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Enhancing non-domestic Passivhaus auditoria ventilation design for improved indoor environmental quality

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ABSTRACT

Enhanced energy conservation strategies often involve tightly controlled ventilation flow rates. However, strategies that don't carefully consider ventilation rates can, in certain contexts, result in inadequate ventilation, with increased risks of poor indoor environmental quality and user acceptance. These design challenges are often exacerbated in non-domestic buildings with highly dynamic occupancy patterns. This study used computational fluid dynamics, supported by field measurements, to investigate the relationship between zonal supply air strategies and thermal comfort in the George Davies Centre, Leicester University, which is the largest non-domestic certified Passivhaus building in the UK. Ventilation strategies involving mechanical ventilation operating with heat recovery turned on and off, and natural ventilation systems were investigated in relation to their ability to maintain thermal comfort in an auditorium space characterised by high internal heat gains and tiered seating. The results show that, depending on the selected thermal comfort criterion, a thermally comfortable environment could be achieved when incoming air is in the range of 9–26 °C for mechanical ventilation with heat recovery and 17–29 °C for natural ventilation. These temperatures are referred to as 'limiting operating temperatures' in the paper. The work showed that in a temperate climate, thermal comfort could be maintained, for up to 80% of the year, using mixed mode ventilation, without space conditioning, in combination with intelligent design and control strategies. Operating in natural ventilation mode also provided increased fresh air supply capacity, a finding which is particularly relevant in the context of mitigating airborne viral transmission.

1. Introduction

In response to anthropogenic climate change, there is a growing trend towards countries tightening their building regulations and introducing advanced building performance standards [1,2]. In 2020, non-domestic building energy consumption accounted for 16% of the UK's total final energy consumption [3]. Predictions across the European Union (EU) stock indicate that HVAC related energy consumption in buildings is likely to rise by around 50% within 15 years. A substantial component of this increase is attributed to the growth in the use of air-conditioning [4]. The increasing risks of overheating in non-domestic buildings is well documented in Europe [5], the UK [6], China [7], United States (US) [8] and Latin America [9,10]; yet, there is a lack of literature on how to prevent overheating in highly energy efficient non-domestic buildings. Two ostensibly different low-energy design concepts: Passivhaus and natural ventilation (NV) offer the potential to significantly reduce HVAC related energy demand and green-house gas (GHG) emissions from non-domestic buildings. The Passivhaus concept has strict energy compliance limits [11] that can be achieved by employing a low heat loss building fabric, typically coupled with mechanical ventilation with heat recovery (MVHR) which aims to capture a high percentage of the waste heat from the ventilation system. In contrast, NV has been widely established as an effective means of providing a thermally comfortable environment in non-domestic buildings either when used in isolation [12] or in conjunction with mechanical ventilation (MV) [13]. However, NV solutions employ naturally occurring forces of wind and buoyancy (temperature difference) to supply fresh air ventilation but typically lack any feasible means of recovering waste heat.

Studies in the domestic Passivhaus sector have shown that in

Abbreviations: CFD, Computational Fluid Dynamics; nZEB, near Zero Energy Building; IEQ, Indoor Environmental Quality.

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temperate climates Passivhaus buildings may be vulnerable to poor ventilation design and overheating [14–19]. Sub-optimal MVHR design and installation [20] combined with a lack of user knowledge and guidance in relation to the operation of such units [21] 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 continuously running an MVHR system, as well as the net primary energy [22] and much greater embodied energy and floor space that such systems require [23] the use of NV appears to be a strong candidate to either replace or augment MVHR in many non-domestic Passivhaus buildings and near zero energy buildings (nZEB). This is particularly so in temperate and warm climates where the energetic and carbon saving benefits of using heat recovery ventilation systems are reduced [24] but also in educational buildings, and other buildings characterised by intermittent occupancy with relatively high occupant heat gains [25].

Moreover, the requirement for MVHR in different climatic conditions for domestic Passivhaus buildings has been questioned by several studies [18,26], which raises the question of whether further investigation of these issues is warranted in the non-domestic building context. To date there has been limited research in the context of non-domestic Passivhaus buildings, especially in the UK climate. The need for further work in this area was highlighted by Wang et al. [21], who concluded that research into the thermal stratification of buoyancy driven ventilation systems for Passivhaus buildings in the UK climatic context is required, due to the pronounced effect of thermal stratification on thermal comfort (which is unaccounted for in steady state energy models such as the Passive House Planing Package (PHPP)). Moreover, since non-domestic buildings have complex spatial and servicing requirements, it is often challenging and expensive to retrofit ventilation systems into those buildings. Thus, a robust understanding of the indoor environmental quality (IEQ) resulting from the use of NV, mechanical ventilation (MV) and MVHR (as well as mixed-mode operational options) is essential for making informed decisions at the early design stages.

Auditoria are considered to be onerous spaces from a design perspective, particularly in relation to satisfying their acoustic [27,28] and thermal comfort requirements [29]. A number of factors contribute to the challenge of achieving good IEQ in such spaces, including: the presence of significant thermal stratifications and cold draughts in the occupied zone, which occur because of intermittent and high-density occupancy conditions, variable ceiling heights and the typical displacement ventilation system design used in such spaces [30,31].

In the context of designing low energy auditoria, these challenges are arguably even more pronounced. Historically a number of studies [12, 32–34] have emphasised the benefits of using stack ventilation in naturally ventilated auditoria design. More recently Ahmad et al. [35] looked at ways to increase both energy efficiency as well as the IEQ in a mechanically ventilated auditorium with air conditioning using different (ceiling and floor level) air outlet positions and concluded that under-seat extraction led to improved thermal comfort. Cao et al. [36] investigated the thermal comfort of a NV auditorium space in a warm climatic region of China, using post-occupancy thermal sensation voting surveys, and concluded that occupant satisfaction levels around 70% could be achieved. Notably however, the study was based on surveys conducted during the autumn and the level of satisfaction might be expected to change during the warmer and cooler periods, which points to the need for longitudinal assessments of thermal comfort. In a study of auditoria using computational fluid dynamics (CFD), Wan et al. [31], observed a significant temperature stratification (up to 4 °C between ankle and head heights) due to the use of an underfloor displacement system. They concluded that the temperature stratification would be reduced significantly when the underfloor displacement system was supplemented with a ceiling mounted displacement system. However, the authors found that the flow rates of the new system had to be tightly controlled to achieve the thermal comfort targets without increasing the energy requirement. Despite such insights, a common problem with many of the existing studies on auditorium spaces is that their findings are difficult to generalise since they are often predicated on a specific design context and/or climate.

To the authors' knowledge there have been no studies on auditoria ventilation conducted in buildings designed to meet the stringent Passivhaus criteria. Such studies are urgently needed, in the transition to nZEBs, because the design constraints of Passivhaus and ultra-low energy auditoria (where there is often no, or little, auxiliary heating and cooling within the space) are likely to be significantly different to more conventional low-energy approaches. Moreover, whilst a number of studies have considered the issues of both natural and mechanical ventilation in isolation, there is only limited work investigating the possibility of combining these methods into a mixed-mode low-energy auditoria ventilation strategy. In the past decade, only a few studies were conducted on auditoria using mixed-mode ventilation. One of these studies, Thomas [37], conducted a post-occupancy thermal comfort survey in three buildings located in Australia and four buildings located in India. Within those buildings, only one of them (located in India) had an auditorium space in the building. Although the specific results were not presented space by space, the overall results showed that the buildings employing a mixed-mode approach achieved the best thermal comfort and productivity results in the surveys. However, the outdoor temperature range between warmer and colder seasons in those countries does not vary as greatly as the UK and other European countries, rendering their results less applicable to other dissimilar climates. Another post-occupancy study, Ali et al. [38], investigated the effect of using ceiling fans in conjunction with windows on thermal comfort in Nigerian auditoria spaces during the hot season. The results from the two auditorium spaces investigated showed that the use of ceiling fans in conjunction with opened windows improved the thermal conditions of the space to a certain extent. However, it could be argued that the suggested mixed-mode system (i.e. ceiling fans in conjunction with openable windows) might not be as effective as a displacement mixed-mode system capable of operating under a wider range of conditions, however alternative strategies were not tested. Finally, Ricciardi et al. [30], investigated the thermal comfort, in an historic auditorium space in Italy, using questionnaires along with temperature measurement, where the occupants were seated at different heights within the building. They concluded that while the building was mechanically ventilated and heated, the temperature difference at different heights in the occupied zone was up to 7 K causing 90% of the occupants in the higher level seats to be disstisfied according to thermal sensation votes due to warmer temperatures. However, when the building was operated in NV mode, the thermal stratification was found to be less prounounced (i.e around 1.5 K) resutling in increased thermal satisfaction. Given that this study was conducted in a building with a relatively high heat loss envelope, the thermal stratification might be even more pronounced in a building with a low heat loss envelope.

In light of some of the gaps in the knowledge identified above, this study investigated the indoor environmental performance of NV, MV and MVHR systems in a UK Passivhaus auditorium (section 2.1) under various outdoor temperature, occupant density and ventilation scenarios. Although the study was conducted prior to the SARS-CoV-2 pandemic, the implications of the findings in relation to the transmission of airborne pathogens are raised in the discussion section. The specific aim of the work was to establish external limiting operating temperatures (LOTs) for each type of ventilation system when used individually and in combination (i.e. a mixed-mode configuration), and to quantify the percentage of thermally comfortable occupied hours corresponding to those operating temperatures, for various UK climatic locations. In this study, the term 'LOTs' refers to the intake air temperatures (i.e. the external air temperature range) that would be required to maintain a thermally comfortable environment under any given ventilation regime, without the use of active heating and cooling technologies. The LOT ranges, needed to maintain acceptable thermal comfort, are determined for each ventilation scenario along with the

corresponding zonal boundary conditions. Subsequently, the LOTs derived by CFD are compared with the seasonal distribution of UK outdoor temperatures, in order to establish the likely acceptability of the various protocols in specific geographic contexts from a thermal comfort perspective. Part of this process involves the identification of overheating risks when outdoor tempertures exceed the LOTs and assessing the efficacy of potential mitigation strategies, as demonstrated by this study.

2. Methods

A combination of advanced numerical simulation methods and field measurements were used for modelling the ventilation system and thermal comfort analysis in this research. The spatial temperature distribution in the case study building's auditorium was predicted using CFD software, with realistic boundary conditions taken from field measurements. The CFD predictions were subsequently validated in comparison to in-situ temperature measurements recorded in the case-study auditorium. A series of different scenarios were then simulated (section 2.4) 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), different opening sizes and positions (for NV), quantity of supply and exhaust terminals/openings (for MV and NV), occupancy densities, and supply air temperatures (for MV and NV). From this parametric analysis it was possible to understand the LOT ranges for various combinations of these parameters. Subsequently, the LOTs were compared with the UK outdoor temperatures to derive the percentage of thermally comfortable occupied hours for specific geographic locations. This analysis was carried out in accordance with the respective national and international thermal comfort standards ASHRAE Standard 55 [39], Building Bulletin 101 [40], and CIBSE Guide A [41]. Following the CFD simulations, a number of design interventions were suggested for extending the LOTs of the ventilation systems, therefore increasing the energy efficiency of the space whilst providing a thermally comfortable environment.

2.1. The case study building

The case study building was the George Davies Centre (formerly known as the Centre for Medicine) at Leicester University, in the UK Midlands (Fig. 1, Left). This building was unique at the time of this study for being the largest non-domestic certified Passivhaus Building in the UK. The auditorium space (Fig. 1, Right) was chosen for investigation due to the complex ventilation design challenges posed by its tiered seating, transient occupancy, and high internal heat gains.

