The Effect of Ventilation Strategies on Thermal Comfort in a Non-Domestic Passivhaus Building in the UK

Abstract

This study has used computational fluid dynamics (CFD) supported by field measurements to investigate the relationship between the zonal supply air strategies (natural ventilation (NV), mechanical ventilation (MV) and mechanical ventilation with heat recovery (MVHR)) and thermal comfort in the largest non-domestic certified Passivhaus building in the UK. The results show that, depending on the selected thermal comfort standard and internal heat gains, a thermally comfortable environment could be achieved by supplying air in a range of 9°C – 26°C for MVHR, versus 17°C – 29°C for NV. The findings indicate that a mixed mode approach, using auxiliary heating and cooling, may be necessary for optimal year-round thermal comfort and energy efficiency in non-domestic Passivhaus buildings in the UK climate.

Introduction

In response to anthropogenic climate change many countries are tightening their building regulations and introducing advanced building performance standards (Feist et al., 2005; Liu and Linden, 2006; McLeod and Hopfe, 2013). In 2017, non-domestic building energy consumption accounted for 14% of the UK's total final energy consumption (BEIS, 2017). Whilst predictions across the European Union (EU) stock indicate that heating, ventilation, and air conditioning (HVAC) related energy consumption might rise to around 50% within 15 years, due to the increase in air-conditioned spaces (Lombard et al., 2008). Two ostensibly different design concepts, Passivhaus and natural ventilation (NV) offer the potential to significantly reduce HVAC related energy demand and greenhouse gas (GHG) emissions from buildings.

Recent studies in the domestic Passivhaus sector (Macintosh & Steemer 2005; Phan et al. 2008; Sassi 2013; McLeod et al., 2013; Perez and Østergaard, 2014) have shown that in milder climates Passivhaus buildings are often vulnerable to poor ventilation design and overheating. Sub-optimal MVHR design and installation (McLeod and Swainson, 2017) combined with a lack of user knowledge and guidance in relation to the operation of such units (Wang et al., 2017) are often cited as reasons why such systems fail to work as intended. When these shortcomings are combined with the capital and maintenance costs of running an MVHR system, the use of NV appears to be a strong candidate to either replace or augment MVHR in non-domestic Passivhaus buildings, particularly in temperate and warm climates.

Although the usage of NV and MVHR has been well documented in relation to domestic Passivhaus buildings; there is limited research in the non-domestic building context, especially in the UK climate. This issue was highlighted by Wang et al. (2017), who stated that further research into the thermal stratification of buoyancy driven ventilation systems for Passivhaus buildings in the UK climatic context is needed. A robust understanding of indoor thermal environment performance of NV, MV and MVHR (including mixed-mode options) is essential to making informed decisions at the early design stages.

In this study, the indoor thermal environment performance of NV, MV and MVHR will be investigated under various outdoor temperature scenarios. The aim of the work is to establish the limiting operating temperatures (i.e. the outdoor (NV) or supply-air temperature (MV)) that are required to provide a thermally comfortable environment under any given ventilation regime. Limiting temperature ranges, needed to achieve acceptable thermal comfort, will be determined for each ventilation scenario, from which the derived ventilation temperatures will be compared to the seasonal distribution of UK outdoor temperatures, in order to establish the likely acceptability of the various protocols in practice.

Although focused on a single case study, this research aims to highlight the benefits of a methodology that has practical benefits for stakeholders wishing to make informed ventilation design decisions at the early stages of non-domestic Passivhaus projects.

Methods

A combination of advanced numerical simulation methods and field measurements were used for the ventilation system modelling and thermal comfort analysis in this research. The spatial temperature distribution in the case study building's auditorium was predicted using CFD software. The CFD predictions were subsequently validated in comparison to empirical temperature measurements taken in the auditorium. A series of different scenarios were then simulated using the validated CFD model to investigate the thermal comfort levels resultant from a variety of different ventilation strategies including NV, MV, and MVHR. This analysis included investigation of a range of design parameters including: different fresh air supply rates (for MV), opening sizes and positions (for NV), quantity of supply and exhaust terminals/openings (for MV and NV), occupancy densities, and supply air temperatures. From this parametric analysis it was possible to understand the UK outdoor air temperature ranges under which it would be possible to provide a thermally comfortable environment, using the least carbon intensive strategy. This analysis was, carried out in accordance with the respective national and international thermal comfort standards ASHRAE Standard 55 (ASHRAE, 2017), Building Bulletin 101 (BB 101, 2006), and CIBSE Guide A (CIBSE, 2016).

