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Analysis and Comparison of Different Airflow Pattern and Introducing Air Outlet Duct in a Cool Storage By Means Of Computational Fluid Dynamics

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Abstract

Freshness and quality of fruits and vegetables as well as products saleable weight depend on heat and moisture transfer rate during air-cooling in the cool storages. Airflow inside a cool storage is investigated using computational fluid dynamics. A transient three-dimensional computational fluid dynamics (CFD) model was developed to predict airflow and heat transfer in a typical full loaded cool storage. The model was validated against experiments by means of velocity and product temperature in cool storage. In this study, comparison of two different air flow pattern is done and air outlet duct is introduced in a cool storage in order to reduce computational cost. The airflow model is based on the transient state incompressible, Reynolds – averaged Navier-Stokes equations (RANS). ANSYS FLUENT 12.0 software and turbulence modelling: the standard k - ε model with a second-moment closure is used for this thermal study. The forced-circulation air cooler unit is modelled with an appropriate body force and resistance, corresponding to the characteristics of the fan and the heat exchanger. The finite volume method of discretisation is used. The three dimensional model was capable of predicting product temperature, air velocity at different air pattern in a cool storage can be calculated with reasonable accuracy and was reliable enough for numerical studies on larger domain with high reduction in computational costs. On the basis of net average temperature, power consumption at different air pattern can be compared.

1 Introduction

The horticultural industries make significant contributions to the Indian economy. In the year of 2001-02 horticultural export earnings was about Rs 5677.50 (Rs in crore) and in the year 2010-11 i.e. after ten years the export earnings of the country was about Rs 13792.20 (Rs in crore), 142.9% was increased (Ministry of Agriculture and Forestry, 2011). To maintain and increase their share of the highly competitive global market, the India horticultural industries have to produce and market products with the highest possible quality.

Effective temperature management is essential to maintain product quality. The temperature of horticultural produce at harvest is close to that of ambient air. Rapid reduction of produce temperature to the optimum storage condition results in the desired produce quality and prolonged storage life. Rapid cooling after harvest is generally referred to as precooling (Wills et al., 1998). Forced-air cooling (pressure cooling) is often adopted for precooling of horticultural produce. Forced-air cooling involves creating a pressure gradient to force cold air through container vents.

The rate of cooling is significantly increased as the surface area available for heat transfer is enlarged by forcing air through packages and thus around each item of produce rather than only over the package surfaces. The time needed for forced-air cooling is only 10-25 percent of that for room cooling (MitcheIl 1992; Watkins, 1989). During forced-air cooling, it is crucial to achieve fast and uniform cooling throughout stacks of bins or pallets. The cooling rates of products inside the packages depend mainly on heat transfer between cooling medium (air) and products in the packages. The heat transfer processes are closely related to airflow transport within the packaging systems (Qian Zou 2002).

Apart from its influence on the heat transfer processes, air distribution plays a crucial role in produce weight loss due to water vapour mass transfer between product and air. Other important quality-related factors, such as the concentrations of gases (oxygen, nitrogen, carbon dioxide, ethylene, etc.) in the storage atmosphere and inside the produce, are also

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2 affected by air movement.

The heat and airflow transport processes are affected by the following factors:

• Materials and configurations of the produce packaging systems (trays, cartons, bins, palletisation patterns, stacking patterns, etc.)

• Characteristics of produce (dimensions, thermal properties, etc.)

• Cooling conditions (inlet airflow rates, inlet air temperature, inlet air moisture content, etc.)

Forced-air cooling requires the produce packages to be ventilated. Effective container venting is essential for efficient forced-air cooling. Cold air must be able to pass through all parts of the package so that all items of produce can be cooled evenly. As packages are usually stacked or palletised during forced-air cooling, the stack patterns should also allow a substantial amount of the airflow to be uniformly distributed over the whole packaging system.

