3D internal forced convection heat-transfer correlations from CFD for building performance simulation

A. Rincón-Casado and F. J. Sánchez de la Flor

ABSTRACT

This study presents new correlations for internal forced convective heat-transfer coefficients (CHTCs) in buildings. CHTCs are used by building energy performance simulation (BES) tools to estimate the annual heating and cooling demands of buildings. Most programs use constant values or correlations but do not consider in detail the 3D geometry or the dimensions and positions of the inlet and outlet openings. The correlations developed here take into account 3D geometry and the dimensions and positions of the inlet/outlet openings in opposite walls. A new methodology has been developed for the calculation of CHTCs in 3D enclosures from two 2D planes, one with a horizontal orientation and another with a vertical orientation. Comparing the results from correlations with those from computational fluid dynamics (CFD) simulations, the errors found are less than 15%. These correlations are easy to use in BES.

Nomenclature

| Symbol | Description |
|--------|-------------|
| λ      | thermal conductivity of air (W/mK) |
| A      | width (m) |
| a      | correlation coefficient |
| b      | correlation coefficient |
| Cp     | specific heat capacity (J·kg⁻¹·K⁻¹) |
| g      | gravitational acceleration (m·s⁻²) |
| G      | production of turbulent kinetic energy (W·m⁻²·K⁻¹) |
| Gr     | Grashof number |
| H      | height or vertical opening position (m) |
| h      | heat-transfer coefficient (W·m⁻²·K⁻¹) |
| k      | turbulent kinetic energy (m²·s⁻²) |
| L      | length (m) |
| Lc     | characteristic length (m) |
| n      | exponent correlation |
| Nu     | average Nusselt number |
| Pr     | Prandtl number |
| q      | density heat flux (W·m⁻²) |
| Q      | heat flux (W·m⁻²) |
| R      | regression coefficient |
| Ra     | Rayleigh number |
| Re     | Reynolds number |
| S      | source terms (W·m⁻²·K⁻¹) |
| T      | temperature (°C) |
| u      | averaged velocity component (m·s⁻¹) |
| V      | volume (m³) |
| W      | height of the opening (m) |
| X      | width of the opening (m) |
| y⁺     | y plus |

Greek symbols

| Symbol | Description |
|--------|-------------|
| α      | inclination angle (°) |
| β      | coefficient of thermal expansion (K⁻¹) |
| δ      | distance of boundary node from wall (mm) |
| ε      | dissipation rate of turbulent kinetic energy (kg·m⁻²·s⁻³) |
| μ      | dynamic viscosity (kg·m⁻²·s⁻¹) |
| ν      | kinematic viscosity (m²·s⁻¹) |
| ρ      | density (kg·m⁻³) |
| σ      | constant standard (k–ε) |
| Ω      | direction vector |

Superscripts

| Superscript | Description |
|-------------|-------------|
| 2D          | 2D geometry |
| 3D          | 3D geometry |
| h           | horizontal orientation |
| v           | vertical orientation |

Subscripts

| Subscript | Description |
|-----------|-------------|
| ∞          | unperturbed fluid |
| c          | cold wall |
CFD use of a CFD tool
FC forced convection
h horizontal orientation
i x-coordinate or wall number
In Inlet
j y-coordinate
k z-coordinate or turbulent kinetic energy
MC mixed convection
n normal direction
NC natural convection
out outlet
s surface
T turbulent flow
t turbulence
v vertical orientation
ε turbulence parameter $k-\epsilon$ model

1. Introduction

The current regulations requiring a reduction in energy consumption in buildings mean that a more accurate assessment of their thermal loads is necessary for preventing the overestimation or underestimation of their annual energy demand. Inaccurate calculations can lead to oversizing of the conditioning and insulation equipment in the envelope, thus causing the cost and energy consumption of buildings to increase (Obyn & Van Moeseke, 2015). At present, most heat-transfer mechanisms in the building envelope are determined accurately; however, there is potential for improvement regarding the precise calculation of the convection mechanism (Peeters, Beausoleil-Morrison, & Novoselac, 2011). Building energy performance simulation (BES) program developers cannot find precise and simple correlations in the scientific literature to implement in BES code – consequently, the correlations used lead to errors in the calculation of energy demands.

The heat exchanged between a building and its environment depends on three heat-transfer mechanisms: conduction, convection and radiation. Of these three mechanisms, convection presents the greatest degree of uncertainty (Peeters et al., 2011). In the convection mechanism, the heat exchange between a wall and the air is determined by the convective heat-transfer coefficient (CHTC). Within this mechanism, natural convection, forced convection, and mixed convection are distinguished when the air movement is caused by buoyancy forces, a pressure difference, or both, respectively (Beausoleil-Morrison, 2002). Natural convection in buildings has been extensively studied. The most recent studies involve correlations that are oriented towards building enclosures to be implemented in BES (Rincón-Casado, Sánchez de la Flor, Chacón Vera, & Sánchez Ramos, 2017). However, internal forced convection is not well resolved in BES programs since they use empirical or numerical correlations that do not take into account the influence of 3D geometry and the positions of the inlet–outlet openings. The CHTC of the internal walls of buildings – the rate of heat-transfer exchange between the interior air and the walls – is a determining parameter in annual cooling and heating demands. A recent study by Obyn and Van Moeseke (2015) compares the annual demands calculated with a CHTC obtained by correlations with annual demands calculated with a constant CHTC, finding a deviation of up to 9.1%. This highlights the importance of correctly calculating the internal CHTC, especially when estimating the energy demand of buildings.

