Evaluating the performance of different window opening styles for single-sided buoyancy-driven natural ventilation using CFD simulations

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Abstract
In many circumstances, natural ventilation is a cost-effective and efficient alternative to higher energy consuming mechanical ventilation systems to provide thermal comfort and good air quality. The aim of this research is to investigate the performances of different types of windows such as bottom hung, horizontal pivot and top hung fanlight for single-sided natural ventilation driven by buoyancy forces. CFD simulations, combining RANS model and SST k-ω turbulence models, were used to numerically evaluate the different window opening styles for the weather conditions of South Tyrol, Italy. In addition to the much-studied parameters of ventilation effectiveness, thermal comfort parameters have been considered based on standards such as EN ISO 7730 and ASHRAE 55, for comparing the performances of these selected windows. The results indicate that horizontal pivot window performs the best among the 3 types, because of its two-part structure ensuring better segregation. It provides 23% higher incoming airflow rate compared to bottom hung windows both in winter and summer conditions, but a 11% higher draught risk in winter.

Key Innovations
- Testing 3 different window opening types for buoyancy-driven single-sided natural ventilation flow
- Evaluating parameters of ventilation effectiveness and thermal comfort for different boundary conditions
- Testing different boundary conditions and geometrical simplifications for high quality CFD simulations

Practical Implications
It will give building designers, engineers and researchers a deeper understanding of:
- How to efficiently use CFD simulations to replicate buoyancy-driven single-sided natural ventilation flow
- How to choose from the different window opening styles based on parameters of ventilation effectiveness and thermal comfort

Introduction
Buildings are a major consumer of energy in all parts of the world. In the 28 EU Member States, household sector consumes up to 26% of the total final energy consumption (Eurostat, 2018), and similarly 28% in the residential and commercial sector in the United States (US EIA, 2019). The biggest proportion of this energy is used by Heating, ventilation, and air conditioning (HVAC) systems to provide ventilation, thermal comfort, and good air quality in the indoor environment (Bangalee et al., 2012). With more effort towards reducing dependence on fossil fuels, the focus is shifting towards natural resources and for promoting innovative renewable applications. Natural ventilation is being constantly recognized as an inexpensive and efficient alternative to the high energy consuming mechanical ventilation systems, but the air flow can be challenging to control (Allocca et al., 2003). An efficient natural ventilation system can provide not only good thermal comfort in the indoor environment but also improve air quality (Bangalee et al., 2012; Belleri et al., 2014; Von Grabe et al., 2014).

The performance of natural ventilation depends on various factors, such as driving force, opening configurations, number, and arrangement of openings, among other factors (Von Grabe et al., 2014; Wang et al., 2017). Moreover, the architectural arrangement of buildings can be an important factor in determining whether a room can have single-sided or cross ventilation through its openings, for example, residential buildings, hostels, dormitories etc. usually have openings only on one side. Furthermore, different window opening styles perform differently in different weather conditions (Allocca et al., 2003). Various experimental, analytical, and numerical approaches can be adopted to study the physical nature of natural ventilation. Using Computational Fluid Dynamics (CFD) to investigate the air and heat flow in buildings helps in early understanding of this physics (Cook et al., 2003). This further enables modifications and systematic optimization of the design and operating conditions (Cook et al., 2003). It is important to appropriately define the boundary conditions in case of natural ventilation flows since the velocities and pressure at the inlet and outlet opening can be affected by many internal and external factors. Some previous studies have been done to study buoyancy-driven natural ventilation in buildings, with different computational domain sizes and for different window configurations, but mainly investigating the performance of ventilation. Cook et al. (2003), and Allocca et al. (2003) investigated the combined effects of single-sided wind and buoyancy driven natural ventilation...
flow using CFD methods. Favarolo and Manz (2005) studied the impact of the temperature difference between outside and inside air on discharge coefficient, but specific to large rectangular openings. Larsen and Heiselberg (2008) predicted the airflow in single-sided natural ventilation based on parameters such as wind direction, temperature difference and wind speed together, including the effect from different incidence angles. Gan (2010) studied the use of large extended domain for simulating air flow for modelling buoyancy-driven natural ventilation around buildings. Wang and Chen (2012) developed a model to predict the fluctuating effects of single-sided natural ventilation flow driven by wind. Wang et al. (2017) investigated 6 different window configurations for single-sided natural ventilation driven by buoyancy, but mainly to estimate the ventilation rates. However, there is a lack of studies on the evaluation of thermal comfort parameters, and there is a need to understand how both air flow rates as well as comfort conditions are affected by the configuration of window opening and indoor-outdoor temperature differences. The objective of this study is to explore three specific window typologies, and to compare their performance for single-sided natural ventilation for buoyancy-driven flow in the context of South Tyrol, Italy. This study shall focus on performance indicators of ventilation effectiveness, window characterization and thermal comfort based on EN-ISO 7730 and American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) 55 standards, which are discussed in the next section.