It is usually considered expensive, time-consuming and situation-specific to use only experimental methods for studying heat and airflow transfer processes. Furthermore it may be difficult to achieve a complete understanding of the phenomena by examining a large amount of experimental data. Alternatively, mathematical modelling is, overall, a cost-effective strategy for predicting the airflow patterns and temperature variation in controlled environments such as ventilated packages. If information on packaging systems, cooling conditions, and produce properties are used as model input data, the results obtained can predict the effects of these factors on the airflow patterns and the product cooling rate. In this case, experiments may still have to be conducted for model verification. Computational Fluid Dynamics (CFD) provides a sophisticated but economic tool for modelling airflow and heat transfer. CFD employs numerical methods to solve the fundamental fluid transport equations that are derived from the laws of conservation of mass, momentum and energy. The increasing capacity and decreasing cost of modem computers have made the application of CFD modelling more and more efficient and popular.

The aim of this project is to develop a CFD modelling system for simulating airflow and heat transfer processes to compare two different airflow pattern and introduction of air duct at the outlet of the room. It is seen that forced convection air cooling is done in all the cool

2 Literature Survey

2.1 Introduction

While many CFD studies have been conducted to simulate airflow in cool storage but there were no studies found that compared the different airflow i.e. changing the direction of airflow by fan inside the cool storage. Also no standards for the design were found and no studies related to outlet duct in the cool storage were found. However, the following studies were useful in setting up proper CFD models for the purposes of this research.

This review focuses on the principles of Computational Fluid Dynamics (CFD), and the mathematical models for airflow and heat transfer processes in air-based cooling of fresh produce. Four classes of relevant literature were reviewed: (a) general modelling methodology; (b) basic principles of CFD; (c) modelling of airflow; (d) modelling of heat transfer during produce cooling.

2.2 Modelling Methodology

Harper & Wanninger (1969) defined mathematical models as the equations that simulate real world situations, because they behave in a manner analogous to the actual situations. Application of mathematical models reduces the scope and cost of experimentation, as modelling allows more alternatives to be considered which may be difficult or expensive

to test (Levine, 1997).

2.2.1 Modelling Procedure

Meerschaert (1993) identified five main stages in a modelling process: asking the questions, selecting a modelling approach, formulating the model, solving the model, and answering the questions. The starting steps are to examine the real world system to be modelled, and to identify the problems to be solved. These would enable the modeller to decide on the objectives of modelling, the required accuracy, and the type and size of computer envisaged (Touber, 1984). Cleland (1990) introduced a general system for equation development ***Literature survey***

suitable for modelling in the area of refrigeration.

2.2.2 Types of Models

• Steady-state and unsteady-state models

According to whether the modelled systems change with time or not, models can be classified as steady state and dynamic (unsteady-state).

Steady-state models are suitable for modelling systems whose major parameters do not change with time. Steady-state models are also used to assess the performance of a system under different sets of operating conditions (Touber 1984). Steady-state modelling may be applied to describe time-averaged behaviour of a transient system, while heat and mass accumulation in the system is negligible. Steady-state models generally demand less computational time and a small amount of input data, as the timevariability in system parameters is not considered.

Dynamic models are applied to assess how the time-variable effects, such as heat load, environmental conditions and start-up transients, influence normal system operation; accordingly advanced control strategies or detailed controllers may be developed (Cleland & Cleland, 1989). Dynamic models are usually considered closer to real world situations because most of them are time-dependent. However, as one more dimension (time) has to be dealt with mathematically, the dynamic modelling approach requires more input data (initial conditions) and more computational capacity while solving the models. Based on the approaches for modelling positional variation of variables (space discretisation), models are divided into zoned and fully distributed models.

In zoned models, the space to be modelled is divided into several zones. For each zone an ordinary differential equation is adequate for each variable as the conditions within the zone are assumed to be uniform. Movement of fluid within a region may be defined by a ***Literature survey***

plug-flow pathway, and the position of zones is arranged along the flow pathway through the system in a sequential fashion (Amos, 1995). Generally, only temperature and fluid concentration are solved in zoned models, while fluid velocity is defined with experimental data instead of by solving the momentum conservation equations. A single zoned model is the simplest case in which the whole calculation region is treated as uniform.