Currently, there are three ways to estimate CHTCs: empirical methods, numerical methods and correlations. The empirical methods are obtained from experimental tests whose boundary conditions are limited, and therefore their applicability is limited. On the other hand, numerical methods such as computational fluid dynamics (CFD) are reliable and accurate but are complicated to use and have a high computational cost. The linking of CFD programs with BES programs to calculate the annual demand of a whole building is currently unattainable due to decreases in the time steps, especially when the conditioning systems are coupled (Zhai & Chen, 2003, 2005; Zhai, Chen, Haves, & Klem, 2002). The method involving correlations, usually developed from numerical or empirical results, is easy to use and to implement in building thermal simulation programs. Therefore, the use of correlations could be the most appropriate method for integrating the CHTC into BES programs. However, the correlations found in the literature, especially in forced convection, have a limited application because they do not contemplate either the dependence of all the geometric variables of 3D enclosures or the geometry of the inlet–outlet openings.

Thermal simulation programs for buildings involve a variety of calculation methods, and Crawley, Hand, Kummer, and Griffith (2008) recently performed a review of the 20 most important programs of the last 50 years. CHTCs were among the parameters analyzed, and were found to differ in terms of temperature variation, airflow variation, CFD use and CHTC user definitions. Most of the programs analyzed use empirical correlations or constant values for the CHTC. Only three programs use CHTC values derived from CFD: Energy Plus (2017), ESP-r (2015), and IES < VE > (2017). Energy Plus uses several algorithms for each interior surface of the building, the most important being the adaptive convection algorithm based on the work of Beausoleil-Morrison (2001), although it has been modified to adapt to new
designs and the appearance of new correlations. The algorithm has 45 different categories and 29 different options depending on the surface, floor, wall or window studied and its location. In general, the correlations used by this algorithm come from experimental studies performed by different authors; however, the correlations used do not take into account all the geometric variables of the enclosure or the geometry of the inlet-outlet openings. Another important program is ESP-r, developed by Clarke (1991, 2001), which implements empirical correlations and a CFD module. This program allows for a bidirectional mode when the CHTC is transferred to the thermal balance module and an iterative mode when there is iteration between the CFD and thermal balance modules. However, the computational cost of these calculation modes is high for small buildings. On the other hand, it is possible to use the correlation developed by Fisher and Pederson (1997), which depends on the air changes per hour (ACH) – but these correlations only take into account the ACH, and not the 3D geometry or the position of the inlet–outlet openings of the room.

The main objective of this study is to develop correlations for the calculation of the interior CHTC in forced convection for the 3D geometry of rooms in buildings. The dimensions of each room and the positions of the inlet and outlet openings are incorporated into the developed correlation. The purpose of these correlations is to facilitate the calculation of CHTCs in building simulation programs. The correlations of 3D geometric parameters obtained from CFD are not accurate enough and are not valid for use in BES. For this reason, a new methodology is used based on 2D planes that allows for a more accurate prediction of the direction of the air flow and the interactions with the walls of the enclosure. This methodology consists of solving the 3D enclosures from two 2D planes – one with a horizontal orientation and one with a vertical orientation. A parametric study of 3D and 2D geometries is performed, and the resulting configurations that are formed as a result are subsequently solved using numerical CFD simulation tools. The results of this are used to construct two correlations, the first of which calculates the CHTCs of the 2D planes and the second of which calculates the CHTCs of the 3D enclosure from the results of the first correlation.

2. Materials and methods

Buildings are generally made up of clustered, connected 3D enclosures that have air inlet and outlet openings. This study was conducted on a 3D enclosure, but all the enclosures of the building can be grouped together when the correlations are implemented in the BES. The developed methodology is applicable to 3D enclosures with parallelepiped geometry whose inlet-outlet openings are located in opposing walls. The mechanism of forced convection was studied. The purpose of this methodology is to obtain the CHTC of a 3D enclosure from the CHTC of two 2D planes, one with a vertical orientation and the other with a horizontal orientation. In this way, the CHTC of each wall of the 3D enclosure is calculated in two steps: the first step develops a correlation to calculate the CHTCs of the 2D planes, and the second step develops a correlation that relates the CHTCs of 3D geometry to the CHTCs of the 2D planes. The sections below describe both the methodology to develop the correlation that relates the 3D geometry to the 2D planes and the methodology that was followed to obtain the correlation of the 2D planes.

