
</tr>
</tbody>
</table>

Table 9.1: Cumulative Comfort Performance Indicators: Heavy and Lightweight

the space increasing comfort. For the “transitional” day the supply temperature is, (initially), higher than the “too cool” zone conditions which acts to improve the comfort conditions. The poor performance of the forced fan ventilation strategy in the “transitional” day is due to the insensitivity of the optimized fan profile having low fan flow rates, (instead of zero flow rate), which increase the already “too cool” discomfort conditions.

In all test days and for each of the mixed mode strategies, (except for “transitional” forced fan ventilation where the optimized profile indicated an “Off” position), it is clear that the addition of the chilled beam always improves the comfort performance. This performance improvement is mainly associated with the reduction of afternoon overheating. A point of interest is that only the “hot” day shows an incremental increase of comfort performance with increasing reliance on mechanical ventilation, showing that for situations other that extreme overheating, ("hot" days), good comfort performance can be achieved using means other than mechanical.

9.2.2 Energy Performance

Figure 9.8 shows the normalised energy performance for all seven mixed mode strategies for each test day. It is clear that, with the exception of the “hot” day, (where displacement ventilation plus chilled beam has the lowest energy per-
Figure 9.7: Heavyweight Normalised Comfort Performance
Table 9.2: Cumulative Energy Performance Indicators: Heavy and Lightweight

performance due to the increased cooling demand of the “hot” day), the full air conditioning has the lowest energy performance, i.e. uses the greatest amount of energy. For all the test days natural ventilation has the best energy performance.

In all the test days the inclusion of the chilled beam to the mixed mode strategy decreases the energy performance, and that in general energy performance falls as the use of mechanical plant increases. The introduction of the chilled beam dramatically increases the energy usage of the mixed mode strategy, i.e. the chiller associated to the chilled beam is, (relative to the other mechanical plant), energy intensive. This is due to the necessity to chill the cooling medium below the saturation temperature of the room to promote condensation mass transfer, and the latent heat transfer addition. An interesting point to note is that the addition of the chilled beam actually reduces the energy usage of the displacement ventilation strategy, (the chilled beam lowers the zone temperature and as a result improves the energy performance of the displacement ventilation (Table 9.2.2)), however the overall energy usage increases when the beam energy use is included, (Figure 9.8).
9.2. Performance Comparison

Figure 9.8: Heavyweight Normalised Energy Performance
9.2. Performance Comparison

<table>
<thead>
<tr>
<th>Test Day</th>
<th>Natural Ventilation + Beam</th>
<th>Forced Fan + Beam</th>
<th>Displacement + Beam</th>
<th>Full HVAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>'Average'</td>
<td>-</td>
<td>0.14</td>
<td>0.45</td>
<td>0.44</td>
</tr>
<tr>
<td>'Hot'</td>
<td>-</td>
<td>0.27</td>
<td>2.51</td>
<td>1.49</td>
</tr>
<tr>
<td>'Large Diurnal'</td>
<td>-</td>
<td>0.27</td>
<td>0.52</td>
<td>0.51</td>
</tr>
<tr>
<td>'Transitional'</td>
<td>-</td>
<td>0.11</td>
<td>0.27</td>
<td>0.26</td>
</tr>
</tbody>
</table>

Lightweight

<table>
<thead>
<tr>
<th>Test Day</th>
<th>Natural Ventilation + Beam</th>
<th>Forced Fan + Beam</th>
<th>Displacement + Beam</th>
<th>Full HVAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>'Average'</td>
<td>-</td>
<td>0.18</td>
<td>0.46</td>
<td>0.45</td>
</tr>
<tr>
<td>'Hot'</td>
<td>-</td>
<td>0.27</td>
<td>2.54</td>
<td>1.53</td>
</tr>
<tr>
<td>'Large Diurnal'</td>
<td>-</td>
<td>0.29</td>
<td>0.53</td>
<td>0.52</td>
</tr>
<tr>
<td>'Transitional'</td>
<td>-</td>
<td>0.11</td>
<td>0.27</td>
<td>0.27</td>
</tr>
</tbody>
</table>

Table 9.3: Cumulative Cost Performance Indicators: Heavy and Lightweight

9.2.3 Cost Performance

Figure 9.9 shows the normalised cost performance for all seven mixed mode strategies for each test day. It is clear that in all the test days that natural ventilation only has the greatest cost performance. Except the “transitional” day, where the full air conditioning has the worst cost performance, the worst cost performance is given by the displacement ventilation plus chilled beam mixed mode strategy.

In all the test days and for each of the mixed mode strategies the inclusion of the chilled beam reduces the cost performance, and the cost performance falls as the use of mechanical means of cooling increases. Cost performance falls dramatically as the chilled beam is introduced, this is because of the disproportionate amount of energy that a chiller plant requires. A breakdown of the costs incurred by each part of the mechanical plant, Figure 9.10, and with reference to the cumulative cost of the chilled beam, Figure 8.17, it is clear that the energy use, (and therefore cost), of cooling plant is much higher than heating, the fan and the humidifier. Also, close inspection shows that the cost to cool the chilled water is much higher than that to cool the air stream in the air conditioning system.

