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Numerical Modelling and Experimental Validation of a Cold Store Ambient Factors

Serap AKDEMİR^a, Thomas BARTZANAS^b

^aNamik Kemal University, Technical Sciences Vocational School, Air Conditioning and Refrigeration Programme, Tekirdağ, TURKEY

^bCentre for Research and Technology-HELLAS, Institute for Research and Technology-Thessaly, Volos, GREECE

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Corresponding Author: Serap AKDEMİR, E-mail: sakdemir@nku.edu.tr, Tel: +90 (537) 529 59 06

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ABSTRACT

The objective of this study was to analyse air temperature and relative humidity distribution in an experimental cold store fully loaded with apples by using both experimental and numerical (CFD) methods. An unsteady three-dimensional computational fluid dynamics model was developed to assess the distribution of temperature and relative humidity in a cold store fully loaded with Granny Smith apples. The storage temperature and relative humidity were maintained at 2 °C and 90%, respectively. The relative humidity and temperature were measured at 36 different points inside the cold store in three different planes. A three-dimensional mathematical model was built for the numerical needs. The numerical model was validated against experimental data from the same facility. Relative error of the model was calculated 13% for temperature and 1.43% for relative humidity. Numerical results obtained from the simulations agreed quite well with experimental data for temperature and relative humidity. Maximum differences were observed near the borders of the cold store which can be attributed to the stronger thermal gradients taking place on these surfaces. A more homogeneous distribution was achieved in the middle of the cold store both for air temperature and relative humidity leading to even smaller errors between measurements and simulations.

Keywords: Computational fluid dynamics (CFD); Cold store; Temperature; Relative humidity; Apple

Bir Soğuk Hava Deposunda Ortam Koşullarının Sayısal Modellenmesi ve Deneysel Geçerliliğinin Belirlenmesi

ESER BİLGİSİ

Araştırma Makalesi

Sorumlu Yazar: Serap AKDEMİR, E-posta: sakdemir@nku.edu.tr, Tel: +90 (537) 529 59 06

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ÖZET

Bu araştırmanın amacı; deneysel bir soğuk depo için sıcaklık ve bağıl nem dağılımını sayısal ve deneysel yöntemleri birlikte kullanarak incelemektir. Granny Smith elma ile dolu bir soğuk depoda hava hızı ve sıcaklığın dağılımını değerlendirmek için kararsız üç boyutlu hesaplamalı akışkanlar dinamiği modeli geliştirilmiştir. Depolama sıcaklığı ve

bağlı nem 2 °C ve % 90 olarak belirlenmiştir. Bağlı nem ve sıcaklık soğuk hava deposu içinde üç farklı düzlemde 36 farklı noktada ölçülmüştür. Sayısal gereksinimler için üç boyutlu matematiksel model oluşturulmuştur. Sayısal modelin geçerliliği aynı koşullarda uygulamadan elde edilen deneysel verilerle test edilmiştir. Modelin bağlı hatası sıcaklık için % 13 ve hava hızı için % 1.43 olarak hesaplanmıştır. Sıcaklık ve bağlı nem için modellemeyen elde edilen sayısal sonuçlar oldukça iyidir. Model ile ölçümler arasındaki en yüksek farklılıklar yüzeyler üzerinde ısı değişimlerinin en güçlü olduğu soğuk odanın sınırlarına yakın saptanmıştır. Soğuk deponun ortasında model ve ölçümler arasında küçük hatalar olmasına rağmen hem sıcaklık hem de bağlı nem için daha homojen bir dağılım elde edilmiştir.

Anahtar Kelimeler: Hesaplamalı akışkanlar dinamiği (HAD); Soğuk depo; Sıcaklık; Bağlı nem; Elma

1. Introduction

Many agricultural products are consumed a long time after their harvest or they have to be kept in storage for long periods of time. Fruits and vegetables remain physiologically active after harvesting. Respiration affects and constrains the life span of fruits and vegetables, as well as quality and nutrition changes during storage. Continuing respiration after harvesting of fruits and vegetables results in quality deterioration and nutrient reduction. Post-harvest storage facilities for agricultural fresh products are usually used in order to preserve the quality of agricultural products by reducing the respiration process and thus the production of carbon dioxide and heat. The indoor climate was automated and controlled in order to keep temperature, humidity and CO₂ concentration stable. However, it is difficult to achieve a homogenous distribution of climate parameters in any agricultural building due to geometrical features of the post-harvest facility, the cooling medium and the influence of products on the airflow distribution as well as the rest climate parameters. However it is important to access in as much detail is possible the distribution of air temperature and then to propose a design modification in the cold store for improving the quality and storage duration of agricultural products. The heterogeneous climate inside a cold store can be accessed experimentally. However the method is expensive, time consuming and usually very difficult, if not impossible, to receive values for the whole experimental domain. On the other hand, numerical modelling techniques such as computational fluid dynamics (CFD) can be used as an alternative technology and tool for