Fully distributed models are also called fluid dynamic models, and use partial differential equations (PDEs) to formulate the full position-variability. These PDEs describe heat, mass, and momentum conservation within the considered region. This approach is usually able to simulate the real world more accurately than the zoned models, since fewer assumptions are needed to develop a fully distributed model. Finite-difference, finite-element, and finite-volume methods are the most commonly used numerical tools for solving the transport equations. Fully distributed modelling requires much larger computer memory and computational time than zoned modelling. When applying the finite difference method, the solution domain is mapped with a grid. At each grid point, the partial derivatives in the PDEs are replaced by algebraic approximations in terms of nodal values of the solution functions. The result is one algebraic equation per grid node for each variable, in which the variable values at the node and a number of its neighbour nodes are unknowns in the equation. The approximations to the first and second derivatives of the variables are usually obtained by Taylor series expansion or polynomial fitting (Anderson et al., 1984). ***Literature survey***

DEVELOPMENT OF GEOMETRY AND MESH

4.1 Cool Storage Description

A cool storage was modelled based on the experimental conditions of the pilot cool storage of a postharvest laboratory in Department of Food Engineering and Post Harvest Technology (Karaj, Iran) with the dimensions of 4 m length, 2.6 m width, and 1.9 m height (Seyed Majid Sajadiye 2012) .The cooling unit was located at the top centre of the storage and consisted of two axial fans of 0.25 m diameter for air circulation and a finned tube heat exchanger (Figure 1.1). Apples (Malus domestica Borkh cv. Golden Delicious) were packed in the vented containers of 30 kg weight. The wall slots were spread over the five faces of each container to allow air to go through about 10% of the surfaces. The pallets or container's wall were made of high-density polyethylene (HDPE) with 0.004 m thickness and modelled as conducting walls. Containers of apples were modelled as a porous media. The enclosure was loaded with 96 containers with in-line array with the dimensions of 0.54 m length, 0.4 m width, and 0.3 m height. Four containers stacked with a small vertical gap of 0.06 m and there were four stacks along Z-direction with 0.06 m horizontal gap. Six columns of stacks positioned alongside X-direction with 0.1 m horizontal gap. All the gaps between the container and room's side walls were 0.1 m. The distance between pallets and roof, bottom, back, and front walls of room considered as 0.46, 0.1, 0.85, and 0.25 m, respectively. These distances were considered according to guidelines of settings in a refrigerated warehouse (IIR, 1966; Woolrich and Hallowell, 1970) (Seyed Majid Sajadiye 2012).

4.1.1 Three-dimensional Model Description

A three-dimensional model of a room in the shape of a rectangular prism was developed. The physical dimensions were set to be 4 m length, 2.6 m width, and 1.9 m height. The model geometry was created using pre-processor ANSYS DESIGN MODELER. On the basis of this data three geometry and mesh as per project work were made and discuss below. ***Development of geometry and mesh***

- **In first case**, that was in normal airflow pattern - The heat exchanger and fan, for both inlets and outlets, was located in the corner of the roof of the model room was at height of 1.6 m from bottom wall or floor and 0.1 m from back wall. The inlet were given to the front of the fan and the outlet were given to the bottom of the heat exchanger as shown in figure 4.1. Air outlet duct Door

4.1.2 Meshing

After creating the geometry, meshing was done in which a uniform tetrahedral mesh size of 0.04 m for the entire domain was selected. The mesh can be refined near the walls to satisfy the standard wall function conditions to bound y between 30 and 300 but because of the complexity it is avoided in this work and a uniform mesh is formed over the entire model. The total numbers of nodes was about 244748 and 1216611 tetrahedral cells (Figure 4.4 and 4.5). ***Development of geometry and mesh***

