



Auxiliary Draught Base Performance Improvement of Air Distribution System of Cold Store

Pankaj Mishra, K. R. Aharwal

Abstract: Air distribution enable convective heat transfer in cold storage operation. Thermal behaviors of the cold storage system are based on air transport arrangements. Transport characteristics can handle with auxiliary arrangements such as induce draught system. Experimental investigation for the impact of auxiliary draught system (ADS) on air transportation is carried out. Air transport velocity was measured in the cold chamber with a hot wire anemometer. Experimental results show significant enhancement by three times mid-section air flow velocity and overall one- and half-time greater flow velocity observed, while return air velocity measured almost tow time of general condition during the experiment. COP of plant improve by 21% with 25% less time required to achieve desired temperature. 26% saving in power consumption observed during experiments. Auxiliary draught ensures homogeneous environment inside the plant through proper mixing of air and support convective heat transfer. Designing and analysis of airflow patterns with temperature distribution in large entity like cold storage is a difficult task thus Computational Fluid Dynamics (CFD) can address the issue with high degree of precision. It has been observed that SST K- ϵ model has average 26% error with experimental values.

Keywords: Cold Storage, Auxiliary Draught System, Computational Fluid Dynamic (CFD), Air Circulation, SST K- ϵ model,

I. INTRODUCTION

India is the second-largest producer of fruits and vegetables, the number one producer of milk and milk products also one of the key producers of meat and fish. Even with these positive points, in India a large part of population is suffering from starvation.

Problem can be handled if large cold storage chain established in the country. Perishable item can store in cold storage for long time with maintaining quality of products.

The advantages of employing cold store is increased availability of perishable in the lean season. Cold storage is divided into three types, Frozen ($<-18^{\circ}\text{C}$), Chilled (0°C to 10°C), Mild Chilled (10°C to 20°C) and Normal ($>20^{\circ}\text{C}$) storage [1].

Perishable product storage takes place in very large spaces with high ceilings. In these spaces, cooling is normally provided by ceiling-mounted evaporator fan coil units. It draws air from the space and discharges it at high velocity directly back to space as the system is isolated. Effectiveness of cold storage system needed large air circulation rates and air velocities, which, combined with the low temperatures, cause high energy consumption in the space. Therefore, air distribution is an essential factor that needs to be prudently taken in order to create an environment capable of maintaining food quality and shelf life without excessive energy consumption.

Mixed airflow (ventilation) is the generally adopted airflow distribution method for cold storage facilities by which cold air is mixed through the entire cold storage capacity. In mixed ventilation or general configuration of cold storage in loaded conditions various temperature zone built inside the store leads to frosting (at low-temperature reason), fouling, nutrition loss, and respiration (at high-temperature zone).

Controlled air distribution can help to maintain the low temperature only to areas occupied storage activities that will lead to significant energy savings and product life with quality. The challenge is to achieve this air distribution in existing cold storages within the techno-economic frameworks. In order to succeed in this task, the complete renewal of the cold storage equipment may be prohibitive. As a result, the air distribution solution should be capable to be adopted by the existing refrigeration equipment.

For efficient cold storage operation various factors are responsible like the design of the chamber, evaporator size and position, insulation, stacking arrangements, goods store, air circulation, etc., Among all the above mentions factors, Airflow pattern and circulation is one of the most crucial factors for life and quality of preserved foodstuff in cold storage. Heterogeneous air distribution in the chamber caused variation of cooling rate and non-uniform temperature of the product inside the chamber. Researchers have observed the stagnant zones with poor ventilation in the rear part of the cold storage chamber which leads the higher temperatures in these zones.

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Researchers who work on cold storage airflow analysis mostly adopt computational techniques for their study's experimental investigation for the same was too low.

II. LITERATURE REVIEW

Many researchers work on cold storage and its air distribution pattern. Most of them adopted computational techniques for understanding the airflow pattern. Basic objectives on which researchers were work are life of product and thermal performance of cold storage. Large office space experimentally analyzed for the airflow mixing and overall temperature gradient with displacement of ventilation port and chilled ceiling diffuser [2]. Analysis of thermal performance with power saving in large room with return grill at various part of the room to controlled air circulation create comfort for occupants [3]. Swirl and direct diffuser with exhaust vents improve thermal comfort of indoor facilities with air quality and energy consumption in case of large building with lots of obstacles in the path of air flow [4]. Lin and Tsai (2014) reported that with higher mixing rate of air flow for diffuser cause reduction in temperature gradient in the space [5]. Jurelionis et al.(2015) investigated on test-chamber for the impact of the airflow as per source selection on mixing efficiency. Visualization of flow is done by aerosol particles. Results explained that one-way mixing diffusers with low flow rate are more efficient compared with four-way mixing and high air exchange rate diffusers [6]. S. Duret et al. experimented in a chilled enclosure pallets field with apple. With simple air flow geometry, they had measured airflow velocity and temperature for the test cold chamber. [7].

Computational widely used in air flow analysis in cold chamber as it is difficult to adopt robust measurement in large rooms. Moureh et al. analyzed the velocity characteristics throughout a long-ventilated enclosure considering different inlet flow arrangements. The authors employed the high and low Reynolds numbers form of the two-equation $k-\epsilon$ model and the Reynolds stress model (RSM). According to the results, the RSM was able to predict correctly the general behavior of primary and secondary airflow recirculation. Subsequently, they investigated the airflow patterns above and within an enclosure with vented boxes [8, 9]. RSM model predict well the air ventilation level values as obtained by experimentation [9]. Son H. Ho et.al works on temperature and velocity simulation in cold stores numerically. Numerical modeling of the fluid flow and heat transfer can be used in this document, as well as for the assessment of thermo-uniformity in cold storage. This article also provides the simulation of 3-D and 2-D systems for a reduction in cost through computational analysis, as well as the finding, was with good accuracy observed [10]. Food stacking guidelines for fast perishable products which can affect by excess air contact, such products stored in solid piles to reduce air circulation around the product to the minimum [1,11]. For reducing the load on computation system porous media of stacking consider as series of wind tunnels, results were validated through experimental setup through measurements of temperature and velocity of airflow inside the chamber. 23.2% error was found in results for flow velocity prediction

