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Computational Depiction of Transportation Phenomenon Control using Auxiliary Draught in Cold Storage

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Abstract: ***Computational Flow Analysis Enables To Demonstrate Large Cold Storage Air Distribution. Thermal Performances Of The Cold Storage Are Based On Airflow Mechanism In A Closed Chamber. Experimental Investigation On Cold Store A Costly Affair Thus, In This Paper The Impact Of Auxiliary Draught System (Ads) On Airflow Distribution Is Carried Out With The Help Of Computational Fluid Dyanamics (Cfd). Experimental Validation Shows A 26% Of Error In Simulation Results. Cfd Results Reflect That Use Of Ads Reduces Cooling Time By 25%. Auxiliary Draught Ensures A Homogeneous Environment Inside The Plant.***

Keywords: ***Cold Storage, Auxiliary Draught System, Computational Fluid Dynamic (Cfd), Air Circulation, Sst K-Ɛ Model,***

I. INTRODUCTION

The cold storage is refrigerated space, using a vapor compression refrigerated system, which reduces the temperature of both the air and products. It is the most commonly used technique for preservation of perishable items with their desirable characteristics; thus, reducing the loss of quality, and increasing the shelf life (ASHRAE, 2009) [1]. Cold storage facilities ensure a safe and prolonged life of perishable stuff by using the refrigeration cycle at large scale. Perishable stuff stored in cold storage at very low almost freezing temperatures. For the best results, it is recommended that the temperature for each vegetable is maintained with fewer fluctuations, with the appropriate distribution of cooling air from the evaporator. Airflow patterns are crucial for life and quality of preserved foodstuff in the cold storage. Heterogeneous air distribution in the chamber caused variation of cooling rate and non-uniform temperature of the product inside the chamber. It leads to deterioration of stuff due to high respiration at high temperatures or frosting injury at low temperatures [2].

1.1. Importance of air distribution cold storage chamber

To maintain uniform and fast cooling inside the cold storage chamber, it is important that supply cool air covers every nook and corners of the chamber, as well as return air reach evaporator uninterrupted and fast. In a cold chamber supply air take heat from stored stuff and from the leaked surrounding air in the chamber. Return air transfer heat to the evaporator. In this manner, air circulation is set up in a cold storage chamber through which cooling is achieved. The process is demonstrated in Fig. 1.

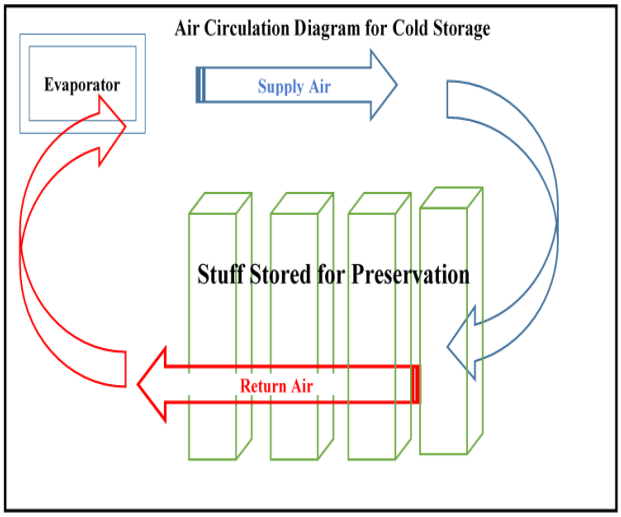

Figure 1. Air Circulation Diagram for Cold Storage Chamber

The poor airflow distribution causes a higher relative humidity zone that responsible for the loss of foodstuff (potato) because of the condensation of moisture at various parts of stacking. Such conditions in potato cold stores raise the storage losses up to 10%, while the prescribed maximum limit is 5% for 8 months storage period [3]. Heat and mass transfer in agricultural products are a necessary part of cooling and storage. It is difficult to attain uniform cooling of products stored in industrial cooling rooms, owing to the existence of an uneven distribution of the airflow [4-5]. Cold air distribution is as important as other parameters like temperature, relative humidity, etc., the velocity of cold air directly affects the heat transfer time. Airflow pattern in the cold chamber is responsible for temperature distribution and its uniformity [6].

