POLITECNICO DI TORINO

International Master's Degree Course in Mechanical Engineering

Master's Degree Thesis

Design of A Single-Stage Ammonia-Water Vapour Absorption Chiller for Industrial Vehicles

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Abstract

The study examines a vapour absorption cycle with an ammonia-water solution through a thermodynamic analysis of the system using waste heat from truck diesel engines. An energy balance of a diesel engine shows that sufficient waste heat can be utilized to run an absorption refrigeration cycle. At the given parameters of the diesel engine and ambient, the obtained coefficient of performance is 0.3556 and the circulation ratio is 2.552. These results are manipulated using a range of operating conditions which maximizes the performance of the system. The results demonstrate that higher performance of the system is obtained at the high generator and low absorber temperature or vice versa to obtain the cabin temperature equals 20 ºC which is the prime objective.

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List of Abbreviations and Symbols

Abbreviations

CFC	Chlorofluorocarbon
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbon
GWP	Global Warming Potential
ODP	Ozone Depletion Potential
IGWP	Indirect Global Warming Potential
TEWI	Total Equivalent Warming Impact
GAX	Generator Absorber Heat Exchange Absorption
IIR	International Institute of Refrigeration
EC	European Commission
ITMC	Integrated Thermal Management Controller
VCHP	Variable Conductance Heat Pipe
CCHP	Combined Cooling, Heating and Power
VCC	Vapour Compression Cooling
LPG	Liquefied Petroleum Gas
PER	Primary Energy Ratio
EES	Engineering Equation Solver
COP	Coefficient of Performance

Symbols

T	Temperature (^{O}C)
P	Pressure (bar)
X	Ammonia Mass fraction
$\dot{Q}$	Heat Transfer Rate (W)
h	Enthalpy (KJ/Kg)
u	Internal Energy (KJ/Kg)
s	Entropy (KJ/kg)

v	Specific Volume (m^3/kg)
q	Quality
W	Work input (W)
$\dot{m}$	Mass flow rate (kg/s)
f	Solution Circulation Ratio
ρ	Density (Kg/m^3)
η	Efficiency

Subscripts

evap	Evaporator
gen	Generator
abs	Absorber
rect	Rectifier
cond	Condenser
ABS,REF	Absorption Refrigeration
P	Pump
C	Cabin
rs	Rich Side
ps	Poor Side

CHAPTER-1

Introduction

1.1 Background

The current situation in the world demands sustainability and less pollution, the possibility of which lies in the optimization of consumption of primary energy resources and diluting the consequences caused by the release of greenhouse gases thereby meeting the standards defined in the Paris agreement 2016. Being a highlighted issue globally, many countries and international institutions are playing their role to achieve the targets in various ways i.e. economic incentives etc. An Increase in raw material prices also calls for addressing the issue of efficient use of energy with immediate importance and seriousness.

Another factor that plays an important role in achieving the efficient use of energy is the design of the equipment. Industries having fairly designed machineries will exploit the use of energy more efficiently than to its contrary.

Since the invention of the steam engine and industrial revolution, there has been research going to increase the amount of heat recovered from the processes (like combustion and waste-gas release from the vehicles) to be utilized in the post-operation stages i.e. in pre-heating of a process or any space. An effective way to use this un-utilized heat is to produce cooling effect from it as there are many refrigeration systems e.g. absorption cooling system which can take advantage of this heat hence saving up the energy wasted otherwise.

Because one of the most used machinery in daily life globally is a vehicle and according to an estimation, almost one-third of the energy produced during the combustion process is wasted from a vehicle in the form of exhaust, there is a need for optimization here. Also, air-conditioning in cars is becoming a vital need. Whether it’s a normal vehicle or an industrial vehicle, there is a requirement for a cooled space. Consequently, these exhaust gases can be an energy source for the absorption cooling systems in vehicles.

Absorption cooing systems come with different types which mainly differs based on the working fluids being used in them. The refrigerants that were used in the 1990s were CFCs and HCFCs but the repercussion caused by CFCs refrigerants (i.e. ozone depletion) made them banned in 1987 by Montreal protocol and with the Kigali amendment, the HCFCs are also declared high GWP (Global Warming Potential) substances. However, the world is now finding its replacements in Hydrofluoroolefins or ‘natural’ refrigerants.

In absorption systems, the most widely used refrigerants are water-lithium bromide (H_2O-LiBr) and ammonia-water (NH_3-H_2O) due to their ozone depletion potential (ODP) equal to zero. The working principle of systems with both of these refrigerants is the same. The difference, however lies in the application. LiBr systems, due to their somewhat higher performance index, are used for air-conditioning purposes but are usually limited to it. Ammonia-water systems are best suited for applications below 5 ^{O}C.