A point of interest is the difference between the energy and cost performance on the “average” and “large diurnal variation” day, in that full air conditioning has the greatest energy use, (lowest energy performance), but the displacement ventilation and chilled beam strategy has the greatest cost, (lowest cost performance). This is due to the “two tier” tariff structure of electrical energy use, the cost of electrical
9.2. Performance Comparison

energy being lower in the period 0:00 to 07:00. Analysis of the operating positions of the valve for the optimized displacement ventilation strategy, (Figure 8.11), shows that beam operation increases during the occupied period, (08:00 to 18:00), when the electrical tariff is highest, which will decrease the cost performance.
9.2. Performance Comparison

Figure 9.10: Full HVAC Mechanical Plant Cost Breakdown
9.2. Performance Comparison

### Chilled Beam Energy and Cost Performance Indicators

<table>
<thead>
<tr>
<th>Test Day</th>
<th>Natural Ventilation</th>
<th>Forced Fan</th>
<th>Displacement</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cost</td>
<td>Energy</td>
<td>Cost</td>
</tr>
<tr>
<td>'Average'</td>
<td>1.87</td>
<td>285.21</td>
<td>2.27</td>
</tr>
<tr>
<td>'Hot'</td>
<td>5.80</td>
<td>796.77</td>
<td>7.37</td>
</tr>
<tr>
<td>'Large Diurnal'</td>
<td>2.37</td>
<td>320.15</td>
<td>2.76</td>
</tr>
<tr>
<td>'Transitional'</td>
<td>2.18</td>
<td>298.76</td>
<td>1.00</td>
</tr>
</tbody>
</table>

### Lightweight

<table>
<thead>
<tr>
<th>Test Day</th>
<th>Cost</th>
<th>Energy</th>
<th>Cost</th>
<th>Energy</th>
<th>Cost</th>
<th>Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>'Average'</td>
<td>1.91</td>
<td>289.52</td>
<td>1.99</td>
<td>250.33</td>
<td>3.73</td>
<td>575.94</td>
</tr>
<tr>
<td>'Hot'</td>
<td>5.80</td>
<td>796.27</td>
<td>7.41</td>
<td>1026.6</td>
<td>7.01</td>
<td>1019.5</td>
</tr>
<tr>
<td>'Large Diurnal'</td>
<td>2.40</td>
<td>322.95</td>
<td>3.09</td>
<td>411.83</td>
<td>4.10</td>
<td>587.24</td>
</tr>
<tr>
<td>'Transitional'</td>
<td>2.20</td>
<td>300.51</td>
<td>0.17</td>
<td>21.48</td>
<td>3.24</td>
<td>470.39</td>
</tr>
</tbody>
</table>

Table 9.4: Chilled Beam Energy and Cost Performance Indicators

9.2.4 Thermal Weight

Figure 9.11 shows the cost, energy and comfort performance for light weight, (lightly shaded bars), and heavyweight, (darker shaded bars), building construction, (section 5.2), for all seven mixed mode strategies on the “average” day. It is clear that, with the exception here of the energy performance for full air conditioning, (where the control algorithms “mask” the effect of thermal weight, section 8.1.2), that better performance is given by the heavyweight building construction. This is due entirely to the dampening of the zone air temperature swing as described previously. The performance indicators for all the tests days mirror that of the “average” day.

9.2.5 Composite Performance: Discussions

Figure 9.12 shows the composite performance, (comfort performance + cost performance), of each mixed mode strategy for each test day. As cost and energy performance indicators essentially show the same performance, (cost being a function of the energy use), it was decided that overall performance would be assessed as using cost or energy performance and comfort performance. Cost and comfort performance are taken to have equal importance in the analysis and composite performance is a sum of the normalised performance indicators. Typically in industry, cost would be the primary concern, comfort being secondary.
9.2. Performance Comparison

Figure 9.11: Performance Comparison: Thermal Weight
9.3. Summary

It is clear that in all the mixed mode strategies the composite performance falls with the inclusion of the sub-dew point chilled beam. Even though the chilled beam improves zone thermal comfort it does this with a cost and energy use penalty. For the "average" and "large diurnal variation" day greatest composite performance is given by the forced fan ventilation strategy closely followed by natural ventilation. Good composite performance is found here due to the low energy usage of these systems and a medium comfort performance rating.

For the "hot" day the full air conditioning strategy gives the best composite performance, with natural ventilation, forced fan and displacement ventilation sharing very similar mid-range levels of composite performance. Hence, for the "hot" day if internal conditions did not have to be kept within a too stringent range, the composite performance indicates that good performance can be achieved with a less energy intensive strategy.

For the "transitional" day the greatest composite performance is given by natural ventilation only. Given that the "cool" ambient conditions of the "transitional" day lends itself closer to a heating rather than a cooling strategy, the displacement ventilation strategy, with its constant supply conditions, also performs well.

With the exception of the "hot" day, (where the energy usage of the chilled beam invalidates its use), both the natural ventilation plus chilled beam and forced fan plus chilled beam show high composite performance in comparison to these systems without the chilled beam, indicating a possible retrofit role for sub-dew point chilled beams.