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As discussed above the thermal gradient inside the cold storage is dependent upon air distribution and its disturbance. Various factors are responsible for airflow and its distribution, like the selection of evaporator coil position, fan speed, height and gape between stakes rows and columns, cold room geometry, etc. Cold storage length and width and height are very important for airflow and its distribution in the chamber. Cold Air is blown from the evaporator coil is picked heat from the preserved items and transfer them to the refrigeration system. Thus, as maximum the reach of cool air, higher the refrigerating effect. Researchers were observed that cold air from the evaporator coil is not able to cover the whole length of the chamber there is a stagnant zone with poor ventilation in the rear part of the cold storage. The poorer reach of air in the farthest part (from the evaporator) of the chamber causes higher temperatures at the section in comparison to parts near to the evaporator [7, 8, 9]. Son H. Ho et al 2010, in his simulation analysis of cold storage, demonstrate that cold air from the evaporator does not move too away from evaporator coil and short circuit back to cooling coil. This phenomenon causes a high-temperature region at the rear portion of the cold store [10]. S. Duret et al. 2014, in his experiment, show that return air velocity is higher as compare to air supply velocity in the rear section of cold storage, it causes relatively low heat transfer at the rear section in comparison with cooling coil side[11, 12, 13].

CFD modeling of cold store units required some specific features as there are large airflow obstacles by means of stockpiling of agricultural goods, air-conditioning systems affect room boundary condition, the evaporator unit can be treated as a source of momentum and low air velocity ± 3m/s with turbulence [14, 15, 20].

the k-ε model provides results with approximately 24% error and is easy to use for modeling, as these models required less user skill and low computing capacity.

II. METHODOLOGY

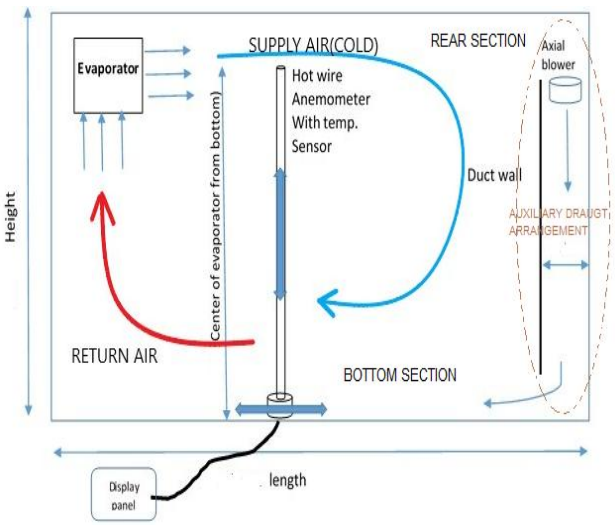


Figure 2. schameric diagram of praposed plan of action

CFD Governing Equations

This section is a summary of the governing equations used in CFD to mathematically solve for fluid flow and heat transfer, based on the principles of conservation of mass, momentum, and energy. Details of how they are actually used in the CFD computations are described below.

Conservation Equations

The conservation laws of physics form the basis for fluid flow governing equations are:

➢ Law of Conservation of Mass: Fluid mass is always conserved.[]

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \qquad (1)$$

➢ Newton's 2nd Law: The sum of the forces on a fluid particle is equal to the rate of change of momentum.

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial p}{\partial x_i} \qquad (2)$$

➢ First Law of Thermodynamics: The rate of head added to a system plus the rate of work done on a fluid particle equals the total rate of change in energy.

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right) \qquad (3)$$

Set up Dimension and boundary conditions for CFD Simulation:

Chamber size:

1.5m (l) x 1m (w) x 1m (h),

Evaporator

Dimensions: 0.5m x 0.16m

Position: Midpoint is 86 cm above the base of the chamber, 14 cm from the back-side wall and 50 cm from sidewall as shown in figure 1. The front face of the evaporator is 22 cm from the back sidewall.

Flow capacity of evaporator: 2.28 m^3/min

Auxiliary Draught System :

Auxiliary draught fan capacity 1.14m^3/min

Auxiliary draught wall size: 1m x 0.8m

10 cm above the ground and 10 cm below the top face.

The duct wall is 0.16m away from the chamber wall opposite to evaporator. Two fans fitted at top of the duct and through air axial downward.

The third condition for the experiment is duct wall having equally spaced slots of 12 cm x 84 cm cuts horizontally lengthwise

Stacking: Size of carats: 17cm(L) X 12cm (w) X 10cm (D). Total no. of carats in chamber 48 in an array of 3x4 with 4 carats in each column. Each carat has 1.3kg (approx.) of onion.