2.1. Methodology for obtaining the correlation between the 3D and 2D geometries

In a 3D enclosure with inlet and outlet openings on its walls, the air enters through the inlet opening at a certain velocity and temperature and then leaves in other conditions through the outlet opening. The air exchanges heat with the walls as it follows the air path, the rate of heat exchange being defined by the CHTC. The methodology developed relates the CHTC of 3D geometries to the CHTC of two 2D planes (one with a horizontal orientation and one with a vertical orientation).

Figure 1(a) shows a 3D enclosure with six walls, with a height \( H \), a length \( L \), and a width \( A \). In addition there is one inlet opening and one outlet opening, which are set in opposing walls. Figure 1(b) shows the vertical 2D plane and Figure 1(c) shows the horizontal 2D plane; in both cases, the orientation angle \( (\alpha_h, \alpha_v) \) depends on the misalignment of the inlet-outlet openings. The variables that define the vertical 2D plane (Figure 2(a)) are the height of the enclosure \( (H) \), the length influenced by the inclination angle \( (L/\cos \alpha_v) \), and the position and size of the inlet opening for the vertical orientation \( (W_{in, out}, H_{in, out}) \). The variables that define the horizontal 2D plane (Figure 2(b)) are the width of the enclosure \( (A) \), the length influenced by the inclination angle \( (L/\cos \alpha_h) \), and the position and size of the inlet opening for the horizontal orientation \( (X_{in, out}, \alpha_{in, out}) \).

The input variables for the 3D and 2D computer models are: the velocity and temperature of the inlet air; the wall temperature; the enclosure geometry; and the geometry of the inlet and outlet openings. Table 1 shows the numerical values used for the study of the 3D model and the 2D models. This methodology was used to find the geometric relationships between the 3D model and the equivalent 2D model, and in the final correlation.
these dimensions vary and are defined by the user. As demonstrated below, 25 3D configurations were obtained from these values, and these were studied for three different input Reynolds numbers. Following the methodology of section 2.3, the mesh was generated, the turbulence model was selected, and the convergence criteria were set to perform the CFD simulations. In the various configurations of the horizontal and vertical 2D planes, the positions of the inlet and outlet openings were combined and the symmetrical configurations were eliminated, resulting in 5 independent configurations for each 2D plane. These configurations were then studied for three velocities. As with the 3D configurations, following the methodology of section 2.3, the CFD models were constructed and then simulated until convergence was reached.

The proposed methodology relates the CHTCs of the 3D geometry to the CHTCs of the 2D planes through the inclination angles, so the dependence on the dimensions is incorporated into the CHTCs of the 2D planes. To obtain results for 2D planes with different inclination angles, the positions of the inlet and outlet openings were varied. The openings in the inlet–outlet walls can be in 3 positions in the horizontal direction and 3 in the vertical direction – thus, each opening can be in 9 different positions, resulting in 81 configurations. However, because the phenomenon under study is forced convection, the number of configurations is considerably

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### Table 1. Boundary conditions of the vertical 2D, horizontal 2D, and 3D simulations.

| Variable | 3D | Vertical 2D | Horizontal 2D |
|----------|----|-------------|---------------|
| \( V_{in} \) (m/s) | 0.5–1.0–1.5 | 0.5–1.0–1.5 | 0.5–1.0–1.5 |
| \( T_{in} \) (°C) | 20 | 20 | 20 |
| \( T_{wall} \) (°C) | 30 | 30 | 30 |
| \( H \) (m) | 3 | 3 | 3 |
| \( L \) (m) | 5 | 5/cos \( \alpha_v \) | 5/cos \( \alpha_h \) |
| \( A \) (m) | 3 | 3 | 3 |
| \( W_{in} \) (m), \( W_{out} \) (m) | 0.42 | 0.42 | – |
| \( X_{in} \) (m), \( X_{out} \) (m) | 0.70 | – | 0.70 |
| \( H_{in} \) (m), \( H_{out} \) (m) | 2.59–1.40–0.21 | 2.59–1.40–0.21 | – |
| \( A_{in} \) (m), \( A_{out} \) (m) | 2.59–1.40–0.21 | – | 2.59–1.40–0.21 |
Figure 3. Symmetric and asymmetric configurations for the 3D model.

reduced due to some configurations being symmetrical. The top part of Figure 3 shows the 9 possible positions of the inlet opening; while the independent positions are 1.1, 1.2, 2.1, and 2.2, the rest can be obtained by symmetry. Likewise, the 9 possible positions of the outlet opening include symmetrical positions, although these depend on the position of the inlet opening. Combining the non-symmetrical input and output positions, 25 independent configurations were obtained (shaded in Figure 3). These 25 configurations of the 3D geometry and their 2D planes were solved using CFD tools. With the results obtained, a correlation was sought to relate the CHTC of the 3D geometry to the CHTCs of the 2D planes, including the dependence on the positions of the openings through the inclination angles of the 2D planes (as shown in section 3.1 below).