analysing design parameters and/or fruit process, environmental data and weather conditions within a virtual environment. Computational fluid dynamics (CFD) is a simulation tool which uses powerful computer and applied mathematics to model fluid flow situations for the prediction of heat, mass and momentum transfer and optimal design in industrial processes. It is only in recent years that CFD has been applied in the food processing industry (Xia & Sun 2002). The amount of experimental work has thus been greatly reduced, although, not eliminated since experimental data is needed for the validation process. CFD is used more and more in many engineering applications that aim to analyse heat and mass transfer and to propose design modifications (Schaldach et al 2000). CFD is also used extensively in agricultural operations in the last decade such as: drying and sterilization, Le Page et al (2009), crop sprayers Nuyttens et al (2009), greenhouse and livestock buildings Bartzanas et al (2013); Ambaw et al (2013); Bjerg et al (2013). CFD has been used to predict and analyse the airflow, air temperature and air gases distribution in post-harvest facilities (Foster et al 2006). Tassou & Xiang (1998) have analysed the buoyancy driver airflow in a post-harvest facility used for vegetables. The 3-D model of (Xu & Burfoot 1999) for the prediction of distribution of air velocity, air temperature and humidity was further improved and used by Moureh et al (2009). In order to predict the possibilities of condensation in stored potatoes, Chourasia & Goswami (2007a) used CFD not only to analyse the distribution of microclimate variables inside a cold store room but also to propose design modifications for improving the airflow distribution system. The same approach for improving the design of

air conditioning systems was adopted and used by Tanaka et al (2012) and Xie et al (2006). Their results clearly indicate the strong influence of design parameters on the distribution of air velocity and air temperature inside the cold store facilities. Nahor et al (2005) developed and experimentally validated a CFD model for predicting air and fruit temperature with reasonable accuracy. Mirade & Picgirard (2006) developed a CFD model which was also validated successfully against experimental data for improving the airflow distribution inside a batch-type meat carcass chiller. Martins et al (2011) used a CFD model with a true apple shape in order to study the cooling of apple fruits in overleaf arrangements on trays. The cooling performance of a partially loaded cold store was numerically investigated and experimentally validated by Tanaka et al (2012) with the aim of a transient 3D CFD model. A multi-scale approach for analysing air velocity distribution cold store facilities was used by Delele et al (2009). The CFD model was used at the smallest scale whereas, at a larger scale, the airflow was assessed through the estimation of pressure loss coefficients around the stacks. Air flow inside a cold store was investigated using computational fluid dynamics by Hoang et al (2000). The validation of the model was performed using a comparison of the calculated time-averaged velocity magnitudes with the mean velocities measured by means of a hot film type omnidirectional velocity sensor. A relative error of 26% was observed on the calculated air velocities (Hoang et al 2000). The SST $k-\omega$ based model was applied to calculate air-flow velocities and temperatures in a model office room. Calculations were compared with experimental results as well as with the results of the standard $k-\epsilon$, the RNG $k-\epsilon$ model and the laminar model. It was concluded that (a) all the three tested turbulent models satisfactorily predict the main qualitative features of the flow and the layered type of temperature fields and (b) computations with the SST $k-\omega$ based model show the best agreement with the measurements (Stamou & Katsiris 2006). Measurements of infiltration through different size entrances of a cold store at two different cold store temperatures were taken and compared against established analytical models and computational fluid dynamics (CFD) models

(Foster et al 2003). To investigate the effectiveness of postharvest storage fungicide fogging systems, a computational fluid dynamics (CFD) model was developed and validated (Delele et al 2012).

Variation of storage conditions greatly affects quality of stored products. Using sensors in cold storage is an easy and rapid method to attain valuable data and monitor the storage conditions on an ongoing basis. Fuzzy clustering was proved to be a sufficient tool to manage spatial variation of cold storage properties and to delineate zones of differential management (Akdemir & Tagarakis 2014). Distribution of the temperature and relative humidity in a cold store was important because these factors affect quality of stored products. Many studies on the effect of storage temperature on quality and storage life of fruits have been carried out which indicate that temperature plays an important role on quality of fruits after harvest (Marcilla et al 2006). The Objective of this study was to analyse air temperature and velocity distribution in an experimental cold store full loaded apples by using both experimental and numerical (CFD) methods. A second objective was to use the validated CFD model for proposing design modifications of the cold store for improving the homogeneity of environmental factors.

2. Material and Methods

2.1. Experimental cold store

An experimental cold store (4.0 m length, 5.0 m width and 3.0 m height) with chiller and air conditioning system was used in this study. The system consists of three main units; water cooling unit, air conditioning unit and control unit. Cold store mainly installed in a concrete building but two of the walls of the cold store were located in front of the outside walls of concrete buildings. These walls called as “outside walls”. One another called as “door-wall” and last was inside walls because of its position. The experimental cold storage general view and the air-conditioning units can be seen in Figure 1.

Apples (Granny Smith variety) were stored during the experiments. Air conditioning unit was



(a) general view of cold storage



(b) air conditioning

Figure 1- The experimental cold storage general view (a) and the air conditioning unit (b)

Şekil 1- Deneysel soğuk odanın genel görünümü (a) ve iklimlendirme ünitesi (b)

operating to maintain an air temperature of 2 °C and an air relative humidity of 90% during the storage. The unit has a tolerance of ± 0.5 °C for air temperature, 5% for relative humidity and the temperature control step was 0.1 °C. The technical specifications of the air-conditioning unit used in the experimental cold storage were given in Table 1.

Water was cooled by using refrigerant 404A in the chiller. Cold air was distributed by the inlet (pressure) channels and exhausted by the outlet (suction) air channels to be released outside. There were 3 inlets and 3 outlets channels for ventilation and suction of the cold air. The cooling system was facilitated with a vapour humidification capacity of 2 kg h⁻¹. A control software was responsible for the supervision and inspection of the cooling system performance. Inlet and outlet temperatures as well as relative humidity could be controlled and checked on a computer screen.

