Albatross Energetics:
Innovative Cooling And Dehumidification Solutions

END - TERM REPORT

SUBMITTED BY :

TEAM 30

CONTENTS

Chapter 1

Model Details

The vapor compression cycle operates on a closed-loop system, using a refrigerant to transfer heat from a low-temperature region (cooling space) to a high-temperature region (external environment). The cycle consists of four primary stages: compression, condensation, expansion, and evaporation. During these stages, the refrigerant alternates between its vapor and liquid phases, absorbing heat from the cooling space and releasing it to the surroundings. This process enables effective temperature control while maintaining energy efficiency.

1.1 Description of the Model

1.1.1 Key Libraries and Tools Used

MATLAB/Simulink tools are used for modeling and analysis. Simscape Fluids provides blocks for fluid systems and thermodynamic analysis. Simulink Control Design facilitates PID controller design and real-time adjustments. Simscape Thermal Systems models heat transfer for condensers and evaporators. MATLAB Function Blocks implement custom control logic, while Simulink Scope and MATLAB plots visualize outputs like temperature and energy efficiency, ensuring system validation under varying conditions.

1.1.2 Control System

Control Logic For Indoor Fan RPM:

The rate of temperature decrease increases with an increase in compressor RPM, whereas the rate of dehumidification decreases. The deviation of both the dry-bulb temperature and relative humidity (which can be derived from the wet-bulb and dry-bulb temperature) must be considered when applying the control logic.

The error for both temperature and relative humidity is calculated with respect to their respective setpoints. These error signals are then processed by two separate PID controllers: one for the temperature error signal and the other for the humidity error signal. The PID controllers generate control signals for the indoor fan RPM.

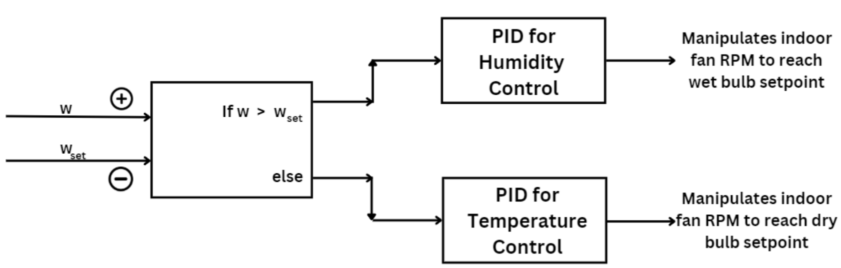

Figure 1.1: Control System

The two PID controllers operate alternately based on the difference between the desired humidity setpoint and the actual room humidity. If the humidity is below the set-point, the PID controller for temperature error controls the fan; otherwise, the PID controller for humidity error adjusts the fan RPM to achieve the desired relative humidity.

The resulting RPM signal is constrained within physical and operational limits using saturation. This ensures the RPM does not exceed the allowable range for safe and efficient operation.

Control Logic for Compressor RPM:

A PID controller is implemented to control both the dry-bulb temperature and humidity (which can be derived from the wet-bulb and dry-bulb temperature) by dynamically varying the compressor RPM.

The rate of decrease in temperature and humidity increases with an increase in compressor RPM. Here, only the deviation of the dry-bulb temperature from the setpoint is used to generate a control signal for the compressor RPM. The error signal for humidity generates a similar control signal as the temperature error signal; therefore, the PID controller uses only the deviation of room temperature from the desired setpoint to adjust the compressor RPM.

The resulting RPM signal is constrained within physical and operational limits using saturation. This ensures that the RPM does not exceed the allowable range for safe and efficient operation. The constrained RPM signal is then fed back into the compressor's shaft speed and used as feedback for error calculation, creating a closed-loop control system.

We have set the RPM of the condenser fan to the maximum speed because the rate of change of both temperature and humidity was highest in it.

1.2 Block/Component Choice

Condenser Block: The condenser block simulates a heat exchanger equipped with fins to enhance heat dissipation to the surroundings. It ensures efficient rejection of heat absorbed from the refrigerant during the cooling cycle.

Evaporator Block: The evaporator block represents a finned heat exchanger that facilitates efficient heat absorption from the surrounding air. It ensures proper cooling and humidity control in the system.

Flow Rate Sensor: The flow rate sensor block is used to measure the fluid flow rate within the system accurately. It ensures precise monitoring and control of the fluid dynamics, which is critical for maintaining optimal system performance.

Positive Displacement Compressor Block: The positive displacement compressor block models the scroll compressor used in the system, which operates by trapping and compressing refrigerant between stationary and orbiting scrolls. This block accurately represents the thermodynamic processes of compression, essential for maintaining desired pressure and temperature levels in the refrigeration cycle.

