ALBATROSS ENERGETICS

INNOVATIVE COOLING
AND
DEHUMIDIFICATION SOLUTIONS

TEAM - 22
END TERM REPORT

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A - Introduction

The growing demand for energy-efficient and environmentally friendly cooling solutions has led to significant advancements in vapor compression-based air conditioning (AC) systems. These systems are widely used for their ability to provide effective cooling and dehumidification in residential, commercial, and industrial applications. This project aims to design and model a detailed vapor compression air conditioning system in MATLAB/Simulink, focusing on achieving optimal cooling performance, dehumidification efficiency, and energy conservation under varying load conditions.

The system incorporates a single compressor and leverages the thermodynamic principles of the vapor compression cycle to regulate indoor temperatures and humidity levels. A critical aspect of the design is the selection of appropriate components, refrigerant, and control logic to ensure the system operates within safe limits while meeting performance and efficiency targets.

This report outlines the methodology for modeling the vapor compression AC system, including component design, refrigerant selection, and control strategy implementation. Simulation results will be analyzed to assess the system's performance, highlighting key metrics such as cooling capacity, coefficient of performance (COP), and dehumidification efficiency.

B- Methodology

Model Description : The design of a vapor compression refrigeration system in Simulink begins with defining system requirements, such as cooling capacity, temperature limits, and the choice of refrigerant . Simscape or Simscape Fluids libraries are utilized to model the key components, including the compressor, condenser, expansion valve, and evaporator. Thermodynamic parameters, such as pressure ratios and mass flow rates, are configured, and appropriate heat transfer models are selected to simulate refrigerant flow and phase changes accurately. Sensors and controllers are integrated to regulate temperature and pressure effectively. The model is validated through simulations under various operating conditions, ensuring optimal performance and efficiency.

1. **Refrigerant Selection**

The selection of the most suitable refrigerant from the provided list was carried out by carefully evaluating parameters such as isentropic efficiency, coefficient of performance (COP), global warming potential (GWP), and ozone depletion potential (ODP), as outlined in the accompanying table.

	R134a	R290	R600	R407C	R513A	R454C	R1234yf	R50C
Boiling Point (°C)	-36.1	-42.16	−11.7	-43.8	-29.7	-45.7	-19.3	-24.2
Critical Temperature (°C) and Pressure (MPa)	101 / 4.509	134.7/3.6	134.7/3.6	86.7/4.5	96.5/3.7	85.2/46.2	101/3.38	126/4.99
Latent Heat (kJ/kg)	215	425	360	220	215	740.4	184	194.5
COP	~3 to 5	~4 to 6	~4 to 6	~3.5 to 4	~3 to 5	~3 to 5	~ 3.2 to 4	~ 3 to 4
GWP	1430	~3	~3	1770	573	148	4	6
ODP	0	0	0	0	0	0	0	0
Flamm-ability	A1	A3	A3	A1	A2L	A2L	A3	A3
Toxicit-y	A1	A3	A3	A1	A1	A1	A	A

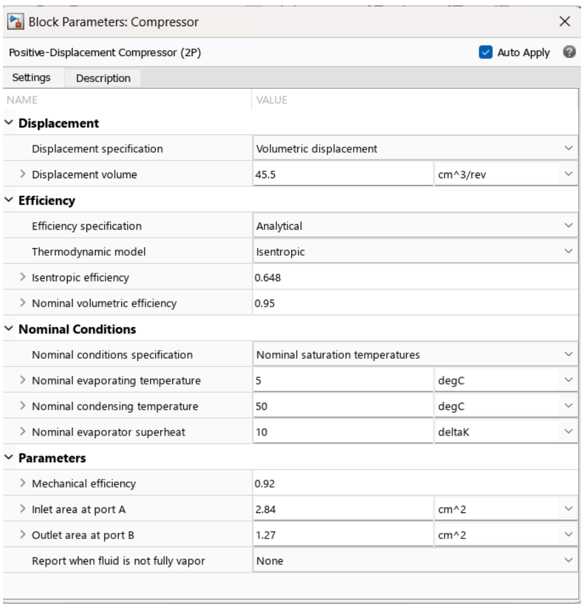
Key parameters for deciding the refrigerant :

1. **Boiling Point:** Low boiling point for efficient heat absorption at the evaporator.
2. **Critical Temperature and Pressure:** Sufficiently high to avoid reaching the critical point.

3. **Latent Heat of Vaporization:** Higher values improve cooling capacity for a given mass flow rate.
4. **Coefficient of Performance (COP):** High COP ensures better energy efficiency.
5. **Global Warming Potential (GWP):** Low GWP to minimize greenhouse gas emissions.
6. **Ozone Depletion Potential (ODP):** Zero or low ODP to protect the ozone layer.
7. **Flammability:** Refrigerants should have a low flammability rating (e.g., A1 class).
8. **Toxicity:** Low toxicity for safety during leakage or exposure.

