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Air Flow and Temperature Field Analysis of a Cold Room

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***Abstract*:** Food is fuel for us by which we get energy. So it is very necessary to preserve that food. Cold room can hold anything at lower temperature. Cold chain is now recognized as a sunrise sector in India. The current scenario reveals that there is a tremendous scope for the development of cold chain facilities and Cold Rooms form the heart of the cold chain. The cooling rate and quality of food stuffs in a cold room are highly dependent on the temperature field which is closely related to flow field. Cold rooms of varying sizes are used to chill and store food. The air in these rooms is usually cooled by a heat exchanger which draws warm air with the aid of a fan which in turn circulates cool air. In this research paper a cold room is analyzed experimentally and by using Computational Fluid Dynamics (CFD). The simulation results reflected the characteristics of airflow and temperature distribution. After that, several design parameters which would affect the flow field in the cold store were analyzed. The results of calculation indicated that all these design parameters influenced the flow and temperature fields inside the cold store greatly.

Keywords: Cold room, Computational Fluid Dynamics, airflow, temperature distribution

I. INTRODUCTION

The refrigeration process is the main process of cold-room systems, which provide proper preservation temperatures to preserve the quality of products. The flow field design in cold store is always a puzzle. Information about the structure of the flow field in cold store can be obtained from measurements in experimental test facilities or from flow visualization studies. A uniform cooling and cold storage of fresh produce are difficult to obtain in industrial cooling rooms because of an uneven distribution of the airflow. The airflow distribution is dependent on the product, the cooling medium, and the geometry and characteristic of the cooling room. The velocity distribution can be determined based on the conservation equations for mass and momentum. An analytical solution can be found only in simple cases. The variables can be examined experimentally, but this is a tedious, costly, time-consuming method and furthermore, it is only applicable to existing storage rooms. Computational fluid dynamics (CFD) is a technique to model fluid flows using a computer simulation. The problem in cold store can be solved y solving the mathematical equations that govern the flow dynamics.

Although previous work on cold room was available, most of the work was focused on developing the models for various cold rooms, and validating the feasibility and definition of the models of cold rooms. The effects of design parameters in flow and temperature fields of cold room were rarely studied. Nowadays, flow structure in many cold rooms is far from rational or not good enough, which is always a technical problem for reducing the cost of energy and improving the uniformity of temperature. CFD technology is a useful method to get notable benefits with little cost.

In this study Ansys CFX is used as a tool to analyze the air flow field and temperature distribution within the cold room. On the basis of previous work, in the current study the K–ε turbulence model in a cold room is proposed. Based on this study, it will be decided whether current methods allow the food engineer to use CFD to optimize cold room designs. The main objective of this study was to develop a validated CFD model for predicting the flow and temperature fields in a cold store, and therefore to determine the factors affecting the uniformity of those two fields.

II. MATHEMATICAL MODEL ANALYSIS

Formulation of the problem:

Mini type constructional cold store was chosen as a study object. Its exterior dimension is 2.3286 mm (L) x 2.9032 mm (W) x 2.5610 mm (H). The cold fan in this study has the exterior dimensions of 1.75 m (l) · 0.46 m (w) · 0.5 m (h). The thickness of the wall of cold store is 125 mm, which is made up of polyurethane foam (PUF).

Governing equations:

Turbulence Model: - K- epsilon:

One of the most prominent turbulence models, the k-ε (k-epsilon) model, has been implemented in most general purpose CFD codes and is considered the industry standard model. It has proven to be stable and numerically robust and has a well established regime of predictive capability. For general purpose simulation, the model offers a good compromise in term of accuracy and robustness.

Within ANSYS-CFX, the k-ε turbulence uses the scalable wall function approach to improve robustness and accuracy when the near wall mesh is very fine. The scalable wall functions allow simulation on arbitrarily fine near wall grids, which is a significant improvement over standard wall functions. While standard two equation models provide good predictions for many flow of engineering interest k is the turbulence kinetic energy and is defined as the variance of the fluctuations in velocity. ε is the turbulence eddy dissipation (the rate at which the velocity fluctuations dissipate) and has dimensions of per unit time. The k-ε model introduces two new variables into the system of equations.

