Research Paper

3D and transient numerical modelling of door opening and closing processes and its influence on thermal performance of cold rooms

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HIGHLIGHTS

• A 3D CFD modelling of opening/closing cycle of cold room's doors is developed.
• The CFD simulation uses the tracer gas concentration decay experimental technique.
• Air infiltration rate through sliding door is 20% lower than through a hinged one.
• Air temperature inside cold room with sliding door is 17% lower than with hinged one.
• The developed model extends the analytical results for door opening/closing periods.

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ABSTRACT

This paper presents the comparison of three-dimensional and transient CFD modelling of the opening and closing processes of hinged and sliding doors and its influence on the thermal performance of cold rooms. A species transport model is used to model a tracer gas. The air infiltration through the door opening is determined by the tracer gas concentration decay technique. The prediction of air temperature and velocity fields in the cold room as function of external air temperature allows quantifying the increase of the air infiltration rate and consequently of the average air temperature inside the cold room. When the hinged door is used, the formation of vortices during the opening movement promotes a larger and faster thermal interaction between the two contiguous air masses. The air infiltration during the sliding door opening/closing is 20% lower than for a hinged door. Consequently, the average air temperature inside the cold room is 17% lower. The air infiltration rate was numerically predicted and compared with analytical models' results. The numerical model predicts closely the air infiltration rate for each door type. Moreover, the transient CFD modelling extends the results of the analytical models allowing the analysis of the influence of door opening and closing processes on the air temperature and velocity fields.

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1. Introduction

The door opening and closing cycles for storing or removing food products from cold rooms imply the infiltration of hot and humid air. This condition may have several consequences on production efficiency, product quality, hygiene, food safety and maintenance [1]: (1) increase of the thermal load (it can reach up to 50% of the total heat load), which reduces the refrigeration system's ability to maintain the desired temperature; (2) higher coil frost formation rates that reduce the refrigeration capacity due to the insulating effect [2–4]; (3) temperature fluctuations that may impair food quality, food safety and induce weight loss [2,5]. Therefore, its analysis and quantification are very important once it can lead to the use of best practice measures and complementary solutions to reduce the frequency with which the refrigeration system is triggered, and consequently to a reduction of energy consumption while ensuring products quality [4,6].

Evans et al. [7] conducted a study to characterize the energy consumption of cold stores. Energy audits were performed in cold stores of several European countries to identify possible energy savings measures and practices. Infiltration/protection of doors represented about 8.3% of problems identified in refrigeration rooms with a volume higher than 100 m³. An average energy saving from 6% to 17% was estimated with the implementation of simple maintenance practices and/or improvement on the cold room doors. Thus, the knowledge of the influence of the warm air infiltration into cold rooms is relevant since it allows setting up strategies, such as loading and unloading maps, collaborators training and awareness to the adoption of solutions/protection devices such
as strip PVC curtains, air curtains, flexible fast-opening doors and vestibules to improve energy efficiency [2–4].

Several analytical, experimental and/or numerical methods can be used to estimate the air infiltration rate into cold rooms. Foster et al. [8] carried out experimental measurements of a tracer gas concentration to calculate the air infiltration rate through cold room entrances with different sizes and with different air temperatures within the chilled space. The experimental results were compared with numerical predictions and with the results provided by analytical models. The results showed that analytical models developed by Gosney & Olama [9] and Fritzsche & Lilienblum [10] are those that best predict the air infiltration rate even with better accuracy than the computational model. However, when a transient analysis is performed, the analytical models provide inaccurate predictions of the air infiltration rate because the door opening and closing movements are not considered. Gonçalves et al. [11] experimentally measured the air infiltration rate into a cold room using a tracer gas technique. The experimental results were compared with numerical model predictions and with the results provided by analytical models. It was concluded that analytical models are not suitable for determining the air infiltration rate in a transient analysis. The predictions obtained with the numerical models are in agreement with experimental results, and then the solution was validated since numerical models give a closest prediction of the air infiltration rate into the cold room as well the air temperature and velocity fields, either inside or outside of the chilled space.

The tracer gas technique, such as found in both studies discussed above, is one of the experimental methods that can be used to determine the air infiltration rate [12,13]. Within the tracer gas technique, the tracer gas concentration decay technique is suitable for this purpose. This method consists in introducing a predetermined quantity of tracer gas, \( C_{\text{initial}} \), inside the cold room volume, \( V \) [m\(^3\)]. This gas will be mixed with indoor air to ensure a uniform concentration, \( C_1 \), can be determined after a time period, \( T \) [s], by the knowledge of the tracer gas concentration, \( C_{\text{final}} \), using Eq. (1) [2,11–13].

\[
I = \frac{V}{T} \ln \left( \frac{C_{\text{initial}}}{C_{\text{final}}} \right)
\]

Several studies have considered the development of analytical models to quantify the air infiltration over the time. Table 1 includes empirical mathematical expressions to determine the air infiltration, \( I \) [m\(^3\) s\(^{-1}\)]. All equations have a common set of variables. The density is an example, once the infiltration rate is directly related to the difference of density between the air of the interior, \( \rho_\text{i} \), and external, \( \rho_\text{ext} \), environments. In addition, the opening area, \( A \) [m\(^2\)], the height of the door, \( H \) [m], and the.acceleration due to gravity, \( g \) [m s\(^{-2}\)], are parameters to be used.

### Nomenclature

**General**
- \( A \): area [m\(^2\)]
- \( b \): wall thickness [m]
- \( C \): tracer gas concentration [%]
- \( C_a \): specific heat at constant pressure [kJ kg\(^{-1}\) K\(^{-1}\)]
- \( g \): gravity acceleration [9.81 m s\(^{-2}\)]
- \( H \): height [m]
- \( i \): infiltration [m\(^3\)]
- \( I \): infiltration rate [m\(^3\) s\(^{-1}\)]
- \( j_i \): diffusion flux [kg s\(^{-1}\) m\(^{-2}\)]
- \( k \): thermal conductivity [W m\(^{-1}\) K\(^{-1}\)], turbulent kinetic energy [m\(^2\) s\(^{-2}\)]
- \( K_{f,L} \): Fritzsche & Lilienblum correction factor
- \( L \): length [m]
- \( m \): mass [kg]
- \( n \): generic quantity
- \( p \): pressure [Pa]
- \( \bar{r} \): position vector [m]
- \( R \): gas-law constant [J kmol\(^{-1}\) K\(^{-1}\)]
- \( S \): source term
- \( t \): time [s]
- \( h \): height [m]
- \( v \): velocity magnitude – component in \( i \) direction [m s\(^{-1}\)]
- \( v \): velocity vector
- \( \nu \): velocity (average) [m s\(^{-1}\)]; specific volume [m\(^3\) kg\(^{-1}\)]
- \( V \): volume [m\(^3\)]
- \( W_g \): molecular weight of gas [kg kmol\(^{-1}\)]
- \( \alpha, \gamma, z \): spatial coordinate system [m]
- \( \chi_i \): spatial coordinate – component in \( i \) direction [m]
- \( \chi_i \): mass fraction of species \( i \) [kg kg\(_m\)^{-1}]

**Greek symbols**
- \( \delta \): Function Delta de Dirac
- \( \delta_y \): Kronecker tensor
- \( \Delta \): increment; range; difference
- \( \phi \): dependent variable (generic)
- \( \theta \): door opening degree [°]
- \( \lambda \): stopping criterion of iterative process
- \( \mu \): dynamic viscosity [kg m\(^{-1}\) s\(^{-1}\)]
- \( \rho \): density [kg m\(^{-3}\)]

**Lower rates**
- \( t_1 \): initial time
- \( t_2 \): final time
- \( \text{avg} \): average
- \( \text{ext} \): air outside the cold room
- \( i \): initial
- \( l, k \): component of cartesian directions \( x, y \) and \( z \)
- \( \text{in} \): air inside the cold room
- \( \text{ref} \): reference
- \( \text{rel} \): relative
- \( \text{tracer} \): tracer gas
- \( \text{var} \): variation of a quantity
- \( \text{vc} \): element; control volume
- \( \text{total} \): reference to the air total infiltration rate
- \( \phi \): generic dependent variable

**Higher rates**
- \( \rightarrow \): vector

<table>
<thead>
<tr>
<th>Authors/Year</th>
<th>Equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brown and Solvason [15]</td>
<td>( I = 0.343A(gH)0.65 \left( \frac{\rho_\text{i} - \rho_\text{ext}}{\rho_\text{i}} \right)^{0.5} \left[ 1 - 0.498(p) \right] )</td>
</tr>
<tr>
<td>Tamm [14]</td>
<td>( I = 0.333A(gH)0.5 \left( \frac{\rho_\text{i} - \rho_\text{ext}}{\rho_\text{i}} \right)^{0.5} \left( \frac{1}{1 + (\rho_\text{ext}/\rho_\text{i})} \right)^{1.5} )</td>
</tr>
<tr>
<td>Fritzsche and Lilienblum [10]</td>
<td>( K_{f,L} = 0.48 + 0.004(T_{\text{ext}} - T_{\text{in}}) )</td>
</tr>
<tr>
<td>Gosney and Olama [9]</td>
<td>( I = 0.333K_{f,L}A(gH)0.5 \left( \frac{\rho_\text{i} - \rho_\text{ext}}{\rho_\text{i}} \right)^{0.5} \left[ 1/1 + (\rho_\text{ext}/\rho_\text{i}) \right]^{1.5} )</td>
</tr>
<tr>
<td>Pham and Oliver [16]</td>
<td>( I = 0.226A(gH)0.5 \left( \frac{\rho_\text{i} - \rho_\text{ext}}{\rho_\text{i}} \right)^{0.5} \left( \frac{1}{1 + (\rho_\text{ext}/\rho_\text{i})} \right)^{1.5} )</td>
</tr>
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</table>
eration of gravity, \( g \) \( [\text{m} \text{s}^{-2}] \), are variables common to all equations. The main difference between equations is the infiltration coefficient obtained by each author through experimental studies \([2]\). The model developed by Tamm \([14]\) is an improvement of Brown & Solvason \([15]\) model, which is not used in this study since it considers the wall thickness in the doorway. Fritzschke & Lilienblum \([10]\) performed measurements of air velocity through anemometers and added a correction factor to the Tamm equation \([14]\). The correction factor takes into account the contracting of the flow (contraction coefficient) and thermal effects. This model assumes that the air volume flow rate entering the cold room is the same that leaves the chilled space. However, if air entering the cold room has a lower temperature, Fritzschke & Lilienblum \([10]\) assumption is not valid. Air flow rates into and out of the cold room are potentially different in both cases of an adjacent air that is cooler or warmer than the air inside the cold room. Gosney & Olama model \([9]\) considers the mass conservation and the experimental measurements allowed to get a different correction factor. Experiments conducted by Pham & Oliver \([16]\) allowed obtaining a correction factor that was included in the Tamm model \([14]\), which new equation was named Tamm’s modified equation.

The adoption of Computational fluid dynamics (CFD) by engineers connected to the food industry began in the 90s due to the larger processing power of computing resources and the emergence of specific and commercial software. Currently, it is a widely used tool due to the aforementioned benefits. Specifically, in the food industry, this technique allows modelling physical and chemical processes related to cooking, sterilizing, mixing, cooling and refrigerated storage of products, in order to study possible improvements to be adopted in practice \([4]\). Foster et al. \([8,17]\) conducted experimental and numerical studies of air infiltration through doors of different sizes and with two different temperatures inside cold rooms. Results were compared with results of analytical and CFD models. The tracer gas concentration decay method was used in the experimental tests. Gonçalves et al. \([18]\) performed a numerical study of the influence of air curtains in the obstruction to the infiltration of thermal loads to the interior of a cold room. For the different simulated case studies, the air curtains placed outside the cold room had better thermal performance. Air curtains with vertical jet have a greater efficiency (>70%) than the air curtains with horizontal jet (about 55%). Gaspar et al. \([19]\) studied the effect of ambient air conditions in the thermal performance and energy efficiency of vertical open refrigerated displays cabinets (ORDC) and its air curtain that provides a barrier to the external environment. Additionally, Gaspar et al. \([20]\) developed a comprehensive and detailed CFD modelling of airflow and heat transfer in an ORDC. Subsequently, Gaspar et al. \([21]\) introduced modifications in the CFD model related to low cost geometrical and functional characteristics to perform parametric studies with the aim to predict improvements of the global performance and energy efficiency.

Orlandi et al. \([22]\) performed a 3D numerical study of a closed refrigerated display cabinet with an air curtain and different types of doors, hinged and sliding doors to evaluate the air infiltration during doors opening and to better understand the physical processes associated with the energy transport. The door motion was modeled using two different methods. One method was based on an automatic remeshing algorithm while the second method consisted on several computational mesh sections, each one corresponding to a different door position. Then, the mesh and the boundary conditions were updated for each door position. This latter method, despite having a smaller movement discretization than the automatic remeshing algorithm, simulated accurately the door motion due to problems associated with the skewness of mesh elements when using automatic remeshing algorithm for large translational movements.

These studies contributed greatly to improve the perception of this type of phenomena and its influence on the performance of refrigeration equipment. Additionally, the approaches used in these studies serve as a theoretical supplement to the development of numerical models, thereby facilitating the approach in certain parameters that influence the results accuracy.

Thus, it was shown that CFD can be used to transient modelling of air masses movement between the cold room and external environment. If the CFD model includes a species model to consider a tracer gas, the numerical predictions of the species fraction can be used to determine the air infiltration rate during the door opening/closing cycle by the tracer gas concentration decay technique. This is the innovative application of CFD to this particular case, providing several benefits: (1) numerical simulation of an experimental technique, while neglecting the costs associated with experimental devices and consumables acquisition as well as the time required to perform the series of experimental tests; (2) detailed spatial and temporal evaluation of the air infiltration rate and of its influence on the air temperature and velocity fields inside the cold room.

So, the main purpose of this paper is to extends the abovementioned research results by using a three-dimensional (3D) transient CFD model to predict the air infiltration rate during the door \((L \times H = 1.2 \times 2.2 \text{ m}^2)\) opening/closing cycle for two door types (hinged and sliding door). The influence of the air infiltration rate during the door opening/closing cycles on the inner air temperature of the cold room is evaluated by setting different values for the vestibule air temperature (sensitivity analysis). Subsequently, the numerical results of the numerical models of two door types are compared for the reference vestibule temperature \((T_{\text{ref, ext}} = 15 \text{ °C})\) to quantify the best thermal performance of the cold room, considering minimization of air infiltration, air temperature and velocity fields. Finally, a comparative analysis between numerical predictions for the reference vestibule temperature of two door types with the results of analytical models was performed to validate the CFD model.

The novelty of this study relies on the use of the species equation for modelling the transport of a tracer gas. The tracer gas concentration decay method is used to determine the air infiltration rate through a hinged or a sliding door of a cold room. The knowledge about the evolution of the air infiltration during the door opening/closing times allows drawing some conclusions about its influence on the air conditions inside the cold room. Besides promoting the adoption and development of equipment with better efficiency, the results can be used to suggest costless best practices and simple technical improvements that can minimize air infiltration, and consequently improve thermal performance and energy consumption rationalization.

2. Materials and methods

CFD codes are based on numerical algorithms that allow the resolution of a set of differential equations, not analytically solvable, which describe the mathematical-physical model that underlies the case study. The numerical methods to solve CFD models are based on the finite volume formulation \([23]\), which involves dividing the entire computational domain into control volumes. The governing equations are integrated in each node to build algebraic equations for the unknown variables. Finally, the linearization of the discretized equations is performed and the resulting linear system of equations is solved. All methods and formulations that allow obtaining the solution should be properly chosen, as well as the computational mesh that must be created according to the flow properties and provided with slightly distorted elements so as not to affect the convergence of the solution \([23]\). Nevertheless,
while the very most of CFD software use finite volumes, finite element formulation of fluid flow equation are feasible.

This study uses the CFD code, Fluent, included in the computational tool ANSYS [24] to obtain the numerical predictions of air temperature and velocity fields in the door plane and inside the cold room, as well as the tracer gas concentration field on an isosurface of constant concentration value.

2.1. Governing equations

The governing equations of fluid flow and heat transfer can be considered mathematical formulations of the conservation laws of fluid mechanics and thermodynamics. When applied to a Newtonian fluid allows describing the rate of a particular fluid property when external forces are acting on it. The models of this study consider the laws of conservation of mass, or continuity, conservation of momentum and conservation of energy, expressed by Eq. (2)–(4), respectively [25]. The conservation of a quantity means that its variation inside any control volume can be described as the amount to be transported across its boundaries due to internal sources and forces as well external forces acting on the control volume. Thus, the flow through the boundary may be diffusive or convective [26].

\[ \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0 \]  
\[ \frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_k}(\rho u_i u_k) = -\frac{\partial p}{\partial x_i} + \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \rho g_i \]  
\[ \frac{\partial}{\partial t}(\rho C_T) + \frac{\partial}{\partial x_i}(\rho u_i C_T) = \frac{\partial}{\partial x_i} \left( \frac{k \frac{\partial T}{\partial x_i}}{C_1} \right) = S_i \]  

This study comprises three-dimensional (3D) and transient simulation of natural convection airflow through a door opening. In natural convection the Grashof number is the dimensionless parameter that governs the fluid flow. Since the value of the Grashof number is below the critical value, \( \text{Gr} = 1.5 \times 10^8 < 10^9 \) [27], it is assumed that the natural convection flow is in the laminar regime. Air behaves as an ideal and incompressible gas. Consequently, it is necessary to associate the thermodynamic properties of the fluid through the ideal gas law [23,25–28]. The equation of state shown in Eq. (5) relates the variations of fluid density with its temperature change while disregarding pressure variation [23].

\[ \rho = \frac{p_{ref} W_a}{RT} \]  

2.1.1. Species transport model

In this study, the calculation of the air infiltration rate allows to evaluate the influence of different variables related to the door opening/closing cycle in the thermal performance of cold rooms. These variables are: type and dimensions of the door, door opening/closing velocity, temperature difference between contiguous spaces, i.e. chilled and external spaced, among others. Thus, to determine the air infiltration rate it is necessary to consider the use of a species transport model which allows the creation of a fictitious fluid (tracer gas) with the same properties of air, but that virtually can be distinguished from it. The transient prediction of the concentration of this “new” fluid inside the cold room allows calculating the air infiltration rate by the tracer gas concentration decay technique using Eq. (1) [2]. The tracer gas component has the same properties than air. A constant concentration of tracer gas inside the cold room was set. The concentration of the tracer gas, \( C_{\text{tracer}} \), was set to 5000 ppm, i.e. \( C_{\text{tracer}} = 0.5\% \), following the procedure set by Foster et al. [8]. The concentration of tracer gas remains constant in the airtight cold room when the door is fully closed. When the door opening/closing cycle begins, the mass fraction of each species will vary due to the entrainment with external air. The mass fraction of each species, \( Y_i \), is predicted solving the convection-diffusion equation for the species \( i \), which is given by Eq. (6) [23,25].

\[ \frac{\partial}{\partial t}(\rho Y_i) + \nabla \cdot (\rho \vec{v} Y_i) = -\nabla \cdot \vec{J}_i + R_i + S_i \]  

2.2. Numerical model

2.2.1. Geometry/computational domain

The 3D geometry consists of two adjacent volumes, the interior volume corresponding to the geometry of the cold room, \( V_{\text{in}} = 96 - m^3 \), and the outside volume corresponding to the environment outside the cold room (vestibule), \( V_{\text{ext}} = 128 \text{ m}^3 \), which dimensions are shown in Table 2. It must be highlighted that both geometries have no other element in addition to its own volume and the geometry corresponding to the door (\( A = 1.2 \times 2.2 \text{ m}^2 \)) because the focus of this study is the analysis of the natural convection fluid flow through the door opening. The model simplification allows a substantial decrease of the computational effort without affecting the accuracy of results [17,18].

The simulation of the transient fluid flow due to the door movement requires using a mesh motion method to update the volume mesh in the deforming regions subject to the motion defined at the boundaries, i.e. the door faces. This procedure can be accomplished by two different methods. The first method implies to use a remeshing algorithm for dynamic mesh update. This algorithm agglomerates cells that surpass the skewness or size criteria and locally remeshes the agglomerated cells or faces [24]. However, due to the large boundary displacement when compared to the local cell sizes, the cell quality deteriorates significantly, i.e. there is a large number of cells with a skewness value higher than 0.9. This condition will lead to convergence problems when the solution is updated to the next time step. The second method consists in defining a computational mesh with several volume sections, each one corresponding to a different door position. During the transient simulation, the effect of the solid motion of the door on fluid flow is accomplished running a user defined function that updates the mesh zones (replacing sequentially a fluid zone by a solid zone) for each door position. This latter method, despite having smaller movement discretization than the automatic remeshing algorithm, simulates accurately the door motion, avoiding the convergence and accuracy problems associated with the skewness of mesh elements for large movements.

The computational domain consists in a different geometry for each door type. The geometry for the sliding door model is shown in Fig. 1 while the geometry for the hinged door model is shown in Fig. 2. 36 individual volumes (zones) were created for the region that simulates the door. Each zone corresponds to a different door position during the door opening/closing cycle (in each time step). Each zone corresponds to a \( \Delta \theta = 5^\circ \) arc in a 180° opening for the hinged door model. It was considered that the hinged door opens 180° instead of only opening 90°. Cleland et al. [29] evaluated

<table>
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<th>Table 2</th>
<th>Geometry dimensions.</th>
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<tr>
<td>Dimensions</td>
<td>Geometry name</td>
</tr>
<tr>
<td>Width (mm)</td>
<td>Length (mm)</td>
</tr>
<tr>
<td>Cold room</td>
<td>5000</td>
</tr>
<tr>
<td>Exterior environment</td>
<td>8000</td>
</tr>
<tr>
<td>Door</td>
<td>1200</td>
</tr>
</tbody>
</table>

* Door thickness.
experimentally the effect of door open angle. It was concluded that there was no significant difference in air infiltration rate when the door was fully opened (90° to the wall) rather than partially opened (45° to the wall). Thus, it can be expected the same air infiltration result between 90° and 180° openings. For the sliding door model, each zone corresponds to a width of Δx = 33.3 mm for a total opening of 1200 mm.

2.2.2. Computational mesh

The flow governing equations are highly non-linear and self-coupled. Therefore, the solution of the conservative governing equations requires using numerical techniques. The finite volume method is used to discretize the partial differential equations into a finite set of numerically solvable algebraic equations as described by Patankar [23]. Therefore, a spatial discretization of the computational domain into control volumes is performed. An unstructured mesh composed by tetrahedral elements is created. The mesh type ensures a better refinement in regions where high gradients are expected. CFD simulations require very high-quality meshes in both element shape and smoothness of sizes changes. The quality of the mesh plays a significant role in the accuracy and stability of the numerical computation. The attributes associated with mesh quality are node point distribution, smoothness, and skewness. The quality of the mesh elements was assessed during the meshes creation in order to balance the results accuracy with the computational cost and simulation time. Three key
parameters to measure the mesh quality were analysed: aspect ratio, skewness and orthogonal quality. The aspect ratio is a measure of the stretching of a cell. It is computed as the ratio of the largest and smallest dimensions of the edges of the faces of the control volume. A good aspect ratio has a value close to one. Skewness is defined as the difference between the shape of the cell and the shape of an equilateral cell of equivalent volume. The mesh quality improves as skewness value is towards null value. The orthogonal quality for cells is computed using the face normal vector, the vector from the cell centroid to the centroid of each of the adjacent cells, and the vector from the cell centroid to each of the faces. The orthogonal quality for a cell is computed as the minimum of these quantities. The mesh quality increases as the value of orthogonal quality is towards unity.

Several computational meshes were created until the size of mesh elements provided high values of aspect ratio and orthogonal quality and a small skewness, ensuring a good relation between the computational cost and the accuracy of the results [23,24].

Since the aim of the study is to simulate air infiltration through door opening for two types of door, it was necessary to create a computational mesh for each of geometries.

The mesh of the hinged door model has 10,828,944 elements, 1,836,073 nodes with an average aspect ratio of 1.87, an average skewness of 0.23 and average orthogonal quality of 0.85. These values for the parameters describing mesh quality were accomplished for a computational mesh of almost 11 million control volumes, which was a value near the computational limit of the hardware. The mesh of the sliding door geometry has a total of 2,818,425 elements.
elements and 494,475 nodes. The values of the mesh quality parameters were similar. Thus, despite the non-totally smooth description of the concentration and temperature gradients in the volume, the balance between volumes discretization (and consequently of results accuracy) and the computational resources required and computational time is suitable for an engineering application.

2.2.3. Initial and boundary conditions

In order to simplify the model some considerations were assumed, such as:

- There is no heat flow through the walls of cold room (adiabatic walls).
- Air can only move through the door opening (airtight cold room).
- Air humidity has a low influence in the air flow (water vapor was not considered in the mixture).
- The cold room walls have no thickness.

Taking into account these considerations, the boundary condition (BC) of pressure outlet type was set in the outside environment to the cold room (vestibule), with the purpose of keeping constant the air conditions in this zone during the simulation. The outer face adjacent to the cold room and the upper and lower surfaces were the only exceptions, being defined as wall BC. In the cold room, all surfaces were set as wall BC. Considering that cold room walls were assumed as adiabatic, a null heat flow was set for all surfaces defined as wall BC. As initial conditions, was assumed that model starts from steady state all over the domain, with air temperature of \( T_{in} = 0 \) °C inside the cold room, and \( T_{ext, ref} = 15 \) °C on the outside. Regarding the tracer gas concentration, in the chilled space was set as \( C_{tracer} = 0.5\% \) and \( C_{tracer} = 0\% \) on the outside. Both boundary and initial conditions were set in order to highlight and promote the airflow between contiguous spaces (cold room and vestibule), neglecting any thermal exchange through the envelope by heat conduction since the Rayleigh number, \( Ra = 1.06 \times 10^{8} \) is much larger than the critical Rayleigh number.

To simulate the door opening and closing movement, it is necessary at each time step to change the door boundary conditions. The procedures for hinged and sliding door are distinct. For hinged door, firstly was created the representative volume for the door opening. In this zone, the faces corresponding to the door opening were defined as interface to subsequently be able to create a mesh interface to associate the two interfaces as an interior zone (fluid

![Fig. 5. Numerical predictions for the sliding door model. (a) Temperature field in yz plane at x = 2750 mm. and (b) 3D iso-surface of tracer gas concentration at 0.25% during the door opening.](image-url)
region), through which there is fluid flow. Therefore, in this region all governing equations are solved. To simulate the door opening and closing movement, the boundary condition type for each of the faces belonging to the 36 geometries is switched between wall and interior according to the level of opening/closing in that time step. Fig. 3 shows how boundary conditions are set in the door zone for the hinged door. The shaded zone represents the mesh interface corresponding to the door opening (this face is a wall BC type in the initial time step), and the zones numbered with 2 and 3 represent the faces corresponding to different positions of door opening which are obtained by alternating the BCs type between wall and interior.

It is also required the creation of a mesh interface which represents the door opening in the case of the sliding door. However, the door opening position in this model is given by the number of faces that compose the interface mesh. As shown in Fig. 4, since each volume has a width of 33.3 mm, the door opening of 33.3 mm requires face n.\textsuperscript{2} to be set as interface BC in order to create a mesh interface. Proceeding to an opening of 66.6 mm in the next time step, the BC of face n.\textsuperscript{2} remains as interface and face n.\textsuperscript{3} (belonging to the next volume), which was set as wall BC, is set as interface BC creating a mesh interface covering these two volumes. The simulation of the door closing movement requires sequentially changing the BC type from interface to wall of the faces of the volumes constituting the door.

2.2.4. Numerical method and solution convergence control

The 3D transient numerical model to predict the airflow through the door opening solves the conservation equations for mass, momentum and energy. The second order Upwind scheme was used to discretize the equations. The linear relaxation method is used to reduce the high variation of dependent variables during the iterative process. A first-order implicit time integration method was used to perform the temporal discretization that involves the integration of each term in the differential equations over a time step. This method shows a good stability relatively to the time step and performs an assessment of the spatial discretization in the later time step as shown by Patankar [23]. The convergence monitoring process evaluates the sums of absolute residuals of mean field variables. The iterative process per time step stops when the convergence criterion is meet, \( \lambda \leq 1 \times 10^{-3} \). This criterion was used for all residuals except energy, which used \( \lambda \leq 1 \times 10^{-6} \). The solution does not diverge throughout the simulations. However, in certain time steps, the solution residuals stabilize in a value close to, but above, the stop criterion preventing the end of the iterative process. In such cases, the iterative process per time

\[ \Delta t = 1200 \text{ mm opening at } t = 11 \text{ sec.} \]

\[ \Delta t = 300 \text{ mm opening at } t = 13.25 \text{ sec.} \]

\[ \Delta t = 33.3 \text{ mm opening at } t = 14 \text{ sec.} \]

Fig. 6. Numerical predictions for the sliding door model. (a) Temperature field in \( yz \) plane at \( x = 2750 \) mm; and (b) 3D iso-surface of tracer gas concentration at 0.25%, during the door closing.
step ends when 50 iterations were performed, since the possible convergence of the solution was reached. This criterion for the maximum number of iterations was defined after performing some tests to assess the convergence of the solution. Both numerical models were performed on a computer with an i7 CPU – 3.40 GHz 64 bit and 8 GB of RAM. For the hinged and sliding door models, each complete door opening/closing cycle took about 168 h and 84 h respectively.

2.2.5. Door opening/closing times

The time considered for the door opening/closing cycle was the same for both models:

- Door opens from \( t = 0 \)– 3 s.
- Door remains fully open during \( \Delta t = 8 \) s, i.e. up to \( t = 11 \) s.
- Door closes in \( \Delta t = 3 \) s, i.e. at \( t = 14 \) s the door is fully closed.

These are the average times per procedure determined by Nunes et al. [30].

The time step, i.e. the time between each door movement, was determined from the total time of the opening/closing cycle. In the case of the sliding door, for example, the door movement is performed in steps of \( \Delta x = 33.3 \) mm. Thus, the time step is approximately equal to \( \Delta t = 83.33 \) ms.

3. Results and discussion

This section analyses and discusses the results of the sensitivity analysis and compares the numerical predictions for both door typologies in relation to the reference model (\( T_{\text{ext, ref}} = 15^\circ \text{C} \)).

The sensitivity analysis determines the effect of the variation of a particular parameter in the reference solution model. Initially, the reference model simulation in which the air temperature outside the cold room is \( T_{\text{ext, ref}} = 15^\circ \text{C} \) was performed for each door model. This temperature value is commonly found in the vestibules of cold rooms [29]. Subsequently, two computational models per door typology were developed that consider a decrease and increase of one degree Celsius in the air temperature value outside the cold room, i.e., models with \( T_{\text{ext, case1}} = 14^\circ \text{C} \) and \( T_{\text{ext, case2}} = 16^\circ \text{C} \).

The log of tracer gas concentration in each time step during all simulation allows calculating the air infiltration rate by Eq. (1). The

![Fig. 7. Numerical predictions for the hinged door model. (a) Temperature field in yz plane at x = 2750 mm; and (b) 3D iso-surface of tracer gas concentration at 0.25%.](image-url)
average value of the air temperature inside the cold room was recorded in each time step to evaluate its variation during the opening/closing cycle of doors and depending on the external ambient conditions.

### 3.1. Influence of air temperature variation outside the cold room

As mentioned above, the sensitivity analysis allows evaluating the influence of the air temperature variation outside the cold room, for each of the door types, on the airflow between the two rooms, as well on the air temperature field inside the cold room. In this sense, the predictions of the air temperature field inside the cold room in a perpendicular plane at the middle of the door (yz plane at \( x = 2750 \text{ mm} \)), and of the tracer gas flow in a 3D perspective of the computational domain are presented. The iso-surface of tracer gas concentration is shown for \( C_{\text{tracer}} = 0.25\% \). This tracer gas concentration was chosen because the airflow has already some development, i.e., there is airflow between interior and exterior of the chilled space, and so, some thermal interaction between the air of two rooms is expected. Figs. 5 and 6 show the abovementioned predictions during the sliding door opening/closing movement. Both figures refer to the numerical predictions of the reference model, \( T_{\text{ext, ref}} = 15 \text{ °C} \). The predictions for the CFD models with \( T_{\text{ext, case1}} \) and \( T_{\text{ext, case2}} \) are very similar to the reference model predictions. The main difference is related to the value of the air temperature between the interior and exterior of cold room, which is lower and upper, respectively. Consequently, the thermal interaction across the door occurs at lower and higher velocity for the \( T_{\text{ext, case1}} = 14 \text{ °C} \) and \( T_{\text{ext, case2}} = 16 \text{ °C} \) respectively.

Fig. 5 shows how the opening movement of the sliding door increases the airflow infiltration and consequently the thermal entrainment. A neutral pressure line, designated by neutral pressure elevation, is predicted. The hot air flows from the external environment to the cold room above this neutral pressure. Below this neutral pressure elevation, the air mass flows in the opposite direction, i.e., the cold air flows from the interior of cold room to its exterior. As larger is the door opening, more easily this phenomenon can be observed. The formation of vortex within the cold room was predicted, which dimension is more evident when the door opening is larger, i.e. in \( x = 900 \text{ mm} \) and \( x = 1200 \text{ mm} \). Fig. 6 shows the sliding door closing movement. After 8 s of the door closing.
the air infiltrated in the chilled space, rises and moves along the ceiling, creating a layer of warm air near the ceiling. The airflow prediction of the iso-surface of tracer gas concentration during the door closing movement shows the drag caused by the door movement. The door movement induces the movement of a large mass of air into the cold room (zone above the neutral pressure elevation).

3.2. Sensitivity analysis for sliding door model

Fig. 9 shows the numerical predictions of the air infiltration rate profile during the opening/closing cycle for different air temperatures outside the cold room. The figure analysis suggests that the air infiltration rate at each time step increases with the difference of air temperature between the two rooms. Therefore, the difference of air temperature between the two rooms and the increase/decrease of exchange area during door opening/closing movements lead to a large airflow between the two environments. This same result was obtained experimentally by Navaz et al. [31].

Table 3 divides the air infiltration rate for each moment of the door opening/closing cycle. The greater the difference of air temperature between the two rooms, the higher the air infiltration rate at each moment. As expected, the air infiltration rate is greater when the door is kept fully open during Δt = 8 s.

The results show that, on average, for each °C of air temperature increase outside the cold room (or the air temperature difference between the two rooms), the air infiltration rate through the sliding door opening with a cross section of A = 2.64 m² is I = 0.012 m³ s⁻¹ i.e., increases 12 L per second. In relation to the reference model (T_{ext, ref} = 15 °C), the air infiltration rate decreases 2.3% and increases 3.7% respectively, decreasing and increasing the air temperature outside the cold room by 1 °C.

A similar analysis can be performed concerning the average value of the air temperature inside the cold room. At the end of the cycle, the average air temperature inside the cold room is T_{in} = 0.78 °C, T_{in} = 0.88 °C and T_{in} = 0.96 °C, for T_{ext, case1} = 14 °C, T_{ext, ref} = 15 °C and T_{ext, case2} = 16 °C, respectively. Thus, on average, for each °C of air temperature increase outside the cold room (or air temperature difference between two rooms), the air temperature inside the cold room with a sliding door of A = 2.64 m² increases ΔT_{in} = 0.09 °C. The variation of the average air temperature inside of cold room can also be divided during the door movement (opening, fully open, closing) as shown in Table 4. The predictions of the air temperature inside the cold room suggests that, as expected, it increases with the increase of air temperature difference between the two rooms.
3.3. Sensitivity analysis for hinged door model

The same kind of analysis was performed for the case study of hinged door. Fig. 10 suggests that the greater the air temperature difference between the two rooms, the greater is the air infiltration rate at each moment. Thus, the difference of air temperature between the two rooms and the increase/decrease of the door opening area (during opening and closing movements) lead to larger flow between the two environments than for the sliding door case. For the time period corresponding to the door closing movement, it can be seen that the air infiltration rate does not decrease suddenly as it is predicted for the sliding door case. Instead, it is predicted that before the air infiltration rate starts to decrease due to the exchange area reduction, there is a period of time that increases due to the drag of ambient air inside the cold room during the closing movement.

Table 5 includes the predictions of the average air infiltration rate for the opening, fully open and closing times of the hinged door. The average air infiltration rate is greater when the door is kept fully open during \(\Delta t = 8\) s.

The average air infiltration rate at the end of the cycle for different air temperatures outside the cold room, for the hinged door case study is: \(I = 0.47\) m\(^3\) s\(^{-1}\), \(I = 0.49\) m\(^3\) s\(^{-1}\), and \(I = 0.51\) m\(^3\) s\(^{-1}\) for the hinged door, respectively. The results show that, on average, air temperature increase outside the cold room (or air temperature difference between two rooms), the air infiltration rate through the door opening is \(I = 0.016\) m\(^3\) s\(^{-1}\), i.e., increases 16 L per second, considering a hinged door with an area \(A = 2.64\) m\(^2\). The results show that relatively to the reference model (\(T_{\text{ext, ref}} = 15^\circ\text{C}\)), the air infiltration rate decreases 3.1% and increases 3.5% respectively, decreasing and increasing the air temperature inside the cold room by \(1^\circ\text{C}\).

The average air temperature inside the cold room at the end of the opening/closing cycle is \(T_{\text{in}} = 0.93^\circ\text{C}\), \(T_{\text{in}} = 1.03^\circ\text{C}\), and \(T_{\text{in}} = 1.14^\circ\text{C}\), for the sliding door model. These results mean that, on average, air temperature increase outside the cold room (or air temperature difference between two rooms), the air temperature inside the cold room increases 0.10 \(^\circ\text{C}\), considering a hinged door with an area \(A = 2.64\) m\(^2\). (see Table 6).

3.4. Comparison of air infiltration rate through hinged and sliding doors

The comparative analysis of the air infiltration rate allows evaluating the type of door more suitable for the air infiltration minimization and, consequently, for the lower thermal entrainment inside the cold room due to door openings.

Fig. 11 shows the profile of the numerical predictions of air infiltration rate for both door types. The figure analysis suggests that during all door opening/closing cycle the air infiltration rate is larger for the hinged than for the sliding door type. The air infiltration rate is approximately equal for both door types in an instant \((t = 4.6\) s\) during the period that the door is fully open. At this time, the air infiltration rate achieves its maximum value in the sliding door type and the minimum value in the hinged door type.

The average air infiltration rate of the hinged door case \((I = 0.49\) m\(^3\) s\(^{-1}\)) is 19.5% higher than the sliding door case \((I = 0.41\) m\(^3\) s\(^{-1}\)), i.e., 7% of the total air volume renovation is performed in a 14 s cycle. Thus, a complete air volume renovation would occur if the door remain fully open during approximately 3 min. Table 7 includes the linear correlations that relate the air infiltration rate with the exterior air temperature.

Fig. 12 and Table 8 show the profile of the average air temperature inside the cold room during the door opening/closing cycle for the reference air temperature outside the cold room \((T_{\text{ext, ref}} = 15^\circ\text{C})\) and for each type of door. The main differences between door type results are predicted in the opening and closing periods. During the door opening period, the average air temperature inside the cold room increases, \(\Delta T = 0.0028^\circ\text{C}\) per second and \(\Delta T = 0.0048^\circ\text{C}\) per second for sliding and hinged door type.
respectively. During the period that the door remains fully open, the average air temperature inside the cold room increases approximately $\Delta T = 0.0067 \degree C$ per second. Finally, for the door closing period, the average air temperature inside the cold room increases $\Delta T = 0.0035 \degree C$ per second and $\Delta T = 0.0061 \degree C$ per second for the sliding and hinged door, respectively. From these results, it is verified that the closing movement of hinged doors drags a large amount of external air into the cold room. At the end of the cycle, the average air temperature inside the cold room is $T_{in} = 0.88 \degree C$ and $T_{in} = 1.03 \degree C$ for the sliding and hinged door cases, respectively. Therefore, the average air temperature inside the cold room is 17% greater in the hinged door case. This fact describes the direct influence of the air infiltration rate in the air temperature of the chilled space.

### 3.5. Comparison between numerical predictions and analytical models results

This section presents the comparison between the results of the air infiltration rate through the door opening determined by the numerical models (sliding and hinged doors) for reference model ($T_{ext, ref} = 15 \degree C$) and the analytical models described in the introduction section.

Figs. 13 and 14 show the comparative results of air infiltration rate for sliding and hinged door cases. The figures observation allows, instinctively, assess that in general the numerical models underestimate the air infiltration rate, especially, in the door open-
ing and closing periods. In the case of the sliding door, the air infiltration results determined with the numerical model predictions underestimate the air infiltration rate during all the door opening/closing cycle, except when compared with the Fritzschke & Lilienblum [10] model results. In this situation, the numerical model overestimates the air infiltration rate, especially during the fully open and closing door periods. When the results are compared with those obtained by Gosney & Olama [9] and Fritzschke & Lilienblum [10] models, it can be concluded that the numerical model predicts closely the air infiltration rate due to the reliability of these analytical models.

For the hinged door case, the numerical predictions follow closely the results obtained Gosney & Olama [9] and Fritzschke & Lilienblum [10] models, especially during the fully open period ($3 \, s < t \leq 11 \, s$).

From the comparative analysis performed for the sliding and hinged door cases, it can be seen that the analytical models consider only the air infiltration in steady state, i.e., only during the fully open period. In the hinged door case, the comparison between numerical and analytical models is reliable only for the period of time that door remains fully open, $3 \, s < t \leq 11 \, s$, because the analytical model do not consider the opening and closing periods. In the calculation of the air infiltration rate through the hinged door case using the analytical models, the opening exposed area to air infiltration is equal to the total opening area (2.64 m²) from the first time step ($\theta = 5^\circ$ opening), although the door constitutes a

![Fig. 13. Air infiltration rate: comparison of numerical predictions and analytical models results - sliding door case.](image)

![Fig. 14. Air infiltration rate: comparison of numerical predictions and analytical models results - hinged door case.](image)
barrier to the air flow. As the analytical models consider only the variation of the opening area, then the air infiltration rate is constant during all door opening/closing cycle.

Two distinct analyses were carried out in order to compare the numerical predictions with analytical models results. The aim was to compare and evaluate the air infiltration rate percentage variation (ΔI) at different periods. The first analysis corresponds to the complete door opening/closing cycle, i.e., the percentage variation of the total air infiltration rate obtained with the numerical in relation to the analytical models. The second analysis corresponds only to the door fully open period (3 s < t ≤ 11 s).

Table 9 shows the results for both analysis. Comparing the numerical predictions with numerical models results, the numerical predictions have similar results to the Gosney & Olama [9] and Fritzsche & Lilienblum [10] models. Therefore, due to the accuracy of the models, it is possible to conclude that the numerical models predicts with proximity the air infiltration rate. Analyzing the percentage variation of the total air infiltration rate relatively to the Fritzsche & Lilienblum [10] model, is possible to conclude that the total air infiltration rate for the sliding door case calculated based on the numerical predictions is ΔI = 1.4% and ΔI = 2.3% greater at the end of the cycle and during the fully open period, respectively. The difference is slightly higher for the hinged door case. The total air infiltration rate calculated based on numerical predictions is ΔI = 4.7% lower and ΔI = 6.7% greater at the end of the cycle and during the fully open period, respectively. For Gosney & Olama [9] model, the percentage variation of the total air infiltration rate for both periods (complete or fully open) are higher. These results, especially, during the period when the door remains fully open are in agreement with results of Hendrix [32], who showed that the air infiltration rate calculated based on the Gosney & Olama [9] model was slightly higher (about 10%) than the determined in experimental analysis.

The results of the percentage variation of the total air infiltration rate for the hinged door case emphasize the fact that analytical models do not consider the door opening and closing periods, so there was a divergence between analytical models results and numerical predictions. When the analysis corresponds to the period when the door remains fully open, it can be seen that the numerical models predictions are accurate due to the proximity to the analytical models results, especially with the Gosney & Olama [9] and Fritzsche & Lilienblum [10] models.

The numerical models predictions are in agreement with the analytical models results during the fully open time, i.e., in steady state. Chen et al. [33] showed experimentally that the airflow through the door while it was opening and closing was equivalent to the airflow if it had been fully open for half of the time required to open or close. For the sliding door reference case, the comparison of the total air volume infiltrated between the numerical predictions (V = 5.663 m³) and by calculation using Chen et al. [33] relationship during opening and closing times and the Fritzsche & Lilienblum [10] model during the fully open time (V = 5.738 m³), accesses the reliability of the numerical results since the difference is in order of 1%. Thus, the numerical modelling can predict the air infiltration rate in transient regime taking into account the 3D effects of the airflow and other effects that affect the linearity of the results.

4. Conclusions

This paper presents the comparison of three-dimensional and transient CFD modelling of the opening and closing processes of hinged and sliding doors and its influence on the thermal performance of cold rooms. These models included the simulation of the experimental technique, tracer gas concentration decay method, to determine the air infiltration rate. This simulation was accomplished using a species transport model for the tracer gas. The tracer gas concentration was predicted each time step, i.e., every 83.3 ms. The interval between measurements is much greater in an experimental analysis reducing the possibility of obtaining a detailed variation profile of the air infiltration rate. Thus the numerical models developed allowed modelling an experimental technique, obtaining accurate results during all the transient door opening/closing cycle to, subsequently, trace the variation of air infiltration rate into the cold room.

The numerical predictions of the parametric study for each type of door evaluated the airflow characteristics between the cold room and exterior environment and the influence of the temperature value of the external hot air on the air infiltration rate and air temperature field inside the cold room.

There is a line where the pressure difference between the two rooms is zero, leading to a pressure gradient between two rooms with a value that depends on the door height. Thus, the pressure below the neutral line increases more rapidly in the cold room than outside, leading to the airflow from the interior of cold room to its exterior in the zone below the neutral line. On the other hand, above the neutral pressure line, the pressure inside the cold room decreases more rapidly as compared with the decrease in pressure outside it. This leads the airflow from the exterior to the interior of cold room.

The main results of the analysis are: The air infiltration rate increases with the difference of air temperature between the two rooms (12 L s⁻¹ and 16 L s⁻¹ for sliding and hinged door types, respectively). Consequently, the average air temperature inside the cold room also increases (0.09 °C and 0.10 °C for sliding and hinged door types, respectively). When a hinged door is used, the formation of vortices during the opening movement is a factor that promotes the rapid thermal interaction between the air masses of the two rooms as result of the existence of forces generated by the movement of the door itself.

The comparative study of two door types for the reference model, i.e., assuming an external temperature, T_{ext, ref} = 15 °C, allowed to determine that the sliding door type is most suitable in terms of air infiltration minimization (19.5% less). Consequently, the average air temperature inside the cold room is 17% lower than the hinged door typology. The door typology influences the airflow development. It is noted that the airflow and, consequently, the thermal interaction occurs more rapidly when a hinged door is used. The movement performed by each door causes this different behavior. The rotational movement performed by the hinged door causes the movement of large air masses. Suction forces are generated during the door opening movement leading to a high air movement between the two rooms. The door closing movement induces forces that push the outside air into the cold room. For the sliding door, the linear motion does not generate
these kind of effects that force the airflow between the two rooms. Thus, the air infiltration occurs only due to the door opening area variation and the air temperature gradient.

A comparison of the air infiltration rate through the door opening into the cold room determined from the numerical predictions with the results of analytical models was performed. The air infiltration rate varies over the time due to the exchange cross section. However, the analytical models results admit it as being permanent. The difference between the numerical predictions and the analytical models results is highlighted for the hinged door case. The analytical models do not consider the existence of the door, which acts like a blockage of the airflow since the door cross section is constant for a hinged door. A comparison of numerical models predictions and analytical models results allowed concluding that for both sliding and hinged door cases, the numerical models predict with proximity the air infiltration rate obtained through the Gosney & Olama [9] and Fritzsche & Lilienblum [10] models, giving a good dynamic estimate for the period under review.

References


