Development of a heat pump system for mobile applications

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by

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Abstract

The heating and cooling sector is one of the biggest sectors at this moment due to drastic fluctuations in climate parameters. Even though this sector helps in making sure everyone gets comfortable temperatures at the home, office, etc., there are environmental consequences attached to them. These consequences are in terms of emissions of greenhouse gases. Almost around 80% of the current greenhouse gas emissions are due to the demand for energy, mostly in the form of electricity and heat [1], [2]. Having looked at these emissions, the demand for heating and cooling in residential and commercial spaces is increasing day by day. The systems which provide the desired effect of cooling and heating use different working mediums i.e. fossil fuels and renewable sources which correspond to 75% and 22% usage in EU currently [3]. So, to reduce the consumption of fossil fuels, heat pumps came into the picture.

A heat pump uses a refrigerant through which heat transfer is carried out to attain the desired temperature at a particular location. The properties of these refrigerants and other component-based parameters affect the COP (Coefficient of Performance) of the system. For a mobile system, i.e. a portable heating and cooling device used for military camps, agricultural purposes currently use fossil fuels which lead to enormous emissions, thus, these systems have to be redesigned by using vapor compression technology. One of the constraints is being able to use a refrigerant with GWP (Global Warming Potential i.e. a measure of the amount of heat a greenhouse gas traps in the atmosphere comparing it with the same amount of CO_2) of less than 150 [4]. Aspen model was created to assess different refrigerant behavior under some constraints like a cooling capacity of 40 kW, a heating capacity of 60 kW, and an air-air system with an airflow rate of less than 10000 m^3/h, these requirements are obtained from fossil fuel-based mobile systems, as these systems are air-air systems and system is kept outside the desired place, hoses have to be used to provide the air into and out of the system, thus, the specific air flow rate has to be achieved to get the desired temperature inside the desired place, similarly, 40-60 kW is a need for these systems due to high capacity applications. Sensitivity analysis of the aspen model was carried out by changing ambient temperature from 21 to 40 °C for cooling and -5 to 15°C for heating respectively, to quantify the behavior of refrigerants R290, R1234yf, R1234ze, R454C, R455A, R1270, R600a, R717, R516A. Exergy analysis was carried out along with the COP calculations to examine the second law efficiency for all the refrigerants to see the feasibility of these refrigerants being an alternative option for widely used R410A refrigerant. By varying a few parameters considered for the heat pump systems, optimization and sensitivity analysis resulted in finding the most suitable refrigerant for this system. For ease of calculations, a few assumptions and constraints were considered while simulating the system.

The maximum COP value obtained for the refrigerants R290, R1270, R1234yf, R1234ze, R454C, R455A, and R516 for cooling effect are 1.75, 1.77, 2.09, 1.91, 2.18, 2.25 and 2.21 respectively and similarly for heating effect are 3.61, 3.26, 2.49, 2.72, 3.56, 3.67 and 3.68 respectively. These COP values are obtained when the pressure drop across the heat exchangers is zero, so, the values with a pressure drop of 0.5 bar will fluctuate a bit from these values. If not COP, it will surely affect all other component parameters. Apart from COPs, second law efficiency is also a crucial parameter to analyze the system performance compared to a Carnot system i.e. an ideal system. Exergy analysis helped in quantifying the second law efficiency of the system for each refrigerant, which is calculated by the ratio of the actual COP of the system and Carnot COP of the system (Calculated by taking the source and sink temperature values). The maximum second law efficiency values obtained for the refrigerants R290, R1270, R1234yf, R1234ze, R454C, R455A and R516 for cooling effect are 0.32, 0.27, 0.2, 0.16, 0.29, 0.31 and 0.21 respectively and similarly for heating effect are 0.4, 0.41, 0.37, 0.45, 0.405, 0.42 and 0.41 respectively. So, by considering all the COP and second law efficiency values it can be concluded that a suitable refrigerant for this system with good performance as well as safe to use and handle is R455A, and other potential refrigerants being R454C almost similar performance with respect to R455A, and R290 and R1270 single component refrigerants with the concern of higher flammability can be used with further optimizations in the model.

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Contents

1 Introduction

The well-known phenomenon ”Climate Change” has constantly been occurring and being studied throughout history by understanding the variations and fluctuations in ice core levels, drastic changes in temperature, and many more, etc [5]. The atmospheric changes on Earth i.e. subsequent changes in the amount of solar energy absorbed or received by the surface of the earth have been observed. However, as mentioned in [6] that ”Since the mid 20th century, global warming has been expediting to an unrivaled extent, for which the main cause being activities of humans.” Also, as mentioned in [7] IPCC (International Panel of Climate Change) report that ”The surface temperature of Earth has been increasing, enhancing the sea level due to melting of snow and ice available in terms of glaciers and mountains, etc.” So, this is not an out-of-the-blue problem but has been ignored by humans on some levels for a long time. From various sources and data, it can be stated that the temperature on the surface of the earth was higher than the current temperature [2]. Nonetheless, the resources also disclose that the global warming rate currently is almost reaching its peak compared to the past few years, hence a concrete step has to be taken to find a solution than ignoring it anymore in order to limit the consequences of climate change in the near future.

Legislation regarding climate change and reducing greenhouse gas emissions have been put forth by many associations belonging to various countries or continents. The Kyoto Protocol in 1997 was the first agreement between nations in order to reduce GHG emissions by 5% compared to the emission level observed in 1990 [2]. As stated in the report from 2012 [8], increases the limit to 20%. The European Union (EU) again in 2014, enhanced this limit of emission percentage to 40% compared to the 1990 level by 2030, which became a foundation of the Paris Agreement, which was then adopted by the United Nations in 2015[2]. In 2015, under the name of the United Nations Framework Convention on Climate Change (UNFCCC). The objective of [8] report along with the above agreements was to limit raising the surface temperature of Earth to 2°C, whilst aiming to pursue efforts to find more solutions to restrict the limit to 1.5°C. EU proposed a strategy to enable and achieve the goal of a neutral climate economy by the year 2050 as stated in [9].

Almost around 80% of greenhouse gas emissions seen nowadays is predominantly due to the usage of electricity and heat [1]. Thus, optimizing these forms of energy and making them decarbonized is something everyone is thriving for. If stats have to be looked upon, the industrial sector gives rise to almost 32% of global greenhouse gas emissions in terms of various processes where usually heat predominantly increases the emission [2].

Demand for heat energy is increasing day by day in terms of heating or cooling for commercial and residential applications across the globe, climate change being the primary reason for the enhanced demand. Comfort in terms of heating or cooling more than a luxury is a necessity due to the drastic changes in atmospheric conditions. Building energy usage may lead up to approximately 73% of the total energy usage in the world. The energy demand for building applications is elevating exponentially as per [10], hence finding long-term solutions for energy usage with lesser consequences on the environment is need of an hour and steps have to be taken as soon as possible for a sustainable world [11].

1.1. Heating and Cooling

The heating and cooling sector is one of the biggest sectors in terms of energy usage share, i.e. more than 50% of energy given to the buildings [12]. In the Europe region, the energy consumption by this sector can be summarised as ”Heating and cooling accounts for half of the energy usage and makes the biggest energy-end use sector leaving behind electricity and transport sector” [3].
The building sector consists of residential, public, and commercial properties, accounting for almost 30% of the final energy consumption which corresponds to around 3100 Mtoe across the globe in a direct and indirect way, consuming almost 55% of global electricity. Building operations have given rise to approximately 28% of global CO_2 emissions, making, decarbonization a key priority to reach climate neutrality goals, [13].
Heating and cooling, the two prominent end-uses in building operations, are particularly critical to address to curb building emissions. Heating corresponds to around 45% of building emissions, but more than 55% of its total energy consumption still has a reliance on fossil fuels for supplying desired heating effect [13]. The floor area of buildings is predicted to increase by the multiplication factor of at least 2 by the year 2070 – the equivalent of adding the built surface of Paris to the buildings’ stock every week. Simultaneously, space cooling will also expand more rapidly than any other building end-use, with access provided to an additional 5 billion people by 2070, [13].
In EU households, from the survey of [3], the stats can be seen like, 79% of the total energy use is accounted for by heating and hot water. Even though cooling has a comparatively smaller share due to colder climates, it does vary due to changes in the temperatures during summer and an increase in the demand for cooling for businesses such as the food industry, where cooling or refrigeration plays a crucial role. Along with households, energy consumption for the industrial sector in the EU also has usage of around 70.6% for space and industrial process heating, 26.7% for lighting and electrical processes, such as machine motors, and 2.7% for cooling respectively. This data shows that Heating and cooling are vital factors in every sector [3].

The desired effect of cooling or heating can be achieved with the help of various systems, which might be using various sources:

- Fossil fuels based heating and cooling: Around 75% of the heating and cooling systems are based on fossil fuels as per European commission.

- Renewable source based heating and cooling: Only about 22% of the heating and cooling systems use renewable sources as per European commission.

1.2. Types of Heating and Cooling Systems

To achieve the heating and cooling demand, various systems can be used, they differ in terms of working principles, working mediums, environmental friendliness, etc, which are discussed below. Depending on the applications, it is mainly divided into 2 basic types which are, Stationary and Mobile/Movable systems [14]:

- **Stationary Systems**: These systems are permanently fixed systems to any of the commercial, industrial or residential spaces. These systems can not be moved after their installations, but surely have a higher performance efficiency. The heating or cooling effect is also permanent, and with proper maintenance, at least for a few years, one can expect optimum performance from the system. Usually, these systems have multiple components and are installed in different locations in a particular region, which enables the system in providing higher capacity outputs from the system, ranging from 4 kW to a few hundred kW.

- **Mobile/Movable Systems**: These are portable systems, and can be moved from one place to the other depending on the requirements. This system has mostly just one component comprised of multiple systems stored inside. Due to its portable nature, it is comparatively lightweight, robust as well as compact, but still provides the expected desired effect. Applications of such systems focus more on the temporary effect i.e. temporary heating, which is for a limited time period ranging from 1 week to 2 years used in tents for events, military camps, etc. Due to the complexity of the system, there are numerous challenges in designing and developing it the system should have as small components as possible to make the system compact, but still give a good performance, due to the small structure less pipe structure is available so flow control becomes a crucial parameter, compared to a fixed system, this can not be divided into an indoor and outdoor unit which might lead to flooding on the refrigerant which can cause compressor damage, good insulation of the system is necessary in order to prevent the loss of heating or cooling effect to the atmosphere, perfectly suitable refrigerant should be chosen to work in a bigger range of ambient temperature conditions, components chosen should be as lightweight as possible, etc.

1.3. Thesis Objective

The mobile/movable heating systems available in the market for various applications are based on fossil fuels, so it is essential to convert them into a system that has lesser environmental consequences. Also, there are a few mobile systems in the market that uses Heat Pump technology having a very low capacity of around 7-8 kW, but with the usage of refrigerants like R410A, whose GWP is more than 2500 [15], and according to the new rules of EU Commission, can not be used in mobile/movable systems anymore [4], but only the refrigerants with GWP less than 150 can be used [4]. So, in this thesis project, the aim is to develop a heat pump system, more precisely a mobile/movable ASHP (Air Source Heat Pump) reversible system (reversible i.e. able to provide heating and cooling) which is not available in the market and this will replace the current fossil fuel-based systems that are produced for multiple applications. Usage of higher GWP refrigerant has to be reduced and Low GWP alternatives that follow the norms given by EU Commission [4] will be examined in this thesis project. The project outline along with the task list is given below:

- Study of refrigerants, focusing on their feasibility by analyzing the specific properties as well as choosing appropriate components of the system.
- Analysis of the chosen Refrigerants in a basic Aspen model to validate the choices made regarding refrigerant, its parameters along with parameters related to components.
- Verifying the performance of the system and comparing it with the heat pump systems using high GWP refrigerants.
- Exergy analysis of the system to validate results when compared to the Carnot system and find possibilities for optimization.
- Comparing the performance parameters for all the potential refrigerants for the system.
- The system is expected to have the following specifications:
  - The system must be having a refrigerant with the least environmental hazards along with GWP less than 150 [4].
  - It should be a reversible heat pump system (heating as well as cooling) depending on the ambient temperature conditions.
  - Air-Air system with a capacity of around 40 kW for cooling and 60 kW for heating, with the ability to provide desired effect in the ambient temperature range of -5 to 40°C.
  - The system should be compact, robust, and as lightweight as possible.
  - For air circulation, hoses have to be used with the volumetric flow rate of around 10000 m^3/h.
  - COP to be as high as possible, approximately in the range of 2 to 5.

1.3.1. Research Questions

The project surely links with research related to emerging technologies in the heat pump sector, and the research questions help in channelizing the project to achieve the expected outcomes in the best possible way. Some of these research questions are given below:

- What refrigerant will perform the best to suit the expected specifications?

- What are the best alternatives for high GWP refrigerants?
- What is the feasibility of low GWP refrigerants?
- What are the maximum temperature and pressure conditions that will be attained during the operation of the system?
- Will the system performance match with the fossil fuel-based system and also Heat Pump systems with high GWP refrigerants?
- What is the performance of the system when compared to the Carnot system by doing an exergy analysis?
- How the system can be optimized by reanalyzing the parameters and components?

These questions will be answered as the project dives deep into the details of every component and the Aspen model. One of the challenges which might be faced is, as there is no other similar kind of system or product available, so validating and verification of the model results will be a task. Hence, a few assumptions might be considered to make calculations easier. Also, the refrigerant mixtures or blends defined in Aspen might differ in behavior compared with their actual behavior, so approximations have to be taken, or only predefined mixtures will be used. The questions listed above will be answered in this project in the consequent chapters of this report.

2
Literature Review

2.1. Heating and Cooling Systems

As discussed in 1.2, there are majorly stationary and mobile/movable types of heating and cooling systems. These differ in terms of their working phenomenon, usage of components, working fluids, and also applications. As for the desired effect of heating, various sources have been used in the past but for cooling Vapour compression cycle or refrigeration cycle in which refrigerant is the working fluid and provides the necessary desired effect. For reversible heat pumps, both refrigerants as well as other fossil fuel sources like gas, oil, etc. can be used, this is discussed in the next section.

2.1.1. Fossil Fuel Based Mobile Systems for Heating and Cooling

The systems which use fossil fuels as a working medium to get the desired effect of Heating might also be combined and used as an auxiliary system with a Vapor compression system called dual fuel or dual source system [16]. The working medium for fossil fuel-based systems can be oil, coal, or gaseous fuels, sometimes even wood is used as a working medium. In order to get the heating effect, the direct or indirect combustion method is used [17]. In order to achieve a cooling effect, the heating effect generated by these fuels in a particular location can be cooled down using cooling towers. Cooling towers do not directly use any of these fuels, but still correspond to emission-related issues and have environmental impacts due to the combustion method used for heating[18].

Working principle

Either a direct or an indirect combustion method is the basic working mechanism, i.e., a direct-fired stove equipped with a fan to blow air into the desired places or an indirect-fired tank to store the water at a particular temperature respectively. The advantages and disadvantages of these systems by classifying them into oil-fired and gas-fired systems are discussed below, with the source being [17] to focus on mobile systems:

- **Oil Fired Systems**
  - Advantages
    - Direct-fired air heaters are used
    - Discharge temperatures of up to 450°C
    - Diesel or petrol is often used as a fuel
    - Applications such as Agriculture, Horticulture, Industries, Buildings, tents, etc.
    - Oil furnaces are comparatively cheaper
    - Oil-burning furnaces need simple and easy maintenance service
    - Oil burns hotter than gas, giving off more heat per BTU for an equivalent amount of fuel
    - These systems are comparatively silent

- Disadvantages
  - Oil is not as cheap as gas
  - Storage tank for oil is necessary
  - Creates soot and dirt buildup on the furnace
  - Oil furnaces reach a maximum of up to 90% annual fuel utilization efficiency
  - Ventilation is necessary

- **Gas Fired Systems**
  - Advantages
    - Direct fired-burner with an axial or centrifugal fan to blow air
    - Discharge temperatures up to 200°C
    - Natural gas or propane is usually used as a working fuel
    - Applications such as Agriculture, Horticulture, etc.
    - Cost of gas is low
    - Gas furnaces are efficient, most of the furnaces have 100% annual fuel utilization efficiency
    - Furnaces need lesser maintenance, along with being cleaner and quieter compared with oil-based equivalents
    - Instant effect due to higher efficiency
    - Lifespan of about 10-20 years
  - Disadvantages
    - Compared with oil, gas gives lesser heat per BTU for an equivalent amount of fuel
    - Gas furnaces are expensive
    - Large and complex structures are usually observed, making it tough to move
    - Gas leaks can be hazardous
    - Ventilation is necessary

Schematic

A picture of a mobile/movable system using fossil fuels as a source to produce heating for various applications is shown below 2.1. The picture is included in order to indicate the compactness of the system as well as the complex assembly which is needed to fit all the components in such a small casing. Hoses are used to deliver the desired output to the tens or other spaces, as these systems are kept outside the region, which is shown in the picture given below in 2.2 & 2.3. Pictures are obtained from the catalog [17].

Advantages of Fossil Fuel based systems

- Efficient in heating and cooling even in the extreme ambient temperature conditions
- In some cases, Cheaper as compared to other types of heating systems
- Time needed to attain the required temperature is less
- Mobile applications can be easily developed

Disadvantages of Fossil Fuel based systems

- Environmental consequences are pretty high
- Energy consumption might be higher
- Pollution increases due to emission of CO, CO_2 i.e Higher carbon footprint overall etc.

Figure 2.1: The picture indicates the mobile/movable system using fossil fuels like oil, gas, etc. The left side cavity/hollow part is used to connect hoses, sometimes multiple cavities are provided to connect multiple hoses. The cavity on the top surface is a purger, through which gas emission/exhaust can be carried out like CO_x, SO_x, NO_x and on the right side (back of the system), it is used as an air capturing fan, through which air enters the combustion stove [17]. Dimensions of the systems are 161 cm ×72 cm ×128 cm as length, width, and height respectively

Figure 2.2: The mobile/movable system having multiple hoses is showcased. As can be seen, it has two cavities on the left side through which more hoses are connected. The system is placed on the top of the building which shows it is totally outside the place from where a heating effect is needed.

Figure 2.3: The mobile/movable system having multiple hoses is showcased. As can be seen, it has just 1 cavity on the left side through which just 2 hoses are connected. The system is placed outside the tent inside which a heating effect is needed.

It has been observed that heating in buildings currently produces near about 6% of global Green-House Gas (GHG) emissions but the reduction of up to 90% of CO_2 emissions can be achieved by sustainable technologies which will almost vanish the environmental impacts due to heating or cooling systems [19]. It will help in limiting the environmental consequences and reaching global neutrality goals as soon as possible. Fossil fuel-based heating gives rise to emissions along with being exorbitant. Increase in the world energy demand, the primary sources used to accomplish the desired energy requirement have to be replaced or optimized. Hence transition to sustainable solutions is crucial to create environmentally friendly and cost-effective heating and cooling systems.

So as to find solutions for sustainable technology to limit GHG emissions, electrification of these systems can be one of the solutions. Electrification of heat increases opportunities for heat storage and demand response that are contributing towards balancing the variable renewable energy (VRE) sources like wind, solar, etc [20]. The research toward a transition to a sustainable environment was carried out and the Heat Pump came into the picture as one of the promising solutions.

2.2. Heat Pumps

A heat pump is a promising solution to overcome or limit environmental consequences like emissions of greenhouse gases, climate change, etc. But when compared to fossil fuel-based systems, these also should provide optimum performance in various applications then only can be replaced. Heat Pump can be defined as per International Energy Conservation Code [21] and [22], ”A mechanical device that is capable of providing heating, cooling and hot water for residential, commercial and industrial applications”. Electricity is its working force, and the mechanism is transferring heat from one source at a particular temperature to the other source at a different absolute temperature. System specifications will be provided ahead.

2.3. System

Heat pumps can provide heating or cooling or both depending on the application requirement. The desired effect that the system is providing predominantly, the machine/system can have a different name like a heat pump, air-conditioner, or cooling/refrigerating machine. The basic and well-known working principle for this is the vapor compression refrigeration cycle, also called the refrigeration cycle.

2.3.1. Vapour Compression Refrigeration Cycle

The vapor compression refrigeration cycle is one of the most widely used working principles of the basic heat pump system. This technology was the up-gradation from the fossil fuel-based systems due to excessive emissions and factors affecting the environment. The use of heat pumps over fossil fuel-based systems can approximately save up-to 20% natural gas in long term along with reducing carbon emission globally by 1.7 Mt [23].

Vapour Compression Technology based system has main components which are as follows:

- Condenser (Heat Exchanger)
- Evaporator (Heat Exchanger)
- Expansion Valve
- Compressor
- Refrigerant (Working fluid/medium)

Brief information about each component is given below:

- **Condenser**

  A condenser is a component of the system which is used to condense the working fluid or change the phase from vapor to liquid. It plays a decisive role in determining the efficiency or effectiveness of the system [24], as it carries heat transfer between the refrigerant and other working fluids like air or water. The refrigerant at higher pressure and temperature rejects heat to the working fluid (air/water), this plays a crucial role mainly when heating is the desired effect. Depending on the type of application, a shell and tube or plate heat exchanger is used. Also, streams flowing inside the condenser directly affect the effectiveness, depending on the direction of streams, it is classified as Co-current and counter-current, also a cross-flow type of stream is widely used in condensers.

- **Evaporator**

  An evaporator is also a heat exchanger, which is used to evaporate or boil the refrigerant after the expansion process in the expansion valve. The refrigerant absorbs heat and evaporates, so the refrigerant in gaseous form is transported into a compressor. Usually, an evaporator coil is used [25]. Heat transfer occurred in the evaporator is the desired effect for cooling or refrigeration purposes. Just apart from the desired effect i.e. pressure and temperature limit, all other things like working principle and method, streams, etc are exactly similar to a condenser.

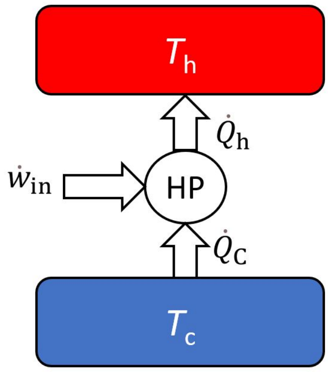

Figure 2.4: Mechanical Driven Heat Pump System representing a reservoir at a higher temperature T_h and another reservoir at a lower temperature T_c, so the heat is pumped from a lower temperature reservoir denoted by $\dot{Q}_c$ to higher reservoir denoted by $\dot{Q}_h$ with the help of work $\dot{W}_{in}$. This is the mechanism of the Heat Pump system. [29]

- **Expansion Valve**

  An expansion valve is also called a pressure-reducing valve, in which pressure and temperature are dropped usually in a drastic manner. The refrigerant condensed in the condenser at a higher pressure and temperature is expanded in the valve so that both of the parameters are optimized for the next part of the process. It also affects the mass flow rate of the refrigerant, as there is a sudden change of area inside the valve which helps in controlling the mass flow rate. Depending on the type of application and requirement, different types of expansion valves are used, which include Thermal expansion valves [26], manual valves, automatic valves, capillary tubes, float valves, electronic expansion valves, etc.

- **Compressor**

  A compressor is a pressure-elevating device, which is used to increase the pressure of an evaporated refrigerant to a supersaturated temperature and pressure which then can be used for condensation. As per the thermodynamics, work is needed to be done by the compressor to achieve the optimum parameters and which is the input energy for the performance of the system. Depending upon the application Reciprocating, a Rotary, Scroll, Rotary Screw, or Centrifugal compressor is used [27]. To optimize the performance, multistage compression is also an option i.e. more than 1 compressor is used in just one system.

- **Refrigerant**

  Refrigerant is a chemical compound that is responsible for the transportation of heat throughout the vapor compression refrigeration cycle. The characteristics of refrigerant define how much heat is absorbed in the evaporator and how much heat is rejected in the condenser. The significance of a Refrigerant will be explained ahead in detail.

2.3.2. Thermodynamics of Heat Pumps

A heat pump in which the driving energy is mechanical can be schematically shown as in Figure 2.4, uses a mechanical power $\dot{W}_{in}$ to withdraw heat from the reservoir at a colder temperature T_c and provides it to the other reservoir at a higher temperature T_h [28].

As per the statement of the first law of thermodynamics, ”the total energy of a system remains constant, even if it is converted from one form to another” [30].

$$\dot{W}_{in} = \dot{Q}_h - \dot{Q}_c \tag{2.1}$$

and the second law of thermodynamics states, ”The total entropy of an isolated system can never decrease over time, and is constant if and only if all processes are reversible. Isolated systems spon-

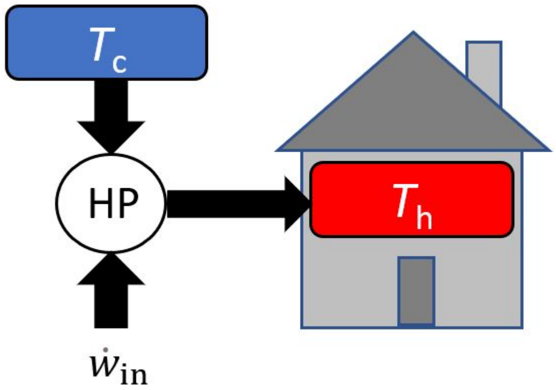

Figure 2.5: Heat Pump in Heating Mode is shown in the figure. In which, the heat from the ambient air which is at a lower temperature T_c is pumped using the heat pump system with the help of work $\dot{W}_{in}$ and provides heating inside the house at an absolute temperature T_h. The heating mode is used in winter. [32]

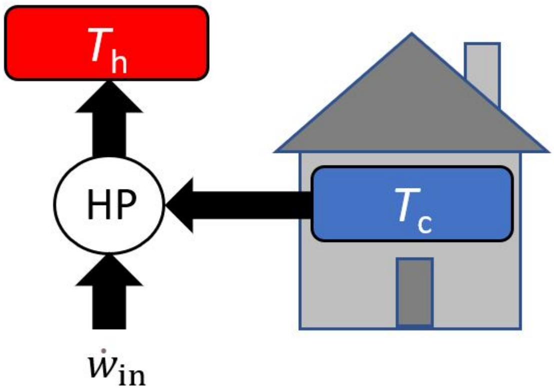

Figure 2.6: Heat Pump in Cooling Mode is shown in the figure. In which, the heat from the air inside the house which is at a lower temperature T_c is pumped outside using the heat pump system with the help of work $\dot{W}_{in}$ and provides cooling inside the house by rejecting heat outside at an absolute temperature T_h. Cooling mode is used in summer. [32]

taneously evolve towards thermodynamic Equilibrium, the state with maximum entropy" [31].

$$\frac{\dot{Q}_h}{T_h} = \frac{\dot{Q}_c}{T_c} \tag{2.2}$$

combining the above mentioned equations 2.1 and 2.2 we get,

$$\dot{W}_{in} = \dot{Q}_h \left(\frac{T_h - T_c}{T_h} \right) \tag{2.3}$$

$$\dot{W}_{in} = \dot{Q}_c \left(\frac{T_h - T_c}{T_c} \right) \tag{2.4}$$

The value of $\dot{W}_{in}$ is ideal in this case. The two equations obtained after combining laws denote heating and cooling application respectively. The equation 2.3 shows the heating as the desired effect whereas the other equation 2.4, denotes the desired effect of cooling. As per the desired effect, the whole system looks something like shown in Figure 2.5 showing the heating mode, and Figure 2.6 shows the cooling mode of the heat pump system.

