Albatross Energetics - Innovative Cooling and Dehumidification Solutions

Inter IIT Tech Meet 13.0

Final Submission Report

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Contents

1 Abstract

This project develops an HVAC model for efficient cooling and dehumidification of a house, maintaining a set temperature within a specified time frame. The system employs custom control logic based on a PID controller, which dynamically adjusts the compressor shaft speed. This in turn regulates the mass flow rate of the refrigerant, optimizing the thermal load and humidity levels to meet the desired environmental conditions effectively. The transfer function and components of the HVAC cycle are derived based on the known initial parameters, providing a mathematical framework to model the system's dynamic behavior.

2 Problem Understanding

With increasing occurrences of extreme heat waves and rising energy demands, providing affordable and efficient cooling solutions has become a critical challenge. In India, space cooling contributes significantly to electricity consumption, yet access to air conditioning remains limited due to high costs. This disparity highlights the urgent need for innovative cooling technologies that are both sustainable and accessible.

The problem statement involves designing a vapor compression-based air conditioning (AC) system to optimize cooling, dehumidification, and energy efficiency within safe operational limits. Participants must model the system in MATLAB/Simulink, incorporating a compressor, refrigerant, evaporator, condenser, and custom control logic to adapt to varying load conditions. The design must maximize EER and ISEER while using environmentally responsible refrigerants and adhering to safety requirements. Scoring emphasizes performance, operational efficiency, and environmental impact, with penalties for delays or safety violations. The challenge addresses the need for sustainable, cost-effective cooling solutions in the context of increasing energy demand and environmental concerns.

3 Vapour-Compression Model developed in Simulink

- The model is based on the standard HVAC loop which contains an evaporator, condenser, expansion valve and a compressor.
- The evaporator and condenser are connected with house subsystem and external environment blocks respectively.The moist air fluid which is a part of heat exchanger cycle for both evaporator and condenser is obtained from the connection at port 2 for both the devices.
- The model contains a P-h diagram block where the final P-h diagram is obtained.It also contains an ISEER calculator block which computes the final ISEER once the input block is updated from running the model

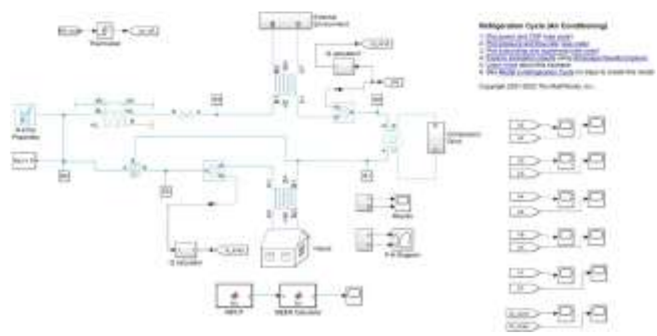


Fig. 1: The Simulink Model

- The house model contains various parameters such as window surface area,roof surface area ,walls and furniture.
- It also contains blocks of occupant heat gain and appliance heat gain, where occupant heat gain sums up the net heat gain by all the live bodies present in the room and appliance heat gain stands for the net heat gain by all the appliances in the room.
- The fan present in the house subsystem plays a role in taking heat from the room and passing it onto the evaporator.
- The house parameters are decided based on the mentioned parameters i.e 100 sq.ft floor area and 8 ft height.

4 Model Details

4.1 Key Considerations for Choosing Refrigerant

The refrigerant chosen for the air condition process is R134a also known as hydrofluorocarbon-134a (HFC-134a).

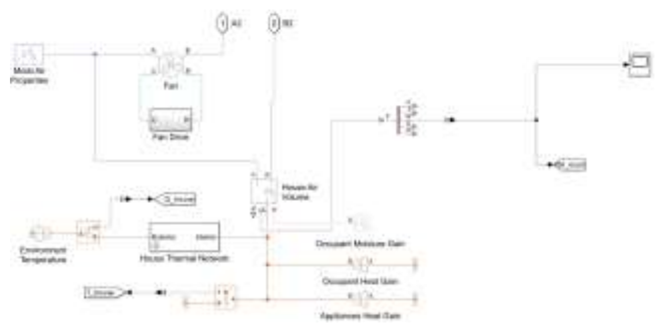
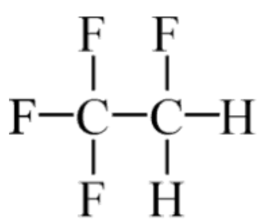


