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INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING
WWW.IJRAME.COM
ISSN (ONLINE): 2321-3051

Vol.13 Issue 1, January 2025
Pg:- 51 – 66

Development of miniature forced convection evaporator unit for transporting temperature sensitive goods

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Abstract *- This paper highlights the development of a forced convection Miniature Evaporator unit for an intelligent portable and battery-powered medical carrier device. The device is being used to store and transport temperature-sensitive medical goods while preserving their potency by maintaining required temperature using a miniature vapour compression-based refrigeration system. In order to provide uniformity of cooling inside the cargo container and faster cooldown time, a forced air convection-based system was considered to be better than the conventional tube type evaporator like those used in refrigerators. Hence, a similar system was developed and its efficiency improved by varying the fan airflow, the number of fins on the evaporator, and the shape of ducts. Overall, uniformity of temperature inside the cargo chamber improved and cool down time reduced significantly with some increase in energy consumption.*

Key Words: *Miniature refrigeration, duct-based, airflow, evaporator design, medical device, forced convection.*

1. INTRODUCTION

With advancements in medical research and an increase in innovation, there has been a steady rise in need for stricter temperature control of medical interventions to ensure their efficacy. Recent examples are the new RNA based Covid 19 vaccines and Covid 19 PCR test kits. Similarly, the need for tissue samples in genetic testing and treatments, donated organ transportation, blood and blood products requiring continuous temperature maintenance has been steadily growing. This has led to a rise in the need for portable carriers that can accurately maintain required temperature in spite of the dynamic ambient conditions during transit. Conventional solutions involve passive cooling with the usage of simple frozen ice/gel packs or a bit more advanced frozen packs with phase-change materials (PCMs) in insulated boxes. However, these carriers, often without any monitoring, face problems with continuous temperature adherence when subject to the vagaries of the environment. They also suffer from human factors related to improper freezing, conditioning and packing processes.

Active Carrier Devices (ACDs) that use active cooling methods represent the latest innovation in addressing the shortcomings of passive solutions. These devices only need charging of their batteries and are easy to operate without any other preparation. For this paper, we refer to an active carrier with a 15-liter capacity, operating at 40 watts using vapour compression refrigeration system, and featuring remote monitoring and precise temperature control. Vapour compression as a cooling mechanism is far more efficient than competing thermoelectric (Peltier) systems. Peltier module based systems can only support smaller volumes of less than 2 liters and have a Coefficient of Performance (COP) less than 0.7 [5]. Hence, using them for larger capacities like 15L is not practical. Our ACD using vapour compression has a COP of 1.1, as shown in figure 14 (the RPM of the compressor was 3900 and an optimum state was achieved by following the ASHRAE test conditions). Other technologies, such as vapour adsorption refrigeration, are still under development for small-scale applications.

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Vol.13 Issue 1,
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Pg:- 51 – 66

Vapour compression uses a sealed system with refrigerant driven by a compressor, a condenser and a fan to expel heat to the external environment and an evaporator that absorbs heat from the cargo payload [6](See 'Figure 1' for reference). Typical evaporators consist of conductive tubing to aid heat absorption. These evaporators suffer from the drawback of non-uniform cooling resulting in hot and cold spots in the cargo compartment. Areas closer to the tubing are typically much colder than other areas inside the cargo compartment. This causes problems for transportation of medical items since they require a narrower temperature range to preserve potency. Hence, a method of forced air convection using ducts that holds the potential for rapid and uniform cooling is explored in this paper.