2.3.3. Performance of Heat Pumps

The performance of heat pumps is measured in a parameter called COP, which is an abbreviation for Coefficient Of Performance, which is a dimensionless number showcasing how efficient the system is. It is the ratio of desired effect (Heating/Cooling) to the power required to achieve that effect. The value of COP usually ranges between 2 to 500, so it is equivalent to efficiency but in terms of magnitude, it might not do justice. It can be denoted as,

$$\mathrm{COP} = \frac{\text{Desired effect Duty (kW)}}{\text{Input Power (kW)}} \tag{2.5}$$

For heating effect, it can be written as,

$$\mathrm{COP_H} = \frac{\dot{Q}_\mathrm{h}}{\dot{W}_\mathrm{in}} \tag{2.6}$$

whereas for cooling,

$$\mathrm{COP_C} = \frac{\dot{Q}_\mathrm{c}}{\dot{W}_\mathrm{in}} \tag{2.7}$$

Second law efficiency is the ratio of the actual value of COP to the ideal which in these cases is considered as equivalent Carnot COP value [33]. This efficiency denotes how well the system performance when compared to an ideal system working between the same temperature limits, for which the formula can be written as:

$$\eta = \frac{\text{Actual COP}}{\text{Carnot COP}} \tag{2.8}$$

The value of η varies between 0 to 1, where 1 is considered as the maximum efficiency in which the system behaves as an ideal system. It gives the effectiveness of the energy utilization in a system compared to the Carnot system. The Carnot COP for an ideal system, delivering heat to an isothermal the reservoir at temperature T_h by means of work input and transfer heat from an isothermal reservoir at a lower temperature T_c, which utilizes only reversible processes [33], is given by:

$$\text{Carnot COP}_\mathrm{H} = \frac{T_\mathrm{h}}{T_\mathrm{h} - T_\mathrm{c}} \tag{2.9}$$

$$\text{Carnot COP}_\mathrm{R,C} = \frac{T_\mathrm{c}}{T_\mathrm{h} - T_\mathrm{c}} \tag{2.10}$$

Equation (2.9) can be used for Heating and equation (2.10) for cooling and refrigeration applications. Finding the values of $\dot{Q}_\mathrm{h}$, $\dot{Q}_\mathrm{c}$ and $\dot{W}_\mathrm{in}$ is crucial to find the exact value of COP. The value can be found by using the formulas given below:

$$\dot{Q}_\mathrm{c} = \dot{m}(\Delta h_\mathrm{evaporator}) \tag{2.11}$$

$$\dot{Q}_\mathrm{h} = \dot{m}(\Delta h_\mathrm{condenser}) \tag{2.12}$$

$$\dot{W}_\mathrm{in} = \dot{m}(\Delta h_\mathrm{compressor}) \tag{2.13}$$

Where, $\dot{m}$ is the mass flow rate of refrigerant and h denotes the specific enthalpy, which is a thermodynamic property found by the addition of internal energy and flow work. Enthalpy is a function of temperature. Δh is the change in the enthalpy, so across each component change in enthalpy is calculated to know the energy transfer. The pressure versus enthalpy curve is shown below in Figure 2.7, which shows the vapor compression refrigeration cycle and the changes in the parameters throughout the cycle. The temperature versus Entropy curve is also shown below in Figure 2.8, more information about the cycle is discussed in [34].

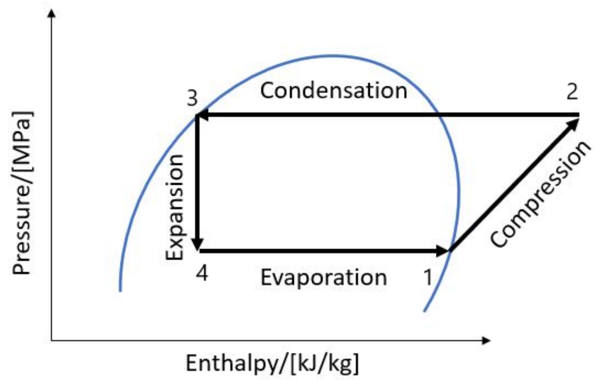

Figure 2.7: Pressure(the quantity Pressure divided by the unit MPa is the dimensionless number that is listed on the y-axis) versus Enthalpy(the quantity Enthalpy divided by the unit kJ/kg is the dimensionless number that is listed on the x-axis) curve is shown. It denotes processes like $4 \rightarrow 1$ as evaporation, $1 \rightarrow 2$ as compression, $2 \rightarrow 3$ as condensation, and $3 \rightarrow 4$ as expansion in terms of the variation of pressure and enthalpy during a vapor compression refrigeration cycle. [34]

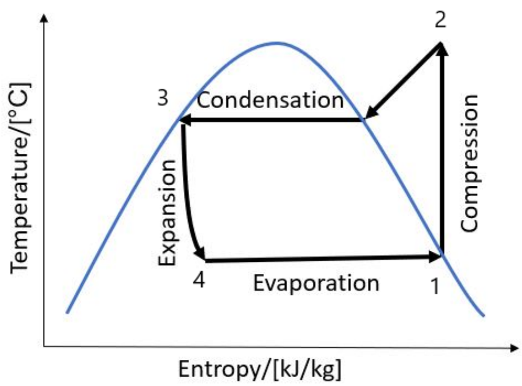

Figure 2.8: Temperature(the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the y-axis) versus Entropy(the quantity Entropy divided by the unit kJ/kg is the dimensionless number that is listed on the x-axis) curve is shown. It denotes processes like $4 \rightarrow 1$ as evaporation, $1 \rightarrow 2$ as compression, $2 \rightarrow 3$ as condensation, and $3 \rightarrow 4$ as expansion in terms of the variation of temperature and entropy during a vapor compression refrigeration cycle. [34]

2.3.4. Practical Vapour Compression Cycle

The cycles shown in figure 2.8 and 2.7 are ideal but the actual cycle differs a bit from this. In the ideal cycle, all the processes are considered to behave in an ideal manner which is not feasible in real life. So, we find temperature and pressure losses across all the components and also the need of superheating the refrigerant beyond the saturated vapor phase as well as subcooling of the refrigerant beyond the saturated liquid phase in order to get the desired output. Sometimes due to the component specifications, superheating and/or subcooling will occur automatically, which might enhance or reduce the overall coefficient of performance. The actual vapour compression cycle can be drawn like shown in figure 2.9 & 2.10.
Superheating increases the span of the evaporation, as well as subcooling, does. Increasing the magnitude of change of enthalpies hence enhancing the desired effect. COP is the ratio of the desired effect over work done, so if just desired effect enhances then COP also increases. But due to superheating and non-isentropic compression, there will be an increase in the work done due to changes in the discharge temperature. So, in order to have higher COP, the isentropic efficiency should be optimum as well as the superheating and subcooling values. Only when the change in work done is minimum but enhances the desired effect, then COP optimization is possible.

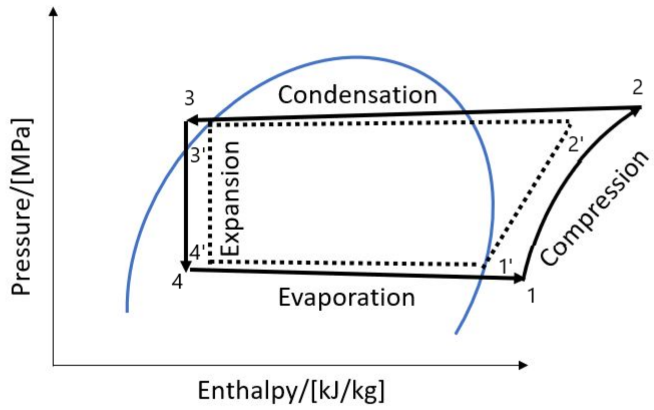

Figure 2.9: Pressure(the quantity Pressure divided by the unit MPa is the dimensionless number that is listed on the y-axis) versus Enthalpy(the quantity Enthalpy divided by the unit kJ/kg is the dimensionless number that is listed on the x-axis) curve is shown. It denotes processes like $4 \rightarrow 1$ as evaporation, $1 \rightarrow 2$ as compression, $2 \rightarrow 3$ as condensation, and $3 \rightarrow 4$ as expansion in terms of the variation of pressure and enthalpy during a vapor compression refrigeration cycle. The evaporation process is continued with superheating, compression is also not isentropic anymore and there is subcooling after the condensation. The dotted lines denote the ideal processes and solid lines denote the processes from the practical cycle containing pressure drop in evaporation and condensation[35].

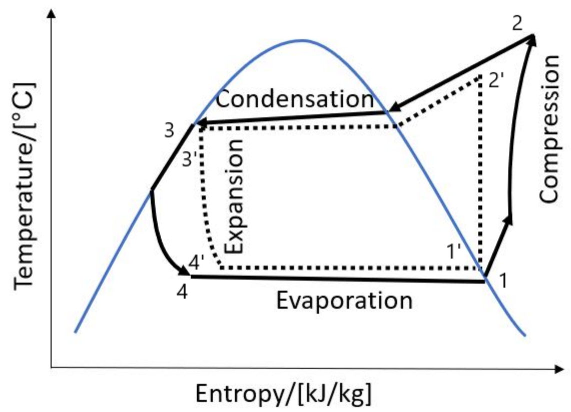

Figure 2.10: Temperature(the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the y-axis) versus Entropy(the quantity Entropy divided by the unit kJ/kg is the dimensionless number that is listed on the x-axis) curve is shown. It denotes processes like $4 \rightarrow 1$ as evaporation, $1 \rightarrow 2$ as compression, $2 \rightarrow 3$ as condensation, and $3 \rightarrow 4$ as expansion in terms of the variation of temperature and entropy during a vapor compression refrigeration cycle. The evaporation process is continued with superheating, compression is also not isentropic anymore and there is subcooling after the condensation. The dotted lines denote the ideal processes and solid lines denote the processes from the practical cycle containing pressure drop in evaporation and condensation[35].

2.3.5. Exergy Analysis

Exergy can be defined as ”the maximum theoretical work obtainable from an overall system consisting of a system and the environment as the system comes into equilibrium with the environment” [36]. Through exergy analysis, the current system is compared with the Carnot system to assess its performance. Exergy analysis validates the current parameter specifications and if there is a possibility of optimization. Exergy analysis is the energy losses in each component and reveals which component can be further optimized corresponding to the quality of the process linked with the component [37],[38]. Exergy or the specific flow availability [36], depicts the work by the reversible process which will modify the flow to attain equilibrium with the environmental conditions [39]. In this process, isentropic expansion to the atmospheric pressure and isothermal expansion to the entropy state environment is considered. Hence, the exergy of a flow with a state can be calculated by ignoring the potential and kinetic energy terms due to negligible values:

$$Ex_{\text{state}} = (h_{\text{state}} - h_0) - T(s_{\text{state}} - s_0) \tag{2.14}$$

By using the above formula, exergy loss across each component is calculated, to find the second law efficiency.

- Compressor: A refrigerant is compressed in a compressor, let the process of compression is from 1 to 2, then the exergy loss across the compressor can be given as:

$$Ex_{\text{loss-comp}} = \dot{m_{\text{ref}}}(ex_1 - ex_2) + \dot{W_{\text{comp}}} \tag{2.15}$$

  So, the exergy loss depends on the mass flow of the refrigerant, exergy loss at the suction side and the discharge side as well as the work done by the compressor.

- Expansion valve: A refrigerant is expanded in an expansion valve, let the process of expansion is from 3 to 4, then the exergy loss across the expansion valve can be given as:

$$Ex_{\text{loss-exp}} = \dot{m_{\text{ref}}}(ex_3 - ex_4) + \dot{W_{\text{exp}}} \tag{2.16}$$

  So, the exergy loss depends on the mass flow of the refrigerant, exergy loss at the inlet side and the outlet side as well as the work done by the expansion valve which is equal to 0 [39].

- Condenser: A refrigerant is condensed in a condenser, let that process be from 2 to 3, and it is using a secondary working medium of air, so the exergy loss across the condenser can be given as:

$$Ex_{\text{loss-cond}} = \dot{m_{\text{ref}}}(ex_2 - ex_3) + \dot{m_{\text{air}}}(ex_{\text{air-in}} - ex_{\text{air-out}}) \tag{2.17}$$

  So, the exergy loss depends on the mass flow rate of the refrigerant, exergy loss for the condensation of the refrigerant at the entrance and at the exit, and similarly on the flow rate of air and exergy loss at the inlet and outlet of air.

- Evaporator: A refrigerant is evaporated in an evaporator, let that process be from 4 to 1, and it is using a secondary working medium of air, so the exergy loss across the condenser can be given as:

$$Ex_{\text{loss-evap}} = \dot{m_{\text{ref}}}(ex_4 - ex_1) + \dot{m_{\text{air}}}(ex_{\text{air-in}} - ex_{\text{air-out}}) \tag{2.18}$$

  So, the exergy loss depends on the mass flow rate of refrigerant, exergy loss for the evaporation of refrigerant at the entrance and at the exit, and similarly on the flow rate of air and exergy loss at the inlet and outlet of air.

2.3.6. Entropy production

Entropy production in a particular component shows how well the processes carried out in the component are irreversible. In order to find the entropy production across each component, it is considered that component is externally adiabatic [39]. The formula for entropy production for each of the components is given below:

- Compressor: A refrigerant is compressed in a compressor, let the process of compression is from 1 to 2, then the entropy production across the compressor can be given as:

$$\Delta \dot{S_{\text{comp}}} = \dot{m_{\text{ref}}}(s_2 - s_1) \tag{2.19}$$

  So, the entropy production depends on the mass flow rate of the refrigerant, entropy after the compression, and before the compression.

- Expansion valve: A refrigerant is expanded in an expansion valve, let the process of expansion is from 3 to 4, then the entropy production across the expansion valve can be given as:

$$\Delta \dot{S_{\text{exp}}} = \dot{m_{\text{ref}}}(s_4 - s_3) \tag{2.20}$$

  So, the exergy loss depends on the mass flow of the refrigerant, entropy at the inlet side, and the outlet of the expansion valve.

- Condenser: A refrigerant is condensed in a condenser, let that process be from 2 to 3, and it is using a secondary working medium of air, so the entropy production across the condenser can be given as:

$$\Delta \dot{S}_{\text{cond}} = \dot{m}_{\text{ref}}(s_3 - s_2) + \frac{\dot{Q}_{\text{cond}}}{T_{\text{h}}} \tag{2.21}$$

  So, the entropy production depends on the mass flow rate of the refrigerant, entropy change due to condensation of refrigerant, heat rejected by the condenser, and temperature of condensation.

- Evaporator: A refrigerant is evaporated in an evaporator, let that process be from 4 to 1, and it is using a secondary working medium of air, so the entropy production across the evaporator can be given as:

$$\Delta \dot{S}_{\text{evap}} = \dot{m}_{\text{ref}}(s_1 - s_4) - \frac{\dot{Q}_{\text{evap}}}{T_{\text{c}}} \tag{2.22}$$

  So, the entropy production depends on the mass flow rate of the refrigerant, entropy change due to the evaporation of a refrigerant, heat absorbed by the evaporator, and temperature of evaporation.

2.3.7. Types of heat pumps based on the source

As discussed for heat exchangers, apart from the refrigerant, there is also a working medium present that helps in extracting or absorbing, and delivering the heat. Most of the time water or air is used as the working fluid to transfer the heat from the absorbed energy of refrigerants, with one more suitable option of Brine solution. Depending on the working medium on the source and sink side, there are majorly four types of heat pumps, which are as follows:

- **Air-Air Heat Pumps**
  - The heat pump system in which heat is absorbed from the air (ambient temperature) and then heat is pumped to the source in terms of air inside the room. i.e., Evaporator transfers heat between air and refrigerant as well as the condenser has the same working mediums [40].
  - Advantages of using such systems are
    - Quick and easy to install
    - Generates hot air
    - Cheapest installation cost
  - On the other side, disadvantages are
    - Need of access to an ample amount of ambient air, which in extreme cases might not be possible due to frosting on the outside heat exchanger
    - Fans are used to pump air into the system which might cause noise
    - Due to the lower density and heat capacity of air, it is not as efficient as other types of systems

- **Air-Water Heat Pumps**
  - The heat pump system in which heat is absorbed from the air (ambient temperature) and then heat is pumped to the source in terms of water inside the room. i.e., Evaporator transfers heat between air and refrigerant as well as the condenser has the water as a working medium. This water might be used in a heat exchanger to deliver the heat to the room [41].
  - Advantages of using such systems are
    - Easy to install
    - Generates hot water, which is then used to heat the room by using radiators or convectors
    - Comparatively cheaper installation cost

- ◇ Efficient compared to Air-Air systems due to water-based heating inside the room
- – Disadvantages
  - ◇ Need of access to an ample amount of ambient air, which in extreme cases might not be possible due to frosting on the outside heat exchanger
  - ◇ Due to the lower density and heat capacity of air, it is not as efficient as other types of systems due to the outdoor heat extraction

- **Water-Air Heat Pumps**
  - – The heat pump system in which heat is absorbed from the water (a ground source or lake-based source etc.) and then heat is pumped to the source in terms of air inside the room. i.e., Evaporator transfers heat between water and refrigerant and the condenser has the transfer directly using air [42].
  - – Advantages of using such systems are
    - ◇ Due to water being the source, the change in temperature is minimal, giving steady and effective performance compared to air-source-based systems
    - ◇ No need for radiators just a blower is sufficient
    - ◇ Due to less usage of fans, comparatively quieter
  - – Disadvantages
    - ◇ Difficult to install due to ground source or lake-based system needs drilling, boring, and piping in the ground
    - ◇ Due to more operations for installation, it is expensive
    - ◇ Installation is not quick and easy

- **Water-Water Heat Pumps**
  - – The heat pump system in which heat is absorbed from the water (a ground source or lake-based source etc.) and then heat is pumped to the source in terms of water inside the room. i.e., Evaporator transfers heat between water and refrigerant as well as the condenser has the same working mediums. Sometimes even brine (highly concentrated salt & water solution) is used as an antifreeze during the circulation of water throughout the system [42].
  - – Advantages of using such systems are
    - ◇ Generates hot water, which is then used to heat the room by using radiators or convectors
    - ◇ Due to the larger density and heat capacity of water compared to air, heat transfer is efficient and hence the system efficiency is also higher
    - ◇ Due to water being the source, the change in temperature is minimal, giving steady and effective performance compared to air-source-based systems
    - ◇ Quieter compared to other systems
  - – Disadvantages
    - ◇ Difficult to install due to ground source or lake-based system needs drilling, boring, and piping in the ground
    - ◇ Due to more operations for installation, it is expensive
    - ◇ Installation is not quick and easy

2.4. Literature Overview

An overview of the study is discussed in this section. Mostly the focus is on studying refrigerants, all the terms related to refrigerants, characteristics, properties, and even nomenclature with the help of [43].

2.4.1. Refrigerant

All of the components of the vapor compression refrigeration cycle surely affect the performance of a heat pump. Let that be condenser and evaporator size & effectiveness, type of compressor used, all the parameters related to the compressor, and type of expansion valve used. But, the most important factor which gives the range up to which a system can perform optimally is the refrigerant. Hence, choosing a suitable refrigerant plays a crucial role in system design. As per [43], Refrigerant can be defined as ”The fluid used for heat transfer in a refrigerating system; the refrigerant absorbs heat and transfers it at a higher temperature and a higher pressure, usually with a phase change. Substances added to provide other functions, such as lubrication, leak detection, absorption, or drying, are not refrigerants”. Refrigerant circulates heat throughout the heat pump system. Refrigerants are normally denoted by R along with a specific combination of numbers and letters which originated from the chemical structure of the refrigerant as shown in table 2.2. If it contains a letter after numbers, then it shows the isotopes of the compound. The boiling point plays an important role, for the standard atmospheric pressure condition, the refrigerants can have boiling point lower than 0°C, on an average between -10°C to -50°C, which helps in delivering low temperatures. Every refrigerant has different characteristics like boiling point, toxicity, flammability, etc., hence depending upon the applications, refrigerants have to be chosen.

Important properties and terms regarding refrigerants are defined below [43]:

- **azeotropic** - An azeotropic blend or a mixture is one that contains two or more refrigerants whose equilibrium vapor and liquid-phase compositions are the same at a given pressure.
- **critical point** - The location or a point on a plot of thermodynamic properties at which the liquid and vapor states of a substance meet and become indistinguishable. The temperature, density, and composition of the substance for the liquid and vapor phases are equal at this location. The density, pressure, specific volume, and temperature at the critical point are also called critical density, critical pressure, critical volume, and critical temperature, respectively.
- **glide or temperature glide** - The difference between the starting and ending temperatures of a phase change process denoted by an absolute value for a refrigerant within a component of the refrigerating system, without considering any sub-cooling or super-heating. This describes the condensation or evaporation of a zeotrope.
- **heat of combustion (HOC)** - The heat released or rejected when a substance is ignited, is determined as the difference in the enthalpy between the reactants and their products after combustion. Heat or enthalpy of combustion is often expressed as a specific value i.e. energy per mass (e.g., Btu/lb or [kJ/kg]).
- **toxic** - chemical possessing any of the properties categorized below can be termed as toxic:
  1. A chemical having a median lethal dose (LD50) of more than 50 mg/kg but not more than 500 mg/kg of body weight when examined on the albino rats weighing between 200 and 300 g each.
  2. A chemical having a median lethal dose (LD50) of more than 200 mg/kg but not more than 1000 mg/kg of body weight when examined by continuous contact for 24 hours (or less if death occurs within 24 hours) with the albino rabbits weighing between 2 and 3 kg each.
  3. A chemical having a median lethal concentration (LC50) within the air of more than 200 ppm but not more than 2000 ppm by volume of gas or vapor, or more than 2 mg/L but not more than 20 mg/L of mist, fume, or dust, when examined by continuous inhalation for one hour (or less if death occurs within one hour) to the albino rats between 200 and 300 g each by weight.
- **toxicity** - the ability of refrigerant to be harmful or lethal due to acute or chronic exposure by contact, inhalation, or ingestion. The effects of concern include carcinogens, poisons, reproductive toxins, irritants, corrosives, sensitizers, hepatoxins, nephrotoxins, neurotoxins, agents that act on the hematopoietic system, along with agents that damage the lungs, skin, eyes, or mucous membranes but are not limited to the only above mentioned consequences. Temporary discomfort at a level that is not impairing, will be excluded from the standardization.

- **refrigerant concentration limit (RCL)** - RCL is the limit of refrigerant in air determined with the usage of standards along with intentions of reducing the risks of acute toxicity, asphyxiation, and flammability hazards in occupied enclosed spaces. [43]
- **flammable concentration limit (FCL)** - FCL is the limit of refrigerant in air determined with the usage of standards along with intentions of reducing the risks of fire or explosion in occupied enclosed spaces.
- **threshold limit value time-weighted average (TLV-TWA)**: the time-weighted average concentration for a normal eight-hour workday and a forty-hour workweek to which nearly all workers may be repeatedly exposed, day after day, without adverse effect [44].
- **workplace environmental exposure level (WEEL)** - an occupational exposure limit set by the Toxicology Excellence for Risk Assessment (TERA) Occupational Alliance for Risk Science (OARS) 5 (previously issued by American Industrial Hygiene Association [AMA]); the TWA concentration, measured in the worker breathing zone, for a normal eight-hour workday and 40-hour workweek for which it is believed that nearly all workers can be repeatedly exposed without adverse health effects. OARS-WEEL values may be expressed as time-weighted average TWA concentrations, short-term exposure levels (STELs), or ceiling values [45].
- **zeotropic** - a blend or a mixture consisting of multiple components with different volatilities that, change the volumetric composition and saturation temperatures as they evaporate (boil) or condense at constant pressure when used in a vapor compression refrigeration cycle. Derivation of the word Zeotropic is from the Greek words zein (to boil) and tropos (to change).

Apart from the characteristics mentioned above, the vital characteristics which have major environmental impacts are Ozone Depletion Potential, Global Warming Potential, and Total Equivalent Warming Impact, the definition for which will be given below [43] :

- **ODP - Ozone Depletion Potential** : It is an indicator of the expected depletion of the ozone layer in relation to the impact caused by R11 (Tricholoroflouromethane). The value of ODP for R11 is 1, hence the ODP value ranges from 0-1 [46].
- **GWP - Global Warming Potential** : It is a measure of the amount of heat a greenhouse gas traps in the atmosphere comparing it with the same amount of $\mathrm{CO_2}$. GWP value for $\mathrm{CO_2}$ is 1, higher the GWP value, higher the emissions [46]. The value ranges from 1 - 15000 approximately.
- **TEWI - Total Equivalent Warming Impact**: The sum of greenhouse gas emissions in a direct and indirect way is the total equivalent warming impact [47]. It does depend on the GWP value of the refrigerant.

There are majorly four types of refrigerants that are generally used for heat pump systems [48], [43]:

- **CFCs** : Abbreviated from ChloroFluoroCarbons, are the compounds having Chlorine, Fluorine and Carbons combined. Examples are R11, R12, R113, R114, R115, etc. having ODP of almost 1 and GWP also very high somewhere around 10000. These are the first type of refrigerants that were used in various applications, but due to higher ODP and GWP values, it is not sustainable to use them to the current date.
- **HCFCs** : Abbreviated from HydroChloroFluoroCarbons, are the compounds having Hydrogen, Chlorine, Fluorine and Carbons combined. Examples are R22, R123, R124, R141b, etc. having ODP and GWP lesser compared to CFCs. These are the first upgradation from CFCs type of refrigerants which were used in various applications, but due to mediocre but still comparatively higher ODP and GWP values, it is not the best solution for refrigerant environmental impacts.
- **HFCs**: Abbreviated from HydroFluoroCarbons, are the compounds having Hydrogen, Fluorine and Carbons combined. Examples are R23, R32, R125, R134a, R143a, etc. These have an ODP of 0 but GWP is higher. These are the leading replacements for CFCs and HCFCs type of refrigerants and are used in almost every application nowadays, but due to the presence of high GWP values, search for alternatives is mandatory.