1.2 Objective

The main aim of this thesis is to design a chiller for industrial vehicles. The chiller employs a single-stage vapour absorption cycle by using ammonia-water as working fluids. The exhaust from the diesel engine of the vehicle is utilized as an energy source by the system. The design is to be used in further studies i.e. prototyping and a fully-scaled fabricated model. Enlisting the objectives as follows:

- Development of a numerical model of a simple ammonia-water single-stage absorption cycle
- Further designing of this single-stage absorption cycle with different configurations
- Comparing the results to select the best configuration
- Verification of the final model with results from a standard model
- Analyzing the selected configuration of the absorption cycle with input conditions

CHAPTER-2

Absorption Technology

2.1 Heat Pump

Before further explanation it is important to have some understanding of the technicality of the absorption chiller systems. In this section the basic philosophy behind the working of the vapour absorption cycle will be discussed along with its various types, different configurations among it and its comparison with the other types of thermodynamic cycles. Fundamentally in thermodynamics, a body that remains at a constant temperature even after heat transfer phenomenon from or to it is termed as thermal or heat source. Practically a thermal source can be a body that has either dimensions large enough that any heat loss or gained would not affect its temperature or a body whose temperature is set fixed by another heat transfer process. The example for the former case is the atmosphere or the ocean as their sizes are relatively very large and generally heat loss or gained by them would not affect their temperatures. The latter case can be seen in the tubes of radiator or cooler where the temperature is maintained at a constant value by a heat transfer process. Another term that is used frequently is a thermal machine, which is defined as a close system in which a thermodynamic process between two states will cause the transfer of heat, work and mass. There is a possibility that no fluid is moved in the space during the process. Usually a thermodynamic state changes either by the energy transfer from work due to the movement of certain boundaries or by the energy exchange from heat transfer across the boundaries (conduction, convection etc.). How these transfers occur can be in succession or spontaneous. If a system has no fluid transfer then the process will occur periodically. [1]

A heat pump can be defined as a closed system that can provide heat by using the input work. It can draw heat from a source that is at a lower temperature and release it into a space with a comparatively higher temperature (sink) with the help of some external work. They are divided into two types based on the effect to be achieved in the end. These are:

1) Heat Pump: In the heat pump cycle, as the name states, the heat from a source that is maintained at a lower temperature is transferred to the space (sink) which is at a higher temperature with external input energy which can in the form of mechanical, chemical

thermal etc. (e.g. ‘Heat’ mode of an air condition unit installed in homes works on this principle).

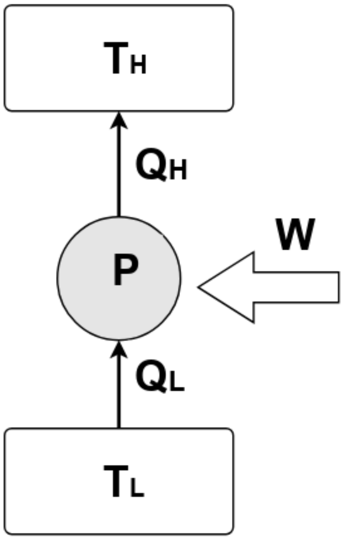

Figure 1: Basic Heat Pump or Refrigeration Cycle

2) Refrigeration Cycle: This cycle provides the cooling effect by removing heat from a source maintained at a lower temperature to a space (sink) at a higher temperature by employing input energy from some external source (e.g. ‘Cool’ mode of an air condition unit works on this principle).

A heat pump is a reversed Carnot cycle. A normal Carnot cycle, which is an ideal thermodynamic cycle for the conversion of heat into work, employs two isothermal processes and two isentropic adiabatic processes. When this cycle is operated in an anti-clockwise way in thermodynamic diagrams (p-v or T-s), a heat pump cycle is obtained.

2.2 Measurement of Performance

For measuring the effectiveness of any process or a cycle, generally a ratio between the useful output energy and the input energy being utilized is calculated. For heat pumps and refrigeration cycles, this ratio is called 'Coefficient of Performance (COP)' and the term is written distinctively by changing the subscript.

For a **heat pump**, since the desired effect is heating so the useful output energy is 'Q_H' which is supplied to the sink with higher temperature. The energy balance can be written as follows:

$$W + Q_L + Q_H = 0 \tag{i}$$

$$|W| + |Q_L| = |Q_H| \tag{ii}$$

Where W represents the mechanical input power, Q_L shows the rate of heat transfer from the low-temperature source (atmosphere) and Q_H denotes the rate of heat transfer released to the high-temperature sink (heating effect).

The coefficient of performance can be written for the heat pump as:

$$COP_H = \frac{Q_H}{W} = \frac{|W| + |Q_L|}{|W|} = 1 + \frac{Q_L}{W} \tag{iii}$$

As described earlier, the COP for the heat pump is denoted by writing 'H' as a subscript. Also as the equation already shows, the value of COP_H will always be greater than 1.

For a **refrigeration** cycle, the same process is repeated. Here the useful energy is the heat removed from the low-temperature source i.e. Q_2. The energy balance will be same as equation (i) i.e.:

$$W + Q_L + Q_H = 0$$

Here Q_L represents the rate of heat transfer from the source to the atmosphere to generate the cooling effect, Q_H is the rate of heat rejected to the atmosphere and W is the input work.