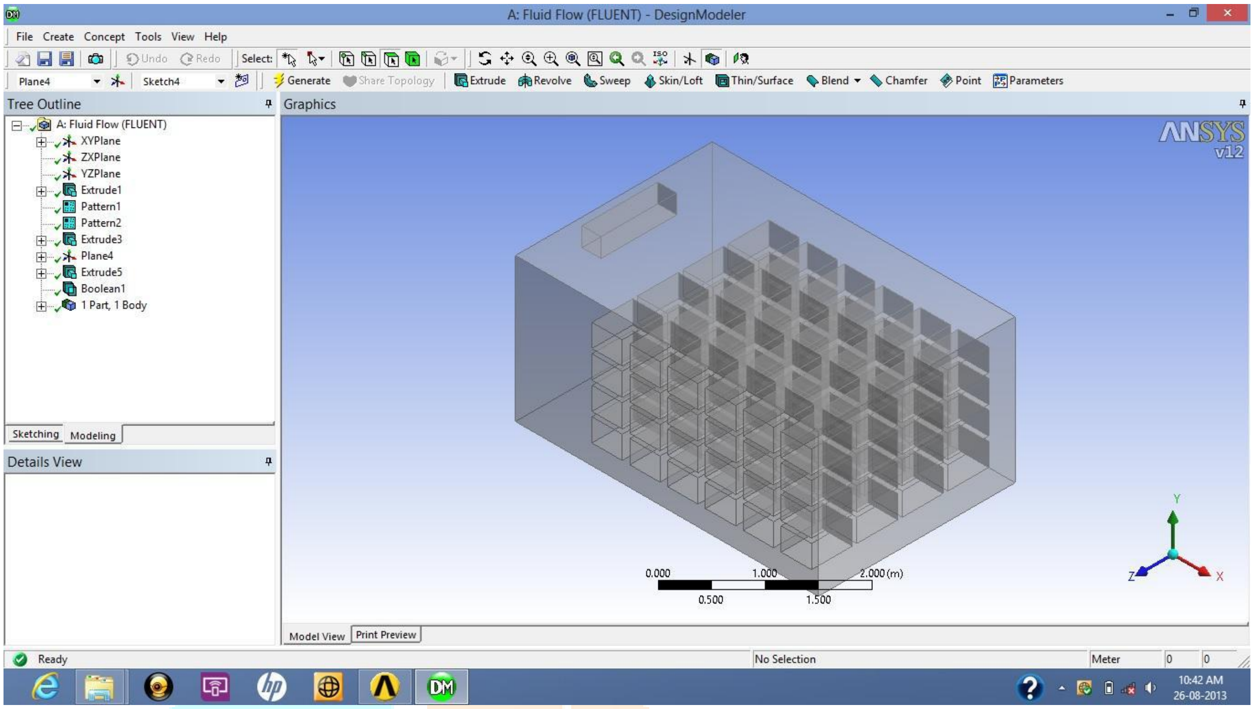

Figure 4.1, model diagram of normal or general airflow pattern

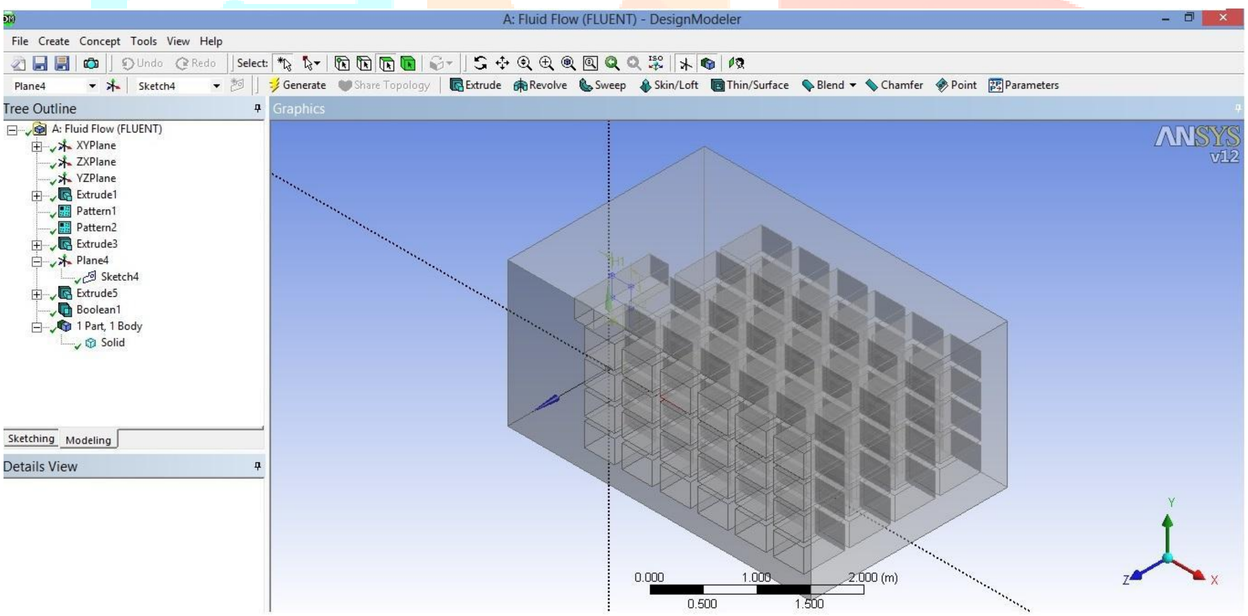

Figure 4.2, model diagram of change airflow pattern

DEVELOPMENT OF AIRFLOW AND HEAT TRANSFER MODEL

5.1 Introduction

This chapter presents the conceptual models for airflow and heat transfer in the cool storage. These conceptual models specify the important airflow and heat transfer processes and make assumptions to simplify the model formulation and solution. Following conceptualisation, the airflow and heat transfer models were developed and presented.

5.2 Conceptual Models

5.2.1 General Analysis of Airflow and heat transfer Processes

Airflow patterns, i.e. the distributions of air velocity and pressure over the domain of interest, are obtained by solving a set of partial differential equations that describe air mass and conservation over the domain. To select and solve air mass and momentum conservation equations over the domain, it is essential to carry out a detailed analysis of the transport processes and associated domain.

Temperature profiles, i.e. the air, produce, and package materials temperatures over the domain of interest are obtained by solving the partial differential equations that describe energy conservation over the domain. To select and solve the energy conservation equations over the domains of cool storage, it is essential to carry out a detailed analysis of the heat transfer processes and associated domain.

- **Conceptualisation conducted for the airflow models**

The airflow patterns predicted by the airflow models are to be used as input data for heat transfer calculations and volume-averaged equations for air and produce energy conservation. Hence the conceptualisation conducted for the airflow models forms a basis for the heat transfer models.

- **Effects of radiative heat transfer**