Table 1- Technical specification of air conditioning unit

Çizelge 1- İklimlendirme sisteminin teknik özellikleri

Parameter	Value
Ventilator type	Axial
Flow rate	5000 m ³ h ⁻¹
Cooling system capacity	10 kW
Heating capacity	5 kW
Inlet air temperature and relative humidity	19 °C 65% RH
Outlet air temperature and relative humidity	15 °C 85% RH
Water inlet and outlet temperature	6/11 °C
Heating system type	Electrical resistance
Air flow rate of ventilator	Invertor changes flow rate

2.2. Data loggers for temperature and relative humidity

Air temperature and relative humidity were measured using a Testo 177 H1 data logger. Number of Testo 177H1 data logger was 36. Measuring Range of the Testo 177 H1 data logger is 0..100 for relative humidity and -40 to +120 °C for temperature. Resolution is 0.1 for RH% and 0.1 °C for temperature. Accuracy is for 2% for relative humidity and 0.2% for temperature (Testo 2015). Testo Comfort Software and Testo IRsoft software were used to transfer the recorded data from the data loggers to the computer.

2.3. Numerical analysis

Computational Fluid Dynamics (CFD) software package ANSYS FLUENT v.14 was used for numerical modelling of air temperature and relative humidity in the cold store (Ansys 2014). The finite volume method was used to solve the partial differential equations for different quantities of interest. The code first performs the coupled resolution of the pressure and velocity fields after which it continues with the other parameters such as temperature. The domain of interest was generated and then meshed using the integrated pre-processor software for creating and meshing the geometry.

The CFD technique numerically solved the Navier-Stokes equations and the mass and energy conservation equations. The three dimensional conservation equations describing the transport phenomena for steady flows in free convection are of the general form:

$$\frac{\partial(U\Phi)}{\partial x} + \frac{\partial(V\Phi)}{\partial y} + \frac{\partial(W\Phi)}{\partial z} = \Gamma \nabla^2 \Phi + S_\Phi \quad (1)$$

In Equation (1), Φ stands for the transport quantity in a dimensionless form, while U, V and W are the components of velocity vector; Γ is the diffusion coefficient; and S_Φ is the source term.

The present flow and transport phenomena are described using the Navier-Stokes equations. The time-averaged Navier-Stokes equations for the continuity and momentum are given in Equation 2 and 3.

Continuity equation:

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (2)$$

Momentum conservation:

$$\rho U_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_t) \frac{\partial U_i}{\partial x_j} \right] + f_b + S_i \quad (3)$$

2.4. Turbulence modelling

Several previous experiments and simulations have shown that airflow distribution inside buildings is turbulent even for very small values of air velocity (Monteith & Unsworth 1990). We have to introduce turbulent models in our solution domain in order to assess the turbulent nature of air flow and to obtain a more realistic solution. The standard k-ε model (Launder & Spalding 1974) assuming isotropic turbulence was adopted to describe turbulent transport. The complete set of equations of the k-ε model as in Equation 4, 5, 6 and 7 can be found in Mohammadi & Pironneau (1994).

The effect of turbulence on flow was implemented via the high Re k-ε model (standard) model (Launder & Spalding 1974).

$$\rho \frac{\partial k}{\partial t} + \rho U_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} + \tau_{ij} \frac{\partial U_i}{\partial x_j} - \rho \epsilon \quad (4)$$

$$\rho \frac{\partial \epsilon}{\partial t} + \rho U_j \frac{\partial \epsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} - \rho C_{\epsilon 2} \frac{\epsilon^2}{k} + C_{\epsilon 1} \frac{\epsilon}{k} P_k \quad (5)$$

Where, the turbulent viscosity is:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (6)$$

and

$$P_k = \tau_{ij} \frac{\partial U_i}{\partial x_j} \quad (7)$$

and

$C_\mu = 0.09$, $\sigma_k = 1$, $C_{\epsilon 1} = 1.44$, $C_{\epsilon 2} = 1.96$, and $\sigma_\epsilon = 1.3$ are model constants.

Boundary and initial conditions, the geometry of the studied domain and the final grid are all important factors that affect the final result. Mean values from experiments were used for boundary and initial conditions (Table 2) and a three dimensional (3D) model for rendering the geometry of the cold store (Figure 4). The final grid, selected after grid independence test studies, was dense enough to secure accurate solutions especially in areas where strong gradients of airflow were expected. Number of the tetrahedral and hexahedral elements was approximately 3250000. The resulting solid model, mesh structure and nodes were presented in Figure 2.

Distributions of temperature and relative humidity in the cold store were modelled using Computational Fluid Dynamic (CFD). Analysis includes geometrical modelling, creating numerical mesh and evaluating of results. Boundary conditions have been given in Table 2.

Tetrahedral and hexahedral meshes were generated and then refined until a solution was obtained. Turbulence was modelled using k-ε. Heat transition through boundaries and the lighting system were taken into account for modelling. The base of cold store was accepted as an isolated surface. Constant heat transition was used as a boundary condition for the walls and ceiling. Storage temperature, air velocity and relative humidity were assumed as 2 °C, 2.99 m s⁻¹ and 90%, respectively.

X-Z, Y-Z and X-Y planes and levels were used to acquire data from CFD models which have been given in Figure 3.