as compared to experimental data, and 9.1 % error for weight loss of product was observed [12]. M.K. Chourasia, et al. did CFD simulation for analysis of cold storage operating parameters and their effects on the performance of the system, for heat transfer and moisture loss in potato stockpiling. Loss of moisture and RH in the bulk medium improved with an enhanced coefficient of skin mass transfer [11]. M.L.Hoang et al. presented an analysis for airflow in cold storage using the CFD approach employing the Reynolds-averaged Navier-Stokes equations with the $k-\epsilon$ turbulence model. A comparison of numerical results with investigational measurements showed an averaged difference of 26%. [2] Tapsoba et al (2006) found that for large cold store truck three-time high flow rate through evaporator required to maintain low temperature till last row of pallets. Consequently, high air renewal requirements computed [13]. H.B. Nahor et al. worked on transient three-dimensional CFD model, with 22% error as compare to measured values of airflow velocity, temperature etc.for empty store and 20% error for loaded cold store [14]. Serap Akdemir et al. studied the local temperature distribution in cold store. Their results are achieved at the ceiling, medium and floor level in the cold store and for different storage temperatures (0 °C, 1 °C, 2 °C, and 3 °C). Mapping software was presented to show the variability. Also, they indicated that the spatial distribution of the temperature and the relative humidity was not uniform in the cold store [15]. As selection of model difficult thus researcher reviewed various models for 2 d and 3d simulations. They also take consideration of use of Combined Heat and Power(CHP) in cold storage. Their work focus on energy saving and high performance of cold store [16, 17, 18]. Chourasia et al. applied the RNG $k-\epsilon$ turbulence model with the finite volume solution technique. Average overall errors of 19.5%, 8%, 0.5°C, and 0.61% were found for air velocity, air flow temperature, product temperature and moisture loss from the potatoes respectively [19, 20,21, 22]. Delele et al.(2013) found that the four different models with their individual prediction accuracy regarding the air velocity were the Standard $k-\epsilon$ model with 24.3 % error, Realizable $k-\epsilon$ model with 23.5 % error, RNG $k-\epsilon$ model with 22.4 % error and SST $k-\omega$ models with 18.2% error. A comparison between predicted air pressure drop and produce temperature and measured values showed a good agreement with an average relative error of 13.80% and 16.27%, respectively [23, 24 25, 26]. Ambaw et al. reviewed that for air velocity prediction, the SST- $k-\omega$ model produced the smallest error compared to the RSM and $k-\epsilon$ models[27, 28]. Laguerre et al. in order to avoid the computational time of a CFD model, created a simplified model of a cold room using the knowledge obtained from experimental measurements of air velocity, air temperature, and food product weight losses. The model was separated into zones and heat balance equations for each zone were developed. The simplified model was found to predict the product cooling rate and the final product temperature at different positions in the cold room quite well [29]. Pasut et al. validate SST $k-\epsilon$ model with experimental results for cold store [30].

2.2 Problem Identification

Literature review illustrated that various research were carried out on cold storage or large chilling facilities for airflow distribution analysis and its impact on refrigeration system performance. Researcher were observed that at rear parts (opposite section to evaporator position) of the cold chamber higher temperatures zones build, because of the airflow stagnant zones, To address the stagnant zone in rear section of the chamber auxiliary draught is adopted in this research work.

Evaporator coil throws supply air axially along the length of the room but supply air is not able to cover the whole length of the room because of flow resistance and obstacles in the flow path. As a result, a stagnant zone or low airflow zone is created at the rear part of the cold room. In stagnant zone heat transfer rate through convection is low and it may cause damage to stored foodstuff.

Presented work demonstrates the effect of auxiliary draught on airflow circulation in cold chamber with the help of flow velocity measurements at a different section of the room for various conditions. Auxiliary Draught is an arrangement in cold chamber, in which a duct section is placed on opposite wall to evaporator side with downward axial blowers fitted on it. It results in development of induce airflow toward rear farthest section to evaporator, in the cold room.

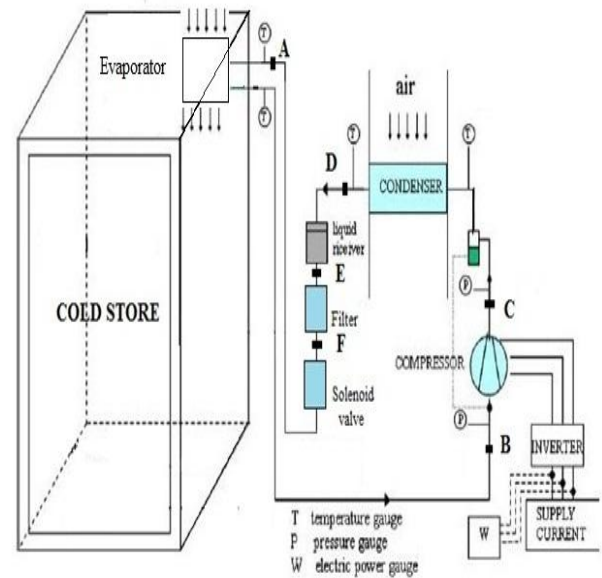
III. RESEARCH PROJECT DESCRIPTION:

The scope of this research is to improve the efficiency of cold air distribution in cold storage. Improved temperature distribution should lead to the reduction of the overall energy consumption of the refrigeration plant. As there has been no work reported on the air distribution systems improvement using auxiliary draught.

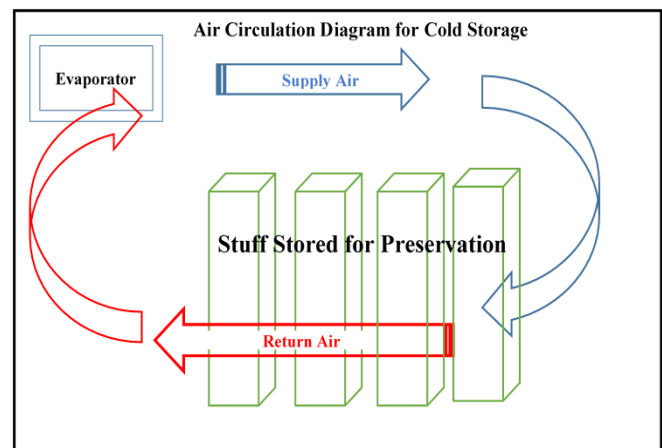
3.1 METHODOLOGY & EXPERIMENTAL SETUP

This chapter deals with the experimental set-up and the initial monitoring of the developed test rig. The experimental set-up was designed and built in order to represent an effect of auxiliary draught on performance of existing cold room and its temperature distribution system. In addition, a prototype cold chamber analysis, experimentally and computationally to understand the air distribution pattern with and without auxiliary draught.

3.2 EXPERIMENTAL SETUP:



(a) Line diagram of general cold room



(b) Air circulation pattern in cold room

Figure 1. Cold Store Basic Arrangement Line Diagram and Airflow Pattern in general configuration

Figure 1 represent basic parts of the cold store with general airflow pattern in chamber. Details of setup are as under:

Dimensions of the cold chamber (L x B x H) = (2.8194m x 2.5654m x 2.6678m)

Dimensions of the crates (l x b x h) = (.54m x .36m x .29m)

Weight of each crate: 1.52 kg/ per crate

Specific Heat of material of crates: 1.67 kJ/kg K

Agriculture product in use: Onion (Specific Heat C= 3.77kJ/kg K)

Quantity of onion in used: 300KG

Figure 2 is a line diagram of the cold chamber with auxiliary draught arrangement.