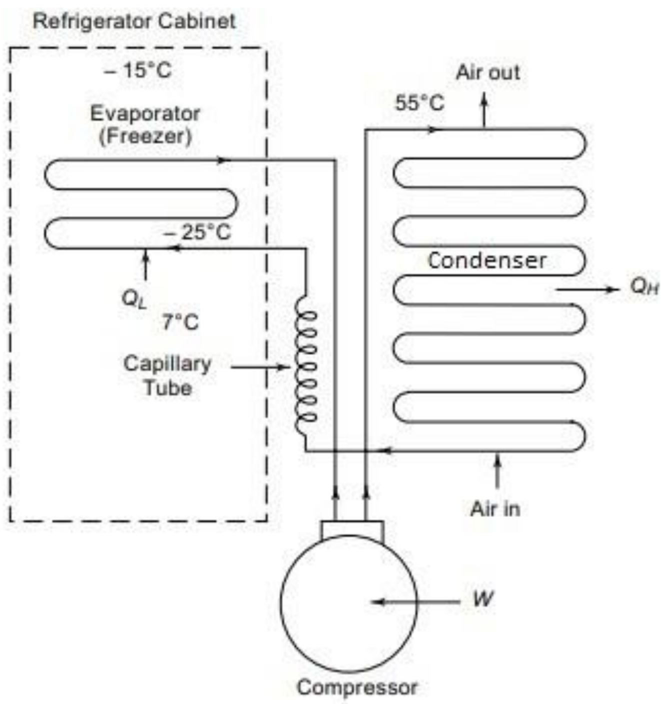

Figure 1: Schematic diagram of a domestic refrigerator

2. ENGINEERING CHANGE

The main components of the ACD's refrigeration system includes a small reciprocating or rotary compressor using R134a refrigerant, a wire-and-tube type condenser with a DC fan for heat removal, a tube type evaporator to cool the cargo, and a capillary tube to create the pressure and temperature differences. All these components were designed to be compact and energy efficient. The evaporator system relied on natural convection to cool the 15-liter cargo chamber. The evaporator was designed as copper tubing wrapped around a plastic cargo container box which was covered with polyurethane insulation around it. The insulation helped to avoid the loss of cooling to the environment. The device was most commonly used to store goods like vaccines and blood samples which were kept within a temperature range of 2–8°C. Since the R134a refrigerant has a boiling point of about -24°C, the system can cool to as low as -20°C in some conditions. To maintain the required 2-8°C range, a PLC control system operates the compressor with on-off cycles around 5°C temperature.

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Vol.13 Issue 1, January 2025
Pg:- 51 – 66

In spite of such accurate control, items near the evaporator were getting damaged due to lack of uniformity of cooling. The natural convection currents were not strong enough to maintain uniform cooling and the temperature of goods near the evaporator was dropping below 2°C in certain conditions. This issue was common in older generation domestic refrigerators which relied on natural convection. To address this, domestic refrigerators started using forced-air cooling and branded as 'Frost-free refrigerators'. Taking inspiration from this change in domestic refrigerators, it was planned to add forced convection airflow system to the ACD.

Designing refrigeration systems requires controlling numerous factors, and relying solely on theoretical calculations or simulations often does not yield accurate results. To create a market feasible system, practical experiments supported by calculations, are commonly used in the industry. A similar approach was used in developing the prototype for forced convection refrigeration system. As the cooling capacity and the required temperature range was same, the refrigerant (R134a), compressor, condenser and the capillary of ACD were left unchanged. The focus was on optimizing the evaporator, fan, and air ducts. The followings section details out the development stages for the prototype.Note that the target cargo temperature was set to 2-8°C temperature since that was the most commonly used by customers. Testing and tuning of the system for other temperature ranges is not in the scope of this paper.

3. PROTOTYPE DEVELOPMENT

The prototype was developed to be similar to the ACD. Like the ACD, the cargo chamber is made from 1.5 mm thick HDPE (High-Density Polyethylene) plastic.To prevent cooling loss, strong insulation was required around the cargo chamber. ACD and domestic refrigerators typically use polyurethane foam which has very low thermal conductivity (0.022-0.028 W/m.k) [7].

Another option was to use 'Polystyrene foam'. It had better workability which is ideal for prototyping. However, the thermal conductivity of polystyrene (0.034-0.038 W/m.k) was less than that of polyurethane foam insulation [7]. After careful consideration, workability was given more weightage and polystyrene sheet was chosen for the prototype. The polystyrene insulation sheets were attached to the cargo chamber using silicone sealant. The inner dimensions of the cargo chamber were L × W × H: 350 × 245 × 175 mm.

The condensing unit, consisting of the compressor and condenser, was mounted on a stainless steel plate beside the chamber. The evaporator and a fan for airflow were mounted above the condensing unit, outside the cargo chamber wall. A miniature fin-and-tube evaporator was used because it provided a larger surface area and better heat transfer than the commonly used wire-and-tube evaporator.The evaporator was also insulated with polystyrene to minimize heat dissipation to the surroundings (see figure 2 and 3).

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Pg:- 51 – 66

Figure 2: Cargo chamber Layout

Figure 3: Complete Prototype

In frost-free domestic refrigerators, a fan is placed near to the evaporator. The fan pushes the hot air over the evaporator which takes away the heat and supplies the cool air to the frozen food storage chamber [3]. The cool air is then forced through ducts to the fresh food storage which is located below the frozen food storage, natural convection also aides in airflow as the cooler air settles below. The hot air then moves through the ducts back to evaporator and the fan completed

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Pg:- 51 – 66

the air circulation (see figure 4). This forced air circulation helps distribute the cooling more evenly between the different chambers.