After evaluating the key parameters for refrigerant selection, R1234yf , R290 and R134a emerged as ideal choices for the vapor compression system. The model was divided into two parts: Compressor and Condenser ,Evaporator and Expansion Valve. This approach helped finalize the system parameters and verify whether the desired **P-H diagram** was achieved. Additionally, it served as a testing platform to evaluate and eliminate refrigerants based on their performance in each setup which led to the selection of R1234yf as an ideal refrigerant.

2. Compressor Selection : Based on the selection of the refrigerant, Scroll Compressor 2 was chosen from the provided list with the following details :



Block Parameters: Compressor

Positive-Displacement Compressor (2P) — Auto Apply

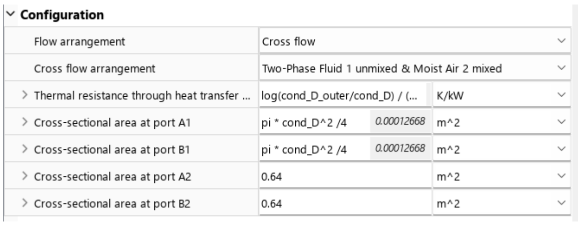
Settings | Description

NAME	VALUE	
Displacement		
Displacement specification	Volumetric displacement	
Displacement volume	45.5	cm^3/rev
Efficiency		
Efficiency specification	Analytical	
Thermodynamic model	Isentropic	
Isentropic efficiency	0.648	
Nominal volumetric efficiency	0.95	
Nominal Conditions		
Nominal conditions specification	Nominal saturation temperatures	
Nominal evaporating temperature	5	degC
Nominal condensing temperature	50	degC
Nominal evaporator superheat	10	deltaK
Parameters		
Mechanical efficiency	0.92	
Inlet area at port A	2.84	cm^2
Outlet area at port B	1.27	cm^2
Report when fluid is not fully vapor	None	

3. Condenser Modelling : The area of the condenser inlet and outlet ports were determined from the stud discharge diameter of the compressor:

A1=B1=π*(0.25”)^2=0.00012668 m^2

A2=B2 = (Condenser Duct Width)*(Condenser Duct Height) = 0.8 * 0.8 = 0.64 m^2



Configuration		
Flow arrangement	Cross flow	
Cross flow arrangement	Two-Phase Fluid 1 unmixed & Moist Air 2 mixed	
Thermal resistance through heat transfer ...	log(cond_D_outer/cond_D) / (...	K/kW
Cross-sectional area at port A1	pi * cond_D^2 /4 — 0.00012668	m^2
Cross-sectional area at port B1	pi * cond_D^2 /4 — 0.00012668	m^2
Cross-sectional area at port A2	0.64	m^2
Cross-sectional area at port B2	0.64	m^2

Since tubes in condenser help increase the heat transfer surface area , overall efficiency , better flow distribution and at the same time ensure proper pressure drop, one tube was used being coiled multiple times having a total length of **60m**. This is calculated by multiplying the duct length with the number of tubes per row and the number of rows. Usually in air conditioners, copper tubes used have no internal fins - in which case, total fin surface area 0. The pressure was initialised to **1.302MPa**, corresponding to the state variables from the P-H Diagram.

Number of tubes per row and number of tube segments per row were optimised to find the best condition for maximum heat rejection. As fins help in effectively removing heat, a high fin surface area was chosen while not crowding the geometry. The area is **35m^2**.

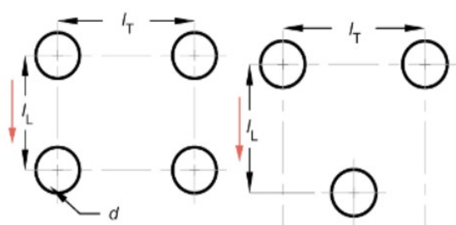
This area was calculated using the basic definition of efficiency :

$$\eta = q/hA\theta$$

Inline arrangement of tubes has been selected to ensure proper heat transfer and the equations used for the calculation of lateral pitch is :

$$m=\rho V(N_t S_t L)$$

Cross-section of Tubing with Pitch Measurements



Other governing equations include :

- $Nu = 0.404L_q^{1/3}(Re+1/Re+1000)^{0.1}$
- $Lq= 1.18Pr[(4l_t\pi\text{-}D)/(l_L)]Hg(Re)$