2.2. Methodology for obtaining the correlation between the 2D geometries

The previous section describes the methodology that was followed to develop a correlation between the CHTC of a 3D geometry and the CHTCs of two 2D planes. The next step is to correlate the CHTCs of a 2D plane with all the geometric variables involved. Figure 2(a,b) shows the variables involved in the vertical and horizontal 2D planes. It is possible to verify that these are the same variables in both cases, meaning that a correlation obtained for the vertical plane can also be used for the horizontal plane.

In order to develop the correlation between the CHTCs of the 2D planes, it is necessary to define the variables on which the correlation depends. As Figure 4 shows, the studied geometric variables are the enclosure height \( H \), the enclosure length \( L \), the input aperture width \( W_{in} \), the output aperture width \( W_{out} \), the input opening height \( H_{in} \), the outlet opening height \( H_{out} \), and the air inlet velocity \( V_{in} \). The variable \( H \) is taken as a variable for the dimensionless value and must represent the typical dimensions of a building (3 m). Consequently, the remaining number of independent variables is 6. For the construction of the correlation, it is necessary to obtain 3 values for each variable within the range of applicability. Combining all the values for the 6 variables studied, a total of 729 (3^6) unique 2D configurations are obtained. However, this number is reduced when the symmetrical configurations are eliminated because there is forced convection. The heights of the inlet and outlet openings (\( H_{in}, H_{out} \)) are affected by symmetry. Figure 5 shows that Configurations 9 and 1 are symmetrical, exchanging the results of Wall 2 for Wall 4. The same happens with Configurations 5, 7 and 8, which are symmetrical to Configurations 4, 3 and 2, respectively. On the other hand, due to the convergence problems of the CFD tool, the outlet opening size (\( W_{out} \)) is limited to being greater than or equal to the inlet opening size (\( W_{in} \)). Applying the previous restrictions, the total number of configurations is reduced to 270. Table 3 shows the dimensionless values adopted for each input variable of the computational model, wherein each variable is assigned three values: the minimum, the average and the maximum. However, the CFD program uses dimensions that are typical for
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Figure 5. 2D configurations for different inlet and outlet opening positions.

buildings \((H = 3 \text{ m})\), and the air inlet velocities studied are 0.5, 1.0, and 1.5 m/s. Once the geometry values of each variable are defined, the geometry and meshing for each configuration can be constructed according to the methodology.

2.3. The computational model

The computational model presented here considers a steady flow, 2D and 3D geometry, and Newtonian fluid. All of the fluid’s properties remain constant and behave as an ideal gas. All of the properties are evaluated at the fluid’s average temperature within the enclosure. The phenomenon under study is forced convection; therefore, the velocities are large enough for all buoyancy effects to be negligible and for gravity to be ignored. For the radiation exchange between walls, all of their temperatures are imposed as a boundary condition. The CFD results were obtained by solving the Navier–Stokes equations and the energy equation via the finite-volume method using the commercial software package Ansys Fluent (2017). The semi-implicit method for pressure-linked equations (SIMPLE) numerical algorithm developed by Patankar and Spalding (1972) was also used. The equations for mass, momentum and energy were solved iteratively for the corresponding boundary conditions using numerical methods until convergence was reached for the variables of interest (velocity, temperature, pressure and wall heat flow).

One of the most important aspects to consider when using CFD tools is the construction of the mesh for the computational environment. DESIGN MODEL was used for the construction of the 2D and 3D geometries, and ICEM was used for the construction of the meshes, both of which are part of the Ansys program package (Ansys Fluent, 2017). The mesh used in the simulation is built from rectangular elements and increases in size from the walls toward the center of the enclosure in a uniform fashion (Figure 6). This type of mesh produces a better convergence because there is a higher density of elements within the thermal boundary layer where the CHTC is calculated. To ensure the correct solution within the boundary layer, nodes must be placed within the viscous sub-layer. Thus, the first element must be located at a maximum distance of 1 mm from the wall, and the growth rate toward the center of the enclosure must be between 10% and 15%. The parameter that controls the correct solution of the viscous sub-layer is \(y^+\). This dimensionless parameter depends on the turbulence model. Thus, for standard \(k–\epsilon\) turbulence models with ‘enhanced wall treatment’, \(y^+\) must have a value of approximately 1 – and to control this, the mesh is refined until this value is reached. This function is implemented in Fluent code and is used to specify both \(\epsilon\) and the turbulent viscosity in the near-wall cells. In this approach, the whole domain is subdivided into a viscosity-affected region and a fully-turbulent region (Ansys Fluent, 2017). Figure 6 shows an example of an enclosure represented by a mesh with an aspect ratio of 1. According to sensitivity
studies developed by Al-Sanea Sami, Zedan, and Al-Harbi (2012), CHTCs have low sensitivity to the inflow turbulence; therefore, the turbulence intensity ($I_{in}$) is set at 5% and the scale length (or size swirl) is set at 10 cm ($L_{in}$). This value must never exceed the size of the inlet opening.