Methodology

A CFD study is carried out for testing different window opening configurations, to evaluate the flow rates based on different internal-external temperature differences. In this section, the model configuration and geometrical assumptions considered for the CFD analysis are presented, followed by the parameters considered to compare the performance of different window opening types. A grid independence study is also carried out to minimize the numerical errors and obtain a reasonably fine mesh to capture the correct physics of the wind flow.

Geometric model and discretization

In natural ventilation flows, the airflow is affected by various factors such as window opening style, temperature differences, heat sources and geometry of the room. The computational domains were modelled following the conclusions of Wang et al. (2017), to accurately replicate the conditions for a single-sided buoyancy driven flow, while at the same time keeping the computational time in a reasonable range.

For the building geometry, the dimensions of the chamber, to be later used for experimental validation, were chosen (4 m x 8 m x 3 m). The geometry and mesh were created with ANSYS ICEM and the CFD simulations were performed using ANSYS CFX (both 2019 R3 version). Firstly, watertight geometries are created with the same dimensions of the chamber and different window opening types. To simulate outdoor conditions for buoyancy-driven natural ventilation, an external domain, same size as the chamber, is placed in front of it. As shown in Figure 1, the external domain is represented on the left side, and the chamber on the right, such that the window opening is practically in the middle of the entire 3D geometry.

Secondly, the discretization of the domains is done following an unstructured meshing technique for better flexibility. The maximum mesh size is set-up in the software with attention to specific elements of the domain to maintain a good grid size. In general, mesh fineness is increased at critical areas, that is, the mesh is finer around the opening, less coarse for the chamber, and coarser for the external domain. The domains are first meshed using Octree method, though, only the surface triangulations are retained. Emerging from these surface triangles, the tetrahedrons are produced throughout the volume using Delaunay triangulation algorithm, followed by ten prism layers at the no-slip walls for better orthogonality.

The three different window configurations considered in this study have been chosen after a preliminary market analysis, based on their likely performance, high market potential and applicability in the context of South Tyrol (Italy). Figure 2 illustrates the 3 window types placed on the front façade of the chamber, all opened 20°. The Type_1 and Type_2 cased have an openable window area of 1190 x 1450 mm, whereas, for the Type_3 top hung fanlight window the openable top area is 1190 x 450 mm, and the part below is fixed.
Basic CFD principles
The chosen CFD software uses a parallel, implicitly coupled multigrid solver. Each set of grids is analyzed in transient condition (120 seconds) using steady-state solutions for initialization, for a buoyant and thermal energy transfer model, with an adaptive time step as a function of root-mean-square (RMS) Courant number of 5. During the transient simulation, the time step size was allowed to vary between 10 and 0.05 seconds, ensuring time steps to complete within 20 coefficient loops to keep the RMS residual target. Boussinesq approximation is used as the fluid is Newtonian (air) with constant properties and the buoyancy-driven flow is incompressible with relatively small density differences due to changes in temperature (Cook et al., 2003; Wang et al., 2017). The Boussinesq model is applied by setting the reference buoyancy temperature equal to the window glass temperature, as an approximate average temperature of the two domains, and applying a local gravitational force throughout the fluid (ANSYS Inc., 2013). The convergence criteria were set at 1e-05 for the RMS residuals and 0.01 for the conservation target (Babich et al., 2017). The airflow characteristics for the computational domains are solved combining Reynolds-averaged Navier–Stokes (RANS) model and Shear Stress Transport (SST) k-omega turbulence model, which is a hybrid model combining the Wilcox k-omega and the k-epsilon models. It is one of the most effective models as it has been well tested for accuracy for many aerodynamic applications (ANSYS Inc., 2013; Babich et al., 2017). The governing equations for this model are discussed in detail by Menter (1994) and ANSYS Inc. (2013). All simulations were performed on a workstation of 16 GB RAM and a 6-core Intel Xenon Gold 6154 CPU.