As produce items are packaged in boxes or bins, the individual produce item would likely receive minimal net radiative heat transfer (Tanner, 1998). Hence, it was assumed that effects of radiative heat transfer were negligible.

- **Constant thermal properties**

Thermal properties of produce and package materials (package walls and trays) are affected mainly by moisture content. As moisture content variation can be negligible during forced- air cooling. Hence it was assumed that produce and packaging materials have constant density, thermal conductivity, and specific heat capacity. In airflow models, constant air thermal properties were assumed, which is still valid for the heat transfer models.

5.3 SOLUTION OF AIRFLOW AND HEAT TRANSFER MODEL

5.3.1 Thermal Boundaries

In addition to the above equations, it was necessary to define the conditions of the container box or pallets containing fruits (apples). As container box are made up of high density polyethylene (HDPE) and therefore heat transfer takes place between the cool air, container box i.e HDPE and apples. To determine the thermal interaction within the model, the boundary consisted of three different elements: cool air, the container walls, and the apples. While each of these sections had distinct thermal properties, a similar method of determining the effective heat transfer coefficients of each part was used. In general, an overall heat transfer coefficients was calculated using a thermal resistance model which took into account the conductive heat transfer within each layer of the boundary walls and the convective heat transfer to the cool air, if applicable. Overall, the heat transfer rate through the pallets can be expressed as follows:

$$q = h_{eff} \cdot A \cdot (T_w - T_{inf})$$

Table 5.1, Properties of materials

Layer	Material	Thermal property	Thickness
1	Air	$h_{inf} = 10$ W/m^{2}K	NA
2	HDPE	K=0.49 W/m K	0.004 m
3	Apple	K=0.560395 W/m K	0.54 m
4	Air	K=0.02397 W/m K	NA