The Case Study Building

The case study building chosen for this research was unique at the time of this study, for being the largest certified non-domestic Passivhaus Building in the UK. The auditorium space (Figure 1) was chosen for investigation due to the complex ventilation design challenges posed by its low ceiling and high internal heat gains conditions. These conditions presented considerable difficulties for appropriate ventilation design solutions due to temperature stratification within the domain. In spaces characterised by high internal heat gains and low ceilings, warm stale air may drop down into the breathing zone, causing a high temperature gradient (>3°C) within the occupied zone which can cause localised thermal discomfort. High temperature gradients, within the occupied space, and poor indoor air quality (IAQ) were noted in similar spaces (lecture auditorium, cinema and a theatre) utilising displacement ventilation, by Mathisen (1989). Similarly, thermal comfort and energy efficiency problems were reported by Kavgic et al., (2008) in a mechanically ventilated theatre. Such studies highlight the importance of investigating the thermal performance of ventilation systems within such domains in the Passivhaus context.

Figure 1: Auditorium Space, The George Davies Centre, Medical School Building, at Leicester University.

Field Measurements

The modelled domain space was 16.8m in length, 14.5m in depth and 8.15m in height. The auditorium had a typical ventilation layout with the fresh air being supplied through a plenum before being distributed via low-level inlet grilles located in the risers below the seats. After passing through the occupied zone, buoyancy-driven stale air passed through ceiling panel perforations and rectangular diffusers in the suspended ceiling, before finally being extracted via the exhaust terminal located above the suspended ceiling (Figure 2).

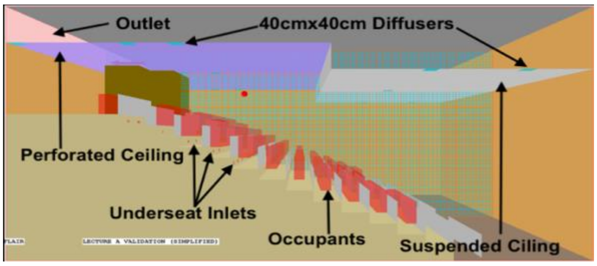

Figure 2: CFD Model Showing Perspective View of Principle Ventilation Components of the Auditorium

The vertical temperature distribution within the case study building's auditorium was predicted, using validated CFD software (PHOENICS, Cham Limited, 2017). In this study, the standard k-ε 2-equation turbulence model was used (Launder and Spalding, 1974) due to the model's broad range of applicability and minimal computational power requirement. Launder and Spalding (1974) demonstrated in their study that the k-ε model was very effective for modelling auditorium ventilation systems. Furthermore, Zhang and Chen (2006) effectively predicted airflow in a room with an underfloor air distribution system using the k-ε model. The success of various k-ε models was also demonstrated by Yang (2004) as a result of the investigation of the mean air flow rates through a naturally ventilated building. However, the standard k-ε model was found to be the most effective of the k-ε models for modelling turbulence in naturally ventilated spaces (Walsh and Leong, 2004).

A hexahedral mesh was used in this study, and it was manually modified to obtain good mesh quality, free of imperfections such as, skewed, squashed, and high aspect ratio cells. Furthermore, ventilation inlets and outlets were divided into at least four computational cells for a realistic representation of the air flow. The grid was manually modified to be sufficiently fine near the wall surfaces to solve boundary layer phenomena effectively whilst becoming coarser further away from the surfaces. The mesh (grid) independence study was performed by generating meshes with 2.13, 3.37, and 7.1 million computational cells. During this study the outlet flow rate was monitored for each simulation run, and it was concluded that all three mesh options yielded the same outlet flow rate (the variation being less than10^{-5}kg/s). Therefore, from a computational efficiency perspective, the coarse mesh was selected for the rest of the study. Moreover, the following convergence criteria were set for the numerical solutions: domain imbalances for energy, mass and momentum were less than 1%, residuals of transport equations were less than 10^{-4}, and the temperature at the monitoring point was checked for stability.

