Numerical and experimental validation procedures for the simulation of refrigerated rooms

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Abstract

This paper presents two techniques (models), one computational and another experimental, to study the three-dimensional turbulent non-isothermal flow in refrigerated chambers. The computational model consists on a numerical procedure which solves, in finite-volume method, the three-dimensional time-averaged equations for the conservation of mass, momentum, energy and concentration of species. The effect of turbulence is described by the k-ε model. The predict results were experimentally validated through the comparison with measurements taken from an experimental reduced-scale model, designed to provide similarity with the real chamber. Due to the limitations of the similarity theory based on Prandtl, Reynolds and Archimedes numbers, a simple physical modelling technique, based on the dimensional analysis, was used to derive the physical properties of the laboratory model. In order to demonstrate the models applicability, two-case study of refrigerated chambers (natural convection and forced convection) was performed for the simulation and visualization of the thermal, humidity and air flow patterns. Agreement between measured and predicted values was very satisfactory for engineering purposes.
1 Introduction

In the past, the design of refrigerated rooms and the choice of operating conditions have largely been achieved by trial and error. They perform their functions reasonably well and, until recently, there was not much need to question seriously their design. Nowadays, however, this need has emerged in the wake of the soaring importance of energy saving and good cold quality for food storage. There are many configurations of refrigerated rooms and a wide range of potential operation conditions. Real chambers are never simple, and the range of shapes, design standards and external conditions is infinitely varied. The expected performance of a refrigerated room with an air distribution system in some cases can be predicted from past experience and established design procedure. However, in other cases and particularly when non-conventional methods of air distribution are employed, physical modelling and or mathematical modelling is crucial to the design of refrigerated chambers in providing cold quality as well as in increasing the energy efficiency of mechanical and electrical systems.

Much can be learned from the careful instrumentation and monitoring of the daily operation of existing chambers. However, full-scale tests of real refrigerated rooms are often prohibitively expensive and time-consuming, requiring sophisticated sensors and instrumentation. Therefore, experimental data on refrigeration chambers, particularly for trucks, trailers and containers, are extremely scarce. An experimental real scale study of refrigerated truck chambers has been presented by Nieuwenhuizen and Meffert [1], which reported comparative temperature measurements on three thin-wall and one conventional road refrigerated vehicle. The use of small-scale models offers an attractive and viable solution to the problem of the full-scale tests, but experimental results must be obtained in similarity conditions. Strict adherence of the similarity theory, based on Prandtl, Reynolds and Archimedes numbers, is the major limitation. Pitarma [2] did experimental measurements in a reduced-scale model of a refrigerated truck chamber with realistic loads and developed CFD simulations. Decreasing costs of computers allied to their development gave way to numerical methods in this field of investigations. Numerical simulation offers the ability to predict cold characteristics over a wide range of parameters and physical configurations at much less cost. However, the existence of reliable experimental results is fundamentally important for the validation and improvement of the mathematical models. A numerical simulation of a refrigerated room with the help of the advanced package Phoenics code was performed by Wang and Touber [3]. Pitarma [4] presents numerical predictions and experimental validation (temperature) of the three-dimensional turbulent flow inside yogurt truck refrigerated chamber with forced convection.

The objective of the present paper is to describe mathematical and experimental (reduced-scale) models to study the three-dimensional turbulent air flow patterns with thermal buoyant effects, the heat transfer and the moist air transport within mechanically or naturally refrigerated chambers. The extension of the computational model to the stored cargo were also presented and
validated. A case study is outlined. The laboratory model, designed to provide similarity with the real chamber, is described in section 2 followed by the mathematical model in section 3. The results are presented in section 4 and the conclusions presented in section 5.

2 Laboratory modelling

2.1 Modelling strategy

Mathematical models require experimental validation. While full-scale measurements of refrigerated rooms are very time consuming and expensive, the use of small-scale models offers an attractive and viable solution. For the results from reduced-scale model tests of problems in fluid dynamics to be applicable to the prototype, geometric, kinematic and thermal similarity between model and prototype must be achieved. Geometric similarity is a pre-requisite for any modelling investigation. For isothermal flows, geometric and kinematic similarity must be present and these can be usually achieved without too much difficulty. However, for non-isothermal flows all three similarity requirements should theoretically be present before a complete simulation of the flow in the chamber can be achieved. Therefore, the generally accepted statement from similitude theory of is that a scale model will perfectly replicate the kinematic response of its prototype, if the Prandtl, Reynolds and Arquimedes numbers are identical in both systems. In practice, it is not possible to provide a complete similarity for non-isothermal flow and, as a result, difficulty in interpreting the model results may be experienced. This constraint will invariably lead to the conclusion that convective heat transfer can only be accurately modelled on a real 1:1 scale (see, for instance, Awbi [5]), but such research is often prohibitively expensive and time-consuming.