Two-Phase Fluid 1

Number of tubes	1		
Total length of each tube	60		m
Configurability	Compile-time		
Tube cross section	Circular		
Tube inner diameter	cond_D	0.0127	m
Pressure loss model	Correlation for flow inside tubes		
Local resistance specification	Aggregate equivalent length		
Aggregate equivalent length of local resis...	0.1 * cond_num_tube_rows * c...		m
Internal surface absolute roughness	15e-6		m
Laminar flow upper Reynolds number limit	2000		
Turbulent flow lower Reynolds number limit	4000		
Heat transfer coefficient model	Colburn equation		
Coefficients [a, b, c] for a*Re^b*Pr^c in li...	[.023, .8, .33]	[0.023,0.8,0.33]	
Coefficients [a, b, c] for a*Re^b*Pr^c in m...	[.05, .8, .33]	[0.05,0.8,0.33]	
Coefficients [a, b, c] for a*Re^b*Pr^c in v...	[.023, .8, .33]	[0.023,0.8,0.33]	
Fouling factor	0.1		K*m^2/kW
Total fin surface area	0		m^2
Fin efficiency	0.75		
Initial fluid energy specification	Vapor quality		
Initial two-phase fluid pressure	1.302		MPa
Initial two-phase fluid vapor quality	[1,0]		

Moist Air 2

Flow geometry	Flow perpendicular to bank of circular tubes		
Tube bank grid arrangement	Inline		
Number of tube rows along flow direction	15		
Number of tube segments in each tube row	5		
Length of each tube segment in a tube row	cond_duct_W	0.8	m
Tube outer diameter	cond_D_outer	0.0137	m
Longitudinal tube pitch (along flow directi...	cond_long_pitch	0.0508	m
Transverse tube pitch (perpendicular to fl...	cond_trans_pitch	0.0508	m
Pressure loss model	Correlation for flow over tube bank		
Heat transfer coefficient model	Correlation for flow over tube bank		
Fouling factor	0.1		K*m^2/kW
Total fin surface area	35		m^2
Fin efficiency	0.75		
Initial moist air pressure	0.101325		MPa
Initial moist air temperature	T_env	35	degC
Initial moisture specification	Relative humidity		
Initial moist air relative humidity	0.5		
Initial trace gas specification	Mass fraction		
Initial moist air trace gas mass fraction	0.001		
Relative humidity at saturation	1		

4. Thermostatic Expansion Valve Modelling : Using the given parameters for nominal operating temperatures, the expansion valve parameters were set. The nominal mass flow rate was calculated by measuring the density at the evaporator outlet using a thermodynamic property sensor and using the compressor volumetric displacement of **45.5 cm^3/rev.** The nominal mass flow rate was found to be **0.055kg/s** which was later verified using a mass flow rate sensor. The maximum mass flow rate at high load conditions was similarly calculated to be **0.12kg/s**.

A maximum operating temperature of 20C was put to ensure that the operating envelope of the compressor was not breached.

Parameters

Valve parameterization	Nominal capacity, superheat, and operating conditions	
Capacity specification	Mass flow rate	
Nominal mass flow rate	0.055	kg/s
Maximum mass flow rate	0.14	kg/s
Nominal pressure specification	Pressure at specified saturation temperature	
Nominal condensing (saturation) tempera...	50	degC
Nominal evaporating (saturation) temper...	5	degC
Nominal condenser subcooling	0	deltaK
Nominal (static + opening) evaporator su...	10	deltaK
Static (minimum) evaporator superheat	2	deltaK
MOP limit	On - Specify maximum operating temperature	
Maximum evaporating (saturation) tempe...	15	degC
Pressure equalization	Internal pressure equalization	
☐ Bulb temperature dynamics		
Leakage flow fraction	1e-6	
Smoothing factor	0.01	
Laminar flow pressure ratio	0.999	
Inlet phase change time constant	0.1	s
Cross-sectional area at ports A and B	2.8553	cm^2

5. Evaporator Modelling : The area of the evaporator inlet and outlet ports were determined from the stud discharge diameter of the compressor:

$A1=B1 =\pi*(0.375")^2=0.00028553 \text{ m}^2$

$A2=B2 = (\text{Evaporator Duct Width})*(\text{Evaporator Duct Height}) = 0.8 * 0.8 = 0.64 \text{ m}^2$

Configuration

Flow arrangement	Cross flow	
Cross flow arrangement	Two-Phase Fluid 1 unmixed & Moist Air 2 mixed	
Thermal resistance through heat transfer ...	log(evap_D_outer/evap_D) / (2...	K/kW
Cross-sectional area at port A1	2.8553	cm^2
Cross-sectional area at port B1	2.8553	cm^2
Cross-sectional area at port A2	0.64	m^2
Cross-sectional area at port B2	0.64	m^2

One tube was coiled back and forth for a total length of 40m, achieving an optimal balance between heat absorption and compactness. The initial pressure was specified as **0.373 MPa**. Additionally, the fin surface area was set to zero, as there are no fins present within the refrigerant tubes.