$$y^+ = y \frac{u_+}{v}$$  \hspace{1cm} (1)

Once the geometric, meshing and boundary conditions have been defined, all the 3D and 2D configurations are simulated with a CFD tool. Two convergence criteria are adopted: first, the residues of the equations must be below $10^{-8}$, and second, the heat flux on the walls is monitored. Convergence is reached when a constant value remains for at least 1000 iterations. The number of iterations required for convergence is usually about 15,000 for 3D models and 5000 for 2D models, resulting in a simulation time of around 12 h for 3D models and 15 min for 2D models. The simulations were conducted on a PC with an Intel Core i7 3770 @ 3.4–3.9 GHz with 8 cores and 16 GB of RAM, which used parallel computing (MPI) and decomposed the domain into two parts. The total calculation time for all configurations was 7 days.

To obtain a mesh configuration that offers a good trade-off between accuracy and computing costs it is necessary to establish a mesh refinement process. The grid convergence index method chosen was developed by Celik et al. (2008). This process involves selecting three grids with different degrees of coarseness: a coarse grid, a medium grid and a fine grid. The mesh is refined near to the surface of the wall, where the velocity and temperature gradient vary rapidly. The Nusselt numbers resulting from the CFD simulations are compared with one another to justify the best compromise between accuracy and computational cost. The convergence criterion imposed on all of the residual equations was above $10^{8}$.

In order to obtain an accurate and meaningful numerical solution, meshing the computational domain is a crucial first step that is particularly important with fast-moving flows due to the steep gradients which occur within the boundary layers. For this reason, the $y^+$ criteria must be satisfied in order to accurately resolve these steep gradients in the buoyancy-driven boundary layers.

Five grids were studied for the 3D geometry under identical operating conditions, and it can be seen from Figure 7 that the $300 \times 300 \times 300$ grid offers the best balance between precision and computational cost.

### 2.4. Governing equations

To calculate the CHTCs, the velocity, temperature and pressure fields of the enclosure must be determined. This was accomplished by using the Navier–Stokes equations, which describe the fluid motion for a given set of boundary conditions. These equations, along with the turbulence model and the energy equation, are solved at each node of the mesh.

The turbulence model used here is the standard $k–\varepsilon$ model, which obtains good results compared to experimental results (Fomichev, 2010). The equations for the 3D model are provided in tensor notation, where $x_i$ represents the variables $X, Y$, and $Z$, and $u_i$ represents the corresponding averaged velocities components.

The continuity equation is as follows:

$$\frac{\partial p}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0$$  \hspace{1cm} (2)

The equation for the conservation of momentum is as follows:

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \right]$$  \hspace{1cm} (3)
The transport equation for \( \varepsilon \) from the turbulence model is as follows:

\[
\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_T}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right) - \frac{\rho C_2 \varepsilon^2}{\lambda + \sqrt{\nu \varepsilon}} + C_{3\varepsilon} \frac{\varepsilon}{\lambda} G_{\varepsilon} + S_{\varepsilon}
\]

where the source terms \( S_{\varepsilon} \) can be defined for each configuration and are optional, and the coefficient \( C_{3\varepsilon} \) is given by

\[
C_{3\varepsilon} = \tanh \left( \frac{\nu}{U^*} \right)
\]

The constants used in the standard \( k-\varepsilon \) model are as follows:

\[
\sigma_k = 1.00, \quad \sigma_\varepsilon = 1.30, \quad C_{1\varepsilon} = 1.44, \quad C_{2\varepsilon} = 1.92, \quad C_2 = 0.43, \quad \text{and} \quad C_{3\varepsilon} = 1.90
\]

The CHTC between the surface of the wall and the fluid in motion at different temperatures is provided by Newton’s law of cooling and depends on the total heat of the wall, the transfer area and the temperature difference between the surface and unperturbed fluid:

\[
\bar{h}_i = \frac{Q_i}{A_i(T_i - T_\infty)}
\]

The total heat flow that is transferred between the fluid and the wall is calculated from the temperature gradient produced inside the thermal boundary layer of the fluid through an integration of Fourier’s law at each wall.
Because the heat flow depends on the temperature gradient, it is solved using CFD. Thus, the heat flow at a wall is obtained by integrating the temperature gradient along the wall and is affected by the conductivity of the fluid as follows:

\[
Q_{\text{CFD}}|_{yi} = -\int_{0}^{x} \int_{0}^{y} \lambda \frac{\partial T}{\partial y} \, dx \, dy \bigg|_{y=0} \quad (12)
\]

\[
Q_{\text{CFD}}|_{xi} = -\int_{0}^{x} \int_{0}^{y} \lambda \frac{\partial T}{\partial x} \, dx \, dy \bigg|_{x=0} \quad (13)
\]

\[
Q_{\text{CFD}}|_{zi} = -\int_{0}^{x} \int_{0}^{y} \lambda \frac{\partial T}{\partial z} \, dx \, dy \bigg|_{z=0} \quad (14)
\]