Due to the limitations imposed by the theory of similitude mentioned above, another method can be proposed (see Imbabi [6]). Dimensional analysis is commonly used in experimental studies to transform physical parameters from prototype to model values. Buckingham's $\pi$-theorem is ideally suited for the manipulation of large numbers of variables, so this technique is used here to obtained similarity between model and prototype (see Pitarma [2] for further details).

In fact, if the physical phenomenon at hand is governed by a functional relation $f$, which involves $n$ variables

$$f (x_1, x_2, x_3, \ldots, x_n)=0$$

(1)

this problem can be equally described by another functional relation $\xi$, more compact, just involving $k=n-m$ dimensionless groups ($\pi$’s)

$$\xi (\pi_1, \pi_2, \ldots, \pi_k)=0$$

(2)
where \( m \) is the number of independent variables in the equation (1). The similarity between model and prototype is obtained through the establishment of the equalities

\[
\pi_{\text{model}} = \pi_{\text{prototype}}
\]  

Figure 1: Sketch of the prototype chamber configuration and the symmetry planes. a) Natural convection; b) Forced convection.

### 2.2 Strategy application

#### 2.2.1 Prototype chamber

The prototype refrigerated room used in this study is a 3.80x1.80x1.65m\(^3\) polyurethane-insulated truck chamber for practical operation of transportation of perishables (used in retail delivery). The chamber, in case of forced convection, is equipped with the common method of refrigerated system and air distribution used in truck chambers, represented in schematic diagram in Figure 1a. The air flows into the room, in the longitudinal direction, through a supply square opening placed close to the ceiling and leaves the chamber through another square hole below. For the version of natural convection, the chamber is equipped with two eutectic plate located in the side walls close to the ceiling as presented in Figure 1b. The geometrical and functional parameters are shown in Table 1.

#### 2.2.2 Laboratory model

Thus, if \( L \), \( T_e \), \( C_p \) and \( \rho \) are chosen as independent variables, the various dimensionless groups, detailed in the Table 1, may be derived. Finally, the similarity between model and prototype is obtained through the establishment of the equalities \( \pi_{\text{model}} = \pi_{\text{prototype}} \), that was used to derive the model parameters listed in Table 2. Naturally that geometric form of room is preserved by the model. In accordance with the modelling strategy, the experimental enclosure consists of a
1,52x0,72x0,66m³ box with 6mm thick Perspex walls. The entire apparatus is insulated with two snugly fitting layers of 50mm thick expanded polystyrene, in order to achieve insulating thermal conditions given in the same table.

Table 1: Dimensionless groups.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>SI Units</th>
<th>Dimension</th>
<th>Dimensionless groups</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside air temperature</td>
<td>Te</td>
<td>K</td>
<td>[0]</td>
<td>—</td>
</tr>
<tr>
<td>Density of air</td>
<td>ρ</td>
<td>kgm⁻³</td>
<td>[ML⁻³]</td>
<td>—</td>
</tr>
<tr>
<td>Specific heat cap. of air</td>
<td>cp</td>
<td>Jkg⁻¹K⁻¹</td>
<td>[L²T⁻²θ⁻¹]</td>
<td>—</td>
</tr>
<tr>
<td>Thermal load</td>
<td>Q</td>
<td>W</td>
<td>[ML²T⁻³]</td>
<td>( π₁ = Qte^{-1.5}cp^{-1.5}ρ^{-1}L^{-2} )</td>
</tr>
<tr>
<td>Wall thermal resistance</td>
<td>R</td>
<td>m²KW⁻¹</td>
<td>[T³θM⁻¹]</td>
<td>( π₂ = Ute^{-0.5}cp^{-1.5}ρ^{-1} )</td>
</tr>
<tr>
<td>Length</td>
<td>L</td>
<td>m</td>
<td>[L]</td>
<td>—</td>
</tr>
<tr>
<td>Width</td>
<td>W</td>
<td>m</td>
<td>[L]</td>
<td>( π₃ = WL^{-1} )</td>
</tr>
<tr>
<td>Heigh</td>
<td>H</td>
<td>m</td>
<td>[L]</td>
<td>( π₄ = HL^{-1} )</td>
</tr>
<tr>
<td>Time</td>
<td>t</td>
<td>s</td>
<td>[T]</td>
<td>( π₅ = tTe^{0.5}cp^{0.5}L^{-1} )</td>
</tr>
<tr>
<td>Convection coefficient</td>
<td>h</td>
<td>Wm⁻²K⁻¹</td>
<td>[MT⁻³θ⁻¹]</td>
<td>( π₆ = hTe^{-0.5}cp^{-1.5}ρ^{-1} )</td>
</tr>
<tr>
<td>Air velocity</td>
<td>v</td>
<td>ms⁻¹</td>
<td>[LT⁻¹]</td>
<td>( π₇ = vTe^{-0.5}cp^{-0.5} )</td>
</tr>
</tbody>
</table>