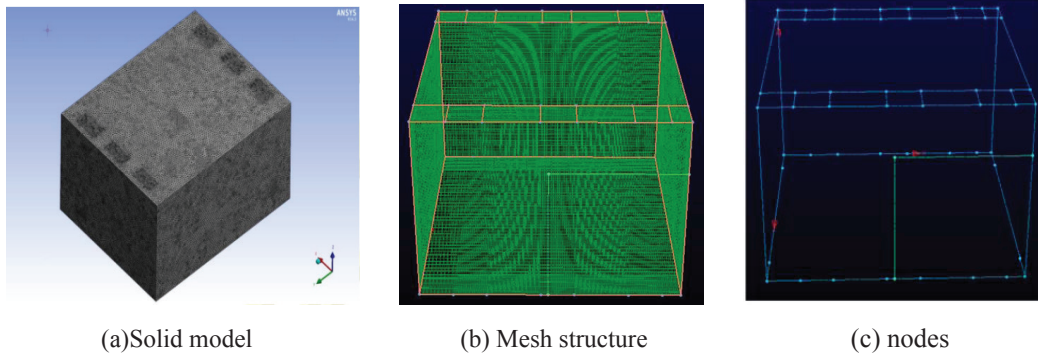


Figure 2- Solid model (a) mesh structure (b) and nodes (c) in the geometrical model

Şekil 2- Geometrik modelde katı model (a) mesh yapısı (b) ve düğüm noktaları (c)

Table 2- CFD Model boundary conditions

Çizelge 2- CFD Model sınır koşulları

Parameter	Value
Inlet	Surface of fluid inlet
Outlet	Surface of fluid outlet
Walls	Solid, proof against flow of fluid
Convective heat transfer coefficient	0.24 W m ⁻² K(for inside walls and ceiling)
Conductive heat transfer coefficient	0.025W m ⁻¹ K (for outside wall, sandwich panel-DIN 4108)
Outside temperature	18 °C
Inside temperature	2 °C
H ₂ O mass fraction	0.0041432 h20
Solution type	Pressure-Based , Double Precision
Fluid regime	Steady State
Turbulence model	k-epsilon
Pressure - velocity relationship	Simple
Discretisation method	Second Order Upwind
Convergence criteria	< 10 ⁻³ (Residuals)

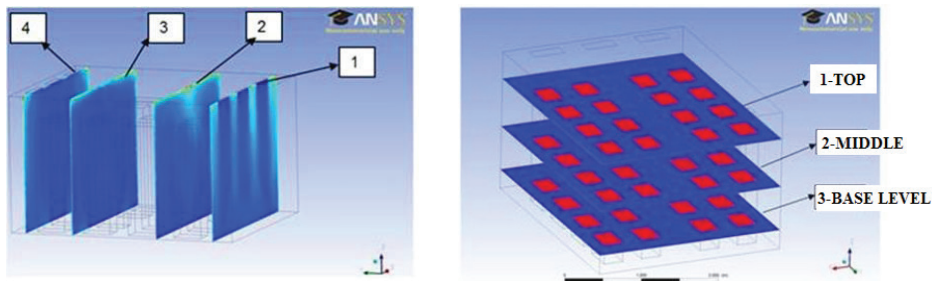


Figure 3- X-Z and X-Y planes

Şekil 3- X-Z ve X-Y düzlemleri

In addition, CFD models of temperature and relative humidity were developed for Y-Z planes, X-Z and X-Y planes by using Ansys Fluent Software.

2.5. Experimental measurements

The cold store was loaded with 160 cases (10 cases/column, 4x4 rows) of 0.6 m length, 0.4 m width and 0.2 m height. The arrangement of apple cases has been shown in Figure 4.

Air temperature and relative humidity were measured at a total of 36 different points inside the cold store for 3 different levels (12 points for each

level). Measurement points (as star), axis and layout of the data loggers can be seen in Figure 5.

Temperature and relative humidity data loggers were located at measurement points. Model data were determined from CFD models by using Ansys Fluent Software.

CFD Model validated with measurements by using Equation 8 (Hoang et al 2000; Nahor et al 2005; Chourasia & Gosvami 2007a).

$$E_{CFD} = \frac{1}{S} \sum_{s=1}^S \frac{|RH_{CFD}^s \cdot t_{CFD}^s - RH_{measurements}^s \cdot t_{measurements}^s|}{RH_{measurements}^s \cdot t_{measurements}^s} \cdot 100 \quad (8)$$

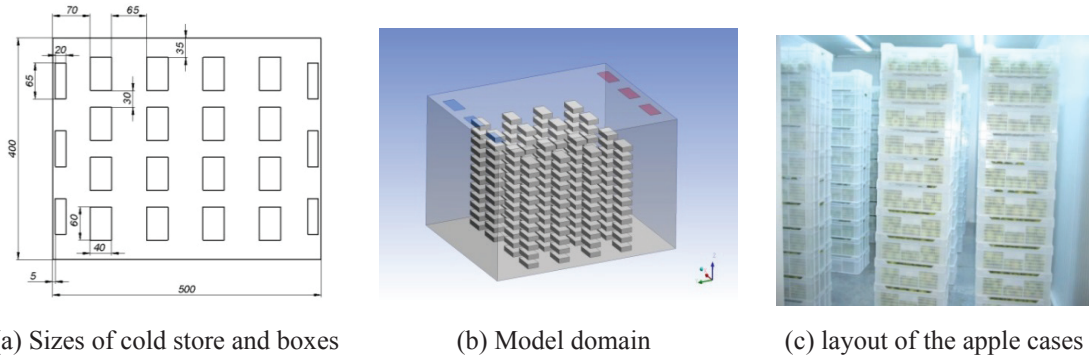


Figure 4- Sizes of cold store and boxes (a) model domain (b) layout of the apple cases (c)