Therefore, at the fluid–solid interface, the heat transfer by conduction is equal to the heat transfer by convection, and the average CHTCs are as follows:

\[
\bar{h}_i = \frac{Q_{\text{CFD}}|_{yi}}{A_i(T_{hi} - T_{\infty})} \quad (15)
\]

\[
T_{\infty} = \frac{\int V T \, dV}{V} \quad (16)
\]

\[
\bar{N}u_i = \frac{h_i L_c}{\lambda} \quad (17)
\]

\[
L_c^{3D} = \frac{4 W_{\text{in}} A_{\text{in}}}{(2 W_{\text{in}} + 2 A_{\text{in}})} \quad (18)
\]

Eq. (15) shows that the temperature difference between the wall \((T_{hi})\) and the free fluid \((T_{\infty})\) is proportional to the temperature gradient of the thermal boundary layer. Thus, increases in the temperature transferred between the wall and the fluid correspond to increases in the gradient within the boundary layer, which maintains a constant ratio, indicating that the CHTC is independent of the temperature. This finding is valid for flat plates where the fluid is not perturbed by other walls and for enclosures with walls whose temperatures are all equal, but it is not valid for cases in which the walls have different temperatures.

3. Results and discussion

3.1. Correlations between the 3D and 2D models

In this section, the methodology presented in section 2.1 is implemented in order to obtain a correlation for the CHTCs of the 2D planes. To find the correlations, it was necessary to calculate the CHTCs of the 3D configurations proposed in Figure 3, where each configuration is associated with two 2D planes – one with a vertical orientation (Li, Wenping, Xu, & Mao, 2017) and the other with a horizontal orientation. The computational model is described in section 2.3 and the governing equations are described in section 2.4. Using the input variables of the computational model, all configurations were simulated with CFD in order to obtain the results in terms of the CHTCs. Subsequently, mathematical optimization was used to obtain the correlations that relate the CHTCs of the 3D models to the CHTCs of the 2D planes.

The CFD program calculated the heat fluxes in each wall using Eqs. (12) to (14), and the average Nusselt numbers were obtained using Eq. (17). For the 3D configurations, the characteristic length used was the hydraulic diameter of the inlet opening (Eq. (18)), while \(W_{\text{in}}\) was used for the 2D vertical configurations and \(A_{\text{in}}\) was used for the 2D horizontal configurations. Using the obtained Nusselt numbers, a correlation with the best fit was found by means of mathematical optimization. The resulting correlation, which relates the Nusselt numbers of the walls of the 3D enclosure as a function of the Nusselt numbers of the 2D planes, is as follows:

\[
N_{\text{Nu}}^{3D} = a_i N_{\text{Nu}}^{2Dh} + b_i N_{\text{Nu}}^{2Dv} \quad (19)
\]

Table 2 shows the coefficients of the correlation as a function of each wall. For example, Walls 1 and 3 of the 3D enclosure \((N_{\text{Nu}}^{3D})\) depend on Walls 1 and 3 of the horizontal 2D plane \((N_{\text{Nu}}^{2Dh})\), the 2D vertical plane \((N_{\text{Nu}}^{2Dv})\), the horizontal inclination angle \((\alpha_h)\), and the vertical inclination angle \((\alpha_v)\). Accordingly, Table 2, Walls 2 and 4 (the floor and the ceiling, respectively) only depend on the Nusselt numbers of Walls 2 and 4 of the vertical 2D plane, and the inclination angle \((\alpha_v)\). Contrastingly, Walls 5 and 6 of the 3D enclosure are a function of Walls 4 and 5 of the horizontal 2D plane and its inclination angle \((\alpha_h)\).

Figure 8 compares the Nusselt numbers obtained by correlation and CFD simulation. Figure 8(a) shows the results for Walls 1 and 3, the former with an inlet opening and the latter with an outlet opening. Figure 8(b) compares the Nusselt numbers of Walls 2 and 4, which only depend on the vertical 2D plane. Finally, in Figure 8(c), the Nusselt numbers of Walls 5 and 6 obtained by simulation are compared with those obtained by correlation.

| Wall | \(a_i\) | \(b_i\) | \(n_i\) | \(R^2\) |
|------|--------|--------|--------|-------|
| 1    | .4352  | .4352  | 1      | .985  |
| 2    |        | .9124  | 1      | .995  |
| 3    | .5123  | .5123  | 3      | .987  |
| 4    |        | .8112  | 1      | .986  |
| 5    | .4491  |        | 1      | .979  |
| 6    | .5512  |        | 1      | .989  |
3.2. Correlations for the 2D models

In this section, the methodology presented in section 2.2 is followed to obtain a correlation to calculate CHTCs for 2D planes. To find the correlations, it is necessary to obtain CFD results based on the variables that must appear in these correlations. Figure 2 shows the variables of the 2D planes under examination, and it can be seen that $W_{in}, W_{out}, H_{in}, H_{out},$ and $H$ in Figure 2(a) are similar to $X_{in}, X_{out}, A_{in}, A_{out},$ and $A$ in Figure 2(b). Therefore, the correlation obtained for the vertical 2D plane is also valid for the horizontal 2D plane. In order to obtain a wide range of configurations, 3 values were assigned to each variable, and by combining the values for the 6 variables and then eliminating the symmetrical configurations, 270 unique configurations were obtained.