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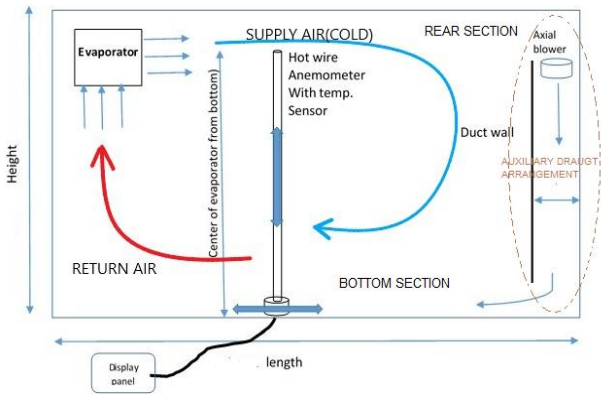


Figure 2. Schematic Diagram of Cold Room with Auxiliary Draught

Various arrangements of auxiliary draught are represented in figure 3.



(a) First Arrangement with fans in passive mode & slot closed.



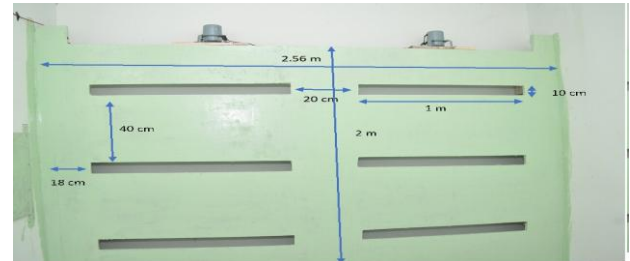
(b) Second Arrangement with fans in active mode & center slots uncovered



(c) Third Arrangement with fans in active mode double slot (M&D) uncovered



(d) Fourth Arrangement with fans in active mode all slots uncovered

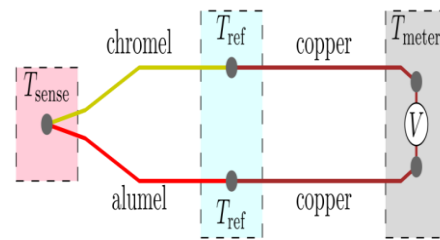


(e) dimension of parts of auxiliary draught

Figure 3. Auxiliary Draught arrangements

Instrumentations used:-

Thermocouples: - T type thermocouples use to measure the temperature on various points of the cold storage.



(a) T-type thermocouple



(b) Digital Temperature Indicator



(c) Selector Switch

Figure 4. Thermocouples Type, Temperature indicator & Selector Switch

Calibrated T-type thermocouple along with digital temperature indicator selector switch were used to measure temperature at various location in cold room.

Hot Wire Anemometer: HTC AVM 08 hot wire anemometer (shown fig. 5) used. It can measure air velocity in the range from 0.1 to 25 m/s with 0.01m/s resolution



Figure 5. HTC AVM -08 Hot Wire Anemometer

Fan size and Duct design calculations: -

Evaporator system rated flow capacity = 255CMH of Air.

If efficiency of the evaporator = 80%

Then airflow through evaporator = $0.80 \times 255 = 204\text{CMH}$

Average Airflow velocity at evaporator = 1.5 m/s

Average Airflow velocity at rear the section = 0.1m/s

Design of auxiliary draught system (ADS) with assumption that 70% of evaporator CMH will flow through it, as it will ensure cold air from evaporator cover the maximum length in the cold room and can reach up to rear section.

At inlet of ADS airflow capacity = 70% of 204CMH = 142.8CMH

Since the length of the ADS is fixed which is = 2.56m

Now for calculating the width of suction area, From discharge equation

$L \times W \times \text{Air velocity} = 142.8\text{CMH}$

$2.565 \times W \times 0.1 = 142.8/3600$

$W = 0.1549 \sim 0.2$

$W = 0.20\text{m}$

$W = 20\text{cm} \sim 8\text{inch}$

Inlet Cross-section of ADS = 2.56m X 0.20m.

As per evenly distribution of airflow through ADS two fans of 200CMH installed on top of ADS. Speed of fan regulated and operate at less than 50% of actual flow capacity, as a result 150 CMH of airflow in total provided. Slots are cuts on the

inner ADS wall to produce uniform air circulation with in the chamber. Inner ADS wall is kept 20cm above the bottom level as the pallet height 15.4cm.

Pallets: - A pallet is the structural foundation of a unit load that permits handling and storage efficiencies. Pallet of wooden type, 1.2m in length, and 1m in width and 0.0154m in height having regular gaps of 3inch for better air circulation in the cold chamber is shown in Figure 6.

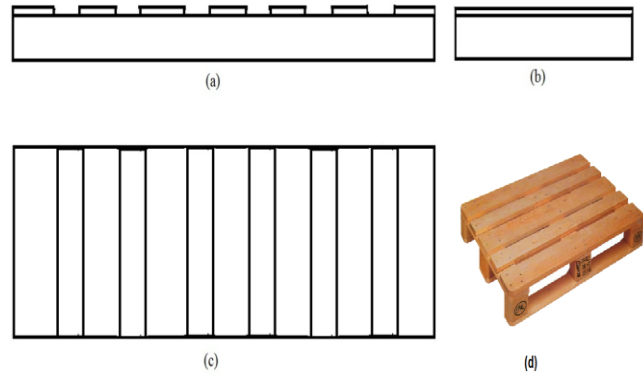


Figure 6. Different Views of a Pallet, 1.2m x 1m x 0.0154m (a) Front view, (b) Side View, and (c) Top View (d) Actual view of pallet

Stacking Arrangements: - In this arrangement side gap from walls of the cold chamber is 38cm, the gap between the columns of the stack is kept 10 cm.

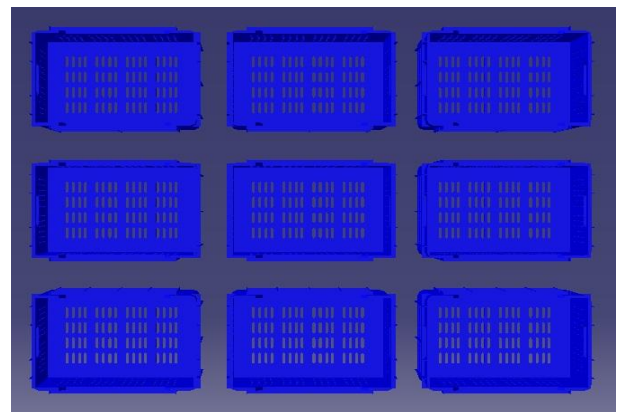


Figure 7 Arrangement of crates inside cold storage with Gap of 10cm, Top View



Figure 8. Actual Stacking Arrangements

3.3 Experimental Setup for Airflow pattern analysis:

A cold storage prototype of 1.5m (l) x 1m (w) x 1m (h), (at MANIT, Bhopal) (fig. 9 & 10) is erected with acrylic sheets of 3mm thickness and covered by insulating material to minimize the heat leak in the system. The model is used to demonstrate an analysis of the effects of the auxiliary draught on airflow velocity pattern inside the cold store. Air flow velocity measured with hot wire anemometer.

Small cold chamber is required for this research as there is no robust and economical means of airflow velocity measurement in large chamber. In large closed enclosure steady state airflow condition is impossible to achieve as a results value of flow velocity always fluctuated and no steady value found during measurement of flow velocity. Turbulence and obstacles in chamber create three-dimensional flow condition thus we do not have same velocity reading at same place even other parameters are constant. These difficulties lead to idea of small size chamber to analysis of flow pattern in the cold chamber.