Fig. 2: The House Subsystem

Chemical Name: 1,1,1,2-tetrafluoroethane
Molecular Formula: CH_2FCF_3
Molecular Weight: 102.0 g/mol
Chemical Structure:



- **Physical Properties:** With a boiling level of -26.3°C (-15.34°F) at atmospheric pressure, it undergoes partial transition from liquid to vapor at comparatively low temperatures, facilitating efficient heat transfer within refrigeration cycles. Additionally, R134a demonstrates a moderate vapor pressure and high volumetric cooling capacity, enabling it to absorb and dissipate heat effectively.
- **Environmental Impact:** R407C and R410A,being popular refrigerants, have even higher GWPs of 1774 and 2088, respectively. Choosing R134a over R407C and R410A can help reduce environmental impact while maintaining safety and efficiency in refrigeration and air conditioning applications.
- **Safety Considerations:** R134a is classified as non-flammable (A1), making it safer to handle and use compared to refrigerants like R290, R454C, and R1234yf, which are flammable to varying degrees. R134a has low toxicity (A1), ensuring it poses minimal health risks during use. R134a's lower GWP, nonflammability and low toxicity make it a comparatively better choice.
- **Cost-Effectiveness:** R134a is a widely used refrigerant known for its non-flammability and cost-effectiveness, with prices ranging from INR 410 to INR 500 per kg. Compared to other refrigerants like R290, R407C, R513A, R454C, R1234yf, R450A, and R410A, R134a offers a balance of safety, affordability, and performance.

4.2 Modelling the T-s graph

For the chosen refrigerant R134a, depending upon the given inlet and outlet parameters the T-s graph is designed as follows:

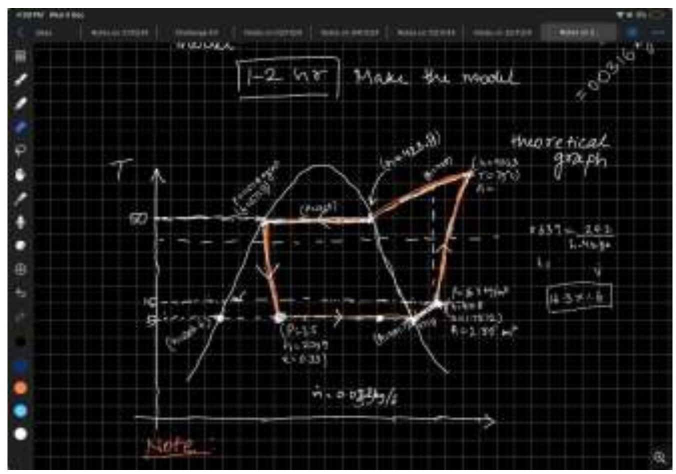


Fig. 3: T-s graph of R134a

- A constant pressure line has been plotted at $5^\circ C$ where the pressure is 3.5 bar which meet the standard curve where h = $401.7 W/m^2K$
- From this point on the graph the liquid is superheated till it reaches $15^\circ C$ where the enthalpy $h = 410.8 W/m^2K$
- Then the liquid undergoes compression isentropically until it reaches a point where $T = 75^\circ C$
- The liquid is then condensed till the point where there is no subcooling i.e. where $h = 271.9 W/m^2K$ and pressure is 13.1 bar.
- Then the liquid passes through the expansion valve which reduces its pressure until it reaches the pressure at 3.5 bar i.e. the pressure line corresponding to the pressure line at evaporation.

4.3 Compressor Selection Explained

When selecting a compressor for an HVAC system[1], several key parameters need to be considered to ensure optimal performance and efficiency:

- Refrigerant Type
- Operating Pressure and Temperature
- Efficiency

Based upon the above parameters, the compressor which deemed to be the best was **Scroll Compressor 2**.

The justification is as follows:

- The refrigerant chosen i.e. R134a is compatible with only two compressors: Scroll Compressor 2 and Scroll Compressor 3.
- Among the two compressors, Scroll Compressor 2 has higher **'Isentropic efficiency'**.
- The chosen compressor also exhibits more displacement compared to the other compressor at the same frequency.