Şekil 4- Soğuk oda ve kasaların ölçüleri (a) model etki alanı (b) elma kasalarının yerleşimi (c)

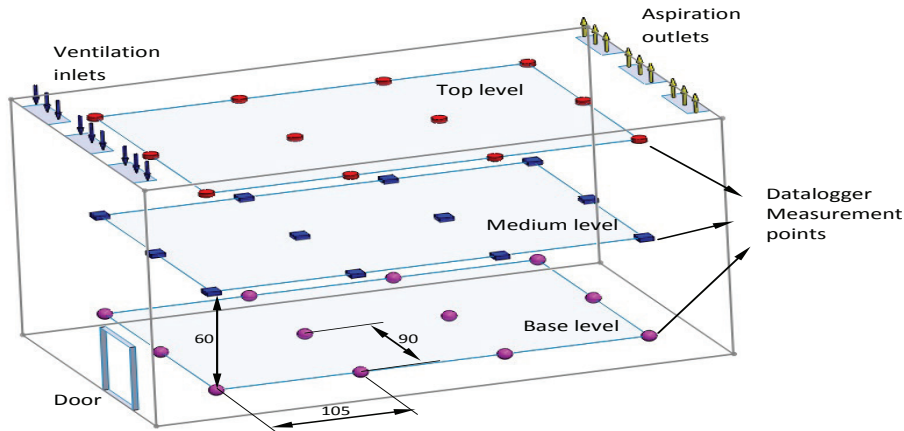


Figure 5- Measurements points, axis and layout of the data loggers in the cold store

Şekil 5- Soğuk odada ölçme noktaları, eksenler ve data loggerların yerleşimi

Where; \overline{E}_{CFD} , relative error (%); s, total number of the measurements; r, indices 0, 1, 2, ..., s; RH_{CFD}^r, t_{CFD}^r , CFD model data for relative humidity and temperature; $RH_{measurements}^r, t_{measurements}^r$, measured data for relative humidity and temperature.

Mean values of ambient factors do not give a correct idea for the evaluation of the spatial distribution of relative humidity and temperature in a cold store. The variation should be evaluated level by level using CFD models and spatial distribution graphics of ambient factors. In addition, variance analysis also gives an idea about the comparison of the CFD model with data logger measurements. In this sense; differences between model and measurements were calculated and evaluated via variance analysis for statistical evaluation. Significant sources such as CFD/Measurements, Y axis, levels or interactions between these factors were grouped by Duncan test. Logarithmic transformation was applied to relative humidity for variance analysis (Soysal 2007). SPSS

and MSTAT statistical software were used to make variance analysis and Duncan grouping tests.

3. Results and Discussion

3.1. CFD model results

3.1.1. Temperature

Air temperature distribution in the Y-Z planes has been given in Figure 6. Air temperature strongly decreased in this specific area due the strong air steam from inlet, increasing in the rest volume of the cold store according to the distribution of airflow. The air temperature varies from 2.40 °C to 2.7 °C near the roof of the cold store and this value was further decreased to the range of 2.10-2.40 °C near the middle height of the cold store.

Air temperature distribution in X-Z planes has been given in Figure 7. The temperature distribution

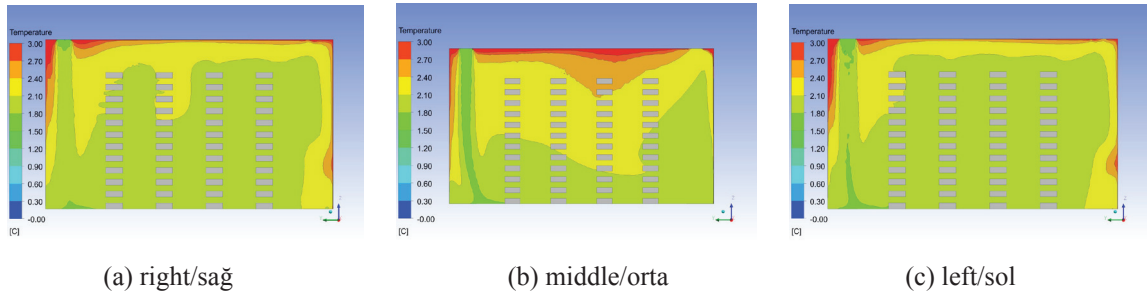


Figure 6- Temperature distributions in Y-Z plane of the cold store

Şekil 6- Soğuk oda Y-Z düzlemlerinde sıcaklık dağılımı

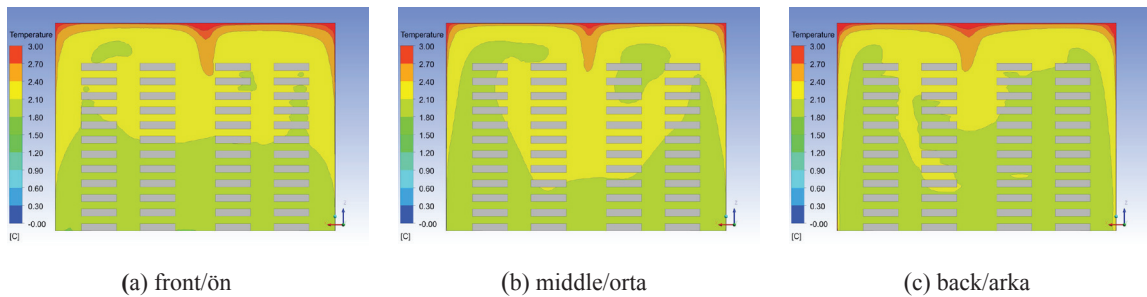


Figure 7- Air temperature distribution in X-Z planes of the cold store

Şekil 7- Soğuk oda X-Z düzlemlerinde sıcaklık dağılımı

was more homogeneous along the length of the cold store. The cold store air temperature varied from 1.50 °C to 1.80 °C near the walls and the ground (Figure 7). Air temperature increased and its value ranged from 2.40 °C to 2.70 °C near the roof of cold store due to reduced air velocities.