Small cold storage is completely different in term of cooling capacity as mini cold store have 0.5 Tons of Cooling capacity with 137 CMH (2.28 m³/min.) of evaporator flow capacity. Mini cold store is geometrically and kinematically similar in term of air flow and turbulence hence we can say it is a reduce scale model of actual one. The geometrical and kinematic similarity is desirable between the model and prototype. The model is geometrically similar to the 1/2 reduction ratio as compared to the prototype. For kinematic similarity common range of Reynold's number for model and prototype is desirable. To calculate Reynold's number for prototype cross-section area for evaporator is 0.5m x 0.4m (0.2m²) (characteristic diameter = 4A/P) where average airflow velocity at the outlet of evaporator is 1.5m/s. Reynold's number for the prototype is calculated at the temperature of 25°C is $Re_p = 43664.09$. Likewise, for model evaporator cross-section is 0.5m x 0.16m and average flow velocity is 3 m/s. Reynold's number for the model at 25°C is calculated $Re_m = 46670.93$.

Reynold's number calculated for both has a difference of 6.85% while taking allowance of 10% for the model, Reynold's numbers are approximately in the same range. The test is conducted on a scaled model for the following three conditions:

- General configuration (without auxiliary arrangement)
- With auxiliary draught arrangement
- With auxiliary draught wall having equally spaced slots cut on it.

General configuration deals with only a closed chamber equipped with evaporator and measuring arrangements, no other auxiliary arrangements are placed inside the chamber. It is similar to the empty cold storage plant. In this case, flow velocity in the chamber was measured along the vertical central plane at different locations which were marked along the length and height of vertical planes as shown in fig. 10. A total of ninety-six (96) stations were selected for velocity measurement.

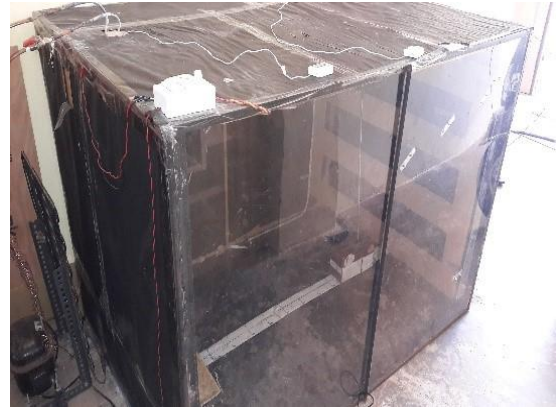


Figure 9. Actual photograph of the experimental setup

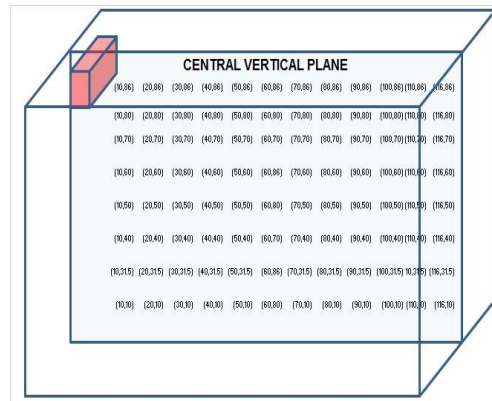


Figure 10. Location Map for observation

For second configuration testing induce draught system is placed at the rear section of the chamber, where reverse duct with two axial blowers, of flow capacity 0.57 m³/min per blower, were mounted. The duct wall is placed 16 cm away from the rear wall of the chamber (fig. 2) with a gap of 11cm from the top plane and 10 cm at the bottom plane. Two axial blower fans are placed in a gap to force air in a downward direction. These fans have a mass flow rate is 1.14 CMM in combination. These fans create pressure drop, as a result, induce airflow through the duct section.

The third experimental configuration consists of two equally spaced horizontal slots of 12 cm x 84 cm cuts on the duct wall along the width of it. Fans in the evaporator section run at their full combine rated capacity i.e. 2.28 m³/min.

Stacking

Size of carats:

- Length: 17cm
- Width: 12cm
- Depth : 10cm

Total no. of carats in chamber 48 in array of 3x4 with 4 carats in each column. Each carats have 1.3kg (approx.) of onion.

CFD Setup:

Various CFD model were used by researcher for cold store air distribution analysis, The SST-k- ω turbulence model is good agreement with experimental values with low computing time. As per open literature the SST k- ω model a more accurate model in comparison to the k- ϵ and k- ω models [23,26,31].

Based on the preceding comments and results, simulations were directly solved using the SST $k-\omega$ turbulence model. The SST- $k-\omega$ turbulence model is a two-equation eddy-viscosity model which was developed by (Menter 1994) [32]. In general, two-equation turbulence models allow the determination of the turbulent length and the time scale by solving two separate transport equations.

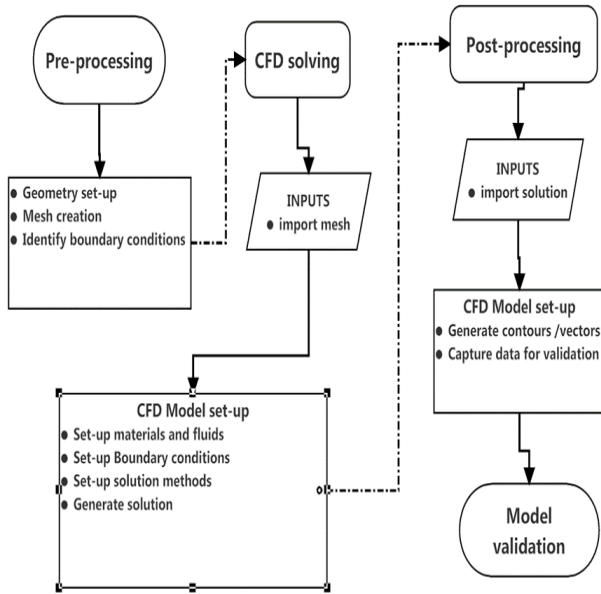
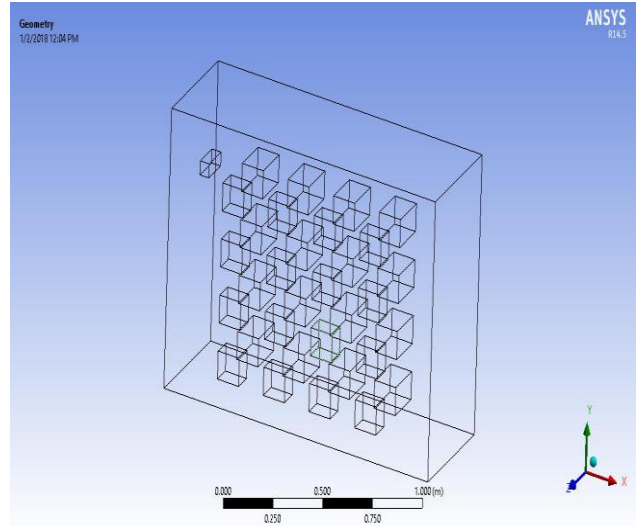
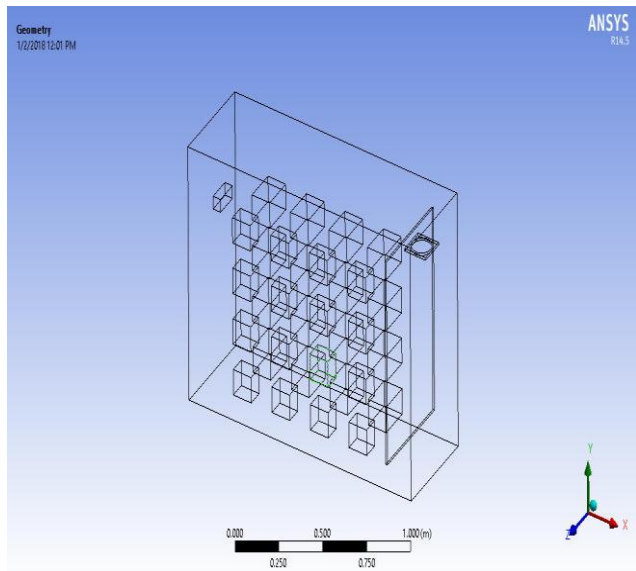