Thermostatic Expansion Valve: The thermostatic expansion valve block regulates the flow of refrigerant entering the evaporator by maintaining the appropriate superheat. This ensures efficient heat exchange and prevents liquid refrigerant from entering the compressor, protecting the system.

Liquid Receiver: The liquid receiver block stores excess refrigerant to ensure a steady supply to the expansion valve, accommodating varying system loads. It also separates liquid and vapor phases, maintaining proper refrigerant flow in the system.

Chapter 2

Assumptions and Justifications

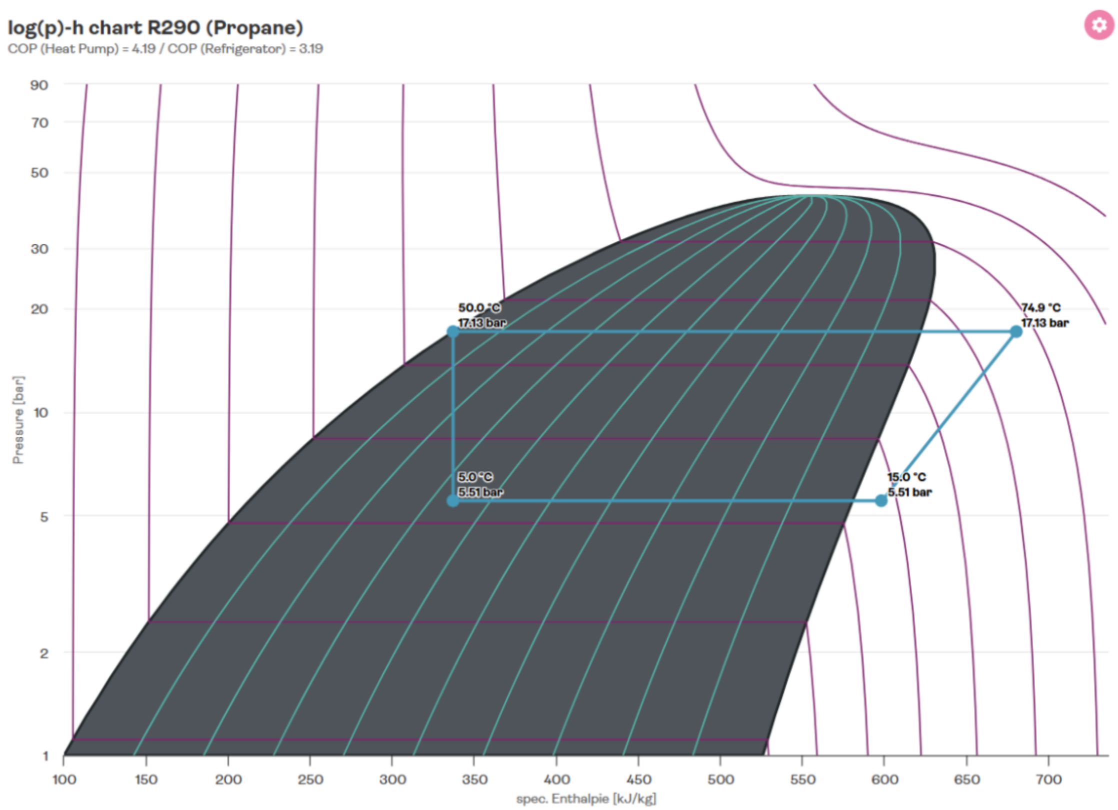

Figure 2.1: Ideal P-h Chart

With R290 as the refrigerant and other provided parameters, an ideal log(P)-h graph can be plotted to obtain the initial conditions for temperature, pressure, and specific enthalpy of all blocks for the ideal vapor compression refrigeration cycle. Keeping the maximum load to be 1.25 tons of refrigeration, **safety factor of 4.5kW** is taken into account. The change in specific enthalpy across the evaporator is determined, and thus from the data tabulated above, a load of 4.7507 kW is obtained at nominal conditions (i.e., at 2900 rpm).

With the nominal load condition, the evaporator and condenser would be designed to exchange this load of energy to their respective sinks. The **cross-flow method** is best suited for heat transmission for the DX system. To increase evaporation, the total surface area was increased. The effective transfer area of 14 m^2 is required, which was achieved by **added fins to the copper tubes** in the confined volume itself. Readily available copper tubes (36 fins per inch), which had a fin area of 0.78 m^2 per unit length, a total length of evaporator tube are estimated to be 20 m, providing a net surface area of 15.6 m^2. The same process was repeated for the condenser as well. The condenser model was enhanced (described in chapter 5 in detail), which deliberately increased the net heat transfer by increasing airflow using a **mist + air system** that would show growth in the system's real-life implication. This model was able to attain a COP of 3.33 with constrained condensation/evaporation temperature.