Two-Phase Fluid 1

Number of tubes	1		
Total length of each tube	40		m
Configurability	Compile-time		
Tube cross section	Circular		
Tube inner diameter	evap_D	0.019	m
Pressure loss model	Correlation for flow inside tubes		
Local resistance specification	Aggregate equivalent length		
Aggregate equivalent length of local resis...	0.5 * evap_duct_H * evap_num...		m
Internal surface absolute roughness	15e-6		m
Laminar flow upper Reynolds number limit	2000		
Turbulent flow lower Reynolds number limit	4000		
Heat transfer coefficient model	Correlation for flow inside tubes		
Fouling factor	0.1		K*m^2/kW
Total fin surface area	0		m^2
Fin efficiency	0.75		
Initial fluid energy specification	Temperature		
Initial two-phase fluid pressure	0.373		MPa
Initial two-phase fluid temperature	[5,15]		degC

The number of tubes per row and the number of tube segments per row were optimized to identify the best configuration for maximum heat rejection. To enhance heat removal efficiency, a large fin surface area was selected, ensuring the geometry remained compact. Based on the principles of heat transfer through fins, the fin surface area was calculated to be **50 m²**.

Moist Air 2			
Flow geometry	Flow perpendicular to bank of circular tubes		
Tube bank grid arrangement	Inline		
Number of tube rows along flow direction	10		
Number of tube segments in each tube row	5		
Length of each tube segment in a tube row	0.8		m
Tube outer diameter	evap_D_outer	0.021	m
Longitudinal tube pitch (along flow directi...	evap_long_pitch	0.056	m
Transverse tube pitch (perpendicular to fl...	evap_trans_pitch	0.056	m
Pressure loss model	Correlation for flow over tube bank		
Heat transfer coefficient model	Correlation for flow over tube bank		
Fouling factor	0.1		K*m^2/kW
Total fin surface area	50		m^2
Fin efficiency	0.75		
Initial moist air pressure	0.101325		MPa
Initial moist air temperature	T_house_init	35	degC
Initial moisture specification	Relative humidity		
Initial moist air relative humidity	0.5		
Initial trace gas specification	Mass fraction		
Initial moist air trace gas mass fraction	0.001		
Relative humidity at saturation	1		

6. External Environment:

The external environment consisted of a moist air reservoir and a fan. The fan parameters were selected to strike an optimal balance between the coefficient of performance (COP) and cooling efficiency. The flow areas were defined by the ducts. The nominal RPM was set to **2500**, and the nominal volumetric flow rate was set to **1.5 m³**. The efficiency was increased from the default value of 0.4 to **0.6**, reflecting the advancements in modern fan technologies that allow for higher mechanical efficiencies.

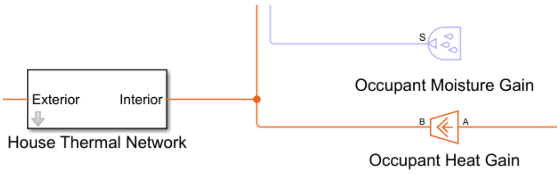
Parameters			
Fan parameterization	Static pressure and flow rate at reference shaft speed		
Shaft power specification	Fan efficiency		
Nominal volumetric flow rate	1.5		m^3/s
Nominal static pressure gain	200		Pa
Nominal efficiency	.6	0.6	
Maximum static pressure gain at zero flow	300		Pa
Maximum volumetric flow rate at zero pre...	2		m^3/s
Reference shaft speed	2500		rpm
Minimum shaft speed threshold as fractio...	1e-2		
Fan diameter scale factor	1		
Mechanical orientation	Positive angular velocity of port R relative to port C correspo		
Inlet flow area (port A)	duct_W^2	0.1225	m^2
Outlet flow area (port B)	duct_W^2	0.1225	m^2

7. House Subsystem:

The house subsystem includes a similar fan setup, along with various heat loads, latent and sensible. All these parameters can be varied for accurate simulation of various loading conditions.

The flow rate for the indoor fan is set to **0.75m³**, and a lower RPM of **1500** is used for low noise levels.

Parameters			
Fan parameterization	Static pressure and flow rate at reference shaft speed		
Shaft power specification	Fan efficiency		
Nominal volumetric flow rate	.75	0.75	m^3/s
Nominal static pressure gain	200		Pa
Nominal efficiency	.6	0.6	
Maximum static pressure gain at zero flow	300		Pa
Maximum volumetric flow rate at zero pre...	.8	0.8	m^3/s
Reference shaft speed	2000		rpm
Minimum shaft speed threshold as fractio...	1e-2		
Fan diameter scale factor	1		
Mechanical orientation	Positive angular velocity of port R relative to port C correspo		
Inlet flow area (port A)	duct_W^2	0.1225	m^2
Outlet flow area (port B)	duct_W^2	0.1225	m^2



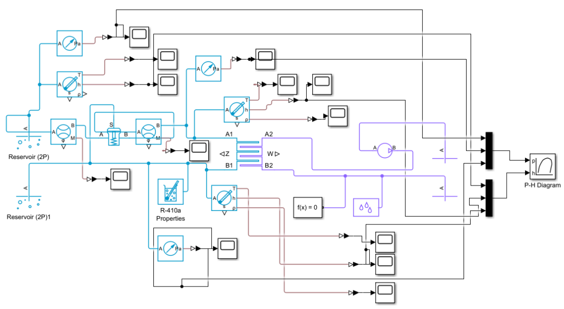
The house thermal network and occupant heat and moisture gain blocks give adequate control over latent and sensible loads.