Figure 11. CFD Sample Algorithm



(a) Cold chamber with slotted auxiliary draught



(b) Cold chamber with plane auxiliary draught

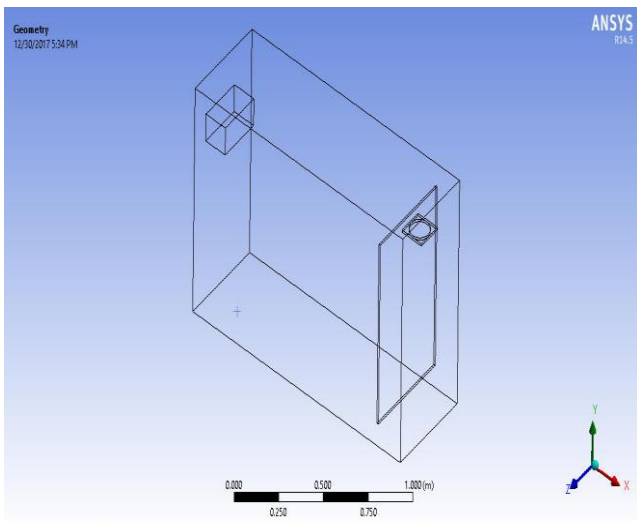
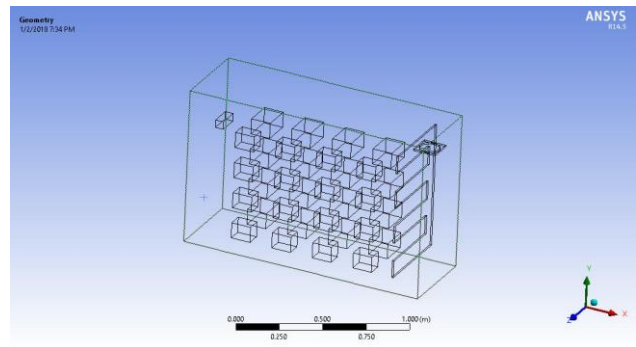


Figure 12. Geometry for Empty Cold storage



(c) Cold chamber with slotted auxiliary draught
Figure 13. Geometry for all Three Cases with loaded condition

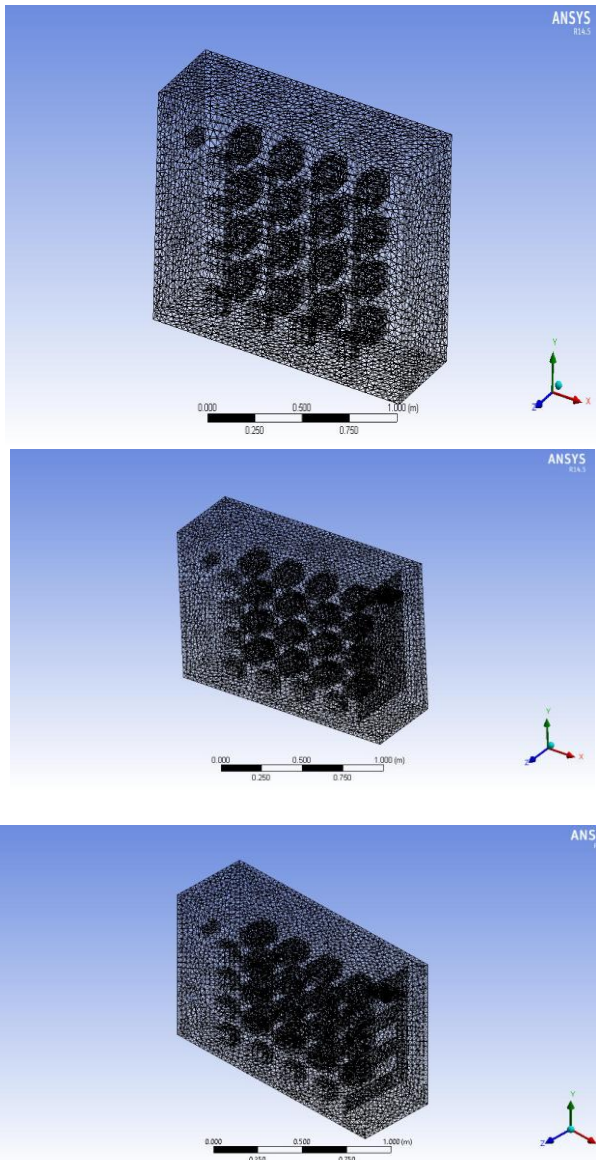
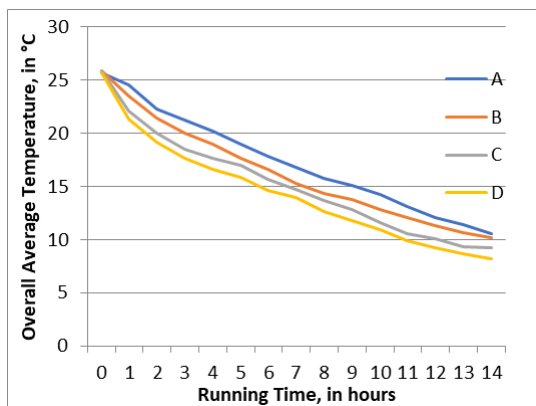