Table 2.1: Safety group classification as per toxicity and flammability [43]. The table shows how a refrigerant is assigned to a particular group. So, if for a refrigerant the Safety group class is A2L, which shows that the refrigerant has lower flammability as well as toxicity respectively.

Properties	Lower Toxicity	Higher Toxicity
Higher Flammability	A3	B3
Flammable	A2	B2
Lower Flammability	A2L	B2L
No flame propagation	A1	B1

Table 2.2: Refrigerant nomenclature is shown in the table 2.2. It contains information like the number assigned to the refrigerant, chemical IUPAC name, and chemical formula. For pure components, the chemical formula is given but for mixtures/blends, the composition of pure components is given[43]

Refrigerant Number	**Chemical Name**	**Chemical Formula**
R11	trichlorofluoromethane	CCl_3F
R22	chlorodifluoromethane	$CHClF_2$
R32	difluoromethane	CH_2F_2
R290	Propane	C_3H_8
R600a	2-methylpropane	$CH(CH_3)_2CH_3$
R717	Ammonia	NH_3
R744	Carbon dioxide	CO_2
R1270	Propylene	$CH_3CH{=}CH_2$
R1234yf	2,3,3,3-tetrafluoro-1-propene	$CH_3CF{=}CH_2$
R1234ze	trans-1,3,3,3-tetrafluoro-1-propene	$CH_3CH{=}CFH$
R410A	Puron	R32/R125(50/50)
R454C	Opteon™ XL20	R-32/1234yf (21.5/78.5)
R516A	ARM-42	R-1234yf/134a/152a (77.5/8.5/14.0)

- **Natural Refrigerants** : Examples of natural refrigerants are R290, R600a, R717, R718 & R744 etc. These types of refrigerants have ODP of 0 and even GWP in a single digit which is comparatively negligible when compared to CFCs, HCFCs, and HFCs. A lot of research is going on to use natural refrigerants in today's applications and hopefully, these will be the prominent refrigerants used worldwide in the upcoming years. [48]

As per the regulations given by EU 517/2014 [4], it is distinctly mentioned that Mobile/Movable air-conditioning systems have to use refrigerants with a GWP value of less than 150 from January 2020. So, the major task is to find a suitable refrigerant that follows this rule along with suitable characteristics for the application.
After doing the market survey, it was found that for Air-Air stationary and mobile systems, the predominantly used refrigerant is R410A having GWP of 1924 as per [15], but as it does not fulfill the criteria of GWP for this project case, alternatives have to be found. GWP versus density table provided by Danfoss (A company working with Heat Pumps) [49], is shown below in Figure 2.11.

So, as observed in figure 2.11, few refrigerants are taken into consideration for this project. The table 2.3 given below shows the refrigerants considered to validate their ability with an analysis in Aspen along with their properties and features, specifically denoting specific parameters which define their ability to work under a long range of applications. Table 2.1 shows the safety group classification of Refrigerants as per [43], which mostly focuses on the toxicity and flammability of the refrigerant. Every refrigerant is given a safety class which denotes its toxic and flammability properties. As an example, R290 (Propane) has a safety group class of A3, so from table 2.1, Propane is less toxic but highly flammable, just like this, every refrigerant is given a specific safety group class.

Table 2.3: Comparison of Refrigerant's properties is given in the table [15]. It compares the refrigerant information which is considered for this project, i.e. the potential refrigerant for mobile applications. Properties like safety group, GWP, crucial thermodynamic parameters like boiling point, critical temperature and pressure, and more, etc. [43] The market availability is given for refrigerants i.e. some of them are still in the research stage and are not available to be used commercially [50]

Refrigerants	**R 290**	**R 454C**	**R 455A**	**R 516A**	**R 1270**	**R 600a**	**R 1234yf**	**R 1234ze**	**R 717**
Safety Group	A3	A2L	A2L	A2L	A3	A3	A2L	A2L	B2L
GWP	3	146	146	131	2	3	<1	<1	0
ODP	0	0	0	0	0	0	0	0	0
Critical Temperature (°C)	96.74	85.67	85.61	96.63	91.06	134.7	94.70	109.4	132.4
Critical Pressure (bar)	42.51	43.19	46.54	36.15	45.55	36.29	33.82	36.35	113.6
Boiling Temperature (°C)	-42.11	-45.56	-52.02	-29.39	-47.62	-11.75	-29.49	-18.97	-33.32
Triple point (°C)	-187.6	-	-	-	-185.2	-159.4	-	-104.5	-
Mixtures	No	Yes	Yes	Yes	No	No	No	No	No
Nominal glide (°C)	-	7.81	12.9	-	-	-	-	-	-
Market Availability	Yes	Yes	Yes	No	Yes	Yes	Yes	Yes	Yes

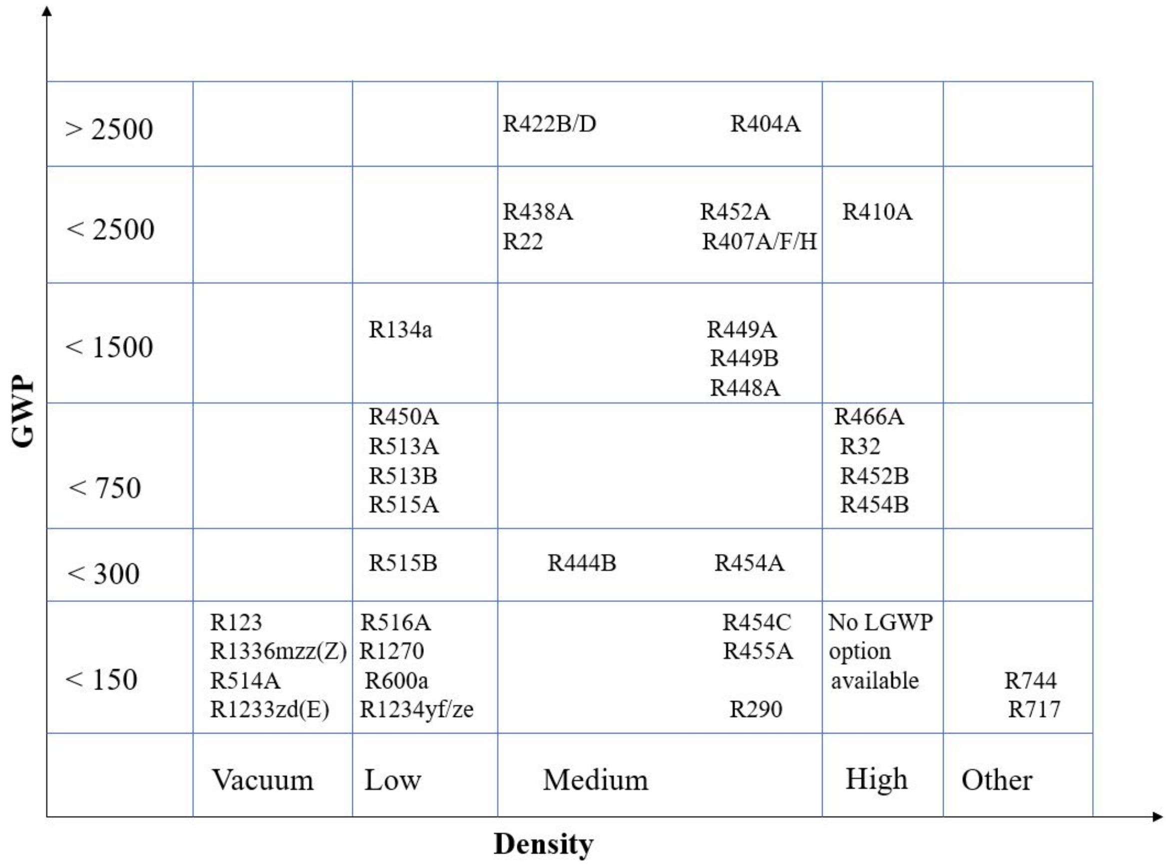

Figure 2.11: GWP of a refrigerant(Global Warming Potential is a dimensionless number denoting the measure of the amount of heat a greenhouse gas traps in the atmosphere comparing it with the same amount of CO_2 for a particular refrigerant is given on the y-axis) versus Density (The quantity divided by unit kg/m^3 is a dimensionless number for a refrigerant is denoted on the x-axis) is depicted in the graph [49]. It helps in choosing a particular refrigerant depending on the regulations given by EU [4]. Source: Danfoss

Through table 2.11 appropriate refrigerant can be selected to follow the GWP rule but still take into consideration of density. If the density is lower, then a higher volumetric flow rate has to be given to attain a higher mass flow rate. As most of the time, the less dense refrigerant will need higher mass flow rates to achieve the desired effect, so density has to be as high as possible. But there is no direct replacement for R410A, with similar density but lower GWP, so less dense refrigerant has to be analyzed to keep it as a potential replacement.

Table 2.3 shows the comparison of refrigerants concerning a few characteristics. Starting with the nomenclature using the letter R. For the nomenclature and chemical compound names refer to table 2.2. After that, a safety group class is provided for all as per table 2.1. The environmental impacts have to be assessed for these refrigerants, [51], [52], So, GWP i.e. Global Warming Potential as defined in 2.4.1, is given for all refrigerants, which satisfies the condition of value is less than 150 [4]. Similarly, ODP i.e. Ozone Depletion Potential as defined in 2.4.1, is provided for all refrigerants, having a magnitude of 0 for all, which makes them less harmful in terms of ozone layer consequences. Critical temperature and pressure as defined in 2.4.1 are given, the higher the value of these parameters, the higher the range of working capacity [53]. The boiling point is the temperature at which refrigerant starts to boil or evaporates, the lower the boiling point, the better the working capacity. The triple point is the temperature at which the solid, liquid, and vapor phase of refrigerant exists, expected to have a value as low as possible. More than these parameters, even other parameters have to be examined [54]. Table consists of refrigerants with blends/mixtures as defined in 2.4.1 & 2.4.1, as well as single fluid components. Glide as defined in 2.4.1, is provided for two blends, as others have zero glide being single fluid components and one being an azeotropic mixture. Next, the market availability showcases which refrigerants are already being sold in the market and how many are still in their research phase but are considered to be potential refrigerants in near future, from the table, just the refrigerant R516A is not available in the market, but still will be considered for the analysis to check the expected performance from it.

3

System Modelling

The system has to be modeled in order to achieve the expected outcomes from the actual system. Modeling simulation will be carried out in Aspen Plus, through which system parameters and performance will be examined for all the considered refrigerants from the table 2.3. In order to provide initial parameters or the bracket in which most of the basic component parameters lie, experimentation should be considered. So, to find out the data set for the initialization of parameters, an experiment was carried out on a system, the details of which are given ahead.

3.1. Experimental Overview

An experiment was carried out on a system that is currently available but uses R410A. The data set obtained from the experiments will be used for the inputs in the Aspen model. The system used for experimentation, whose specifications are given below in table 3.1:

Table 3.1: System specifications are listed in the table, showcasing the basic parameters which are used to define an Air-Air heat pump system, like capacity, the refrigerant used, system dimensions, the weight of the system along with operation limits, etc.

Model	C17R
Rated Cooling capacity (kW)	17
Rated Heating capacity (kW)	20
Rated power input (kW)	5.7
Air circulation (m^3/h)	3500
Operating limit for cooling (Max & Min)[indoor/outdoor,$^\circ$C]	30/40 & 22/20
Operating limit for heating (Max & Min)[indoor/outdoor,$^\circ$C]	20/20 & 10/11
Refrigerant used	R410A
System dimensions (mm^3)	1200×1400×1800
System weight (kg)	305

The experiment was carried out to check both the effects i.e. Heating and cooling for a particular ambient temperature. The values received for which are listed below in tables 3.2 and 3.3. It shows all the parameters which were directly obtained from the system, it was run 5 times to see how each of the parameters fluctuates in every run.

The parameters for the cooling experiment are listed as follows in table 3.2. Similarly, the parameters for the heating effect are shown in table 3.3

Table 3.2: System parameter specifications are listed in the table, showcasing the thermodynamic parameters like pressure, temperature, etc. These will be used to calculate the performance of the cooling effect provided by the system.

Data set number	1	2	3	4	5
Higher Pressure (bar)	18	18.7	18.3	19.1	19.2
Lower Pressure (bar)	6	6.6	6.5	6.6	6.7
Condensation Temperature (°C)	29	31	30	32	32
Evaporation Temperature (°C)	-4	-1	-2	-1	-1
Temperature of refrigerant leaving condenser (°C)	27	29	27	30	29
Temperature of refrigerant leaving evaporator (°C)	1	2	1	3	2
Subcooling (°C)	3	2.3	3	2.1	2.7
Superheating (°C)	5.5	4.6	4.6	4.7	4.2
Compressor discharge temperature (°C)	46	53	47	56	52
Ambient temperature (°C)	22	21.5	22	22	22
Room temperature (°C)	21	20.5	21	21	21
Air leaving the condenser temperature (°C)	29	28	29	29	29
Air entering the room temperature (°C)	10	9	9.8	9	9
Valve position (%)	68	57	63	54	60

Table 3.3: System parameter specifications are listed in the table, showcasing the thermodynamic parameters like pressure, temperature, etc. These will be used to calculate the performance of the heating effect provided by the system.

Data set number	1	2	3	4	5	6
Higher Pressure (bar)	28.2	30.1	30.7	31.3	31.6	31.6
Lower Pressure (bar)	8.5	8.8	8.8	8.8	8.8	8.8
Condensation Temperature (°C)	48	50	51	51	52	52
Evaporation Temperature (°C)	5	6	6	6	6	6
Temperature of refrigerant leaving condenser (°C)	46	48	49	50	50	50
Temperature of refrigerant leaving evaporator (°C)	10	10	10	10	11	10
Subcooling (°C)	2.9	2	2	1.8	1.8	1.7
Superheating (°C)	5.4	4.1	4	4.4	4.7	4.4
Compressor discharge temperature (°C)	70	70	72	75	76	76
Ambient temperature (°C)	18	17	17	17	17	17
Room temperature (°C)	21	21	21	21	21	21
Air leaving the condenser temperature (°C)	35	35	36	41	38.2	42
Air entering the room temperature (°C)	14	15	13.2	14	13.2	15
Valve position (%)	60	48	42	32	28	28

3.1.1. Remarks

Parameter data obtained from the experiment were analyzed to check how the system was designed in terms of components. The evaporator and condenser area was measured and were found to be approximately 0.5 m^2 and 1.5 m^2 respectively. As, these parameters could provide a maximum of 17 kW of cooling capacity and 20 kW of heating capacity, but in this project, the system has to be designed to attain a maximum of 40 kW of cooling and 60 kW of heating capacity, for which all the parameters have to be assessed. The experiment did not give any pressure drop values across the component as well as the exact capacity provided at a particular ambient temperature, so this data is not purely complete but can surely provide a basis for the fundamental parameters which can be used for the input for Aspen Modelling.

Table 3.4: Peng Robinson parameters for all considered refrigerants are provided. It consists of acentric factor ω, critical pressure P_c and critical temperature T_c respectively. These values will be used in the equation 3.1 to find vapor pressure at a particular temperature.

Refrigerant	ω	P_c (bar)	T_c (K)
R290	0.15251	42.51	369.901
R1270	0.14325	45.55	364.21
R1234yf	0.2753	33.82	367.85
R1234ze	0.31314	36.35	382.55
R717	0.25609	113.6	405.55
R600a	0.18428	36.29	407.85
R32	0.27661	57.84	351.28
R744	0.22551	73.82	304.17
R134a	0.32694	40.56	374.18
R152a	0.27524	45.17	386.438

3.2. Aspen Modelling

The heat pump system modeling is a crucial method to create a prototype of the system. Through this, parameters can be analyzed, so that the actual performance of the system can be predicted. It predicts the approximate values but still, a range of values can play a critical role in summarizing a system. Aspen Plus is a process or system simulator, through which processes in the systems can be modeled [55]. Due to its ability to do complex calculations in comparatively lesser time, inbuilt features, and libraries for calculation methods, it is being used for this project. Heat pump systems containing all components along with air and refrigerant being the working medium are used and discussed ahead.

3.2.1. Calculation Method

Aspen Plus has numerous calculation methods to find out specific properties. For refrigerants, REFPROP software is used to calculate all the physical properties like boiling point, critical point, pressure fluctuations with temperature, saturated vapor, liquid conditions, etc. REFPROP method is a feature in Aspen, REFPROP uses data obtained from the NIST (National Institute of Standards and Technology), this data can be used for calculations, but due to the presence of air as a secondary medium, Aspen does not allow usage of this method due to air being in the supercritical region even at standard temperature and pressure conditions. So, alternatively, the Peng-Robinson equation of state can be used as a base method to do all the calculations related to refrigerant physical and chemical properties. Peng-Robinson uses the equation 3.1 given below, [56].

$$P = \frac{RT}{V_m - b} - \frac{a\alpha}{V_m^2 + 2bV_m - b^2}$$

Where,

$$\begin{aligned} a &= \frac{0.45724R^2T_c^2}{P_c} \\ b &= \frac{0.07780RT_c}{P_c} \\ \alpha &= \left(1 + (0.37464 + 1.55426\omega - 0.26992\omega^2)(1 - T_r^{0.5})\right)^2 \\ T_r &= \frac{T}{T_c} \end{aligned} \tag{3.1}$$

Peng-Robinson parameters i.e. a, b and α are dependant on R, T_c, P_c and ω values, which denote universal gas constant with a constant value of 8.3145 Jmol^{-1}K^{-1} but rest of the parameters differ for each refrigerant. For mixtures, all the individual component values are used and are given in the table 3.4.

To compare the values obtained from REFPROP and using the Peng-Robinson equation, a P-T

Table 3.5: Antoine coefficients for all considered refrigerants are provided. The table consists of A, B, and C coefficients respectively. These values will be used in the equation 3.2 to find saturation pressure at a particular temperature for the refrigerants which is shown in figure 3.1

Refrigerant	A	B	C
R290	9.688	2158.962	265.481
R1270	9.783	2153.292	268.501
R1234yf	9.813	2198.239	253.334
R1234ze	9.874	2236.922	245.265
R717	10.684	2271.954	245.914
R600a	9.342	2240.393	251.756
R32	10.575	2235.928	263.683
R744	10.940	2055.394	278.138
R134a	9.865	2128.419	242.117
R152a	9.832	2182.773	246.313

curve is drawn for the refrigerant R1234yf. The values are obtained by varying the temperature from -50 to 90°C, and the pressure variation is checked for both the calculation methods as shown in figure 3.1. As can be seen, the error between the values is minimum, hence Peng-Robinson will be used as a base method for the calculation in the Aspen Model. The graphs are calculated by the Antoine equation which is given by equation 3.2, where P^s is the saturation pressure, T is temperature, and A, B, and C are Antoine coefficients specific to substances. These coefficients for all the refrigerants listed in table 3.4 are given below in table 3.5.

$$log(P^s) = A - \frac{B}{C + T} \tag{3.2}$$

3.2.2. Solver

Aspen Plus has multiple options of solvers to choose from. By default, it is set to the Sequential Modular method. In this method, the model simulation executes one unit operation at a time, but the sequence of operation of blocks has to be provided by the user. The calculated streams for a specific block will act as an input for the consecutive block operation, in a single mode of operation. The algorithm and its improvement can be found at [57]. For this project, Simulation Modular Method will be used to find the system parameters.

3.2.3. Convergence Method

Multiple convergence methods are available in Aspen Plus, for tear streams, optimization, etc. So, it is important to know what is the base method for the convergence of the simulation. These will be briefly discussed. A tear stream is a stream that recycles in the system containing information like mass flow, pressure, temperature, enthalpy, etc. calculated by iterations. It can be any of the streams in the system loop. So, for tear stream convergence, the parameters are as follows: Tolerance of 0.0001, and the stream converges when equation 3.3 is valid for all of the variables. State option, in this either Pressure or Enthalpy or both, can be kept to be converged for every simulation. In this model, the state of Pressure & Enthalpy convergence option has been chosen.

$$-tol \leq \frac{X_{calculated} - X_{assumed}}{X_{assumed}} \leq tol \tag{3.3}$$

Default convergence methods are given as follows:

- **Tears convergence - Wegstein Method**: This method is used for tear stream convergence due to its quickness and reliability. It uses extrapolation for the direct substitution iteration method. It can be applied to any number of streams simultaneously. At the default settings, it keeps 30 as the maximum number of flowsheet calculations. This method can be controlled by specifying the upper and lower limits of the acceleration parameter q, which can be calculated as given in the

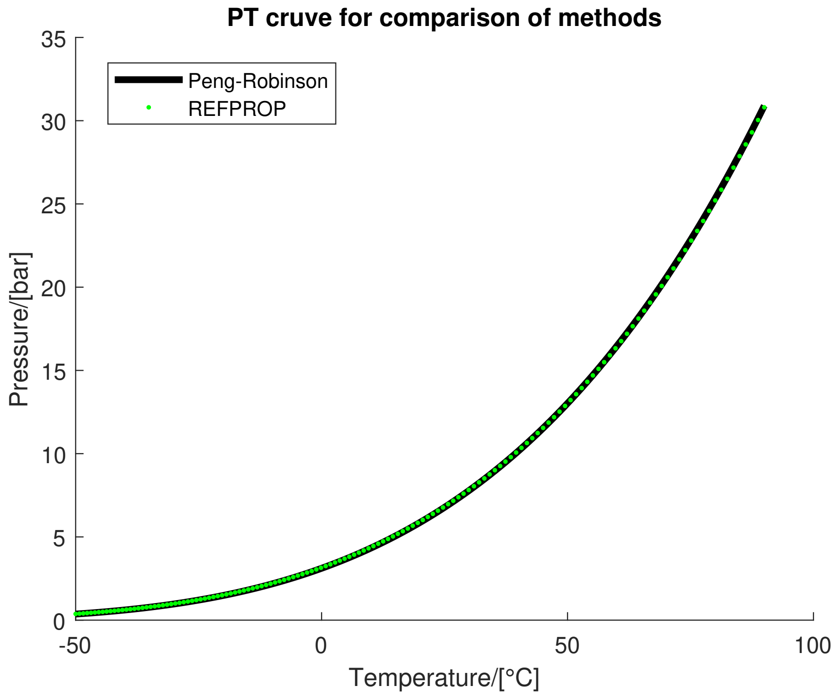

Figure 3.1: Pressure (the quantity Pressure divided by the unit bar is the dimensionless number that is listed on the y-axis) versus Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) curve is shown. The black solid line depicts the values obtained using the Peng-Robinson equation and the green points on it depict the values from REFPROP. The values almost match at lower and medium pressures, but at higher pressure, an offset is found.

equation 3.4, where, X is an estimate of the tear stream variable, k is the iteration number and G(X) is the resulting calculated value of the variable.

$$q = \frac{s}{s-1}$$
$$s = \frac{G(X_k) - G(X_{k-1})}{X_k - X_{k-1}} \tag{3.4}$$

The new estimate is calculated as,

$$X_{k+1} = X_k + (1-q)(G(X_k) - X_k)$$

The default lower and upper limits for q are -5 and 0 respectively. The effect of q on convergence varies depending on the magnitude. If the magnitude is less than 0, then it is termed acceleration. If the value is equal to zero, then it resembles with direct substitution method. If the value is between 0 to 1, excluding the extremities, then the damping effect is seen in the convergence.

- **Tears convergence - Direct Method**: The calculated value from the previous pass is used to estimate the new value. It can be presented as in equation 3.5, where, X is an estimate of the tear stream variable, k is the iteration number and G(X) is the resulting calculated value of the variable.

$$X_{k+1} = G(X_k) \tag{3.5}$$

  This method is slow but provides better results and stability compared to other methods and also ease of finding errors during the calculation. It is the same as Wetgstein if the upper and lower limit is set to 0.

- **Single Design Spec. - Secant Method**: It is a linear approximation method with higher-order enhancements. It is the default method for single design specifications convergence for a model. Maximum 30 flowsheet evaluations are carried out, with a regular step size of 0.01 to a maximum of 1 with tolerance of $1\text{x}10^{-8}$.

- **Multiple design specs and Tears & design specs - Broyden Method**: It is an approximate linearization method, which is a modification of Newton's method. It is faster than Newton's but not as reliable. It can converge multiple design specifications or tear streams and design specifications simultaneously. Similar to other methods has by default a maximum of 30 flowsheet evaluations, with a tolerance of 0.001, and uses direct substitution for the first two iterations then acceleration is used and has lower and upper bounds of -5 and 0 respectively.

- **Optimization - BOBYQA**: This method is being used for this model to carry out the optimization feature. It is an acronym for Bound Optimization BY Quadratic Approximation. It has maximum flowsheet evaluations of 1000. It uses constraints to do the simultaneous optimization. For detailed information, please refer to [58].

3.2.4. Pressure-Temperature dependency

The pressure-Temperature curve is an important curve to determine the quality of a refrigerant at a particular point. So, at a specific temperature, what is the magnitude of pressure that denotes if the refrigerant is saturated liquid, saturated vapor, superheated or subcooled? This can be determined for single component fluids using the relation given in equation 3.1 and 3.2. The Pressure versus temperature graph for all of the considered single components is given in figure 3.2. Through this, we can determine the system parameters by checking how pressure fluctuates with temperature and try to keep both of these values under critical conditions. For binary mixtures i.e. a mixture containing two single component fluids, a T-xy and P-xy curve is obtained instead of a P-T curve. Due to the presence of two individual components having different properties, it is observed that due to changes in the boiling and dew points, with respect to the vapor and liquid fraction, we get two curves for the temperature at constant pressure and similarly for pressure at constant temperature [59]. Hence, for the refrigerant R454C which is a mixture of R32 and R1234yf, P-xy and T-xy diagrams are shown in figures 3.4 & 3.3 for different values of temperatures and pressures respectively. But for a ternary mixture, it is not possible to find either of the two graphs mentioned above, but a ternary diagram or Residue curve map is obtained with the change of mole fraction of all the individual components through which it can be understood what are the pressure and temperature conditions, belonging to each individual component in liquid or vapor state [60]. It is a crucial diagram for distillation processes [61]. So, for R455A and R516A, the residue curve map has to be plotted, for which refer to [60].