2.2. Experimental set-up and field measurements

In this study, the indoor air temperature distribution of the auditorium was monitored under occupied conditions, for a three-day period (which represented the typical occupant density of the space) using HOBO™ pendant temperature loggers. According to the manufacturer's specifications, the temperature loggers have an accuracy of ±0.5 K for the measurement of indoor air temperatures. Prior to installation, the accuracy of all temperature loggers was checked to confirm the manufacturer's data, using a temperature-controlled water bath in the Loughborough University laboratories. Following this calibration, twenty-nine HOBO™ pendant temperature loggers were installed at various positions within the domain. The layout of the space and the (x, y,z) coordinates of the sensors are given in Fig. 2 and Table 1 respectively. The locations of the loggers were chosen to capture a realistic three-dimensional representation of the spatial temperature distribution in the room. The loggers recorded the temperature data at a 10-min sampling frequency.

2.3. Computational fluid dynamics modelling for validation

In the MV mode, the air is supplied to the auditorium by a decentralised air handling unit (AHU) located in the services room of the building. The air supplied by the AHU fills the large plenum (beneath the tiered seating) and then the air passes through the under-seat inlets (located in the risers) before flowing into the occupied space. Then the air in the occupied space moves upwards and passes through the suspended ceiling (via 40 × 40 cm diffusers and perforated suspended ceiling panels) filling the suspended ceiling void above. Finally, the air in the suspended ceiling is extracted out of the room by the extract fan connected to the large outlet located at the AHU (Fig. 3). In NV mode, the air is induced to move through the space mainly by the buoyancy forces generated by the internal heat gains. The location of the under-seat inlets remains unchanged in the NV mode. The auditorium utilises a ventilation system which may (theoretically) be driven either by displacement MV or NV stack forces.

The modelling work in this study was conducted under steady state conditions. The field measurement results showed that the space achieved a steady state condition (i.e. stable indoor air temperatures) at around 50 min from the start of the first 2 h morning lecture (Appendix, Table A2) with stable ventilation parameters (i.e. ventilation flow rates and air temperatures) justifying the use of steady state boundary conditions in the modelling. Furthermore, the modelling results also showed that, the monitored spot values in the CFD validation model solver reached steady state values while the residual errors dropped below the required thresholds (Appendix, Fig. A.1), highlighting the attainment of the steady state CFD solution.

The vertical temperature distribution within the case study

Fig. 1. The George Davies Centre, Medical School Building, at Leicester University, UK (left), and the case-study (lecture theatre 1) auditorium space (right).

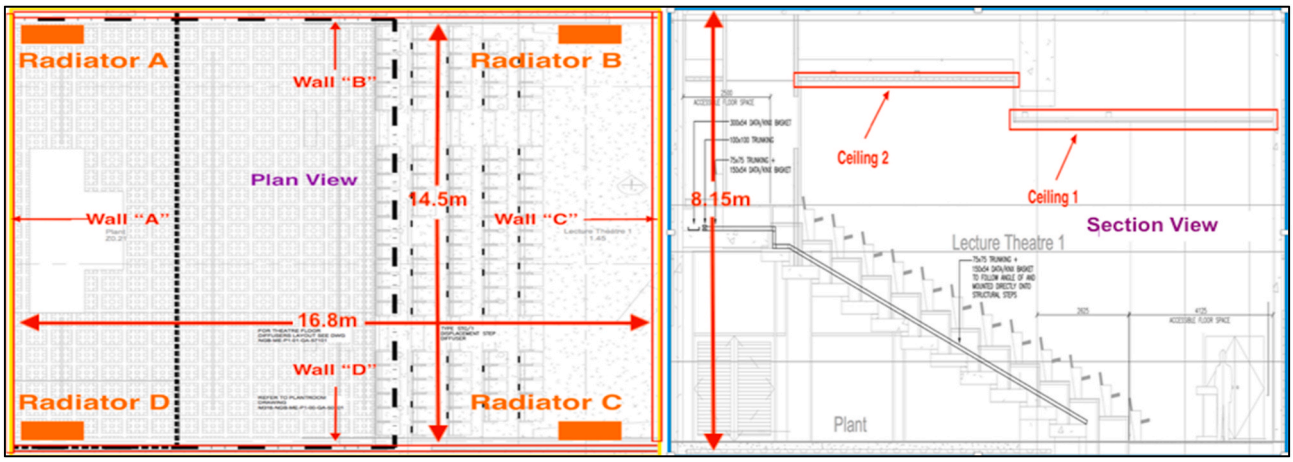

Fig. 2. Auditorium coding key for sensor locations; showing plan-view (left) and section-view (right).

Table 1
Locations of the sensors in the auditorium (to be read in conjunction with Fig. 2).

Sensor ID	Distance from wall A [m]	Distance from wall B [m]	Distance from wall C [m]	Distance from wall D [m]	Distance from ceiling 1 [m]	Distance from ceiling 2 [m]
H-03	–	6.95	3.82	–	5.23	–
H-04	2.19	0.00	–	–	–	1.54
H-05	4.35	–	–	2.95	–	2.66
H-07	–	–	8.83	4.10	–	4.60
H-08	4.36	4.10	–	–	–	2.65
J-01	–	–	5.52	2.91	4.75	–
J-02	7.00	–	–	0.00	–	2.19
J-03	5.26	4.09	11.31	–	–	3.45
J-04	3.00	–	–	0.00	–	0.49
J-05	2.7	0.53	–	–	–	2.39
J-07	6.97	–	12.37	–	–	2.90
J-09	4.05	0.00	–	–	–	0.63
KC-1	–	7.00	13.46	–	–	1.63
KC-4	4.40	0.53	–	–	–	3.03
KC-7	1.20	–	–	1.14	–	0.00
KC-9	1.70	0.00	–	–	–	0.47
M3	–	6.27	5.92	–	4.89	–
M4	–	–	6.38	2.92	4.05	–
M5	–	6.00	9.02	–	–	4.45
RJN-10	–	–	11.40	2.91	–	3.50
RJN-4	–	2.69	3.69	–	5.68	–
Z10	–	0.53	7.19	–	–	5.04
Z14	6.97	2.93	–	–	–	3.71
Z16	–	2.92	8.80	–	–	4.60
Z18	–	6.27	5.92	–	4.03	–
Z3	2.60	–	–	1.95	–	2.45
Z4	1.53	1.60	–	–	–	0.00
Z6	–	0.00	6.20	–	2.48	–
Z7	–	–	9.70	4.10	–	3.73

building's auditorium was predicted, using PHOENICS CFD software [42]. In this study, the standard k-ϵ (kinetic energy (k) and its dissipation rate (ϵ)) 2-equation turbulence model [43] was used for the MV scenarios. Launder and Spalding [43] suggest that the k-ε model is effective in modelling an auditorium ventilation system as a result of the close agreement between predicted and measured air velocity values in a case study performed by Nielsen [44]. Furthermore, Zhang and Cheng [45] 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 [46] as a result of the investigation of mean air flow rates through a naturally ventilated building. Although the standard k-ε model was also found by Wash and Leong [47] to be the most effective of the k-ε models for modelling the turbulence in naturally ventilated spaces, in this study, convergence difficulties were encountered when using the k-ϵ model with natural ventilation. As a result, the LVEL model was used for the NV scenarios. The LVEL model is an algebraic model requiring the distance to the nearest wall (L), the local velocity (VEL) and the laminar viscosity. The LVEL model approach has been shown to yield similar results to the k-ϵ model by CHAM [48] and Dhinsa et al. [49].

A simplified rectangular occupant geometry was used in the model (Fig. 3) in order to achieve an acceptable mesh quality and element numbers whilst making the CFD solution feasible and stable, in terms of convergence and computational expense. The use of the simplified occupant geometry in the model was thought not to result in inaccurate air temperature predictions in the regions of interest, around the occupants, since the simulation predictions were shown to be comparable with the field measurements under occupied conditions.

In the CFD models, the walls were largely modelled with an adiabatic boundary condition. The reason for this is that the modelled space had only one exposed external wall (wall B) and during the field measurements the air temperature difference between the outdoor and indoor

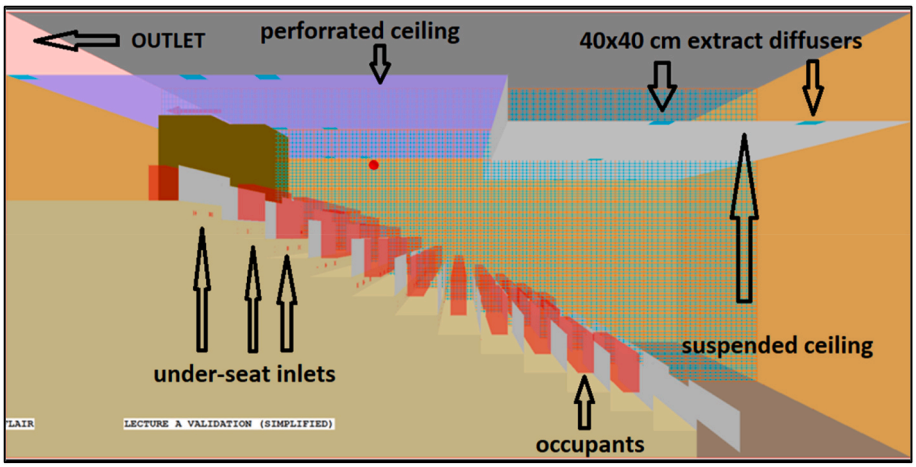

Fig. 3. CFD model showing a perspective view of the principle ventilation components of the auditorium.

wall surfaces was less than 3 °C. More importantly, since the building was a Passivhaus building (using highly insulated walls to minimize conductive heat transfer) and the auditorium space lacked windows, the assumption of the indoor air temperature being approximately equal to the wall surface temperatures is well justified. However, unlike the wall surface temperatures, some objects within the space (i.e. occupants, lights and audio and visual equipment) have considerably different radiant surface temperatures than the dry-bulb air temperature. In order to incorporate the radiative heat exchange component from these objects into the models, heat fluxes into the domain sourcing from the top of the tables, ceiling and some parts of the walls (e.g. those hosting audio and visual equipment) were added to the models. The radiative heat gain component of the lights was attributed to the tabletops, whilst the radiative heat gains from the occupants and audio and visual equipment were attributed to the closest surfaces with a direct view factor.

In preparing the CFD model for validation, boundary conditions taken from the field data, manufacturers specifications, the literature, and design guides were used to derive the key parameters (Table 2). The fresh air supply rate and its temperature were not available for each inlet. Therefore, the total fresh air supply rate from the AHU was subdivided equally 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 due to the well-insulated supply ductwork, relatively small temperature difference between the supply air and the internal distribution space temperatures, and the direct and relatively short air flow path linking the AHU to the inlets via the large supply plenum.

A hexahedral mesh was used in this study, and it was manually modified to improve the mesh quality, avoiding mesh elements with high skewness, high aspect ratio, and poor orthogonality (Fig. 4a). Each air inlet and outlet was represented with a mesh cell number proportional to their opening cross-sectional areas (Fig. 4b).

A mesh independence study was performed by generating meshes with 2.13, 3.37, and 7.1 million computational cells. The outlet flow rate was monitored for each simulation run, and it was shown that all three mesh options yielded the same outlet flow rate with differences in the range of 10^{-6} m^3/s Therefore, from a computational efficiency perspective, the coarse mesh (i.e. the mesh with 2.13 million cells) was selected for this study. The coarse mesh solver time for each simulation was about 27 h when running on a laptop computer with an Intel i5 (2 core) processor and 8 GB of RAM. The CFD solutions were assumed to be converged when the domain energy and mass imbalances were less than 1%, the root mean square (RMS) residuals of the transport equations were less than 10^{-5} and the monitored values (air temperature, air speed and velocity, air pressure and turbulence components) at six locations within the domain were steady (e.g. changes in variables between iterations were less than 0.1%).

Table 2
Boundary conditions for validation simulation.