Table 3 shows the dimensionless values adopted for each input variable of the computational model. Each variable is assigned three values: the minimum, the average, and the maximum. However, in the CFD program typical building dimensions were used ($H = 3 \text{ m}$) and the air inlet velocities chosen were 0.5, 1.0, and 1.5 m/s.

Once the geometric values of each variable had been defined, the geometry and meshing for each configuration were constructed according to the methodology in section 2.3. Subsequently, the convergence criterion was defined as $10^{-8}$ in the residuals of all the equations, and the heat flows were monitored until they maintained a constant value for at least 1000 iterations. The average number of iterations needed to reach convergence was 15,000, yielding a mean simulation time of 120 min for each configuration. The ability to calculate 8 simulations in parallel reduced the simulation time from 67 days to 8.7 days.

All simulations were completed satisfactorily. The mean Nusselt value for each wall was calculated from the CFD heat fluxes using Eqs. (12) and (13). The characteristic length used in the 2D configurations was the size of the input opening $W_{in}$ ($A_{in}$ in the horizontal 2D plane). From all the Nusselt values for each wall and the values of the dimensionless variables, a search was conducted for the correlation that best fits the simulation results. The desired correlation should have a similar shape to the flat plate correlation in forced convection (see Eq. (20) below), where the average Nusselt number depends on the dimensions of the enclosure ($H, L$), the dimensions of the inlet and outlet openings ($W_{in}, W_{out}$), and their position ($H_{in}, H_{out}$; see Figure 4):

$$Nu = CRe^nPr$$

**Table 3. Application range of the variables in the 2D models (dimensionless).**

| Variable | Minimum | Average | Maximum |
|----------|---------|---------|---------|
| $W_{in}/H$ | 0.017 | 0.067 | 0.400 |
| $W_{out}/H$ | 0.017 | 0.067 | 0.400 |
| $H_{in}/H$ | 0.025–0.083–0.217 | 0.500 | 0.975–0.917–0.783 |
| $H_{out}/H$ | 0.025–0.083–0.217 | 0.500 | 0.975–0.917–0.783 |
| $L/H$ | 0.5 | 1.0 | 2.0 |
| $Re_{in} (W_{in}, V_{in})$ | 1657–3314–4971 | 6627–13,255–19,882 | 39,764–79,529–119,293 |
Table 4. Correlation coefficients for the 2D models.

| Coefficient | Wall 1   | Wall 2   | Wall 3   | Wall 4   |
|-------------|----------|----------|----------|----------|
| a           | 0.019512 | 0.041169 | 0.066311 | 0.009220 |
| b           | 0.006094 | -0.06679 | -0.027515| -0.04526 |
| c           | -0.005944| -0.03410 | 0.005247 | -0.02107 |
| d           | 0.030444 | -0.024028| 0.005527 | 0.06855  |
| e           | -0.00913 | -0.02876 | -0.03623 | -0.01066 |
| f           | 0.16337  | 0.041022 | 0.020702 | 0.03956  |
| n           | 0.770041 | 0.782311 | 0.771561 | 0.868345 |
| $R^2$       | 0.987    | 0.978    | 0.988    | 0.986    |

The coefficients of the correlations $a$, $b$, $c$, $d$, $e$, $f$, and $n$ were obtained for the four walls of the two 2D enclosures, using mathematical optimization to determine the values with the best correlation coefficient ($R^2$) and the least mean square error. Table 4 shows the correlation coefficients for each wall, and Figure 9 shows the Nusselt numbers obtained by simulation versus correlation for the four walls of the two 2D enclosures.

3.3. Test cases and comparison with other authors

To validate the methodology and test the precision of the developed correlations, the results of CFD test cases were compared with those obtained by correlation. The test case consists of an enclosure that is 3 m high, 3 m wide and 5 m long, with inlet and outlet openings defined as having the same dimensions (0.1 m), simulated in several different positions. Figure 10 shows an example with 3D geometry and 2D planes, and Table 5 presents the geometric variables used in the 3D and 2D models, along with the CHTCs obtained from both simulation and correlation. The methodology proposed in this work was applied to obtain the 2D planes and the inclination angles. Subsequently, the Nusselt numbers of the horizontal and vertical 2D planes were calculated using Eq. (21), while Eq. (19) was used to calculate the Nusselt numbers of the 3D geometry using the Nusselt numbers of the 2D planes. Finally, the CHTCs were calculated from the hydraulic diameter of the inlet openings of the 3D geometry using Eq. (17). The results show errors of less than 15% in 95% of the test cases; it is thus reasonable to draw the conclusion that the computed errors are overall less than 20%, consistent with the CFD results reported in other works (Mou, He, Zhao, & Chau, 2017; Obasaju, 1992; Zhao & He, 2017).