Coefficient of Performance for the refrigeration cycle (COP_C) can be written as:

$$COP_C = \frac{Q_L}{W} \tag{iv}$$

The value of this coefficient can be higher or lower than 1 as shown by the equation.

From the two equations of coefficient of performances, another relation between them can be found as:

$$COP_H = COP_C + 1 \tag{v}$$

This relation can be used to calculate the COP for the cycles operating between the same temperature bounds. This also shows that, based on the desired effect, a single cycle can have two COPs (one for the heating effect and the other for the refrigeration effect). [1]

2.3 Refrigeration Systems

The refrigeration cycle, as discussed earlier, requires external input work to achieve the desired cooling effect. This input energy can be supplied in the form of various ways (mechanical, chemical, electrical and thermal etc.). The most common is the mechanical input work via a compressor or a pump. In this part, the types of refrigeration systems based on the method of mechanical input will be discussed.

2.3.1 Compression Refrigeration Cycle

As the name shows, the mechanical work here is supplied using a compressor driven by an electric motor. The main component of the cycle are listed below:

1) Evaporator: It provides the main cooling effect by removing the heat Q_E from the space to be cooled which is at low temperature T_L.
2) Compressor: It is the mechanical device to provide the required amount of work W.
3) Condenser: It releases the heat Q_C to the atmosphere and it is generally cooled by the use of a cooling network.
4) Throttling Valve: This device is used to reduce the pressure of refrigerant leaving the condenser before entering the evaporator.

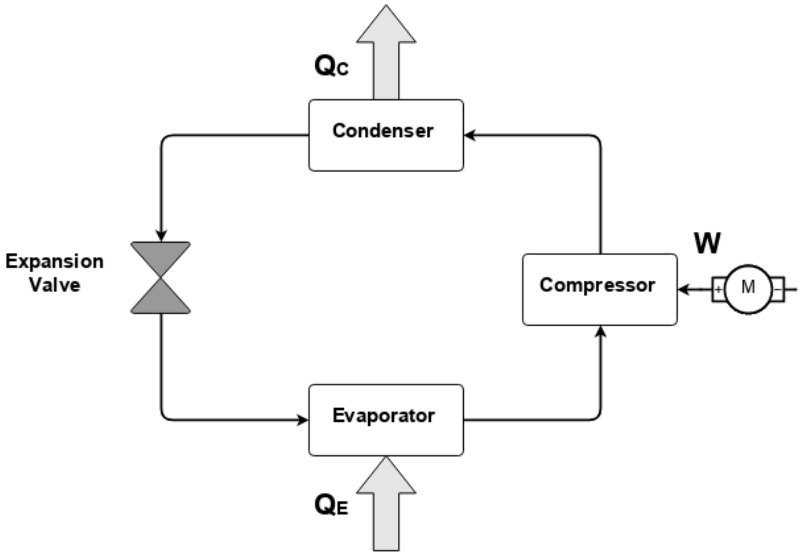

Figure 2: Vapour Compression Refrigeration Cycle Schematic

The process can be explained via a T-s diagram. The sequence of thermodynamic processes during the compression refrigeration cycle is explained as:

1) Compression: The refrigerant enters the compressor as a saturated vapour form having low pressure. From here it is compressed to the superheated state at a higher pressure. Ideally, the process is isentropic but in real-life it is modeled as an irreversible adiabatic process.
2) Condensation: The high-pressure, high-temperature superheated vapour is passed through the condenser where it is cooled by the cooling network (e.g. air or water etc.) and brought to the saturated temperature. The process occurs at a constant temperature and the refrigerant becomes the high-pressure liquid at the end.
3) Throttling: The high-pressure liquid then enters the throttling valve for the expansion valve where the isentropic expansion occurs to bring the liquid to the low pressure of the cycle by keeping the enthalpy constant. The temperature also reduces during this process.
4) Evaporation: The low pressure, low-temperature refrigerant enters the evaporator to draw the heat from the hot source to provide the required cooling effect. The process takes place at constant temperature and the refrigerant becomes low-pressure vapour at the end of the stage. From here it enters the compressor to repeat the cycle.