Grid independence study
The grid independence study is carried out for Type_1 case only, since the other window typologies differ only in the opening style, while the domain broadly remains the same. For this study, the recommendations of Celik et al. (2008) are followed. Since the cell size varies throughout the domain, a representative cell size h is calculated using equation (1).

\[
h = \left[ \frac{1}{N} \sum_{i=1}^{N} (\Delta V_i)^{1/3} \right]^{1/3} \tag{1}
\]

where \(\Delta V_i\) is the volume of the \(i^{th}\) cell, and \(N\) is the total number of cells in the domain used for the computations. In order to ensure significant difference in the sizes of grids, a grid refinement factor, \(r = h_{\text{coarse}}/h_{\text{fine}}\), of at least 1.3 has been suggested by Celik et al. (2008). An iterative procedure is followed, by changing the maximum mesh size to obtain the right number of total cells and thus have a representative cell size 1.3 times smaller than the previous (coarser) mesh. Different set of grids are created for the same geometry, with increasing fineness of the mesh and hence increasing total number of cells.

The following boundary conditions were assigned (for the purpose of mesh independence study only):
- All chamber surfaces are no-slip smooth walls, at 20°C, except the floor which is at 23°C
- The window glass is a no-slip smooth surface at 14°C
- The window opening is an interface between indoor and outdoor domain
- The external domain is set at 10°C, with opening boundary condition of 0 Pa pressure imposed on the top surface, while all other surfaces as no-slip walls
- An under-door opening at 0 Pa relative pressure and 20°C on the opposite wall of the chamber.

Each test grid is then analyzed for the same boundary conditions for a residual RMS error value of 1e-05 in transient state (60 seconds). Three monitoring point are chosen in the indoor domain to record the temperature, velocity, and pressure values for each iteration of all the simulations.

(a) Velocity at monitoring points 1,2,3

(b) Temperature at monitoring points 2,3

Figure 3: Grid independence results
The Figure 3 shows the graphs for temperature and velocity approaching an asymptote after around 3 million elements. The computational time is around 8.5 hours for test with 3.3 million elements, whereas, 19.6 hours for the one with 7.1 million elements. Thus, we can assume a basic mesh independence at test with 3.3 million elements as the simulation time and computational power is significantly saved while accurately capturing the physics of the airflow, and further studies for buoyancy-driven flow will be carried out with similar maximum mesh size.

**Boundary conditions**

The boundary conditions for all the surfaces in the domains are summarized in Table 1.

<table>
<thead>
<tr>
<th>Location</th>
<th>Type</th>
<th>Winter</th>
<th>Summer</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a) Chamber</td>
<td>Window Glass</td>
<td>14°C</td>
<td>28°C</td>
</tr>
<tr>
<td>Walls</td>
<td>No-slip</td>
<td>20°C</td>
<td>30°C</td>
</tr>
<tr>
<td>Ceiling</td>
<td>No-slip</td>
<td>20°C</td>
<td>30°C</td>
</tr>
<tr>
<td>Floor</td>
<td>No-slip</td>
<td>23°C</td>
<td>30°C</td>
</tr>
<tr>
<td>Window Glass</td>
<td>No-slip</td>
<td>14°C</td>
<td>28°C</td>
</tr>
<tr>
<td>(b) External Domain</td>
<td>Top</td>
<td>Opening, 0 Pa</td>
<td>10°C</td>
</tr>
<tr>
<td></td>
<td>Ground</td>
<td>No-slip</td>
<td>10°C</td>
</tr>
<tr>
<td></td>
<td>3 Sides</td>
<td>No-slip</td>
<td>10°C</td>
</tr>
<tr>
<td></td>
<td>Window Glass</td>
<td>No-slip</td>
<td>14°C</td>
</tr>
</tbody>
</table>

To replicate the external conditions most reasonably for a single-sided buoyancy driven flow, the suggestions of Wang et al. (2017) have been followed, and the top boundary of the external domain has been set up as an opening. The weather conditions most representative to the climate of Bolzano, South Tyrol (Italy) have been chosen.