9.3 Summary

The "benchmark" full air conditioning system has been investigated and the effect of enclosure thermal weight and external heat disturbances upon the zone has been analysed. Control algorithms have been used to approximate "real" air conditioning ventilation strategies to provide accurate comfort, cost and energy performance datums for the comparison of all the mixed mode strategies.

It was shown that in all but the highest internal heat load conditions the full air conditioning strategy can control the internal air conditions to a pre-determined set-point within a proportional band, and as a result reduces the amplitude of the diurnal PMV value. Additionally, the full air conditioning strategy produces PMV
Figure 9.12: Mixed Mode Composite (Comfort + Cost) Performance
profiles that are closer to thermal neutrality than any of the previous ventilation strategies throughout the whole day. Also, it was shown that the response time of the full air conditioning strategy "masked" the effects of thermal weight upon the thermal comfort in the zone.

The comparative performance of all the mixed mode strategies for comfort, cost and energy performance has been investigated. It was found that in all cases the sub-dew point chilled beam improves comfort performance, and, in conjunction with displacement ventilation reduces the energy use and cost of mechanical plant. However, in all cases the cumulative energy use and cost of the mixed mode strategy plus the chilled beam was much higher, and that the energy and cost performance fell. In the very hottest conditions overall performance increases as the use of mechanical means of cooling increases, but it was shown that the overall performance falls with the inclusion of the chilled beam, invalidating its use in high internal load conditions.

It was shown throughout the test days that if strict levels of comfort or internal conditions are not a prerequisite of the zone, good overall performance in comparison to full air conditioning can be achieved by natural and forced fan ventilation strategies, but at a fraction of the energy cost.

A retrofit application within natural and forced fan ventilation strategies which are failing to meet internal comfort conditions was shown to be the most likely role for the chilled beam. The composite performance, (cost + comfort), of these strategies with the chilled beam, was shown to be of a similar value to the strategies without the chilled beam.
Chapter 10

Conclusions and Further Work

10.1 Conclusions

This thesis has described the analysis of the performance and integration of a sub-dew point chilled beam within a range of mixed mode strategies. The mixed mode strategies considered were, natural ventilation, forced fan ventilation and displacement ventilation. Optimization of the mixed mode strategies was carried out to provide minimum load conditions for the sub-dew point chilled beam and to prevent any ambiguities in the analysis. The operation of the sub-dew point chilled within the low load conditions of all the mixed mode strategies was optimized, and the energy, cost, and comfort performance investigated. Analysis of the energy, cost and comfort performance of a full air conditioning system was also performed. Comparative performance analysis of all the mixed mode strategies to establish the role and performance of the sub-dew point chilled beam has been investigated, with the traditional full air conditioning acting as a "benchmark".

The sub-dew point chilled beam design and operation encouraged the (otherwise) unwanted production of condensation and included a latent heat transfer addition to the sensible heat transfer. Hence, performance analysis relied upon the investigation of both the temperature and moisture response of a zone. Thermal and moisture models described the zone air, fabric and chilled beam response to external heat disturbances, ventilation and plant models described the response of the mixed mode strategies. The models describing the zone and beam response in each mixed mode strategy must be dynamic, accurate, easily integrated, and robust in allowing the examination of different thermal mass.
The approach to the thermal modelling of the zone and the sub-dew point chilled beam is described in Chapter 2. Both the zone and the chilled beam are modelled using electrical analogy, (a well established thermal modelling strategy), which ensured easy integration of the separate models, an essential prerequisite of the modelling technique. The zone was split into 4 “massive” building elements, the floor, ceiling, external and internal walls, resistors representing thermal resistance and capacitors representing the thermal storage capacity of each building element. Air conditions were represented by a single node and radiant and convective heat flux paths were treated separately. The sub-dew point chilled beam model used a heat exchanger analogy and a sensible heat ratio correction to model latent heat transfer, (condensation mass transfer associated with the latent heat transfer was modelled in Chapter 3). The heat exchange between the zone and the chilled beam can be modelled. The integrated zone and sub-dew point chilled beam model parameters can be easily defined from real material properties, which enabled the investigation of thermal mass. Three thermal mass weights, (light, medium, and heavy), were initially considered. It was shown that for the material properties chosen, medium and heavy weight structures showed little difference in thermal response and thus the rest of the investigation focused on light and heavy weight constructions.

The validation and robustness of the integrated zone and chilled beam model have also been described in Chapter 2. A four level test for model validation was introduced and instigated. Throughout the thesis the derived models were subject to the same validation procedure. This included a theoretical test which established, through extensive literature reviews, the appropriateness of the model techniques chosen, and known response testing upon the chosen models which established the robustness and accuracy of the chosen models. An important aspect of the validation was that the model produced a “characteristic” behaviour of the zone, which was shown. Absolute accuracy of the model was considered less critical for this study as performance analysis was evaluated as a comparison between each mixed mode strategy.

The approach to the moisture modelling is described in Chapter 3. The moisture model is a linear differential equation which, using discrete time steps, enabled the linearization of the highly non-linear moisture transport properties. The model described the interaction of the moisture transport phenomena and the resultant response of the indoor air humidity. The linearized moisture transport properties considered within the latent model were: moisture absorption/desorption, chilled
beam surface condensation, infiltration, exfiltration, evaporation, generation, and natural and mechanical ventilation. The integration of the thermal and moisture model along with the validation and robustness of the moisture model have also been described in Chapter 3. The moisture model was shown to be robust and was easily integrated with the thermal model and was considered an appropriate choice for this project.