The modulation and manipulation of fan RPM and compressor RPM via controllers are mentioned in the **control system** section.

Few assumptions, like peak values being 20% more than nominal values, are taken into account. Fan property table is used and a curve-fitting method is implemented, which is explicitly mentioned and calculated in the model as well in the 'Work Function Subsystem'.Fans used in this model work on 2500 RPM derived from model available in market. The efficiency and power consumption of the fans and compressor used in this model are taken from the existing models. The efficiency can be improved further by material selection and appropriate appliances like better motors in fans and compressors, which are eventually used in high ISEER-rated ACs, like **DC switch reluctance motors** and many more. For any other parameters used either the default values or values derived via required calculation are used.

Chapter 3

Performance Metrics

To calculate the ISEER, the indoor dry bulb temperature of 27℃ and the indoor wet bulb temperature of 19℃. The corresponding indoor specific humidity is calculated to be 46.3%. The EER is averaged over 600 seconds (10 minutes) for every temperature from 24℃ to 43℃.

3.1 EER (Energy Efficiency Ratio)

EER is the ratio of the amount of cooling output of a system per unit of input power over a set period of time:

$$EER = \frac{\text{Cooling Output (BTUs)}}{\text{Power Input (Watts)}}$$

3.2 ISEER (Indian Seasonal Energy Efficiency Ratio)

ISEER is the ratio of the amount of heat an air conditional units can remove per unit of energy consumption:

$$ISEER = \frac{CSTL}{CSEC}$$

where CSTL : Cooling Seasonal Total Load (BTUs)
CSEC : Cooling Seasonal Energy Consumption (Watts)

Temperature (℃)	**24**	**25**	**26**	**27**	**28**	**29**	**30**	**31**	**32**
Average Annual Hours	527	590	639	660	603	543	451	377	309
Fraction	9.1	10.2	11.1	11.4	10.4	9.4	7.8	6.5	5.4
Bin Hours	146	163	177	183	167	150	125	104	86
EER (BTU/h/W)	12.21	12.02	11.85	11.68	11.51	11.34	11.18	11.02	10.88

33	**34**	**35**	**36**	**37**	**38**	**39**	**40**	**41**	**42**	**43**	**Total**
240	196	165	130	101	79	59	44	31	20	10	5774
4.2	3.4	2.9	2.3	1.7	1.4	1.0	0.8	0.5	0.3	0.2	100
67	54	46	36	28	22	16	12	9	6	3	1600
10.75	10.60	10.47	10.34	10.23	10.12	10.01	9.89	9.79	9.69	9.60	

Table 3.1: EER Values for Outdoor Conditions

Average annual temperature throughout the year is 29℃. The ISEER value calculated is 3.33 (W/W).

Chapter 4

Performance and Output Requirements

The indoor conditions are set to be 27℃ dry bulb temperature and 19℃ wet bulb temperature and the outdoor temperature is set to be 35℃.

4.1 Pressure-Enthalpy (P-h) Chart

Below is the Pressure-Enthalpy chart of our model:

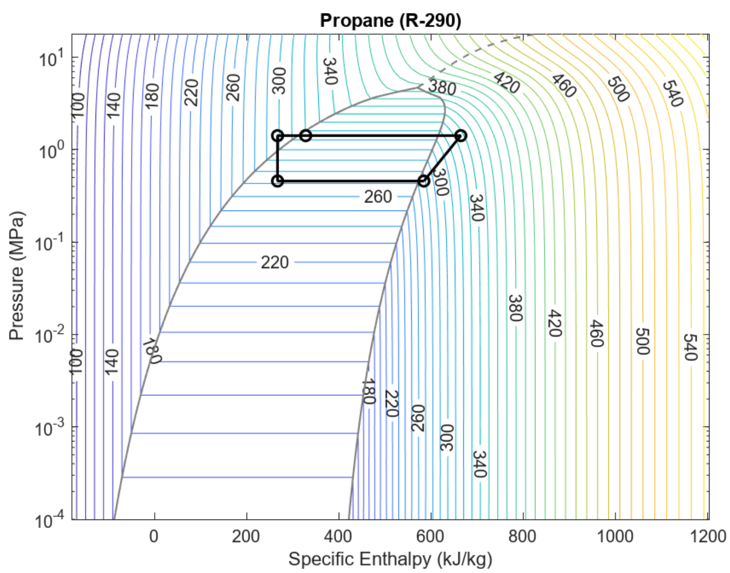

Figure 4.1: P-h Chart

4.2 Other Graphs Displaying Key Metrics

4.2.1 Indoor Conditions

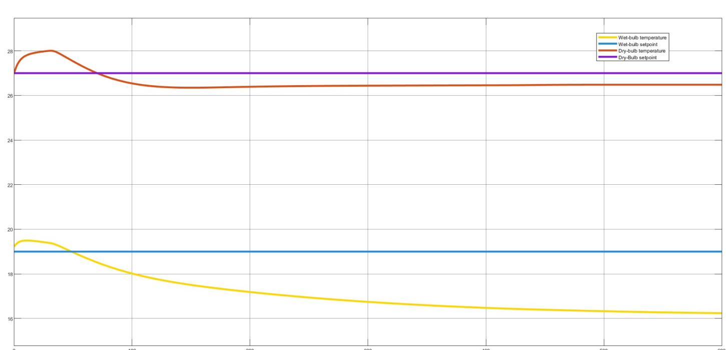

Figure 4.2: Dry Bult and Wet Bulb Temperature vs Time

4.2.2 System Performance

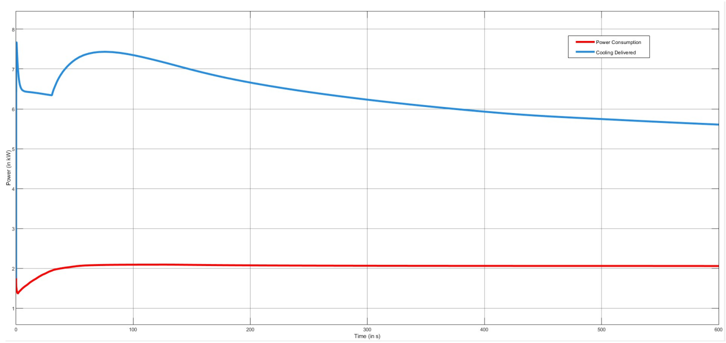

Figure 4.3: Cooling Delivered and Power Consumption vs Time

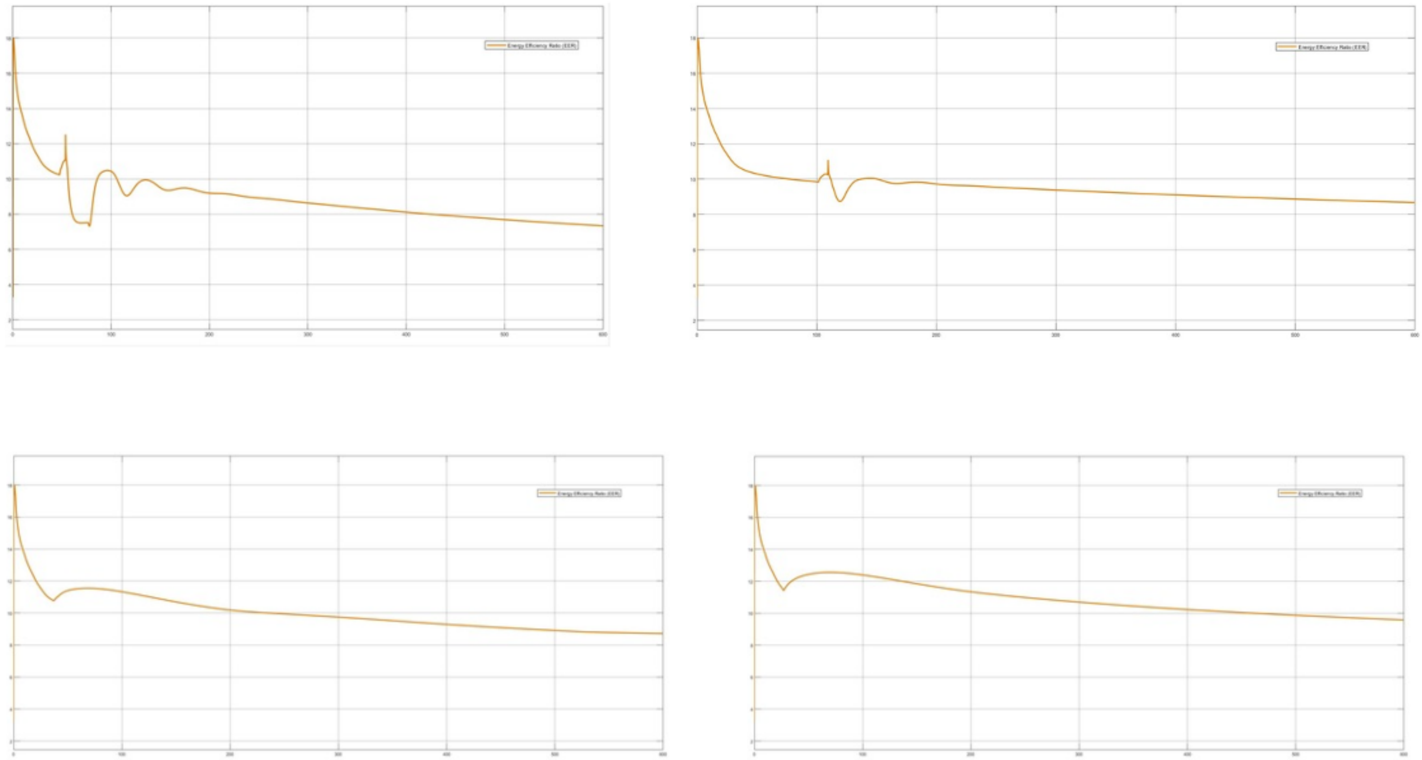

Figure 4.4: Energy Efficiency Ratio (EER) vs Time at different loads

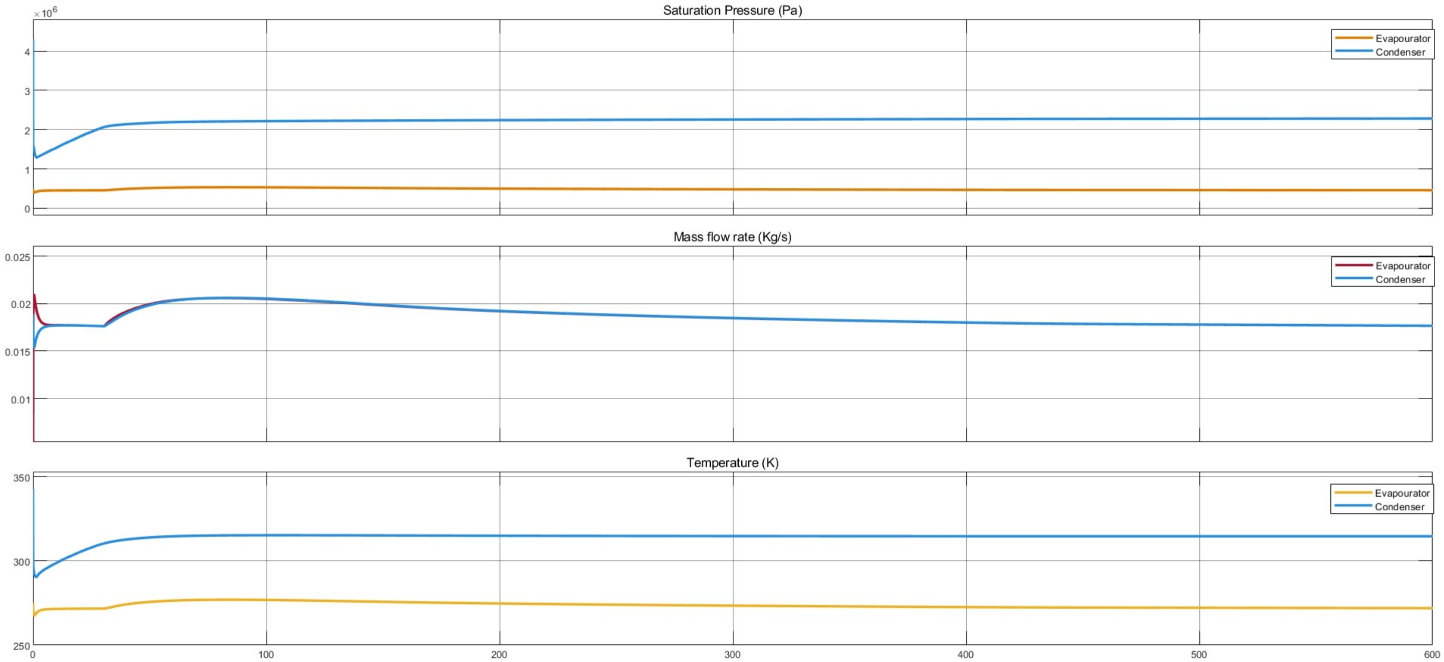

Figure 4.5: Saturation Pressure and Saturation Temperature vs Time

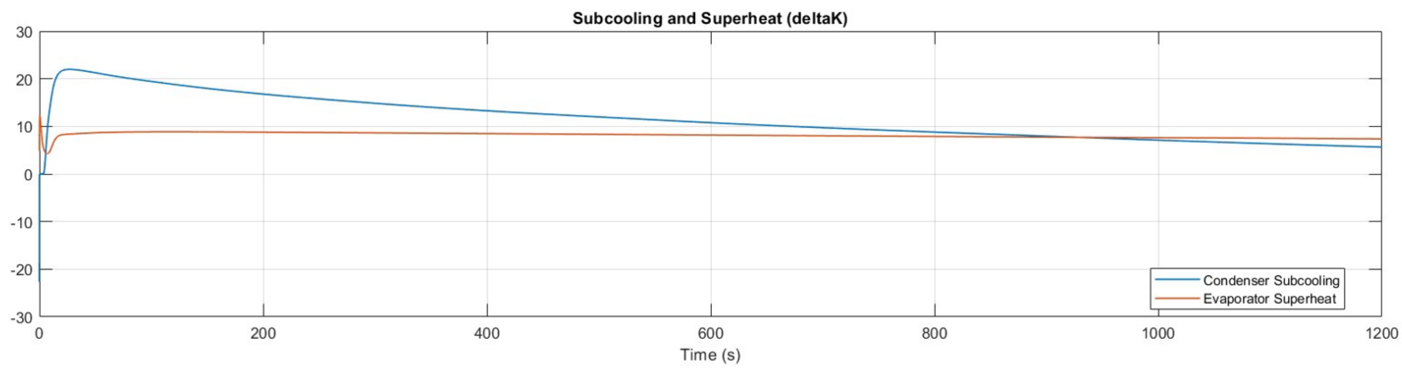

Figure 4.6: Superheat and Subcooling vs Time

Chapter 5

Key Innovations and Environmental Considerations

5.1 Innovation

While optimizing the performance of HVAC systems there exist few trade-offs that come with its higher efficiency. One of the major disadvantages being space and weight constraints.