**Performance parameters**:

For comparing the performance of the windows, the following parameters are considered:

1. Temperature distribution – The temperature profiles are presented at the vertical plane in the middle of the geometry. They are presented with ranges set up between initial outdoor and indoor temperature, that is, 10°C to 20°C for winter (on the left) and 25°C–30°C for summer (on the right).
2. Airflow fields – The airflow fields are presented at the same plane, showing the speed and direction of the flow and are set up at a range of 0–0.6 m/s.
3. Incoming airflow rate (Q) – The airflow rate of the air entering the room through the window opening.
4. Mean Age of air (\( \Delta t \)) – It is the time elapsed by the air entering a space through an opening in the envelope until it reaches a particular point in the room. A scalar quantity is defined in ANSYS CFX software to calculate the local age of air (Ning et al., 2016).
5. Air changes per hour (ACH) – The number of times air is added or removed from a space. It is calculated as the ratio between incoming airflow rate to the volume of the space.

\[
ACH = \frac{Q}{V}
\]  

(2)

where, \( Q \) is the incoming airflow rate through the opening (m³/s), and \( V \) is the space volume (m³).

6. Effective penetration depth – The longitudinal distance traveled by fresh air from the inlet opening up to the position where the room air dominates. Theoretically, this value is calculated with a domain long enough to simulate the entering air without reaching the opposite end of the room, but for the purpose of this study and context in which the windows and climate are considered a maximum length of only 8m is considered.

7. Discharge coefficient (\( C_d \)) – A characteristic parameter to relate volume flow rate and pressure difference when a fluid flows through an opening. It depends on buoyancy, flow area and window type (Wang et al., 2017).

\[
C_d = \frac{Q}{\bar{A}} \sqrt{\frac{\rho}{2\Delta p}}
\]

(3)

where, \( Q \) is the incoming airflow rate through the opening (m³/s), \( \bar{A} \) is the opening area with incoming airflow (m²), \( \rho \) is the air density in the room (kg/m³) and \( \Delta p \) is the pressure difference between the opening area with incoming airflow and the indoor pressure in the room (Pa).

8. Temperature stratification – Taking considerations from ISO 7730 and ASHRAE 55, temperature difference along different planes are calculated to estimate the local thermal discomfort in winter: (i) Vertical air temperature difference between head level (from floor lever: 1.8 m for standing, and 1.1 m for sitting) and ankles (0.1 m above floor), and (ii) Horizontal air temperature difference at 0.5 m, and 1 m, 1.5 m and 2 m from the window.

9. Draught risk (DR) – The discomfort caused in winter due to draught is calculated using the equation (4) (ISO 7730):

\[
DR = (34 - t_a)(v_a - 0.05)^{0.62}(v_a \cdot T_a + 3.14)
\]

(4)

where, \( t_a \) is the local air temperature (°C), \( v_a \) is the local mean air velocity (m/s), and \( T_a \) is the local turbulence intensity (%).

10. Mean air velocity at levels – The average air velocity is calculated at levels 1.8 m, 1.1 m, and 0.1 m above the floor. The effect on perceived air temperature due to air velocity is verified based on ISO 7730.

11. Mean air velocity near surfaces – The average air velocity near the walls, ceiling and floor are calculated to compare convection effect. For this purpose, the velocity is averaged at a plane 5 cm away from the respective surface.
Results and Discussion
Figure 4 presents the temperature profiles of the three window types at the central vertical plane. The Type_1 and Type_2 cases show a well-distributed temperature stratification, and air with initial warm temperatures near the ceiling. It is worth noting that the temperature contours of both Type_1 and Type_2 are consistent with the thermal profiles at the central plane of Wang et al. (2017). The Type_3 case still maintains the initial temperature for a major part of the plane in both weather conditions due to lesser airflow, which is also evident in Table 2, as the mean temperature in the chamber is close to the initial temperature.