Temperature distribution in X-Y planes along the length of the cold store has been given in Figure 8. Air temperature distribution along the cold store height varies from 1.5 °C (a few centimetres above cold store ground and close to the airflow outlets) to 3.0 °C at regions on the ground of the cold store and in the corners usually at regions unaffected from air velocities (Figure 8).

3.1.2. Relative humidity

Relative humidity distribution in the Y-Z planes has been given in Figure 9. It can be seen from Figure 9 that the ventilated cold air entering the cold store from the inlet was directly guided to the ground

after which it was distributed into the rest of the cold store volume following the cold store walls. While inlet relative humidity was approximately 88%, outlet relative humidity was decreased to 85%. Relative humidity was changed between 84% and 87% in the middle areas of the cold store.

Relative humidity distribution in X-Z planes along the length of the cold store has been given in Figure 10. This area has no direct contact with inlet and outlet. So the values of relative humidity were further decreased in middle and top areas of the cold store. Relative humidity varied from 82% to 87%. Relative humidity at base level of this plane was 87%.

Relative humidity distribution in X-Y planes has been given in Figure 11. The relative humidity was changed from 88% to 85% in this plane. Even there wasn't big variability for relative humidity in X-Y plane, amount of the relative humidity was decreased from bottom to top.

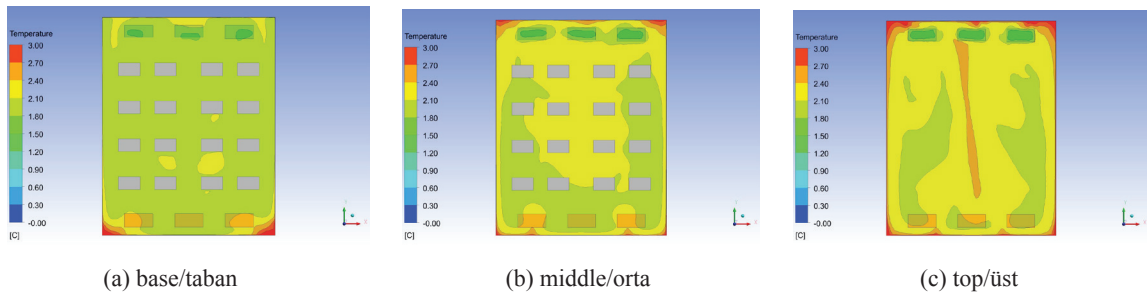


Figure 8- Air temperature distribution in X-Y planes of the cold store

Şekil 8- Soğuk oda X-Y düzlemlerinde sıcaklık dağılımı

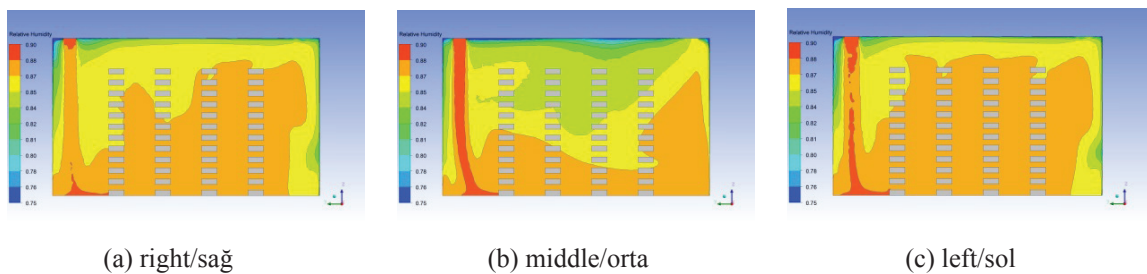


Figure 9- Relative humidity distribution in Y-Z plane of the cold store

Şekil 9- Soğuk oda Y-Z düzlemlerinde bağıl nem dağılımı

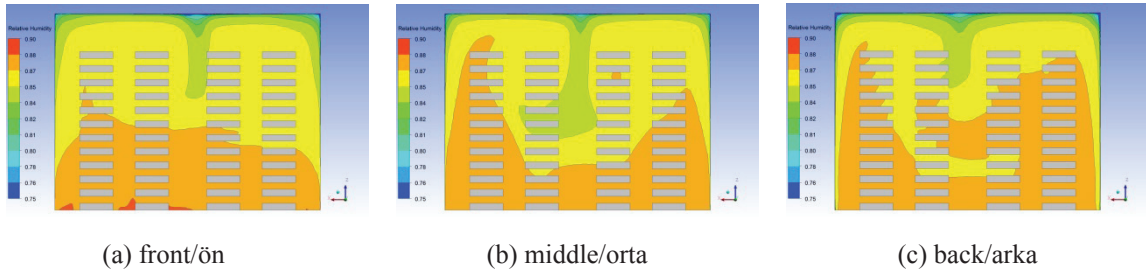


Figure 10- Relative humidity distribution in X-Z planes of the cold store

Şekil 10- Soğuk oda X-Z düzlemlerinde bağıl nem dağılımı

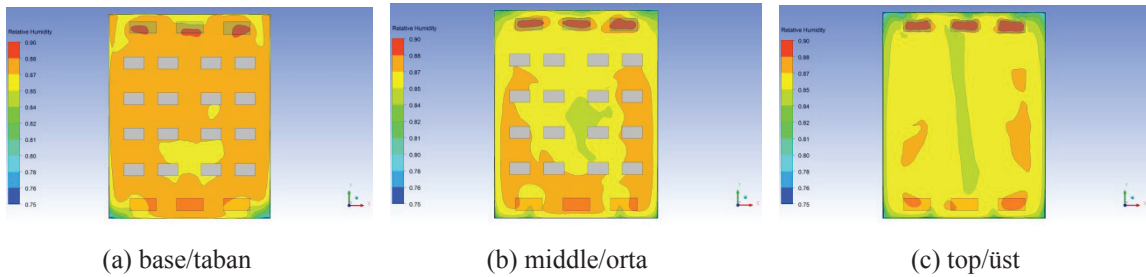


Figure 11- Relative humidity distribution in X-Y planes of the cold store

Şekil 11- Soğuk oda X-Y düzlemlerinde bağıl nem dağılımı

3.2. Numerical model validation

3.2.1. Air temperature

CFD model (t_{CFD}) and data logger measurements ($t_{measurements}$) and differences between model and measurements (Δt) of temperature in the cold storage during full loaded apple storage have been given in Table 3.