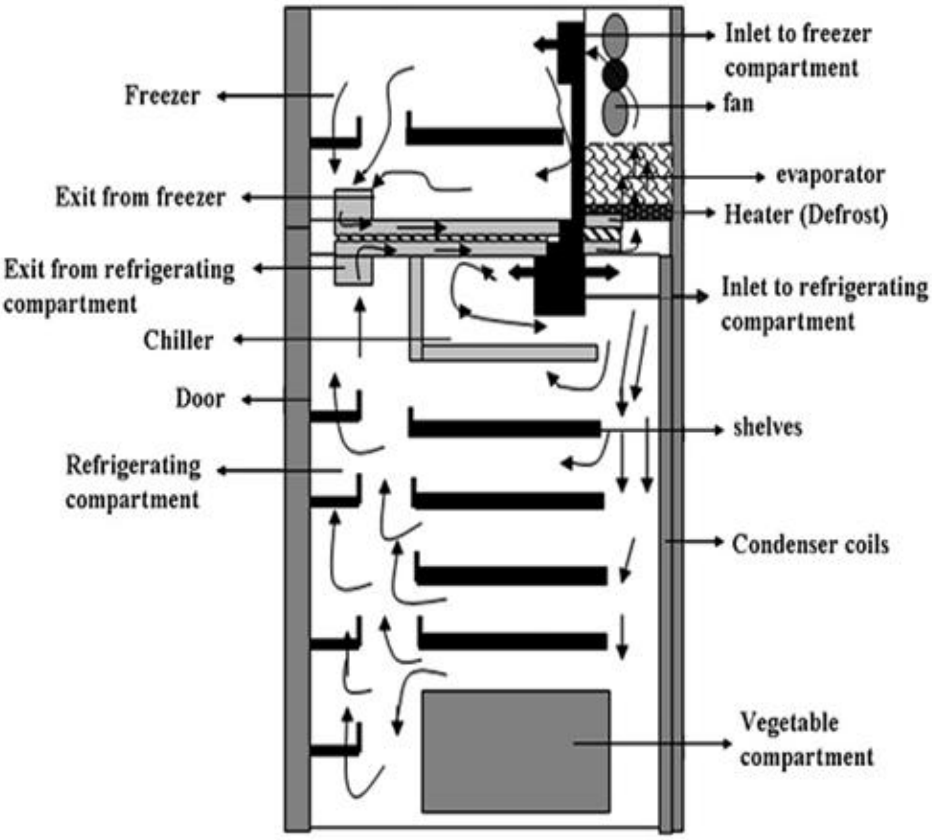

Figure 4: Air-circulation in single door frost free refrigerators

Taking reference from domestic refrigerators, the cool air was supplied to the cold chamber from the top of the side face, allowing it to settle downward naturally, while warm air was drawn from the bottom on the same side (see figure 5). An opening was provided behind the evaporator fan to allow airflow between the fan and the evaporator. The isolated refrigeration system with the radial blower setup is shown in the figure 6.

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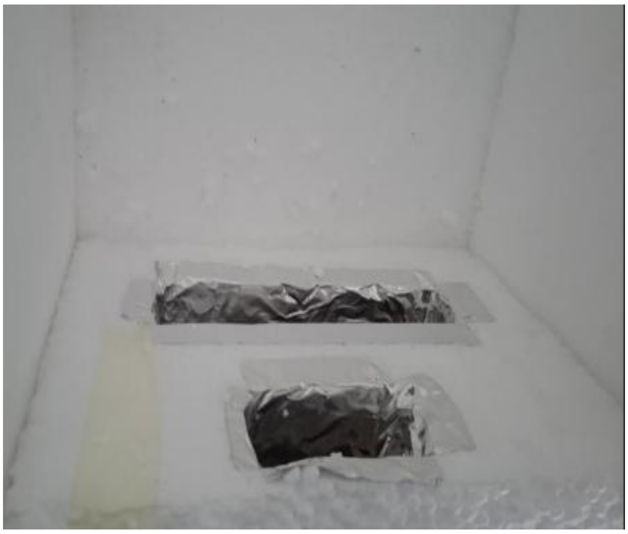

Figure 5: Cargo chamber air outlets

Figure 6: Refrigeration system

4. CHOICE OF FAN

Fan is the critical part of the forced air convection system. After literature review, the choice of fan was narrowed down to an axial fan or a radial blower [4] (see figure 7 and 8). Due to its construction, a radial blower provides more pressure head than the axial fan, but the axial fan moves more air (cubic feet per minute) than the radial one[6]. The radial blower is usually employed in industrial air conditioning systems. For domestic refrigerators, axial fan is the most commonly used

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Vol.13 Issue 1,
January 2025
Pg:- 51 – 66

fan. The size and length of the ducts in domestic refrigerators is small and the pressure drop is less compared to the industrial air conditioning system. So, the axial fan setup was expected to perform better than the radial one for the prototype. However, due to the novelty and miniaturisation of the ducts, it was decided to conduct experiments with both the fans to determine the performance of the prototype. Based on the results, a comparison was made to choose the better performing fan.

Figure 7: DC axial Fan

Figure 8: DC radial blower

The compressor and condenser fan operated on a 12V DC power supply, making it practical to use a 12V DC fan for the evaporator. Both the axial fan and radial blower selected for testing were compatible with this power supply. The specifications for the radial blower were as follows: outer dimensions (L × W × H): 52 × 52 × 15 mm, RPM: 7000, and airflow: 10 CFM (cubic feet per minute). The axial fan specifications were: outer dimensions: 92 × 92 × 25 mm, RPM: 2400, and airflow: 20 CFM. The radial blower evaporator assembly is shown in figure 9 and 10.

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Pg:- 51 – 66

Figure 9: Radial Blower

Figure 10: Radial blower type Evaporator Assembly

The cooling performance of the two fans was evaluated by conducting cool-down experiments. During the tests, various temperatures inside the cargo chamber were recorded as the refrigeration system operated. The tests were continued until the cargo chamber temperature reached a stable state.