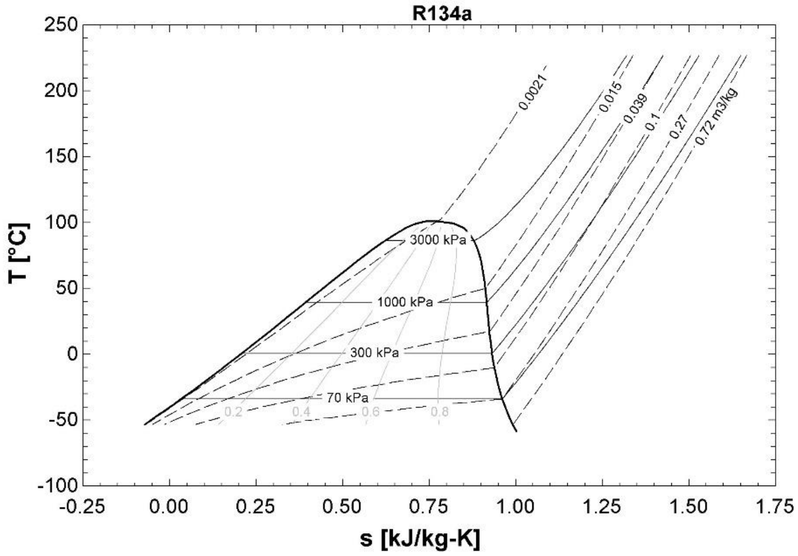

Figure 3: T-s diagram of a commercial refrigerant (R134a)

2.3.2 Absorption Refrigeration Cycle

In absorption refrigeration, the compressor is replaced by the absorber and generator. This technology, as discussed earlier, employs different refrigerant mixtures. An example with an ammonia-water mixture is taken to explain the absorption refrigeration cycle. In this cycle, ammonia is the refrigerant that will produce the required cooling effect whereas water is used as an absorbent as it has a strong affinity for ammonia.

The main components for the absorption refrigeration cycle include:

1) Evaporator: Its function is the same as in the compression refrigeration cycle i.e. it provides the required cooling effect.
2) Absorber: It contains the solution of ammonia and water. An external cooling network is also attached to cool the mixture during the process.
3) Generator: It also contains a mixture of ammonia and water. An analyzer is placed above it whose main function is to separate the ammonia from the water.
4) Pump: It transfers the refrigerant mixture from absorber to generator by increasing the pressure.
5) Pressure Reducing Valve: It reduces the pressure of the mixture going from the generator back to the absorber.

6) Condenser: As in the compression refrigeration cycle, it converts the vapour refrigerant coming from the analyzer into the liquid by changing its phase. An external cooling network is attached with it too.
7) Expansion Valve: It reduces the pressure and temperature of the refrigerant coming from the condenser via the expansion process. From here the refrigerant enters into the evaporator to repeat the cycle.

The working process of the ammonia-water absorption refrigeration cycle is explained below:

Firstly, a chilled, low-temperature and low-pressure ammonia refrigerant coming from the expansion valve enters into the evaporator. As mentioned earlier, the desired cooling effect produces here when this refrigerant absorbs the heat Q_E from the surroundings. In this process, the chilled low-pressure and low-temperature ammonia turn into a low-pressure ammonia vapour.

This low-pressure ammonia vapour then enters into the absorber which already contains a weak solution of ammonia and water. As water has a strong affinity to absorb ammonia, it will start absorbing this incoming ammonia refrigerant from the evaporator. This will increase the concentration of ammonia in the solution and the weak solution present in the absorber will start becoming a strong solution. As more and more ammonia is being absorbed by the water, the mixture will be heated up and decrease the capacity of water to absorb ammonia further. To keep the temperature of the absorber up to an optimum level, an external circuit of cooling medium (e.g. water etc.) is attached. It stops the temperature to rise and maintains the capacity of water to absorb the incoming ammonia refrigerant continuously hence producing the strong ammonia-water solution.

The pump then transfers this strong ammonia-water solution to the generator. Here, the input heat Q_G is supplied from the external source e.g. heater, steam or exhaust heat from any process etc. As the strong solution of ammonia and water is heated, both ammonia and water will start turning into vapour form. Ammonia turns into vapour faster than water but eventually, the vapours of both of them will fill the generator. The rectifier placed above the generator functions to separate the water vapour from the ammonia vapour and only allows the ammonia vapour to pass. This is because if water vapours enters in to the system it will affect the efficiency of the refrigeration system and a larger quantity of this water vapour can damage the system as well. The water in the vapour form is condensed by the rectifier and sent back to the generator. From here, this residue water along with some ammonia (weak solution) is sent back to the absorber by reducing its pressure through a pressure reducing valve.

From the rectifier, the high temperature and high-pressure ammonia vapour enters into the cold condenser where it loses its latent heat because the condenser is supplied with the external cooling circuit. Thus, the ammonia vapour entering into the condenser, leaves it as high-pressure liquid ammonia.

After condensation, the high-pressure liquid ammonia enters the expansion valve. Here the expansion process occurs which decreases the pressure of refrigerant considerably along with its temperature thus converting it into a chilled low temperature and low-pressure ammonia. From here, the refrigerant enters into the evaporator to produce the cooling effect and the whole cycle is repeated over and over again.