The approach to the ventilation and thermal plant modelling is described in Chapter 4. Models of natural ventilation, air infiltration and models of the thermal plant associated with the enclosure were described, these included a chiller plant model that describes the energy consumption of the chilled beam, and a model of a HVAC mechanical plant, plus the control strategies for the HVAC mechanical plant. Using a typical HVAC scheme, the mechanical plant simulation described steady state numerical models of a mixing box, heating coil, cooling coil, steam humidifier, fan and associated ductwork. The mechanical plant models were solved using a successive approximation algorithm. The chiller plant associated with the chilled beam utilized a black box numerical model approach using curve fits of manufacturers data. The control strategies and the HVAC component models were shown to be robust and were validated against expected performance, thus were considered an appropriate choice for this project.

The integration of the thermal, moisture, ventilation and thermal plant models into a composite model and the cost and comfort performance indicators are described in Chapter 5. The structure of the composite model is described, the output of which is used to evaluate the enclosure comfort performance which forms the objective function of the optimization. The enclosure comfort performance, a function of the percentage persons dissatisfied along with the optimization routine, the Complex method, is described. The objective function of the Complex method was found from the enclosure comfort performance calculated during the occupied period, (08:00 to 18:00). The Complex method was shown to be robust and, (except for insensitive periods of operation), produced optimised operation profiles which satisfied the objective function. Therefore the complex method was considered to be an appropriate choice for this project.

Three different summer ambient conditions and one transitional day were chosen for the performance analysis, an average UK summer day, a hot summer day, and a day with a large diurnal temperature variation. The hot summer day was chosen to investigate the performance for a potentially overheated building, whereas the day having a large diurnal temperature variation was selected to investigate the
impact of ventilation night cooling. The transitional day was chosen to investigate
the influence of chilled beam use in moderate climatic conditions. Additionally, the
tariff structure of primary energy cost, (both gas and electricity), was discussed.

Performance analysis was split into four main areas; passive analysis, (natural
ventilation), active analysis, (forced fan and displacement ventilation), benchmark
analysis, (full air conditioning), and performance analysis. Optimized strategies
of chilled beam operation were found using optimized strategies, (for natural and
forced fan ventilation), and control strategies, (for displacement ventilation), which
provided minimum load conditions. The effect of enclosure thermal weight, ex-
ternal heat disturbances and ventilation flow rates upon the sensible and latent
fluxes in the zone have been clearly identified and analysed for all the test days.
Optimized strategies and control algorithms have, in general, produced expected
results.

There are a number of specific discussions that should be noted for each of the
mixed mode strategies:

Natural Ventilation

The natural ventilation mixed mode strategy used window opening position as
the parameters for the optimization, the ventilation rate was determined by inter-
nal/external temperature and pressure gradients. It was shown that the optimized
ventilation strategy could be largely replicated by a simple ruled based approach,
and that the sensitivity of the optimization increases with the difference in the
ambient and zone air temperatures, (not only did a large temperature increase the
potential for cooling (or heating), it also increased the buoyancy driven ventilation
rate). The effect of a minimum ventilation rate and the optimized operation of
the sub-dew point chilled beam were also investigated. The investigation can be
summarised as:

- the optimization method clearly identified the optimum natural ventilation
  characteristic but for some insensitive periods of ventilation, the solutions
  may only be near optimal;
- with the exception of the “transitional” day, the role of a minimum ven-
  tilation requirement as a constraint in the natural ventilation optimization
  improved the comfort performance of the natural ventilation strategy during
  periods of insensitivity;
Conclusions 247

- without exception increasing the thermal mass reduced the amplitude of the zone diurnal air, mass and surface temperature swing, this for the test days that were sinusoidal around thermal neutrality, ("average", "large diurnal" and "transitional"), improved thermal comfort, and the zone comfort performance. For the "hot" day increased thermal mass improved the thermal comfort in the occupied period, and due to the nature of the objective function, (integration during occupied periods only), improved the overall comfort performance;

- in order to suppress afternoon overheating, the chilled beam operation optimization produced a period of "cool" comfort conditions during early occupancy, a more conventional control strategy, in which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling), would produce a different optimum;

- the zone comfort performance is based upon a single temperature node, and as such it is not possible to analyse location specific thermal comfort. In a zone with a fully opened window an occupant positioned centrally in the zone may experience thermally neutral conditions, whereas an occupant beside the window may experience "too cool" conditions based on the localised air speed, a one node approximation can not show this;

- the evaluation of energy and comfort performance of the chilled beam for "at and below" the room saturation temperature showed that the "sub-saturation" case gave a higher cooling capacity that improved the comfort performance of the zone, however, this led to an increase in both the energy usage and cost and a reduction in the energy performance;

- the running cost of the chiller associated to the chilled beam was found to be less for heavy weight construction than for light weight, indicating that the dampening of the diurnal temperature swing by using increased thermal mass can reduce thermal plant running costs.