Three-dimensional Model Description:

A 3-dimensional model of a chamber in the shape of a rectangular prism is developed for Case 1 and Case 2 & case 3 of the evaporator and auxiliary draught arrangements.

The physical dimensions set to be 1.5m length, 1m width, and 1m height. The model geometry will be created using preprocessor ANSYS DESIGN MODELER.

Onions were packed in carets with a capacity of 1.3 kg/caret. The pallets or container's walls were made of high-density polyethylene (HDPE) and modeled as conducting walls. Containers of apples were modeled as porous media. The enclosure was loaded with 48 carats with an in-line array with dimensions of 1.2 m length, 1m width, and 0.75m height.

Case 1:

The cooling unit is located at the top center of the storage and consisted of axial fans for air circulation and a finned tube heat exchanger at a height of 0.86 m from the bottom wall or floor and 0.1 m from the back wall. The inlet was given to the bottom of the fan and the outlet was given to the front of the heat exchanger. The chamber is considered empty. Flow analysis was done for three conditions of the chamber first without auxiliary draught, second with auxiliary draught and the third condition with auxiliary draught having slots.

Case 2

Same as the first case but the chamber is loaded with onion in carats.

2.1 Boundary conditions

Operating Condition: Pressure = 101325 Pa
Velocity inlet = 3 m/s
Turbulent intensity = 1%
Hydraulic Diameter of evaporator = 0.243m

Pressure – Standard

Momentum – Second order

Turbulent Kinetic Energy (k) – First order upwind

Turbulent Dissipation Rate (ε) - First order upwind

Figure 3 represent CFD algorithm for problem solving.

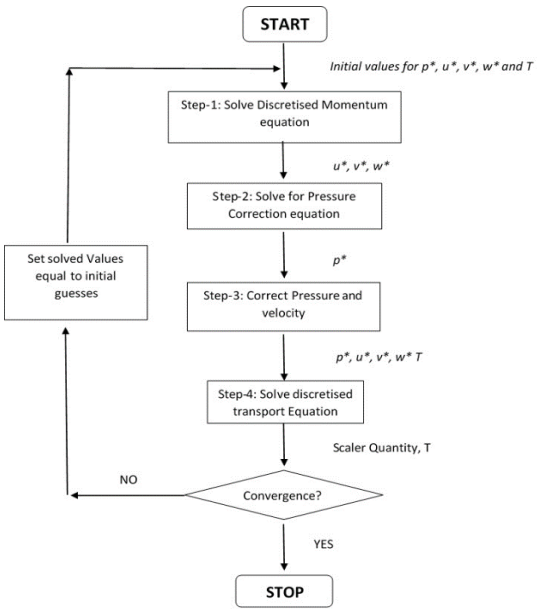


Figure 3. CFD Sample Algorithm

2.2 Modeling of cold chamber

Geometrical modeling and boundary conditions set according to the above discussion and parameter values set as per table 1.

Table 1: Model parameters and parameter values

Parameter	Value
Onion mean diameter	0.029 m
Onion true density	970 kg/m^3
Onion bulk density 462.313 kg m–3	547.48kg/m^3
Onion bulk porosity 0.453	0.4234
Onion moisture content (wet based)	83.45%
Onion average surface area	0.001404m^2
Onion skin mass transfer coefficient	0.8867kg/m^{2}MP
Onion heat capacity	a
Onion heat conductivity	3.77KJ/kg °C
Moist air density	0.356 W/m K
Moist air relative humidity	1.29 kg/ m^3
Moist air heat capacity	91%
Moist air heat conductivity	1007.41 J/kg °C
Moist air viscosity	0.02396 J/kg °C
Water vapor diffusivity in the air	1.72x10^{-5} kg/ms
Latent heat of water (at 0°C)	2.1x10^{-5} m^2/ s
HDPE density	2.495 KJ/ kg
HDPE heat capacity	952.5 kg/ m^3
HDPE heat conductivity	2,250 J/kg °C
Heat exchanger's heat transfer area	0.49 W /m C
Heat exchanger's heat transfer coefficient	0.2 m^2
Heat exchanger's average surface temperature	42 W /m^2 °C
	-3 °C

Geometry for all cases and conditions built on ANSYS 14.5 workbench.