After analysing the experimental results, it was determined that the axial fan setup was more efficient. It achieved faster cooling and better temperature uniformity compared to the radial blower setup. Additionally, the airspeed required for the small refrigerator (15-liter capacity) was relatively low, and it was better to move a higher volume of air (CFM) using an axial fan rather than a small, high-pressure stream of air as in the case of the radial blower. The radial blower also likely did not adequately cover the entire surface area of the evaporator. The performance and temperature zones of the optimized axial fan and evaporator combination are illustrated in Figures 15 and 16. The supply airflow measurement for this setup, as shown in Figure 11, was recorded at 1.0 m/s.

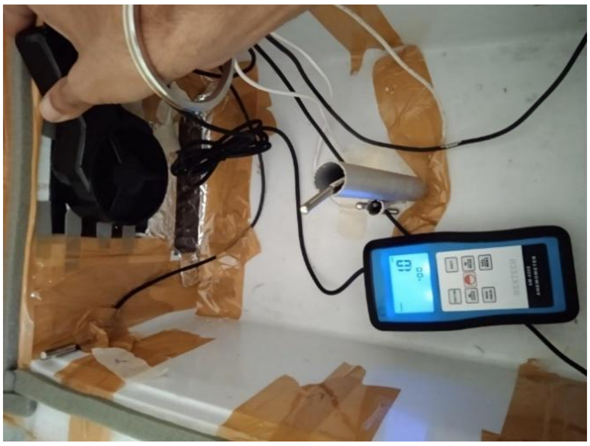

Figure 11: DC fan airflow measurement

5. EVAPORATOR TUNING

Further, tests were conducted on the prototype with the axial fan to measure the refrigeration system's temperature points. These temperature points were return gas temperature (temperature of the inlet copper pipe of the compressor), discharge temperature and liquid sub-cooled temperature (condenser exit temperature). These temperature values along with the cool down performance are used by refrigeration engineers for tuning the system for optimum health and performance. The common rule of the thumb is to keep these values within 5°C degrees of the recommended ones. The reference for the temperature points were taken from the ASHRAE Standard Refrigeration tests results shared by the compressor manufacturer (see figure 12).

Running Speed	Hz	75	60	45	
	RPM	4,500	3,600	2,700	
Cooling Capacity	W	49	33	27	±10%
Power Consumption	W	41.3	32.7	30.7	-
COP	W/W	1.19	1.01	0.88	±7%

Test Condition : ASHRAE LBP

Evaporating Temperature	-23.3℃
Condensing Temperature	54.4℃
Ambient Temperature	32℃
Return Gas Temperature	32℃
Liquid Sub cooled Temperature	32℃

Figure 12: ASHRAE LBP test for the compressor

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Pg:- 51 – 66

After initial cool down experiments on the prototype, the temperature gradient across the evaporator was found to be from -25°C to the ambient temperature over the initial part of the evaporator itself. Also, the return gas temperature into the compressor was the same as the ambient temperature. As per conventional practices, the return gas temperature is recommended to be kept as close to the ambient temperature as possible, but based on empirical testing during the development of the ACD, we determined that it needs to be atleast 5°C lesser than the ambient temperature.

From these two observations, it was deduced that the evaporator was slightly oversized and required a reduction in the surface area to reduce the thermal heat rejection.In order to reduce the surface area of the evaporator, the best bet was to remove few aluminium fins from the copper pipe. The fins were made from aluminium sheets press fitted onto the copper pipes of the evaporator. These fins have a very small thickness of 0.4mm and it was easier to manually remove them without damaging the copper pipes. The length of the copper pipe used was 1.2 metre and the covering body was made out of Stainless Steel 304 material (see figures 13 and 14). After reducing the number of fins from 60 to 40, the return gas temperature became 4-5 °C lower than the ambient temperature and also improved the cool down performance. Figure 15 demonstrates the positions at which temperature probes were kept inside the cold chamber and figure16 shows their respective temperatures values. The cool down performance results and refrigeration system's temperatures points are also shown in figure 16.

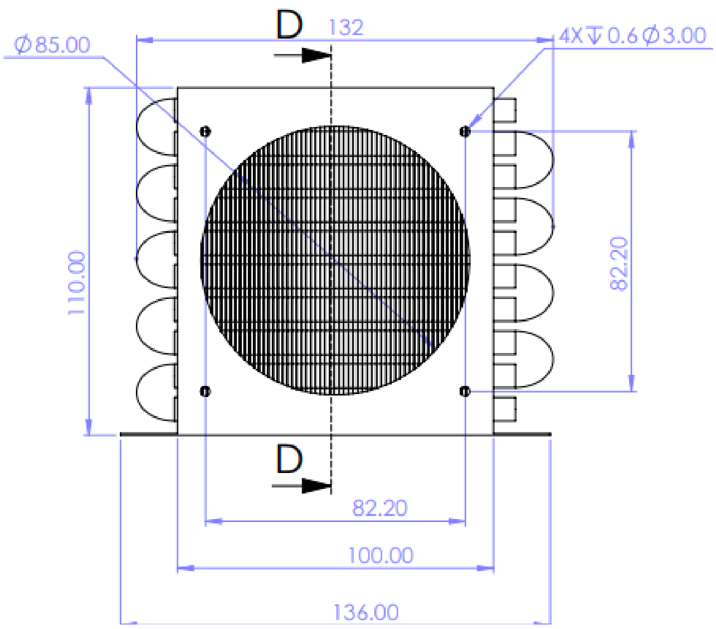

Figure 13: Evaporator (front view)