Sensible Internal Heat Gains [W]		
Modelling Parameter	Value	Reference Source
Male occupants	90	CIBSE Guide A, 2021
Female occupants	78	CIBSE Guide A, 2021
Average occupant heat gains (50% male and 50% female assumed based on the field observations)	84	
Each LED luminaire (Total of 55 LEDs)	22	Manufacturer
Tablet (electrical appliance) for each occupant	3	Manufacturer
Audio and visual equipment	1000	Manufacturer
Fresh Air Supply		
Supply temperature	18.3 °C	Building Management System (BMS)
Supply air flow rate from each inlet	25.9 l/s	(Total Air Flow Rate from BMS divided by #inlets)
Inlets and Outlets		
Exhaust opening above suspended ceiling	1.15 m × 14.5 m	Facility Management (FM)
Rectangular ceiling diffuser dimensions	40 cm × 40 cm	Manufacturer
Rectangular ceiling diffuser discharge coefficient	0.61 (Representing sharp edged orifice)	Literature [50],
Perforated ceiling discharge coefficient	0.193	Estimate
Exhaust flow rate	3.9 m^3/s	BMS
Number of Under-Seat Inlet	150	Site Inspection
Size of Under-Seat Inlet	11 cm × 24 cm	Site Inspection

2.4. *Computational fluid dynamics modelling scenarios for identification of natural and mechanical ventilation limiting operating temperature ranges*

In order to assess the thermal comfort performance limits of the NV and MV systems, 34 different ventilation scenarios were modelled. Scenarios were created for typical occupancy at 68% capacity (n = 220 people/84 W/m^2) as observed in field studies and at full capacity (n = 324 people/124 W/m^2). These scenarios were then simulated with different supply (MV) or outdoor (NV) air temperatures, supply air flow rates (MV), opening sizes and positions (NV), and number of inlet and outlets (MV and NV) (Appendix, Table A1). The total number of scenarios (n = 34) required to explore the LOT ranges of the ventilation

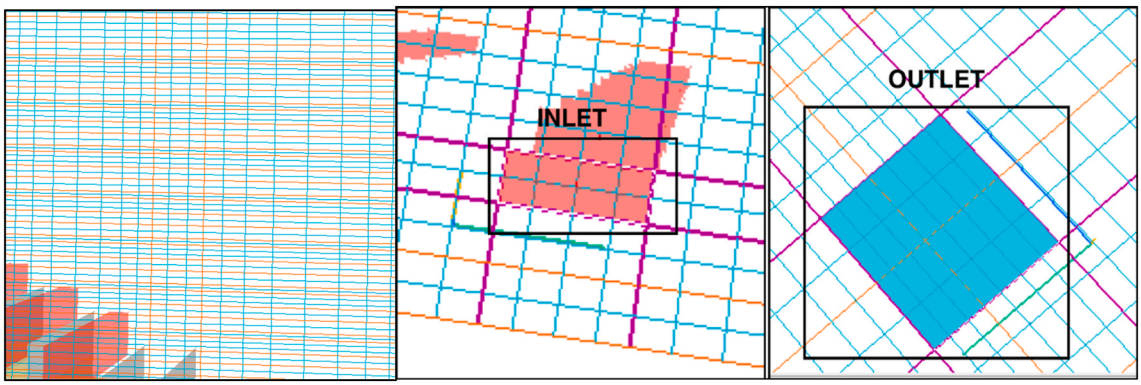

Fig. 4. Mesh elements demonstrating the low aspect ratio, high orthogonality, and low skewness properties of the general mesh (left), the number of mesh elements (8 for the inlets and 28 for outlets) used to represent ventilation terminals proportional to their sizes (middle and right).

solutions was decided upon based on the design parameters. For instance, supply air temperatures and effective opening areas for both occupancy densities (i.e. n = 324 and n = 220) and ventilation options (i.e. MV and NV) were explored (i.e. gradually increased or decreased) until the tested configuration was not able to provide a thermally comfortable environment without the need for active heating and cooling. Detailed visual illustrations of some of the key scenarios (Appendix, Table A1) highlighting specific ventilation design and operation interventions (such as varying the suspended ceiling opening percentage, inlet and outlet opening sizes and number of inlet and outlets) are reported in section 4 (Figs. 12 and 13). These particular scenarios aimed to investigate means to extend the LOT ranges through design and operation.

2.5. 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 [39], which is referred to as *criterion one* in this study, was chosen as it provides a robust standard for free running buildings by accounting for the adaptive behaviour of occupants. The equation for the allowable operative temperature limits within the 80% acceptability limits (i.e. the temperature limits wherein 80% of the occupants would be thermally comfortable) is given by the following formula:

$$T_{ot} = 0.31^{*}\ T_{DB,m} + 17.8 \pm 3.5 \tag{1}$$

where.

T_{ot} = Acceptable operative temperature range (at the 80% acceptability limits) [°C]; and
$T_{DB,m}$ = Monthly Mean Outdoor Air Dry-Bulb Temperature [°C].

Note that the operative temperature term in equation (1) refers to the average of the air and mean radiant temperatures in the auditoria and it is not the same as the LOT term we have introduced in this study.

One of the UK non-statutory guidance documents, Building Bulletin 101:2018 [40] which is widely used to define acceptable indoor environmental conditions in educational buildings (in relation to overheating), was used to define *criterion two* in this study. BB101 specifies that two of the following three requirements must be met.

1. The average indoor to outdoor air temperature difference should not exceed 5 K (in summer);
2. Indoor air temperature should not exceed 28 °C for more than 120 h in a year; and
3. When the space is occupied, the maximum indoor air temperature should not exceed 32 °C.

In this study only requirement three was tested.

UK design standard, CIBSE Guide A:2021 [41] which is identified as *criterion three* in this study, provided a similar approach to *criterion one*, where the comfort temperature band ("*Bands within which comfortable conditions have been found to lie are shown in relation to the running mean outdoor temperature.*" [40, p. 1–14] is given by the following formula:

$$\Theta_{com} = 0.33^{*}\Theta_{rm} + 18.8 \pm 3.5 \tag{2}$$

where.

Θ_{com} = Comfort Temperature [°C]; and
Θ_{rm} = Running Mean of Daily Mean Outdoor Air Temperature [°C].

If the maximum and minimum temperatures in the scenarios modelled by CFD complied with the acceptable temperature ranges defined by the above standards, then the ventilation case was deemed to "pass", and otherwise "fail". Due to the uncertainties inherent in the modelling process, a tolerance factor of ±0.5 K was applied to the upper and lower temperatures boundaries of the thermal comfort criteria respectively before assessing the CFD predicted temperatures. This value (i.e. ±0.5 K) was selected as it is the magnitude of the mean absolute error (MAE) between the CFD predicted and the measured temperature values (Fig. 6). In addition to assessing the above criteria, evaluation of localised thermal discomfort was considered by examining the temperature gradient from ankle to head level when the occupants were in seated positions. In accordance with CIBSE Guide A:2021 the vertical temperature gradient should not exceed 3 K in order to keep the Predicted Percentage Dissatisfied (PPD) index below 6% (BS EN ISO 7730:2005 [51]).

It should be noted that the aforementioned ASHRAE and CIBSE criteria use the operative temperature as the benchmark temperature, whereas the CFD model does not include an explicit radiation balance model and predicts dry-bulb air temperatures. However, the difference between dry-bulb and operative temperatures was considered to be negligible in the context of this study (as previously mentioned) due to the presence of highly insulated walls, with only one external façade (and relatively mild external temperatures) and the absence of windows on the external façade.

2.6. Climate analysis of the UK

A frequency distribution of UK outdoor air temperatures was created (Fig. 5) in order to make a comparison with the MV and NV LOTs. The result of the comparision was then used to determine the percentage of thermally comfortable occupied hours, for the UK climate, without the use of active heating and cooling systems for each tested scenario. This method was adopted as it was not feasible (due to the computational expense) to perform transient CFD simulations covering a whole year

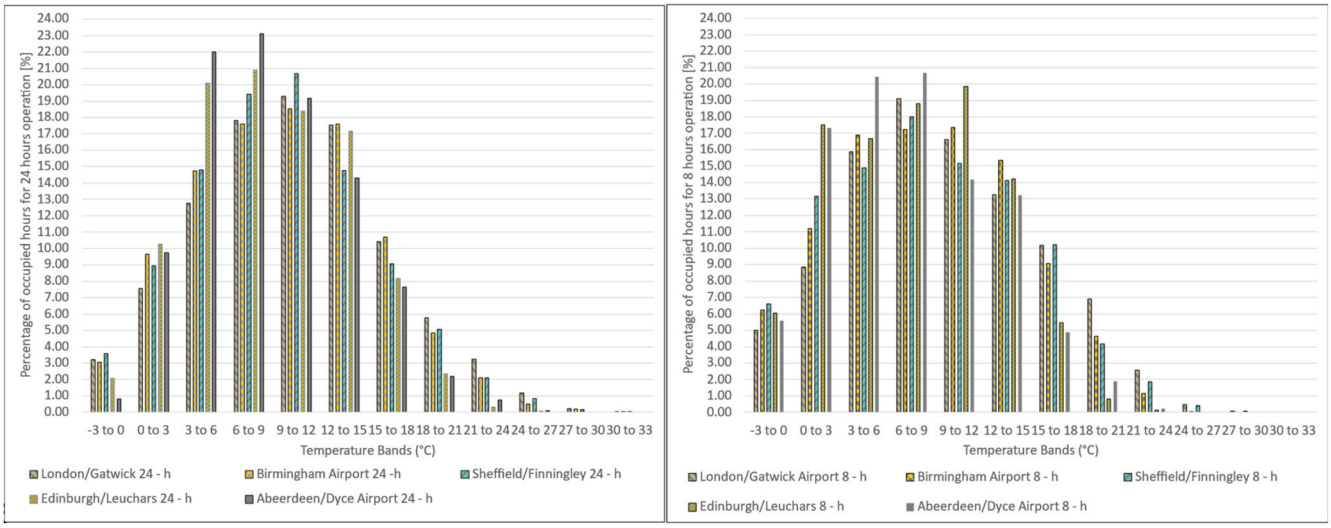

Fig. 5. Histogram showing the outdoor temperature frequency distribution of the major UK cities for 24 h (left) and 8 h (right) operating schedules.

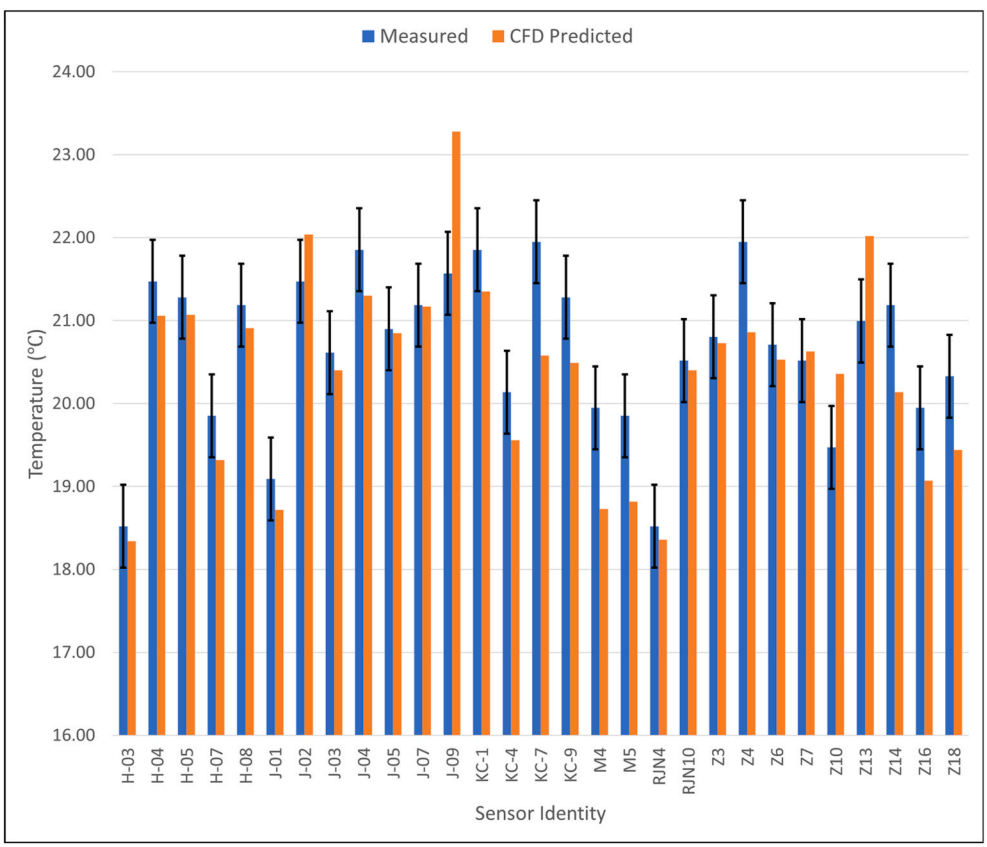

Fig. 6. Steady state spot temperature comparison between CFD predictions (orange bars) and site measurements (blue bars), black error bars showing the manufacturer's stated equipment uncertainty (see Fig. 2 and Table 1 for sensor locations). (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

with different occupancy and ventilation configurations.