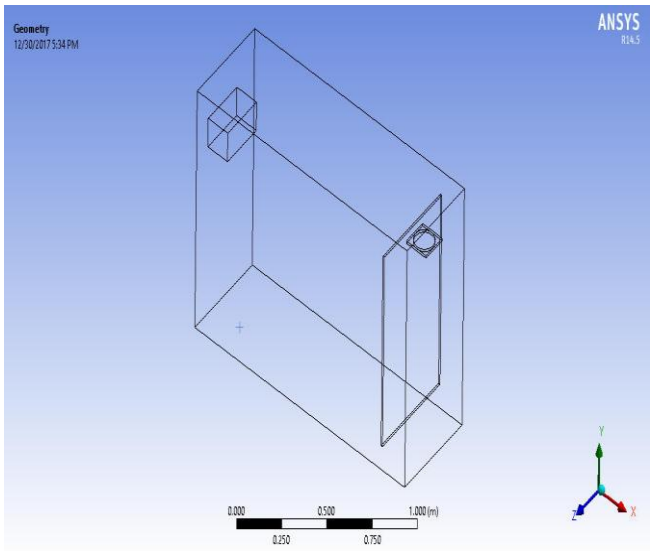
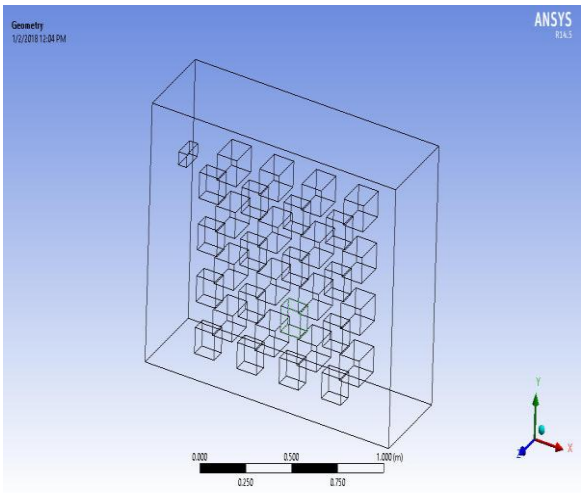
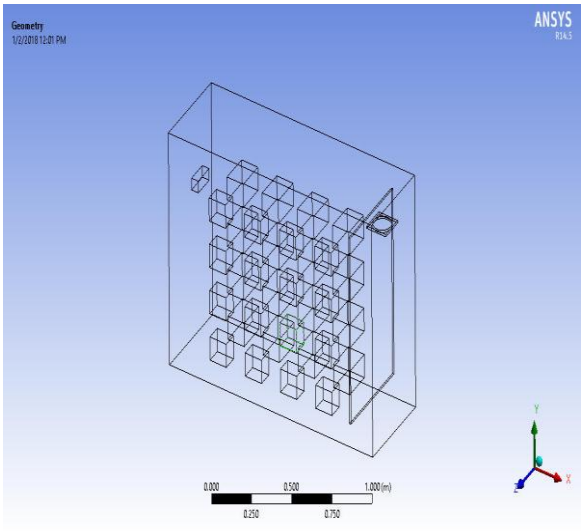


Figure 4. Geometry for Empty Cold storage

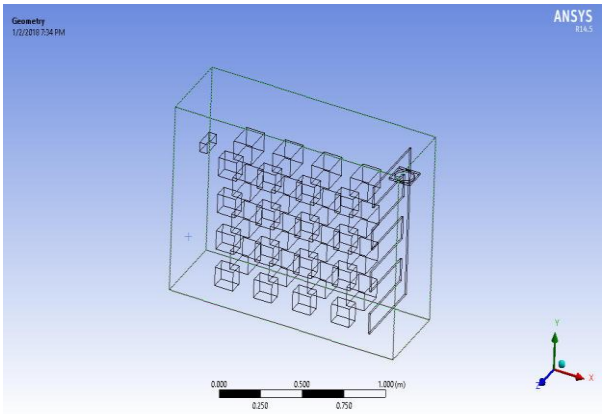
Figure 4 represents the geometry of the room without a loaded condition. The analysis was done on all three auxiliary draught conditions in the empty room.



(a) Cold chamber with slotted auxiliary draught



(b) Cold chamber with plane auxiliary draught



(c) Cold chamber with slotted auxiliary draught condition

Figure 5. Geometry for the chamber with carats

Figure 5 and Figure 6 represent the geometry of cold storage and meshing of the same. Tetrahedral mesh with about 950000 cells used for meshing.