Where k is the turbulence kinetic energy and is defined as the variance of the fluctuations in velocity. It has dimensions of ($L^2 T^{-2}$); for example, m^2/s^2.

ε is the turbulence eddy dissipation (the rate at which the velocity fluctuations dissipate), as well as dimensions of k per unit time ($L^2 T^{-3}$) (e.g., m^2/s^3).

The k-ε model introduces two new variables into the system of equations.

The continuity Equation is then:

$$\frac{\partial \rho}{\partial t} + div\, U = 0$$

and the momentum equation becomes

$$\frac{\partial \rho U}{\partial t} + div(\rho U vi) - div(\mu_{eff}\, grad\, vi) = \frac{\partial P}{\partial xi} + S_i$$

$$\mu_{eff} = \mu + \mu_T$$

III. EXPERIMENTAL SETUP

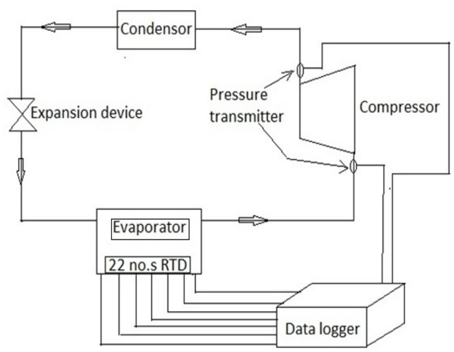

Fig. 1. Schematic diagram of experimental setup

In this work G-Tech Data logger is used to record the temperature and the pressure. G-Tech Data logger can sense 24 quantities at a time. In this experiment RTD is employed to measure temperature at different nodes within the cold room in channel 3 to 24 of data logger. The outlet velocity of the cold fan was measured by an air velocity testing anemometer. The cold store was pre-set for temperature range between -18 to -22 in order to achieve a basically steady condition.

Table 1. Position of RTDs within the cold room

RTD	X mm	Y mm	Z mm
3	Air out from evap.		
4	150	1280	500
5	1164	1451	2800
6	2170	380	2780
7	2170	1435	2780
8	200	2160	2750
9	200	770	2750
10	Evap. coil		
11	1164	100	1450
12	Air in to evap.		
13	1164	1400	1450
14	2148	2100	1450
15	2148	690	1450
16	100	1760	1450
17	100	280	1450
18	2078	1560	300
19	2078	450	300
20	125	1810	125
21	125	360	125
22	1164	1585	170
23	Refrigerant inlet to evap.		
24	Refrigerant outlet from evap.		