An important aspect here is the functioning of the pump. Since the pump is acting upon the liquid phase (mixture of water and ammonia), the overall mechanical power required is less than that of the compressor in the compressor refrigeration cycle. [1]

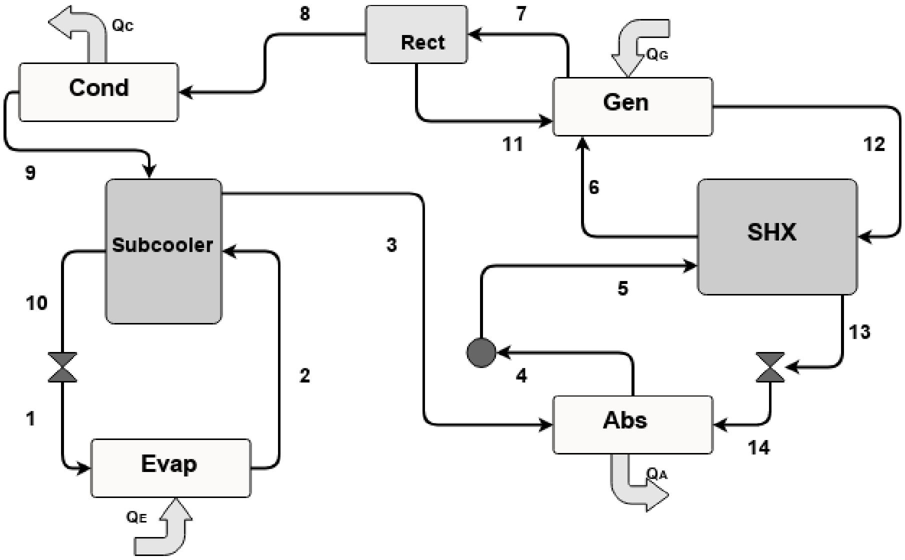

Figure 4: Ammonia-Water Absorption Refrigeration Cycle

2.4 Coefficient of Performance of Absorption Refrigeration System

The expression for the performance of the absorption refrigeration system is developed in the same way as done previously as follows:

$$COP_{ABS,REF} = \frac{Q_E}{Q_G + W_{pump}} \qquad \text{(vi)}$$

Here the term Q_G is the heat supplied to the generator, W_{pump} is the mechanical power applied by the pump and Q_E is the useful heat removed for the cooling effect.

One of the main advantages of using an absorption refrigeration cycle is that it provides the opportunity to recover heat even at low temperatures ($120\ ^{O}C - 160\ ^{O}C$). The heat energy available in the surplus in industries e.g. waste heat from combustion or steam etc. which is otherwise wasted is utilized by the absorption refrigeration cycle.

2.5 Essential Characteristics of Working Fluids of Refrigeration

The performance of any cycle is greatly affected by the characteristics of the working fluids that are being used. For the efficient working of the refrigeration cycles, the working fluids must have some desirable properties. These are discussed below as:

1) At the given heat rejection temperature, the condensing temperature is not very high.
2) At the given cooling temperature, the evaporation pressure is not very low.
3) High value of latent heat during the transition of phase so it will reduce the volume of working fluid for the same heat rate at the evaporator.
4) For a reduced volumetric flow, a low saturated vapour specific volume is preferred. Due to this, compressor size will also be reduced as there will be higher mass flow for the same volume flow.
5) Non-toxicity.
6) In case of leakage there must not be any risk of inflammability and explosiveness.

7) It must be compatible with the construction materials that are used in refrigeration equipment i.e. stainless steel, brass and copper etc.
8) In case of leakage, fluids must be environmental friendly.

As discussed earlier, initially the chlorofluorocarbons were used for refrigeration and heating but after the Paris agreement in 2016, these fluids are considered as the major contributors to the depletion of the ozone layer, causing global warming and thus affecting the environment adversely.

2.6 Indication of Effects on Environment

To measure the compatibility of a fluid with the environment, a few parameters are defined as follows:

1) GWP: It is the “Gross Warming Potential”. It is the amount of radiant energy in the infrared band that gas can absorb in 100 years compared to that of the carbon dioxide (which is used as standard i.e. ($GWP_{CO_2} = 1$). Its value depicts the effect of the emission from the refrigerant on global warming.
2) ODP: it is the “Ozone Depletion Potential” and indicates the ability of a gas to deplete the ozone layer. Its value ranges from 0 to 1. The ideal refrigerant has ODP equal to zero. The refrigerant trichlorofluoromethane (CFC-11 or R11) is used as standard which has ODP equal to 1.

There are some other terms like IGWP (“Indirect Global Warming Potential”) and TEWI (“Total equivalent Warming Impact”) which describes the impact of a refrigerant on the environment due to its working characteristics in the form of radiative heat. These terms quantify the energy transformations during any process (e.g. combustion, nuclear, renewable etc.). All these factors impose the necessity to choose a refrigeration system with high values of COP.