The modelled space had only one exposed external wall, and since the building was a Passivhaus building (using highly insulated walls to minimise conductive heat transfer), it is assumed (as a first approximation) that the indoor air temperature was equal to the wall surface temperatures. Hence, walls were modelled as adiabatic for model simplification. However, a heat flux towards the domain was added to some walls and surfaces (yielding radiative heat gains) to approximate the radiative heat gains from the occupants, lights and electronic devices (e.g. radiative heat gain component of lights attributed to desks due to the direct view factor between the lights and tables). Thus, the effect of radiative heat gains on individual components was crudely accounted for in the indoor air temperatures. In preparing the CFD model for validation, boundary conditions (BCs) derived from the field data, manufacturers specifications, the literature, and design guides were used where possible (Table 1). The fresh air supply rate and its temperature were not available for each inlet. Therefore, the total fresh air supply rate from the air handling unit (AHU) was sub-divided into the number of inlets. It was assumed that the fresh air temperature at the inlets was equal to the fresh air off-coil temperature at the AHU. These assumptions were justified because of the well-insulated supply ductwork.

Table 1: Boundary Conditions for Validation Simulation

	Internal Heat Gains (W)
Av. Occupant Heat Gains	84 (CIBSE Guide A, 2016)
Number of Occupants	220 (Site Inspection)
Each Luminaire	22 (Manufacturer)
Number of Luminaire	55 (Site Inspection)
Tablet for Each Occupant	3 (Manufacturer)
Audio & Visual Equipment	1000 (Manufacturer)
	Fresh Air Supply
Supply Temperature	18.3°C (BMS)
Supply Air Flow Rate	25.9l/s (FM)
	Inlets and Outlets
Exhaust Opening Dimensions	1.15m x 14.5m (FM)
Ceiling Diffuser Dimensions	40cm x 40cm (Manufacturer)
Ceiling Diffuser's Quadratic Loss Coefficient (QLC)	2.69 (Literature)
Perforated Ceiling QLC	26.9 (Assumption)
Exhaust Flow Rate	3.9m^3/s (BMS)
Number of Under-Seat Inlet	150 (Site Inspection)
Size of Under-Seat Inlet	11cm ×24cm (Site Inspection)

CFD Modelling for NV and MV Performance Assessment

In order to assess the performance limits of the NV and MV systems, 35 different ventilation scenarios were modelled. These scenarios were then simulated with different occupancy density, supply air (outdoor) temperatures, supply air flow rates (for MV), openings sizes and positions (for NV), and number of inlet and outlet terminals/openings (for MV and NV) (Table 2).

Table 2: Boundary Conditions Range of MV and NV CFD Simulations.

Ventilation Modes	NV	MV
# Occupants	220 & 324	220 & 324
Sup. Temp. Range [°C]	14 to 32	12-32
Supply Air Flow Rate [l.s^{-1}.person]*	8 to 31.2	8-17
# of Inlets/Outlets	7&13/1&3	75&150/1
Inlet/Outlet Total Effective Area [m^2]	11.2&20.7& 53.7/16.7&42.4	1.6&3.2/16.7
Ceiling Opening [%]	23&33&56&100	Fixed

* MV supply air flow rates are defined by the user as a BC, on the other hand, for NV, the supply rates are predicted by CFD.

Thermal Comfort Performance Criteria

The performance of the NV, MV and MVHR systems were benchmarked against three national and international thermal comfort performance indices. ASHRAE standard 55 (2017) which is referred to as *criterion one* in this study was chosen as it provides a robust standard for free running buildings by including the adaptive behaviour of occupants. The equation for 80% acceptability limit which is used in this study for evaluation of a ventilation case from thermal comfort perspective is given in ASHRAE Standard 55 (2017).

One of the UK national standards, Building Bulletin 101 (2006), which is widely used to define acceptable indoor environmental conditions in educational buildings, was defined as *criterion two* in this study and dictates the following:

1. The difference between average indoor and outdoor air temperature should not be higher than 5°C;
2. Indoor air temperature should not exceed 28°C for more than 120 hours in a year; and
3. When the space is occupied, the indoor air temperature should not exceed 32°C.

However, the second clause cannot be readily tested with the current computational limitations in CFD, since it would require transient analysis across an entire year using a specific weather file. Nevertheless, this is not a limitation since the standard dictates that only two of the above-mentioned criteria need to be fulfilled in order to achieve a successful design.

UK design standard, CIBSE Guide A (2016) which is identified as *criterion three* in this study, provided a similar approach to criterion one. Comfort temperature equations which are used in this study for evaluation of a ventilation case from the thermal comfort perspective are given in CIBSE Guide A (2016).