A total of **200 kg** of furniture was placed inside the house, accompanied by a heat gain of **500 W** from occupants and a moisture gain of **0.04 g/s**. The volume of the house was **50 m³**.

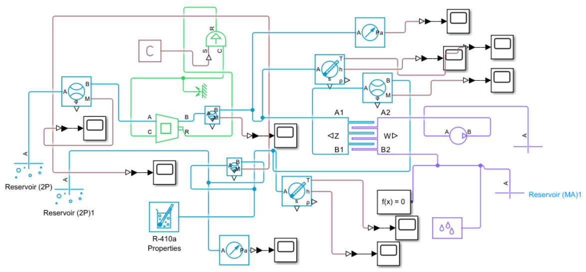
8. Testing Models:

Two testing models were employed in order to verify and optimise design parameters and variables.

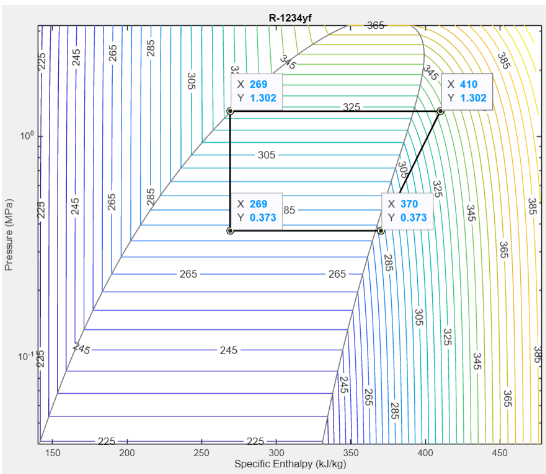
<u>Expansion valve- Evaporator subsystem</u>



This subsystem models the flow of refrigerant between the condenser and the compressor. Both the refrigerant and moist air are drawn from reservoirs at specified temperatures and pressures, with a series of sensors verifying each parameter. The PH diagram of this subsystem closely aligns with the theoretical model, confirming the system's feasibility. A similar testing setup was established for the compressor and condenser.



A similar approach is used. The sensors validated the data and the PH diagram was satisfactory.
Thus, the theoretically feasible PH diagram is:



The **theoretical COP** (Coefficient of Performance) obtained is **2.92**. The final COP depends upon other factors like indoor and outdoor fans, and slight differences in state variables.

The two models are combined using a **liquid reservoir** with a volume of **0.03m³**. This reservoir allowed for fluctuations in refrigerant levels while the system was attaining steady state.

8.1 Data Gathered

Sensors and calculations yielded a host of data and results to optimise. Mass flow rate sensors gave accurate mass flow data. Thermodynamic and pressure sensors gave enthalpy and pressure data respectively to plot PH diagrams. The heat transfer and work input was used to calculate the COP and temperature sensors were used to track the indoor temperature. The wet bulb temperature and relative humidity data was found using the moist air humidity sensor

9. Control logic

In this air conditioning system, a PI controller is implemented to achieve the desired room temperature by dynamically adjusting the speed of the compressor, indoor fan, and outdoor fan. The proportional gain (K_p) is fixed at 1, and the integral gain (K_i) is scheduled based on the target temperature (T_desired) to improve precision and responsiveness. The given gain values have been determined through manual tuning to ensure optimal performance under varying conditions.

The inclusion of an integral term in the controller was necessary to eliminate the steady-state error observed when only proportional control was used. Proportional control alone was insufficient to fully correct the temperature difference, especially under steady-state conditions. The integral term accumulates the error over time and adjusts the control action accordingly, ensuring the system reaches and maintains the desired temperature.

9.1 Gain Schedule and Lookup Table

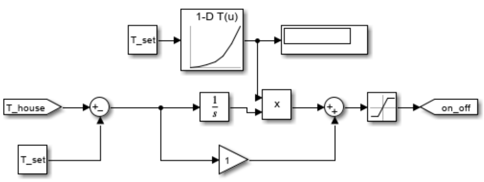
Gain scheduling is employed by defining specific Ki values for discrete temperature breakpoints, as shown below:

- Breakpoints 1 (Temperature in °C): [17 19 21 23 25 27 29]
- Table Data (Corresponding Ki values):[0.00004 , 0.00005 , 0.000064 , 0.00009 , 0.00014 , 0.00030 0.0004]

A lookup table is used to interpolate the Ki values for temperatures within the range, ensuring smooth transitions and precise control across the entire operating range of the system. Linear interpolation allows the controller to determine the most appropriate Ki value for any temperature, even if it is not explicitly defined in the breakpoints.