2.7 Absorption Cycle Working Fluids

In the previous discussion, we explained the use of absorption cycles for industrial and domestic purposes. The presence of the enormous amount of low-grade heat which would otherwise be wasted is employed by the absorption cycles to produce the required heating and cooling effect. As these cycles use a mixture of refrigerant, it benefits the environment by reducing carbon dioxide emissions and using non-toxic working fluids. The development of the absorption cycle started in the 1700s (in 1784 particularly by William Cullen). As it was established previously that with the evaporation of pure water from a vessel placed inside an evacuated container in the presence of sulfuric acid, ice can be formed [2]. In 1859, Ferdinand Carre introduced a novel machine using ammonia water as the working fluid. This machine took out a US patent in 1860. Furthermore, in 1950's, a system using water-LiBr as the working fluid was developed for industrial applications. A critical review of absorption technologies is given which contained a short introduction of working fluids reported before 2001 [3].

In an absorption cycle, the working fluid is composed of a refrigerant and an absorbent. The absorber contains the solution of refrigerant and absorbent which becomes a strong solution as more and more refrigerant is absorbed from the evaporator. This solution is then pumped into the generator where heating is done to separate the refrigerant from the absorber. The weak solution thus formed inside the generator is moved back to the absorber to keep the cycle continuous. The refrigerant separated is condensed in the condenser to keep the pressure of the generator and condenser constant. The thermodynamic properties of the refrigerant greatly affect the performance of the absorption cycle [4]. The ideal characteristics for the working fluid are described below: [5]

1) The elevation of boiling (the difference in boiling point between the pure refrigerant and the mixture at the same pressure) should be as large as possible.
2) Refrigerant should have high heat of vapourization and high concentration within the absorbent to maintain a low circulation rate between the generator and the absorber per unit of cooling capacity.
3) Transport properties that influence heat and mass transfer e.g. viscosity, thermal conductivity, and diffusion coefficient should be favourable.
4) Both refrigerant and absorbent should be non-corrosive, environment friendly, and low-cost

In a survey on absorption fluids, it has been discussed that there are about 40 refrigerant compounds and 200 absorbents available [6]. For clarity, the working fluids are generally classified in to five series: Ammonia series, alcohol series, water series, halogenated hydrocarbon series and other refrigerants.

2.8 Absorption Cycle Configurations

The absorption cycle discussed up till now is of the simplest type to develop the basic understanding of the fundamental processes and working. In reality, there have been many configurations of this absorption cycle based on the factors as discussed below [7]:

Number of Stages

This feature is related to the structure of the cycle. The simple absorption cycle is termed as a single-stage i.e. 4 heat exchangers as the evaporator, absorber, generator, condenser and solution pump are used in a basic single-stage absorption cycle. The number of times this configuration is repeated indicates the number of stages. Each stage can have the optional heat exchangers as sub-cooler and solution heat exchanger.

Number of Effects

The number of times the high-temperature heat is used in the cycle either directly or indirectly depicts the number of effects. This is done by merging or connecting two heat exchangers especially when the temperature of heat from one heat exchanger is high enough to be used again by the other heat exchanger(s).

Number of Pressure Levels

The number of expansions that occur inside the cycle shows the number of pressure levels of that cycle. Generally the number of pressure levels corresponds to the number of stages in the absorption system but there can be configurations where there are several stages or effects in a cycle with only one pressure level.

2.8.1 Double Effect Absorption Cycle

As discussed earlier the concept of effect in an absorption cycle, the double effect cycle is developed to increase the overall efficiency of the system. As per the concept, the heat at a high temperature is exploited twice in the cycle by introducing two new heat exchangers i.e. high-temperature generator and a high-temperature condenser. The output heat from the high-temperature condenser is supplied to low-temperature generator thus increasing the COP of the cycle.

As by employing this configuration, the COP increases from 0.6-0.7 (of a single effect absorption cycle) to 1.0-1.2, hence the double effect absorption cycle competes well economically with the conventional vapour compression cycle than the single effect absorption cycle (which is only advantageous from the point of view of utilizing the low-temperature waste heat). The double effect absorption cycle can have several other configurations as well which may increase the number of absorbers and heat exchangers thus affecting the number of expansion stages and pressure levels. The generator inlet temperature is usually around 140-160 ^{O}C.

2.8.2 Triple Effect Absorption Cycle

The previous explanation for the double effect cycle can be used to discuss this technology as well. As the name suggests, the triple effect cycle will be able to employ the high-temperature heat thrice during the whole process. Though the experimentation and research were done on the development of this technology shows the ascent in the performance indices reaching up to 1.4-1.5 by employing the heat of 180-200 ^{O}C at the inlet of the generator, yet this comes with the expense of complexity in manufacturing the system especially in designing of the internal piping, construction material and maintaining the internal flows.

2.8.3 Generator-Absorber Heat Exchange Absorption (GAX)

This technology is developed to obtain the higher values of COP while keeping the complexity of the overall system to the minimum. In GAX cycle the configuration of a single effect absorption cycle is used to increase the performance of the system. The only difference is the temperature overlap between the generator and the absorber is exploited here. With the rise in the pressure of the absorber, the output temperature of the absorber also increases up to the point that it overlaps with generator temperature. The heat from the absorber is thus, transferred internally to increase the coefficient of performance of the system. This configuration is only achievable with working fluids such as ammonia-water. The COP of the cycle is around 1.0 with input heat at 140-150 ^{O}C. Therefore, despite utilizing the configuration of a single effect absorption cycle, the performance of GAX cycle is comparable to the double effect absorption cycle, not only in terms of COP but also from the economical point of view. Due to this reason, the GAX technology is recently being used widely for refrigeration and space heating or cooling.