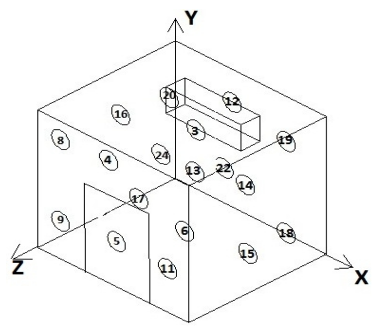

Fig. 2. Schematic diagram of RTDs in cold room

Fig. 3. RTDs within the cold room

Table 2. Observation table

TIME	TEMPERATURE																					CURRENT Amp
	AIR IN	OUT	AIR				COIL													REF. OUT	REF. IN	
	Ch 3 ° C	Ch 4 ° C	Ch5 °C	Ch 6 ° C	Ch 7 ° C	Ch 8 ° C	Ch 9 ° C	Ch 10 ° C	Ch11 °C	Ch 12 ° C	Ch 13 ° C	Ch14 °C	Ch15 °C	Ch 16 ° C	Ch17 ° C	Ch18 °C	Ch19 °C	Ch 20 °C	Ch21 ° C	Ch22 ° C	Ch 23 ° C	Ch24 ° C	
11:30	33.34	32.42	34.64	32.47	32.99	33.8	32.47	23.55	31.81	32.95	32.8	32.2	33.54	33.51	31.94	32.6	33.15	33.54	31.94	33.03	27.72	22.47	2.8
11:40	29.85	28.29	28.33	27.5	28.1	29.3	27.5	-11.41	26.8	29.7	28.1	27.38	29.81	29.25	26.55	27.94	28.77	28.98	26.94	25.98	-5.09	-17.28	2.8
11:50	18.69	18.55	19	18.52	19.04	19.91	18.52	-15.27	17.48	20.51	19.26	18.09	20.04	19.73	17.61	18.39	19.04	19.73	18.22	18.04	-8.96	-21.09	2.7
12:00	6.79	9.64	9.86	9.9	9.69	10.16	9.9	-8.88	8.91	10.21	10.29	9.08	10.51	10.81	9.17	9.64	9.77	10.47	9.9	9.69	-2.34	-15.4	2.7
12:10	-0.06	2.43	2	2.18	1.7	1.87	2.18	-2.04	1.87	2.25	2.91	1.83	3.12	3.08	2.22	2.74	2.43	2.91	3.08	2.78	-2.68	-8.1	2.8
12:25	-8.66	-6.81	-7.16	-7.11	-7.46	-7.46	-7.11	-10.42	-7.16	-6.94	-6.47	-7.24	-6.25	-6.3	-6.94	-6.56	-6.9	-6.56	-6.43	-6.43	-6.73	-13.98	2.7
12:35	-12.87	-11.32	-11.58	-11.6	-11.9	-11.8	-11.6	-14.71	-11.7	-11.5	-10.9	-11.62	-10.72	-10.89	-11.41	-10.98	-11.37	-11.11	-11.07	-10.89	-10.98	-17.67	2.8
12:45	-15.4	-14.07	-14.28	-14.3	-14.6	-14.6	-14.3	-17.28	-14.3	-14.2	-13.7	-14.33	-13.38	-13.6	-14.07	-13.68	-14.03	-13.86	-13.81	-13.6	-13.68	-19.85	2.9
12:55	-18.18	-16.98	-17.1	-17.2	-17.5	-17.5	-17.2	-20.03	-17.2	-17.1	-16.7	-17.23	-16.43	-16.6	-17.07	-16.73	-16.98	-16.8	-16.77	-16.47	-16.6	-22.16	2.9
13:05	-20.32	-19.21	-19.3	-19.4	-19.6	-19.6	-19.4	-21.99	-19.4	-19.2	-19	-19.38	-18.66	-18.78	-19.21	-18.87	-19.17	-19.04	-18.9	-18.6	-18.78	-24.05	2.9
13:15	-21.91	-20.88	-20.97	-21.1	-21.3	-21.4	-21.1	-23.49	-21.1	-20.9	-20.7	-21.09	-20.41	-20.5	-20.88	-20.58	-20.88	-20.75	-20.67	-20.32	-20.5	-25.37	3.1
13:25	-23.15	-22.12	-22.25	-22.3	-22.6	-22.6	-22.3	-24.6	-22.3	-22.1	-22	-22.38	-21.65	-21.7	-22.08	-21.8	-22.12	-21.99	-21.91	-21.57	-21.7	-26.39	3
13:35	-24.17	-23.19	-23.28	-23.4	-23.6	-23.6	-23.4	-25.63	-23.4	-23.2	-23.1	-23.4	-22.76	-22.7	-23.15	-22.93	-23.19	-23.1	-22.98	-22.63	-22.81	-27.38	3
13:45	-22.21	-21.05	-20.8	-21.7	-22.1	-21.8	-21.7	-23.6	-21.6	-21.4	-21.6	-21.78	-21.52	-21.01	-21.1	-21.61	-21.61	-21.05	-20.97	-20.88	-21.69	-25.88	3.2

IV. CALCULATIONS

Mass flow rate of air coming out of evaporator is measured by anemometer = 0.6 kg/s
For air Cp = 1.0005 kJ/kg
R_E = m Cp ΔT
= 0.6 x 1.0005 x (10.21-6.79)
=2.05 kW
(Considering ΔT is the maximum temperature difference of air within the readings)

W = √3 V I cosΦ
The refrigeration system is working on 415 V and power factor is taken 0.8
W= √3 x 415 x 3.2 x 0.8
=1840.13 Watt
=1.84 kW