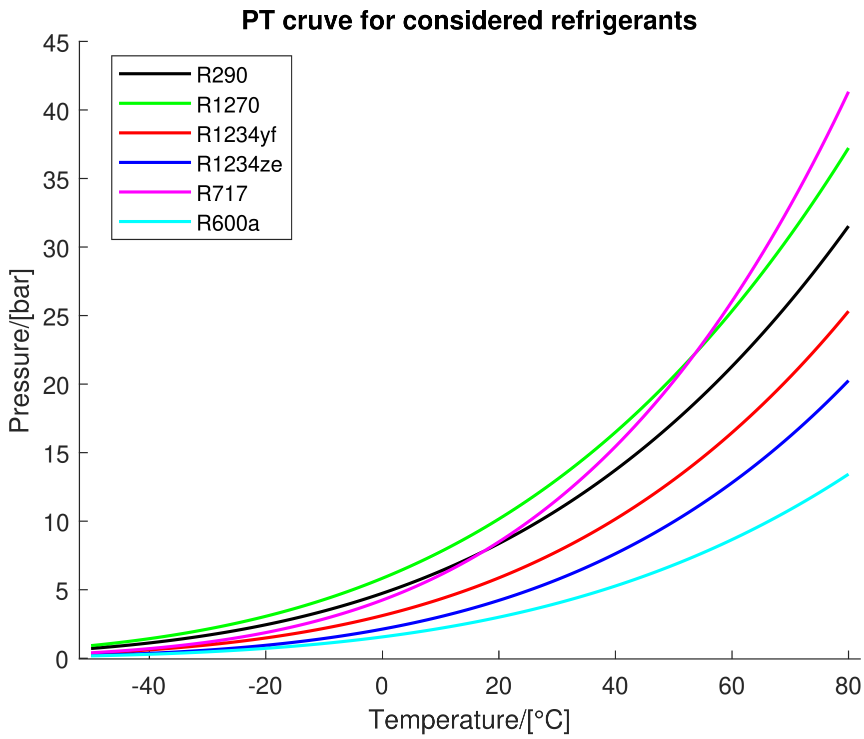

Figure 3.2: Pressure (the quantity Pressure divided by the unit bar is the dimensionless number that is listed on the y-axis) versus Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) curve is shown for refrigerants R290, R1270, R1234yf, R1234z, R717, and R600a respectively.

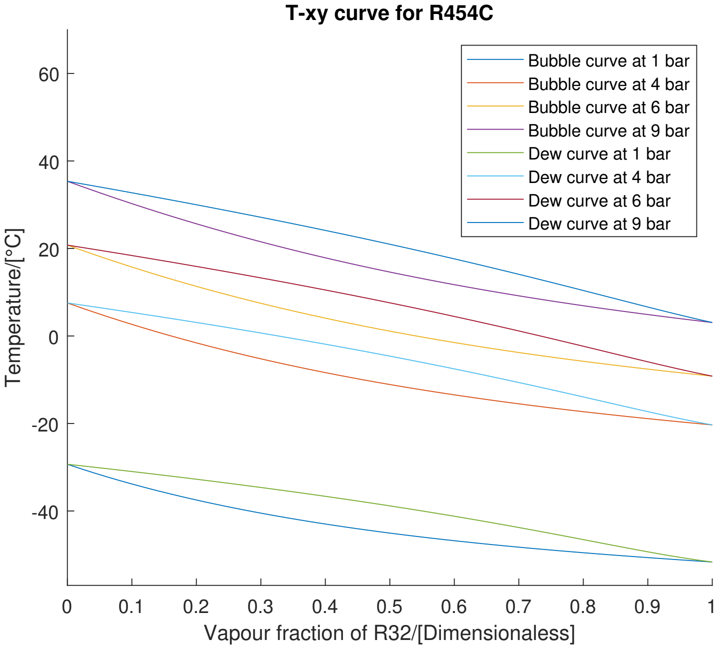

Figure 3.3: Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the y-axis) versus Vapour fraction (the quantity Vapour fraction is the dimensionless number that is listed on the x-axis) curve is shown for refrigerant R454C. The curves are obtained at different pressure levels, 1, 4, 6, and 9 bar. The figure shows the bubble and dew curves for each of the pressure levels.

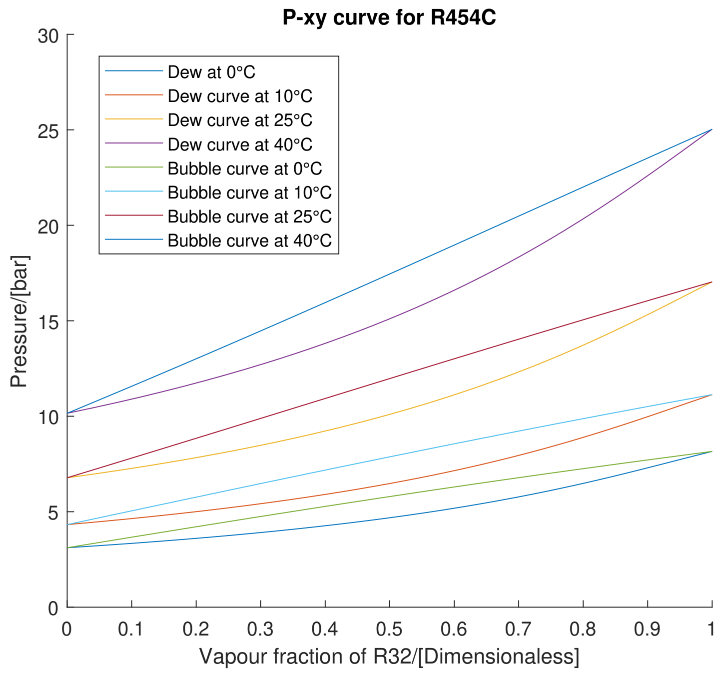

Figure 3.4: Pressure (the quantity Pressure divided by the unit bar is the dimensionless number that is listed on the y-axis) versus Vapour fraction (the quantity vapor fraction is the dimensionless number that is listed on the x-axis) curve is shown for refrigerant R454C. The curves are obtained at different temperature levels, 0, 10, 25, and 40 °C. The figure shows bubble and dew curves for each of the temperature levels.

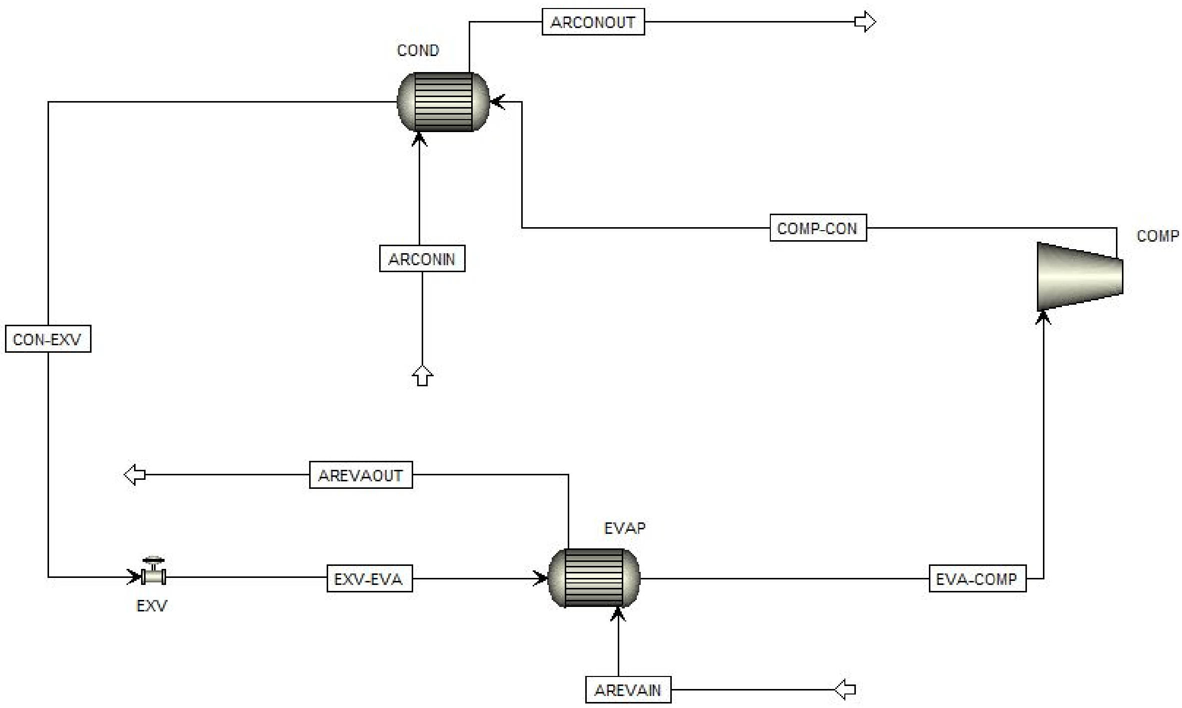

Figure 3.5: Schematic of heat pump system model created in Aspen plus, consisting of all components Compressor, Condenser, Expansion valve, and evaporator, along with the streams of air entering and leaving both the heat exchangers. This is a vapor compression refrigeration cycle. A compressor is denoted by COMP, evaporator by EVAP, condenser by COND, and expansion valve by EXV respectively connected with respective streams.

3.2.5. Aspen Model Schematic

The heat pump system model with all the components like the compressor, condenser, expansion valve, and evaporator is shown in figure 3.5. Due to the constraint of putting a maximum of 8 letters to name a component or a stream, short forms of the names have been used, which are listed as follows:

- COMP: COMP denotes the compressor used in the system. An isentropic compressor is used, the calculations for which will be discussed ahead. It is connected by the suction and discharge stream of EVA-COMP and COMP-CON respectively.
- COND: COND denotes Condenser used in the system, the calculations for which will be discussed ahead. It is connected by two inlet and two outlet streams, each for refrigerant and air. COMP-CON and CON-EXV are inlet and outlet streams for refrigerant, whereas, ARCONIN and ARCONOUT are inlet and outlet streams for air entering and leaving the condenser.
- EXV: EXV denotes the expansion valve used in the system, the calculations for which will be discussed ahead. It is connected by an inlet and outlet stream i.e. CON-EXV and EXV-EVA respectively.
- EVAP: EVAP denotes Evaporator used in the system, the calculations for which will be discussed ahead. It is connected by two inlet and two outlet streams, each for refrigerant and air. EXV-EVA and EVA-COMP are inlet and outlet streams for refrigerant, whereas, AREVAIN and AREVAOUT are inlet and outlet streams for air entering and leaving the evaporator.

3.3. Modelling the system for Cooling

The system has to be designed separately for cooling and heating as the desired effect changes, and the component delivering the effect also changes, i.e. when cooling is taken into consideration, the coefficient of performance is obtained by verifying the evaporator capacity over work done by the compressor, whereas for heating, it becomes, condenser capacity over work done. So, the calculations, initialization, and assumptions for the model can be given as follows.

3.3.1. Calculations

Every component follows a set of methods to find a particular parameter required to complete the loop of the flowsheet. So, the methods under every component are listed below:

- **COMP**: It is the compressor used in the system. The method through which calculations of the compressor were carried out is the Isentropic method, for which isentropic efficiency and pressure ratio are given or obtained to find discharge temperatures, enthalpies, entropies etc. So, the isentropic efficiency can be found using pressure ratio [39], and the relation is:

$$\eta_{\text{is}} = -0.0014\left(\frac{P_{\text{discharge}}}{P_{\text{suction}}}\right)^4 + 0.0297\left(\frac{P_{\text{discharge}}}{P_{\text{suction}}}\right)^3 - 0.236\left(\frac{P_{\text{discharge}}}{P_{\text{suction}}}\right)^2 + 0.8477\left(\frac{P_{\text{discharge}}}{P_{\text{suction}}}\right) - 0.4579 \quad (3.6)$$

  The equation 3.6 can be used only if the pressure ratio is in the range of 2-9, so if the pressure ratio is above 9, equation 3.7 is used.

$$\eta_{\text{is}} = 0.874 - 0.0135\left(\frac{P_{\text{discharge}}}{P_{\text{suction}}}\right) \quad (3.7)$$

  The calculated isentropic efficiency can be used to find the enthalpy after compression using the equation 3.8 given in [39], the ratio of isentropic enthalpy difference, in which the process does not have any entropy change with respect to the enthalpy difference of real process in which entropy changes during the compression. This can be used to find actual enthalpy after compression, which gives the overall work done by the compressor.

$$\eta_{\text{is}} = \frac{\Delta h_{\text{is}}}{\Delta h_{\text{real}}} \quad (3.8)$$

  Work done can be found using actual enthalpy difference times the refrigerant mass flow rate as given in the equation,

$$\dot{W}_{\text{in}} = \dot{m}(\Delta h_{\text{compressor}}) \quad (3.9)$$

- **COND**: Condenser used in the system is a heat exchanger, that helps in condensing the superheated refrigerant to saturated liquid with the transfer of heat with ambient air. This component rejects heat into the atmosphere, increasing the air temperature and decreasing the refrigerant temperature. It is defined by the area (surface) and overall heat transfer coefficient U and the temperature of all the streams entering and leaving the heat exchanger. So, the heat rejected can be given as,

$$\dot{Q}_{\text{h}} = \dot{m}(\Delta h_{\text{condenser}}) = UA(\Delta_{\text{LMTD}}) \quad (3.10)$$

  Where LMTD is the log mean temperature difference between the air streams and refrigerant streams. LMTD value should be as high as possible to have a larger driving force and reduces the heat exchanger area in order to get higher capacity [39], [62]. The value of LMTD should exceed 5 K in order to have optimum heat duty and heat exchanger specifications. The air entering and leaving the condenser at a particular temperature attains a specific magnitude of temperature at the condenser, which varies according to the overall heat transfer coefficient, area of the heat exchanger, the mass flow rate of air, the specific heat capacity of air at that temperature as well as the temperature of the air entering and leaving the condenser. One of the assumptions used to calculate condensation temperature is refrigerant has infinite heat capacity. The equation is given below:

$$T_{\text{COND}} = \frac{T_{\text{H-out}} - T_{\text{H-in}} e^{-\frac{(UA)_{\text{COND}}}{\dot{m} c_{p_{\text{H}}}}}}{1 - e^{-\frac{(UA)_{\text{COND}}}{\dot{m} c_{p_{\text{H}}}}}} \tag{3.11}$$

Hence, it is crucial to choose an optimum area for the heat exchanger in order to have desired condensation temperature as well as heat duty. It also decides the quality of refrigerant after the condensation, if it will be saturated liquid or subcooled, or a mixture of liquid and vapor. The minimum approach temperature is the temperature difference between the refrigerant stream and the air stream. This value is also provided in the model in order to avoid temperature crosses along any of the streams. The usual value of the minimum temperature approach is 5-15 K [39], it maintains the entropy production low hence enhancing the second law efficiency as shown in equation 2.8. The optimum value of the minimum approach can also make the heat exchanger economically feasible.

- **EXV**: The expansion valve used in the system, reduces the pressure of the condensed refrigerant to the evaporator pressure. It is a crucial component to decide the lower pressure of the system as well as the mass flow rate. The process is considered isenthalpic, where enthalpy remains constant but not isentropic, the entropy increases after the expansion as the randomness increases, hence even the quality changes from 0 (ideally) to 0.1-0.5.

- **EVAP**: Evaporator used in the system is a heat exchanger, that helps in evaporating or boiling the condensed and expanded refrigerant to saturated vapor with the transfer of heat with air inside a room or a specific location in case of cooling purposes. This component absorbs heat from that space, reducing the temperature and increasing the refrigerant temperature. It is defined by the area (surface) and overall heat transfer coefficient U and the temperature of all the streams entering and leaving the heat exchanger. So, the heat absorbed can be given as,

$$\dot{Q}_{\text{c}} = \dot{m}(\Delta h_{\text{evaporator}}) = UA(\Delta_{\text{LMTD}}) \tag{3.12}$$

Where LMTD is the log mean temperature difference between the air streams and refrigerant streams. LMTD value should be as high as possible to have a larger driving force and reduces the heat exchanger area in order to get higher capacity [39], [62]. The value of LMTD should exceed 5 K in order to have optimum heat duty and heat exchanger specifications. The air entering and leaving the evaporator at a particular temperature attains a specific magnitude of temperature at the evaporator, which varies according to the overall heat transfer coefficient, area of the heat exchanger, the mass flow rate of air, the specific heat capacity of air at that temperature as well as the temperature of the air entering and leaving the evaporator. One of the assumptions used to calculate evaporation temperature is refrigerant has infinite heat capacity. The equation is given below:

$$T_{\text{EVAP}} = \frac{T_{\text{C-out}} - T_{\text{C-in}} e^{-\frac{(UA)_{\text{EVAP}}}{\dot{m} c_{p_{\text{C}}}}}}{1 - e^{-\frac{(UA)_{\text{EVAP}}}{\dot{m} c_{p_{\text{C}}}}}} \tag{3.13}$$

Hence, it is crucial to choose an optimum area for the heat exchanger in order to have desired condensation temperature as well as heat duty. It also decides the quality of refrigerant after the condensation, if it will be saturated liquid or subcooled, or a mixture of liquid and vapor. The minimum approach temperature is the temperature difference between the refrigerant stream and the air stream. This value is also provided in the model in order to avoid temperature crosses along any of the streams. The usual value of the minimum temperature approach is 5-15 K [39], it maintains the entropy production low hence enhancing the second law efficiency as shown in equation 2.8. The optimum value of the minimum approach can also make the heat exchanger economically feasible.

3.4. Modelling the system for Cooling

3.4.1. Initialization & Assumptions

As discussed in 3.1, the values similar to those are expected from the new system, so by taking reference to those values and literature, some assumptions are considered and accordingly initialization of the system was carried out.

Assumptions

- The system behaves like the ideal vapor compression refrigeration cycle and has no pressure drop across any component or stream.
- Due to the application of the system being tents and agricultural purposes, the temperature outside the tent and inside will not be that different without any system being used. So, in this model, the temperature of the ARCONIN stream has to be greater than or equal to the temperature of AREVAIN, only for a few temperature values, this will be violated in the aspen model.
- Due to ideal behavior, superheating and subcooling are considered zero, i.e. after evaporation refrigerant is saturated vapor and after condensation, it is saturated liquid.
- Maximum duty by evaporator should be around 40 kW and 60 kW for the condenser.
- All the processes are steady-state and steady-flow.
- Kinetic and potential energy variations are negligible.
- The reference or dead state of the refrigerant is taken at ambient conditions, T_0 = 298.15 K and P_0 = 1 atm.
- 1°C = 273.15 K considered throughout the calculations.

Initialization

The initial values for all the components are as follows:

- **COMP**: The input for the compressor is given by pressure ratio and isentropic efficiency. So, the magnitude for which is, Pressure ratio = 6 and isentropic efficiency for this pressure ratio is calculated by the formula 3.6 which calculates, η_{isen} = 0.7118.
- **COND**: The input for the condenser is given by minimum approach temperature, area of the heat exchanger, overall heat transfer coefficient, and condition to calculate the duty of the heat exchanger which are as follows:
  - Area of the heat exchanger = 5 m^2, a bit higher than the experimental value as the desired effect is almost 3 times more. Even though this area is comparatively higher, the required area for a particular condition will be obtained through aspen to get the exact value.
  - Overall heat transfer coefficient = 1500 W/m^{2}K [39].
  - Condition for the duty calculation = The refrigerant should be saturated liquid after the condensation, i.e. quality being zero.
  - Minimum approach temperature = 10 K [39].
  - Pressure drop across both the streams = 0
- **EXV**: The input for the expansion valve is given by the reduced pressure, i.e. pressure obtained after the isenthalpic expansion. The value for which is taken as 4 bar.
- **EVAP**: The input for the evaporator is given by minimum approach temperature, area of the heat exchanger, overall heat transfer coefficient, and condition to calculate the duty of the heat exchanger which are as follows:

- Area of the heat exchanger = 2.5 m^2, a bit higher than the experimental value as the desired effect is almost 3 times more. Even though this area is comparatively higher, the required area for a particular condition will be obtained through aspen to get the exact value.
- Overall heat transfer coefficient = 1200 W/m^2K [39].
- Condition for the duty calculation = The refrigerant should be saturated vapor after the evaporation, i.e. quality being one.
- Minimum approach temperature = 5 K [39].
- Pressure drop across both the streams = 0

EVA-COMP: This is the input stream for the refrigerant flow, which connects the evaporator and compressor. The input parameters for this stream are:

- Pressure of the stream should be equal to the outlet pressure of the expansion valve. So, this value is 4 bar.
- Vapour fraction to be 1, i.e. saturated vapor is obtained after evaporation and ready for compression.
- Mass flow rate of refrigerant = 0.3 kg/s

ARCONIN: The stream indicated the ambient condition for air entering the condenser. The input parameters for which are:

- Temperature = 28°C
- Pressure = 1.01325 bar
- Mass/volume flow rate = 10800 kg/hr or 9580 m^3/hr

AREVAIN: The stream indicated the condition for air entering the evaporator and denotes the air from the room/tent. The input parameters for which are:

- Temperature = 25°C as an initial condition
- Pressure = 1.01325 bar
- Mass/volume flow rate = 9000 kg/hr or 7983 m^3/hr

3.4.2. Optimization

The assumptions and initialization values are given for all the refrigerants in the respective component and streams. But these values have to be modified for every change in temperature to get the highest coefficient of performance. So, the optimization feature is used to change a few parameters and get the maximized COP for those sets of conditions. The design condition for which is changing the isentropic efficiency for change in pressure ratio of all configurations using the formula 3.6. The change of parameters is as given in the table 3.6. Most of these parameters have a specified lower and upper bound, but the mass flow rate parameter upper and lower bound was different for every refrigerant due to their flammable and toxic properties. For some refrigerants, having a higher mass flow rate can lead to hazardous situations so the limit has to be changed. Apart from this property, the mass flow rate bounds have to be changed so as to satisfy the heat duty condition for condenser = 60 kW and evaporator = 40 kW respectively, as every refrigerant gives this duty at different mass flow rates due to their pressure-temperature dependency, and other parameters.
Along with these parameters some constraints have been given for the optimization method which are:

- Maximum evaporator duty should be 40 kW
- Maximum condenser duty should be 60 kW
- Vapour fraction of refrigerant entering the compressor should be 1, i.e. only vapor should enter the compressor.
- Vapour fraction of the refrigerant leaving the condenser should be 0, i.e. saturated liquid.

Table 3.6: System parameter specifications are changed from a lower to an upper limit to find the optimum values for every other parameter in the system. Component and stream variables are fluctuated to optimize the coefficient of performance. The values like pressure ratio, lower pressure of the system, U and area for heat exchangers, and mass flow rate with the initial values along with their lower and upper bounds are given.

Parameter	Initial value	Lower bound	Upper bound
Pressure ratio	6	2	10
Outlet pressure of expansion valve (bar)	4	1.5	6
Condenser Area (m^2)	5	2	7
Evaporator Area (m^2)	2.5	1	7
U value for condenser (W/m^{2}K)	1500	1000	2000
U value for evaporator (W/m^{2}K)	1200	600	2000
Mass flow rate of the refrigerant (kg/s)	0.3	0.01	0.5
Minimum temperature of approach for condenser (K)	15	‘10	20
Minimum temperature of approach for evaporator (K)	5	‘5	10

- The temperature after compression should be less than the critical temperature of the refrigerant in order to prevent super-criticality in the system. This is the maximum temperature obtained in the cycle so has to be less than the critical temperature.
- Discharge pressure after compression should be less than the critical pressure of the refrigerant.

After applying these bounds, the objective is to find the optimum values of the parameters from the table by taking into consideration all the constraints and maximizing the coefficient of performance. So, the solution convergence is obtained to achieve the maximum COP value for a particular ambient and room temperature.

3.4.3. Sensitivity analysis

Sensitivity analysis is carried out by changing the temperature of ARCONIN and AREVAIN, i.e. ambient temperature and temperature inside the room. By changing these two parameters, the system is optimized for every temperature change as discussed in 3.4.2 and gets the maximized value of the coefficient of performance. The sensitivity analysis parameters:

- AREVAIN - The temperature of the stream AREVAIN remains constant at 25°C.
- ARCONIN - The temperature of the stream ARCONIN changes from 21 to 40°C with a step size of 1°C.

The change of temperature is taken from 21°C, even though the desired temperature for cooling is 16°C. But according to [63] and [64], 16 to 20°C is considered to be buffer temperature where cooling or heating is not essential. So, this assumption is used throughout this model and hence the range of sensitivity analysis is decided.

3.4.4. Calculator block

In this block, all the necessary calculations are carried out and desired properties are obtained. So, for the cooling purpose, the COP is calculated as well as exergy calculation is done to compare the system with the Carnot system. All the variables like pressure, temperature, enthalpies, entropies, mass flow rates, vapor fraction, exergy losses across components, and component-specific parameters are tabulated in this block. This block gives the values after optimization of the variation of parameters in the sensitivity analysis. All the equations used to calculate all these variables are given below:

- COP: Coefficient of Performance for cooling purposes is obtained by the heat duty of the evaporator with respect to the work done. i.e.,

$$\mathrm{COP} = \frac{Q_{\mathrm{EVAP}}}{W_{\mathrm{COMP}}} \tag{3.14}$$

- Condensation and evaporation temperature are calculated by the equations 3.11 and 3.13.

- Exergy loss in compressor: The exergy loss is calculated using the formula given below which uses the mass flow rate of refrigerant, reference temperature of T_0 = 298.15 K, and entropy change due to compression.

$$Ex_{\text{COMP}} = \dot{m_{\text{ref}}} T_0 (s_2 - s_1) \tag{3.15}$$

- Exergy loss in condenser: As the reference state is considered to be the environment, hence the higher temperature attained in the system is replaced by the reference temperature. Hence, the exergy loss depends on the reference temperature T_0 = 298.15 K, the mass flow rate of refrigerant, entropy change, and specific heat duty given by:

$$Ex_{\text{COND}} = \dot{m_{\text{ref}}} T_0 \left(s_3 - s_2 + \frac{q_{\text{COND}}}{T_0} \right) \tag{3.16}$$

- Exergy loss in expansion valve: The exergy loss is calculated using the formula given below which uses the mass flow rate of refrigerant, reference temperature of T_0 = 298.15 K, and entropy change due to expansion.

$$Ex_{\text{EXV}} = \dot{m_{\text{ref}}} T_0 (s_4 - s3) \tag{3.17}$$

- Exergy loss in evaporator: The exergy loss in the evaporator depends on reference temperature T_0 = 298.15 K, evaporation temperature, the mass flow rate of refrigerant, entropy change, and specific heat duty given by:

$$Ex_{\text{EVAP}} = \dot{m_{\text{ref}}} T_0 \left(s_1 - s_4 - \frac{q_{\text{EVAP}}}{T_{\text{EVAP}}} \right) \tag{3.18}$$

- Total destructed exergy: Total exergy destructed is the summation of exergy losses across each component.

$$Ex_{\text{Total-loss}} = Ex_{\text{COMP}} + Ex_{\text{COND}} + Ex_{\text{EXV}} + Ex_{\text{EVAP}} \tag{3.19}$$

- Second law efficiency: Second law efficiency is the value of the ratio of actual performance to the Carnot performance of the system as given in equation 2.10, in this the T_{COND} is replaced by T_0. Through exergy losses also, we can find second law efficiency, with either of the formulas used, we should get the same value [36], [39].

$$\eta_{\text{second-law}} = 1 - \frac{Ex_{\text{Total-loss}}}{W_{\text{COMP}}} \tag{3.20}$$

  So, the second law efficiency depends on actual COP and Carnot COP as well as total exergy loss and work done by the compressor. One more method that can be used to calculate it is,

$$\eta_{\text{second-law}} = \frac{Q_{\text{EVAP}}}{W_{\text{COMP}}} \left(\frac{T_0}{T_{\text{EVAP}}} - 1 \right) \tag{3.21}$$

  So, using equations, 2.10, 3.20 and 3.21, the magnitude should be exactly the same.