Table 2: Chamber temperature and velocity

<table>
<thead>
<tr>
<th>Type</th>
<th>Mean Temperature (°C)</th>
<th>Mean Air Velocity (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Winter: 17.91</td>
<td>0.110</td>
</tr>
<tr>
<td></td>
<td>Summer: 28.70</td>
<td>0.071</td>
</tr>
<tr>
<td>2</td>
<td>Winter: 17.44</td>
<td>0.113</td>
</tr>
<tr>
<td></td>
<td>Summer: 28.50</td>
<td>0.073</td>
</tr>
<tr>
<td>3</td>
<td>Winter: 19.85</td>
<td>0.088</td>
</tr>
<tr>
<td></td>
<td>Summer: 29.55</td>
<td>0.050</td>
</tr>
</tbody>
</table>

The airflow fields are shown in Figure 5. The Type_1 case has wind flowing inside through the two opposite side openings and warm air leaving the chamber from the top part. The Type_2 case, instead, has an airflow that is similar at the entry and exit due to its mirrored geometry at the center pivot point, which ensures better segregation of airflow and thus a higher velocity is observed. Also, the airflow fields for Type_1 and Type_2 are consistent with the results of Wang et al. (2017). The Type_3 case has a much lower opening area (0.23 m²) and very less amount of air is thus entering at low velocity, immediately descending down due to convection. Also, the Type_3 geometry with glass hanging outwards from the top side, does not support the outward passage of warm air well enough to facilitate air movement. The penetration depth in all cases is observed until the end of the chamber. The mean temperature and mean air velocity in the chamber are summarized in Table 2. The Type_1 and Type_2 cases show a higher velocity and higher temperature decrease as compared to Type_3.

Figure 6 presents the comparison of incoming airflow rate. The Type_1 case, despite its larger opening area (1.32 m²) as compared to Type_2 (0.96 m²), has a lower airflow rate. The Type_2 case, because of its two-part structure angled such that it opens both indoors and...
outdoors ensuring better segregation, shows a higher incoming airflow.

The comparison of mean age of air in Figure 7, corresponds directly to the incoming airflow rate. With the lowest airflow rate, the Type_3 case has the highest age of air. Air changes per hour is a function of incoming airflow rate and hence a similar pattern is observed in Figure 8. For Type_1 and Type_2, it is worth noting that the air changes per hour are in agreement with the findings of Wang et al. (2017). The discharge coefficients are presented in Figure 9. The Type_1 summer case recorded a high discharge coefficient because of low pressure difference between the opening area with incoming airflow and the indoor pressure in the room. For the Type_2, the discharge coefficient values are in same range as presented by Wang et al. (2017) in the relationship of discharge coefficients versus buoyancy effects and effective flow areas. And the Type_3 case values are lower due to its low flow rate.

The mean values of draught risk in winter, 7.56%, 8.46% and 5.22% for Type_1, Type_2 and Type_3 respectively, are acceptable in accordance with the Category A of ISO 7730, but is expectedly higher near the window due to lower temperature and higher airflow rate, as presented in Figure 10.

Table 3 presents the other thermal comfort parameters. The vertical temperature difference between head level (1.1 m while sitting and 1.8 m while standing) and ankles (0.1 m) are always lower than 2°C, and thus in accordance with Category A of ISO 7730. Similarly, horizontal temperature differences are quite low for all the windows. The air velocity at head and ankle levels are quite low to produce any positive effect on the perceived air temperature (ISO 7730). Table 4 presents a summary of

<table>
<thead>
<tr>
<th>Type</th>
<th>Air Temperature (°C)</th>
<th>Air Velocity (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Vertical difference at levels</td>
<td>Horizontal difference at distances from the window</td>
</tr>
<tr>
<td></td>
<td>1.1 – 0.1m</td>
<td>1.8 – 0.1m</td>
</tr>
<tr>
<td>1</td>
<td>Winter</td>
<td>0.71</td>
</tr>
<tr>
<td></td>
<td>Summer</td>
<td>0.36</td>
</tr>
<tr>
<td>2</td>
<td>Winter</td>
<td>0.48</td>
</tr>
<tr>
<td></td>
<td>Summer</td>
<td>0.33</td>
</tr>
<tr>
<td>3</td>
<td>Winter</td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>Summer</td>
<td>0.18</td>
</tr>
</tbody>
</table>
The results for each performance parameter with regards to each window type.