COP_{Actual} = R_E / W
= 2.05/1.84
=1.11

Table 3. Comparison of COP

COP		
Carnot	Theoritical	Actual
3.8	1.8	1.11

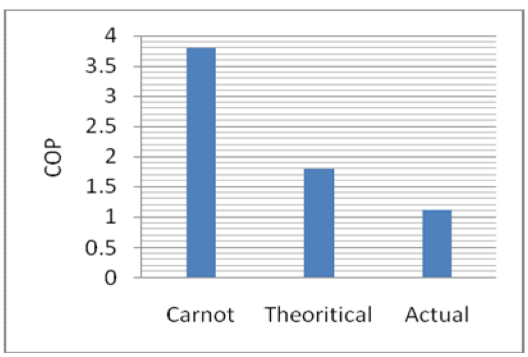

Fig. 4. Comparison of COP

V. CFD ANALYSIS OF COLD ROOM

Boundary conditions:

The air flow in the cold store was assumed to be closed circular flow field driven by a cold fan. The turbulent flows were significantly affected by the presence of walls. In order to make the numerical simulation results approach the practice closely, the heating load was calculated carefully with general heating load calculation method of cold room.

- The air flow in the cold store was assumed to be closed circular flow field driven by a cold fan.
- The fans in the evaporating unit is considered as rotating wall and H.E. is considered as a porous wall of porosity 0.8
- Angular velocity of fan = 1350 RPM

Inlet conditions:

1. K, assumed as 0.5–1.5% mean kinetic energy of coming flow.
2. Initial velocity and temperature were determined by experimental results.

CFD simulation

After discretising the governing equations with the control-volume method, the differential equations that can be solved by computer were obtained. The calculation region was meshed with proportional spacing. Analysis of temperature distribution and air flow is done in Ansys CFX.