Forced Convection

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>SI Units</th>
<th>Dimension</th>
<th>Dimensionless groups</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet air temperature</td>
<td>To</td>
<td>K</td>
<td>[0]</td>
<td>( π₈ = ToTe^{-1} )</td>
</tr>
<tr>
<td>Inlet width</td>
<td>Wi</td>
<td>m</td>
<td>[L]</td>
<td>( π₉ = WiL^{-1} )</td>
</tr>
<tr>
<td>Inlet height</td>
<td>Hi</td>
<td>m</td>
<td>[L]</td>
<td>( π₁₀ = HiL^{-1} )</td>
</tr>
<tr>
<td>Outlet width</td>
<td>WR</td>
<td>m</td>
<td>[L]</td>
<td>( π₁₁ = WRL^{-1} )</td>
</tr>
<tr>
<td>Outlet height</td>
<td>HR</td>
<td>m</td>
<td>[L]</td>
<td>( π₁₂ = HRL^{-1} )</td>
</tr>
<tr>
<td>Air flow rate</td>
<td>V</td>
<td>m³s⁻¹</td>
<td>[L³T⁻¹]</td>
<td>( π₁₃ = VTe^{-0.5}cp^{-0.5}L^{-2} )</td>
</tr>
<tr>
<td>Air velocity</td>
<td>v</td>
<td>ms⁻¹</td>
<td>[LT⁻¹]</td>
<td>( π₁₄ = vTe^{-0.5}cp^{-0.5} )</td>
</tr>
</tbody>
</table>

Natural Convection

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>SI Units</th>
<th>Dimension</th>
<th>Dimensionless groups</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate temperature</td>
<td>Tp</td>
<td>K</td>
<td>[0]</td>
<td>( π₈ = TpTe^{-1} )</td>
</tr>
<tr>
<td>Plate with</td>
<td>Wp</td>
<td>m</td>
<td>[L]</td>
<td>( π₉ = WpL^{-1} )</td>
</tr>
<tr>
<td>Plate height</td>
<td>Hp</td>
<td>m</td>
<td>[L]</td>
<td>( π₁₀ = HpL^{-1} )</td>
</tr>
</tbody>
</table>

The experiments in the natural convection model were performed using the installation schematic shown in Figure 2. The installation is described in detail in Pitarma [2] and consists of two eutectic plates linked by a refrigerant line to a thermostatic bath in order to achieve the operation conditions given in Table 2.
Table 2: Prototype and model parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SI Units</th>
<th>Prototype (value)</th>
<th>Scale factor</th>
<th>Model (value)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Te</td>
<td>K</td>
<td>298,0</td>
<td>1,0</td>
<td>298,0</td>
</tr>
<tr>
<td>θ</td>
<td>kgm⁻³</td>
<td>1,15</td>
<td>1,0</td>
<td>1,15</td>
</tr>
<tr>
<td>cp</td>
<td>Jkg⁻¹K⁻¹</td>
<td>1006,0</td>
<td>1,0</td>
<td>1006,0</td>
</tr>
<tr>
<td>R</td>
<td>m²KW⁻¹</td>
<td>3,45</td>
<td>1,0</td>
<td>3,45</td>
</tr>
<tr>
<td>L</td>
<td>m</td>
<td>3,80</td>
<td>2,5</td>
<td>1,52</td>
</tr>
<tr>
<td>W</td>
<td>m</td>
<td>1,80</td>
<td>2,5</td>
<td>0,72</td>
</tr>
<tr>
<td>H</td>
<td>m</td>
<td>1,65</td>
<td>2,5</td>
<td>0,66</td>
</tr>
<tr>
<td>t</td>
<td>s</td>
<td>1,00</td>
<td>2,5</td>
<td>0,4</td>
</tr>
<tr>
<td>v</td>
<td>ms⁻¹</td>
<td>(Test)</td>
<td>1,0</td>
<td>(Test)</td>
</tr>
<tr>
<td>h</td>
<td>Wm⁻²K⁻¹</td>
<td>(Test)</td>
<td>1,0</td>
<td>(Test)</td>
</tr>
</tbody>
</table>