9.2 Benefits of Gain Scheduling and Interpolation

1. **Elimination of Steady-State Error**: The integral action ensures the system maintains the desired temperature without deviation over time.
2. **Improved Precision:** By tailoring the integral gain to the target temperature, the system achieves more accurate control, reducing overshoot and settling time.
3. **Dynamic Adaptability:** Gain scheduling enables the system to respond effectively to varying operating conditions, such as large or small temperature differences.
4. **Enhanced Stability:** Using a lookup table with interpolated Ki values ensures smooth changes in control action, avoiding abrupt shifts that could destabilize the system.
5. **Energy Efficiency**: Optimized PI parameters help minimize unnecessary energy consumption by avoiding overcorrection or prolonged operation at high power.



C- Performance Metrics

ISEER Calculation Steps :

1. **Data Preparation**
2. **Cooling Capacity and Power Consumption**
3. **Energy Output and Input For Each Bin**

$$\text{ISSER} = \frac{\sum(\text{Bin Hours} \times \text{COP at each temperature bin})}{\text{Total Bin Hours}}$$

Following values were calculated for estimating the value of ISEER :

Temperature	Fraction	Q_evap	Work Input	COP	Bin hours
28	10.4	2.186	0.5533	3.950840412	167
29	9.4	2.209	0.5874	3.760640109	150
30	7.8	2.231	0.6219	3.587393472	125
31	6.5	2.253	0.6574	3.427137207	104
32	5.4	2.274	0.6944	3.274769585	86
33	4.2	2.296	0.7321	3.136183581	67
34	3.4	2.317	0.7717	3.002462097	54
35	2.9	2.338	0.812	2.879310345	46
36	2.3	2.359	0.8542	2.761648326	36
37	1.7	2.38	0.8974	2.652106084	28
38	1.4	2.4	0.9423	2.546959567	22
39	1	2.42	0.9887	2.447658542	16
40	0.8	2.44	1.037	2.352941176	12
41	0.5	2.46	1.087	2.263109476	9
42	0.3	2.481	1.139	2.178226514	6
43	0.2	2.502	1.195	2.093723849	3
SUM	**58.2**				**931**

Using the mentioned formula the value of ISSER was found to be 3.385.

EER Calculations :

The Energy Efficiency Ratio (EER) is a measure of how efficiently a vapor-compression refrigeration (VCR) system operates. It is defined as the ratio of the cooling capacity to the power input, typically expressed as:

EER=Cooling Capacity (Btu/hr) / Power Input (Watts)

Now , $Q_c(kJ/s)=M\cdot(h_1-h_4)$ where M is the mass flow rate and h1 and h4 are the enthalpies across evaporator

Therefore , $Q_c(Btu/hr)=Q_c(kJ/s)\cdot 3412$

Average EER = ISEER* 3.412

ISEER = 3.385

Average EER= 11.55

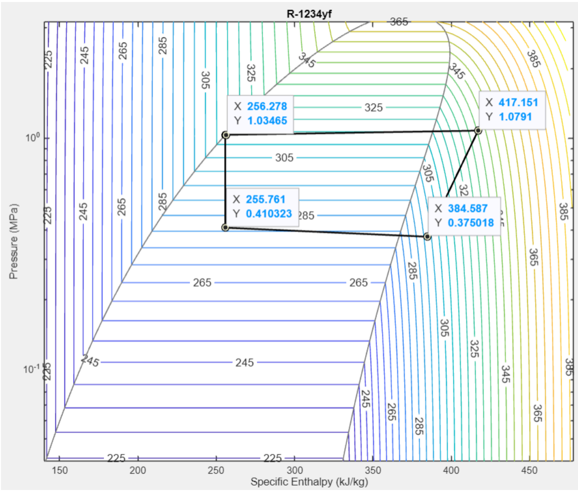
D -Performance and Output Requirements

All performance parameters were evaluated under the ISEER (Indian Seasonal Energy Efficiency Ratio) testing conditions, which are designed to simulate real-world operating environments for air conditioning systems. The specific conditions include: Outdoor Temperature- 35 deg C