Larger Components: High-efficiency heat exchangers or compressors may be bulkier, which can be an issue in systems where space is limited.

Additional Insulation Needs: To prevent thermal losses, more space might be required for insulation materials.

5.1.1 Redesigning Condenser

To counter that the design was optimized such that it achieves better heat transfer in constrained space:

Figure 5.1: Alternate Designs For Condenser

Tower/Cylindrical Fan:- This type of fan is usually used in evaporator and can be used in condenser as well so as to generate desired air flow in confined volume. Moreover, the airflow thus generated flows through entire system catering more heat transfer thus increasing the efficiency of condenser. This arrangement of fan allows air to flow through the system enhancing the net heat transfer rate. The parallel arrangement of tubes supports the mass flow rate.

Arrangement of Tubes:- Rather than parallel arrangement, the following arrangement of tubes will perform better in condenser allowing air to span completely through the condenser.

Cooling Mode:- The efficiency of the condenser can be increased further by improving its cooling which is done either by air or water. Water cooling performs better but adds more weight to the system making it less convenient for small load usage.

So as to improve this, **cooling via mist and air** was opted. Mist is generated by small pump which sprays it at entrance the mist is then mixed with the circulating air in result decreasing its temperature. The decrease in temperature of air thus increases the temperature difference between the condenser pipes (refrigerant) and air

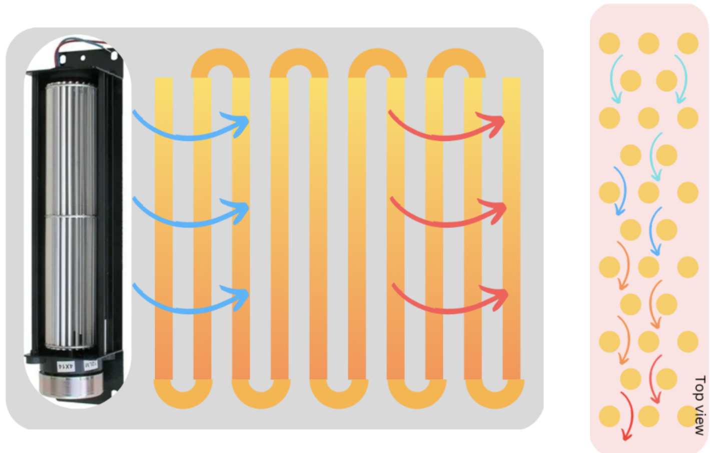

Figure 5.2: Condenser Tube Arrangement

which in result increases the rate of heat transfer thereby increasing the efficiency of condenser by Newton's law of cooling. The water used in this process will be derived from the evaporator.