Forced Convection

| To        | K        | 278,0             | 1,0          | 278,0         |
| Wi        | m        | 0,60              | 2,5          | 0,24          |
| Hi        | m        | 0,15              | 2,5          | 0,06          |
| WR        | m        | 0,80              | 2,5          | 0,32          |
| HR        | m        | 0,30              | 2,5          | 0,12          |
| V         | m³s⁻¹    | 0,279             | 6,25         | 4,464x10⁻²    |
| Q         | W        | 309,0             | 6,25         | 49,4          |

Natural Convection

| Tp        | K        | 253,0             | 1,0          | 253,0         |
| Lp        | m        | 2,6               | 2,5          | 1,04          |
| Hp        | m        | 0,6               | 2,5          | 0,24          |
| Q         | W        | 190,0             | 6,25         | 30,4          |

For the forced convection model, an auxiliary experimental installation which applies a computer linked air conditioning laboratory unit schematically illustrated in Figure 3, will be used to reproduce the inlet air flow conditions in

Figure 2: Sketch of the auxiliary installation in the natural convection model.
the experimental room given in Table 2. A complete description of the experimental installation can be found in Pitarma [2]. The inlet air temperature, \( T_0 = 5.0^\circ C \), was controlled with a precision of 0.25\(^\circ\)C. The mean inlet velocity was \( V_0 = 3.1\) m/s, but velocity measurements were made to characterize the velocity distribution at the inlet section. The internal room load was produced, with an accuracy of 2\%, by electrically heated tapes laid over the floor area to produce a uniform load distribution.

![Sketch of the auxiliary installation to conditioning supply air in the forced convection model.](image)

**Figure 3:** Sketch of the auxiliary installation to conditioning supply air in the forced convection model.

### 2.3 Measurements

A vertical rake of 7 thermocouples (T type, 200\( \mu \)m wire diameter) was used for temperature measurements. Temperature signals were acquired by a Data Translation board (DT2811/DT756Y), connected to an HP Vectra microcomputer, with an accuracy of 0.25\(^\circ\)C (±5\%). The sampling rate, selected after few preliminary runs, was 1Hz (i.e. each one of the 7 channels was sampled every 7 seconds).

The air velocity magnitude \( V = (u^2 + v^2 + w^2)^{1/2} \) measurements were obtained by a hot-wire anemometer system (Dantec StreamLine) connected to an HP Vectra microcomputer, with an omnidirectional hot-film sensor type (Dantec 55R49). This hot-wire anemometer system uses miniature probes and therefore low disturbance to the flow occurs. The control of the whole system, including acquired data and its treatment, was made by a software support (Dantec StreamLine) in Windows environment. A detailed description of this system is given by Jorgensen [7]. The accuracy of measurements is within ±1\% (typically 0,5\%)+0,02m/s.

Complementary information of the airflow was obtained by flow visualization. Neutrally thermal smoke was injected to trace the air movement characteristics within the test room for natural convection configuration. Due to its mixing, flow visualization in the forced convection configuration was carried out through orientation acquired by nylon fibre placed inside the model, in two longitudinal planes (close to symmetry plane and close to lateral wall) and in a traverse plane near the downstream wall.
3 Mathematical modelling

The Reynolds-averaged form of the governing equations for a steady, incompressible and turbulent flow expressed in tensor notation read:

\[
\frac{\partial U_i}{\partial x_i} = 0
\]  

\[
\frac{\partial}{\partial x_j} \left(U_j \Theta \right) = - \frac{\partial}{\partial x_j} (\langle u_i \theta \rangle)
\]  

\[
\frac{\partial}{\partial x_j} (U_j U_i) = - \frac{1}{\rho} \frac{\partial P}{\partial x_i} - \frac{\partial}{\partial x_j} (\langle u_i u_j \rangle) + \beta g_i \Theta
\]