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Pg:- 51 – 66

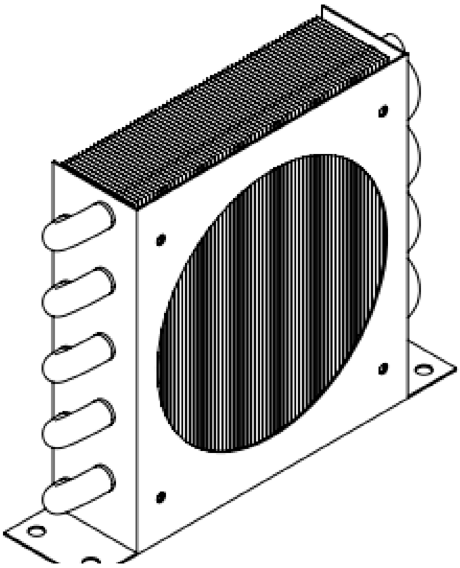
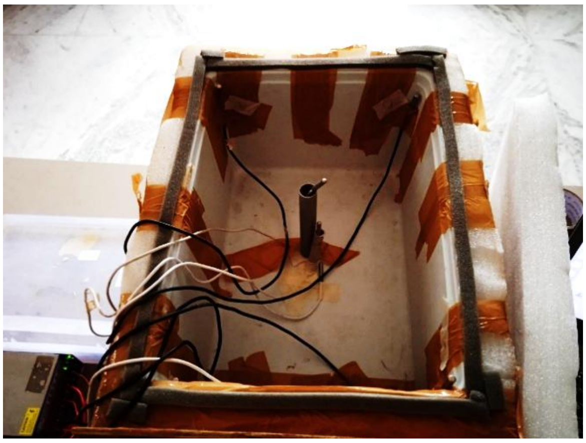


Figure 14: Evaporator (Isometric view)



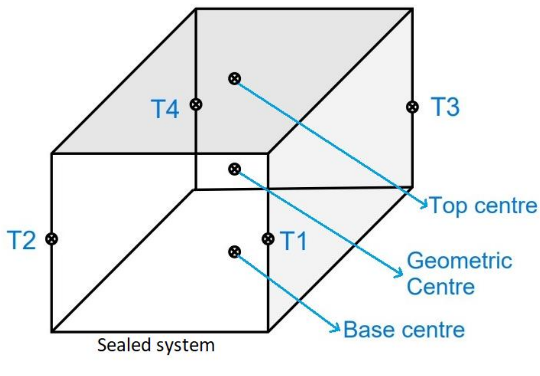


Figure 15: Temperature Zones

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ISSN (ONLINE): 2321-3051

Vol.13 Issue 1, January 2025
Pg:- 51 – 66

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Aim	To check for sealed system performance at room temp for reverification		
Apparatus	15l unit		All Air leakages covered with tape
	Capillary	0.026 inch, 5 meters	
	Evaporator	find and tube type	Capillary wounded on suction and rest unwounded length insulated
	condenser	wire and tube type	cooling axial fan is throughing air over the evap. and throwing cool air from top
	fan speed inl	1.0 m/s	Temp. point measured with thermal camera

Time	Comp. input voltage	comp. input current	Cabinet temp. Top centre	Cabinet temp. Geometric centre	Cabinet temp. base centre	T1	T2	T3	T4	Air inlet to cabinet (cool air)	Air outlet from cabinet (hot air)	return gas temp.	Discharge temp	Condenser exit temp	liquid sub-cooled temp.	Ambient temperature
0	12.1	0.03	30	29.9	29.9	30	30	30	30	29.1	30					
2	12.05	4.16	25.2	25.8	26	29	29	27	28	20.3	25					
4	12.05	4.35	14.8	16.3	16.2	24	26	21	24	7.6	15					
6	12.05	4.16	4.4	5.9	5.3	17	21	13	16	-1.2	6					
8	12.05	3.99	-0.3	0.8	0.4	13	18	7	11	-4.8	2					
10	12.05	3.72	-4.8	-4.7	-4.3	7	14	3	6	-8	-3					
12	12.05	3.51	-7.7	-7.3	-7.4	3	10	-1	2	-10.4	-6					
15	12.05	3.35	-9.9	-9.7	-9.7	0	6	-4	-2	-12.3	-8	25	55	43	42	32
18.3	12.05	3.21	-12.1	-12	-11.9	-4	2	-7	-5	-14	-10					
20	12.05	3.17	-12.7	-12.7	-12.5	-5	1	-8	-6	-14.4	-11	25	56	43	42	32
30	12.05	3	-14.8	-14.9	-14.6	-9	-4	-11	-9	-16.3	-14	26	55	42	41	32
35	12.05	2.94	-15.4	-15.5	-15.2	-10	-5	-11	-10	-17	-14					
40	12.05	2.9	-16	-16	-15.6	-10	-6	-11	-10	-17.3	-15					
51	12.05	2.87	-16.4	-16.5	-15.9	-11	-7	-12	-11	-17.8	-15					