If the predicted spot temperatures generated by the CFD model, within the occupied zone, comply with the above standards, then ventilation case was deemed to "pass", and otherwise "fail". Due to the uncertainties inherent in the modelling process, it was decided to apply a tolerance factor of +/-0.5°C at the upper and lower temperatures boundaries of the thermal comfort criteria when assessing the CFD predicted temperatures. In addition to assessing the above criteria, evaluation of localised thermal discomfort was considered by examining the temperature gradient from ankles to head level when the occupants were in seated positions. In accordance with CIBSE Guide A, (2016) the vertical temperature gradient should not exceed 3°C in order to keep the Predicted Percentage Dissatisfied (PPD) index below 6% (BS EN ISO 7730, 2005).

Climate Analysis of the UK

A frequency distribution of UK outdoor air temperatures was created in order to determine which ventilation system is more suitable in the UK climate throughout different outdoor temperatures from a thermal comfort point of view. The International Weather for Energy Calculation (IWEC) weather files in Energy Plus database (Energyplus.net, 2017) were used to create the outdoor temperature frequency distribution of the UK. Five regions from the Energy Plus database were selected for a broad representation of the UK climate: London (Gatwick Airport), Birmingham Airport, Sheffield (Finningley), Edinburgh (Leuchars), and Aberdeen (Dyce Airport). Since this study intended to cover a wide range of operational schedules, the climate data was analysed for 8-hour and 24-hour periods. The climate data was sorted by 3ºC temperature bands so that a temperature frequency distribution could be determined for each location and operating schedule.

Results

Validation of the CFD Model

It is imperative to validate any CFD model, in order to ensure the reliability of the subsequent predictions. In this study the CFD predicted spatial domain temperatures were compared to field temperature measurement results (Figure 3).

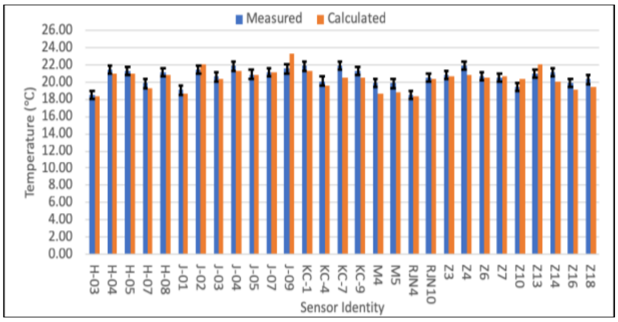

Figure 3: Measured data against CFD Predicted Spot Temperature Values

The relatively small discrepancy between the predicted and measured results can be explained by measurement uncertainties. The error bars shown on the measured values represent only the manufacturers stated instrument accuracy data, but not any other experimental related uncertainties. Some of the measured values tend to be slightly higher than the calculated values (Figure 3). The reason for that difference can be attributed to the radiative heat exchange between sensors and surrounding surfaces with higher temperatures. While measured values were fully affected by all of the radiative heat exchange occurring within their field of view, the predicted values were only partly accounted for this due to the simplified solution used for radiative heat exchange. Additionally, simplifications in certain assumptions, such as: the assumption of uniform occupancy distribution, the division of the total air flow rate equally to inlets, the assumption of supply air temperature measured at the AHU location being the same at supply inlet locations, and the assumption about the pressure loss coefficient assigned to the perforated ceiling are expected to contribute to the overall uncertainty. Moreover, any errors in geometric measurements taken in the field and inconsistencies between the geometry of the actual building and the architectural plans (which were used in creating the CFD geometry) also played a contributing role in these uncertainties. Likewise, these geometrical positioning errors meant that the locations of the objects affecting the airflow pattern and temperature distributions (e.g. air inlets, ceilings, and furniture) were similarly approximated.

NV and MV Thermal Comfort Performance Assessment in the UK Climate

By applying various BCs, the spatial temperature distributions for NV and MV systems were predicted using the validated CFD model and the results compared to the thermal comfort performance criteria to determine whether a ventilation scenario was viable or not from a thermal comfort perspective (Table 3). Additionally, the extended range of operating conditions (as °C in operating temperature) that a HR system would provide an acceptable MV scenario for the winter cases was calculated by using the conservation of energy principle at the heat recovery terminal, based on the MVHR operating at a stated efficiency of 75%.