4. Expansion valve

- An expansion valve is a device used in refrigeration and air conditioning systems to regulate the flow of refrigerant into the evaporator.
- It reduces the pressure of the refrigerant, causing it to expand and cool before entering the evaporator coil.
- This expansion leads to a decrease in temperature, enabling the refrigerant to absorb heat from the surroundings.
- The valve adjusts the refrigerant flow based on the system's demand, ensuring optimal evaporator performance and efficiency.

4.5 Evaporator and Condenser Design

The **Evaporator** and **Condenser** are designed with the same principle as that of a **Heat Exchanger**. The R134a refrigerant and moist air are the two fluids that flow through the cross-flow finned heat exchanger.

When the refrigerant acts as cold fluid, the heat exchanger acts as an evaporator and when it acts as hot fluid, the heat exchanger behaves as a condenser. The procedure to design a heat exchanger is as follows:

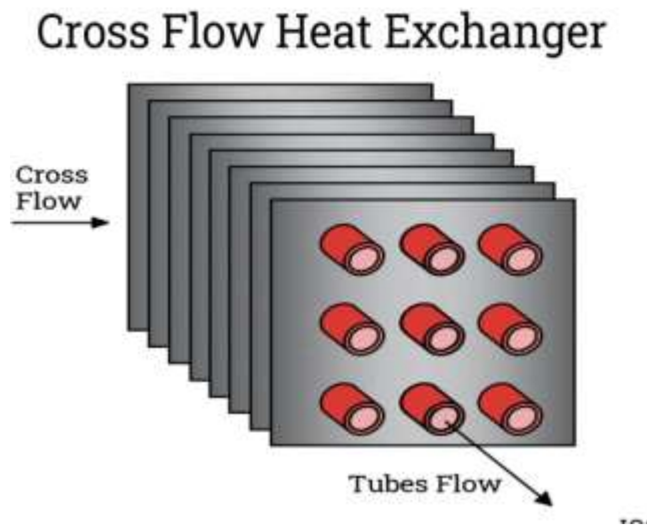


Fig. 4: Cross flow finned heat exchanger

1. Initial Parmeters

- All the initial parameters such as mass flow rate, inlet and outlet temperatures of the fluids, etc. which are required for the subsequent analysis are determined.
- The parameters such as inner diameter of tube, outer diameter of tube, length of tube are initially assumed.

Assumptions:

- For evaporator:
  Inner diameter = 10mm
  Outer diameter = 12mm
  Length of each tube = 1m
  Overall heat coefficient = 25 W/m^2k
- For condenser:
  Inner diameter = 10mm
  Outer diameter = 12mm
  Length of each tube = 1
  Overall heat coefficient = 25 W/m^2k

2. Duty of heat exchanger

- The heat transfer rate is determined from the mass flow rate, average heat transfer coefficient and temperature difference.

$$Q = \dot{m}_c C_{p,c}(T_{c,out} - T_{c,in})$$

$$= \dot{m}_h C_{p,h}(T_{h,in} - T_{h,out})$$

3. Size of the heat exchanger

- The total heat transfer rate Q can be related to the temperature difference between hot and cold fluids as:

$$Q = UA\Delta T$$

$$\Delta T = T_h - T_c$$

- Where U is the overall heat transfer coefficient and A is the reference area (size of heat exchanger). Since ΔT varies with position in the heat exchanger, a log mean temperature difference (LMTD) T_{lm} is necessary

4. Overall Heat Transfer Coefficient U

- The total thermal resistance of the series circuit between two fluids is the sum of all compositions:

$$R_{tot} = R_{conv_h} + R_{conv_c} + R_{foul}$$

  where subscripts h and c represent hold and cold fluid, respectively. The next step is to calculate each thermal resistance in the circuit.
- For convection:

$$R_{conv_h} = 1/h_h A_h$$

$$R_{conv_c} = 1/h_c A_c$$

- Therefore, the overall heat transfer coefficient U is given by:

$$U = 1/AR_{tot}$$

5. Convective heat transfer coefficients

- The convective heat transfer coefficients for hot and cold fluids are to be determined using the relation between Nusselt number,Reynold's number and Prandlt number which relate as follows:

$$Nu = 0.023Re^{4/5}Pr^{2/5}$$

$$Re = \rho vd/\mu$$

$$h = Nuk/D$$

- Where Re is Reynold's Number,Pr is Prnadlt number, Nu is Nusselt number,D is the diameter of tube and ρ is the liquid density.