Using all these assumptions, initialization, optimization, sensitivity analysis, and calculations, the refrigerant performance and the corresponding variables are examined for the desired effect of cooling.

3.5. Modelling the system for Heating

The system has to be designed separately for cooling and heating as the desired effect changes, and the component delivering the effect also changes, i.e. when heating is taken into consideration, the coefficient of performance is obtained by verifying the condenser capacity over work done by the compressor. So, as discussed in section 3.3, similarly for heating apart from the calculations, initialization and assumptions for the model will be given as follows. Calculations discussed in section 3.3.1, remain the same for heating as well, but the calculator block changes.

3.5.1. Initialization & Assumptions

As discussed in 3.1, the values similar to those are expected from the new system, so by taking reference to those values and literature, some assumptions are considered and accordingly initialization of the system was carried out.

Assumptions

- The system behaves like the ideal vapor compression refrigeration cycle and has no pressure drop across any component or stream.
- Due to the application of the system being tents and agricultural purposes, the temperature outside the tent and inside will not be that different without any system being used.
- Due to a change in the desired effect, now the evaporator acts as an outdoor unit dealing with ambient air whereas the condenser acts as an indoor unit dealing with the air inside the room/tent.
- Due to ideal behavior, superheating and subcooling are considered zero, i.e. after evaporation refrigerant is saturated vapor and after condensation, it is saturated liquid.
- Maximum duty by evaporator should be around 40 kW and 60 kW for the condenser.
- All the processes are steady-state and steady-flow.
- Kinetic and potential energy variations are negligible.
- The reference or dead state of the refrigerant is taken at ambient conditions, T_0 = 283.15 K and P_0 = 1 atm.
- 1°C = 273.15 K considered throughout the calculations.

Initialization

The initial values for all the components are as follows:

- **COMP**: The input for the compressor is given by pressure ratio and isentropic efficiency. So, the magnitude for which is, Pressure ratio = 4.5 and isentropic efficiency for this pressure ratio is calculated by the formula 3.6 which calculates, η_{isen} = 0.7101.
- **COND**: The input for the condenser is given by minimum approach temperature, area of the heat exchanger, overall heat transfer coefficient, and condition to calculate the duty of the heat exchanger which are as follows:
  - Area of the heat exchanger = 5 m^2, a bit higher than the experimental value as the desired effect is almost 3 times more. Even though this area is comparatively higher, the required area for a particular condition will be obtained through aspen to get the exact value.
  - Overall heat transfer coefficient = 1500 W/m^{2}K [39].
  - Condition for the duty calculation = The refrigerant should be saturated liquid after the condensation, i.e. quality being zero.
  - Minimum approach temperature = 10 K [39].
  - Pressure drop across both the streams = 0
- **EXV**: The input for the expansion valve is given by the reduced pressure, i.e. pressure obtained after the isenthalpic expansion. The value for which was 4 bar.
- **EVAP**: The input for the evaporator is given by minimum approach temperature, area of the heat exchanger, overall heat transfer coefficient, and condition to calculate the duty of the heat exchanger which are as follows:
  - Area of the heat exchanger = 2.5 m^2, a bit higher than the experimental value as the desired effect is almost 3 times more. Even though this area is comparatively higher, the required area for a particular condition will be obtained through aspen to get the exact value.

- Overall heat transfer coefficient = 1200 W/m^{2}K [39].
- Condition for the duty calculation = The refrigerant should be saturated vapor after the evaporation, i.e. quality being one.
- Minimum approach temperature = 5 K [39].
- Pressure drop across both the streams = 0

EVA-COMP: This is the input stream for the refrigerant flow, which connects the evaporator and compressor. The input parameters for this stream are:

- Pressure of the stream should be equal to the outlet pressure of the expansion valve. So, this value is 4 bar.
- Vapour fraction to be 1, i.e. saturated vapor is obtained after evaporation and ready for compression.
- Mass flow rate of refrigerant = 0.3 kg/s

ARCONIN: The stream indicated the ambient condition for air entering the condenser. The input parameters for which are:

- Temperature = 10°C
- Pressure = 1.01325 bar
- Mass/volume flow rate = 9000 kg/hr or 7983 m^3/hr

AREVAIN: The stream indicated the condition for air entering the evaporator and denotes the air from the room/tent. The input parameters for which are:

- Temperature = 12°C as an initial condition
- Pressure = 1.01325 bar
- Mass/volume flow rate = 10800 kg/hr or 9580 m^3/hr

3.5.2. Optimization

The assumptions and initialization values are given for all the refrigerants in the respective component and streams. But these values have to be modified for every change in temperature to get the highest coefficient of performance. So, the optimization feature is used to change a few parameters and get the maximized COP for those sets of conditions. The design condition for which is changing the isentropic efficiency for change in pressure ratio of all configurations using the formula 3.6. The change of parameters is as given in the table 3.7. Most of these parameters have a specified lower and upper bound, but the mass flow rate parameter upper and lower bound was different for every refrigerant due to their flammable and toxic properties. For some refrigerants, having a higher mass flow rate can lead to hazardous situations so the limit has to be changed. Apart from this property, the mass flow rate bounds have to be changed so as to satisfy the heat duty condition for condenser = 65 kW and evaporator = 45 kW approximately, as every refrigerant gives this duty at different mass flow rates due to their pressure-temperature dependency, and other parameters.
Along with these parameters some constraints have been given for the optimization method which are:

- Maximum evaporator duty should be 40 kW
- Maximum condenser duty should be 60 kW
- Vapour fraction of refrigerant entering the compressor should be 1, i.e. only vapor should enter the compressor.
- Vapour fraction of the refrigerant leaving the condenser should be 0, i.e. saturated liquid.
- The temperature after compression should be less than the critical temperature of the refrigerant in order to prevent super-criticality in the system. This is the maximum temperature obtained in the cycle so has to be less than the critical temperature.

Table 3.7: System parameter specifications are changed from a lower to an upper limit to find the optimum values for every other parameter in the system. Component and stream variables are fluctuated to optimize the coefficient of performance. The values like pressure ratio, lower pressure of the system, U and area for heat exchangers, and mass flow rate with the initial values along with their lower and upper bounds are given.

Parameter	Initial value	Lower bound	Upper bound
Pressure ratio	6	2	10
Outlet pressure of expansion valve (bar)	4	1.5	6
Condenser Area (m^2)	5	2	7
Evaporator Area (m^2)	2.5	1	7
U value for condenser (W/m^2K)	1200	600	2000
U value for evaporator (W/m^2K)	1500	1000	2000
Mass flow rate of the refrigerant (kg/s)	0.3	0.01	0.5
Minimum temperature of approach for condenser (K)	15	‘10	20
Minimum temperature of approach for evaporator (K)	5	‘5	10

- Discharge pressure after compression should be less than the critical pressure of the refrigerant.

After applying these bounds, the objective is to find the optimum values from the table by taking into considering all the constraints and maximizing the coefficient of performance. So, the solution convergence is obtained to achieve the maximum COP value for a particular ambient and room temperature.

3.5.3. Sensitivity analysis

Sensitivity analysis is carried out by changing the temperature of AREVAIN and ARCONIN, i.e. ambient temperature and temperature inside the room, following the condition as the temperature of ARCONIN is greater than or equal to AREVAIN. By changing these two parameters, the system is optimized for every temperature change as discussed in 3.5.2 and gets the maximized value of the coefficient of performance. The sensitivity analysis parameters:

- ARCONIN - The temperature of the stream ARCONIN remains constant at 10°C.
- AREVAIN - The temperature of the stream AREVAIN changes from -5 to 15°C with the step size of 1°C.

Change of temperature is taken up to 15°C, even though the desired temperature for cooling is 20°C. But according to [63] and [64], 16 to 20°C is considered to be buffer temperature where cooling or heating is not essential. So, this assumption is used throughout this model and hence the range of sensitivity analysis is decided.

3.5.4. Calculator block

In this block, all the necessary calculations are carried out and desired properties are obtained. So, for the cooling purpose, the COP is calculated as well as exergy calculation is done to compare the system with the Carnot system. All the variables like pressure, temperature, enthalpies, entropies, mass flow rates, vapor fraction, exergy losses across components, and component-specific parameters are tabulated in this block. This block gives the values after optimization of the variation of parameters in a sensitivity analysis. All the equations used to calculate all these variables are given below:

- COP: Coefficient of Performance for heating purposes is obtained by the heat duty of the condenser with respect to the work done. i.e.,

$$\mathrm{COP} = \frac{Q_{\mathrm{COND}}}{W_{\mathrm{COMP}}} \tag{3.22}$$

- Condensation and evaporation temperature are calculated by the equations 3.11 and 3.13.
- Exergy loss in compressor: The exergy loss is calculated using the formula given below which uses the mass flow rate of refrigerant, reference temperature of T_0 = 283.15 K, and entropy change due to compression.

$$Ex_{\mathrm{COMP}} = \dot{m_{\mathrm{ref}}} T_0 (s_2 - s1) \tag{3.23}$$

- Exergy loss in condenser: As the reference state is considered to be the environment, hence the higher temperature attained in the system is replaced by the reference temperature. Hence, the exergy loss depends on the reference temperature T_0 = 283.15 K, the mass flow rate of refrigerant, entropy change, and specific heat duty given by:

$$Ex_{\text{COND}} = \dot{m_{\text{ref}}} T_0 \left(s_3 - s_2 + \frac{q_{\text{COND}}}{T_{\text{COND}}} \right) \tag{3.24}$$

- Exergy loss in expansion valve: The exergy loss is calculated using the formula given below which uses the mass flow rate of refrigerant, reference temperature of T_0 = 283.15 K, and entropy change due to expansion.

$$Ex_{\text{EXV}} = \dot{m_{\text{ref}}} T_0 (s_4 - s3) \tag{3.25}$$

- Exergy loss in evaporator: The exergy loss in the evaporator depends on reference temperature T_0 = 283.15 K, evaporation temperature, the mass flow rate of refrigerant, entropy change, and specific heat duty given by:

$$Ex_{\text{EVAP}} = \dot{m_{\text{ref}}} T_0 \left(s_1 - s_4 - \frac{q_{\text{EVAP}}}{T_0} \right) \tag{3.26}$$

- Total destructed exergy: Total exergy destructed is the summation of exergy losses across each component.

$$Ex_{\text{Total-loss}} = Ex_{\text{COMP}} + Ex_{\text{COND}} + Ex_{\text{EXV}} + Ex_{\text{EVAP}} \tag{3.27}$$

- Second law efficiency: Second law efficiency is the value of the ratio of actual performance to the Carnot performance of the system as given in equation 2.9, in this the T_EVAP is replaced by T_0. Through exergy losses also, we can find second law efficiency, with either of the formulas used, we should get the same value [36], [39].

$$\eta_{\text{second-law}} = 1 - \frac{Ex_{\text{Total-loss}}}{W_{\text{COMP}}} \tag{3.28}$$

  So, the second law efficiency depends on actual COP and Carnot COP as well as total exergy loss and work done by the compressor. One more method that can be used to calculate it is,

$$\eta_{\text{second-law}} = \frac{Q_{\text{COND}}}{W_{\text{COMP}}} \left(1 - \frac{T_0}{T_{\text{COND}}} \right) \tag{3.29}$$

  So, using equations, 2.9, 3.29 and 3.28, the magnitude should be exactly the same.

Using all these assumptions, initialization, optimization, sensitivity analysis, and calculations, the refrigerant performance, and the corresponding variables are examined for the desired effect of heating. These simulations were carried out with zero pressure drop, but how pressure drop affects the behavior of every refrigerant will also be observed, which is discussed from 4.3.

4
Results

As discussed in 3.3, the model was analyzed and optimized for every refrigerant. But as per the literature survey, it was found that R717 (Ammonia) and R600a (Isobutane) are not compatible with this kind of system. Even though R717 has ODP = 0 & GWP = 0, due to its highly toxic nature, can not be easily used in domestic or residential applications but rather in industrial applications and focusing more on the refrigeration systems over air-conditioning [65], [66]. It is also being used in heat pumps for capacities ranging from 200kW and up to 8 MW [67].
Similarly, R600a, is widely used in refrigeration installations as well as in heat pumps to deliver temperatures higher than 80°C [65], but due to fire and explosion hazards, it is widely used in freezers, refrigerators and dehumidifiers [68]. R600a is a better alternative for R134a and R12 [69] and can work for domestic and industrial refrigeration due to optimum thermodynamic properties. Due to these reasons, these refrigerants do not qualify for mobile air-air systems and will not be considered for the modeling as well.

4.1. Results for Cooling Mode

The methods discussed for modeling, optimization, and sensitivity analysis carried out, and the parameters like evaporator duty, condenser duty, work done by the compressor, mass flow rate of the refrigerant, pressure ratio in the compressor, isentropic efficiency, second law efficiency was calculated and plotted against the ambient temperature conditions varying from 21 to 40°C and room temperature condition kept constant at 25°C. The results for which are given below for each of the refrigerants considered apart from R717 and R600a from the table 2.3. Data shown in table 3.6, was varied for each of the refrigerants due to changes in their thermodynamic properties. Errors in the constraints were analyzed during the optimization and sensitivity analysis.

The results of the comparison of evaporator duty, work done, the pressure ratio of the compressor, the mass flow rate of the refrigerant, COP of the system, and second law efficiency for all refrigerants are shown in figure 4.1, 4.2, 4.3, 4.4, 4.5 and 4.6 respectively.

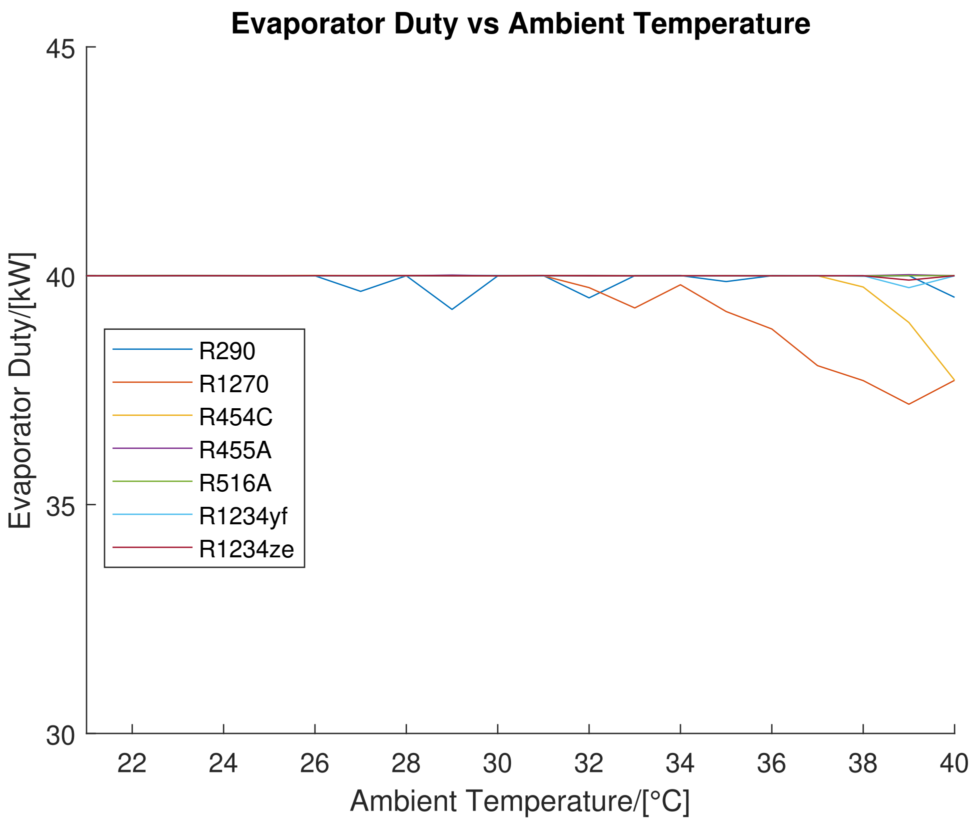

Figure 4.1: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Evaporator duty (the quantity Evaporator duty divided by the unit kW is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of evaporator duty when ambient temperature changes along with constant room temperature. The desired value of duty is 40 kW.

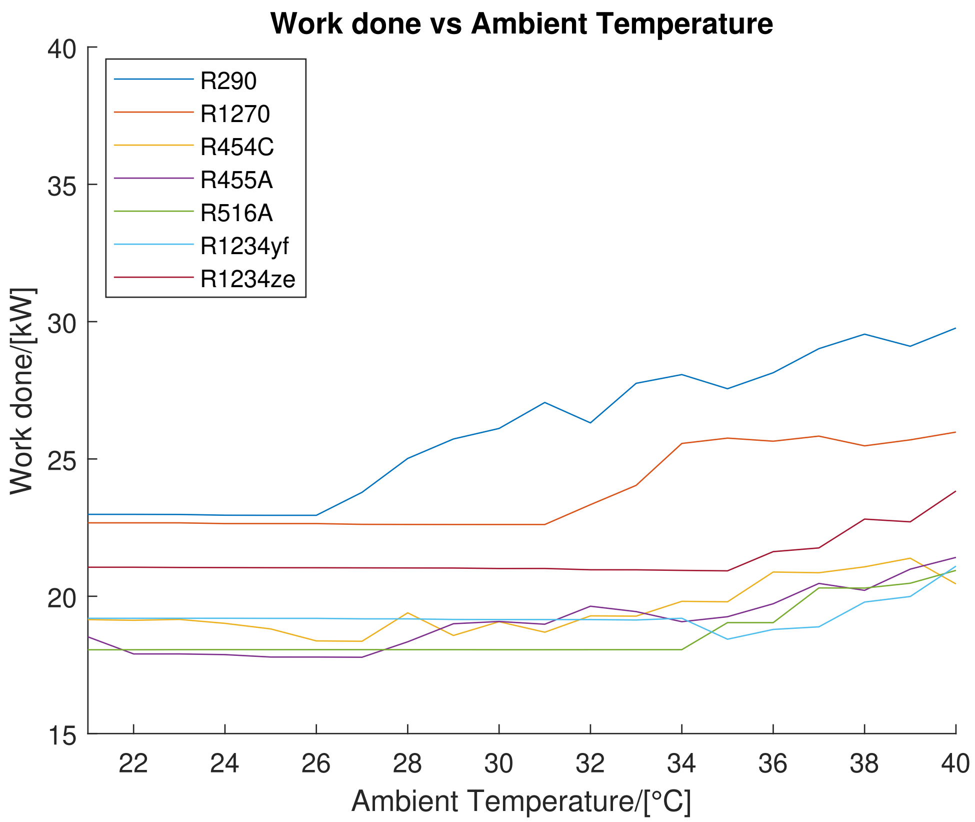

Figure 4.2: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Work done (the quantity Work done divided by the unit kW is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of work done by the compressor when ambient temperature changes along with constant room temperature.

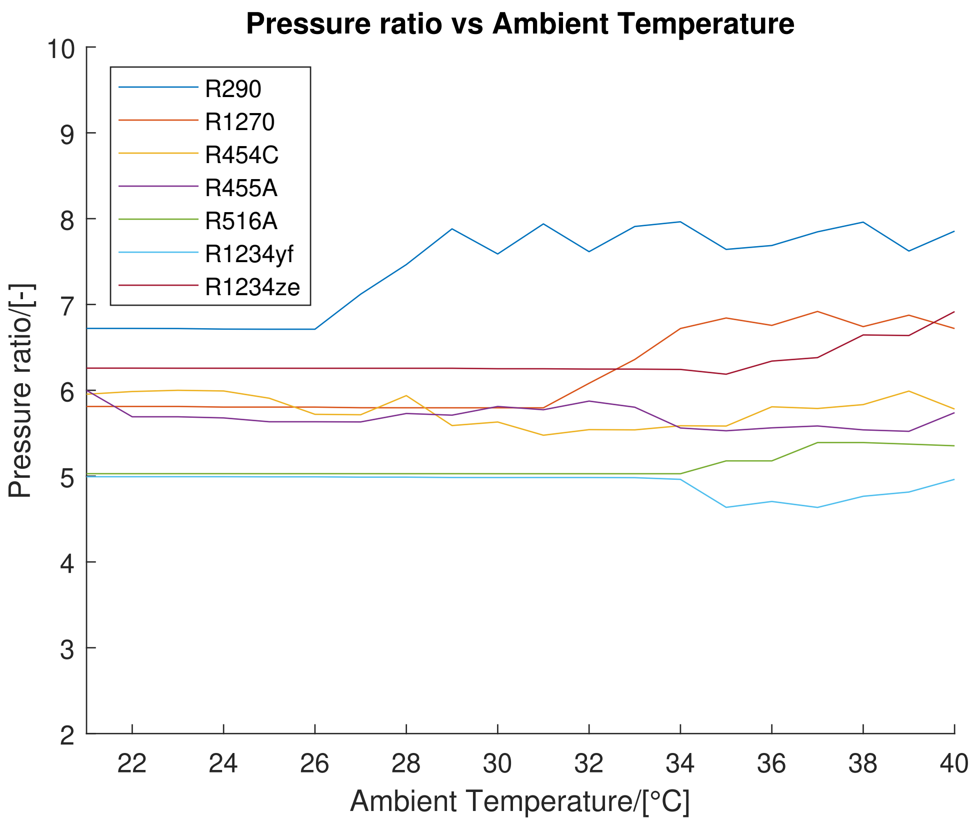

Figure 4.3: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Pressure ratio (the quantity Pressure ratio is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of pressure ratio attained by the compressor when ambient temperature changes along with constant room temperature.

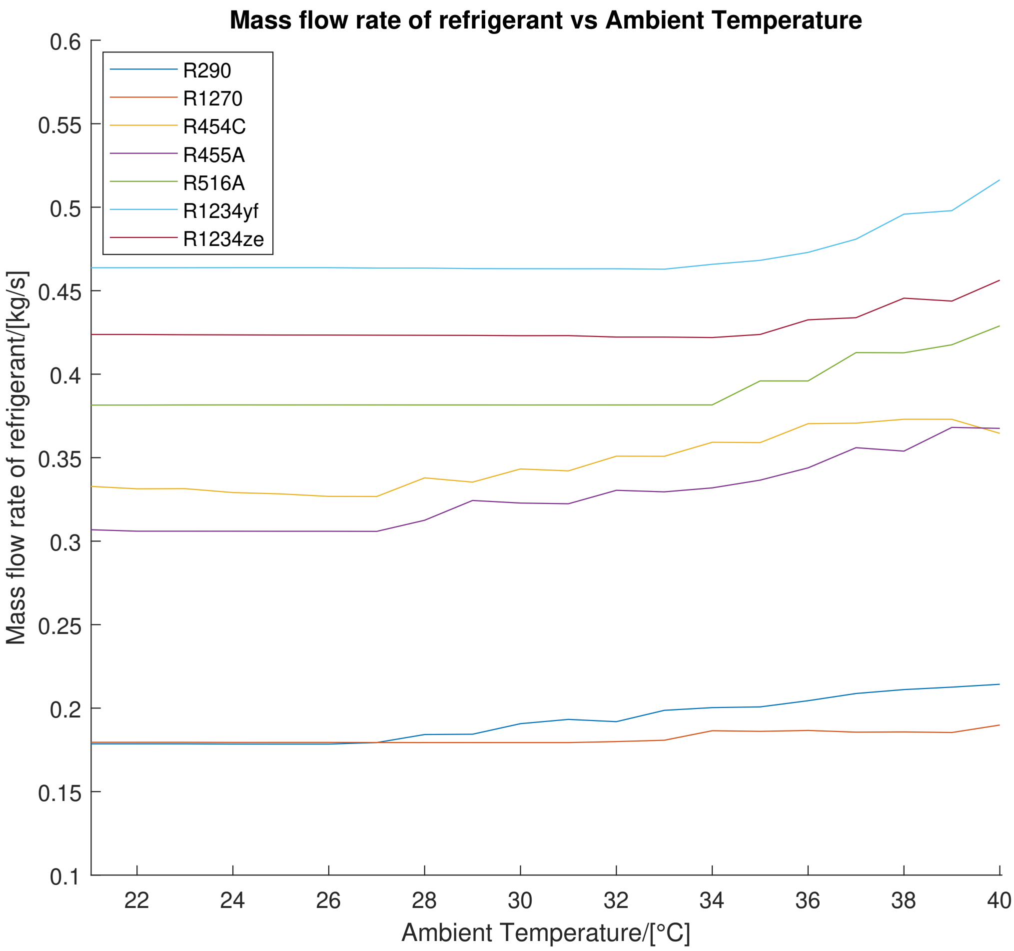

Figure 4.4: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Mass flow rate (the quantity Mass flow rate divided by the unit kg/s is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of the mass flow rate of refrigerant throughout the system when ambient temperature changes along with constant room temperature.