International Weather for Energy Calculation (IWEC) weather files in the Energy Plus database [52] were used to create the outdoor temperature frequency distribution of the UK locations studied. IWEC was an ASHRAE project for creating simulation weather data that is derived from a 17-year historic record, spanning from 1982 to 1999 [52]. 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 the results of this study are intended to be generalisable for various non-domestic building types with similar characteristics (i.e. high internal heat gains and mixed mode ventilation systems) and various operational schedules (e.g. lecture, cinema and live performance theatres), the climate data was differentiated for 8-h (i.e. 09:00–17:00) and 24-h operational schedules. The climate data was grouped in 3 K temperature bands so that a temperature frequency

distribution could be determined for each location and operating schedule. The frequency distribution approach used was preferable to weighting all outdoor temperatures equally since logically the extreme temperatures only occur during a small proportion of the occupied hours throughout the year. Weighting all outdoor temperatures equally would yield a ventilation design meeting the demands under extreme temperatures as well, without considering the frequency of their occurrence. In a predominantly temperate climate this approach would result in an inefficient ventilation system design.

The climate analysis confirmed that the number of extreme temperature hours were relatively low in comparision to the total hours in a given weather year (Fig. 5). Since the IWEC climate data set used in this study is based on historic records and the literature states that warmer extreme temperatures are expected in the future, due to climate change [53,54], temperatures at the cold end of the spectrum (i.e. < -3 °C) were excluded from the original IWEC database. Conversely, low frequency but warmer outdoor temperatures (i.e. > 24 °C) are increasingly likely in the UK [55] and were therefore retained for the analysis.

3. Results

3.1. Validation of the CFD model

In order to validate the CFD model, the CFD temperature predictions were compared with field temperature measurements from the case-study building (Fig. 6). Temperature comparison was the preferred validation metric in this study since the temperature is the main parameter of interest to assess the LOTs of the various ventilation configurations.

The relatively small discrepancy (mean absolute error (MAE) 0.5 K) between the CFD predicted and measured results (Fig. 6) can be explained by measurement uncertainties. The error bars shown on the measured values represent only the manufacturers stated instrument accuracy (i.e. ±0.5 K), but not any other experimental related uncertainties. The majority of the measured values tend to be slightly higher than the calculated values which can be attributed to the inclusion of radiative heat exchange (between sensors and surrounding surfaces with higher temperatures, such as occupants and lights). While the measured values were influenced by the radiative heat exchange occurring within the sensor's field of view, the predicted values only partly accounted for these effects due to the simplified solution used for radiative heat exchange. 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. In relation to the distance between the ceiling and floor, differences from 10 cm to 30 cm were found between the model and the site. The reason for this measurement discrepancy was that the heights of the tiers in the plans (used in creating the model geometry) were different to the as-built configuration. Based on a ceiling height of 8.15 m, the magnitude of this error was between 1.2% and 3.7% in the Z-axis. Furthermore, simplifications and assumptions that were made in the CFD model geometry in order to reduce the mesh size and improve the mesh quality, contributed to this uncertainty. Additionally, simplifications in certain assumptions, such as: the assumption of uniform occupancy distribution (under part-occupancy conditions), the division of the total air flow rate equally between 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. Finally, the CFD simulation tool itself has a relatively small degree of uncertainty due to computer rounding, iterative solution methods, and discretisation errors [56,57]. Considering all of the above-mentioned uncertainties, the measured and predicted values were considered to be in good agreement (Fig. 6).

3.2. Limiting operating temperature ranges of natural and mechanical ventilation, and mechanical ventilation with heat recovery systems, in the UK climate

The validated CFD model was used to simulate the scenarios in Table A1 (see Appendix) and the results were analysed using the thermal comfort performance criteria in section 2.5 to determine whether a ventilation scenario was viable or not (Fig. 7). Additionally, the extended range of operating conditions (expressed as a temperature range) that an HR system would provide for the winter cases was calculated using the conservation of energy principle at the heat recovery terminal, based on the MVHR operating at a stated efficieny of 75% (which is the minimum efficiency required for compliance with the Passivhaus standard).

The findings indicated that NV is capable of extending the summer LOT range by 3 K–6 K for all the tested comfort criteria as a result of providing higher fresh air flow rates compared to MV (Fig. 7). Interestingly, the use of NV also extended the winter LOTs by 3 K under full occupany conditions (i.e. n = 324 occupants) according to the thermal comfort criteria two (BB 101) and three (CIBSE Guide A) when compared with MV without heat recovery. This was as a result of the ability of the NV to match the MV performance in providing the threshold requirement for fresh air (8 l/(s.person)) by reducing the effective opening sizes and changing the opening locations accordingly (as detailed in section 3). Compared to the NV mode, MV provided better performance in only two situations, the n = 220 and n = 324 occupancy scenarios when benchmarked against *criterion one*. In these two cases, MV extended the winter LOTs by 3 K. However the effect of the 3 K extension of the winter LOT range had a more pronounced effect on the percentage of thermally comfortable hours compared with the extension of the summer LOT (as shown in the NV cases). The reason for this is that the UK is a heating dominated climate. Additionally, when HR was added to the MV mode, the winter LOTs were further extended from 3.4 K to 8 K compared to the NV. Nonetheless, using MV (with or without HR), occupants were more prone to local thermal discomfort (as shown by lightning bolt symbols in Fig. 7) compared to the NV set-up due to the vertical temperature stratification exceeding 3 K from ankle to head height in a seated position. In the NV scenarios tested, no local thermal discomfort was observed in association with the temperature stratification since the temperature gradient from ankle to head levels remained less than 3 K.

The CFD predictions were able to give significant insights into the general performance of NV, MV and MVHR systems for various outdoor temperature ranges making the results extensible to comparable climates. However, to quantify the performance in the context of the UK climate, ventilation system LOTs (from Fig. 7) needed to be compared against a frequency distribution of the UK outdoor temperatures (Fig. 5). As a result of this comparison, the capability of NV, MV and MVHR, to provide a thermally comfortable environment for occupants (were the auditorium to be located in different cities) according to operational schedules (i.e. 8 or 24 h) is expressed as percentages of thermally comfortable occupied hours in Fig. 8 (for n = 220 occupants) and Fig. 9 (for n = 324 occupants). In order to highlight the potential energy savings when no active heating and cooling systems are used, it should be stressed that 1% of the occupied hours equates to 88 h in a 24-h and 22 h in an 8-h operating schedule.

The results showed that NV has the largest potential for application in warmer locations (e.g. London) and the least in colder (e.g. Edinburgh/Aberdeen) cities. This finding was expected in a temperate climate since NV was found to extend the summer LOTs, and warmer outdoor temperature hours are more common in the South of the UK compared to the North. Furthermore, it can be seen that NV performed poorly compared to MV in relation to performance *criterion one*, but it performed better in *criteria two* and *three* (Figs. 8 and 9). Evaluation of the total thermally comfortable occupied hours in Figs. 8 and 9 showed that, the thermally comfortable hours, using NV according to *criterion*

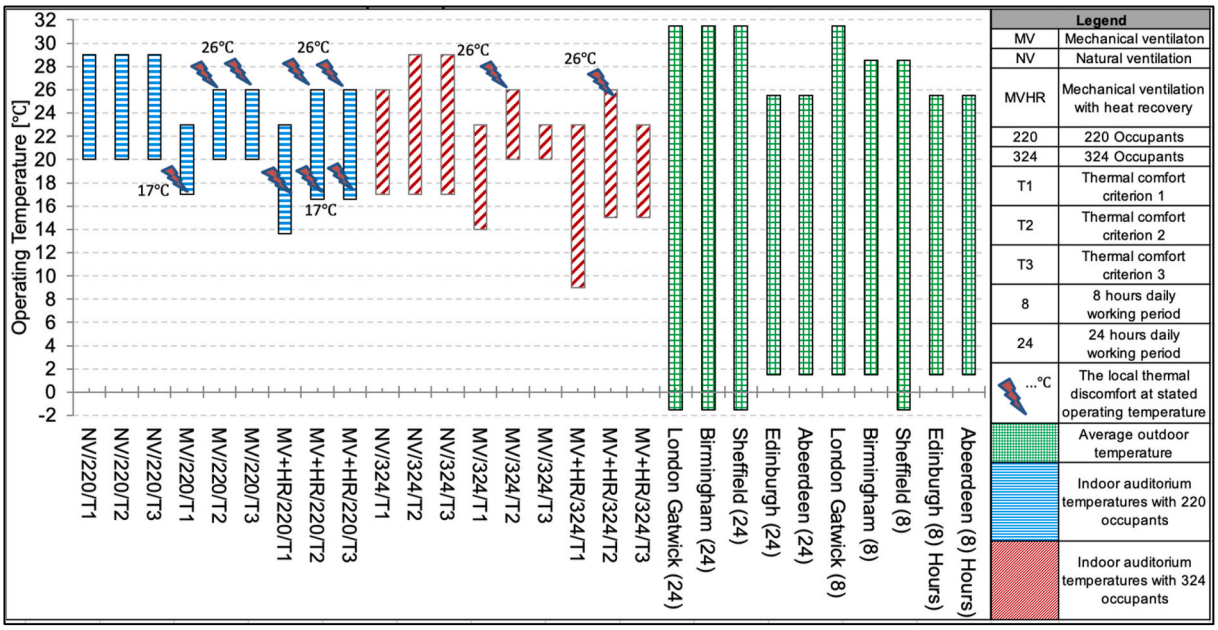

Fig. 7. Summary of the scenario based parametric CFD modelling results showing LOT ranges for n = 220 (blue striped bars) and n = 324 (red striped bars) occupants, against the UK outdoor temperature ranges (green hatched bars), showing the presence of local thermal discomfort for particular scenarios (lightning bolt symbol) due to the temperature gradient. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