In order to validate the numerical results, the CHTCs were recalculated by applying the correlation developed by Fisher and Pederson (1997) for floors, ceilings and walls, and the correlation developed by Goldstein and Novoselac (2010) for floors and walls. Figure 11
compares the results of all three correlation methodologies, although the correlation developed according to the position of the input–output openings is not featured, as this parameter is not factored into the correlations developed by these other authors. The correlations of Fisher and Pederson and the correlations of Goldstein and Novoselac are represented continuously according to the ACH, and the correlations developed in the present study are represented by the CHTCs for 4, 6, 8, 10 and 12 ACH. The average values are shown for the range of variation according to the position of the inlet and outlet openings. The results show that neither Fisher and Pederson nor Goldstein and Novoselac distinguish between vertical walls (Walls 1, 3, 5 and 6), although they do differentiate between floors (Wall 2) and ceilings (Wall 4). All the CHTCs obtained in the present study are similar to those obtained by these other authors, with a greater difference in the ceiling for the Fisher and Pederson correlation, which is due to the fact that the tests were carried out for an enclosure that has an inlet opening near to the ceiling. The rest of the CHTCs are within the margins of the correlations proposed by these authors. The advantages of the methodology of the present study over these two alternatives lie in the distinction between the walls and the positions and dimensions of the openings, as well as in the dimensions of the room.

### 3.4. Applicability to BES tools

The correlations developed in this study have a range of applications, as well as some limitations. They can be applied to typical buildings whose rooms have parallelepiped geometry and feature inlet and outlet openings in opposite walls. Table 6 shows the minimum and maximum values of the application range of the dimensionless variables.

The focus of this study is on the mechanism of forced rather than natural convection. The Richardson number is the parameter which denotes the predominant convection mechanism ($Gr/Re^2$); a Richardson number of less than 1 indicates predominantly forced convection, whereas a number much greater than 1 indicates predominantly natural convection. For the calculation of CHTCs in natural convection, the correlations developed by Beausoleil-Morrison (2002) and Rincón-Casado et al. (2017) can be used. In cases of mixed convection, where both mechanisms are taken into account, Eq. (22) below – developed by Churchill (1977) – can be used, where $n$...
Figure 11. Comparison of the CHTCs according to ACH with the correlations developed by other authors.

Table 6. Application range of the variables of the 3D models.

| Variable | Minimum | Maximum |
|----------|---------|---------|
| H (m)    | 3.0     | 3.0     |
| L (m)    | 0.5     | 6.0     |
| A (m)    | 0.5     | 6.0     |
| \(W_{in}\) (m), \(W_{out}\) (m) | 0.05 | 2.10 |
| \(X_{in}\) (m), \(X_{out}\) (m) | 0.05 | 2.10 |
| \(H_{in}\) (m), \(H_{out}\) (m) | 0.025 | 3.000 |
| \(A_{in}\) (m), \(A_{out}\) (m) | 0.025 | 6.000 |
| \(Re_{in}\) | 1657 | 119,293 |

4. Conclusions

This study aimed to develop correlations from CFD that can be used to calculate the CHTCs for 3D geometric enclosures which are subject to forced convection and which have inlet and outlet openings in opposing walls. An additional aim was for the correlations to be easy to implement in BES and reduce the computational costs required by CFD techniques. The methodology is novel and the results are satisfactory.

Two new correlations have been developed in this study. The first can be used to relate 3D geometry to two 2D planes, and is based on the CHTCs of the 2D planes and the angles of the horizontal and vertical inclination of the planes contained within the 3D enclosure. The comparison of the results for the first correlation and the CFD simulations reveals an error rate of less than 11%. The second correlation calculates the CHTCs of 2D planes using the inlet and outlet openings in opposing walls, and is based on both the dimensions of the 2D planes and the dimensions and positions of the inlet and outlet openings. A parametric study was carried out in order to develop a correlation which incorporates all of the relevant geometric variables. The comparison of the results of the second correlation and the CFD simulations reveals an error rate of less than 13%.

The comparison of the combined results of both correlations and the results of the CFD simulations for the test cases reveals an error rate of less than 15% for 95% of the cases. Therefore, the objective has been satisfactorily achieved.

The correlations developed in this study are easy to use and to apply in BES programs, although they are...
not as accurate as CFD simulations. Therefore, future studies are required in order to improve the mathematical methods used to obtain the correlation coefficients. Alternatively, functions such as proper orthogonal decomposition order reduction could be developed that can be incorporated into the BES programs and that work in interpolation mode in order to bring the accuracy of the correlations closer to that of the CFD simulations.

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No potential conflict of interest was reported by the authors.

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