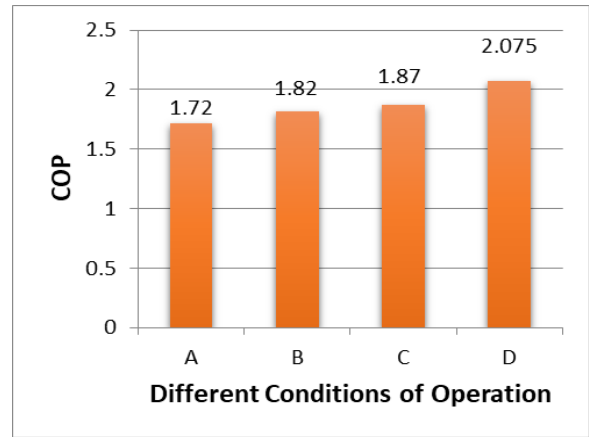
Figure 14. Meshing for all Three Cases

IV. RESULT & DISCUSSION

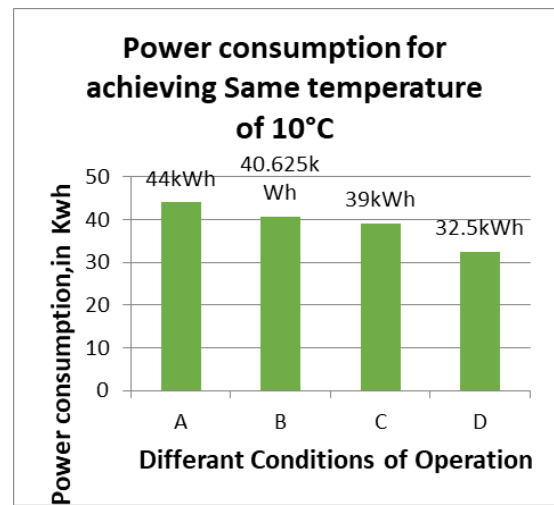
4.1 Results for First Set of Experimental setup:



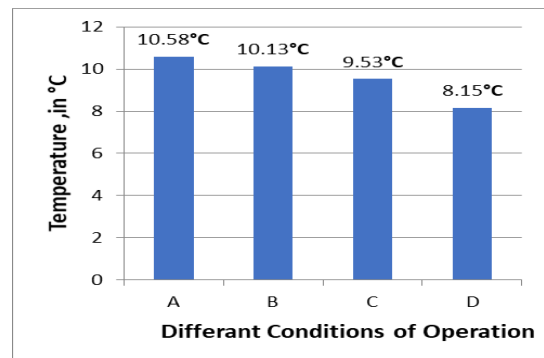
Comparison of Overall temperature drop rate of products in different Conditions



Comparison of COP in different Conditions



Comparison of Power Consumed in Different Conditions



Minimum Temperature Achieved in different Conditions for the same Power Input

- (A = First arrangement with fans in passive mode and without any slot open (without Duct)
- B = Second arrangement with and single (Middle) uncovered
- C = Third arrangement with fans in active mode and two slots (M&D) uncovered
- D = Fourth arrangement with fans in active mode and all slots uncovered)

Figure 15. Results graphs of the first setup

The above graphs in figure 15 represent the time, temperature COP and power consumption for various cases of the cold store room. Use of reverse duct with slots cuts on it is significantly improving the COP of the system by on an average 21% and the same time reduces cooldown period by 23%, as a result of that saving of energy by 26% achieved during experimentation.

The cold room is 25% cooler in the same time period of operation.

4.2 Results for Scaled cold Store: -

Airflow velocity measurements were conducted inside the model cold storage chamber for all three above stated configurations. The velocity profile plotted between airflow velocity and length of the room. Fig. 16 represents the velocity profile for general configuration. Evaporator fans run at their full capacity without any flow affecting medium in the chamber.

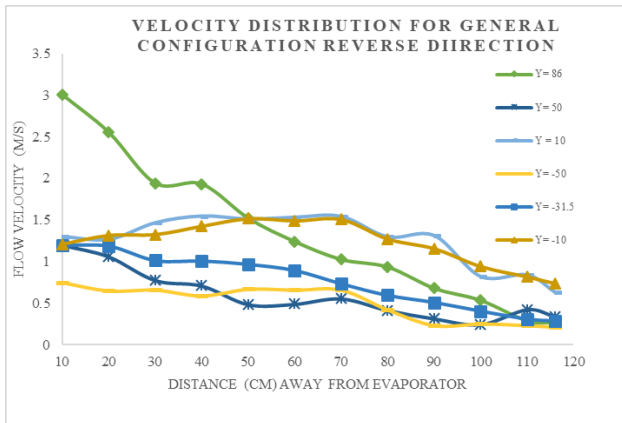


Figure 16. Flow velocity v/s Distance from evaporator graph; for the cold chamber in general configuration.

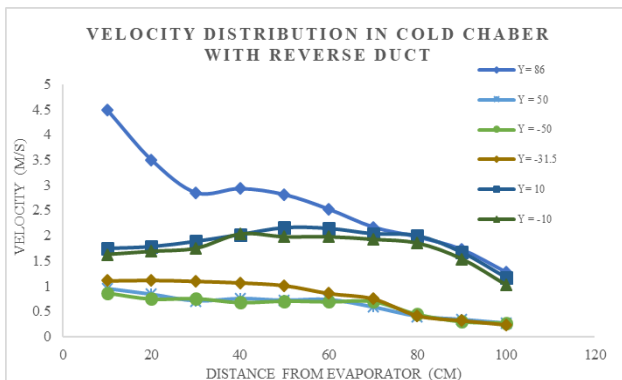


Figure 17. Flow velocity v/s Distance from evaporator graph; for the cold chamber with reverse duct

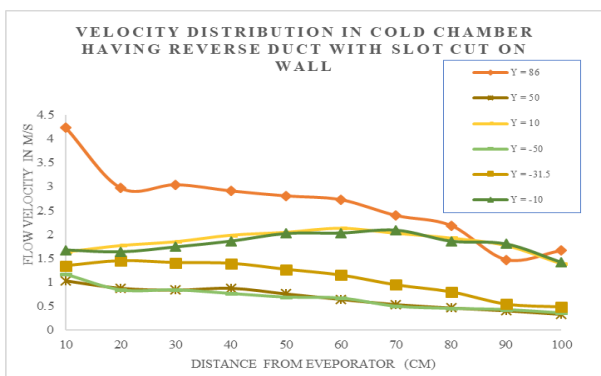


Figure 18. Flow velocity v/s Distance from evaporator graph; for the cold chamber with reverse duct and slot cuts on the duct wall

* Flow away from evaporator & flow toward evaporator (value with (-) sign shows flow toward evaporator)

Supply airflow velocity 10 cm in front of the evaporator is almost the same as that of the fan’s air thrust velocity i.e. 3m/s. Flow velocity continuously drops from the evaporator end to the rear part of the chamber, as supply airflow away from the evaporator section. Initially, the rate of velocity drop is quite high as just 50cm away from evaporator flow velocity remains half of the initial velocity. At 90cm away from evaporator coil and 86 cm above ground, velocity became almost 83% less than of the initial value. Supply air velocity at 50cm above ground recorded 1.3m/s near evaporator and 0.25m/s at the rear section of the chamber. The results indicate that supply airflow velocity is very low at the farthest part of the chamber, because of that poor mixing conditions are prevailing. Results indicate that airflow near the evaporator coils is highly turbulent and because of this good convective heat transfer rate is likely in this part of the chamber. The lowest supply air velocity was observed near the rear wall of the chamber at 10cm above ground was 0.27m/s. Return airflow through the evaporator’s bottom section as shown in fig.16. Circulation of air in the chamber is because the chamber is closed and there is no leak of the air neither outward nor inward to cold storage.