where \(U_i\) and \(u_i\) represent the \(x_i\) component of mean and fluctuating velocity, respectively; \(P\) is the pressure and \(\rho\) the fluid density; \(\Theta\) represents the mean temperature difference, \(T-T_0\), where \(T_0\) is the room average temperature. \(\theta\) is the temperature fluctuation; \(\beta\) is the coefficient of thermal expansion and \(g\) the gravitation constant. Finally, \(\langle \rangle\) stands for the temporal average of the quantity. A detailed description of the transport equations is given, for example, by Pitarma [2]. The turbulence model employed to calculate the turbulent fluxes is a two-equation turbulence model representing the kinetic energy, \(k\), and its rate of dissipation, \(\varepsilon\) (\(k-\varepsilon\) turbulence model). The effects of buoyancy are included both on the vertical component of velocity and on the turbulence model. Further details of the \(k-\varepsilon\) turbulence models are found in Launder and Spalding [8]. The boundary conditions at the walls for velocity components, \(k-\varepsilon\), and thermal energy are specified using wall functions. Details about the wall functions can be found in Launder and Spalding [8].

The numerical procedure is based on a finite-volume discretization of the governing equations, employing a staggered grid for mean-velocity components relatively to scalar properties. The hybrid central/upwind differencing scheme is used to approximate the convection terms. One method based in the Simple algorithm was chosen for the pressure-velocity coupling correction (see Patankar [9]). The solution of the individual equation sets was obtained by a form of Gauss-Seidel line-by-line iteration.

The simulation model was used to predict the velocity and temperature fields in the reduced-scale model, described above. The computations were performed using a 9x11x13 (natural convection case) and 11x9x19 (forced convection case) control volumes grids. In accordance with the symmetry of the experimental chamber and their operating conditions, only a quarter (natural convection) and half (forced convection) of the flowfield was covered by computational domain. Grid-dependence tests were made indicating that the differences between the results in this grid and a 22x18x38 control volumes grid are not significant. The sums of the absolute residuals of mean field variables were used for monitoring
convergence. The iterative process was terminated after the normalized residuals have fallen below 0.05%. The computations were performed in a HP Apollo 720 workstation, and the required time of CPU to achieve convergence was 2h37m (natural convection) and 39 minutes (forced convection).

Figure 4: Visualization of flow field (4.a) and predicted velocity vectors and temperature distribution (4.b) in the traverse symmetry plane of the natural convection configuration.

Figure 5: Predicted velocity vectors and temperature distribution in the symmetry plane of the forced convection configuration.

4 Results

The numerical predictions were validated against experimental data acquired in the reduced-scale model. Experiments include measurements of mean air velocity and mean air temperature distribution that allows testing the experimental and numerical models performance. Figure 4 shows an example of the visualization (4.a) and the calculated (4.b) flow field at the traverse symmetry plane, for the natural convection configuration. An example of the temperature
distribution is given in the same figure (4.b). Figure 5 gives the velocity vectors and temperature field for the symmetry plane in the forced convection configuration.

In Figure 6 the calculated results are compared with measured velocity profiles. For the forced convection configuration, inlet conditions \((V_0, T_0)\) were used to normalize the computational and measured results. Figure 7 shows a comparison between computed and measured velocity and temperature non-dimensional profiles. It can be seen from these figures that predicted values of air velocity and temperature are consistent with measurements and that the major approach was obtained in the forced convection study. In addition, flow visualization experiments confirm, for both configurations, that the predicted and experimentally observed airflow patterns are qualitatively consistent.

Figure 6: Comparison of calculated and measured velocity profiles (o experiment, + simulation) a) Natural convection; b) Forced convection.

Figure 7: Comparison of calculated and measured temperature profiles (o experiment, + simulation). a) Natural convection; b) Forced convection.
5 Conclusion

Computational and experimental reduced-scale models for the simulation of the three-dimensional non isothermal flows in refrigerated chambers are presented. The similarity between reduced-model and prototype room was achieved through a simple physical modelling technique, based on dimensional analysis. To validate the computer model, the calculated results are compared with those from the experimental tests. The paper shows that the results of the numerical calculation procedure are in close agreement with experimental data from model scale experiments. According to the results, three conclusions can be drawn: i) Firstly, the suggested modelling techniques offers a means of studying a wide range of refrigerated chambers problems, with low costs and simplicity; ii) Before the real construction, still in design phase, it is highly desirable that a preliminary assessment of indoor cold storage conditions based on these types of prediction methods be made. Here, numerical simulation is very promising; iii) Finally, due to its lower costs and acceptable versatility, the use of small-scale models offers an attractive and viable solution for the development and experimental validation of the codes.

References

[10] Ramos, J.E., Pitarma, R.A. & Carvalho, M.G. Thermal and Environmental Simulation of Buildings, XXX IAHS World Congress on Housing,
Housing Construction – An Interdisciplinary Task, University of Coimbra, Coimbra, September, 2002.