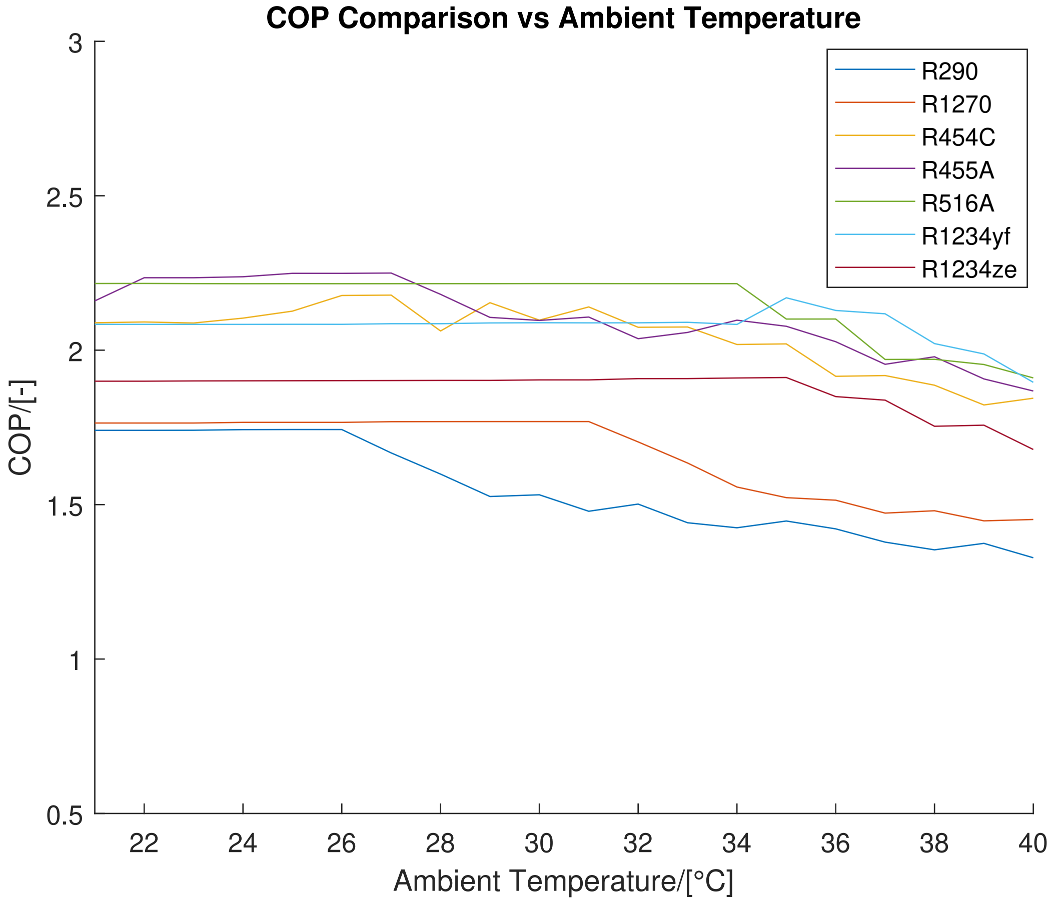

Figure 4.5: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus COP (the quantity COP is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of the Coefficient of Performance of the system given by the specific refrigerant when ambient temperature changes along with constant room temperature.

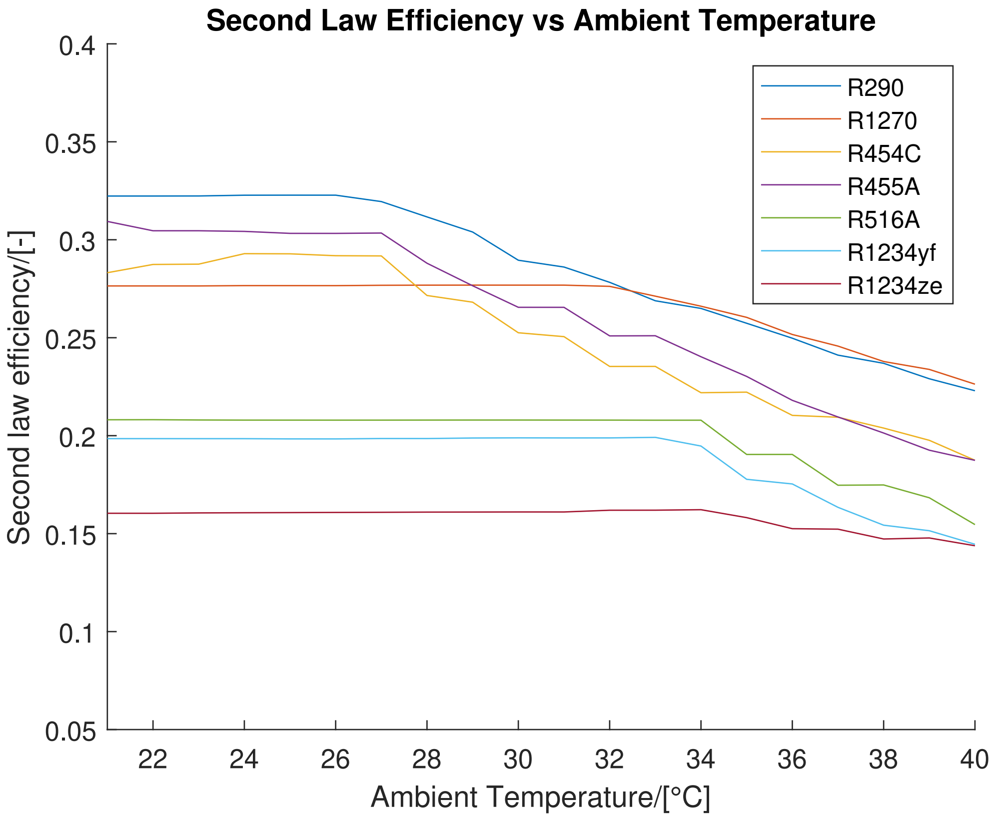

Figure 4.6: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Second law efficiency (the quantity Second law efficiency is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of second law efficiency of the system i.e. COP w.r.t Carnot COP value given by the specific refrigerant when ambient temperature changes along with constant room temperature.

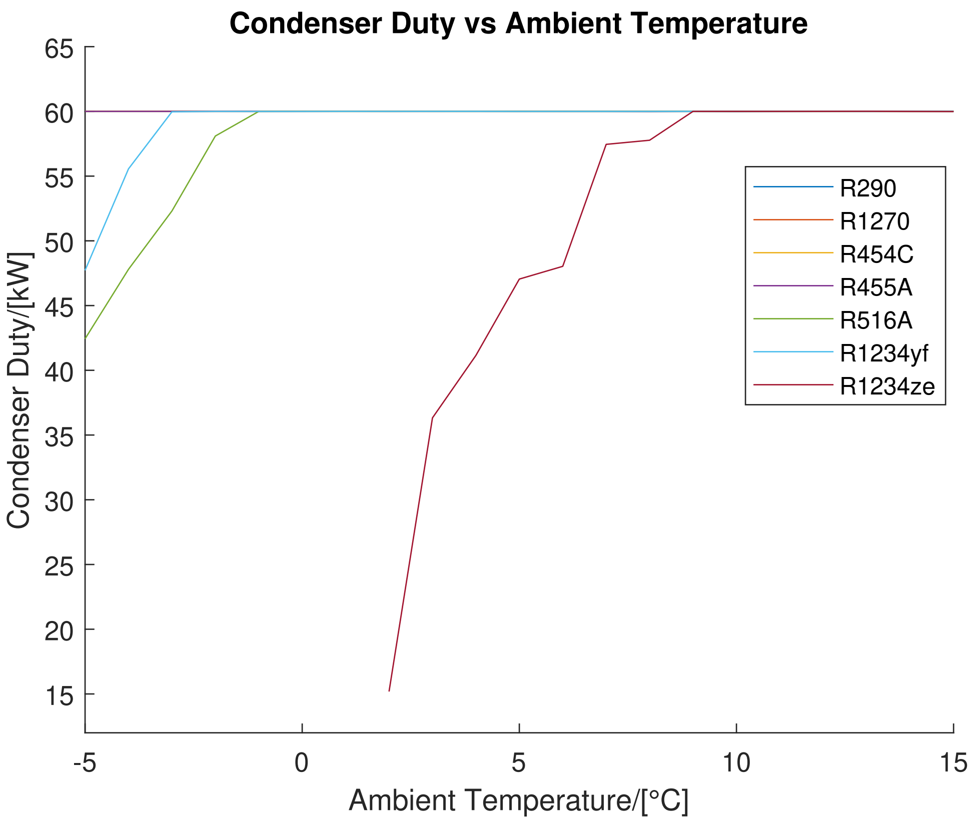

Figure 4.7: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Evaporator duty (the quantity Evaporator duty divided by the unit kW is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of condenser duty when ambient temperature changes along with constant room temperature. The desired value of duty is 40 kW.

4.2. Results for Heating Mode

The methods discussed for modeling, optimization, and sensitivity analysis carried out, and the parameters like evaporator capacity, condenser duty, work done by the compressor, mass flow rate of the refrigerant, pressure ratio in the compressor, isentropic efficiency, second law efficiency was calculated and plotted against the ambient temperature conditions and room temperature conditions varying from 21 to 40°C. The results for which are given below for each of the refrigerants considered apart from R717 and R600a from the table 2.3. Data shown in table 3.6, was varied for each of the refrigerants due to changes in the thermodynamic properties. Error in the constraints was analyzed during the optimization and sensitivity analysis.

The results of the comparison of evaporator duty, work done, the pressure ratio of the compressor, the mass flow rate of the refrigerant, COP of the system, and second law efficiency for all refrigerants are shown in figure 4.7, 4.8, 4.9, 4.10, 4.11 and 4.12 respectively.

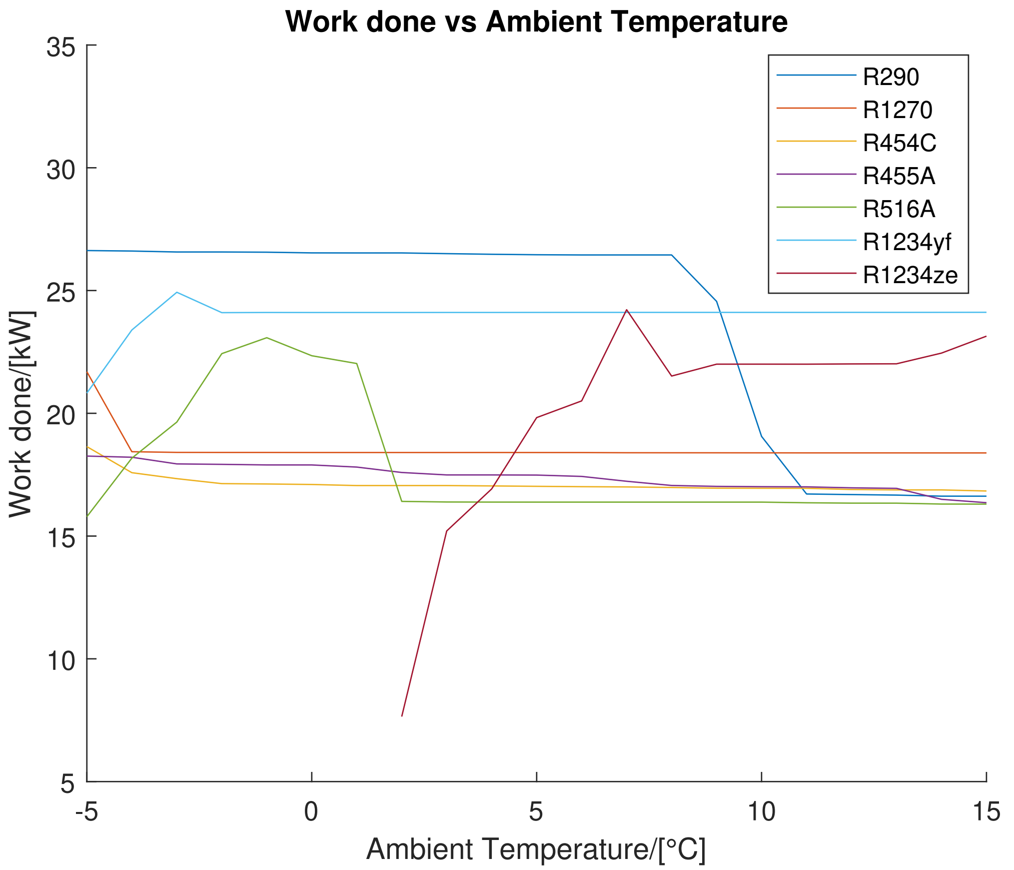

Figure 4.8: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Work done (the quantity Work done divided by the unit kW is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of work done by the compressor when ambient temperature changes along with constant room temperature.

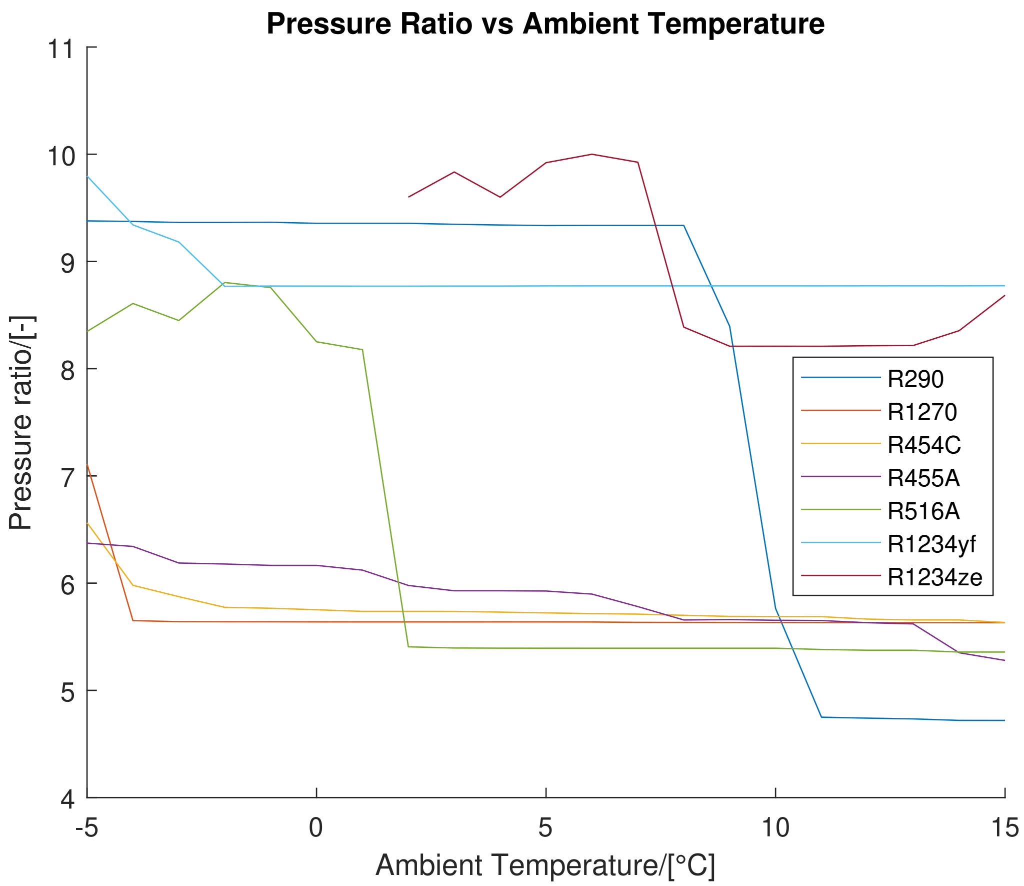

Figure 4.9: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Pressure ratio (the quantity Pressure ratio is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of pressure ratio attained by the compressor when ambient temperature changes along with constant room temperature.

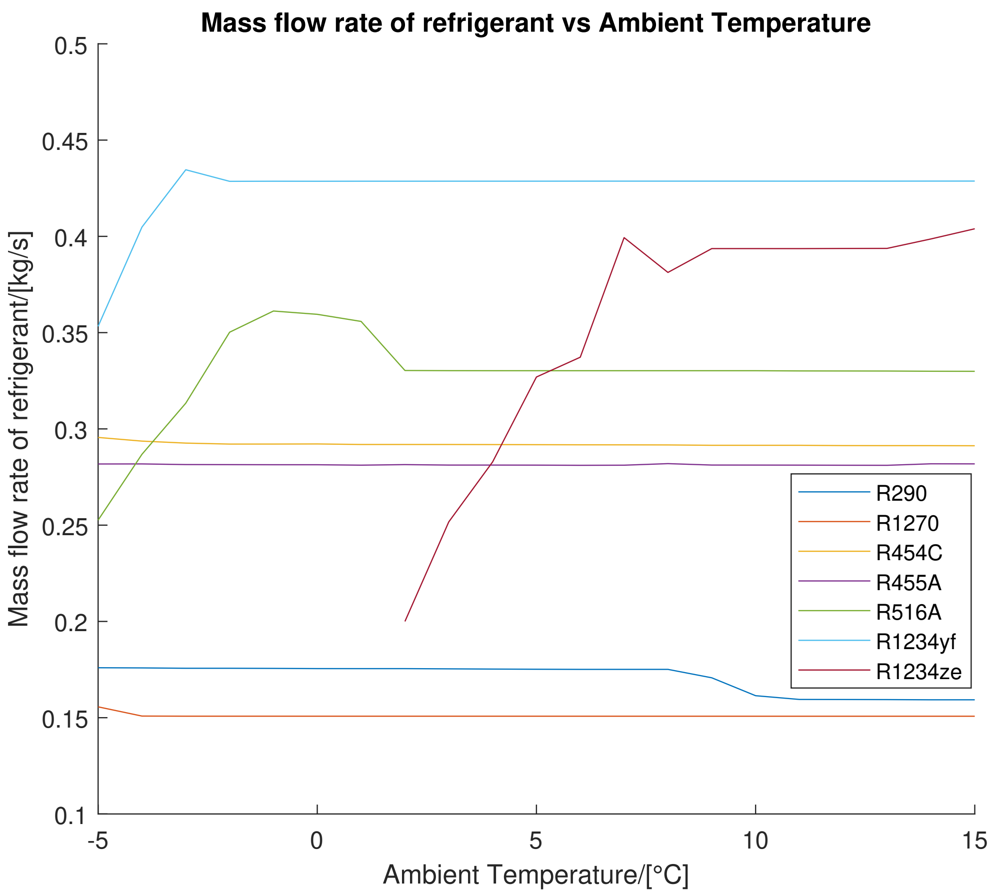

Figure 4.10: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Mass flow rate (the quantity Mass flow rate divided by the unit kg/s is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of the mass flow rate of refrigerant throughout the system when ambient temperature changes along with constant room temperature.

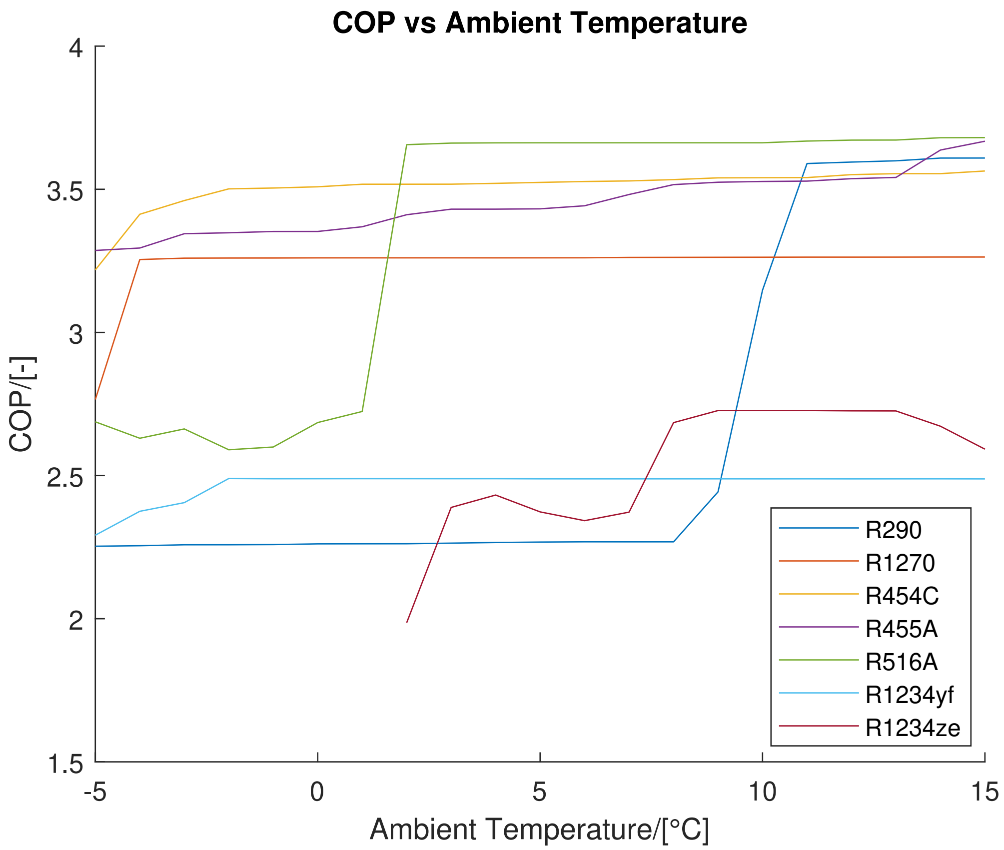

Figure 4.11: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus COP (the quantity COP is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of the Coefficient of Performance of the system given by the specific refrigerant when ambient temperature changes along with constant room temperature.

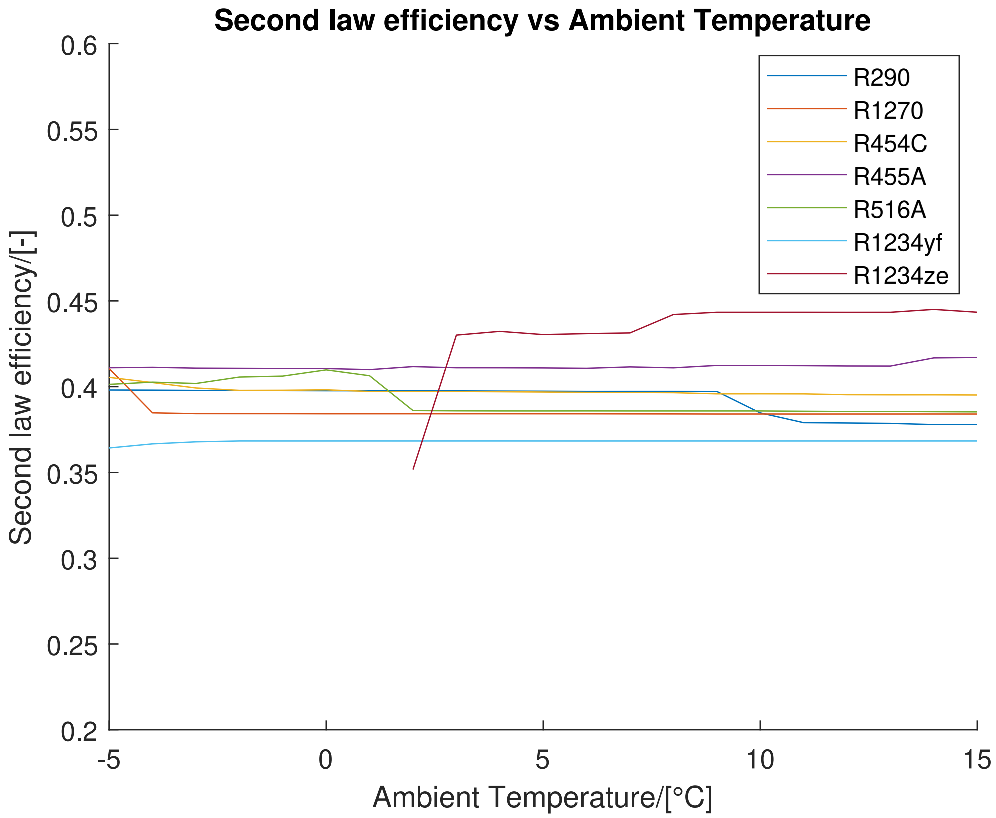

Figure 4.12: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Second law efficiency (the quantity Second law efficiency is the dimensionless number that is listed on the y-axis) curve is shown for refrigerants R290, R1270, R454C, R455A, R516A, R1234yf, and R1234ze. It depicts the fluctuations of second law efficiency of the system i.e. COP w.r.t Carnot COP value given by the specific refrigerant when ambient temperature changes along with constant room temperature.

4.3. Results for Cooling Mode with Pressure Drop

For the modeling of the cooling mode of the system, one of the assumptions was pressure drop across the heat exchanger is zero as given in 3.5.1. But it is not feasible to achieve zero pressure drop across streams in real life, hence validating how pressure drop affects the parameters of the components and influences the performance has to be examined.

So, in this section effect of pressure drop is studied. As per [70], for designing a heat exchanger, a value of 10 psi i.e. around 0.689 bar for the pressure drop is considered initially and after which it is fluctuated as per component specifications, thus, by taking this as a reference, taking the approximate value of 0.5 bar across both the heat exchanger for refrigerant stream only. As, air streams are ambient and room respectively, the drop across it is comparatively lower, hence it is neglected and pressure drop across the refrigerant stream is analyzed.

All conditions discussed in section 3.3 are used for the model along with a pressure drop of 0.5 bar. The graph contains information about fluctuations in evaporator duty, condenser duty, work done by the compressor, coefficient of performance, pressure ratio, isentropic efficiency, the mass flow rate of refrigerant, and second law efficiency, and these parameters are compared for a pressure drop of 0 and 0.5 bar respectively.

The parameters in the legend in figure 4.13 are indicated by the symbols as evaporator duty as Qeva in kW, Work by the compressor as Work in kW, Coefficient Of Performance for cooling as COP, and pressure ratio for the compressor as Pratio. Similarly, in figure 4.14, the legend consists of the mass flow rate of the refrigerant as Mass flow rate in kg/s, Isentropic efficiency, and Second law efficiency as Sec-law efficiency in terms of the magnitude. COP, pressure ratio, isentropic efficiency, and second law efficiency have no units. These legend notations will be used in all of the graphs used ahead in this section.

It can be observed from figure 4.13 & 4.14, that the evaporator capacity is enhanced with the pressure drop, but this led to an increase in the work, due to an increase in the pressure ratio. But for a pressure drop of zero, the evaporator duty is comparatively less similar to the work done by the compressor as the pressure ratio is less. Isentropic efficiency is surely higher in the presence of a pressure drop due to a higher pressure ratio. Higher evaporator duty and work done to demand a higher mass flow rate of refrigerant compared to zero pressure drop. Hence, the coefficient of performance due to pressure drop decreases and also decreases the second law efficiency. Thus it can be concluded that the system has to be designed in order to avoid as many pressure losses as possible to avoid a reduction in the performance of the system. Similarly, the effect of pressure drop on the parameters of the system components for each refrigerant for cooling is shown in figure 4.13 & 4.14 for refrigerant R290, figure 4.15 & 4.16 for R454C, figure 4.17 & 4.18 for R455A, figure 4.19 & 4.20 for R516A, figure 4.21 & 4.22 for R1234yf, figure 4.23 & 4.24 for R1234ze and figure 4.25 & 4.26 for R1270 respectively. These figures depict the difference of change in the parameters for a pressure drop of 0 and 0.5 bar.