Figure 16: Performance results of the prototype with optimised evaporator and axial fan

6. DESIGN OF DUCTS

The final and most critical stage of prototype development was the design and integration of ducts. Extensive literature research was conducted to understand the principles of ventilation and air conditioning in industrial and domestic settings. The study referred to *A. Bhatia, HVAC - How to Size and Design Ducts* [1] and *Sulekha Walunj, Shantanu Shirsath et. al., "Study and Analysis of Air Flow through Duct"* [2] to grasp the general principles of airflow inside ducts. The design parameters under consideration were cross section, length and layout of the ducts.

The duct layout comprised two separate paths: one for supplying cool air to the cargo chamber and the other for returning hot air from the cargo chamber to the evaporator. Experimental development of the ducts was deemed unfeasible due to the excessive time and resources required to test multiple parameter combinations. Consequently, a virtual simulation of air circulation within the cargo chamber was used to design and optimize the ducts.

SolidWorks 3D modelling software was selected to create a CAD model of the cargo chamber and ducts, while *SolidWorks Flow Simulation Software* was used to simulate airflow. Dimensions of the current prototype were inspected and recorded, and a solid model of the system was created in the software. Based on insights from earlier experiments, it was decided to supply cool air directly from the evaporator into the cargo chamber. For this purpose, a square opening of 100mm X 100mm (matching the fan dimensions) was included in the cargo chamber for the supply air.
The return air was designed to enter from the opposite side and flow back to the evaporator through return ducts. The return ducts were divided into two branches with smooth transitions, and fillets were added near the edges to minimize flow restrictions. Please refer to figure 17 to understand the initial design of the ducts and cargo chamber; the supply air is shown in blue colour and the return air is shown in red.

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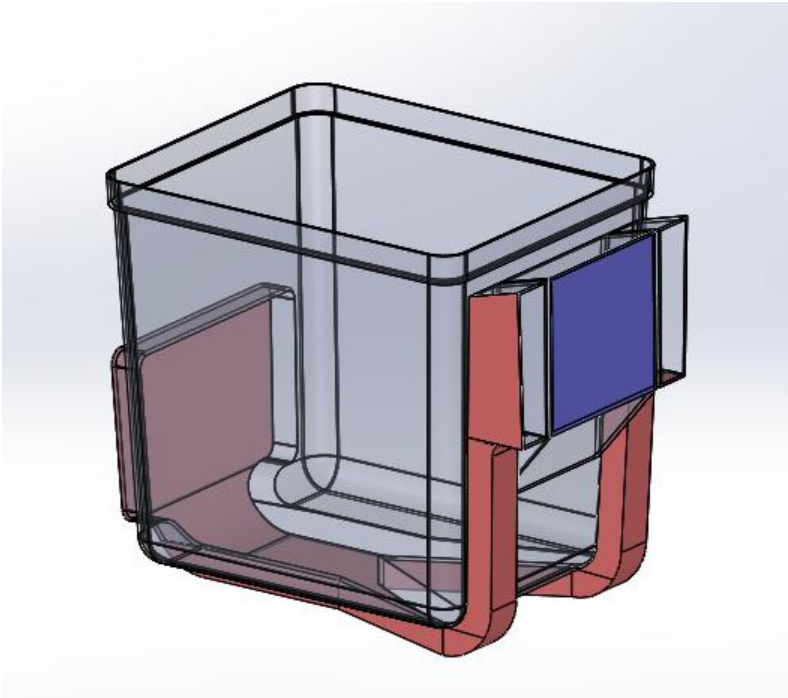

Figure 17: Initial design of the ducts

The CAD model was imported into a simulation environment to analyse airflow. The fan's specifications, including CFM, RPM, and static pressure, were provided as input parameters. Since the airspeed of a fan varies with resistance to airflow (static pressure), the *SolidWorks Flow Simulation Software* enabled accurate airflow simulation by allowing input of a CFM vs. static pressure graph. The specifications of the axial fan used in the current prototype were applied to the simulation (see Figure 18).