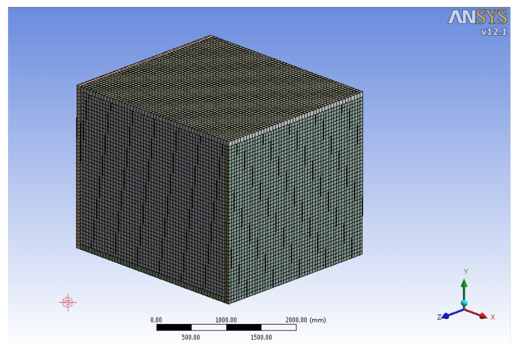

Fig. 5. Meshing of Cold Room

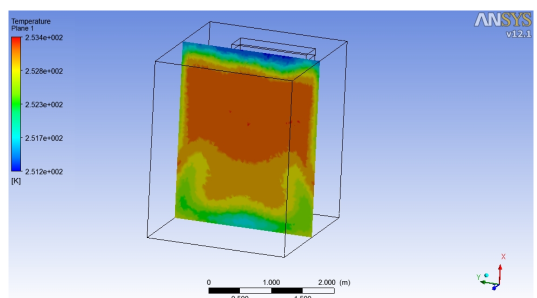

Fig. 6. Temperature Contour

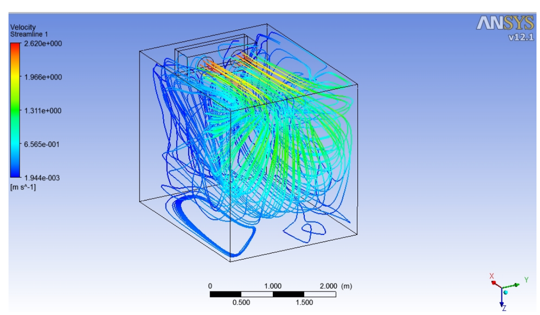

Fig. 7. Velocity Contour

Table 4. Observation Table

Sr. No.	TIME	Temperature							
		Ch 3 °C		Ch 4 °C		Ch 5 °C		Ch 6 °C	
		Exp.	CFD	Exp.	CFD	Exp.	CFD	Exp.	CFD
1	11:30	33.34	33.31	32.42	31.54	34.64	33.52	32.47	31.87
2	11:45	26.2	25.89	24.07	23.89	24.07	23.89	23.38	22.84
3	12:00	6.79	5.78	9.64	8.97	9.86	9.12	9.9	8.89
4	12:15	-2.77	-3.15	-0.62	-0.74	-1.05	-2.012	-1	-1.87
5	12:30	-10.89	-12.54	-9.22	-10.21	-9.52	-10.57	-9.52	-10.87
6	12:45	-15.4	-16.2	-14.07	-15.21	-14.28	-15.84	-14.33	-15.27
7	13:00	-19.34	-20.18	-18.18	-19.84	-18.3	-19.84	-18.36	-19.42
8	13:15	-21.91	-22.04	-20.88	-21.48	-20.97	-21.54	-21.09	-21.57
9	13:30	-23.66	-22.35	-22.68	-23.87	-22.81	-23.14	-22.85	-23.84
10	13:45	-22.21	-22.15	-21.05	-22.84	-20.84	-21.87	-21.74	-22.15
		Ch 7 °C		Ch 8 °C		Ch 9 °C		Ch 10 °C	
1	11:30	32.99	31.87	33.86	32.71	32.47	31.87	23.55	22.87
2	11:45	24.12	23.87	25.38	24.87	23.38	22.17	-18.53	-19.87
3	12:00	9.69	8.74	10.16	9.82	9.9	9.85	-8.88	-9.82
4	12:15	-1.39	-2.14	-1.31	-2.14	-1	-1.57	-4.01	-5.28
5	12:30	-9.77	-10.57	-9.77	-10.47	-9.52	-10.24	-12.83	-13.98
6	12:45	-14.63	-15.98	-14.63	-15.27	-14.33	-15.87	-17.28	-18.97
7	13:00	-18.61	-19.87	-18.61	-19.82	-18.36	-19.24	-21.09	-22.05
8	13:15	-21.27	-21.87	-21.35	-22.21	-21.09	-21.87	-23.49	-24.89
9	13:30	-23.06	-23.87	-23.1	-23.4	-22.85	-23.54	-25.16	-26.82
10	13:45	-22.12	-23.17	-21.82	-22.01	-21.74	-22.58	-23.66	-24.18
		Ch 11 °C		Ch 12 °C		Ch 13 °C		Ch 14 °C	
1	11:30	31.81	30.89	32.95	31.87	32.86	32.16	32.2	31.15
2	11:45	22.42	21.87	26.38	25.91	23.94	22.18	23.07	22.18
3	12:00	8.91	7.52	10.21	9.82	10.29	9.82	9.08	8.95
4	12:15	-0.96	-1.21	-0.75	-0.85	-0.23	-0.31	-1.18	-1.82
5	12:30	-9.52	-10.27	-9.31	-10.52	-8.83	-9.82	-9.56	-10.24