Forced Fan Ventilation

The forced fan ventilation mixed mode strategy used fan operating position as the parameters for the optimization, the ventilation rate determined by the speed of the fan. As for natural ventilation, it was shown that the optimized ventilation strategy could be largely replicated by a simple ruled based approach. The forced
fan strategy optimization showed greater sensitivity than the natural ventilation strategy. This was because the enclosure comfort performance depended not only upon the temperature of the zone but also the relative air speed within it, and due to the controllable nature of the fan, (no longer dependant upon temperature or pressure gradients like natural ventilation), higher sustainable flow rates were available. The investigation can be summarised as:

- the optimization method clearly identified the optimum ventilation characteristic, forced fan ventilation optimizations having increased sensitivity due to the combined cooling effect of temperature potentials and higher air supply mass flow rates;

- as for natural ventilation, the inputs for the comfort algorithm are based on the single zone temperature node. Both temperature and air flow rates in the zone will change with location, (for example, beneath the chilled beam it is likely that a cold downward plume of air will exist whose temperature and flow rate will change in relation to the proximity to the beam), hence the one node approximation only provides an "averaged" indication of thermal comfort;

- without exception increasing the thermal mass reduced the amplitude of the zone diurnal air, mass and surface temperature swing, and as for natural ventilation, improved the overall comfort performance;

- for certain test days, unlike natural ventilation strategies where a period of "cool" comfort conditions during early occupancy was required to prevent afternoon overheating, the forced fan ventilation optimiums provided "closer to thermal optimum conditions" and zone pre-cooling using the chilled beam was found to be un-necessary in all but one case;

- the forced fan optimization was shown to have, (in comparison to natural ventilation), an increased latent heat transfer contribution. This was shown to be a function of the high air supply rates which gave zone conditions closer to ambient and consequently decreased the total heat transfer to the beam. The condensation mass transfer rates were similar which increased the proportion of the latent heat transfer;

- as for natural ventilation, the running cost of the chiller was found to be less for heavy weight construction than for light weight.
Displacement Ventilation

The displacement ventilation mixed mode strategy used control algorithms to maintain a controlled constant supply air mass flow rate at constant temperature. The investigation can be summarised as:

- the control algorithms kept the supply air flow rate and temperature within \( \approx 3\% \) of the desired set-points;
- with low flow rates and supply temperature displacement ventilation ensured conditions for the “hot” day that vastly improved the comfort in comparison with both natural and forced fan ventilation;
- as for natural ventilation and forced fan ventilation increased thermal mass improved comfort conditions in the zone;
- again, as for the natural ventilation, for all the test days, in order to suppress the afternoon overheating, the chilled beam optimization produced a period of “cool” comfort conditions during early occupancy;
- the optimization of chilled beam operation was shown to be highly sensitive to the air temperature response, (valve closure at occupancy only occurred for the test days when afternoon over-heating was less of a problem, (“average” and “transitional” day), and more definite valve operating position response for the larger internal temperature swing typified in light weight construction);
- as for natural and forced fan ventilation, increased thermal weight reduced the running cost of the chiller associated with the chilled beam.
- the one node approximation of the air conditions in the space cannot model the interaction of the air flow of the displacement ventilation with occupants, or the sub dew point chilled beam. The single air node assume fully mixed zone air conditions, (this is not typical of displacement ventilation where temperature stratification occurs), and as such the analysis of comfort performance is based upon an approximation of displacement ventilation, (a constant low supply air mass flow rate and a low supply air temperature).
10.1. Conclusions

Full Air Conditioning

Like the displacement ventilation strategy, the full air conditioning strategy utilized conventional control algorithms, (not optimization algorithms), to determine the ventilation strategy. The full air conditioning ventilation strategy controlled the zone air temperature and humidity to set-points of 20 °C and 50 % relative humidity within a proportional band of 4 °C and 20 % relative humidity respectively. The investigation can be summarised as:

- the close control of the internal zone conditions markedly reduced the diurnal amplitude of the internal temperatures and as a result the diurnal amplitude of the PMV is also reduced;

- the close control of the internal zone conditions produced PMV profiles which were much closer to thermal neutrality than any of the other mixed mode strategies, hence the enclosure comfort performance was much higher;

- the close control of the zone air temperature "masked" the effect of thermal mass upon thermal comfort shown in the other mixed mode strategies;

- localised phenomena of air conditioning systems which effect thermal comfort, such as "dumping" beneath supply terminals, or analysis of occupant location in the zone have not been investigated.

Analysis of the performance and integration of the sub-dew point chilled beam was performed by investigating enclosure comfort performance, energy performance and cost of each mixed mode strategy with and without the chilled beam plus the "benchmark" full air conditioning strategy. Enclosure comfort performance is a cumulative sum of the percentage persons dissatisfied, (PPD), integrated over the occupied period, (08:00 - 18:00). Energy performance is a cumulative sum of the total energy use for each mixed mode strategy throughout the considered test day. Cost performance is a cumulative sum of the total cost of primary energy throughout the considered test day for each mixed mode strategy based on the tariff structure described in Chapter 5. The performance indicators were normalised to enable comparison.