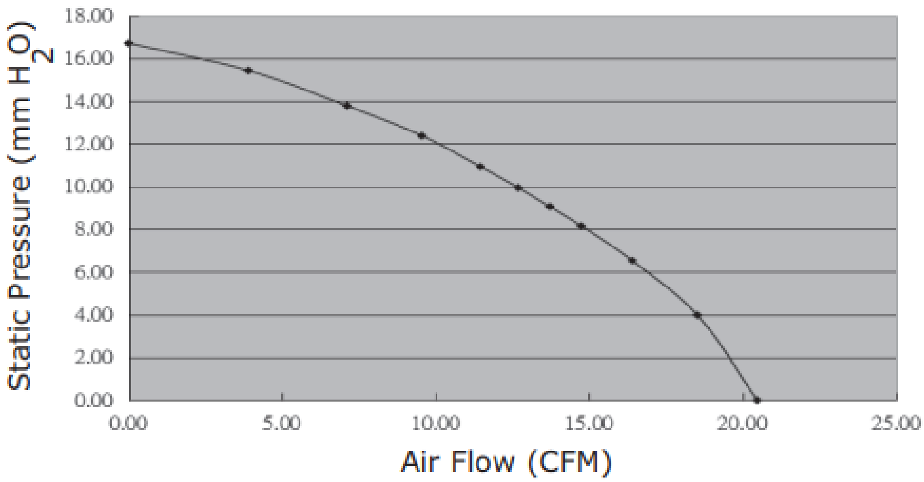

Figure 18: Variation of airflow against Static pressure

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Pg:- 51 – 66

The simulation's objective was to ensure even airflow inside the cargo chamber while maximizing return air supply. Return air velocity was used as a reference for the amount of air being circulated. The height of the return ducts was fixed at 20 mm, while other dimensions and layouts were adjusted during the simulation process. To evaluate different duct designs, velocity cut graphs at various cross-sections and airflow trajectories were analysed. Higher velocity indicated better return air supply. Improvements to the initial return duct design included a smooth transition from a single duct branch into two, with the division placed closer to the fan's suction. These changes enhanced airflow distribution and minimized resistance (see Figures 19 and 20).

The simulation results demonstrated that air velocity was higher in the branch closer to the fan suction (right-hand side) compared to the other branch (left-hand side). The optimized design, as shown in Figures 19 and 20, featured this improved layout on the right-hand side and was selected as the final configuration.