6	12:45	-14.33	-15.87	-14.16	-15.86	-13.73	-14.89	-14.33	-15.32
7	13:00	-18.36	-19.87	-18.18	-19.82	-17.97	-18.42	-18.36	-19.87
8	13:15	-21.05	-22.08	-20.88	-21.52	-20.71	-20.85	-21.09	-22.01
9	13:30	-22.85	-22.98	-22.68	-23.51	-22.51	-23.81	-22.85	-23.58
10	13:45	-21.61	-21.87	-21.39	-22.89	-21.57	-22.57	-21.78	-22.8
		Ch 15 °C		Ch 16 °C		Ch 17 °C		Ch 18 °C	
1	11:30	33.54	32.41	33.51	32.41	31.94	30.14	32.6	31.18
2	11:45	25.64	24.41	25.2	24.89	22.34	21.81	23.73	22.18
3	12:00	10.51	10.24	10.81	9.82	9.17	8.52	9.64	8.82
4	12:15	-0.06	-0.8	-0.06	-0.2	-0.83	-0.91	-0.4	-0.8
5	12:30	-8.66	-9.52	-8.69	-9.82	-9.31	-10.21	-9.01	-10.52
6	12:45	-13.38	-14.24	-13.64	-14.28	-14.07	-15.23	-13.68	-14.58
7	13:00	-17.63	-18.81	-17.76	-18.23	-18.14	-19.52	-17.84	-18.35
8	13:15	-20.41	-20.81	-20.5	-20.81	-20.88	-21.87	-20.58	-21.96
9	13:30	-22.16	-22.83	-22.25	-22.81	-22.59	-22.38	-22.42	-23.21
10	13:45	-21.52	-21.81	-21.01	-21.09	-21.14	-22.15	-21.61	-22.13
		Ch 19 °C		Ch 20 °C		Ch 21 °C		Ch 22 °C	
1	11:30	33.15	32.18	33.54	32.81	31.94	30.21	33.03	32.81
2	11:45	24.42	23.18	24.98	22.81	22.73	21.87	21.99	20.47
3	12:00	9.77	8.97	10.47	9.82	9.9	9.1	9.69	8.51
4	12:15	-0.7	-0.8	-0.23	-0.4	-0.14	-0.18	-0.19	-0.21
5	12:30	-9.22	-10.82	-8.96	-9.81	-8.83	-9.81	-8.69	-9.81
6	12:45	-14.03	-15.97	-13.86	-14.23	-13.81	-14.52	-13.6	-14.53
7	13:00	-18.14	-19.64	-18.01	-19.05	-17.93	-18.92	-17.63	-18.98
8	13:15	-20.88	-21.97	-20.75	-21.09	-20.67	-21.41	-20.32	-21.58
9	13:30	-22.68	-23.51	-22.59	-22.98	-22.46	-23.57	-22.12	-23.51
10	13:45	-21.61	-22.28	-21.05	-21.81	-20.97	-21.45	-20.88	-21.82

In cold room the maximum temperature is observed at RTD channel 13 and minimum temperature is observed at RTD channel 3. The temperature distribution by experiment is obtained. The comparison between experimental and CFD results is shown in figure 9.3 and fig. 9.4. Within the acceptable error range in simulation, the distribution of temperature in both cases is quite consistent, which indicated that the model established in this study is acceptable and reliable.

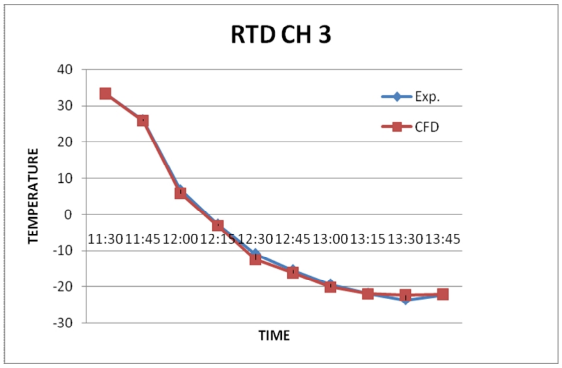

Fig. 8. Comparison of CFD Results and Experimental Results at RTD 3

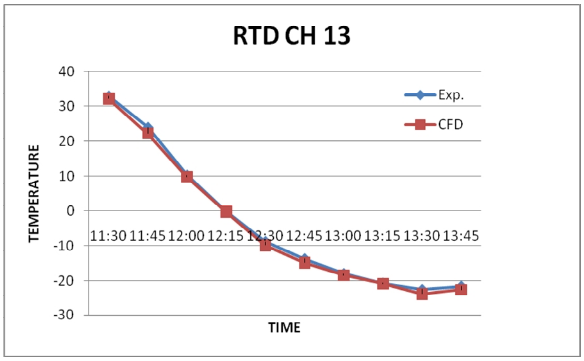

Fig. 9. Comparison of CFD Results and Experimental Results at RTD 13

VI. CONCLUSION

A cold room as per customer requirement is designed. The cold room has estimated total refrigeration capacity of 2kW. Its operating ambient temperature is 43°C with evaporator capacity of 3.08 kW, condenser capacity of 3.24 kW and compressor capacity of 1.16 kW. The theoretical COP of the system is calculated which is 1.8 and Carnot COP is 3.3 and the experimental COP is 1.11. Carnot COP is always higher than Actual COP. By this study, the efficient system can be designed for various purposes. Factors which are affected the actual performance include superheating and sub cooling of vapour, throttling process in expansion valve, and heat transfer during compression. The airflow structure of cold store was simulated by using CFD results were validating with the experimental results.

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