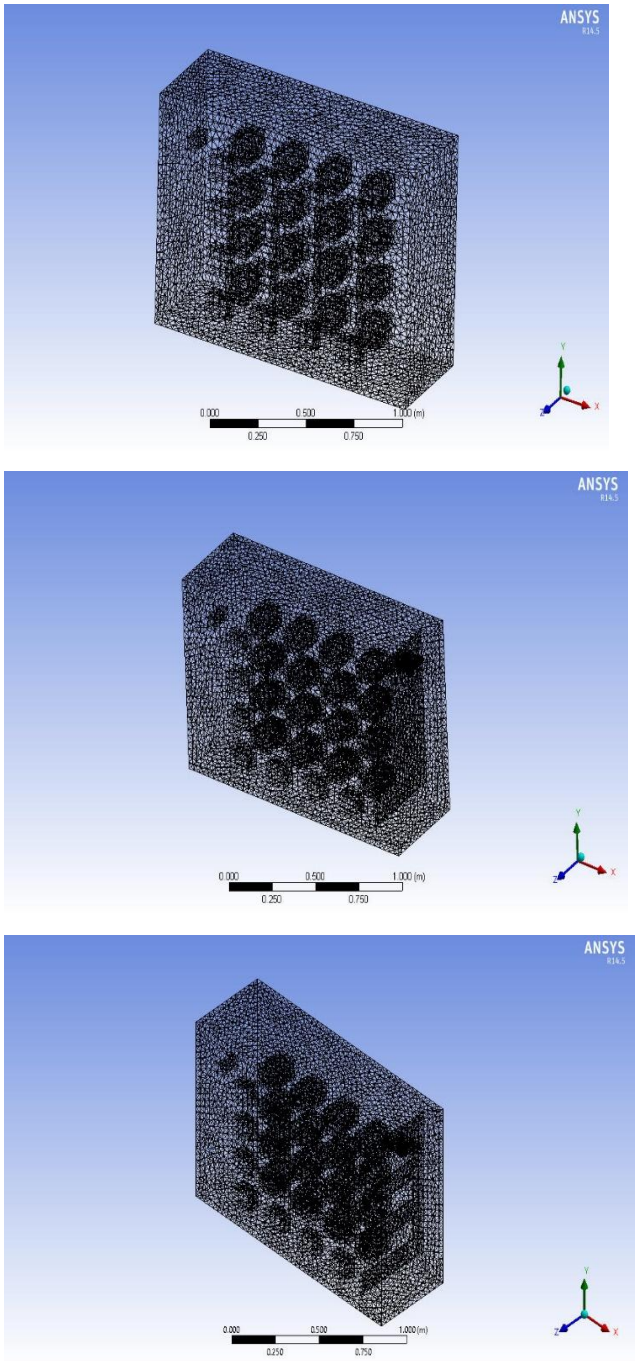


Figure 6. Meshing for all Three Cases

2.3 Validation method:

As very high cost is involved in experimentation of actual loaded cool storage, thus a geometrical similar chamber erected for validation. The existing validation process adapted by Hoang et al. [12] and Nahor et al. [2] was used in this investigation. Product temperature, airflow temperature, and air velocity were measured.

For flow measurement, a telescopic hotwire anemometer (HTC AVM 08) was used. It can measure velocity from 0.1m/s to 25m/s. The anemometer is also capable to measure instantaneous airflow temperature. T- type thermocouple was used to measure product and spatial temperatures.

III. RESULTS AND DISCUSSIONS

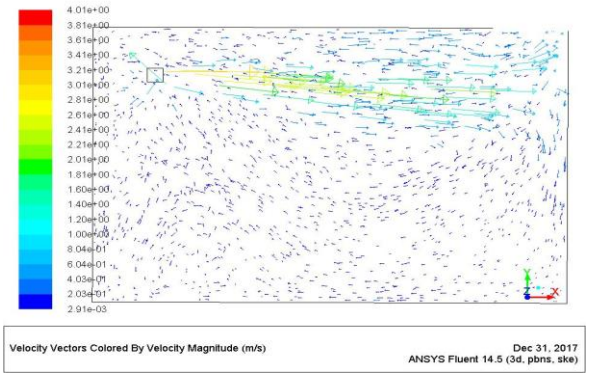


Figure 7. Airflow velocity distribution in the empty cold chamber

Figure 7 demonstrates the flow pattern the same as discussed in figure 1 in the first part of the paper. Average flow velocity observed 2.5m/s in the room if there is no obstacle to flow.