The analysis of comfort performance showed that wherever the chilled beam operation optimization indicated an "on" position, the zone comfort conditions in all of the mixed mode strategies improved. The increased comfort performance
was shown to be mainly due to the reduction in afternoon overheating, which was achieved in some test cases by producing a period of "cool" conditions at the start of occupancy. It was also shown that, with the exception of extreme overheating, ("hot" day), good comfort performance can be achieved without using mechanical ventilation.

The analysis of the energy performance showed that whenever the chilled beam was integrated into a mixed mode strategy the energy performance fell, and that in general energy performance fell as the use of mechanical plant increased. It was shown that the reduction in energy performance was due to the chiller plant associated with the chilled beam being energy intensive, (relative to the other mechanical plant). It was shown that the energy use associated with cooling was far greater than any of the other thermal plant. The analysis of the cost performance showed that whenever the chilled beam was integrated into a mixed mode strategy the cost performance fell, and that in general cost performance fell as the use of mechanical plant increased.

Analysis of the effect of thermal weight upon comfort, cost and energy performance showed that in all but the full air conditioning strategy, (where the close control of the internal conditions "masked" the effect of thermal mass), better performance, (cost, comfort and energy), was given by the heavy weight building construction.

Overall performance of chilled beam integration was given by the composite performance indicator, (normalised cost + normalised comfort performance). Analysis of the overall performance showed that whenever the chilled beam was integrated into a mixed mode strategy the composite performance fell. With the exception of the "hot" day, (where the energy usage of the chilled beam invalidated its use), both the natural ventilation plus chilled beam and forced fan plus chilled beam show high composite performance in comparison to these systems without the chilled beam, indicating a possible retrofit role for sub-dew point chilled beams.

In summary there are a number of specific conclusions that can be drawn:

- A composite thermal, moisture and mechanical plant model that can accurately model the heat exchange, (both sensible and latent), between the sub-dew point chilled beam surface and the zone is of prime importance for the investigation of the performance and integration of the chilled beam within mixed mode strategies; model robustness is also essential to allow the optimization of each mixed mode strategy, the optimization of chilled beam
operation within each mixed mode strategy and the investigation of thermal weight; the chosen models were shown to be robust and were considered an appropriate choice for this project.

- the use of zone comfort performance as the objective function in the optimization of chilled beam operation tended to produce a period of "cool" comfort conditions during early occupancy, a more conventional control strategy, in which the chilled beam would only be operated if the PMV was positive, (indicating a definite need for cooling), would have produced a different optimum;

- the zone comfort performance is based upon a single temperature node, and as such it is not possible to analyse location specific thermal comfort. In a zone with high internal loads and natural ventilation an occupant positioned centrally in the zone may experience high levels of "too warm" thermal discomfort, whereas an occupant beside the window may experience "too cool" conditions based on the localised air speed, a one node approximation can not show this;

- if strict levels of comfort or internal conditions are not a prerequisite of the zone, good overall performance in comparison to full air conditioning can be achieved by natural and forced fan ventilation strategies, but at a fraction of the energy cost;

- the integration of a sub-dew point chilled beam into a mixed mode strategy always improves the comfort performance, however it always reduces the cost and energy performance; integration of the sub-dew point chilled beam into a zone with a high internal load would dramatically reduce the overall performance of the zone;

- a retrofit application within natural and forced fan ventilation strategies which are failing to meet internal comfort conditions was shown to be the most likely role for the chilled beam, (chilled beam integration improving the comfort performance without greatly reducing the overall performance).

10.2 Suggestions for Further Work

This thesis has shown that the likely role for sub-dew point chilled beams is a retrofit application within naturally and forced fan ventilated buildings which have
summer overheating problems. Further work should address the specific comfort issues associated with the chilled beam, namely analysis of the convective heat transfer caused by the negatively buoyant plumes which would form beneath the chilled beam and analysis should focus on the radiant exchange between the chilled beam surface and occupants. Occupant comfort should be analysed as a function of position in the zone, relative to the chilled beam. In this way the limitations and errors created by the use of the one node approximation of zone temperature can be investigated.

Additionally, further work should assess the geometry of the chilled beam design to improve the mechanisms of heat transfer. The chilled beam was approximated as a horizontal cylinder, this should be extend to include multiple horizontal cylinders, vertical cylinders, vertical plates and chilled beams with extended surfaces which enhance convective heat transfer. Also, empirical data showing sub-dew point chilled beam performance would enhance the knowledge of the physical mechanisms of heat and mass transfer enabling actual comfort assessment which would be used to further validate the composite model, and validate the use of sub-dew point chilled beams in office environments.

Further work is still necessary to evaluate the role of sub-dew point chilled beams. All the above recommendations can be used to extend the work in this thesis. Further research will result in an enhanced understanding of the performance, integration and role of sub-dew point chilled beams.
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Appendix A

Comparison of Radiant Network Method to Radiant Star Temperature Method

A.1 Introduction

Heat gain to the chilled beam is achieved by three mechanisms; convection, condensation, and radiation. Each of these mechanisms can be approximated using heat transfer equations. Convection and condensation are governed by first order linear equations. To calculate the effect of radiation the fourth power of the surface temperature has to be known. M.G.Davies[44] developed a linearized method to approximate the difference of the fourth power of temperature, the radiant star temperature. The radiant star temperature has already been shown to fit into lumped capacitance models well[32]. An established method, the radiant network method, is compared to the radiant star temperature method.