In Fig. 16 airflow velocity in reverse direction plots with legends having negative signs. Plots show that return airflow velocity pickup just below the evaporator section, as evaporator draw maximum airflow from its bottom and throw it axially in positive X distraction. Return air velocity at 10 cm in front of the evaporator, 31.5 cm above the base is 1.2m/s. and it is 1.3 m/s at 40 cm above the base. Measurements for general configuration clearly illustrate that there are certain areas in the chamber where airflow is not properly reached because of that heat transfer suffers in those sections.

It is clear from the above discussion that there is a need to build such arrangements that would be able to improve flow distribution in such a manner so that supply air can reach the farthest parts of the chamber. On the basis of the above discussion concept of auxiliary draught is introduced here as a solution. In fig. 16 and fig. 17 results were plotted which indicated the impact of induce draught on the cold storage chamber in terms of the velocity profile. Fig. 18 represent the effect of the duct on airflow pattern in the chamber, it indicated that there is significant rise in flow velocity inside the chamber because of induce draught arrangements. It reflects that because of auxiliary draught the reach of supply airflow improved in farthest parts of the chamber. The flow velocity was almost 50% at mid-way if induce draught was not used.

The graph plotted for velocity profile represent that at 86 cm above the ground and 110 cm in front of the evaporator, the flow velocity is improved form 0.5 m/s to 1.4 m/s. On the observation of velocity profile (fig.17) for return airflow when induce draught is in operation, it is found that return airflow velocity is 35 to 44% more as compared to a general configuration. Induce draught clearly improve flow at the top and bottom section as flow velocities were higher as compared to general configuration, but the central region is merely affected.

Auxiliary Draught Base Performance Improvement of Air Distribution System of Cold Store

Flow velocity in central zone i.e. at 40 cm in front of the duct and 40 mm above the base is marginally affected by reverse duct and values of velocity similar to in the first condition. To overcome this situation, slots were cuts on the duct wall as discussed in the experimental setup section and measurements were taken for the same, results are as indicates in fig.18. Observations made for the third configuration indicted, that flow velocity slightly drops at the top level of the room but overall flow velocity was improved in remaining parts of the chamber. Even though return airflow is also improved as compared to the first two cases. Velocities were recorded for third configuration, 9 to 16% higher as compared to the second configuration at mid-parts of the chamber. More ducting arrangements may be tested for a better solution.

Effects of induce draught on cold storage can better understand with following plots of instantaneous temperature and length of the room.

The impact of the reverse duct clearly reflects on the instantaneous temperature of the airflow in the chamber. After ten hours of operation, it has been observed that for duct arrangement temperature in the rear section of the chamber is higher by 1.4°C as compare to the general section. But a significant drop by 2°C is noted in the central part of the chamber for the third case. The bottom layer temperature is higher at just below the evaporator indicate more heat is carried by return air for the third case.

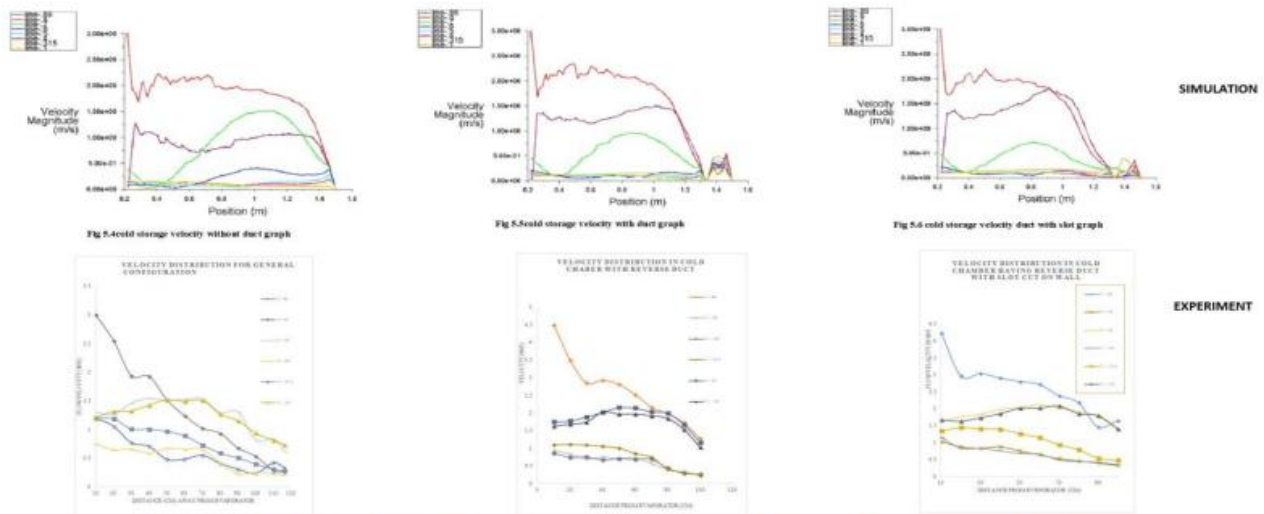


Figure 21. Airflow Velocity comparisons between simulation and experimental results

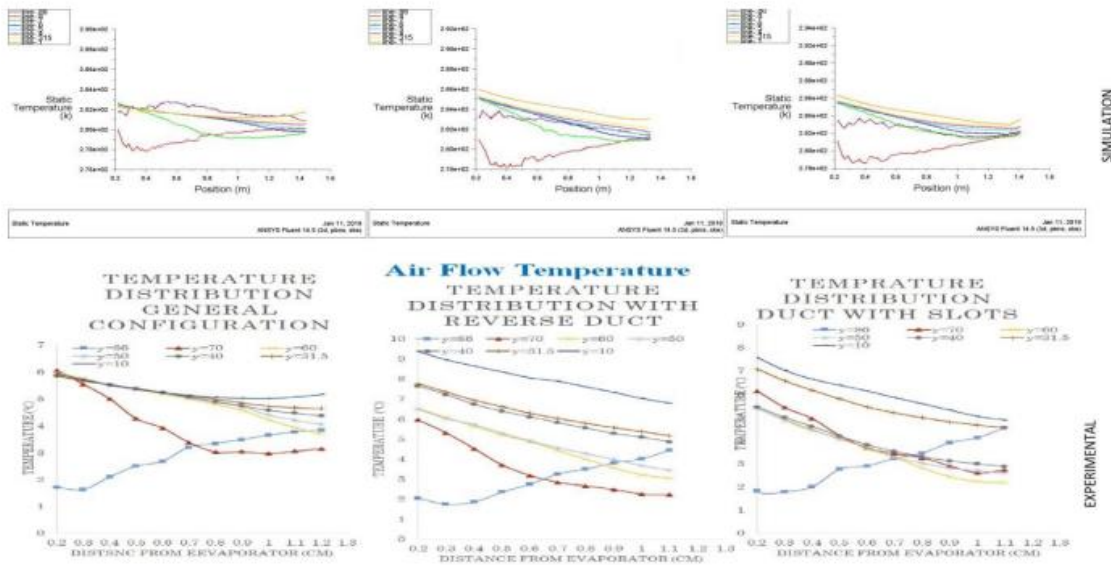


Figure 22. Airflow Temperature (Instantaneous) comparisons between simulation and experimental results.