The findings indicated that NV is capable of extending the summer operating temperatures from 3°C to 6°C as a result of providing larger fresh air flow rates compared to MV. Interestingly, the NV also extended the winter operating performance range by 3°C in accordance with thermal comfort criteria two and three. This was a result of NV's ability to match MV performance in providing the threshold requirement for fresh air (8l.s^{-1}.person). Compared to the NV mode, MV provided better performance in only two situations, this occurred in the 220 and 324 occupancy scenarios when benchmarked against criterion one. In these two cases, MV extended the winter operating temperatures by 3ºC. Moreover, when HR was added to the MV mode, the winter operating temperature were further extended from 3.4ºC to 8ºC compared to the NV. However, using MV (with or without the HR), occupants were more prone to local thermal discomfort compared to the NV due to the vertical temperature stratification exceeding 3ºC from ankle to head level in a seated position.

Table 3: Scenario based CFD Modelling Results.

Vent. Mode	Occ. #	Thermal Comfort Criterion	Operating Temperature Range Providing Comfortable Environment [ºC]	Presence of Local Thermal Discomfort
NV	220	1	20 to 29	No
MV	220	1	17 to 23	Yes, T=17 ºC
MV+HR	220	1	13.6 to 23	Yes, T=17 ºC
NV	220	2	20 to 29	No
MV	220	2	20 to 26	Yes, T=26 ºC

MV+HR	220	2	16.6 to 26	Yes, T=17 ºC and 26 ºC
NV	220	3	20 to 29	No
MV	220	3	20 to 26	Yes, T=26 ºC
MV+HR	220	3	16.6 to 26	Yes, T=17 ºC and 26 ºC
NV	324	1	17 to 26	No
MV	324	1	14 to 23	No
MV+HR	324	1	9 to 23	No
NV	324	2	17 to 29	No
MV	324	2	20 to 26	Yes, T=26 ºC
MV+HR	324	2	15 to 26	Yes, T=26 ºC
NV	324	3	17 to 29	No
MV	324	3	20 to23	No
MV+HR	324	3	15 to 23	No

In order to assess that performance in the context of the UK climate, outdoor operating temperatures (from Table 3) needed to be compared with a frequency distribution of the UK outdoor temperatures. As a result of this comparison, Table 4 tabulates the capability of NV, MV, MVHR, to provide a thermally comfortable environment for occupants (were the auditorium to be located in different cities) according to working schedules expressed as percentages of occupied hours. Table 4 indicates that; NV has the largest potential in warmer locations (London) and the least in colder (Edinburgh/Aberdeen) cities. This finding was expected since NV was found to extend summer operating temperatures and warmer outdoor temperature hours are more common in the South compared to the North. Furthermore, from the total column in table 4, it can be seen that NV performed poorly compared to MV in the performance *criterion one*, but it performed better in *criterion two and three*. Moreover, the MVHR performed better than the NV and MV scenarios. This was also expected considering that the MVHR extends winter operating temperatures significantly and the UK is a heating dominated country. It is also important to emphasise that, when NV and MVHR was used in conjunction, an optimal solution was found in all cases with resultant occupied comfort hours ranging between 8.4% (220 occupants, criterion three, Edinburgh, 24 hours schedule) - 82.3% (324 occupants, criterion one, London, 8 hours schedule).

Table 4: Scenario based CFD Modelling Results.