A.2 Radiant Network Method

The radiant network method analyses the radiant exchange between all surfaces within an enclosure. The analysis of the enclosure is complicated by the fact that surfaces in the enclosure are not black bodies, radiation leaving a surface can be reflected around all the surfaces in the enclosure. Partial absorption can occur
at any of these reflections. Analysis should try to take into account the multiple reflections. The network radiation model assumes the following[30].

- Radiative properties (reflectivity, emissivity and absorptivity) are uniform and independent of direction and frequency.
- The surfaces are diffuse emitters and diffuse reflectors.
- The radiative heat flux leaving the surface is uniform over the whole of the surface.
- The irradiation is uniform over the whole of the surface.
- Either a uniform temperature or a uniform heat flux is prescribed over the whole of the surface.
- The enclosure is filled with a non-participating medium.

By performing radiative flux balances at the surfaces of the enclosure and between the considered surfaces, the following equation set can be found:

\[
Q_i = \frac{E_{bi} - J_i}{(1 - e_i)/A_i} = \Sigma \frac{\Delta J}{(A_iF_{i-j})^{-1}}
\]  

(A.1)

\(Q_i\) is the heat flux to surface \(i\). \(E_{bi}\) is the emissive power of surface \(i\). \(J_i\) is the radiosity of surface \(i\). \(F_{i-j}\) is the view factor from surface \(i\) to surface \(j\). \(e_i\) is the emissivity of surface \(i\). \(A_i\) is the surface area of surface \(i\). The emissive power \(E_{bi}\) is given by:

\[E_{bi} = e_i\sigma T_s^4\]  

(A.2)

\(T_s\) is the absolute temperature of the surface, \(\sigma\) is the Stefan Boltzmann constant.

The equation set can be reduced to:

\[
Q_i = \frac{E_{bi} - J_i}{R_s} = \Sigma \frac{\Delta J}{R_{bs}}
\]  

(A.3)

\(R_s\) is the radiation resistance of the surface and \(R_{bs}\) is the radiative resistance of between all the surfaces. This is shown graphically as Figure A.1.
Equation A.3 has to be set up for each surface; ceiling, floor and the chilled beam. The wall are treated as a black body and given the equation:

\[ J_i = \sigma T_i^4 \]  

(A.4)

This equation and the equation for emissive power, equation A.2, are dependant upon the absolute surface temperature raised to the fourth power. The equation set A.3 when produced for all the surfaces can be solved using gaussian elimination. This then gives the surface temperature and the change in radiative flux for the surface.

### A.3 Radiant Star Index Temperature

Where the network method of radiant exchange considers the interaction of radiant fluxes between surfaces, the radiant star temperature method considers the fluxes incident upon a fictitious construct, the radiant star temperature. The radiant star index temperature is given by all surfaces in an enclosure as[44]:

\[ \text{[44]} \]
Figure A.2: Radiant Star Network

\[
T_r = \frac{K_i T_i + K_j T_j + K_k T_k}{K_i + K_j + K_k} \quad (A.5)
\]

\(T_{i,j,k}\) are the surface temperatures. \(T_{rs}\) is the radiant star node. \(K\) is the radiant conductance between the surfaces at \(T_{i,j,k}\) and the radiant star node \(T_{rs}\), and can be represented as in Figure A.2.

The radiant conductance is given by:

\[
K_{ri} = A_i E h_r \quad (A.6)
\]

\(h_r\) is the linearised radiation thermal conductivity. \(A_i\) is the surface area of the considered surface. \(E\) is an area weighted emissivity factor given by[44]:

\[
\frac{1}{E_i} = \frac{1 - e_i}{e_i} + Y_i \quad (A.7)
\]

\(e_i\) is the emissivity of the surface being considered, \(Y_i\) is shown in the thermal network chapter.
A.3.1 Linearisation of Temperature to the Fourth Power

Temperature difference is the driving potential for conduction through a material, this is also the case for surface convection. The driving potential for radiation is; $\sigma(T_i^4 - T_j^4)$, a fourth power temperature difference. Linearizing this difference enables the use of the radiant star construct. Consider a radiant heat balance between a surface $i$ and the radiant star construct:

$$Q_i = A_i (J_i - J_{rs})$$  \hspace{1cm} (A.8)

Hence;

$$Q_i = A_i \sigma(T_i^4 - T_{rs}^4)$$  \hspace{1cm} (A.9)

Introducing $h_{rj}$, the linearized heat transfer coefficient:

$$Q_i = A_i h_{rj}(T_j - T_{rs})$$  \hspace{1cm} (A.10)

where:

$$h_r = \sigma(T_{js} + T_{rs})(T_i^2 + T_{rs}^2)$$  \hspace{1cm} (A.11)

Some enclosure temperatures will be above $T_{rs}$ and some below and it is sufficient to take an average value. If $T_{rs}$ is taken as 20 degree Celsius, $h_r$ is approximately $5.7 \, W/m^2K$.