**Table 4: Summary of results**

<table>
<thead>
<tr>
<th>Performance parameters</th>
<th>Type_1 (Bottom hung)</th>
<th>Type_2 (Horizontal pivot)</th>
<th>Type_3 (Top hung fanlight)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Temperature distribution</td>
<td>Well distributed vertical temperature stratification.</td>
<td>Poor temperature distribution, maintains initial temperature.</td>
<td></td>
</tr>
<tr>
<td>2 Airflow fields</td>
<td>High speeds at inlet and outlet.</td>
<td>Low speeds.</td>
<td></td>
</tr>
<tr>
<td>3 Incoming airflow rate</td>
<td>High incoming airflow.</td>
<td>Very low incoming airflow. 83% lower than Type_1.</td>
<td></td>
</tr>
<tr>
<td>4 Mean Age of air</td>
<td>Low mean age of air.</td>
<td>Very high mean age of air.</td>
<td></td>
</tr>
<tr>
<td>5 Air changes per hour</td>
<td>Values analogous to incoming airflow</td>
<td>Maximum number of air changes per hour.</td>
<td>Minimum air changes.</td>
</tr>
<tr>
<td>6 Effective penetration depth</td>
<td>Similar to other types.</td>
<td>Similar to other types.</td>
<td>Similar to other types.</td>
</tr>
<tr>
<td>7 Discharge coefficient</td>
<td>Very high value in summer, due to low pressure difference.</td>
<td>High discharge coefficient in winter.</td>
<td>Lowest values in both seasons.</td>
</tr>
<tr>
<td>8 Temperature stratification</td>
<td>Vertical temperature difference at head and ankle level are quite similar in both standing and sitting positions.</td>
<td>Vertical temperature difference at head and ankle level while standing is more than double as compared to sitting.</td>
<td>Very low temperature differences in both seasons.</td>
</tr>
<tr>
<td>10 Mean air velocity at levels</td>
<td>High air velocity near ankle level. Difference is larger at different levels.</td>
<td>Lowest mean air velocity at different levels.</td>
<td></td>
</tr>
<tr>
<td>11 Mean air velocity near surfaces</td>
<td>High air velocity near floor surface.</td>
<td>Lowest mean air velocity near all surfaces.</td>
<td></td>
</tr>
</tbody>
</table>

**Conclusion**

The aim of this study was to investigate the effectiveness of different types of windows for single-sided buoyancy driven natural ventilation using CFD simulations. The initial conditions are correctly defined, and transient simulations provided the most accurate results as they can better capture the unsteady and cyclic nature of buoyancy-driven flows.

To explore the performance of windows, different parameters of ventilation effectiveness and thermal comfort were evaluated for winter and summer weather conditions. CFD has been used effectively to numerically analyze complex equations and have a thorough understanding of the characteristics of natural ventilation driven by buoyancy forces. Among the three windows studied, the Type_2 (Horizontal pivot) case supports the
highest airflow rate, 23% higher than Type_1 (Bottom hung) and provides the maximum air changes per hour in both winter and summer. However, in terms of thermal comfort parameters, Type_1 (Bottom hung) performs better, and maintains a lower draught risk as compared to Type_2 (Horizontal pivot) in winter conditions, which still maintains an acceptable value according to the comfort limits. Overall, the Type_1 (Bottom hung) case performed best for both ventilation performance and thermal comfort due to its two-part uniform structure, whereas the Type_3 (Top hung fanlight) case performed the worst due to its small size and thus lower opening area. Future studies will investigate these window types in full-scale experiments, and a more comprehensive validation of the CFD results shall be done. Moreover, additional research can be done with other window types and including other performance parameters such as energy efficiency.

Acknowledgement
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References