Vent. Mode / Occ. # / Per. Cri. #	**London**		**Birmingham**		**Sheffield**		**Edinburgh**		**Aberdeen**		**Total**	
	24h (%)	8h (%)	24h (%)	8h (%)	24h (%)	8h (%)	24h (%)	8h (%)	24h (%)	8h (%)	24h (%)	8h (%)
NV/220/1	7.5	15.1	5.2	10.4	5.7	4.4	1.6	3.7	1.9	4.6	21.9	38.1
MV/220/1	12.5	21.5	10.5	18.8	10.2	9.4	5.4	11.0	5.5	11.2	44.1	71.9
MVHR/220/1	28.2	38.6	26.5	37.7	23.6	23.3	19.5	30.4	17.8	27.1	115.5	157.1
NV+MVHR/220/1	35.7	53.7	31.7	48.1	29.3	27.7	21.0	34.1	19.7	31.6	-	-
NV/220/2	7.5	15.1	5.2	10.4	5.7	4.4	1.6	3.7	1.9	4.6	21.9	38.1
MV/220/2	6.4	12.9	4.2	8.8	4.6	3.7	1.2	2.8	1.6	3.8	18.0	31.9
MVHR/220/2	20.0	31.6	17.9	29.6	16.7	16.4	10.9	20.6	10.7	20.1	76.2	118.3
NV+MVHR/220/2	27.6	46.7	23.1	40.0	22.3	20.9	12.5	24.2	12.6	24.7	-	-
NV/220/3	7.5	15.1	5.2	10.4	5.7	4.4	1.6	3.7	1.9	4.6	21.9	38.1
MV/220/3	6.4	12.9	4.2	8.8	4.6	3.7	1.2	2.8	1.6	3.8	18.0	31.9
HR/220/3	15.4	26.3	12.8	22.5	12.5	11.5	6.8	13.5	6.9	13.6	54.5	87.5
NV+MVHR/220/3	22.9	41.3	18.0	32.9	18.2	16.0	8.4	17.2	8.8	18.2	-	-
NV/324/1	8.3	16.3	5.8	11.8	6.3	5.0	2.0	4.6	2.3	5.4	24.7	43.1
MV/324/1	25.3	35.9	23.5	34.8	21.2	20.9	16.6	27.1	15.4	24.7	102.0	143.5
MVHR/324/1	56.3	66.1	53.8	63.6	51.7	45.5	46.4	59.1	44.1	54.8	252.3	289.2
NV+MVHR/324/1	64.6	82.3	59.6	75.5	58.0	50.6	48.4	63.7	46.4	60.2	-	-
NV/324/2	13.9	24.6	11.2	20.0	11.2	9.9	5.5	11.1	5.6	11.4	47.4	77.0
MV/324/2	7.5	15.1	5.2	10.4	5.7	4.4	1.6	3.7	1.9	4.6	21.9	38.1
MVHR/324/2	20.6	32.9	18.2	30.2	17.1	16.6	10.9	20.6	10.7	20.2	77.5	120.6
NV+MVHR/324/2	34.5	57.5	29.4	50.2	28.3	26.6	16.4	31.7	16.3	31.6	-	-
NV/324/3	13.9	24.6	11.2	20.0	11.2	9.9	5.5	11.1	5.6	11.4	47.4	77.0
MV/324/3	4.9	9.8	3.5	7.2	3.6	3.0	1.0	2.4	1.4	3.3	14.3	25.7
MVHR/324/3	19.4	30.3	17.7	29.0	16.2	16.2	10.9	20.5	10.6	20.0	74.8	116.1
NV+MVHR/324/3	33.3	54.9	28.9	49.1	27.5	26.1	16.3	31.6	16.2	31.4	-	-

Temperature Stratification in NV and MV

A large temperature stratification is not desirable in a domain as it can cause the stale air to drop down to the breathing zone resulting in IAQ problems (Gilani et al., 2016). Additionally, it could also result in local thermal discomfort when the temperature gradient exceeds 3°C from ankle to head level (ISO 7730, 2005).

NV demonstrated less thermal stratification in the domain compared to MV, and this can be attributed to the provision of larger airflow rates. The NV strategy (Figure 4, Top) resulted in significantly less temperature stratification (<2°C) compared to the MV strategy (Figure 4, Bottom) which was greater than 3°C, under identical supply/outdoor air temperature boundary condition. Moreover, using NV it was possible to achieve a better

IAQ compared to the MV due to the increased fresh air flow rate (30l.s^{-1}.person in NV as opposed to 17l.s^{-1}.person with MV). Another consequence of the reduced flow rates associated with the MV system is that under warm conditions occupants in the centre of the zone experience localised temperatures that are several degrees higher than those experienced with NV, a factor which may place them at increased risk of heat stress (Figure 4, Bottom).

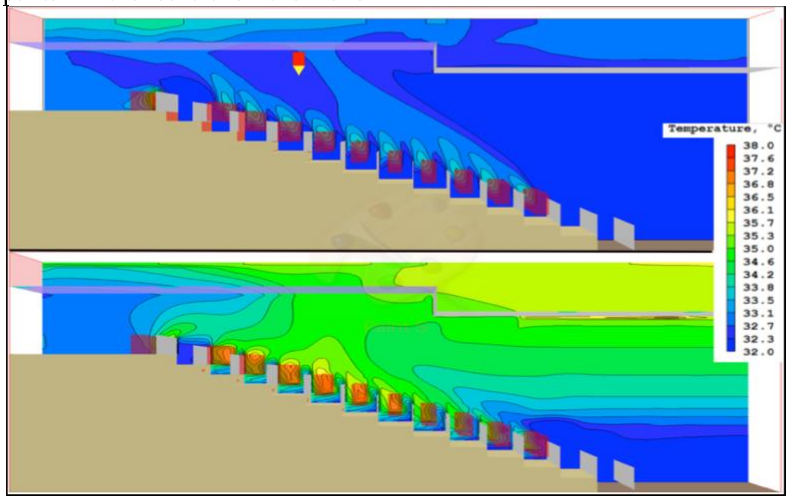

Figure 4: Vertical Temperature Distribution, 220 Occupants, 32°C Supply Air, NV (Top), vs. MV (Bottom)

Improving the NV and MV Performances by Design Interventions

The CFD model predictions have indicated the areas where the ventilation design could be improved both for MV and NV. These improvements could extend the NV and MV operating temperature limits at both the low-end (winter) and high-end (summer) of the outdoor temperature spectrum.