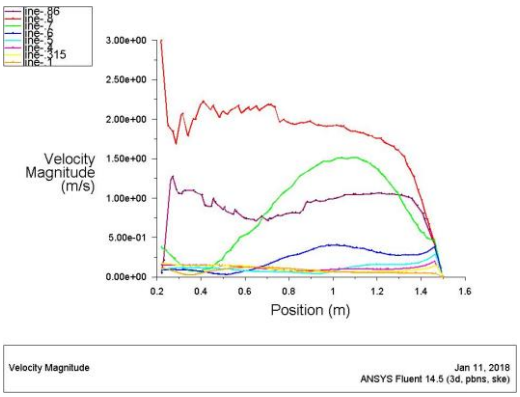


Figure 8. velocity profile for general configuration

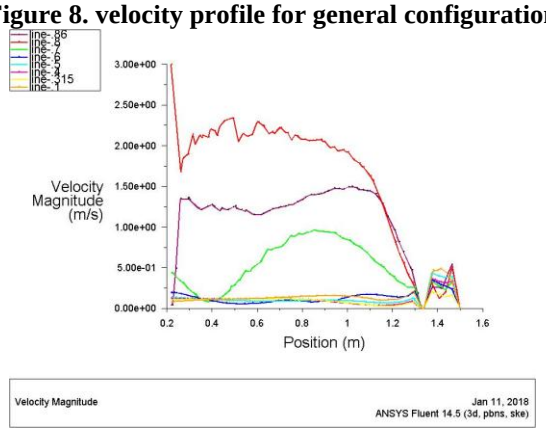


Figure 9. velocity profile for ADS slot closed configuration

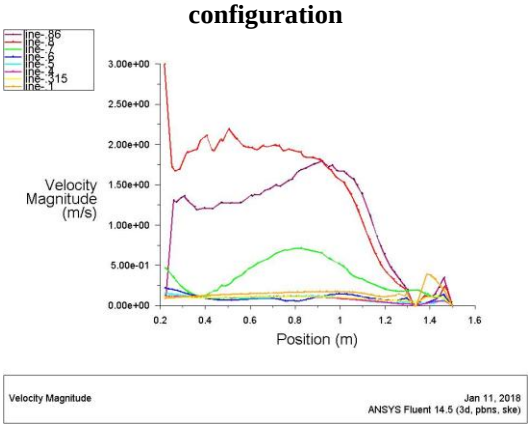


Figure 10. velocity profile for ADS slots open configuration

Fig 8 shows that velocity at the top layer or in front of the evaporator is in order of 2.5 m/s to 3 m/s. but drop drastically away from the source. Velocity at the rear section is in range of 0.47 m/s to 0.947 m/s at 80cm above the ground and 90 cm away from the evaporator. Velocity at mid-section and around the buckets was observed in the range of 0.15 m/s to 0.6 m/s. The return air velocity just below the evaporator is 1.1 m/s. With duct fitting at rear section i.e. 110 cm away from evaporator velocity at 86 cm above and ground now maintaining velocity in a higher range as compared to the last case. i.e. velocity is 0-9 m/s to 1-74 m/s from 80 cm to 100cm from evaporator.

With the use of ADS, the return air velocity at the bottom now is in the range of 0.9 m/s to 1.11 m/s. Return air velocity just below evaporator is possessed much higher speed as compared to the first case. Near buckets velocity at mid-section is not improved too much.

In fig.10 slotted ADS is used and because of slotting improvement at the midsection is sighted. Velocity near buckets, in this case, is in the range of 0-9 m/s to 1-2 m/s at mid parts of the chamber.

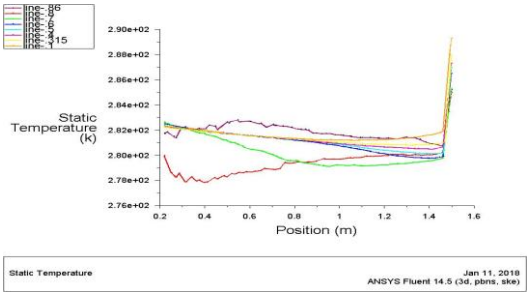


Figure 11. Temperature profile for general configuration

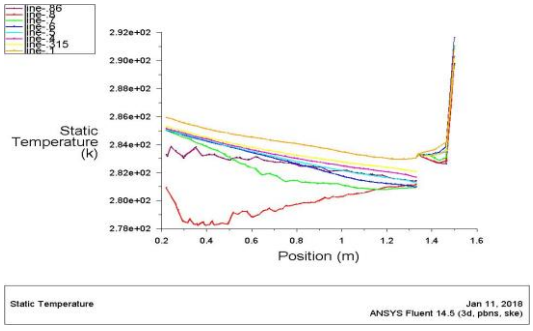


Figure 12. Temperature profile for ADS slots closed configuration

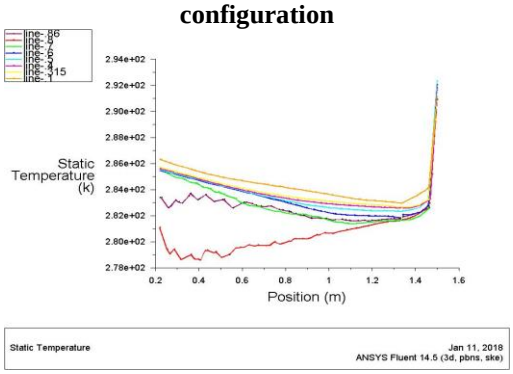


Figure 13. Temperature profile for ADS slots open configuration.

On compare temperature graphs in the above figures it is clear that the temperature of return air increase with ADS in both cases confirms that air circulation is improved as a result convective heat transfer rate is faster as compared to a general configuration.

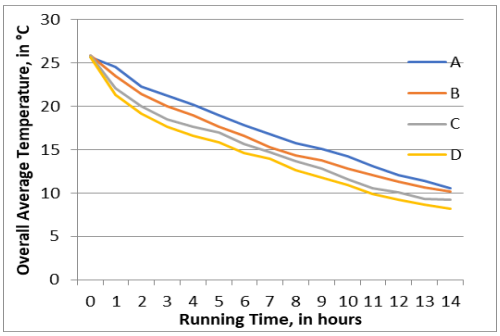


Figure 14. Room temperature drop rate for various configuration

The above graph shows that in due course of time ADS promotes 21% fast cooling as compared to other configurations.

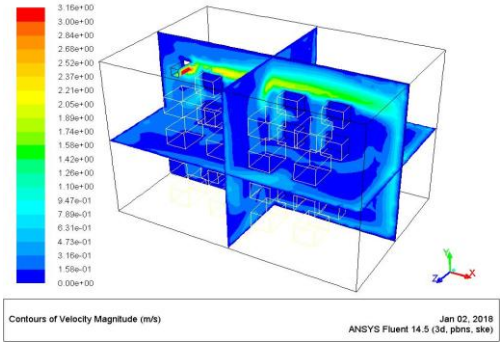


Figure 15. Cold Room velocity profile in filed condition general configuration

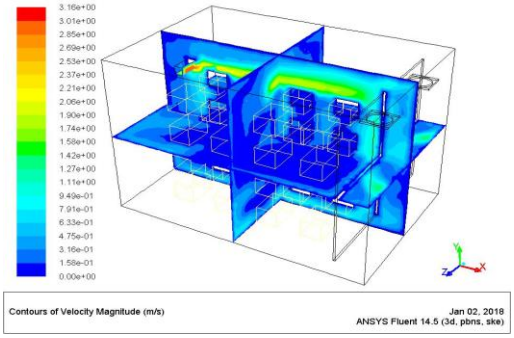


Figure 16. Cold Room velocity profile in filed with ADS

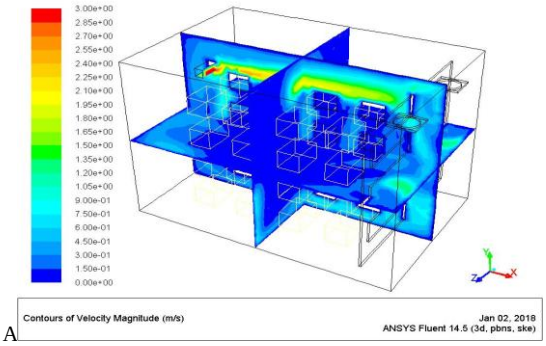


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Figure 17. Cold Room velocity profile in filed condition with slotted ADS

On comparison of the velocity profiles in figure 15, 16, 17 it is clear that mixing in mid-section improved with the sloted ADS, eventually, it reduces the cooling time of the product, also uniform temperature distribution is ensured. Over-all average airflow velocity in the cold room increased by 2.3 times as of general configuration. The airflow velocity in the bottom section is improved considerably.

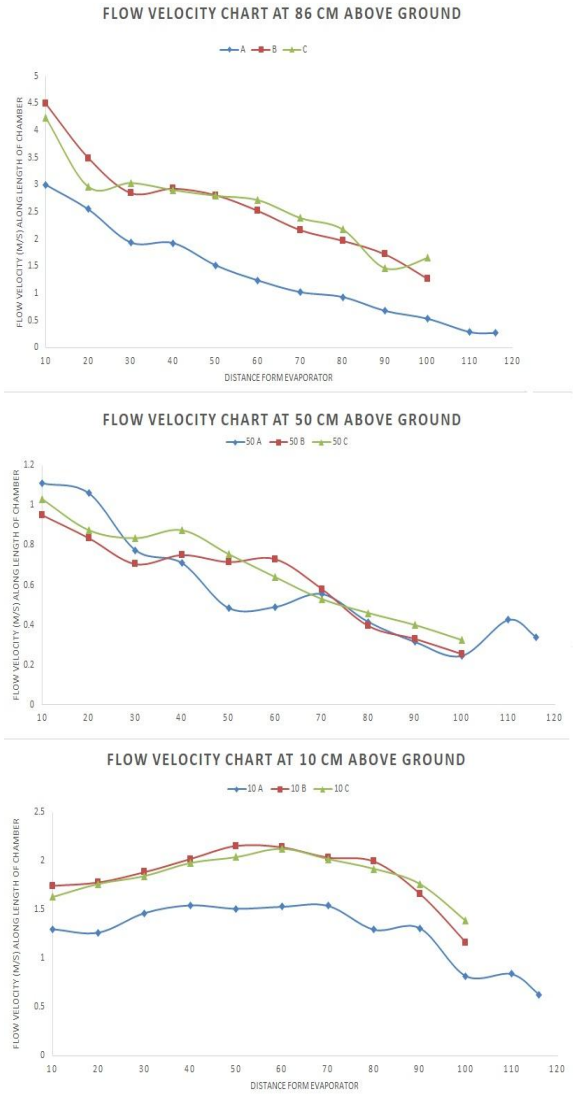

Figure 18. Experimented results of Cold Room airflow velocity distribution

Experimental airflow velocity values (figure 18) were compared with CFD results and it has been observed that the velocity profile has almost similar. There is a 23% error in velocity results as given by CFD. Error is considerable as per the literature review suggested the same, it because of standard assumptions made during CFD analysis. M. L. Hoang et al., demonstrate a relative error of 26% for airflow velocity calculation thru SST k- ε model [12, 13, 16, 18], also Zhang et al indicated that the prediction accuracy of 3D CFD model for air temperature was with the average percentage errors of 7.5%, when compared to measured values [23] and Moonne et al. found error 10% for the same [24].

As discussed above indicate the auxiliary draught system can improve the airflow distribution in the cold chamber. Computational model selection is validated with prototype experimental value and also justified with the literature review.

IV. CONCLUSION

Results suggested that ADS can play a vital role in cold chain industry as it can be implemented over existing facilities of cold storages. The use of auxiliary draught significantly reduces the cooldown period by 23% as a result of the saving of energy. For the same time period, the room is 25% cooler. ADS raises the airflow velocity almost double at the top section of the chamber and 36% improved in the bottom section of the chamber. The average product temperature is lower for the same time duration (i.e. 8.15^{0}C) as compared to general arrangements.

Experimental results conclude that the use of auxiliary draught is advantageous in cold storage operation as because of this uniform flow distribution reduces the load on the cooling system. More convective heat transfer can be possible because of the proper mixing of air.

Nomenclature: -

ρ	density	kg/m^3
μ	viscosity	N-s/m^2
u	velocity (x axis)	m/s
v	velocity (y axis)	m/sr
w	velocity (z axis)	m/s
k	conductivity	W/m-k
T	Temperature	^{o}C
P	pressure	Bar
C_p	Specific Heat	kJ/kg K

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