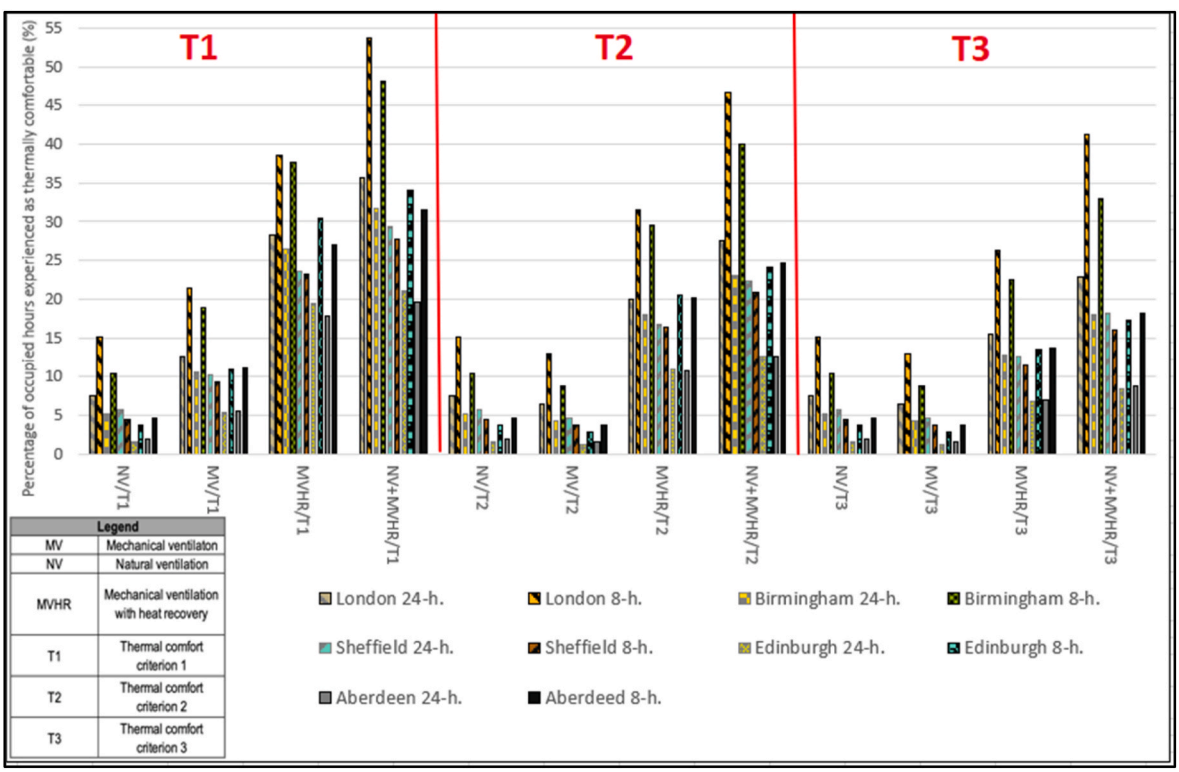

Fig. 8. Summary of the CFD parametric modelling results at partial capacity (n = 220 occupants) as a percentage of the thermally comfortable occupied hours in 5 UK climatic contexts.

one, were about 90% less compared to the use of MV. However, when evaluated according to *criterion 2 and 3*, NV provided 15% more thermally comfortable hours compared to MV. This can be explained by virtue of the fact that *criteria two and three* have more relaxed requirements for upper temperature ranges than *criterion 1*. Expectedly, the scenarios with MVHR performed better than either the NV or MV scenarios in relation to their range of applicability. This was because the use of MVHR extended the winter LOTs significantly downward (by pre-warming the supply air) and the UK is a heating dominated country. It is also important to emphasise that, when NV and MVHR were used together, an optimal solution was found in all cases with the resultant occupied comfort hours ranging between 8.4% (n = 220 occupants, criterion three, Edinburgh, 24 h schedule) - 82.3% (n = 324 occupants, *criterion one*, London, 8 h schedule) (Fig. 8 and 9).

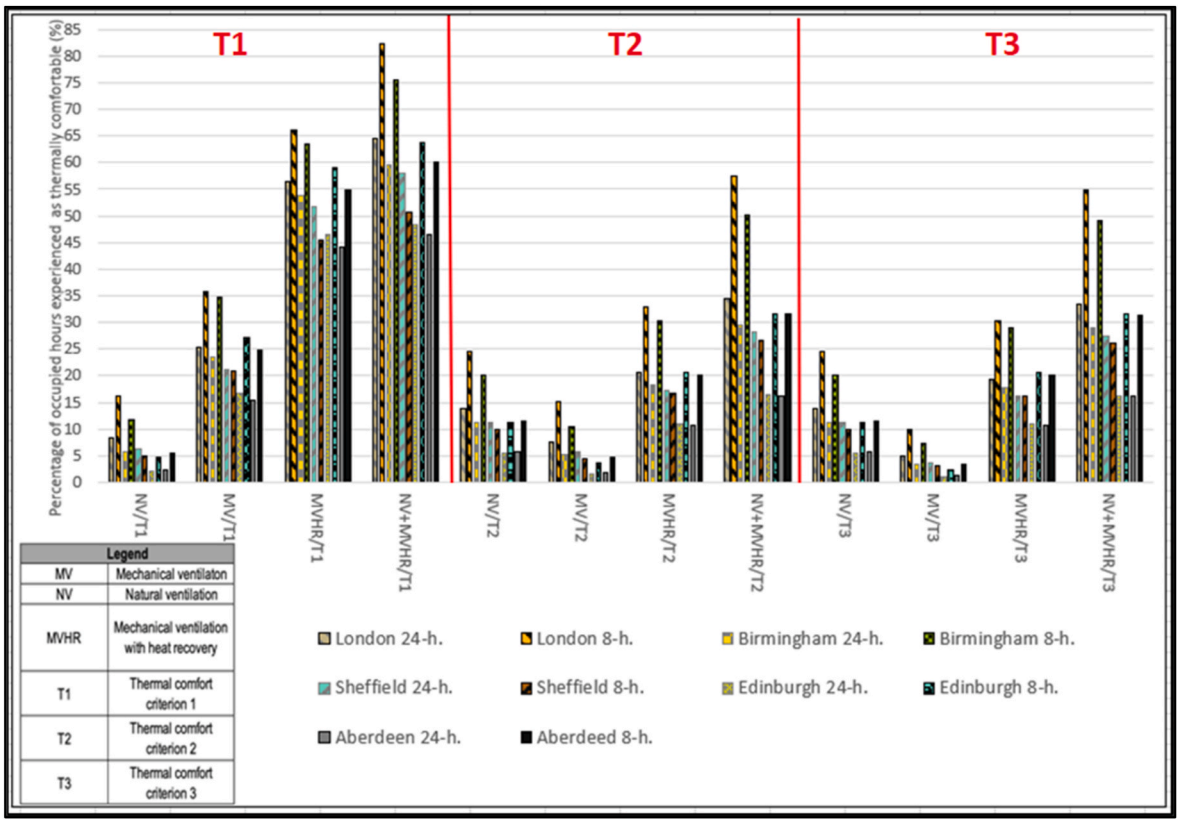

Fig. 9. Summary of the CFD parametric modelling results at full capacity (n = 324 occupants) as a percentage of the thermally comfortable occupied hours in 5 UK climatic contexts.

3.3. Temperature stratification in naturally and mechanically ventilated spaces

Significant temperature stratification is undesirable in rooms with low ceilings as it can cause the stale (and potentially contaminated) room air to remain in the breathing zone, resulting in IAQ problems [58]. Additionally, it could also result in local thermal discomfort when the temperature gradient exceeds 3 K from ankle to head level [50].

The NV scenarios demonstrated a smaller temperature gradient in the domain compared to MV, and this can be attributed to the provision of larger airflow rates. The NV strategy (Fig. 10, top) resulted in a significantly lower temperature gradient (<2 K) compared to the MV strategy (Fig. 10, bottom) which was greater than 3 K, under identical outdoor air temperature boundary conditions. Moreover, using NV it would be possible (depending on the indoor air flow patterns) to achieve a better IAQ compared to the MV due to the increased fresh air flow rate (30 l/(s.person) in NV as opposed to 17 l/(s.person) with MV). This finding is particularly relevant in the context of COVID-19 and other airborne viruses where transmission risks have been shown to be directly correlated to the air exchange rates [59,60]. In this case, the increased ventilation rate by NV without the expense of additional electric power (which would be required if the same ventilation rate was to be achieve by MV) might be beneficial (depending on compounding factors) for mitigating airborne disease transmission risks [61].

Another consequence of the reduced flow rates associated with the MV system is that under warm conditions, occupants in the centre of the zone experienced localised temperatures that are up to 3 K higher (Fig. 10, bottom) than those experienced with NV (Fig. 10, top), a factor which may place them at increased risk of heat stress.

4. Ventilation design and operation interventions to improve thermal comfort

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 LOT limits at both the low-end (winter) and high-end (summer) of the outdoor temperature spectrum.

The case study building had a constant air volume (CAV) AHU providing a volumetric flow rate of 3.9 m^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 LOT range of the MV system. The findings showed that the winter LOT could be extended from 17 °C down to 15 °C for the n = 220 person occupancy condition, if the fixed air supply rate of 17 l/(s.person) was reduced to 10 l/(s.person) (which is the minimum fresh air supply rate requirement according to EN 16798–1:2019 [62]. This finding results from the reduced ventilation heat losses occurring from the domain, which consequently increases the indoor temperatures (Fig. 11). According to IWEC climate data for each city, assuming a typical 8 h working day, this 2 K extension in the LOT would equate to an extension of between 310 and 340 occupied hours (depending on the location) per year in the UK climate, in which thermal comfort (with an 80% acceptability rate) can be achieved without the use of an active heating system.

Using NV, the fresh air flow rate is not as easy to control as MV because the buoyancy forces for NV are less stable than the momentum forces of the MV. However, the simulation results showed that it was possible to influence both the spatial domain temperatures and fresh air flow rates in NV mode by adjusting the inlet and outlet opening sizes and locations, including any transfer paths such as suspended ceilings.

The standard NV setup for the summer conditions, prior to this design intervention, was comprised of: three outlet openings above the

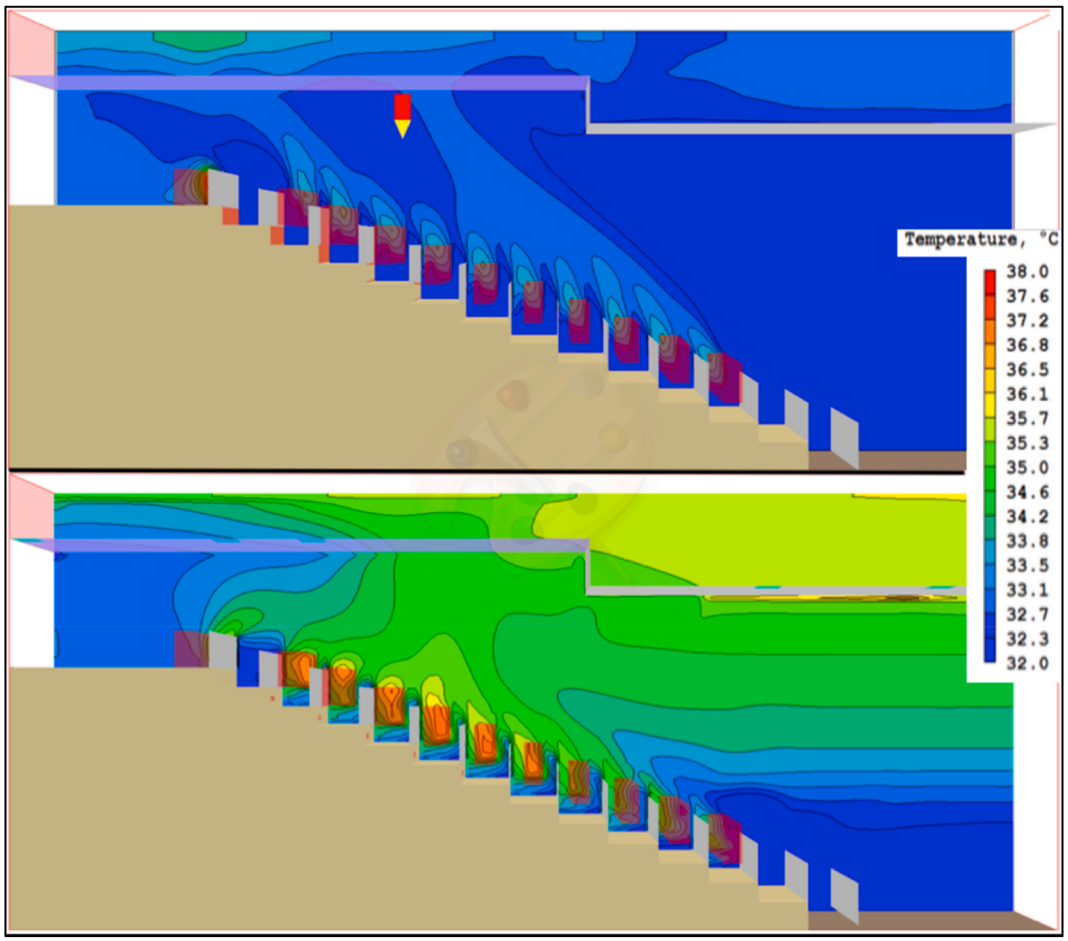

Fig. 10. Vertical temperature distribution (n = 220 Occupants) 32 °C supply air, NV (top) vs. MV (bottom).

suspended ceiling level, thirteen inlet openings (comprising of 11 cm high inlet grilles at each tier, spanning from one side of the room to the other) and two suspended ceilings (one entirely covered with perforations and ceiling diffusers, and the other with ceiling diffusers only) (Fig. 12, top left). When the supply air (outdoor) temperature was increased from 26 °C to 29 °C, the standard NV setup (with 11 cm inlets) failed to comply with thermal comfort *criterion one* because of the high internal temperatures, but satisfied *criteria two* and *three* (Fig. 12, top right). However, when the height of each inlet opening was increased (from 11 cm to 35 cm) to give an opening area of 53.7 m^2 compared to 20.7 m^2 (Fig. 12, bottom left), the overall temperatures in the domain were reduced, with the maximum temperature dropping from 32.3 °C to 30.5 °C (Fig. 12, bottom right). 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.person) to 20.2 l/(s.person) when the inlet sizes were increased from 11 cm to 35 cm. This finding suggests that the initial higher volumetric fresh air flow rates (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.