4.3 Comparatives results with Computational Analysis: - Experimental results were compared with CFD results (fig 21& 22) and it has been observed that temperature and velocity profile have almost similar. There is a 23% error in velocity results as given by CFD and 14% error in the temperature profile. Both errors are considerable as per the

literature review suggested the same, it because of standard assumptions made during CFD analysis. M. L.

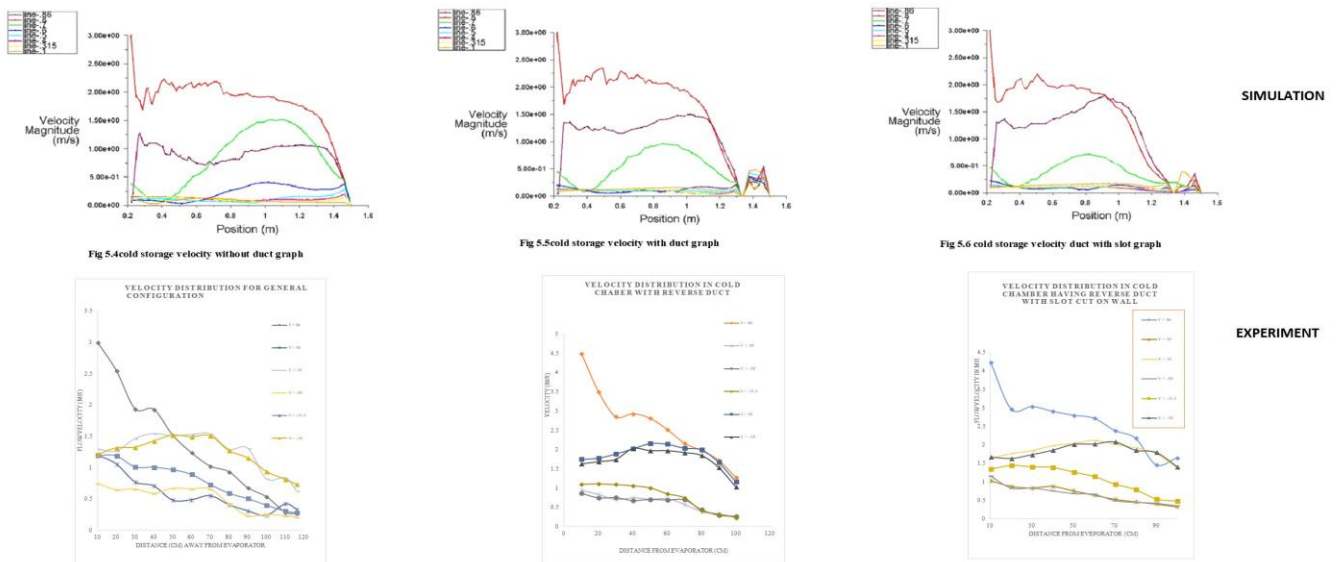


Figure 21. Airflow Velocity comparisons between simulation and experimental results

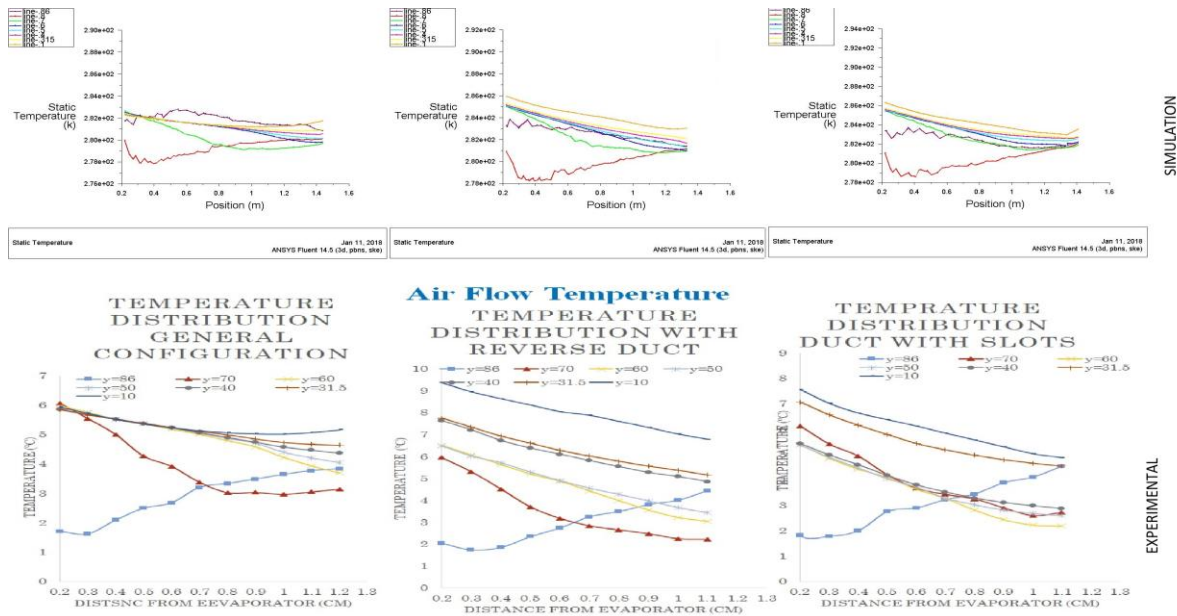


Figure 22. Airflow Temperature (Instantaneous) comparisons between simulation and experimental results.

Hoang et al., demonstrate a relative error of 26% for air flow velocity calculation thru SST k-ε model [12], 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 [33] and Moonne et al. found error 10% for the same [34]. 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 literature review. 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. Results suggest that reverse draught air movements in the chamber are cover all parts that ensure even temperature distribution. Uniform

temperature distribution ensures a reduction in frosting chance inside the chamber. Slots cut on auxiliary draught in the last case further improve velocity in the especially in mid-section of the cold chamber.

V. CONCLUSION:

Two different setups along with CFD analysis were the parts of investigation. Auxiliary draught arrangement with and without slots were investigate. Use of auxiliary draught significantly improve COP of the system by on an average 21%, same time reduce cooldown period by 23% as a result saving of energy by 26%. For same time period, room is 25% cooler. Auxiliary draught system raises the airflow velocity almost double at top section of chamber and 36% improved in bottom section of chamber.

Auxiliary Draught Base Performance Improvement of Air Distribution System of Cold Store

Mid-section was merely affected by auxiliary draught arrangements when slots were covered. Average product temperature is lower for the same time duration (i.e. 8.15^oC) as compared to general arrangements. Performance and airflow distribution improved with the auxiliary draught system in cold store.

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