A.4 Numerical Comparison

Direct numerical comparison between radiant star and network radiation method is necessary to see how applicable radiant star temperature is to the application. The network radiation method was programmed in 'C' code and the value of heat gain to the chilled beam was found for different combinations of surface temperatures in the enclosure. The radiant star method was also analysed using the same surface temperatures. A direct manual comparison of the resultant fluxes was then taken.
The percentage difference between the results of both methods was low, between 3 and 15 percent, depending upon the surface temperature. The accuracy of the radiant star method was greater when surfaces were given temperatures that were close in value. Accuracy reduced when the surfaces had greater differences in temperature. However, given that the differences between the two methods was small, the radiant star method was considered applicable for the project.
Appendix B

Reynolds Analogy - $\mathcal{T}$ derivation

B.1 Introduction

The derivation of the convective mass transfer coefficient, $\mathcal{T}$, analogises free turbulent convective heat and mass transfer. Provided the conditions of the Reynolds Hypothesis are met the convective mass transfer coefficient is given by the ratio of the convective heat transfer coefficient and the specific heat capacity.

B.2 Reynolds Hypothesis

The Reynolds Hypothesis gives the conditions when the Reynolds Analogy is appropriate:

- Convective transport of the mass flux is independent of direction.
- Magnitude of mass transfer conductance, $\mathcal{T}$, not dependant upon concentration gradients or chemical reaction.

These conditions are met for convective heat and mass transfer when:

- Mass transfer rates are low.
- Ratio of Prandtl and Schmidt numbers is unity.
These conditions apply for condensation and evaporation in ambient room conditions.

### B.3 Reynolds Analogy

The controlling equation for turbulent convective heat transfer over a flat surface is given as [71], Figure B.1:

\[
\frac{u}{\partial x} + v \frac{\partial \bar{\theta}}{\partial y} = \frac{\partial}{\partial y} \left[ (\alpha + \epsilon_H) \frac{\partial \bar{\theta}}{\partial y} \right] \tag{B.1}
\]

The left hand term in the equation represents convection, the right thermal diffusion. \( \bar{\theta} \) is the mean boundary layer temperature, \( \alpha \) is the thermal diffusivity, \( \epsilon_H \) is the turbulent diffusion coefficient for heat transfer. \( u \) is the boundary layer horizontal component of velocity, \( v \) is the boundary layer vertical component of velocity.

Assuming that the vertical velocity component is small in comparison to the hor-
horizontal velocity component, and that the temperature gradient in the x-direction is much smaller than in the y-direction, ie:

\[
v \approx 0 \quad \text{and} \quad v \ll u
\]

\[
\frac{\partial \bar{v}}{\partial x} \approx 0 \quad \text{and} \quad \frac{\partial \bar{v}}{\partial x} \ll \frac{\partial \bar{v}}{\partial y}
\]  

(B.2)

Then integrating between \( Y = 0 \) and \( Y = \delta \), noting that the temperature gradient at \( \delta \) is zero, yields:

\[
q'' = -\rho C_p (\alpha + \epsilon_H) \frac{\partial \bar{T}}{\partial Y}
\]  

(B.3)

Similarly the controlling equation for turbulent convective mass transfer over a flat plate is given as:

\[
u \frac{\partial \bar{b}}{\partial X} + v \frac{\partial \bar{b}}{\partial Y} = \frac{\partial}{\partial Y} \left[ (D + \epsilon_M) \frac{\partial \bar{b}}{\partial Y} \right]
\]  

(B.4)

Again the left hand term in the equation represents convection, the right hand side represents mass diffusion. \( D \) is a mass diffusion coefficient, \( \epsilon_M \) is a turbulent diffusion coefficient for mass transfer. \( \bar{b} \) is given as \((m_y - m_s)/(m_s - m_T)\), where \( m_s \) and \( m_T \) are standard states, constant in any integration. \( m_y \) is \( m \) at any point \( Y \) in the concentration boundary layer.

As for heat transfer, Equation B.4 reduces to:

\[
\dot{m}'' = -\rho (D + \epsilon_M) \frac{\partial \bar{T}}{\partial Y}
\]  

(B.5)

In turbulent flow conditions heat and mass transfer is controlled by turbulent diffusion, the thermal and mass diffusivity coefficients are considered negligible, \( \epsilon_H \gg \alpha \) and \( \epsilon_M \gg D \). Dividing Equation B.3 by B.5 and assuming \( \epsilon_H \approx \epsilon_M \) gives:

\[
\frac{q''}{\dot{m}''} = C_p \frac{\partial \bar{T}}{\partial \bar{b}}
\]  

(B.6)

Integrating the right hand side of the equation yields:
\[
\frac{\dot{q}''}{\dot{m}''} = \frac{C_p [\bar{\theta}_G - \bar{\theta}_s]}{(m_G-m_x) - (m_x-m_T)} \tag{B.7}
\]

But:

\[
\frac{\dot{q}''}{[\bar{\theta}_G - \bar{\theta}_s]} = h_c \tag{B.8}
\]

and

\[
\frac{\dot{m}''}{(m_G-m_x) - (m_x-m_T)} = \Upsilon \tag{B.9}
\]

Hence:

\[
\Upsilon = \frac{h_c}{C_p} \tag{B.10}
\]
Appendix C

Publications To Date

Publications associated with this thesis to date are: