

VALIDATED CFD STUDY OF INDOOR ENVIRONMENTAL CONDITIONS IN A HIGHLY GLAZED, CROSS-VENTILATED MEETING ROOM

Magdalena Hajdukiewicz^{1,2,3}, Marco Geron^{1,2,3}, and Marcus M. Keane^{1,2,3}

¹Civil Engineering, National University of Ireland Galway, Ireland

²Ryan Institute, National University of Ireland Galway, Ireland

³Informatics Research Unit for Sustainable Engineering (IRUSE) Galway, Ireland

Corresponding author: m.hajdukiewicz1@nuigalway.ie

ABSTRACT

This paper investigates the environmental conditions inside a highly-glazed cross-ventilated meeting room. A 3D computational fluid dynamics (CFD) model of an indoor environment is developed with the support of the field measurements performed in a normally operating room.

The work presented here follows the steps of the formal calibration methodology for the development of CFD models of naturally ventilated environments. This paper utilises the calibration methodology in order to predict environmental conditions within the highly-glazed cross-ventilated room occupied by people. The CFD model is verified and validated with field measurements performed in an operating building. Moreover, parametric analysis determines the most influential boundary conditions on indoor air temperatures and air speeds.

INTRODUCTION

The building sector is responsible for up to 40% of the total energy consumption in the European Union (European Union, 2012). Half of the energy consumed by buildings is due to the use of heating, ventilation and air conditioning systems (Pérez-Lombard et al., 2008). Recently, natural ventilation has been actively promoted as an effective solution to save energy in buildings (e.g. (Build Up, 2012), (SEAI, 2012)). Well planned natural ventilation systems provide fresh air movement, leading to comfortable and healthy indoor conditions. However, an effective design of a naturally ventilated space requires a good understanding of the complexity and difficulty to predict airflow patterns inside and outside the building (Allocca et al., 2003).

For the last 50 years the development of computational fluid dynamics (CFD) technology has been remarkable. CFD has become progressively more popular and accessible for research and industry sectors, mainly because of the improvements in the computational power and availability of commercial software. The ability of CFD in dealing with complex flows within built environments has become very important to provide safe, healthy and comfortable conditions for the occupants, test energy efficient designs, or apply specific environmental requirements. However, the accuracy of the CFD

predictions is a big concern (AIAA, 1998; Mehta, 1998; Pelletier, 2010). For this reason, the reliability of CFD simulations depends on the modeller's knowledge in fluid dynamics, expertise to handle complex boundary conditions and skills in numerical techniques. Furthermore, the reliable CFD model should be created using verified software, systematic grid refinement studies and experimental data to support model validation (Pelletier, 2010).

The study presented here follows the steps of the formal calibration methodology for the development of CFD models of naturally ventilated environments (Hajdukiewicz et al., 2013). The methodology systematically explained how to verify and validate computational models; and perform parametric analysis in order to evaluate the most influential boundary conditions on model results. This article investigates indoor conditions within the highly-glazed cross-ventilated meeting room in a normally operating building. This is done by the verification of the CFD model and validation with field measurements. Moreover, a parametric analysis aided in estimating the variation of which boundary conditions had the strongest impact on indoor air temperatures and air speeds.

FIELD MEASUREMENTS

Overview of the building

The demonstrator used in this research was the Engineering Building at the National University of Ireland (NUI) Galway, Ireland (figure 1). With a gross floor area of 14 100 m², the Engineering Building is the largest school of engineering in Ireland. The building was designed as a 'living laboratory' in order to provide real-time measured data of structural and environmental building performance (NUI Galway, 2012).



Figure 1 Engineering Building at NUI Galway

A naturally ventilated meeting room (figure 2) on the top floor of the Engineering Building was chosen in order to study indoor environmental conditions. The dimensions of the room are 4.90 m (D) x 5.89 m (L) x 3.47 m (H). The room is cross-ventilated with two external walls consisting of windows (facing north and east directions). One internal wall is a concrete structural wall adjacent to the internal staircase. The second internal wall is a partition wall adjacent to an office space.



Figure 2 Modelled meeting room

Outdoor measurements

The measurement of outdoor weather conditions was facilitated by the automatic weather station located at the NUI Galway campus, approximately 500 m from the Engineering Building (IRUSE Weather, 2011). The outdoor measurements of dry bulb air temperature, solar irradiance and barometric pressure were taken with a time step of 1 min. The accuracies of the weather sensors were as follows: dry-bulb air temperature $\pm 0.13^{\circ}\text{C}$; barometric pressure ± 0.5 mBar ($+20^{\circ}\text{C}$), ± 1.0 mBar (0°C to 40°C), ± 1.5 mBar (-20°C to $+50^{\circ}\text{C}$), ± 2.0 mBar (-40°C to $+60^{\circ}\text{C}$); global solar irradiance ± 5 $\text{W}/\text{m}^2 \pm 12\%$; diffuse solar irradiance ± 2.0 $\text{W}/\text{m}^2 \pm 15\%$.

Indoor measurements

For the CFD model validation purposes indoor air temperatures and air speeds were measured using fourteen wireless air temperature sensors (denoted by the letter S) (Onset Data Loggers, 2011) and three air speed Egg-Whisk sensors (denoted by letters EW) (NAP, 2008). Those air temperature sensors were located in four horizontal layers (ankles level $h = 0.1$ m; sitting person's waist level $h = 0.6$ m; sitting person's head or standing person's waist level $h = 1.1$ m and standing person's head level $h = 1.7$ m) in order to observe the air temperature stratification inside the room that affected occupants' thermal comfort. The Egg-Whisk sensors were placed at three random locations inside the room to observe the airflow behaviour inside the room. Figure 3 shows the location of the sensors. The accuracy of the air temperature sensors was $\pm 0.35^{\circ}\text{C}$ in a range between 0°C and 50°C . The air speed sensors were capable to measure indoor convection air speeds between $0.05 - 1$ m/s with an accuracy of ± 0.01 m/s.

Two air speed sensors (Onset Data Loggers, 2011), located at the window inlet, provided the components of the inlet air velocity boundary condition for the CFD model (horizontal component parallel to the window plane (X) and vertical component (Y)). Moreover, an air speed sensor (Onset Data Loggers, 2011), located at the window outlet (V1) provided the measurement of the speed of air (vertical component) exiting the room, which was used for the validation purposes. The air speed sensors located at the window openings could measure air speeds between 0.15 m/s and 5 m/s, with an accuracy greater of 10% of reading or ± 0.05 m/s or 1% full-scale. Flags placed at both window openings provided an indication of the direction of the indoor flow (inlet – outlet).

The temperatures of internal walls were measured at one location for each wall by the surface temperature sensors (Onset Data Loggers, 2011). The floor temperature was measured at two locations inside the room by the temperature sensor (Onset Data Loggers, 2011). The accuracy of both temperature sensors was $\pm 0.25^{\circ}\text{C}$ in a range between 0°C and 50°C . The surface temperatures of the column, external walls and ceiling were measured using a thermal camera (FLIR, 2012). The measurement accuracy of the camera was $\pm 2^{\circ}\text{C}$ or 2% of the reading.

The measurements of indoor air/surface temperatures and air speeds at the windows were taken with a time step of 1 min; while, the indoor air speeds (EW) were measured by the Egg-Whisk sensors every second.

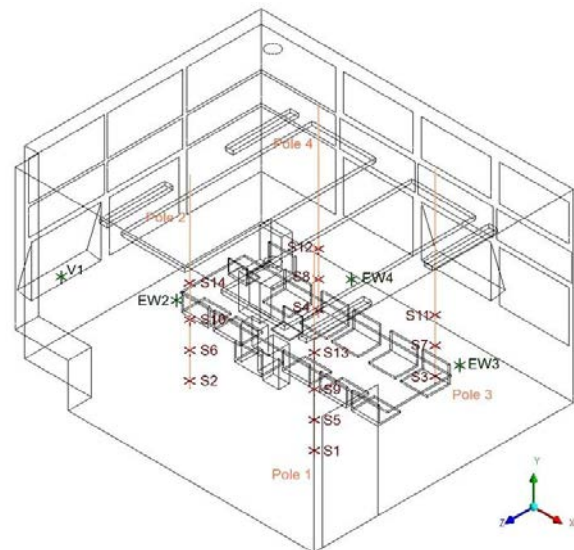


Figure 3 Measurement setup

CFD ANALYSIS

The CFD simulation of the meeting room was performed using the commercial software Ansys CFX v.13.0 (Ansys 13.0, 2012). The airflow and the air temperature stratification were simulated in a naturally ventilated room occupied by two people working on the laptops. The settings of the

CFD model imitated the conditions inside the room during the field measurements.

Geometry

The 3D geometry of the meeting room was created based on the technical drawings, operation and maintenance manuals and site visits. The elements of the modelled room, i.e. occupants, windows, chairs and tables, were simplified. The position of those elements was measured during the field experiment. Figure 4 shows the level of detail in the geometry of the modelled room.

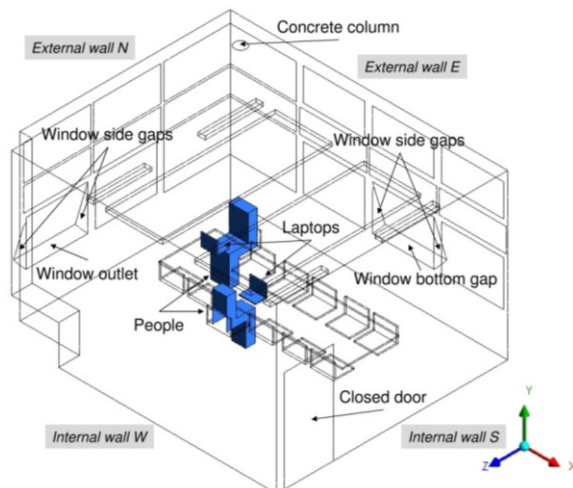


Figure 4 Geometry of the meeting room

Boundary conditions

The data supporting CFD simulations were gathered during the field measurements in the meeting room on August 17th, 2012. The external weather conditions, as well as indoor air speeds and air temperatures, were monitored throughout the day. The outdoor air entered the room through the open window (window bottom gap) and exited the room through another open window (window outlet). The meeting room was occupied by two sitting people working on the laptops. The conditions over a 45 minute period in the afternoon, when outdoor and indoor conditions were relatively steady, were chosen to be used in the CFD simulation. Two occupants of the room were constantly sitting during the 45 minute period. Their position was measured and any slight movements were not considered in the CFD model. Thus, the influence of uncertainties in position and movement of people on the simulation results was not analysed.

The average values over the 45 minute period provided the boundary conditions and validation data for the model. Furthermore, the variations of the measurements were used to define the ranges of input boundary conditions for the parametric analysis. Table 1 presents the boundary conditions for the CFD model of the meeting room.

Table 1
Boundary conditions for the CFD model

Boundary	Type	Heat transfer	Mass & momentum	Radiation
Window bot. gap	Inlet	$T_o = 20.82$ [°C]	$V_x = 0.47$ [m/s] $V_y = 0.56$ [m/s]	n/a
Window side gaps	Open.	$T_o = 20.82$ [°C]	$P_{rel} = 0$ [Pa]	n/a
Window outlet	Open.	$T_o = 20.82$ [°C]	$P_{rel} = 0$ [Pa]	n/a
Internal door	Wall	Adiabatic	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Windows	Wall	$h_c = 2.30$ [W/m ² K] $T_o = 20.82$ [°C] $q_{rad} = 203.10$ [W/m ²]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Internal wall S	Wall	$T_{wall} = 23.64$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Internal wall W	Wall	$T_{wall} = 24.45$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
External wall N	Wall	$T_{wall} = 22.6$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
External wall E	Wall	$T_{wall} = 22.6$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Column	Wall	$T_{wall} = 22.8$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Ceiling	Wall	$T_{wall} = 23.0$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Floor	Wall	$T_{wall} = 23.5$ [°C]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
People	Wall	$q_{con} = 18$ [W/m ²] $q_{rad} = 42$ [W/m ²]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Laptops	Wall	$Q_{con} = 131$ [W]	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$
Other surfaces	Wall	Adiabatic	No slip wall	$\varepsilon = 0.9$ $DF = 1.0$

The meeting room is a highly glazed space, with windows covering the majority of external walls. Thus, the radiation model was included in the CFD simulation. A discrete transfer model (assumption of an isotropic scattering and reasonably homogenous system) was chosen in the analysis, since it is very efficient and provides accurate results (Ansys, 2012). For the period monitored the average global solar irradiance was 445.63 W/m² and the average diffuse solar irradiance was 203.10 W/m² (IRUSE Weather, 2011). The windows of the meeting room are facing north and east directions; and the field measurements, to support the CFD model, were performed in the afternoon (no direct sun light was present in the room for the period monitored). Thus, only a diffuse part of measured solar irradiance was considered in the CFD model. Moreover, the amount of solar irradiance transmitted through the window depends on the transmittance of the glass. In this case, the glass transmittance was not known. Thus, it was assumed that 100% of diffuse solar irradiance was transmitted inside the room. However, the variation of solar irradiance through the period monitored was included in the parametric analysis.

The diffuse solar irradiance was measured on a horizontal surface (received from the entire hemisphere, i.e. 180° field of view). The calculation of average (over period monitored) diffuse solar irradiance and approximate reflected solar irradiance (ground reflectivity was assumed 20% (ASHRAE, 2005)) on a vertical window was not significantly

different to the measured diffuse solar irradiance on a horizontal surface. Thus, in order to simplify the analysis, the average diffuse solar irradiance measured on a horizontal surface was used as a boundary condition for the vertical window.

The heat sources inside the room consisted of two people working on their laptops. The convective and radiative heat fluxes from the people were set to 18 W/m² and 42 W/m² respectively (recommended convection to radiation ratio of 30:70 (Srebric et al., 2008)). The laptops generated 131 W each. The surface temperatures of internal walls and the floor in the model were specified based on the average measurements taken with the temperature sensors (Onset Data Loggers, 2011).

The surface temperatures of the ceiling, external walls and the concrete column were specified based on the thermal images. The closed internal door, chairs, table, lamps and sound panels were all assumed adiabatic. The emissivity and diffuse fraction (a ratio between the diffuse reflected energy and the total reflected energy at an opaque boundary) of all surfaces inside the room were assumed 0.9 and 1.0 respectively.

The outdoor air entered the room through the velocity inlet at the 0.27 m wide gap of the window awning. Two air velocity components were measured at the centre of the gap (X component – horizontal and parallel to the window plane, Y component - vertical) and the model's boundary condition was specified as the average measured values, constant at the 'window bottom gap' velocity inlet. The air exited the room through the window outlet opening (0.27 m wide gap), which was specified with no pressure difference between the boundary and outdoor conditions. Window side gaps (vertical triangular planes formed after opening the window) were modelled as openings at the same reference pressure as the outdoor conditions. The influence of the airflow through the window side gaps was not expected to be significant for the airflow inside the room.

Model setup

The steady state conditions were used in the CFD analysis of the single phase airflow inside the meeting room. The full buoyancy model, where the fluid density is a function of temperature or pressure, was applied. The air was modelled as an ideal gas with the reference buoyancy density of 1.173 kg/m³ (calculated based on the ideal gas law and field measurements). A blend factor of 0.75 was applied in the model in order to maintain robustness and accuracy of the results (Ansys, 2012).

The standard *k-ε* turbulence model was chosen for a good accuracy of the results with the robustness of the solution (Srebric et al., 2008; Ansys, 2012). Satisfactory convergence was achieved using the criteria of 0.01% of root mean square residuals for mass and momentum equations, 1% of the energy

conservation target and the numerical results at points of interest no longer changing with additional iterations. Simulations were performed with a single precision accuracy. The simulation with a double precision accuracy did not produce different results from the results of a single precision accuracy; thus, the round-off error was not considered significant in this case.

Model verification

The grid independence study of indoor air temperatures was performed in order to verify the CFD model solution. Three different meshes, successively refined, were created using unstructured elements. The number of grid elements (*A*) and the refinement ratio (*r*) for each mesh are presented in table 2. The recommended grid refinement ratio should be greater than 1.3 (Celik, 2008).

Table 2
Grid parameters for three different meshes

A_3	A_2	A_1	r_{32}	r_{21}
479 289	1 066 002	2 480 318	1.31	1.33

Figure 5 represents the qualitative grid verification. Axial air temperature profiles along the room height are plotted for three mesh sizes (A_3 , A_2 and A_1). Presented profiles indicate a very close prediction between the medium (A_2) and fine (A_1) meshes (except the lower locations at the pole 3, where the prediction of A_2 mesh is closer to A_3 than A_1). The coarse mesh (A_3) failed to predict accurately air temperatures along the poles 2, 3 and 4. Thus, the medium mesh (A_2) was chosen for further analysis, providing the balance between a good accuracy of the results and a reasonable computational cost.

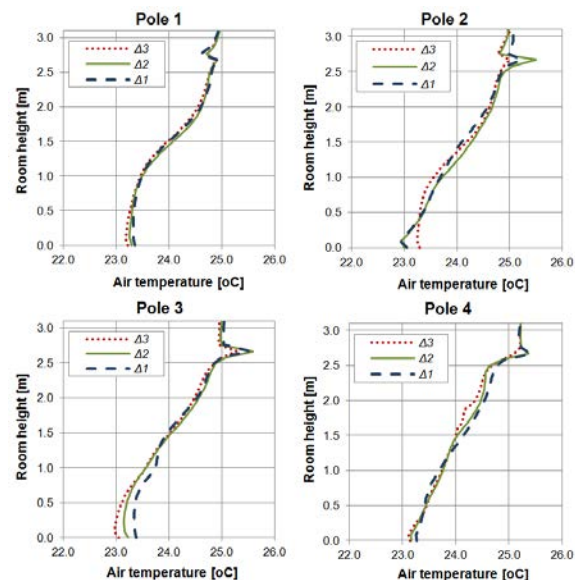


Figure 5 Simulated indoor air temperatures

Model results

Figure 6 shows the locations of three vertical planes: A and C (through the occupants), B (through the middle of the window inlet) are used to show the CFD model results. Figures 7 & 8 demonstrate the air

temperature stratification and airflow distribution inside the modelled room.

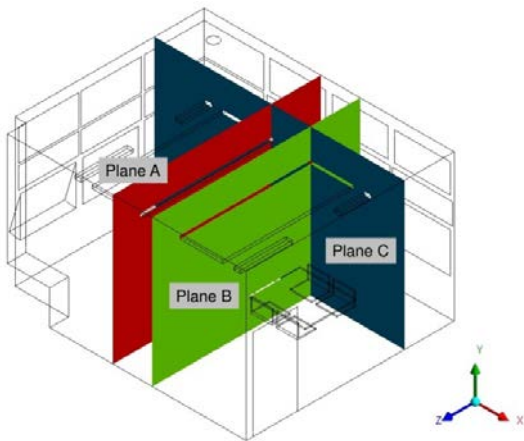


Figure 6 Location of the vertical planes

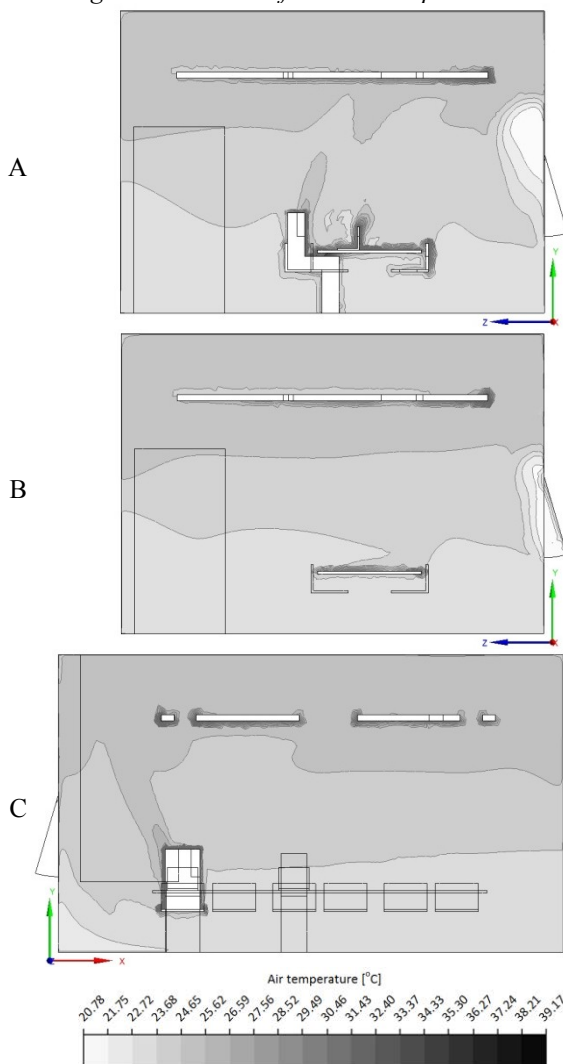


Figure 7 Verified model results of air temperatures

The results showed the airflow inside the room was strongly wind driven. The cold air entered the room through the window inlet (plane B) and exited through the window outlet. The heat plumes above the sitting people (planes A & C) were relatively small. This was due to two reasons: (i) it was a wind,

as opposed to buoyancy, driven flow and (ii) in the model the majority of the heat was transferred through radiation, as opposed to convection.

The average (over a particular plane) air speeds inside the room ranged from 0.14 m/s at the ankles level ($h = 0.1$ m); 0.08 m/s at the sitting person's waist level ($h = 0.6$ m); 0.13 m/s at the sitting person's head or standing person's waist level ($h = 1.1$ m); and 0.08 m/s at the standing person's head level ($h = 1.7$ m). Respectively, the average air temperatures were 23.11°C at the ankles level; 23.41°C at the sitting person's waist level; 23.61°C at the sitting person's head or standing person's waist level; and 24.16°C at the standing person's head level. The vertical air temperature differences between the head ($h = 1.1$ m) and ankles ($h = 0.1$ m) level were less than 2 °C. This caused a dissatisfaction level of only 6% (ISO, 2005).

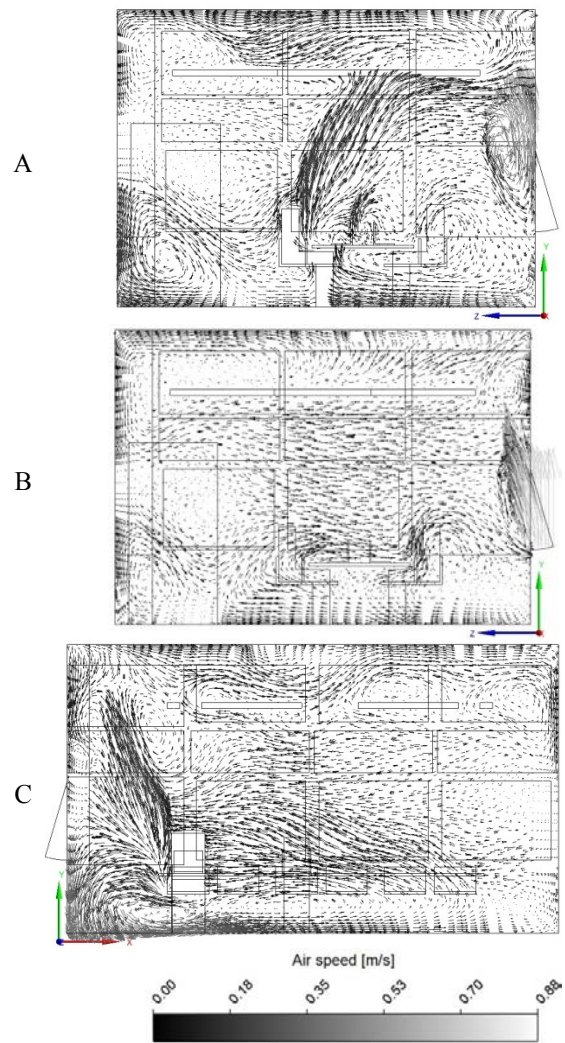


Figure 8 Verified model results of air velocities

Draughts are the common complaints in indoor environments, especially naturally ventilated. The previous research found that the level of occupant thermal acceptability at preferred temperatures was unaffected by air speeds of 0.25 m/s or lower (Berglund & Fobelets, 1987). Figure 9 presents an isosurface of indoor air speeds

of 0.25 m/s. It is clear from the figure 9 that the airflow jet generated at the window inlet entered the room almost vertically, dropped down through the corner of the room and aimed towards the window outlet near the floor level. This airflow jet omitted the occupants, except the ankles level of the person situated in the room corner. Moreover, there were two small airflow jets of 0.25 m/s air speed generated above the laptops with convective heat fluxes. There were no higher air speed jets created above the occupants. This was due to the fact that the heat generated by the occupants was transferred mostly through radiation, as opposed to convection.

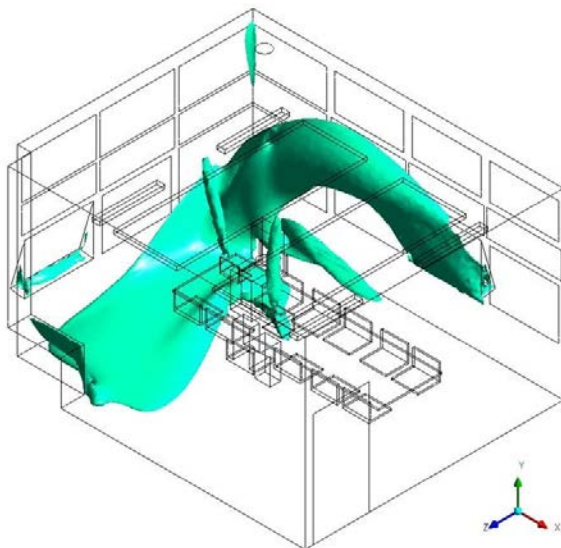


Figure 9 Indoor air speeds of 0.25 m/s

CFD MODEL VALIDATION

The first step of the validation procedure consisted of a qualitative comparison between the measured and simulated data (figures 10 & 11).

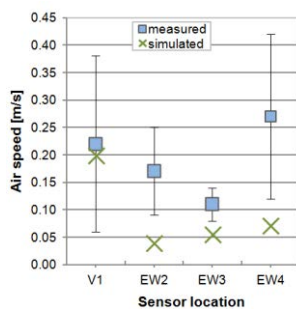


Figure 10 Qualitative validation of indoor air speeds (range of measured data shown)

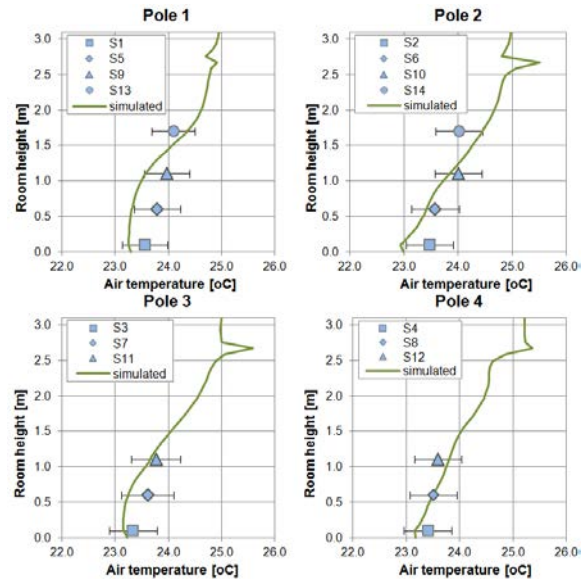


Figure 11 Qualitative validation of indoor air temperatures (range of measured data shown)

Office spaces require calculation of air speeds and air temperatures in order to evaluate the thermal comfort of the occupants. The validation criteria for this work were defined as 0.15 m/s absolute difference between measured and simulated air speeds; and 0.50°C absolute difference between measured and simulated air temperatures. The validation criteria were based on the maximum uncertainties in measured data (0.15 m/s for air speeds and 0.49°C for air temperatures). The uncertainties included the measurement uncertainty (standard deviation) and sensors' accuracy (± 0.01 m/s for air speeds and ± 0.35 °C for air temperatures).

Additionally, the air speed criterion was supported by the strictest design specification of the air velocity in buildings (the mean air velocity in a single office or kindergarten during the winter season may not exceed 0.10 m/s) (ISO, 2005). The criterion for air temperatures was supported by the allowable peak-to-peak variation in indoor air temperatures, which, if less than 1K, causes no influence on the occupants' comfort (ISO, 2005). Also, the maximum indoor operative temperature change allowed in a 15 minute period is 1.1°C (ANSI/ASHRAE, 2004).

Table 3 shows the absolute differences between measured and simulated indoor air speeds and air temperatures. The model met the validation criterion when predicting air temperatures (except the location S2, where the air temperature was slightly under predicted). At the EW4 Egg-Whisk location, which was the closest to the window inlet, the model failed to predict air speeds accurately. However, despite the excess over the validation criterion, the difference between the measured and simulated value at the location EW4 was not extremely high; and may be considered acceptable close to the window inlet, where the air speed underprediction by the model might have been caused by the simplification of

the inlet boundary condition and the assumption of the uniform conditions along the window gap.

Table 3

Quantitative model validation
(values in bold red exceed validation criteria)

Data type	Location	Measured	Simulated	Absolute difference
Air speed [m/s]	V1	0.22	0.20	0.02
	EW2	0.17	0.04	0.13
	EW3	0.11	0.06	0.05
	EW4	0.27	0.07	0.20
Air temperature [°C]	S1	23.56	23.26	0.30
	S2	23.48	22.95	0.53
	S3	23.34	23.16	0.18
	S4	23.40	23.16	0.24
	S5	23.79	23.31	0.47
	S6	23.58	23.42	0.16
	S7	23.61	23.25	0.36
	S8	23.50	23.49	0.01
	S9	23.97	23.57	0.40
	S10	24.00	23.84	0.16
	S11	23.76	23.65	0.11
	S12	23.59	23.80	0.21
	S13	24.09	24.35	0.26
	S14	24.02	24.24	0.42

PARAMETRIC ANALYSIS

In order to perform parametric analysis, the plausible ranges of eight input boundary conditions were assessed (table 4). Those ranges were related to the variation in the measurements and uncertainty in assumptions of the boundary conditions. The range of the air density was calculated using the ideal gas law equation based on the minimum and maximum expected indoor air temperatures. The ranges of the outdoor air temperature, inlet air velocity components (vertical and horizontal) and solar radiation were based on the average measurements \pm standard deviation in air temperature, solar irradiance (weather station) and air speeds (window inlet). The ranges of the convective and radiative heat fluxes from the people were specified as the average values \pm 20%. The range of the heat generated by the laptops was based on the manufacturer's specification \pm 20%.

Table 4

Input boundary conditions and their ranges

Input boundary conditions	Lower bound	Upper bound
Air density	1.163 [kg/m ³]	1.183 [kg/m ³]
Outdoor air temperature	20.63 [°C]	21.02 [°C]
Inlet air velocity (vertical component)	0.38 [m/s]	0.73 [m/s]
Inlet air velocity (horizontal component, parallel to the window plane)	0.30 [m/s]	0.64 [m/s]
People convective heat flux	14.4 [W/m ²]	21.6 [W/m ²]
People radiative heat flux	33.6 [W/m ²]	50.4 [W/m ²]
Laptops convective heat source	105 [W]	157 [W]
Solar radiation at windows	135.74 [W/m ²]	270.46 [W/m ²]

The eight input boundary conditions were varied within their ranges, which generated a set of 81 CFD

models to be solved. Changing the inputs within their ranges caused changes in the output indoor air speeds up to 0.11 m/s from the base model and indoor air temperatures up to 0.72°C.

The vertical component of inlet air velocity influenced the air speeds inside the room the most. Generally, higher the inlet air velocity vertical component, higher were the indoor air speeds (except the location EW3, where the measured air speed was inversely proportional to the inlet air velocity vertical component).

The indoor air temperatures were mostly influenced by the inlet velocity vertical component (inversely proportional), solar radiation through the windows, inlet velocity horizontal component and outdoor air temperature (all proportional).

CONCLUSIONS

Properly designed and operated naturally ventilated indoor environments can provide healthy and comfortable environmental conditions, while contributing to a significant reduction in the energy consumed by buildings. However, the practical modelling of natural ventilation systems is not straightforward. The accurate and robust CFD models may contribute in better understanding of the indoor airflow patterns and temperature stratification.

This paper presents the development of a valid CFD model in order to investigate environmental conditions in a highly-glazed cross-ventilated meeting room. The work shown here follows the steps of previously proposed calibration methodology for the development of reliable CFD models of naturally ventilated environments. The CFD model presented here is verified and validated with field measurements. Furthermore, since the results of the CFD simulations highly depend on the model parameters input by the user, a parametric analysis aided in estimating the variation of which boundary conditions had the strongest impact on indoor air temperatures and air speeds.

In the future work, the calibrated CFD model will be applied to analyse thermal comfort of the room occupants. The predicted mean vote (PMV), predicted percentage of dissatisfied (PPD) and local discomfort indices will be evaluated. This will be done in order to investigate if operation of the meeting room is satisfactory in terms of thermal environment. Moreover, calibrated CFD models can be used in the design and deployment of wireless sensor networks in buildings. This can improve understanding, support design and operation, as well as retrofit processes in naturally ventilated buildings.

NOMENCLATURE

T_o = outdoor temperature

T_{wall} = wall temperature

h_c = heat transfer coefficient

q_{con} = convective heat flux

q_{rad} = radiative heat flux

Q_{con} = convective heat source;

V_x = air speed horizontal component parallel to the windows' plane

V_y = air speed vertical component

P_{rel} = pressure relative to the reference pressure ($P_{reference} = 100200$ Pa - absolute pressure datum from which all other pressure values were taken)

ε = emissivity

DF = diffuse fraction

ACKNOWLEDGEMENT

The Irish Higher Education Authority (HEA) funded this research through the NEMBES Project.

REFERENCES

- AIAA, 1998. *Guide for the verification and validation of Computational Fluid Dynamics simulations*, American Institute of Aeronautics and Astronautics.
- Allocca, C., Chen, Q. & Glicksman, L.R., 2003. Design analysis of single-sided natural ventilation. *Energy and Buildings*, 35(8), pp.785–795.
- ANSI/ASHRAE, 2004. Standard 55: *Thermal Environmental Conditions for Human Occupancy*, American Society of Heating, Refrigerating and Air-Conditioning Engineers.
- Ansys 13.0, 2012. Ansys Inc. Available at: <http://www.ansys.com/>.
- Ansys, 2012. *Ansys, Inc.* Available at: <http://www.ansys.com/>.
- ASHRAE, 2005. *ASHRAE Handbook - Fundamentals*, Atlanta, USA: American Society of Heating, Refrigerating and Air-Conditioning Engineers.
- Berglund, L.G. & Fobelets, A.P.R., 1987. Subjective human response to low-level air currents and asymmetric radiation. *ASHRAE Transactions*, 93(1), pp.497–523.
- Build Up, 2012. *Energy solutions for better buildings*. Available at: <http://www.buildup.eu/>.
- Celik, I.B., 2008. Procedure for estimation and reporting of discretization error in CFD applications. *In Statement on the Control of Numerical Accuracy of the Journal of Fluids Engineering (Editorial Policy)*, ASME Fluids Engineering Division.
- NUI Galway, 2012. Available at: <http://www.nuigalway.ie/new-engineering-building/>.
- European Union, 2012. Available at: <http://www.europa.eu>.
- FLIR, 2012. Available at: <http://www.flir.com/IE/>.
- Hajdukiewicz, M., Geron, M. & Keane, M.M., 2013. Formal calibration methodology for CFD models of naturally ventilated indoor environments. *Building and Environment*, 59(0), pp.290–302.
- IRUSE Weather, 2011. Available at: <http://weather.nuigalway.ie/>.
- ISO, 2005. ISO 7730: *Ergonomics of the thermal environment - Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria*, International Organization for Standardization.
- Mehta, U.B., 1998. Credible computational fluid dynamics simulations. *AIAA Journal*, 36(5), pp.665–667.
- NAP, 2008. Available at: <http://www.tyndall.ie/nap/>.
- Onset Data Loggers, 2011. Available at: <http://www.onsetcomp.com/>.
- Pelletier, D., 2010. Verification, validation, and uncertainty in Computational Fluids Dynamics. *Canadian Journal of Civil Engineering*, 37(7), pp.1003–1013.
- Pérez-Lombard, L., Ortiz, J. & Pout, C., 2008. A review on buildings energy consumption information. *Energy and Buildings*, 40(3), pp.394–398.
- SEAI, 2012. *Sustainable Energy Authority of Ireland*. Available at: <http://www.seai.ie/>.
- Srebric, J. et al., 2008. CFD boundary conditions for contaminant dispersion, heat transfer and airflow simulations around human occupants in indoor environments. *Building and Environment*, 43(3), pp.294–303.