Benefits of Redesigned Condenser :-

- Cost effective cooling enhancement.
- Improved COP due to enhanced heat rejection.
- Peak Load Conditions: Misting can help AC systems perform better during periods of high thermal load.

5.1.2 Refrigerant Safety Management

Although it is widely used, R290 (propane) is a low potential global warming refrigerant that is friendly to the environment. But it is rather flammable and has a flammability range of 2.1-9.5% of air concentration. Moreover, it becomes life-threatening by being an asphyxiant at higher levels as it can cause oxygen displacement and hence suffocation in confined spaces. This is particularly hazardous in poorly ventilated, enclosed spaces. This is especially dangerous in enclosed spaces with poor ventilation.

These dangers necessitate the need for R290 to be managed with the highest safety measures. To solve this problem, the Simulink model incorporates a safety subsystem that accounts for any **fluctuation in mass flow rate** to detect for any possible leak. If the fluctuation remains for a time the refrigerant may accumulate in room and if the volumetric composition exceeds **LEL (Lower Explosive Limit, 2.1%)** in the room then the subsystem stops by cutting off the expansion valve and the compressor. This way the leakage of refrigerant is monitored and the harmful effects of high propane concentration is minimized.

5.2 Environmental Consideration

5.2.1 Refrigerant Analysis

The primary characteristics of refrigerants with their compatible compressor and their parameters were tabulated:

Compressor	Compressor 1	Compressor 2	Compressor 2	Compressor 2	Compressor 2	Compressor 2	Compressor 3	Compressor 3	Compressor 3	Compressor 3	Compressor 4
Refrigerant	R290	R134a	R407C	R513A	R454C	R1234yf	R134a	R407C	R450A	R513A	R410A
Dispalcement (cm^3/rev)	33.1	45.5	45.5	45.5	45.5	45.5	33.1	33.1	33.1	33.1	22.8
Isoentropic Efficiency	68.40%	68.80%	69.10%	67.50%	67.90%	64.80%	63.90%	65.30%	65.60%	65.80%	65.10%
COP (Refrigeration)	3.19	3.28	2.76	3.1	3.1	2.92	3.04	2.61	3.08	3.02	2.77
T_Max_Out (Celsius)	74.9	75.6	91	69.9	68	66.7	78.6	93.6	68	70.7	93.6
T_Max_In (Celsius)	5	5	1.1	5	5	5	5	5	5	5	4.9
P_Max_Out (Bar)	17.13	13.18	22.16	13.77	23	13.02	13.18	22.16	11.7	13.77	30.71
P_Max_In (Bar)	5.51	3.5	5.47	3.85	6	3.73	3.5	5.47	3.1	3.85	9.33
GWP	3	1430	1774	573	148	4	1430	1774	602	573	2088

Figure 5.3: Refrigerants Characteristics

COP, Temperature and Pressure data was determined from P-h (Pressure-Enthalpy) plot for every refrigerant.

Refrigerant	R290 (C1)	R134a (C2)	R513A (C2)	R454C (C2)	R1234yf (C2)	R134a (C3)	R450A (C3)	R513A (C3)
GWP (0.35)	5	4	4	5	5	4	4	4
COP (0.5)	5	5	4	4	3	4	4	4
Vol_disp (0.15)	4	5	5	5	5	4	4	4
TOTAL	**4.85**	**4.65**	4.15	4.5	4	4	4	4

Figure 5.4: Multi-Criteria Decision Making (MCDM) Analysis

On the basis of **COP**, **GWP** and **Displacement Volume** values, the two of most suitable refrigerants for modeling the HVAC System are R134a (Compressor 2) and R290 (Compressor 1).

Refrigerant	Displacement Volume (cm^3/s)	Nominal RPM	Nominal Temp (T_N)(K)	Density @ T_N (kg/m^3)	Mass flow (kg/s)	del H (kJ/kg)	Cooling Power
R290	0.0331	2900	288.15	11.36	18.1741 x 10^(-3)	261.4	4.7507
R134a	0.0455	2900	288.15	16.23	35.6924 x 10^(-3)	139	4.96124

Figure 5.5: R290 vs R134a

Finally, we opted to use **R290** (propane) as the refrigerant with **Scroll Compressor 1**, owing to its favorable thermophysical characteristics and environmental sustainability.