Similarly, the winter CFD modelling results showed that the NV LOT could be extended with an adjustment to the air inlet configuration. Thus, an appropriate adjustment of openings would result in a reduction of fresh air flow rates, and hence an increase in indoor temperatures. Moreover, the results demonstrated that, reducing the number of inlet and outlet openings, and decreasing the suspended ceiling opening percentage would result in an increase in the resistance to air flow path through the domain. As a result, the fresh air entering the domain could be reduced and rerouted to the desired path in order to prevent cold down-draughts at the occupied area. In order to extend the winter LOTs, a series of three design and operation interventions were investigated to establish whether all three thermal comfort criteria could be met.

1. In order to restrict the fresh air supply into the domain, the number of inlets was reduced from one opening at each tier to one opening at every two tiers (making the total number of inlets 7), thereby reducing the effective opening area from 20.7 m^2 to 11.2 m^2 (Fig. 13, top left). Furthermore, the exhaust opening outlets (nos. 2 and 3) were remodelled as a wall, with only outlet no. 1 remaining as an exhaust outlet. The closed inlets and outlets were marked as "closed" in Fig. 13 (top left of figure). This new inlet configuration resulted in a fresh air flow rate of 10.2 l/(s.person) into the domain, but thermal comfort *criteria one* and *three* could not be met as the CFD predicted temperatures were below the thermal comfort requirements (Fig. 13, Top Right). The presence of colder air was more evident in the front rows compared to the others. The air velocity vector plot on a rectangular two-dimensional plane (Fig. 13, top right) showed that outlet no. 1 acted as a bi-directional opening; thus, the rising warmer air at the rear of the room was mixed with the cooler fresh air. Consequently, this resulted in a cool down-draught from Ceiling 2, but more predominantly from Ceiling 1 (Fig. 13, top right).
2. As the second intervention, in order to prevent the cold air down-draught, Ceiling 1 was re-modelled as a closed ceiling with no diffusers (Fig. 13, middle left) (where it previously contained a significant number of 40 by 40 cm ceiling diffusers). This intervention created greater resistance in the air flow path, which consequently prevented the cooler air from falling into the domain from the suspended Ceiling 1 (Fig. 13, middle right). As a result, higher

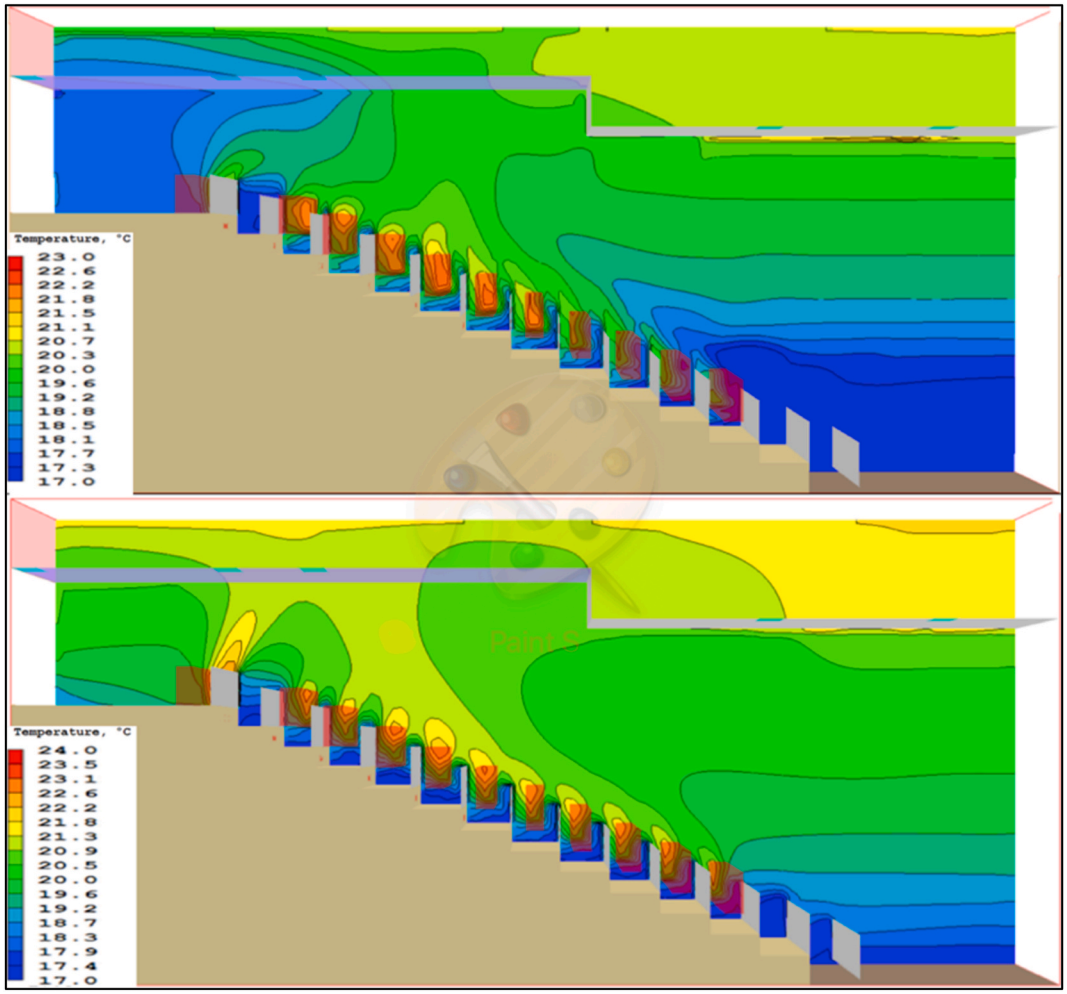

Fig. 11. Vertical temperature distribution, MV (n = 220 occupants) 17 °C supply air, at 17 l/(s.person) flow rate (top) and 10 l/(s.person) flow rate (bottom).

temperatures were generated in the occupied zone. Furthermore, the fresh air supply rate into the domain was reduced from 10.2 l/(s.person) to 8.6 l/(s.person) by merely closing the 40 by 40 cm openings, whilst keeping the remaining inlet and outlet openings the same. As a result of this intervention, acceptable temperatures were achieved (19.2–21 °C) at most seats (Fig. 13, middle right). However, this intervention was insufficient to comply with *criteria one and three* since the last three rows of seats were still exposed to the cold air down-draught coming from Outlet 1 and passing through suspended Ceiling 2.

3. A final intervention was applied in order to determine whether further adjustment of the suspended ceiling could enable the NV scenario to comply with all three thermal comfort requirements. In this scenario 60% of suspended Ceiling 2 was modelled as a closed ceiling in the location affecting the cool air down-draught at the back of the room, and the remaining 40% was modelled as an opening (Fig. 13, bottom left). This final intervention aimed to increase the temperatures at the rear of the room without exceeding the upper temperature limit of the comfort criteria. The desired result was achieved as the cool down-draught was prevented and the resultant temperatures at the rear were within thermal comfort limits (Fig. 13, bottom right). When the spatial temperature distribution of the whole domain was analysed, it was found that this ventilation scenario complied with all three thermal performance criteria.

5. Discussion and limitations

5.1. Natural and mechanical ventilation in non-domestic passivhaus and near zero energy buildings in temperate climates

This study has explored supply air temperature and ventilation flow rate ranges suitable for providing a thermally comfortable and well-ventilated indoor environment (without the use of active heating and cooling technologies). The implications of using NV, MV and MVHR strategies have been studied individually and in combination, for an auditorium zone within an existing non-domestic Passivhaus building in the UK climatic context. However, the findings are extendable to other high-performance, energy efficient buildings with similar climatic, operational, design and occupancy conditions. The findings of this study provide detailed insights for stakeholders wishing to make informed ventilation design and operational decisions at the early and in-use stages of non-domestic Passivhaus and nZEBs. Furthermore, the results will have practical benefit for building services operators and facility managers as the relationship between supply air temperatures, ventilation configuration, and spatial thermal comfort are clealry identified. In particular, the methodology used in this study provides a workable approach to resolving complex design trade-offs between ventilation design, thermal comfort, energy efficiency and carbon emissions.

The findings suggest that all of the ventilation systems examined (whether used individually or in combination) would require some degree of additional heating or cooling, due to the dynamic effects of occupancy, seasonality, and the challenging geometrical design

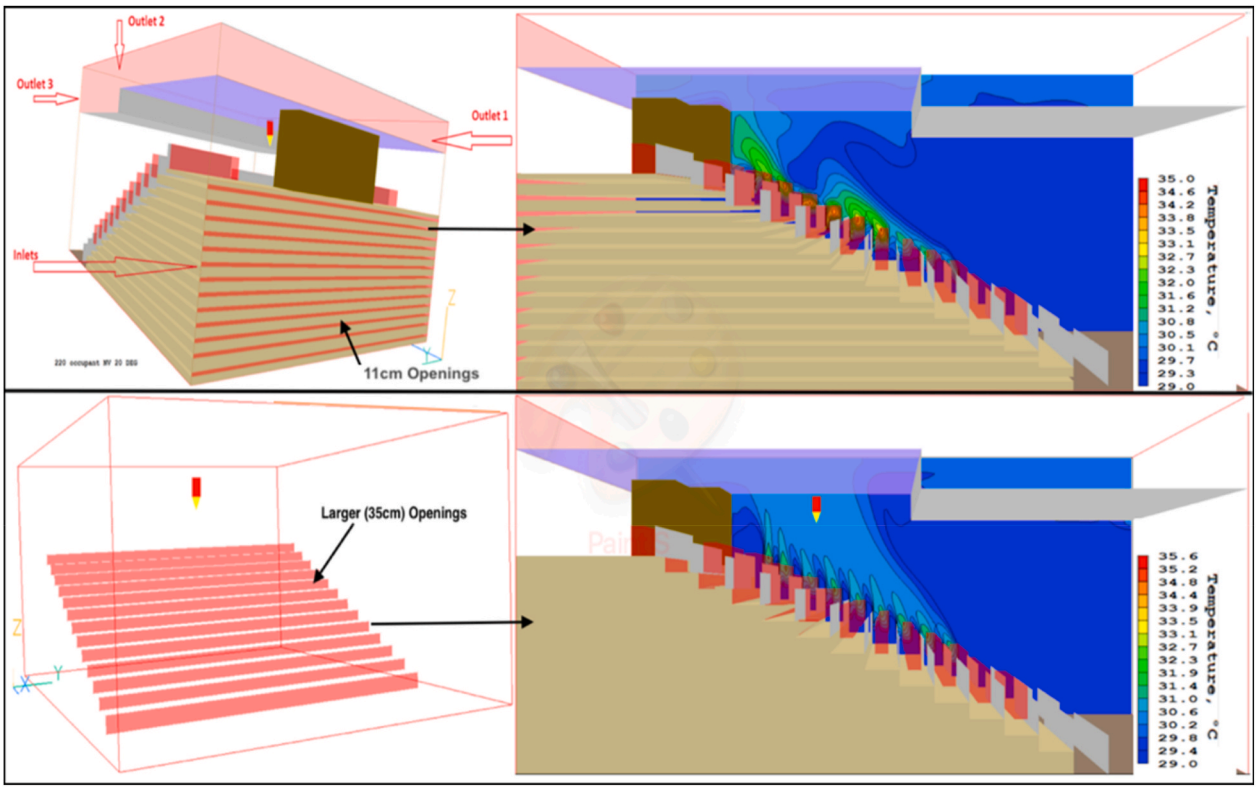

Fig. 12. Influence of small (top) and large (bottom) NV inlet openings and vertical temperature distribution (n = 220 occupants) 29 °C supply air.

characteristics of auditoria. Both the simulation findings and site inspections (where a hydronic heating system is used routinely during colder periods) confirmed that it was not always possible to maintain thermal comfort whilst avoiding the use of a dedicated supply air preheating system, even with transient internal heat gains as high as 124 W/m^2 in the temperate UK climate. This was due to the presence of localised thermal discomfort, in particular temperature gradients of more than 3 K from ankle to head level. The modelling results showed that such localised thermal discomfort conditions are often encountered in mechanical displacement ventilation systems, a finding which is in common with a number of previous studies [29–31,35]. However, the NV modelling results demonstrated that these localised discomfort issues can often be avoided when NV is used instead of MV, supporting the findings of Cook and Short [12] and Ricciardi et al. [30]. Furthermore, as suggested by Cook and Short [12] and Cao et al. [36], NV should be preferred over MV whenever net energy savings are possible and additional heat recovery through a MV system is not required. Since the modelling results show that, a thermally comfortable environment can be achieved with the same LOTs for both NV and MV, it confirms that the energy savings associated with NV can be achieved without compromising thermal comfort. Although, the use of NV might not be always possible due to unfavourable outdoor conditions, as suggested by Cuce et al. [34], the modelling results identified specific LOTs where the MV would cause local discomfort, allowing the selective use of NV for those MV LOTs, thereby minimising the risks of operating NV in unfavourable conditions.

Based on the modelling results, it was evident that the use of natural ventilation extends the upper limit of the summer LOTs by 3 K–6 K depending on the tested criterion and occupant density compared to mechanical ventilation. As expected, the natural ventilation approach showed a higher potential in warmer regions (i.e. London) compared to the cooler regions (north UK). However, in order to influence the indoor temperatures and acquire the most benefit out of the natural ventilation system, it is essential to employ controllable inlet and outlet opening sizes. Despite the undeniable potential of NV, the benefits of hybrid systems involving some form of mechanical ventilation should not be overlooked, particularly where stringent low energy targets are sought. When the heat recovery system is operational, the use of mechanical ventilation extended the winter limiting operating temperature downwards by 2 K (to 8 K) compared to the natural ventilation system (without using auxiliary heating). Furthermore, mechanical ventilation with heat recovery is better suited for cooler climatic regions compared to natural ventilation, since it provides more thermally comfortable hours in cooler regions compared to the natural ventilation systems (see Figs. 7 and 8). Conversely, it has been shown that the Passivhaus concept [17] and similar low energy concepts [20], if poorly implemented, can be prone to overheating, whilst Perez and Østergaard [18] found that 90% of the time mechanical ventilation with heat recovery is not required in milder climates. This study concurs that non-domestic Passivhaus auditoria are prone to overheating (even in a temperate climate) when ventilated solely via mechanical ventilation, due to the pronounced temperature stratification occurring in the occupied zone. However, in relation to thermal comfort and energy efficiency, the findings from this study are in agreement with those of Lambea et al. [26] which found that mechanical ventilation with heat recovery is an essential requirement for Passivhaus certification, unless the climate is consistently hot. Collectively these findings point to the benefits of incorporating seasonal mixed-mode systems in non-domestic Passivhaus and nZEB buildings, where natural ventilation is used in preference to mechanical ventilation whenever the local climate (and air quality) permits.

The requirement for active cooling along with mixed-mode ventilation for achieving thermal comfort when the outdoor air is too hot or cold was highlighted by Cuce et al. [34]. Similarly, the result of this study demonstrates that additional measures would be required in the present UK climate to achieve year-round thermal comfort fully due to the heating and colling loads. Despite the requirement for some auxiliary space conditioning, the results show that, when intelligent NV

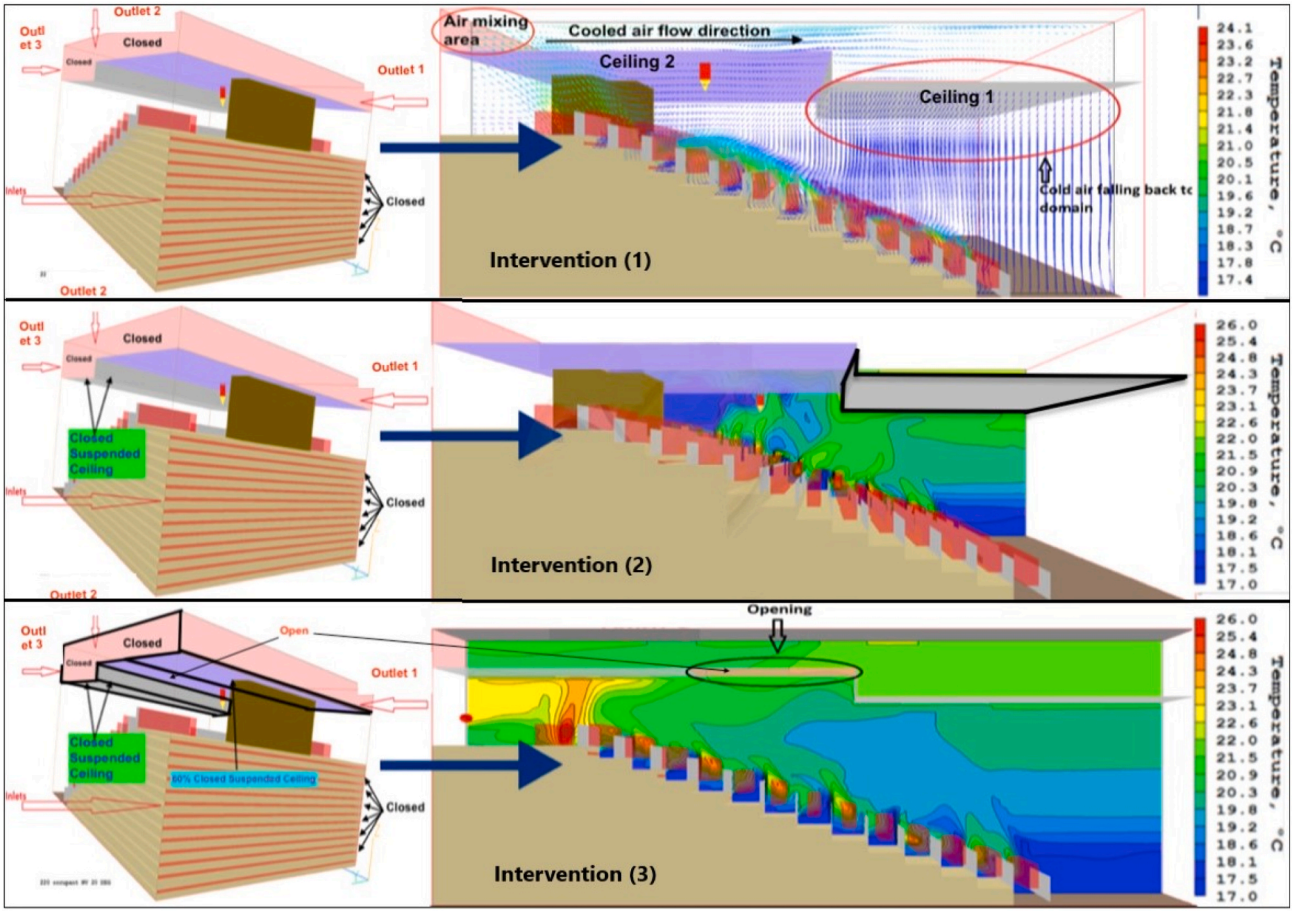

Fig. 13. Design interventions to the standard NV configuration to achieve thermal comfort standards (n = 324 occupants) 17 °C supply air, (1). (top left) closed inlets and outlets, (top right) resultant cold down draughts, predominantly from Ceiling 1, (2). (middle left) closed suspended Ceiling 1, (middle right) resultant warmer indoor temperatures below Ceiling 1 and 2, (3). (bottom left) partially (60%) closed Ceiling 2, (bottom right) resultant warmer indoor air temperatures.

design is used in conjunction with MVHR, it was possible to provide a thermally comfortable environment up to 82% of the time in certain contexts (e.g. in London with a typical 8-h working schedule). This could be considered a significant finding where HVAC related energy usage in non-domestic buildings in such cities is high. However, when using a hybrid approach (e.g. NV + MVHR) it is essential to employ variable opening configurations, with intelligent controls, which respond to internal heat gains and outdoor air temperatures to achieve such performance. Although modifying 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. Relatively minor design changes to the indoor inlet and outlet openings (such as changing the suspended ceiling opening percentage and closing and opening in-domain air inlets and outlets) can achieve the same objective. This makes retrofit solutions a possibility without the requirement for major structural changes in the façade. It also means that improved mixed-mode ventilation designs offer a viable means of improving localised occupant thermal comfort whilst reducing reliance on mechanical heating, cooling and ventilation systems in the context of complex non-domestic Passivhaus and nZEB buildings. Moreover, the work has highlighted the benefits of such an approach in relation to improving volumetric flow rates which have been shown to have a marked impact in reducing airborne viral transmission rates [63,61].

5.2. Limitations

The assumptions and simplifications in relation to the boundary conditions used in this study will have influenced the accuracy of the validation model. Factors such as flow rates and supply temperature for each individual inlet were not precisely known. Additionally, the modelled geometry was not a perfect representation of the real domain due to small 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. Despite the aforementioned simplifications and assumptions, the validation model predictions closely matched the field measurements (MAE of 0.5 K). Moreover, a more accurate representation of internal surface temperatures is needed to derive the operative temperatures in the occupied zone, which would provide a more accurate assessment of the thermal comfort and acceptability of the limiting outside temperature range. In order to achieve this dynamic thermal simulation incorporating radiative heat exchange (i.e. use of a radiance model to predict surface temperatures) could be used to provide more accurate boundary conditions for the CFD model. In this regard the findings of the current approach can be seen as conservative, since the use of operative temperature is likely to further extend the upper and lower bounds of the acceptable limiting outside temperature range. This is due to the fact that in winter the surrounding surfaces of a well-insulated building are likely to be significantly warmer than the external air temperature. Whilst conversely, under warm summer conditions the surrounding surfaces of a well-insulated thermally massive building are likely to be cooler than the warmest air temperatures entering the domain. Finally, whereas the effect of relative humidity (RH) on thermal comfort is not of great concern in the UK, in climates with significantly higher or lower RH, the outcomes might be different.