The case study building had a constant air supply AHU providing a volumetric flow rate of 3.9m^3/s. Nonetheless, the modelling results demonstrated that, if the fresh air supply rate could be reduced based on the occupancy number (i.e. demand controlled), it would be possible to extend the winter operating range of the MV. The findings showed that, the winter operating supply-temperature range could be extended from 17°C down to 15°C for the 220 person occupancy condition, if the fixed air supply rate of 17l.s^{-1}.person was reduced to 10l.s^{-1}.person (which is the minimum fresh air supply rate requirement according to CIBSE Guide A, 2016). This finding results from the smaller ventilation heat losses occurring from the domain, which consequentially increases the indoor temperatures. According to IWEC climate data, for each climatic region with an 8 hours working schedule this 2°C extension in the operating range would equate to an extension of between 310-340 operating hours per year in the UK climate.

Using NV, the fresh airflow rate was not as easy to control as MV because the buoyancy forces for NV were less stable than the momentum forces of the MV. However, the simulation results showed that it was possible to alter both the spatial domain temperatures and fresh air flow rates in NV mode by adjusting the inlet and outlet opening sizes and locations. The standard NV setup prior to this design intervention was composed of: three outlet openings above the suspended ceiling level (comprising of 11cm tall inlet openings at each step from one side of the room to the other) and two suspended ceilings (one completely covered with perforations and ceiling diffusers, and the other with ceiling diffusers only). When the supply air (outdoor) temperature was increased from 26ºC to 29ºC, the standard NV setup with 11cm inlets failed to comply with thermal comfort *criterion 1* because of the high internal temperatures, but satisfied *criteria 2* and *3*. However, when the height of each inlet opening was increased from 11cm to 35cm to give an opening area of 53.7m^2 compared to 20.7m^2, the overall temperatures in the domain were reduced, with the maximum temperature dropping from 32.3°C to 30.5°C. This temperature decrease resulted in compliance with all of the thermal comfort performance criteria. It is interesting to note that, the fresh air supply rate was reduced from 30 l/s^{-1}.person to 20.2l/s^{-1}.person when the inlet sizes were increased to 35cm from 11cm. This finding suggests that higher volumetric fresh air flow from the larger inlets cooled down the domain temperatures thereby reducing the buoyancy forces, which in turn resulted in a reduced fresh air flow rate into the domain.

Discussion and Limitations

NV and MV in Non-Domestic Passivhaus Buildings in the UK

This study has explored the operating temperature boundaries and implications of NV, MV and MVHR strategies individually and in combination, for a specific zone within a non-domestic Passivhaus building in the UK climatic context. However, the findings are extendable to other high-performance (i.e. energy efficient) buildings with similar BCs.

The findings suggest that none of the systems (individually or combined) would be able to provide a thermally comfortable environment all of the time, when considering the dynamic effects of occupancy and seasonality. This finding is interesting because the literature (Feist et al., 2005) states that internal and solar heat gains should be sufficient to overcome heating loads in Passivhaus buildings; thus, hydronic heating systems should not normally be necessary. However, both the simulation findings and site inspections (where a hydronic heating system is used during colder periods) concluded that it was not always possible to avoid the usage of a dedicated hydronic heating system even with internal heat gains as high as 124W/m^2 in the milder UK climate. In this context additional measures were required to provide a continuously comfortable thermal environment as dictated by the heating and cooling loads. Nonetheless the result show that, when intelligent NV design is used in conjunction with MVHR, it was possible to provide a thermally comfortable environment around 80% of the time in some cases (e.g. in London with an 8-hour working schedule). This could be considered as a significant finding where HVAC related energy usage in non-domestic buildings in such cities is high. However, it is essential to employ variable opening configurations driven by internal heat gains and outdoor air temperatures to achieve such a ventilation performance. Even though altering indoor spatial temperatures and air flow rates is commonly achieved by varying façade opening configurations, the research here has shown that it is also possible to achieve this result without major structural changes to the façade openings. This makes retrofit solutions a possibility without the requirement for major structural changes in the façade. It also means that improved hybrid NV/MV designs are possible in the context of complex non-domestic Passivhaus buildings, achieving improved localised occupant comfort whilst reduce reliance on mechanical systems.