6. Fin Surface area and number of tubes

- The fin surface area on refrigerant side is given by:

$$A_{ref} = \pi D_{inner} L$$

- where L is length of tube D_{inner} is inner diameter.
- The fin surface area on air side is given by:

$$A_{ref} = \eta_{ef} \pi D_{outer} L$$

- where L is length of tube D_{outer} is outer diameter and η_{ef} is the fin efficiency.
- The number of tubes is given by:

$$N = A_{tot}/A_{tube}$$

- where A_{tot} is the total surface area and A_{tube} is tube area.

4.6 Transfer Equation

Transfer equations[2] help in selecting appropriate control strategies, tuning controllers, and optimizing system response.In the model, The transfer equation is essential to control the shaft speed in the compressor. The transfer equation is obtained from the balance of heat rates inside the house, which can be schematically represented as:

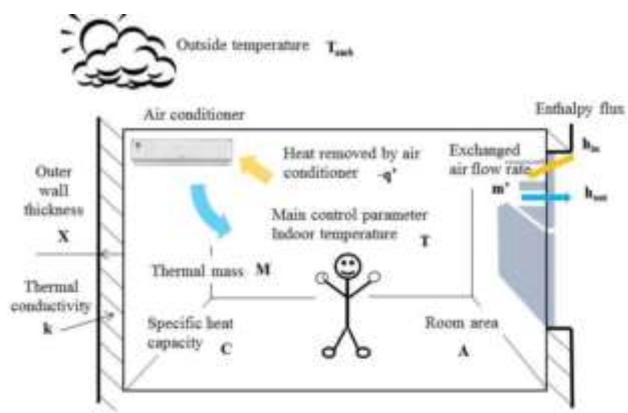


Fig. 5: Mathematical modeling of temperature control in air conditioning space.

Taking the above flow as a reference we develop the mathematical equation based upon the law that the net rate of heat entering into the house equals to the summation of net rate of heat leaving the house and the internal change in the heat with respect to time. Mathematically,

$$\dot{Q_c} = \dot{Q} + MC\frac{dT}{dt}$$

where, $\dot{Q}_c$ represents the conductive heat transfer,$\dot{Q}$ represents the rate at which the heat is lost,M is the thermal mass of the room, C is the Specific heat capacity.

The net rate of heat lost from the room is same as the net rate of heat removed by the evaporator. Hence the rate of heat can be related with the mass flow rate inside the evaporator tube with the following relation:

$$\dot{Q} = \dot{m}\Delta h_{evap}$$

where, $\dot{m}$ represents the mass flow rate in the evaporator tubes and Δh_{evap} represents the enthalpy difference across the evaporator.
we aslo know that,

$$P_{comp} = \dot{m}\Delta h_{comp} = \tau_{comp}\omega(t)$$

where, P_{comp} represents the power of the compressor, τ_{comp} represents the torque on the compressor and $\omega(t)$ represents time varying shaft speed

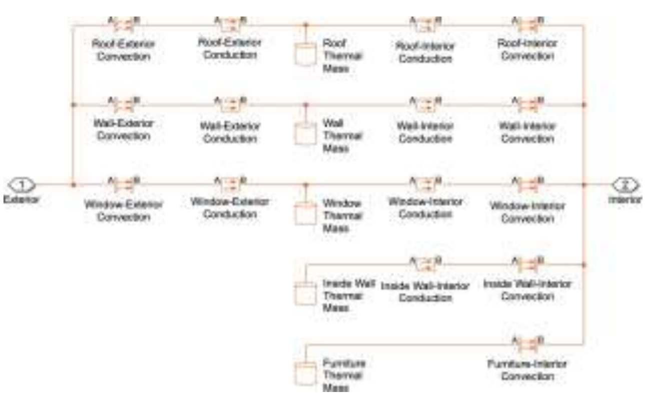


Fig. 6: Thermal resistance network.