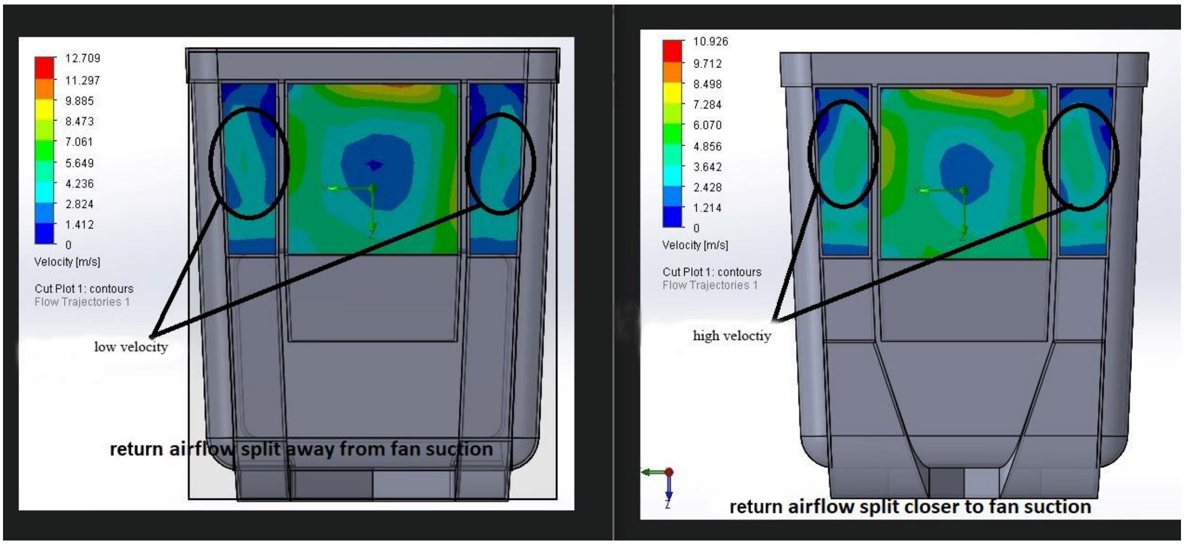

Figure 19: Velocity contour graph from the flow simulation

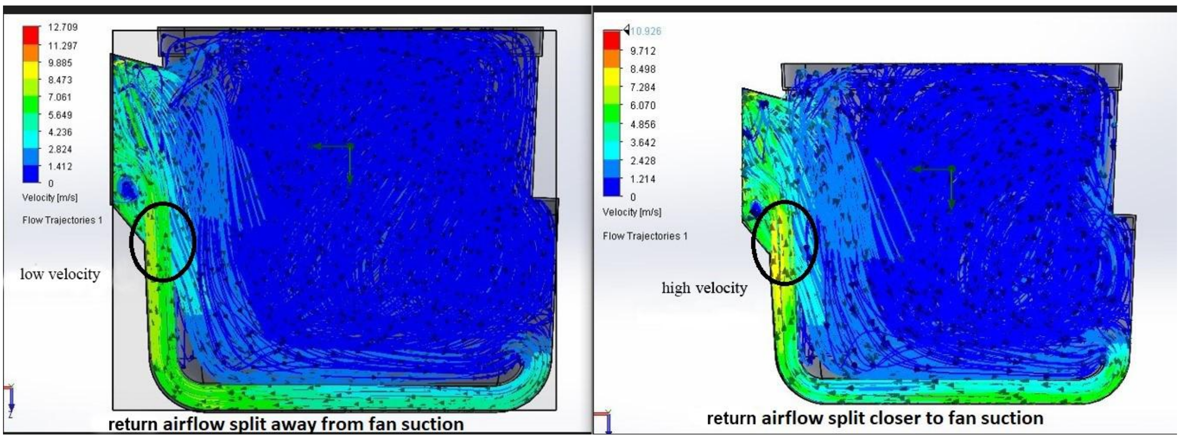

Figure 20: Flow Trajectories from the simulation

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Pg:- 51 – 66

7. RESULTS

The next step was to create and attach the ducts to the main prototype. To achieve this, we opted for FDM (Fused Deposition Modeling) 3D printing—a widely used method for prototyping plastic parts with complex designs. This approach was not only cost-effective but also efficient. For the material, we used PLA (Polylactic Acid), which is ideal for such applications. Once the ducts were fabricated, they were integrated with the existing axial fan setup. A cool-down experiment was then conducted to evaluate the performance of the system with the new ducts compared to its previous configuration. The output air speed from the axial fan reduced to 0.7m/s due to pressure loss inside the ducts, this caused degradation in the cool down performance of the unit. Hence, a more powerful axial fan with 65 CFM capacity was used, which provided the necessary air speed of 1.0 m/s.

To understand the temperature distribution, an experiment was devised to simulate the actual use case scenario. In this test, a standard cooling load .i.e. water bottles precooled at 5C were kept inside the cargo chamber and the temperature retention of the unit was tested by running multiple on/off cycles of the refrigeration system. The compressor was switched off when the temperature reached 5C inside the cargo chamber and switched on again when it increased to 6C. Also, temperatures inside various zones of the cargo chamber were noted down. The results of the temperature retention experiments of ACD (15L V1) and the duct based prototype (15L V2) are shown in the figure 21 below.

	ON/OFF Cycle Experiment												
	Air speed (in m/s)	Amperage (at 5C GC)	Ambient (C)	load	cool down time at 5C (mins)(with load)	Avg On time (mins) (with load)	Avg Off time (mins) (with load)	On time ratio	T1 (min-max) during on-off cycles	T2 (min-max) during on-off cycles	T3 (min-max) during on-off cycles	T4 (min-max) during on-off cycles	power consumption in a day (W-hr)
15L V1	0	4.65	30.-31	50% vol. capacity of vials	14.15	1.39	4.13	0.252	1--2	2--4	1--3	2--4	285
15L V2	1.1	4.15	30-31	50% vol. capacity of vials	12.35	5.42	16.28	0.249	4--6	3--5	4--6	5--7	317

Figure 21: Comparison between 15L V1 (ACD) and 15L V2 (the duct based prototype) for cool down and temperature retention experiments

T1 to T4 represents the temperature range maintained across the four corners of the cargo chamber during the compressor's on/off cycles. The figure also highlights the cool-down time (time required to reach 5°C inside the cold chamber) and the average on/off cycle ratio. Notably, the cool down time was faster for the *duct-based prototype* by 2 minutes (approx.) and the temperature zones of the cold chamber were slightly better than those of the ACD. The values were closer to the mean value of 5C and the deviation was under 2°C compared to 4°C of the ACD. This indicates that the addition of the *forced convection evaporator unit* to the ACD design enhanced both cooling performance and temperature distribution.

However, the duct-based system consumed more energy than natural convection type. Given, the results there is still scope for cool down performance improvement and also a reduction in energy consumption. The insulation used was different in the prototype compared to the ACD. It could help to have a proper comparison if a new prototype with similar polyurethane insulation is made. Also, Using a PWM control-enabled fan to vary the airflow inside the cargo chamber depending on the load and ambient temperature conditions could further increase the efficiency. Moreover, replacing R134a refrigerant with R1234yf would reduce the Global Warming Potential of the unit and still give a similar performance.

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INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING
WWW.IJRAME.COM
ISSN (ONLINE): 2321-3051

Vol.13 Issue 1,
January 2025
Pg:- 51 – 66

8. CONCLUSION

In conclusion, the forced convection miniature evaporator unit on an Active Carrier Device (ACD) demonstrated significant improvements in cooling efficiency and temperature uniformity, essential for transporting temperature-sensitive medical goods. A similar prototype to the ACD was developed using an experimental approach, which involved selecting an axial fan for improved airflow and modifying the evaporator by removing fins. The duct design was optimized using simulation software, resulting in an integrated system. Performance tests revealed that the new design achieved faster cooldown times, with temperature zones more uniform and closer to the mean value of 5°C, albeit with a slight increase in power consumption.

9. ACKNOWLEDGEMENT

I would like to sincerely thank Mr. Sivakrishna Vattikunta for his assistance in prototype development and Mr. Abhishek Chavan for his dedicated efforts in conducting experiments. Your contributions have been instrumental in achieving the research goals.

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