Indoor Dry Bulb Set Point - 27 deg C

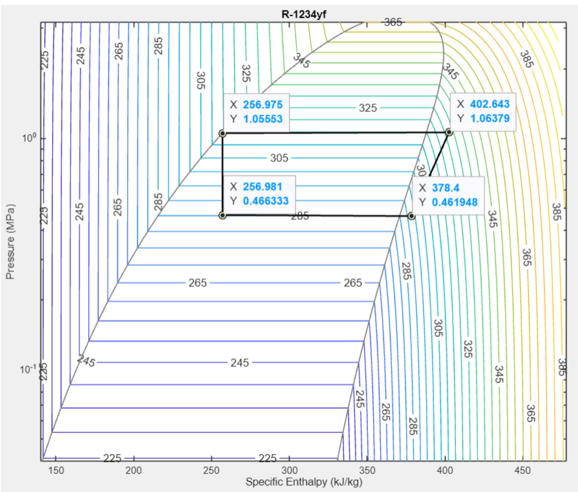
Wet Bulb Set Point - 19 deg C

While cooling



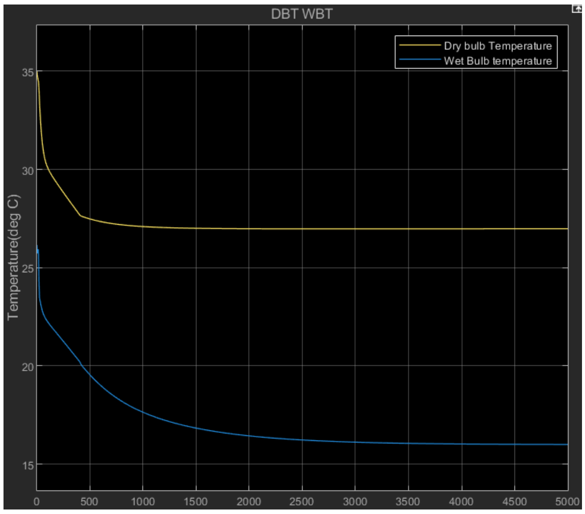
The P-H diagram illustrates the cooling cycle of R-1234yf refrigerant. It shows the relationship between pressure and specific enthalpy, with isobars and isenthalpic curves representing lines of constant pressure and specific enthalpy, respectively. The highlighted points and lines likely represent the condenser, expansion valve, evaporator, and compressor stages of the refrigeration cycle.

Steady State (Temperature maintained)



The P-H diagram illustrates the cooling cycle of R-1234yf refrigerant under steady-state conditions. This means that the system has reached a stable state where all parameters remain constant over time. The highlighted points and lines represent the condenser, expansion valve, evaporator, and compressor stages of the refrigeration cycle under these steady-state conditions.

DBT and WBT



The DBT curve decreases over time. This is typical of the **evaporator** phase in a vapor compression refrigeration (VCR) system, where the refrigerant absorbs heat from the air, lowering its temperature. The WBT also decreases, indicating a reduction in the moisture content of the air. This suggests that the system is **removing humidity**, due to condensation on the evaporator coil, which is common in both cooling and dehumidification modes.

The increasing gap between the DBT and WBT indicates a significant **increase in the air's relative dryness**, which is typical when the system is not only cooling but also dehumidifying the air. The greater the gap, the lower the moisture content in the air.

The combined decrease in both DBT and WBT points to the VCR system performing efficiently, providing both **temperature reduction and moisture removal**, which is critical in environments requiring precise humidity control.

Cooling delivered and power consumption

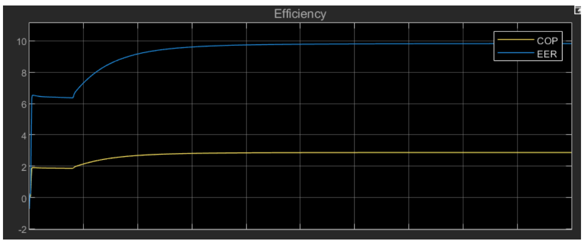
Initially, the heat transfer rate is around **6 kW**, and it decreases sharply before stabilizing. This suggests that the evaporator is absorbing heat initially at a high rate, but as the system reaches equilibrium, the absorption reduces.

The work input to the compressor starts at a higher value of approximately **2.5 kW** and then decreases slightly before stabilizing at around **0.8 kW**. This reflects the compressor's initial high work demand to compress the refrigerant and its eventual stabilization as the system operates at a steady state.

The work input for the outdoor fan starts at about **0.5 kW**, decreases to nearly **0.2 kW**, and then remains constant. This shows the fan's energy consumption drops as the system reaches a stable operation mode.

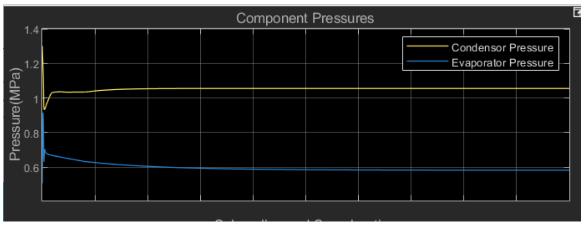
Similar to the outdoor fan, the work input to the indoor fan remains almost constant after a brief decrease, indicating **steady operation** once the system stabilizes.

EER



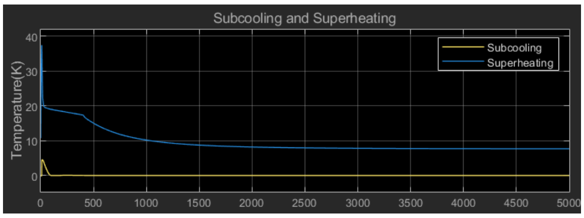
The graph illustrates the transient behavior of a refrigeration system's efficiency metrics, COP (Coefficient of Performance) and EER (Energy Efficiency Ratio), over time. Initially, the system undergoes a transient phase characterized by a rapid increase in both COP and EER. This initial surge is likely due to the system's startup and stabilization process, where the components reach their optimal operating conditions. As the system progresses towards steady-state, the rate of increase in COP and EER diminishes, eventually leveling off at a constant value.