6. Conclusions and future work

The thermal comfort performance of natural and mechanical ventilation and mechanical ventilation with heat recovery has been explored in a non-domestic UK Passivhaus building using state-of-the-art numerical modelling. The model fidelity was validated using field data. The findings conclude that a mixed-mode approach to ventilation (combining natural ventilation and mechanical ventilation with heat recovery) is well suited to the UK climatic context. The mixed-mode approach was able to provide a thermally comfortable environment (without using any additional heating or cooling) for up to 82% of the annual occupied hours. Notably, the mixed-mode approach achieved the target thermal comfort criteria for up to 82% of the occupied hours for the London climate. Considering the largest space conditioning related energy demand of UK non-domestic buildings is found in London, achieving thermally comfortable occupied hours for up to 82% of the total occupied hours (7–25% for natural ventilation, 5–36% for mechanical ventilation, 16–66% for mechanical ventilation with heat recovery and 23–82% in mixed mode operation) without recourse to any auxiliary heating and cooling could result in significant energy savings. Moreover, these figures are likely to significantly underestimate the full range of thermally comfortable occupied hours achievable in practice. This is because the thermal inertia of highly insulated building envelopes would help to maintain the internal operative temperatures within acceptable thermal comfort limits even when the outdoor air temperatures begin to drift outside of an acceptable comfort range. Determination of the limiting external operating temperatures of natural and mechanical ventilation (with and without heat recovery) is a critical requisite for maximising the benefits of both approaches. The design suggestions provided here will help to extend the limiting operating temperature ranges of these systems and are likely to be of value in enhancing the performance and resilience of Passivhaus and near zero energy building auditoria, as well as similar non-domestic buildings.

In choosing the appropriate combination of natural and mechanical ventilation system specification, consideration should be given to the resultant operational and embodied energy and carbon impacts (and resource emissions), which are beyond the scope of this study. Whilst the Passivhaus standard does not currently address these issues, mechanical ventilation systems with heat recovery often carry a large primary energy and embodied carbon penalty due to the requirements for both intake and exhaust fans and ductwork, compared to simpler mechanical extract ventilation systems [63]. It is therefore suggested that further work could include quantifying the potential energy and greenhouse gas emission savings of using a range of hybrid ventilation system configurations in a wide range of non-domestic Passivhaus and near zero energy buildings.

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CRediT authorship contribution statement

Murat Mustafa: Writing – original draft, Visualization, Validation, Software, Methodology, Formal analysis, Data curation, Conceptualization. **Malcolm J. Cook:** Writing – review & editing, Supervision, Software, Project administration, Methodology, Conceptualization. **Robert S. McLeod:** Writing – review & editing, Writing – original draft, Visualization, Supervision, Formal analysis, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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Appendix

Table A.1
Boundary conditions of the MV and NV CFD simulation scenarios

CASE ID	V.* Mode	Nos. of Occ.* (n)	Supply Temp. [°C]	Supply Air Flow Rate [l/(s. person)]	Nos. of Inlets and Nos. of Outlets	Tot. Eff. Opening Area of Inlets and Outlets [m^2]	Suspended Ceiling Opening [%]
S1*	NV	220	17	N/A	13 and 1	20.7 and 16.7	100
S2	NV	220	20	N/A	13 and 1	20.7 and 16.7	100
S3	NV	220	23	N/A	13 and 1	20.7 and 16.7	100
S4	NV	220	26	N/A	13 and 1	20.7 and 16.7	100
S5	NV	220	29	N/A	13 and 1	20.7 and 16.7	100
S6	NV	220	29	N/A	13 and 1	53.7 and 42.4	100
S7	NV	220	32	N/A	13 and 1	53.7 and 42.4	100
S8	NV	324	14	N/A	7 and 1	11.2 and 16.7	23**
S9	NV	324	14	N/A	7 and 1	11.2 and 16.7	56**
S10	NV	324	17	N/A	7 and 1	11.2 and 16.7	23**
S11	NV	324	17	N/A	7 and 1	11.2 and 16.7	33**
S12	NV	324	17	N/A	7 and 1	11.2 and 16.7	56**
S13	NV	324	17	N/A	7 and 1	11.2 and 16.7	100
S14	NV	324	20	N/A	13 and 1	20.7 and 16.7	100
S15	NV	324	23	N/A	13 and 1	20.7 and 16.7	100
S16	NV	324	26	N/A	13 and 1	20.7 and 16.7	100
S17	NV	324	29	N/A	13 and 1	53.7 and 42.4	100
S18	NV	324	32	N/A	13 and 1	53.7 and 42.4	100

(*continued on next page*)

Table A.1 (*continued*)

CASE ID	V.* Mode	Nos. of Occ.* (n)	Supply Temp. [°C]	Supply Air Flow Rate [l/(s. person)]	Nos. of Inlets and Nos. of Outlets	Tot. Eff. Opening Area of Inlets and Outlets [m^2]	Suspended Ceiling Opening [%]
S19	MV	220	14	8	150 and 1	3.2 and 16.7	100
S20	MV	220	17	17	150 and 1	3.2 and 16.7	100
S21	MV	220	17	10	150 and 1	3.2 and 16.7	100
S22	MV	220	20	17	150 and 1	3.2 and 16.7	100
S23	MV	220	23	17	150 and 1	3.2 and 16.7	100
S24	MV	220	26	17	150 and 1	3.2 and 16.7	100
S25	MV	220	29	17	150 and 1	3.2 and 16.7	100
S26	MV	220	32	17	150 and 1	3.2 and 16.7	100
S27	MV	324	12	8	75 and 1	1.6 and 16.7	100
S28	MV	324	14	8	75 and 1	1.6 and 16.7	100
S29	MV	324	14	11.6	150 and 1	3.2 and 16.7	100
S30	MV	324	17	11.6	150 and 1	3.2 and 16.7	100
S31	MV	324	20	11.6	150 and 1	3.2 and 16.7	100
S32	MV	324	23	11.6	150 and 1	3.2 and 16.7	100
S33	MV	324	26	11.6	150 and 1	3.2 and 16.7	100
S34	MV	324	29	11.6	150 and 1	3.2 and 16.7	100

*S= Scenario, V=Ventilation, Occ. = Occupants; ** In these scenarios the opened area of the suspended ceiling letting the air to pass through gradually increased or decreased and the effect of the altering the opening percentage on the internal temperatures were observed.

Table A.2
Spot temperature measurement results in auditoria during "Lecture A". The temperature values recorded at 09:30 were selected for the CFD validation.

Lecture A											
Temperature Measurements (°C)											
Time (hh:mm)											
Sensor ID	08:40	08:50	09:00	09:10	09:20	09:30	09:40	09:50	10:00	10:10	10:20
H-03	18.3	18.3	18.5	19.2	18.6	18.5	18.5	18.6	18.6	18.0	18.1
H-04	19.2	19.2	19.5	21.0	21.7	21.5	21.7	21.8	21.9	20.7	20.0
H-05	19.9	19.9	20.2	21.5	21.6	21.3	21.2	21.3	21.4	20.2	19.9
H-07	19.4	19.4	19.6	20.8	20.2	19.9	19.7	19.6	19.7	18.8	18.8
H-08	19.7	19.7	20.0	21.2	21.4	21.2	21.2	21.2	21.2	20.2	19.9
J-01	19.0	19.0	19.3	19.9	19.4	19.1	19.0	18.9	18.9	18.7	18.6
J-02	19.9	19.9	20.3	21.3	21.6	21.5	21.5	21.4	21.5	20.5	20.0
J-03	19.6	19.6	19.9	21.1	20.8	20.6	20.6	20.5	20.6	19.4	19.2
J-04	20.1	20.3	20.5	21.6	22.0	21.9	21.9	21.9	21.9	21.1	20.8
J-05	19.2	19.2	19.5	20.8	21.4	20.9	20.9	20.8	20.7	20.1	19.7
J-07	19.6	19.5	19.9	20.8	21.2	21.2	21.2	21.2	21.2	20.5	20.0
J-09	19.6	19.7	20.0	21.3	21.7	21.6	21.7	21.7	21.7	20.9	20.9
KC-1	20.3	20.3	20.5	21.5	22.0	21.9	21.9	21.9	22.0	21.2	20.6
KC-4	19.2	19.2	19.5	20.7	20.3	20.1	20.0	20.0	20.0	19.4	19.2
KC-7	20.2	20.3	20.6	22.1	22.1	22.0	21.9	21.9	21.9	21.2	21.1
KC-9	19.3	19.3	19.5	20.8	21.4	21.3	21.3	21.3	21.3	20.6	20.4
M4	19.2	19.3	19.6	20.5	20.2	19.9	19.9	19.8	19.8	19.2	18.9
M5	19.1	19.1	19.2	20.0	20.0	19.9	19.8	19.7	19.7	19.1	19.0
RJN4	18.3	18.3	18.7	18.9	18.6	18.5	18.5	18.5	18.5	18.1	18.1
RJN10	19.6	19.6	19.8	20.8	20.8	20.5	20.2	19.9	20.0	19.0	19.1
Z3	19.7	19.8	19.9	21.0	21.0	20.8	20.8	20.7	20.8	20.1	19.8
Z4	19.8	19.9	20.3	22.2	22.1	22.0	21.9	21.8	21.9	21.1	21.2
Z6	19.3	19.3	19.5	20.6	20.8	20.7	20.7	20.7	20.7	19.8	19.4
Z7	19.6	19.6	20.0	21.0	20.9	20.5	20.4	20.3	20.2	19.0	19.2
Z10	18.8	19.0	19.4	20.0	19.9	19.5	19.5	19.5	19.7	19.1	18.9
Z13	19.5	19.5	19.9	21.0	21.1	21.0	21.0	21.0	21.0	20.1	19.7
Z14	19.5	19.5	20.0	21.3	21.4	21.2	21.1	21.1	21.1	19.9	19.4
Z16	19.4	19.4	19.7	21.2	20.1	19.9	19.9	20.0	20.1	19.2	19.0
Z18	19.3	19.3	19.9	21.3	20.7	20.3	20.1	20.0	20.0	19.1	18.8

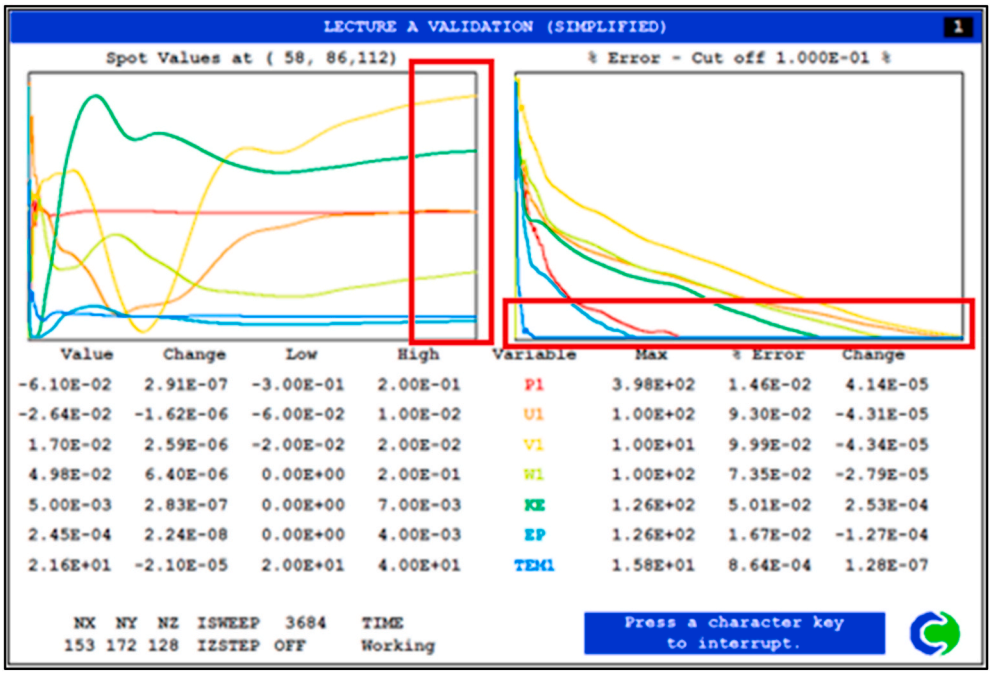

Fig. A.1. Solver results from the CFD validation model for lecture A, showing the monitored values (P1,U1,V1,W1,KE,EP and TEM1) at selected spot in the space (left) and residual errors (right). The red rectangular annotations show the monitored values achieving the steady state condition (left) and residual errors dropping below required thresholds (right).

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