The difference between CFD model and data logger measurement was 0.33 °C. CFD model estimations were generally lower than data logger measurements. Standard deviation was calculated as 0.02 °C for measurement and 0.04 °C for CFD Model. Relative error of CFD model was calculated as 13.01%. This value was higher than that mean average absolute deviation of temperature value of 4.42% (Gilani et al 2013). Even differences between CFD model data and data logger measurements for temperature was small, uneven distribution of the temperature was a good agreement with results of Moureh & Flick (2004). The uneven distributions of

Table 3. CFD model (t_{CFD}) and measurements (t_m) and differences (Δt)

Çizelge 3- CFD model (t_{CFD}) ve data logger ölçümleri (t_m) ve farklar (Δt)

X-Z Planes	Level	t_{CFD} (°C)	t_m (°C)	Δt (°C)	E_{CFD}
1	1	2.14	2.53	0.39	15.42
1	2	2.14	2.54	0.40	15.75
1	3	2.09	2.50	0.41	16.40
2	1	2.22	2.53	0.31	12.25
2	2	2.20	2.50	0.30	12.00
2	3	2.15	2.48	0.33	13.31
3	1	2.20	2.53	0.33	13.04
3	2	2.18	2.50	0.33	12.80
3	3	2.18	2.51	0.32	13.15
4	1	2.25	2.53	0.28	11.07
4	2	2.24	2.48	0.24	9.68
4	3	2.21	2.49	0.33	11.24
Mean		2.18	2.51	0.33	13.01
Minimum		2.09	2.48	0.24	
Maximum		2.25	2.54	0.41	
Standard deviation		0.04	0.02	0.05	

temperature were also compatible with the results of Akdemir & Tagarakis (2014) who determined 4 management zones for this cold store. The developed CFD model can be used to estimate temperature distribution in this kind of cold store.

Differences between CFD model and data logger measurement ($F= 1622.25, \alpha= 0.001$), X-Z Planes ($F= 7.244, \alpha= 0.002$), Levels ($F= 4.866, \alpha=0.005$), CFD/Measurements x Levels interactions ($F= 11.482, \alpha= 0.001$) were found to be statistically significant according to the variance analysis of the air temperature in the cold store. Means of main factors and interaction effects statistically found to be significant were grouped via Duncan tests and the results have been given in Table 4.

It can be seen from Table 4 that the temperature values (mean 2.124 °C) was lower in the first X-Z plane which was close to the cold air input (ventilation) channels in comparison with the other X-Z planes. There were no statistically significant differences between X-Z planes. Model estimation was good for temperature.

There were also spatial variations for temperature of different measurement planes in the experimental cold store. The spatial variations of temperature were also determined by Tassou & Xiang (1998)

for wet air-cooled store after 5, 10 and 15 hours, Akdemir & Arin (2006) for evaporative cold store by using a mapping software with measured data, storage temperature varied between 0-5 °C for Tassou & Xiang (1998), 2.7-5.3 °C for Akdemir & Arin (2006). Average temperature differences during cooling period was found as 1.2 °C by Chourasia & Goswami (2007b) used CFD in their research.

3.2.2. Relative humidity

CFD model (RH_{CFD}) and data logger measurements ($RH_{measurements}$) and differences between model and measurements (ΔRH) of relative humidity in the cold storage during fully loaded apple have been given in Table 5.