As the thermal network is shown, we can compute $\dot{Q}_C$ as:

$$\dot{Q}_C = \dot{Q}_{roof} + \dot{Q}_{furniture} + \dot{Q}_{ext-wall} + \dot{Q}_{int-wall} + \dot{Q}_{window}$$

Considering the exterior is at almost same temperature as that of environment, exterior convection terms are ignored.

The thermal resistances are calculated using the following equations: For convection,

$$R_{conv} = 1/hA$$

where, h is the convective heat transfer and A is the area exposed. For conduction,

$$R_{cond} = l/kA$$

where, k is the thermal conductivity,l is the thickness, A is area exposed.

Upon simplifying all the above equations:

$$MC\frac{dT}{dt} + T*(C_1) + \dot{Q} - T_{amb}*(C_1) = 0$$

Also given that, compressor is 92 percent efficient,

$$\dot{Q} = 0.92\tau(t)\omega(t)$$

where, T_{amb} is ambient temperature,C_1 is the equivalent resistance Applying Laplace transforms, The final transfer function is as follows:

$$\frac{T(s)}{\omega(s)} = \frac{-0.92\tau\omega(s)\Delta h_{evap}}{\left(\frac{MCs}{C_1} + 1\right)\Delta h_{comp}}$$

7. Control Logic

- To regulate the temperature of a house using a compressor drive, we designed a control system around a first-order plant.
- Since the exact transfer function of the system was initially unknown, we began by performing empirical tests to understand the system dynamics.
- Step response testing was used to observe the plant's time constant and steady-state behavior, while ramp and impulse tests were conducted to further characterize its dynamic response.
- From the data collected, a rough transfer function model was derived, approximating the system's behavior in response to input changes.This provided an initial foundation for control system design.
- Using MATLAB's Control System Designer, we applied the **root locus** approach to determine preliminary PID controller parameters (K_p, K_i, K_d) that stabilize the system and meet basic performance criteria.
- The controller was then refined using heuristic tuning methods like the **Ziegler-Nichols** method, which systematically adjusted gains based on the plant's critical oscillation properties.

Finally, MATLAB's PID Tuner was employed for further fine-tuning, iteratively adjusting the controller parameters to achieve the desired steady-state performance and a response time within the constraint of **10 minutes**. Similar control logic using the PID controller were applied to both the **evaporator fan** and the **compressor shaft**. This combined approach ensured a robust controller design, providing precise temperature regulation while maintaining stability and responsiveness.[3]

5 Performance and Output Requirements

All the graphs that represent the output performance are as follows in Figures 9, 10, 11, 13:

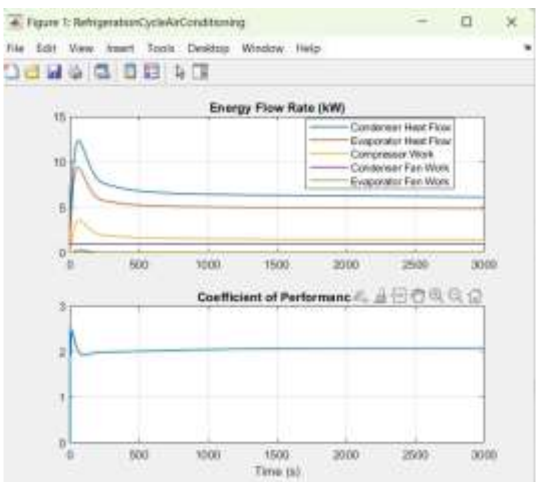


Fig. 7: Energy flow rate and COP at 35°C

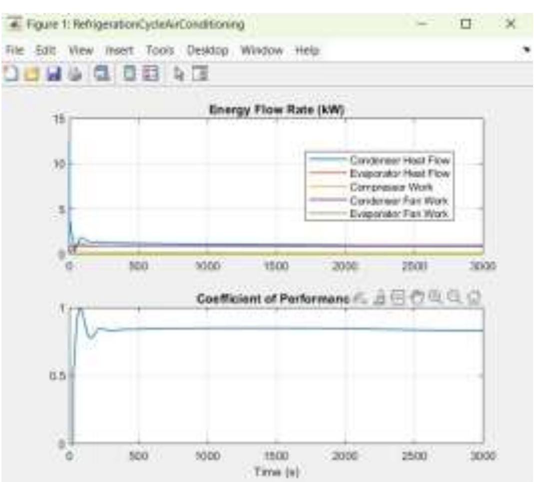


Fig. 8: Energy flow rate and COP at 27°C

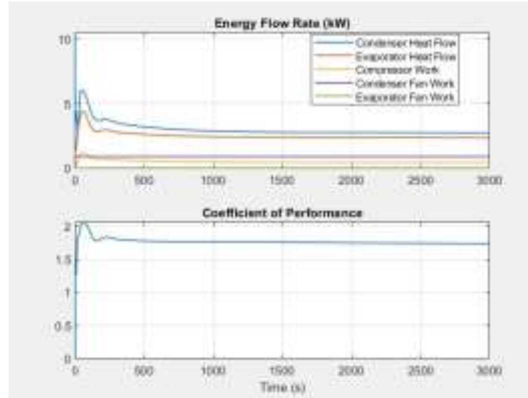


Fig. 9: Energy flow rate and COP at 30°C

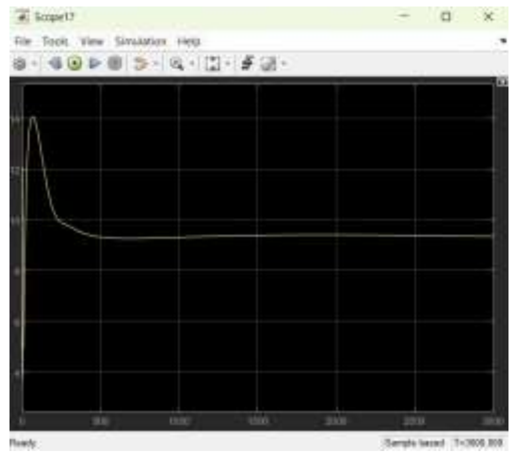


Fig. 10: Superheat at 35°C

Pressure-Enthalpy(P-h) Chart of the Refrigerant Cycle

A P-H (Pressure-Enthalpy) diagram is a graphical representation of the thermodynamic properties of a substance, commonly used in the study of refrigeration cycles and heat engines. In this diagram:

- **Pressure (P)** is plotted on the vertical axis.
- **Enthalpy (h)** is plotted on the vertical axis.

The P-H diagram helps visualize the relationship between pressure and enthalpy in various thermodynamic processes, such as compression, expansion, heating, and cooling. The diagram is often used to analyze refrigeration cycles, as it shows the changes in pressure and enthalpy through various stages like compression, expansion, and heat exchange.

Key regions in a P-H diagram for a refrigerant or working fluid include:

- **Saturated liquid region:** Where the substance is

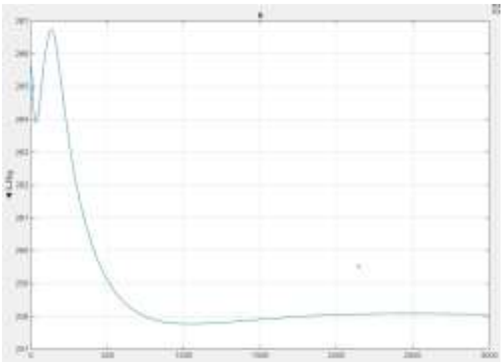


Fig. 11: Cooling capacity

Fig. 12: Dry bulb temperature (Kelvin) on Y-axis and time (Seconds) on X-axis

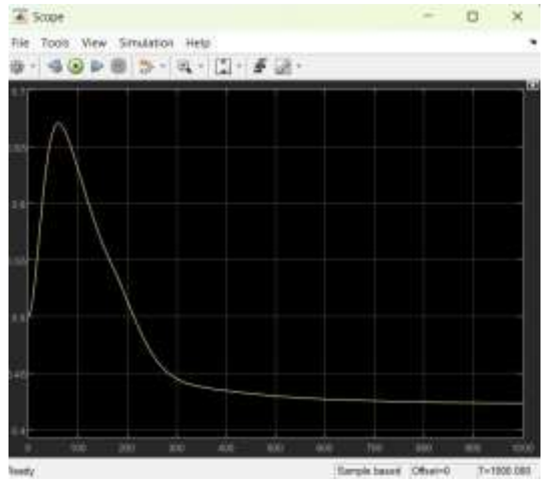


Fig. 13: Wet bulb temperature (Kelvin) on Y-axis and time (Seconds) on X-axis

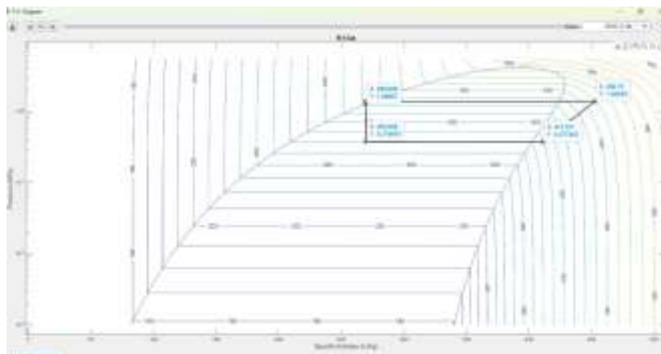


Fig. 14: P-h Graph

in the liquid state at its boiling point.

- **Saturated vapor region:** Where the substance is in the vapor state at its boiling point.
- **Superheated vapor region:**Where the vapor is heated beyond the saturated vapor point.
- **Subcooled liquid region:** Where the liquid is cooled below the saturation temperature.

6 Key Innovations & Environmental Considerations

One of the key innovations in our design is the implementation of a **single PID controller** to regulate both the **evaporator fan speed** and the **compressor shaft speed**, rather than using two separate controllers. This approach simplifies the control system, reducing the complexity of the design while maintaining precise control over both components.

By using a single PID controller, we can effectively manage the relationship between the compressor and fan speeds, optimizing the overall performance of the system. This integration ensures that the cooling and dehumidification processes are dynamically balanced, improving energy efficiency and responsiveness to varying load conditions. The unified control not only reduces hardware requirements but also streamlines system tuning and maintenance, making the system more cost-effective and easier to manage.

Apart from this implementation, we considered several other approaches for our custom control logic including the feasibility of using **Reinforcement Learning** to tune our PID controller but we identified several challenges in implementation and integrating the controller with our model.
Challenges that we identified with using RL for PID Tuning:

- **System Complexity:** The refrigeration system's nonlinear dynamics and multiple interacting components (compressor, fan, refrigerant) make it difficult for the RL agent to efficiently learn optimal PID parameters. This complexity requires significant training data and computational power, leading to slow convergence and high resource demands.
- **Simulink Integration:** Interfacing RL with Simulink adds complexity, as Simulink doesn't natively support RL. The communication between Simulink and the RL framework, especially for real-time data exchange, could be cumbersome, slowing down the learning process.
- **Exploration vs. Exploitation:** RL's exploration-exploitation trade-off may result in poor performance during exploration, potentially leading to system instability or inefficiency before the agent settles on an optimal policy.

We also explored the feasibility of using **Fuzzy Logic** for control over standard PID control and identified the following challenges:

- **Complex Rule Design:** Fuzzy logic requires a comprehensive set of if-then rules to handle multiple variables (temperature, humidity, load), which is time-intensive and requires deep domain expertise.
- **Scalability:** The rule base increases exponentially with additional inputs and outputs, complicating management for complex systems like refrigeration.
- **Tuning Difficulty:** Unlike PID, fuzzy logic lacks standardized tuning methods. Adjusting membership functions and rules relies on trial-and-error or advanced optimization techniques like genetic algorithms.
- **Integration in Simulink:** While supported, fuzzy logic demands real-time data processing for inference and defuzzification, adding computational overhead and potential latency in simulations.
- **Stability Challenges:** Stability depends on carefully designed rules and membership functions, unlike PID, which follows well-defined stability criteria. Poor designs can cause oscillations or instability.
- **Performance in Dynamic Conditions:** Fuzzy logic may struggle to generalize across varying operating conditions (e.g., load and temperature changes) without frequent updates to rules and membership functions

Given the limitation of time, we decided to focus our approach on traditional PID tuning methods for implementing our control logic.

For environmental considerations, although refrigerants like R1234yf are much more environmentally friendly, they are extremely expensive. We chose to go with R134a for our design given its cost-effectiveness and properties that were discussed earlier.