Property	Value/Description
Chemical Name	Propane (R290)
Molecular Weight	44.1 g/mol
Boiling Point at 1 atm	-42.1℃
Critical Temperature	96.7℃
Critical Pressure	42.5 bar
Latent Heat of Vaporization	335 kJ/kg
Thermal Conductivity (at 25℃)	0.014 W/m·K
Specific Heat Capacity (at 25℃)	1.67 kJ/kg·K
Global Warming Potential (GWP)	$<$3 (approximately zero)
Ozone Depletion Potential (ODP)	0
Flammability Limits	2.1–9.5% Vol

Table 5.1: Thermophysical Properties of R290 (Propane)

Chapter 6

Cost Analysis

6.1 Capital Expenditure

Material	Specification	Price
Pressure Safety Valve	15 inch	Rs. 2220
Fin Copper Tube	40 fins/inch, 50 metres	Rs. 6666
Thermostatic Expansion Valve	Compatibility with R290	Rs. 3560
Air Filter	34 cm x 22 cm x 0.1 cm	Rs. 399 x 2
Refrigerant	R290, Weight 1.2 kg	Rs. 1400
High Efficiency BLDC Motor	2000-2500 RPM, 0.138 kW	Rs. 3835 x 2
Fan	Aerodynamic ABS Anti-Tower, 13.5 inch	Rs. 400 x 4
Thermostat	Range: -10℃ to 60℃	Rs. 2055
Wiring	Length: 2 metres, Diameter: 4 mm	Rs. 10 x 3
PCB	Thickness: 1.6 mm	Rs. 6500
Gas/Smoke Sensor	Propane Detection	Rs. 1289
Active Peizo Buzzer	85 dB	Rs. 46
Mist Maker	24 Watt	Rs. 290
AC Aluminium Outdoor Cabinet	1.5 Ton	Rs. 2745
AC Aluminium Indoor Cabinet	1.5 Ton	Rs. 2130
Scroll Compressor	33.1 cm$^3/rev$	Rs. 14000

Table 6.1: Capital Expenditure

Total Capital Expenditure = Rs. 52,999

6.2 Operational Expenditure

Electricity Costs -
Power Rating: 1.25 Ton of refrigeration (4.4 kW)
Usage Hours: 12 hours
Electricity Tariff: The average cost of electricity in India is INR 8 per kilowatt-hour
Days per Year: 365 days

$$\begin{aligned}
\text{Annual Operational Cost (in INR)} &= (\text{Power Rating in kW}) \times (\text{Usage hours per day}) \\
&\quad \times (\text{Days per year}) \times (\text{Electricity Tariff}) \\
&= (1.323\ \text{kWh}) \times (12\ \text{hours}) \\
&\quad \times (365\ \text{days}) \times (\text{Rs. } 8) \\
&= \text{Rs. } 46357.92
\end{aligned}$$

6.3 Potential Savings

In India, the CoP for residential air conditioners typically ranges between 2.5 and 3.0, for our calculations, we have taken 2.8. We have achieved a CoP of 3.33 indicates that your system operates more efficiently than the average unit.

For AC Unit with 2.8 CoP:-

$$\begin{aligned}\text{Power Input for 1.25 Ton} &= \frac{4.4\text{ kW}}{2.8} \\ &= 1.57\text{ kWh}\end{aligned}$$

For AC Unit with 3.33 CoP:-

$$\begin{aligned}\text{Power Input for 1.25 Ton} &= \frac{4.4\text{ kW}}{3.33} \\ &= 1.32\text{ kWh}\end{aligned}$$

Operational Cost Savings:-

$$\begin{aligned}\text{Energy Saved per day} &= (\text{Power Input Difference}) \times (\text{12 hours/day}) \\ &= 3.0\text{ kWh} \\ \text{Annual Energy Savings} &= (\text{Energy Saved per day}) \times (365) \\ &= 1095\text{ kWh} \\ \text{Annual Operational Cost Savings} &= (\text{Annual Energy Savings}) \times (\text{Electricity Tariff}) \\ &= 1095\text{ kWh} \times \text{Rs. 8/kWh} \\ &= \text{Rs. 8,760 per year}\end{aligned}$$

Capital Cost Excess:-

$$\begin{aligned}\text{Capital Cost Excess} &= (\text{Our Capital}) - (\text{Standard AC Cost}) \\ &= (\text{Rs. 52,999}) - (\text{Rs. 34,999}) \\ &= \text{Rs. 18,000}\end{aligned}$$

Cost Recovery Time:-

$$\begin{aligned}\text{Time (in months)} &= (\text{Capital Cost Excess})/(\text{Operational Cost Savings}) \\ &= (\text{Rs. 18,000})/(\text{Rs. 8,760}) \\ &\approx 2\text{ years}\end{aligned}$$