Condenser evaporator saturation temperatures and pressures



The condenser consistently maintains a higher pressure than the evaporator, as expected in a vapor-compression cycle where the refrigerant pressure is elevated by the compressor. The observed transients reflect stabilization processes, including heat transfer and refrigerant flow regulation. The steady-state pressures indicate proper operation of the system.

Superheating and subcooling



The initial transients indicate the system's stabilization phase, where heat transfer and flow dynamics adjust. Steady-state values confirm proper operation, as subcooling ensures efficient condensation, and superheating prevents liquid refrigerant from entering the compressor. This graph is critical for diagnosing performance, identifying imbalances, or optimizing system efficiency.

E - Key Innovations and Environmental Considerations

After carefully comparing various refrigerants, on paper it was clear what should be our choice. Despite that we decided to try a few other refrigerants with promising potential, which led us to our final choice. Our data was validated using our 2 separate models which we finally combined into one at the end.

A careful balance was struck between achieving higher efficiencies and optimizing the cooling rate. Although a very slow cooling rate would result in a significantly higher coefficient of performance (COP), it would not be a practical or desirable solution for consumers, as it would lead to extended cooling times and potentially affect comfort. Thus, the design aimed to find an optimal cooling rate that maintains both efficient energy use and user satisfaction.

Thermal comfort and PMV (Predicted mean vote) values were taken into consideration to ensure customer comfort.

A brief evaluation of vapor absorption systems was found to be unfeasible, as they introduced unnecessary complexity and significantly increased costs. Additionally, these systems posed a higher risk of failure.Although some refrigerants offered higher efficiencies or cooling rates under specific conditions, we prioritized environmental sustainability in our design. The refrigerant selected for our system has an ozone depletion potential (ODP) of 0 and a global warming potential (GWP) of less than 500, ensuring that the system has a minimal environmental impact while maintaining performance.

The system parameters were carefully selected to align with the given conditions, optimizing both efficiency and cooling rate. This thoughtful calibration ensures that the system performs at its best, balancing energy consumption with effective cooling to meet both performance and sustainability goals..

F- Cost Analysis

1. Capital Cost

Component	Cost(Rs)
Scroll Compressor	16000
Condenser	5000
Evaporator	5000
Expansion Valve	3000
Refrigerant	12000
Control System	4500
Misc	3500
Safety Measures	2500
Fans	6000
Piping	3500
Total	**61000 /-**

2. Operational Cost

The cost of electricity would vary based on factors such as environmental temperature, time of operation, electricity cost per unit, humidity levels, and other operational conditions. Additionally, the maintenance cost for the system is estimated to be Rs 4000 per year, covering routine servicing and upkeep to ensure optimal performance.

3. Savings

Considering one unit of electricity costs on average **Rs 6/kWh**, an average EER of commercially sold **1.25 TR** air conditioning units of **8.5**. If the air conditioning unit runs for one hour, it will cost approximately **Rs 3.10**. On the other hand, our system with an average EER of **11.55** would cost **Rs 2.28** per hour.

G-Feasibility

Technical:

A cooling capacity of 1.25TR along with a high average EER of 11.55 is a good milestone for future systems. The use of R1234yf as a refrigerant comes with its challenges of compatible components, flammability and leakages. The system needs to have better pressure handling capabilities. However the refrigerant allows high energy efficiency while also being very environmentally friendly. Precise design of the evaporator and condenser coils and fins are needed for maximum heat transfer. The thermostatic expansion valve chosen should be able to limit the increase of pressure while allowing efficient change of state by varying its opening fraction. The scroll compressor chosen has a great impact on the final energy efficiency, owing to its mechanical and isentropic efficiency values.

Economic:

As outlined above, although such a system would incur higher initial costs to the end user, it would be largely offset by the savings achieved during the run time due to the much larger EER than currently available systems. As the demand for environmentally friendly systems and greater energy efficiency, our system captures a unique market.

H - References

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I- Conclusion

The design and optimization of a vapor compression system for the given outdoor conditions were successfully completed, incorporating advanced control strategies and thermodynamic analysis. Key system parameters, such as subcooling, superheating, and pressure stabilization, were optimized to ensure reliable and efficient operation.

The integration of Proportional-Integral-Derivative (PID) control enabled precise regulation of system variables, enhancing stability during transients and maintaining steady-state performance. This control strategy minimized deviations and improved the overall responsiveness and efficiency of the system.

The performance evaluation, based on ISEER and EER metrics, demonstrated high energy efficiency and compliance with industry standards. The steady-state subcooling and superheating values confirmed effective heat transfer, ensuring proper refrigerant behavior and system reliability.

This study highlights the importance of combining optimized design with advanced control techniques to achieve superior performance under varying conditions. The results provide a strong foundation for real-world implementation and further refinements tailored to dynamic operational requirements.