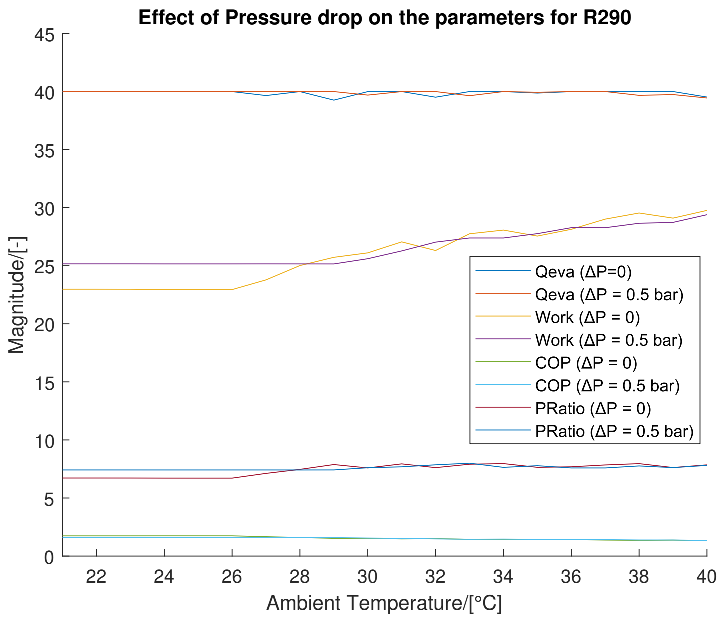

Figure 4.13: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R290. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

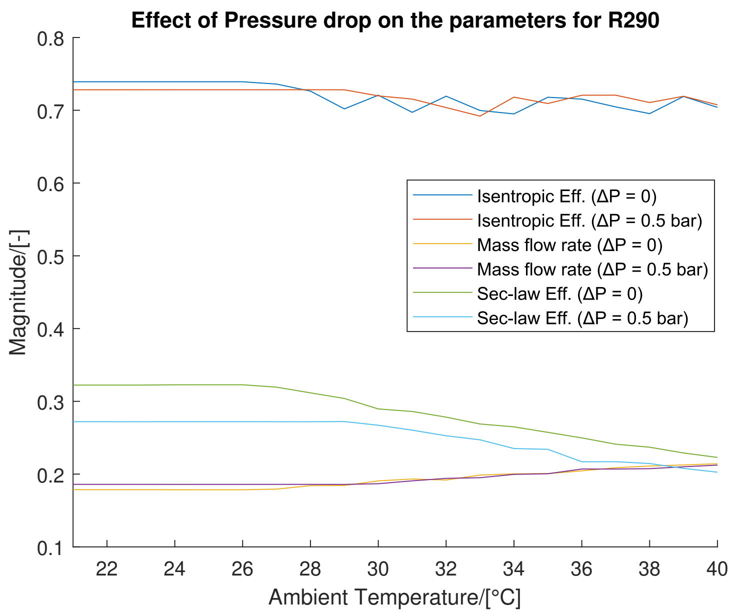

Figure 4.14: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R290. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

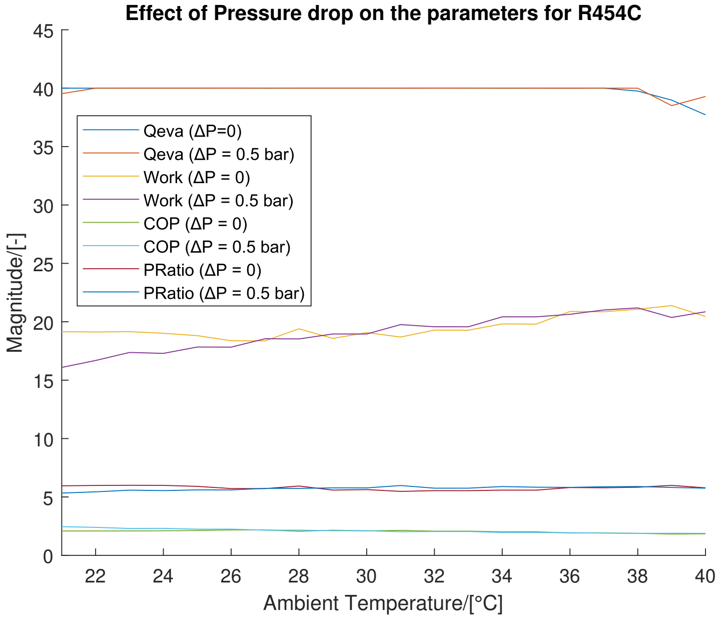

Figure 4.15: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R454C. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

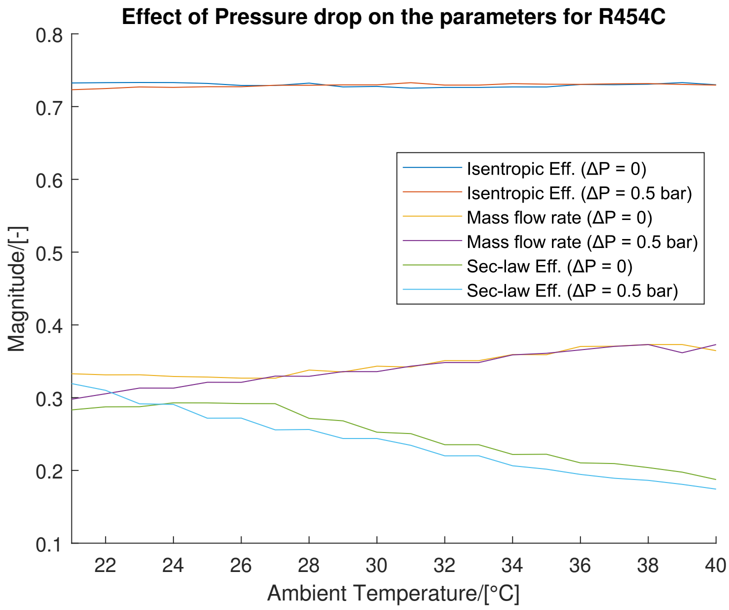

Figure 4.16: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R454C. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

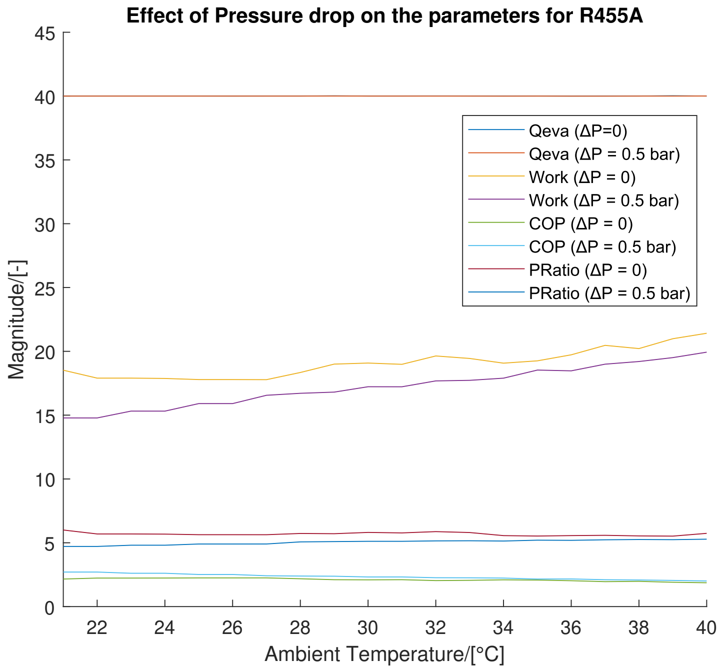

Figure 4.17: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R455A. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

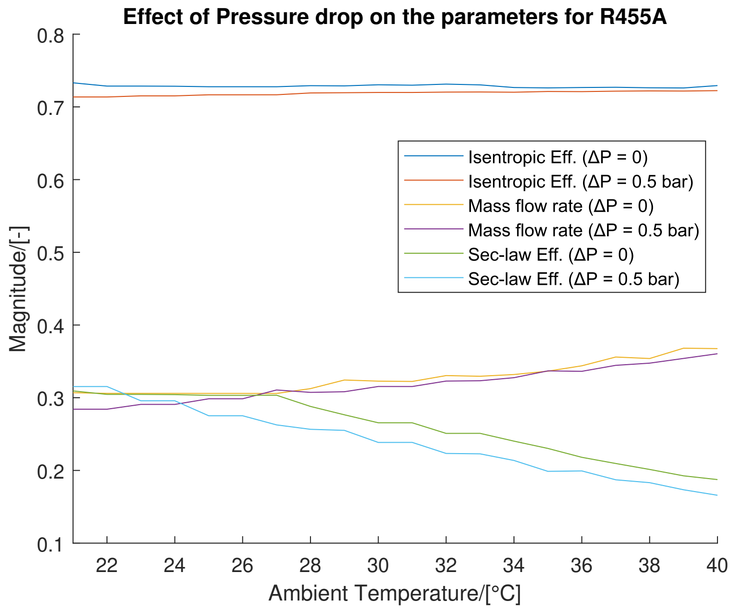

Figure 4.18: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R455A. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

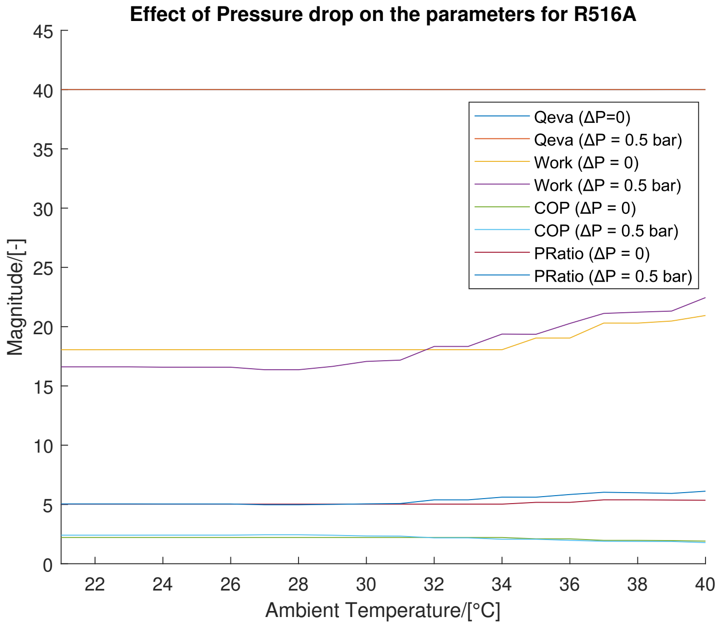

Figure 4.19: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R516A. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

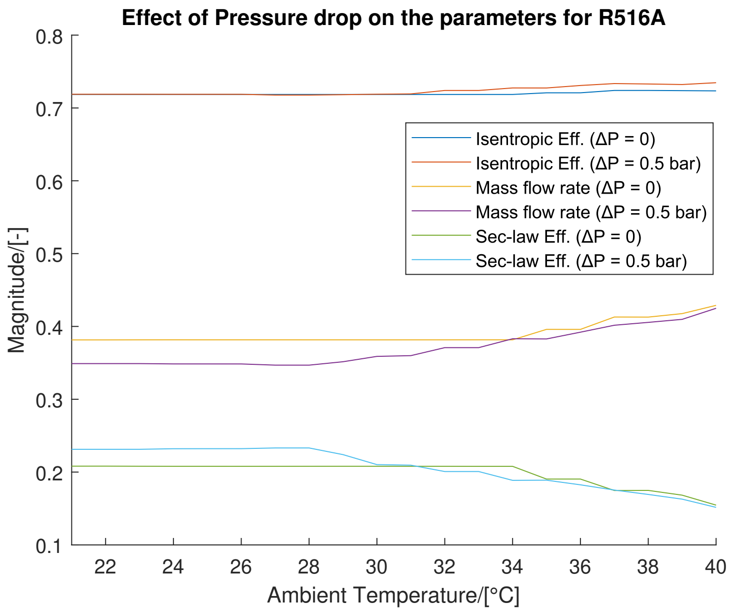

Figure 4.20: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R516A. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

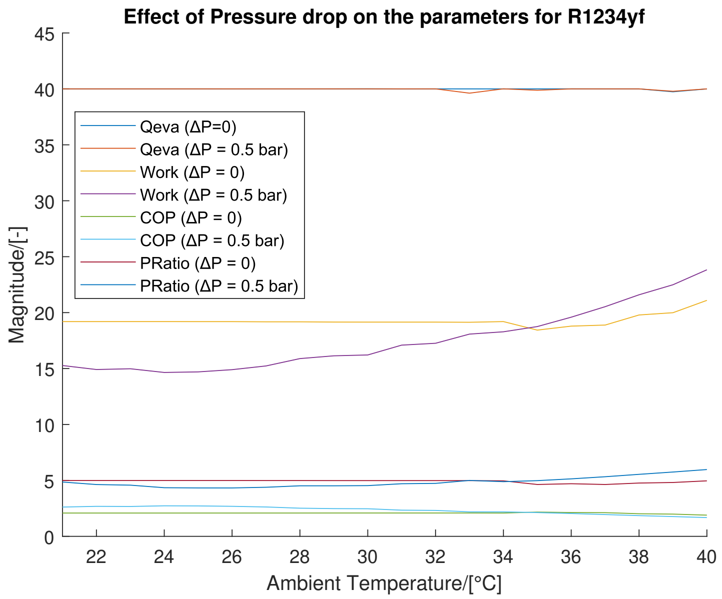

Figure 4.21: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234yf. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

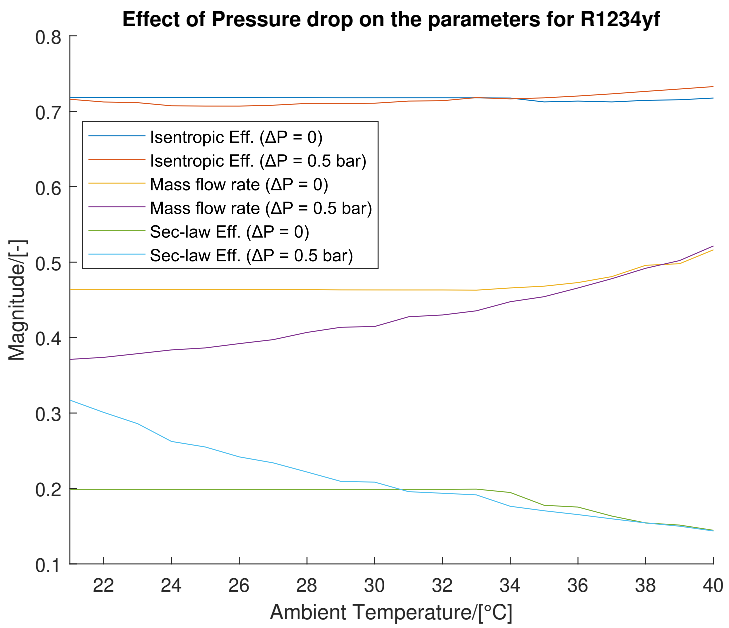

Figure 4.22: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234yf. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

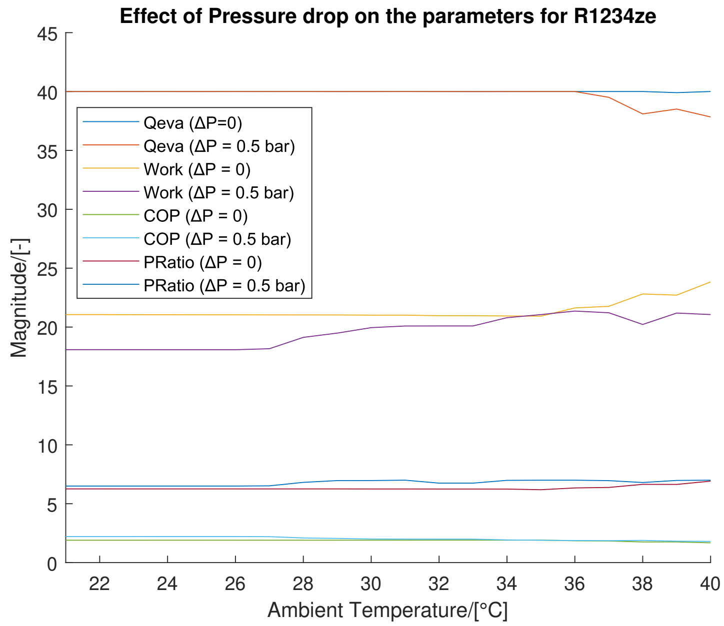

Figure 4.23: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234ze. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

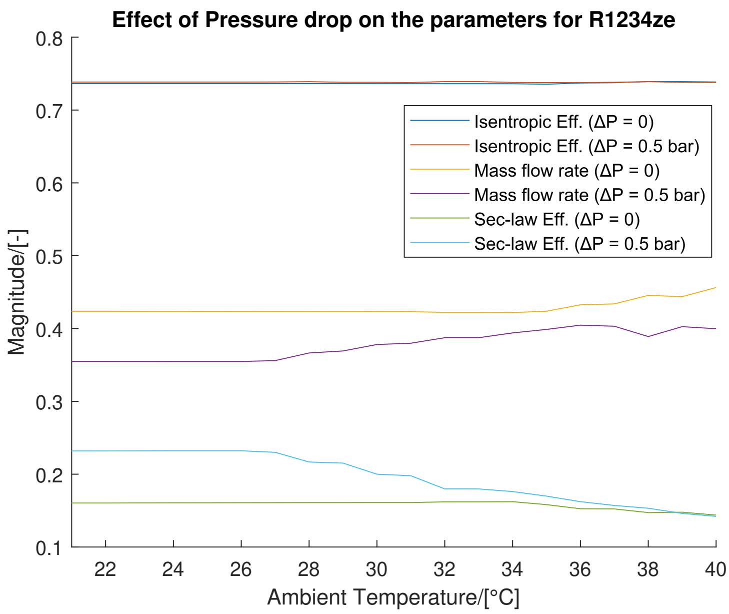

Figure 4.24: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234ze. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

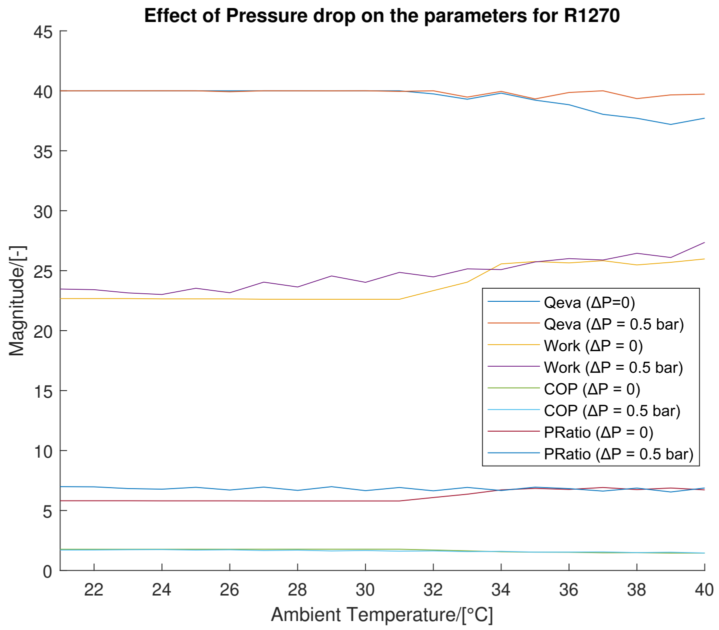

Figure 4.25: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1270. It depicts the fluctuations of evaporator duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

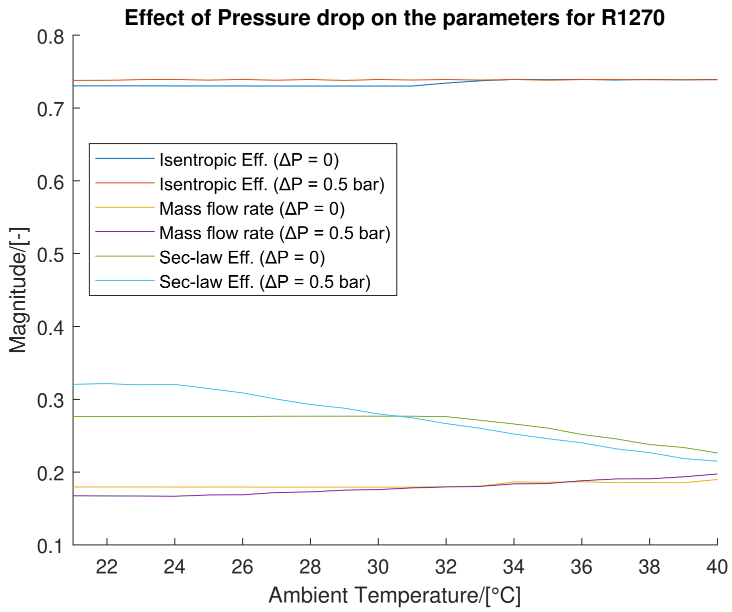

Figure 4.26: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1270. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

4.4. Results for Heating Mode with Pressure Drop

For the modeling of the heating mode of the system, one of the assumptions was pressure drop across the heat exchanger is zero as given in 3.5.1. But it is not feasible to achieve zero pressure drop across streams in real life, hence validating how pressure drop affects the parameters of the components and influences the performance is crucial.

So, in this section effect of pressure drop is studied. As per [70], for designing a heat exchanger, a value of 10 psi i.e. around 0.689 bar for the pressure drop is considered initially and after which it is fluctuated as per component specifications, thus, by taking this as a reference, taking the approximate value of 0.5 bar across both the heat exchanger for refrigerant stream only. As, air streams are ambient and room respectively, the drop across it is comparatively lower, hence it is neglected and pressure drop across the refrigerant stream is analyzed.

All conditions discussed in section 2.5 are used for the model along with a pressure drop of 0.5 bar. The graph contains information about fluctuations in evaporator duty, condenser duty, work done by the compressor, coefficient of performance, pressure ratio, isentropic efficiency, the mass flow rate of refrigerant, and second law efficiency, and these parameters are compared for a pressure drop of 0 and 0.5 bar respectively.

The parameters in the legend in figure 4.27 are indicated by the symbols as condenser duty as Qcond in kW, Work by the compressor as Work in kW, Coefficient Of Performance for cooling as COP, and pressure ratio for the compressor as Pratio. Similarly, in figure 4.28, the legend consists of the mass flow rate of the refrigerant as Mass flow rate in kg/s, Isentropic efficiency, and Second law efficiency as Sec-law efficiency in terms of the magnitude. COP, pressure ratio, isentropic efficiency, and second law efficiency have no units. These legend notations will be used in all of the graphs used ahead in this section.

It can be observed from figure 4.27 & 4.28, that the evaporator capacity is enhanced with the pressure drop, but this led to an increase in the work, due to an increase in the pressure ratio. But for a pressure drop of zero, the evaporator duty is comparatively less similar to the work done by the compressor as the pressure ratio is less. Isentropic efficiency is surely higher in the presence of a pressure drop due to the higher pressure ratio. Higher evaporator duty and work done to demand a higher mass flow rate of refrigerant compared to zero pressure drop. Hence, the coefficient of performance due to pressure drop decreases and also decreases the second law efficiency. Thus it can be concluded that the system has to be designed in order to avoid as many pressure losses as possible to avoid a reduction in the performance of the system. Similarly, the effect of pressure drop on the parameters of the system components for each refrigerant for cooling is shown in figure 4.27 & 4.28 for refrigerant R290, figure 4.29 & 4.30 for R454C, figure 4.31 & 4.32 for R455A, figure 4.33 & 4.34 for R516A, figure 4.35 & 4.36 for R1234yf, figure 4.37 & 4.38 for R1234ze and figure 4.39 & 4.40 for R1270 respectively. These figures depict the difference of change in the parameters for a pressure drop of 0 and 0.5 bar.