DEVELOPMENT OF TURBULANCE MODEL FOR AIRFLOW AND HEAT TRANSFER MODEL

6.1 Introduction

Almost all fluid flow which we encounter in daily life is turbulent. Typical examples are flow around (as well as in) cars, aeroplanes and buildings. The boundary layers and the wakes around and after bluff bodies such as cars, aeroplanes and buildings are turbulent. Also the flow and combustion in engines, both in piston engines and gas turbines and combustors, are highly turbulent (André Bakker 2002) Air movements in rooms are also turbulent, at least along the walls where wall-jets are formed. Hence, when we compute fluid flow it will most likely be turbulent. The characteristic features (Tennekes & Lumley) such as:

I. **Irregularity.** Turbulent flow is irregular, random and chaotic. The flow consists of a spectrum of different scales (eddy sizes) where largest eddies are of the order of the flow geometry (i.e. boundary layer thickness, jet width, etc). At the other end of the spectra we have the smallest ed- dies which are by viscous forces (stresses) dissipated into internal energy. Even though turbulence is chaotic it is deterministic and is described by the Navier-Stokes equations.

II. **Diffusivity.** In turbulent flow the diffusivity increases. This means that the spreading rate of boundary layers, jets, etc. increases as the flow becomes turbulent. The turbulence increases the exchange of momentum in e.g. boundary layers and reduces or delays thereby separation at bluff bodies such as cylinders, airfoils and cars.

III. **Large Reynolds Numbers**. Turbulent flow occurs at high Reynolds number. For example, the transition to turbulent flow in pipes occurs that ReD 2300, and in boundary layers at Rex 100000.

IV. **Three-Dimensional. Turbulent** flow is always three-dimensional. However, when the equations are time averaged we can treat the flow as two-dimensional.

V. **Dissipation**. Turbulent flow is dissipative, which means that kinetic energy in the small (dissipative) eddies are transformed into internal energy. The small eddies receive the kinetic energy from slightly larger eddies. The slightly larger eddies receive their energy from even larger eddies and so on. The largest eddies extract their energy from the mean flow. This

process of transferred energy from the largest turbulent scales (eddies) to the smallest is called cascade process.

PARAMETERS AND BOUNDARY CONDITIONS

7.1 PROPERTIES AND MODEL PARAMETERS

All the parameters related to the apple, fan, and heat exchanger were extracted from several measurements or different references, shown in Table 7.1(Seyed Majid Sajadiye 2012). True density, bulk density, porosity, moisture content, and surface area of 100 apple samples were measured in order to calculate other parameters. Bulk specific area was estimated from the correlation developed by Dullien (1979). Apple thermo- physical properties were extracted from correlations developed by USDA (1986). based on the apple moisture content. Moist air properties extracted using psychometric table developed by ASHRAE (1994) Fan parameters obtained from the manufacturer pressure–flow curve. Heat exchanger parameters were obtained from measurements and also using correlation developed by McQuiston (1981). Apple bulk viscose and inertial resistances were extracted from correlation developed by Reichelt (1972) and Gaskell (1992).

Parameter	Value
Apple mean diameter	0.059 m
Apple true density	845.4 kg m^{-3}
Apple bulk density	462.313 kg m^{-3}
Apple bulk porosity	0.4531
Apple moisture content (wet based)	83.65%
Apple average surface area	0.0138 m^2
Apple specific surface area	60.43 m^2 m^{-3}
Apple skin mass transfer coefficient	1.67×10^{-10} kg m^{-2} s^{-1} Pa^{-1}
Apple heat capacity	3639.275 J kg^{-1} °C^{-1}
Apple heat conductivity	0.560395 W m^{-1} °C^{-1}
Moist air density	1.2893 kg m^3
Moist air relative humidity	90%
Moist air heat capacity	1006.408 J kg^{-1} °C^{-1}
Moist air heat conductivity	0.02397 J kg^{-1} °C^{-1}
Moist air viscosity	1.72×10^{-5} kg m^{-1} s^{-1}
Water vapor diffusivity in the air	2.1×10^{-5} m^2 s^{-1}
Latent heat of water (at 0 °C)	2495460 J kg^{-1}
HDPE density	952.5 kg m3
HDPE heat capacity	2250 J kg^{-1} °C^{-1}
HDPE heat conductivity	0.49 W m^{-1} °C^{-1}
Heat exchanger's heat transfer area	8.81 m^{-2}
Heat exchanger's Heat transfer coefficient	42 W m^{-2} °C^{-1}
Heat exchanger's average surface temperature	-2.1 °C^{-1}

SIMULATION RESULTS AND MODEL VALIDATION

8.1 Introduction

This chapter presents the predicted airflow velocity and temperature profiles during forced-air cooling of produce in different airflow patterns in a cool storage which were considered in this research work. The predicted temperature profiles were compared with experimental data for model validation.

The data collected using ANSYS FLUENT 12.0 included the temperature data and airflow velocity at the monitor point and the temperature distribution and airflow velocity at all node points in the model room at the end of the 30 minutes (1800 seconds) simulation. The normalized residuals of all computations were required to be at a 0.001 level. The above mentioned three models were simulated at five different velocities from 1 m/s to 5 m/s in this research work but the comparison made in this research was at 5 m/s velocity only.

For detailed discussion and comparison at different points inside the model room, Iso – planes were created at different location .The locations of plane were selected as per model geometry are shown below.

Table 8.1, iso – planes distances Iso-plane name	X - distance(m)	Y - distance(m)
1	1.30	0.05
2	1.80	0.67
3	2.30	1.27
4	2.80	1.87
5	3.30	2.47
6	3.80	–

Table 8.2, the above drawn curves can be concluded in the above table Airflow velocity measuring points	Normal airflow pattern (Ist case) (m/s)	Change airflow pattern (IInd case) (m/s)	Air outlet duct (IIIrd case) (m/s)
Maximum	6.697	6.237	13.029
Net average	1.880	1.667	2.711
Interior surface	2.108	1.881	2.980
X-planes			
x1	0.767	0.746	1.939
x2	0.702	0.673	1.251
x3	0.633	0.784	1.203
x4	0.647	1.099	1.205
x5	0.640	1.379	1.188
x6	0.426	0.718	1.278
Y-planes			
y1	3.286	2.480	5.698
y2	1.678	1.669	4.366
y3	3.431	1.014	6.589
y4	1.600	1.302	4.158
y5	2.225	3.361	4.338

CONCLUSION & DISCUSSION

A three-dimensional CFD simulation was developed to compare airflow and heat transfer models in different airflow patterns in a fully loaded cool storage Dynamic behaviour of the fan, and heat exchanger were considered in the model. In this project work three different airflow patterns were compared and analyzed. The calculation domains were discretised with a tetrahedral meshing with mesh size of 0.004 m for the entire domain was selected. Using ANSYS FLUENT 12.0 software a finite volume code was used for the numerical implementation of the models. Standard k - ε turbulence model was enabled based on the previous studies (Nahor, et al., 2008). The overall accuracy of the model was selected as second order upwind. The pressure–velocity coupling was ensured using SIMPLE algorithm, the model was solved for velocity field. Initial conditions were kept same for all the three cases. The comparisons were made on inlet velocity of 5 m/s.

After the airflow velocity comparison we found that best velocity distribution was found in air outlet duct condition or IIIrd case. It was found that the magnitude of maximum velocity was 13.029 m/s. The velocity magnitude at the outlet was 12.216 m/s and in the interior surface of the model room was 2.98 m/s and net average velocity was 2.667 m/s.

Then we found in Ist case the magnitude of maximum velocity was 6.697 m/s. The average velocity magnitude in the interior of the model room was 2.108 m/s and net average velocity was 1.880 m/s.

The airflow velocity distribution in IInd case i.e. with change airflow pattern, we found the magnitude of maximum velocity was 6.237 m/s. The velocity magnitude in the interior surface of the model room was 1.871 m/s and net average velocity was 1.669 m/s.

After the cooling comparison it was found that the best cooling was achieved in IIIrd case i.e. with air outlet duct. The magnitude of net average temperature inside the room was 271.461 K or –1.689 oC, the temperature of pallets was 272.954 K or –0.196 oC and the temperature in the interior surface was 271.333 K or –1.817 oC.

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