2.9 Advantages of Absorption Refrigeration over Compression Refrigeration

The absorption refrigeration is more advantageous than vapour compression systems in the following ways [8]:

1) Refrigeration systems run on cheap fuels (e.g. furnace oil, waste heat, geothermal heat etc.) as compared to the vapour compression cycles which requires electricity as input energy incurring a high cost of running.
2) In refrigeration systems, the vibrations are quite low and the requirement of lubrication is very less due to no moving mechanical parts (except a solution pump) than those of the mechanical compression systems.
3) Vapour compression systems undergo frequent maintenance outages due to reciprocating parts whereas absorption system presents high plant availability due to no reciprocating parts (except a solution pump which requires less mechanical power as compared to a compressor).
4) Absorption machines have no loss of efficiency at part load, the efficiency improves at part load while the vapour compression machines have fixed mechanical losses at part load hence efficiency is lost.
5) The working fluids used by the absorption systems are environment friendly (e.g. water, ammonia etc.) which have minimal effect on the ozone layer and global warming compared to the compression machines which mostly use CFCs which have severe effects on the environment.

2.10 Limitations of Water-Lithium Bromide Systems

As discussed earlier, the two most commonly used working fluids for the vapour absorption cycle are Ammonia-water and Water-Lithium Bromide. Though the coefficient of performance of Water-LiBr is slightly higher there are several limitations of these systems [9]:

1) Using water as a refrigerant has a disadvantage due to its freezing point at 0 ^{O}C hence the machine cannot operate at an evaporation temperature below 0 ^{O}C. It hinders the application of these systems in cryogenic refrigeration or domestic heating of water in colder areas.

2) There is a possibility of crystallization of the water-LiBr if the evaporation temperature is relatively low or absorption temperature is high hence air-cooled absorption is quite difficult.
3) To avoid the degradation of the overall performance of water-LiBr systems, high vacuum conditions must be maintained.

Due to the reasons explained above, using ammonia-water as working fluids in the vapour absorption systems is a suitable choice especially for cryogenic conditions and air-cooled absorption. Moreover, there is not any risk of crystallization of the fluids and the necessity of vacuum for ammonia-water systems.

CHAPTER-3

Literature Review

3.1 Expedition for Alternate Refrigeration Technologies

The need for cooling, refrigeration, and air conditioning technologies has increased dramatically in recent decades to meet a variety of engineering and comfort requirements. The consumption of electricity produced globally by conventional devices like vapour compression refrigeration systems is about one-fifth as per the statistics published by the International Institute of Refrigeration (IIR) and this rate of consumption is expected to rise to one-half in the future due to the increased demand by domestic and commercial sector [10]. This huge increase in the demand of energy has not only escalated the prices, but has also put enormous pressure on the sources used conventionally for energy. Hence there is an alarming situation as in the absence of any appropriate actions, the world will run out of fossil fuels soon. Therefore, there is a need to overcome this problem either by discovering new energy sources or by bringing down the consumption rate thereby conserving the existing sources of energy. In the pursuit of exploring new energy sources, the research in recent years has opened the venue of utilizing renewable energies (particularly due to their long-term and abundant availability) e.g. wind energy, solar energy and the waste heat from industrial operations etc. as energy sources to power refrigeration systems. [11] [12]

The vapour compression refrigeration systems prove to be disadvantageous not only due to their huge consumption of energy but also because of their production of greenhouse gases during their operation which is a great threat to the environment. This is mainly due to the usage of conventional refrigerants (e.g. HCs and HCFCs) as working fluids in vapour compression refrigeration systems which affect the ozone layer adversely causing global warming. To prevent the depletion of the ozone layer and degradation of the environment, restrictions were imposed on the usage of these working fluids by the global environmentalists as per the protocols established in 1987 (Montreal) and 1997 (Kyoto). The statistics issued by NASA has shown that there is a 4,300,000 km^2 increase in the hole size in the ozone layer from 1994 [13]. Under such circumstances, the adaptation of a resolution by the European Commission (EC) to prohibit the use of all HCFCs by 2015 took place in October 2000 [14] [15]. Another report by an institute on climate change revealed that from the start of the present century, there is an increase of 0.6 K in the average global temperature and by forecasting the ongoing trend, this number is predicted to rise to 1.4-1.5 k by 2100 [16].