This study concluded that NV provides a better indoor thermal climate compared to MV when they both operate at the same outdoor (supply) temperatures in the context of this non-domestic Passivhaus building. This is because of the increased ventilation rates of NV and the capability of buoyancy-driven air to remove more internal heat from the domain, hence reducing temperature stratification. The superior performance of NV compared to MV in this regard has been previously highlighted by a number of researchers including: Cook and Short (2005), Short and Lomas (2007) and Stephan et al., (2011). Clearly NV would be more energy efficient within its operating temperature range compared to MV because of the absence of fan power consumption. Moreover, NV would extend both the summer and winter operating temperature ranges from 3°C to 6°C compared to MV. However, NV cannot replace MV (in the context of a Passivhaus building) because of the lack of heat recovery capability. When combined with heat recovery capability, an MV system can extend winter operating temperature ranges from 2°C to 8°C compared to the NV. Additionally, Perez and Østergaard, (2014) have shown that the Passivhaus concept can be prone to overheating, and 90% of the time MVHR is not required in milder climates. This study concurs that non-domestic Passivhaus buildings are prone to overheating, when ventilated via MV, due to the pronounced temperature stratification occurring in the occupied zone. However, in relation to the use of MVHR, for thermal comfort and energetic reasons, the findings from this study agree with those of Lambea et al., (2016) which found that MVHR is an essential requirement for Passivhaus buildings unless the climate is consistently hot. Collectively these findings point to the benefits of incorporating seasonal mixed mode systems in non-domestic Passivhaus building; where NV is used in preference to MV whenever the outdoor temperature permits.

Limitations

The assumptions and simplifications in relation to the BCs used in this study will have influenced the accuracy of the validation model, and hence the magnitude of the findings. Factors such as the spatial occupancy distribution, and flow rates for each individual inlet were not precisely known. Additionally, the modelled geometry was not a perfect representation of the real domain due to the inconsistencies between the plans and the real building. Furthermore, geometrical simplifications were made by aligning modelled objects in the domain to avoid excessive mesh sizes and to improve the mesh quality. This was necessary in order to achieve a manageable mesh in terms of the mesh density and quality, in relation to the available computational power. Additionally, although typical of an auditorium, the modelled space lacked windows on the external façade, thus the findings would not be extendable to similar spaces with high solar heat gains. Finally, a more accurate representation of radiative heat exchange would be obtainable with the use of using dynamic thermal simulation (DTS) to predict surface temperatures, which could then be used as BCs in the CFD model.

Conclusions and Future Work

The following conclusions were drawn from this work:

1. Neither NV nor MVHR was always sufficient to provide thermally comfortable environments in this non-domestic Passivhaus building, in the UK climate. When used in conjunction, as a mixed-mode system, they could provide a comfortable environment between 8.4% – 82.4% of the annual occupied hours depending on the outdoor temperature, selected thermal comfort performance criterion and working schedule (8/24 hours). Thus, a mixed mode approach in ventilation (NV+MVHR) with an additional means of providing auxiliary cooling and heating, would represent the most appropriate design solution in this context.
2. NV showed a higher potential in warmer regions (i.e. London) compared to the cooler regions (North of the UK). Whilst conversely, the potential of MV was higher in the North and lower in the South. Moreover, NV provided more comfortable occupied hours compared to MV when benchmarked against *criterion 2* and *3* but less so in relation to *criterion 1*. Furthermore, the use of NV extended the summer operating temperature by 3°C to 6°C compared to MV, whilst the MVHR system extended the winter

operating temperature by 2°C to 8°C compared to the NV system.
3. Variable openings in size and location for NV and a variable (i.e. demand controlled) supply air flow rate for MV are required to obtain the best performance out of these systems.
4. When the fresh air is supplied at the same ambient air temperature the NV approach provides less thermal stratification within the domain than MV, and as a result NV produces less local thermal discomfort.

This study suggests that a mixed-mode strategy is likely to also provide significant energy savings and GHG emission reductions relative to the default practice of MV. Further work could include quantifying the potential energy and GHG emission savings across a range of Passivhaus building typologies and climatic contexts.

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