The difference between CFD model and data logger measurement was 1.20%. It was very good for the model estimation. Standard deviation was calculated as 0.38% for measurement and 0.78% for CFD Model. Relative error of CFD model was calculated as 1.43%. The CFD model can be used to estimate relative humidity distribution. There was a good agreement between CFD model estimations and data logger measurements for distribution of relative humidity (%) in the cold store (Chourasia & Goswami 2007a). The relative error (E_{CFD}) of relative humidity was better than 15.9% as overall

Table 4- Means and Duncan test group code of main factors and interaction effects on temperature at different X-Z Planes and Levels

Çizelge 4- Farklı X-Z düzlemlerinde ve seviyelerde ana faktör ve interaksiyon etkilerinin sıcaklık ortalamaları ve Duncan test grupları

		<i>X-Z planes</i>						
		<i>Levels</i>	<i>1</i>	<i>2</i>	<i>3</i>	<i>4</i>	<i>Mean</i>	<i>Mean</i>
CFD	Top (1)		2.141	2.221	2.199	2.248	2.202 C	
	Middle (2)		2.140	2.202	2.175	2.239	2.189 CD	2.18 A
	Base (3)		2.092	2.154	2.183	2.214	2.161 D	
	Mean		2.124	2.192	2.186	2.234		
Measurements	Top (1)		2.530	2.530	2.530	2.530	2.530 A	
	Middle (2)		2.537	2.497	2.500	2.480	2.503 B	2.51 B
	Base (3)		2.503	2.480	2.507	2.493	2.496 B	
	Mean		2.523	2.502	2.512	2.501		

$Sx_{(CFD/measurements) \times Levels} : 0.0091, \alpha=0.05$

Table 5- CFD model (RH_{CFD}) and data logger measurements ($RH_{measurements}$) and differences (ΔRH)

 Çizelge 5- CFD model (RH_{CFD}) ve datalogger ölçümleri ($RH_{measurements}$) ve farklar (ΔRH)

<i>X-Z Planes</i>	<i>Level</i>	$RH_{CFD}(\%)$	$RH_{measurements}(\%)$	$\Delta RH(\%)$	E_{CFD}
1	1	82.00	83.42	1.42	1.71
1	2	83.00	84.59	1.59	1.88
1	3	83.00	83.67	0.67	0.80
2	1	82.00	83.69	1.69	2.02
2	2	83.00	83.61	0.61	0.73
2	3	82.00	83.75	1.75	2.09
3	1	82.00	83.31	1.31	1.57
3	2	83.00	83.61	0.61	0.73
3	3	82.00	83.96	1.96	2.33
4	1	82.00	83.58	1.58	1.89
4	2	84.00	83.01	0.99	1.20
4	3	84.00	83.83	0.17	0.21
Mean		82.67	83.67	1.20	1.43
Minimum		82.00	83.01		
Maximum		84.00	84.59		
Standard deviation		0.78	0.38		

Table 6. Means and Duncan test group code of main factors and interaction effects on Relative Humidity at different X-Z Planes and Levels

Çizelge 6- Farklı X-Z düzlemlerinde ve seviyelerde ana faktör ve interaksyon etkilerinin bağlı nem ortalamaları

		<i>X-Z Planes</i>						
		<i>Levels</i>	<i>1</i>	<i>2</i>	<i>3</i>	<i>4</i>	<i>Mean</i>	<i>Mean</i>
CFD	Top (1)		82.000	82.000	82.333	82.333	82.167	
	Middle (2)		83.000	83.000	82.667	84.000	83.167	82.639 B
	Base (3)		83.000	81.667	82.000	83.667	82.583	
	Mean		82.667	82.222	82.233	83.333	82.639	
Measurements	Top (1)		83.423	83.687	83.310	83.583	83.501	
	Middle (2)		84.590	83.607	83.610	83.007	83.703	83.669 A
	Base (3)		83.673	83.750	83.957	83.827	83.802	
	Mean		83.896	83.681	83.626	83.472	83.669	

average relative error of the previously reported CFD models of Delele et al (2009) and 11.5% of Chourasia & Goswami (2007b).

Differences between CFD model and data logger measurement ($F= 13.193$, $\alpha= 0.001$) was found to be statistically significant according to the variance analysis of the relative humidity in the cold store. Means of main factors and interaction effects

statistically did not found to be significant were in Table 6.

It can be seen from Table 6 that there was only small differences between CFD model estimation (mean 82.639%) and data logger measurements (mean 83.669%) as 1.03%. Relative humidity of the base level a little beat higher than other levels for model and measurements.

There is always variation for temperature and relative humidity in cold stores. There is not any cooling system to establish completely homogeny distribution inside the cold stores. This variation also can be affected by replacement of storage boxes. It is important that manage of the spatial variation of ambient factors to decrease effect on quality of stored products.

4. Conclusions

The objective was to develop a CFD model to estimate temperature and relative humidity in a cold store which includes a chiller cooling system. A numerical model was used to access air temperature and relative humidity distribution inside a cold store filled with boxes of apples. Ansys Fluent software was used to develop the model. Before its use the numerical model was validated against experimental data.

Analysis of the simulation with the CFD and experimental data reveals a maximum difference indicating a good agreement between measurements and simulations for temperature and relative humidity. Relative error of CFD model was 13% for temperature and 1.43% for relative humidity. The image as seen in the distribution of temperature inside the cold store was not exactly a homogeneous distribution. Heterogeneity was due to the design of the cold storage system and the arrangements of apple boxes. Ambient temperature was generally increased from bottom to top contrary relative humidity decreased from bottom to top. There is a good agreement results of distribution of relative humidity and temperature in the cold store with Chourasia & Goswami (2007a). Blown air was distributed directly by cold air inlet channels into the room, slamming the air into the ground. Part of the air hitting the ground while flowing parallel to the ground portion enters the circulation in the room. According to research results; ambient temperature and relative humidity in the cold store with chiller unit can be predicted effectively by CFD model.

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