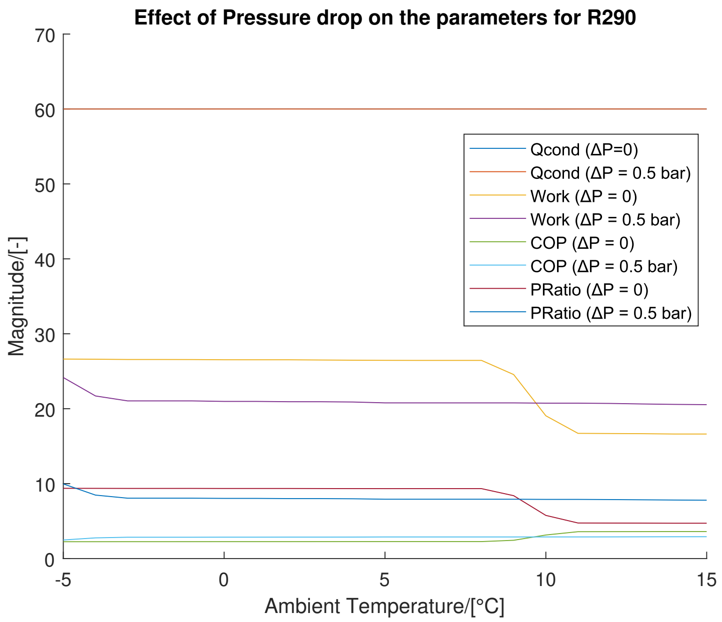

Figure 4.27: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R290. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

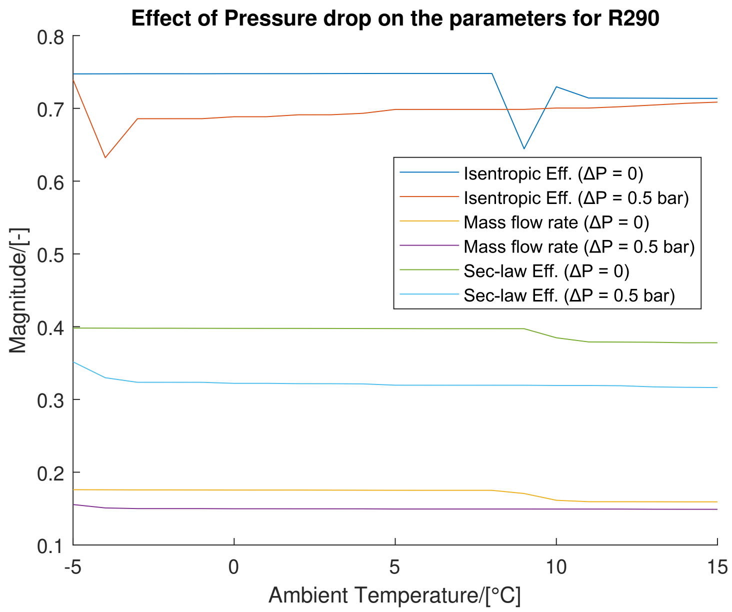

Figure 4.28: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R290. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

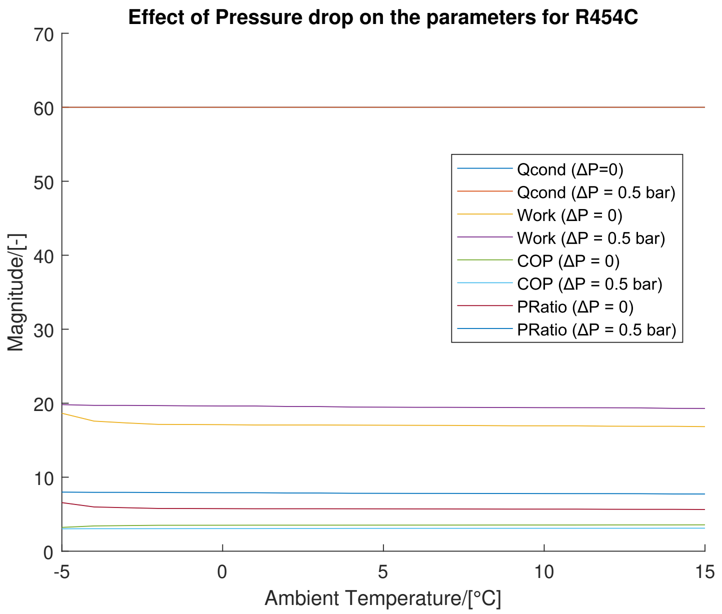

Figure 4.29: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R454C. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

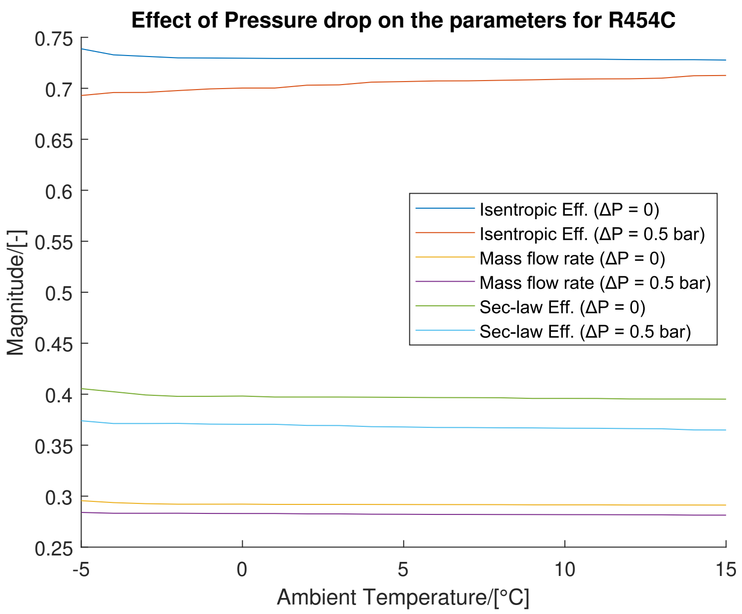

Figure 4.30: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R454C. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

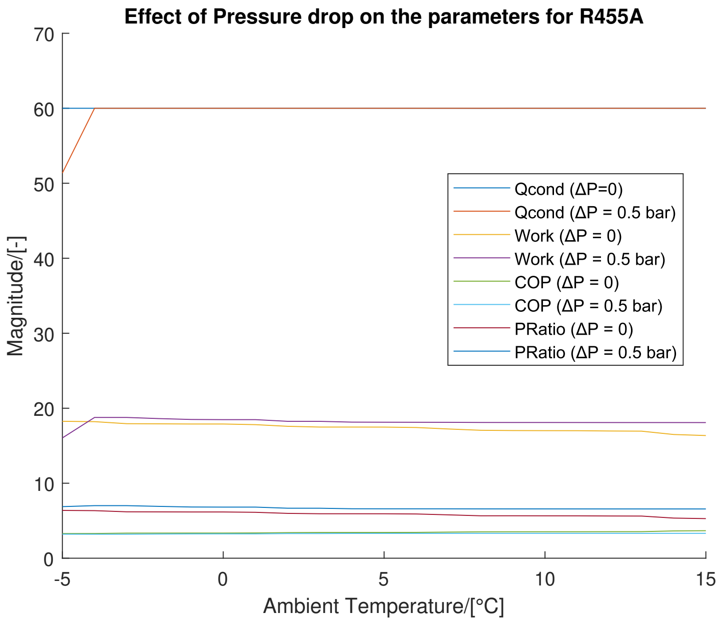

Figure 4.31: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R455A. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

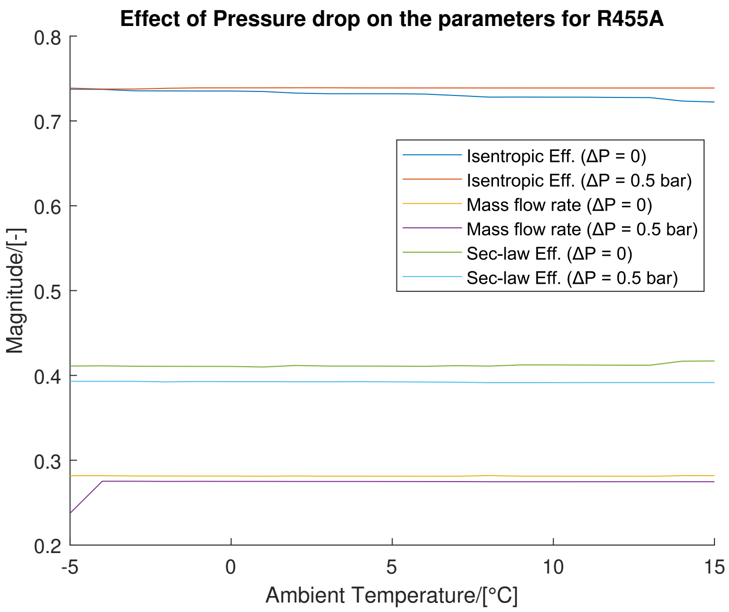

Figure 4.32: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R455A. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant and Second law efficiency when ambient temperature changes along with constant room temperature when pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

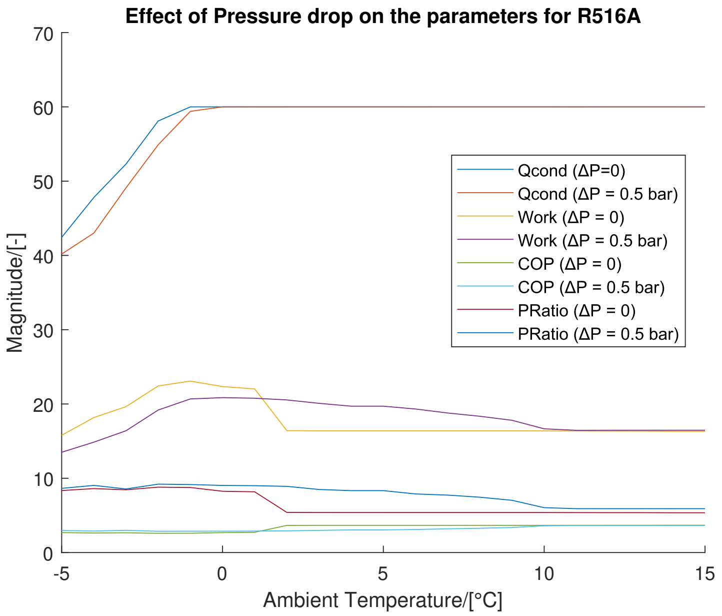

Figure 4.33: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R516A. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

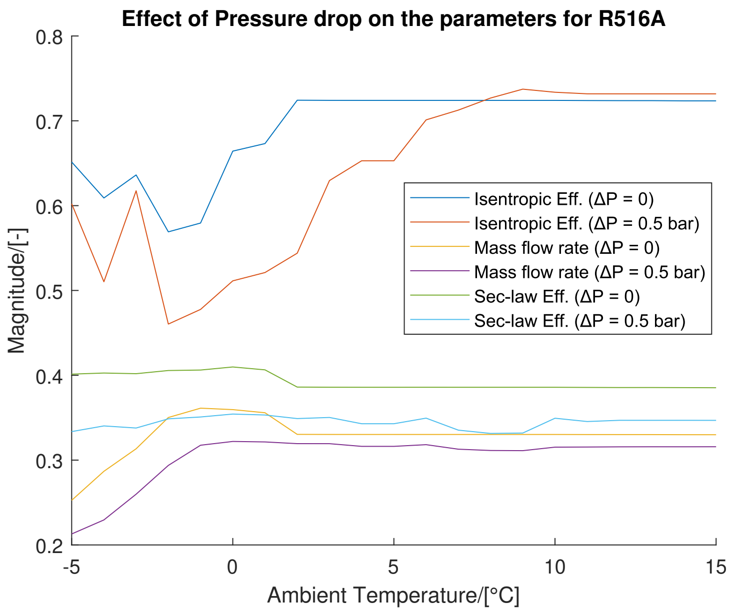

Figure 4.34: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R516A. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

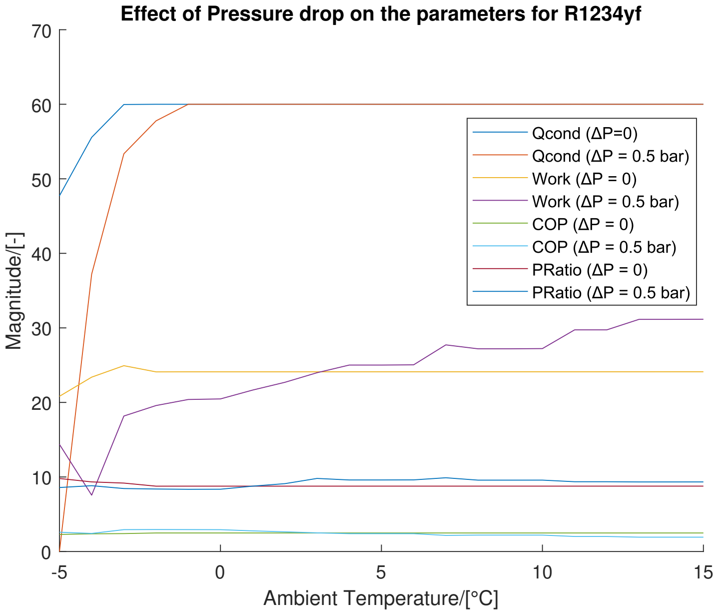

Figure 4.35: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234yf. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

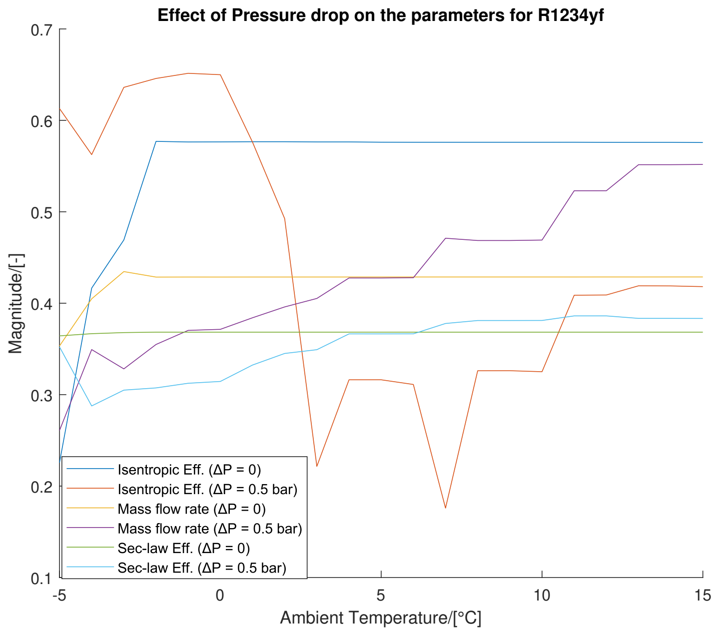

Figure 4.36: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234yf. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

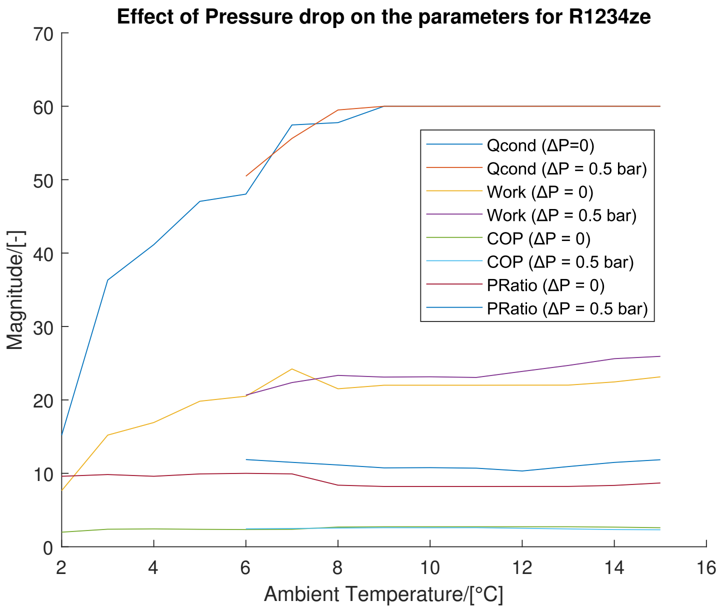

Figure 4.37: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234ze. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

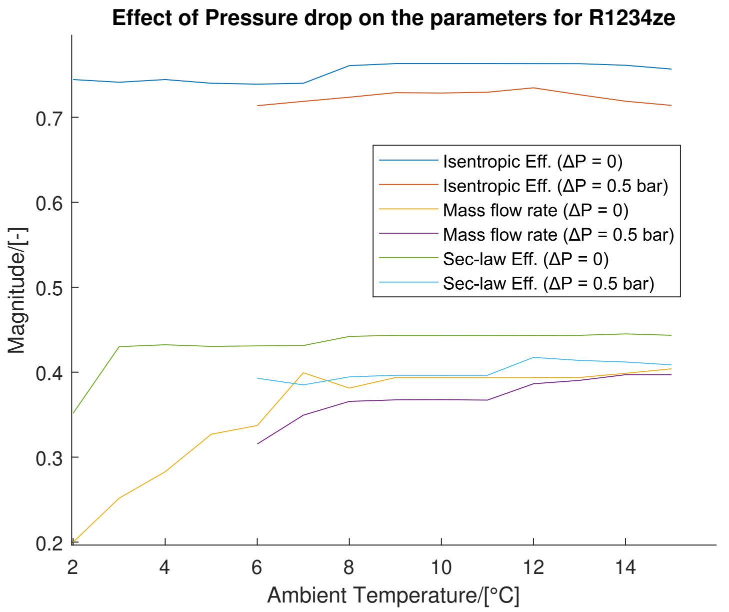

Figure 4.38: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1234ze. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

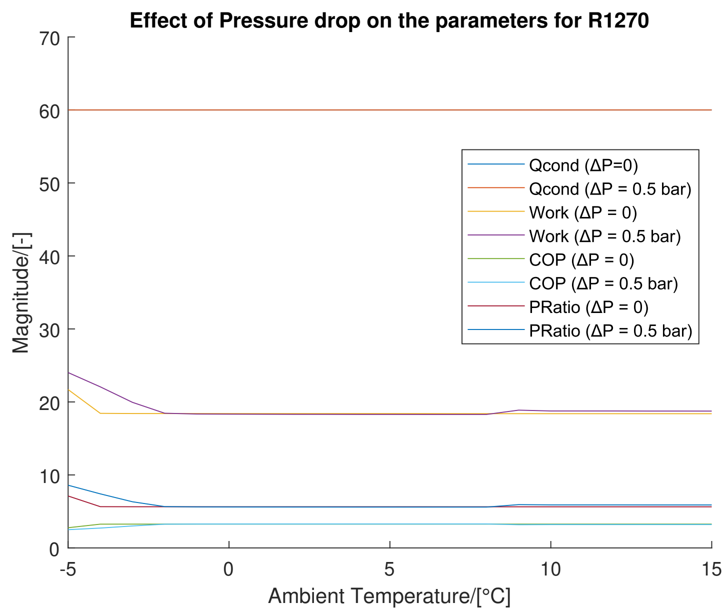

Figure 4.39: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1270. It depicts the fluctuations of condenser duty, work done by the compressor, COP of the system, and Pressure ratio when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

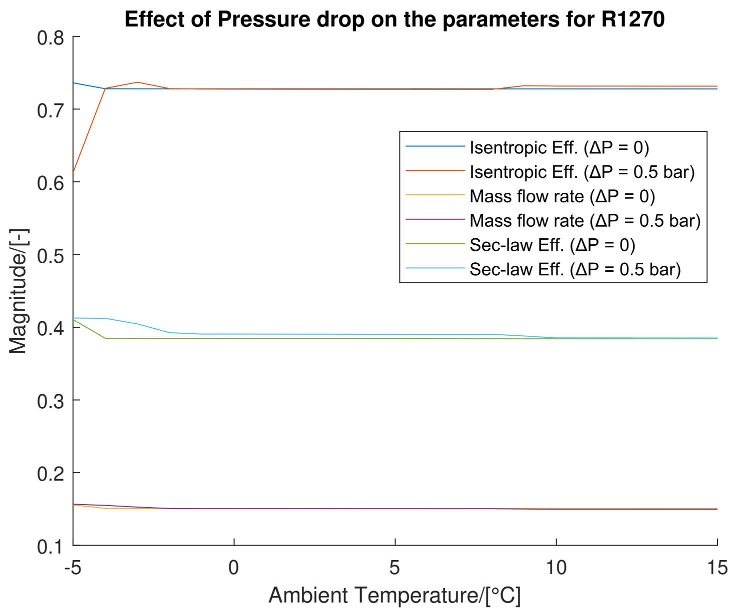

Figure 4.40: Ambient Temperature (the quantity Temperature divided by the unit °C is the dimensionless number that is listed on the x-axis) versus Magnitude (the quantity Magnitude is the dimensionless number that is listed on the y-axis indicating the magnitude of different parameters) curve is shown for refrigerant R1270. It depicts the fluctuations of Isentropic efficiency, Mass flow rate of the refrigerant, and Second law efficiency when ambient temperature changes along with constant room temperature when the pressure drop across the heat exchangers are 0 and 0.5 bar respectively.

5 Conclusion and Recommendations

This project focuses on the feasibility of low GWP refrigerants as compared to R410A (high GWP) in order to develop a heat pump system with the least environmental consequences. Chapter 1 focused on the overview of heating and cooling and how it is linked to climate change. Chapter 2 focused on the heating and cooling systems, their evolution, and the thermodynamics related to them. Chapter 3 focused on the modeling of the system in Aspen, with all the conditions, assumptions, and constraints. Chapter 4 focused on the results obtained after simulating the model created in Aspen, the remarks on each refrigerant are given below.

5.1. Conclusion

As discussed in section 4, the behavior of all the refrigerants was observed under given conditions and constraints. Looking at each refrigerant behavior individually, we can conclude as follows:

- R1234ze: This refrigerant possesses zero ODP and almost zero GWP too, which surely makes it an environmentally friendly refrigerant. It has a higher critical point but the boiling point is comparatively high making it difficult for the refrigerant to cover a longer range of operational temperatures. Looking at its performance for cooling, it did satisfy the evaporator duty condition of 40 kW when the pressure drop was zero but as pressure drop was considered after 36°C, evaporator duty dropped till approximately 37 kW. The pressure ratio needed for the operation was also optimal i.e. around 6-7. The mass flow rate required to attain the duty was high and COP of a maximum of 1.9 and second law efficiency of 0.2 for no pressure drop was attained. Similarly, for heating mode, without pressure drop, and with pressure drop, the bivalent temperature (the temperature below which it can not attain the expected duty with just heat pump system) of 2°C and 6°C respectively was obtained. This temperature can also be TOL (Operation Limit Temperature, below which it stops converging for given conditions). These values are high and do not satisfy the objective of the system. So, out of all the considered refrigerants, R1234ze did not perform as expected and is not a fit for this kind of system.

- R516A: This refrigerant is an azeotropic ternary mixture of R1234yf, R134a, and R152a with no glide present due to the formation of an azeotrope. It has zero ODP with a GWP of 131, a higher critical point, and a moderate boiling point. In terms of cooling, it did satisfy the 40 kW evaporator duty condition for all ambient temperatures, the second lowest pressure ratio of around 5-5.5, with a maximum mass flow rate of 0.5 kg/s and one of the highest COP of 2.25 but second law efficiency of around just 0.22, which can be modified by further optimizing the system. A pressure drop of 0.5 bar did not significantly affect all these parameters, which means this can sustain well even under high-pressure drop conditions. If heating mode is observed, for zero pressure drop, below 0°C, it started dropping the condenser duty and decreased it further till -5°C. Especially, for pressure drop, condenser duty decreased more, decreasing the COP of the system. Due to its good performance in cooling, it can be a good fit for at least just cooling purposes, but for heating, reanalysis and optimization are required. Also, this refrigerant is not readily available in the market, so can not be considered for this system at this moment.

- R1234yf: This refrigerant is a single component, with low density, zero ODP, and almost zero GWP, and has other optimal thermodynamic properties. A better boiling point and critical point are observed with mild flammability. Analyzing the refrigerant behavior for cooling purposes, the evaporator duty of 40 kW was thoroughly satisfied even with the presence of a pressure drop. COP was moderate but second-law efficiency was comparatively low. Due to lower density, a high mass flow rate is needed to achieve the desired effect, which can be tricky to control. Pressure drop did not affect the parameters on a higher level, a small offset was observed in the values. But in terms of heating, condenser duty was not satisfied for all the temperatures, i.e. with zero pressure drop, duty dropped to around 47 kW at -5°C and with pressure drop, it reached 0 kW at -5°C. Even with this, it showed second law efficiency of around 0.4 and COP of around 3-3.5 with and without pressure drop. This refrigerant can be suitable for the system for cooling purposes, but for heating purposes, the operation temperature limit will be around just 0°C for optimum performance.

- R290: Commonly known as Propane, is a single component refrigerant, with ODP of zero and GWP of just 3, optimal critical and boiling point but safety group class being A3 i.e. highly flammable, which is difficult to handle. Due to higher density, a lower mass flow rate is required but if the mass flow rate is higher, there is a possibility of a leak which can lead to hazardous situations. Looking at the behavior under the cooling mode, unstable evaporator duty fluctuates between 37-40 kW. The highest pressure ratio is required, and in some cases, multistage compression might need to be used. COP value is the lowest amongst all but the highest second law efficiency. Similarly, for heating mode, condenser duty of 60 kW is achieved for all the ambient temperature conditions, with a higher pressure ratio of around 9.5 at lower temperatures, but COP and second law efficiency are moderate. Pressure drop impacts are negligible but flammability is the risk factor along with unstable output, hence, for heating, it can be a good alternative but for a reversible system, it is not a good fit.

- R454C: This refrigerant is a binary zeotropic mixture of R32 and R1234yf with a nominal glide of 7.81°C but has zero ODP and GWP of 146 which is still below 150. It has a low and optimal boiling point and optimal critical point too. Mild flammability makes it comparatively safe to use in a system. Observing the graphs, for heating purposes, satisfies all the conditions perfectly, with good COP and second-law efficiency. The effect of pressure drop is almost negligible in most of the parameters. But in terms of cooling, after 37°C, it did not satisfy the 40 kW evaporator duty constraint due to which it affected the COP and second law efficiency. Apart from this 37°C, this refrigerant behavior was optimal for heating but for cooling, reanalysis is essential. R454C can be a potential fit for the system.

- R1270: Commonly known as Propylene, is a single-component refrigerant, with ODP of zero and GWP of just 2, optimal critical and boiling point but safety group class being A3 i.e. highly flammable, which is difficult to handle. Looking at the behavior under the desired effect of cooling, evaporator duty decreases up to 37 kW at higher temperatures of around 36-40°C. The lowest mass flow rate among all the refrigerants is required to attain the evaporator duty, poor COP but moderate second law efficiency is observed. Again due to safety class, it is tough to handle this refrigerant. For heating mode, it satisfies the condenser duty condition for all the temperature values with full convergence. Good COP and second law efficiency of around 0.4 throughout are observed. The effect of pressure drop is minimal and negligible. Due to this nature, for heating, this is the perfect refrigerant but for cooling, the system has to be designed to undertake safety aspects too. This is a potential fit for this kind of mobile system.

- R455A: This refrigerant is a ternary zeotropic mixture of R32, R1234yf, and R744 with a nominal glide of 12.9°C but has zero ODP and GWP of 146 which is still below 150. It has a low and optimal boiling point and optimal critical point too. Mild flammability makes it comparatively safe to use in a system. Due to the presence of 3 components, has a higher value of glide. Optimal density and mild flammable nature make it easier to handle. Observing the cooling mode behavior of R455A, the evaporator duty of 40 kW was satisfied in the considered ambient temperature range, moderate pressure ratio, mass flow rate, highest COP value at some temperatures, and one of the highest second law efficiency too. The effect of pressure drop is minimalistic. Considering the heating mode, the condenser duty of 60 kW was satisfied in the considered ambient temperature range, lower pressure ratio, moderate mass flow rate, one of the best COP values, and the best second law efficiency. The effect of pressure drop in heating is also minimalistic. Just apart from a glide, the rest of the thermodynamic properties are optimal, and is the most suitable refrigerant among all considered refrigerants for this system.

5.2. Recommendations for Future Work

Following are the recommendations to approach this project differently and develop it further:

1. As this project was carried out using the software Aspen plus, so to validate these results, an alternative software or a programming language can be used to assess all of these refrigerant properties and compare the results.

2. Apart from this, the same work can be done by using different calculation methods like NRTL, SRK, etc. instead of Peng-Robinson, to crosscheck which is the best method to do the heat pump calculations and their feasibility.

3. Parameters assumed in this project can be changed or all the constants can be kept as a variable to check how refrigerant behavior changes.

4. Dynamic analysis can be done for any of the potential refrigerants to validate the reliability of that refrigerant.

5. Practical cycle analysis can be done instead of the ideal cycle which was considered in this project. The real cycle consists of pressure drops, superheating, and subcooling too.

6. Component modeling can be done by considering the basis of this analysis to further develop the prototype of the system.

7. Compare this data by analyzing the data of behavior of all the refrigerants with GWP of more than 150 to validate the low GWP refrigerant significance and feasibility.

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