7 Cost Analysis

7.1 Capital Expenditure

1. Expansion Valve

Considering the working conditions of the modeled air conditioning cycle and chosen refrigerant, the Danfoss TXV TN2 Thermostatic Expansion Valve suits the best, with a rated evaporating temperature of 4°C and condensation temperature of 38°C, and a maximum working pressure of 34 bar.

- It amounts to a cost of **INR 3097**

Fig. 17: Expansion Valve

2. Compressor

Embraco Scroll Compressor (SE6018GK-C) - suitable for the chosen refrigerant R134a and the given load at an evaporating temperature of 5°C and condensing temperature of 50°C.
It amounts to a cost of **INR.10,000**

Fig. 17: Compressor

3. Indoor Blower (evaporator fan)

We choose an evaporator fan suitable for 1.25-ton cooling load. It amounts to a cost of **INR 10,000**.

Fig. 18: Evaporator Fan

4. Outdoor Fan Motor

We can choose an Outdoor fan with a nominal speed 1500 RPM, suitable for 1.25-ton cooling load

- It amounts to a cost of **INR 3,900**

5. Thermistor (Temperature Sensor)

A typical thermistor costs around **INR 299**.

6. Evaporator

- **Tubes**
  - Tube length per segment = 1m
  - Number of tubes = 25
  - Hence total tube length = 1×25 = 25 m
  - Tube inner diameter = 0.01m
  - Tube thickness = 0.002m, Hence volume of tube required would be
    V=length × π × inner diameter × thickness
    Upon calculating, we get V = **1570.8** cm^3
  - Material used for tubes: **Copper** (due to its high thermal conductivity)
  - Density of copper = 8.96 g/cm^3 Hence, Mass of copper required = **14.07 kg**
  - As per the London Metal Exchange, the cost of Copper metal is $9122.5 per tonne. Hence Total cost = $128.353 = **INR 10868**
- **Fins**
  - Total fin surface area facing the air = 5.2m^2
  - Total fin surface area facing the refrigerant (inside the tubes) = 4.7124m^2
  - Material used for fins : **Aluminium** (Light and cost effective)

- Assuming fin thickness = 1mm,
  Volume = Surface Area × thickness
  Hence, Volume = $\mathbf{9912.4cm^3}$
- Density of aluminium = $2.7g/cm^3$. Hence, Mass = **26.763 kg**
- As per the London Metal Exchange, the cost of Aluminium metal is $2603.5 per tonne. Hence, Cost = $69 = **INR 5842**

Hence, total cost for evaporator = Cost for Tubing + Cost of Fins = **INR 16710**

7. Condenser

- **Tubes**
  - Tube length per segment = 1m
  - Number of tubes = 25
  - Hence total tube length = 1×25 = 25 m
  - Tube inner diameter = 0.01m
  - Tube thickness = 0.002m, Hence volume of tube required would be
    V=length × π × inner diameter × thickness
    Upon calculating, we get $V = 1570.8cm^3$
  - Material used for tubes: **Copper** (due to its high thermal conductivity)
  - Density of copper = $8.96g/cm^3$ Hence, Mass of copper required = **14.07 kg**
  - As per the London Metal Exchange, the cost of Copper metal is $9122.5 per tonne. Hence Total cost = $128.353 = **INR 10868**
- **Fins**
  - Total fin surface area facing the air = $3.25m^2$
  - Total fin surface area facing the refrigerant (inside the tubes) = $4.7124m^2$
  - Material used for fins : **Aluminium** (Light and cost-effective)
  - Assuming fin thickness = 1mm,
    Volume = Surface Area × thickness
    Hence, Volume = $7962.4cm^3$
  - Density of aluminium = $2.7g/cm^3$. Hence, Mass = **21.49 kg**
  - As per the London Metal Exchange, the cost of Aluminium metal is $2603.5 per tonne. Hence, Cost= $56 = **INR 4742**

Hence, total cost for condenser = Cost for Tubing + Cost of Fins = **INR 15,610**

Given the above considerations, the cost breakdown and total cost comes to a sum of **INR 51,298**

Component	Cost
Expansion Valve	₹3,097
Compressor	₹10,000
Indoor blower	₹1,981
Outdoor fan motor	₹3,900
Evaporator	₹16,710
Condenser	₹15,610
Total	₹51298

Fig. 20: Cost Breakdown

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