The problems caused by the conventional vapour absorption refrigeration system to the environment has enabled the discovery of new refrigeration technologies which are neither destructive to the environment (as they use environment-friendly refrigerants) nor are the source of energy depletion as these technologies employ renewable energies and waste heat. One of the highly efficient examples of such technologies is absorption refrigeration systems. The absorption refrigeration system is defined as ‘thermally driven refrigeration technology for exploiting the heat from low-grade energy sources for cooling purposes, [17]. Based on the advantages of these absorption refrigeration machines, the technology is widely used in the European region and about 60% of all the thermally driven refrigeration systems in Europe are absorption refrigeration systems [18]. Moreover the application of absorption refrigeration machines is now being extended for not only small-scale commercial and domestic applications but also micro-industrial refrigeration purposes (e.g. food storage and ice-making) in developing regions of the world. [19]

3.2 Ammonia-Water Absorption Refrigeration System

From the previous discussion, it is clear that the absorption refrigeration cycle utilizes different refrigerants from the conventional vapour compression cycles which are beneficial to the environment too. A study on the major groups of working fluids of absorption cycles is studied in which they are divided into five major categories based on the type of refrigerant used. It comprises of series of refrigerants using ammonia, water, alcohol, halogenated hydro-carbon and other mediums [20]. Various analytical, numerical and experimental studies have been carried out for the performance analysis of absorption systems with different working fluid pairs at varied operating conditions where the working pairs are ammonia and lithium-bromide based refrigerants along with some other additional mixtures [21] [22]. However, the most widely used working fluid pairs for absorption refrigeration are NH_3-H_2O and H_2O-LiBr [23] [24] . The lithium bromide and water offer a higher coefficient of performance but the limitations of this working pair hinder in choosing them in certain cases. As previously discussed, the H_2O-LiBr cannot be used for domestic heating due to their limitation of operating in cryogenic regions [25]. Moreover, the mixture tends to crystallize when the absorption temperature is high or evaporator temperature is low thus restricting the possibility of air-cooled absorption [26] [27]. Also for the efficient operation of H_2O-LiBr systems, there is a need of maintaining high vacuum conditions which are also a drawback of using this working fluid pair [28]. These limitations provide ammonia-based working fluids with an edge over the water-lithium bromide solution, thus making a better choice for absorption refrigeration application in domestic heating, subfreezing conditions and air-cooled absorption. Due to the non-crystallization characteristic and absence of the requirement of vacuum conditions, this working pair can gather much attention. For example, the adaptation of a single-stage absorption machine with NH_3/Water working pair to achieve the refrigeration temperature from -35 ^{O}C to 5 ^{O}C has yielded impressive results [29].

3.3 Waste Heat Utilization of Absorption Machines

The vapour absorption refrigeration cycle, as discussed earlier is advantageous when it comes to the utilization of renewable energy and waste heat. This waste heat can be a by-product of any industrial process or combustion process of an engine. There have been several attempts in employing the energy of exhaust gases from an internal combustion engine.

An experimental study of an ammonia-water absorption refrigeration system using the exhaust of an internal combustion engine as an energy source is carried out by Manzela et al. [30] where it was observed that the refrigerator reached a steady-state temperature between 4 and 13 ^{O}C after about 3 hours of system startup and this cooling capacity can be increased if a dedicated system is installed for the required air conditioning in automotive. This system also resulted in reduced carbon monoxide emissions in the environment.

Vicatos et al. [31] researched on a passenger car in which the energy from the exhaust gas of an internal combustion engine was used to power an absorption refrigeration system for air-conditioning. A prototype is developed and tested in the laboratory and road-test conditions which verified the theoretical analysis. The exhaust gases were able to provide enough heat for the cooling operation without any dependence on the mechanical output of the engine thus reducing the fuel consumption and emissions. The aqua-ammonia solution proved to be a good working pair however, the COP can be increased further by using high purity refrigerants.

Another prototype of a single-stage ammonia-water absorption refrigeration system was designed, fabricated and tested by Koehler et al. [32] for truck refrigeration using the exhaust gases. The truck-driving conditions of city traffic, mountain roads and flat roads were simulated and the results indicated the COP of 27% which can be improved to values over 30% this system can be a beneficial refrigeration alternative for the trucks travelling long distances on flat roads.

Meunier [33] described waste heat adsorption cooling as totally clean process that hasn't any disruptive effects on the environment. However, its application in automotive air-conditioning could be an important challenge but despite the low coefficient of performance, this would be a quite competitive solution for the sorption systems from the point of view of global warming. Moreover, the smart design of the components is required to make them compact and light-weighted for improved efficiency.

A novel adsorption air-conditioning system was suggested by Jiangzhou et al. [34] which is driven by waste heat from the exhaust gas of the internal combustion engine. The experiments and prototype showed that the system is simplified in structure, has reliable operation and convenient control. The mean refrigeration power is about 5kW and the air outlet temperature of the coil-fan is approximately 18 ^{O}C. This system, as per the author, can meet the demands of air-conditioning for locomotive driver cabin and with the improvement in the design of evaporator and air-cooled absorber, the refrigeration capacity can be increased.

Qin et al [35] developed a new hydride pair of working fluid to be used in an exhaust-gas driven automobile metal hydride refrigeration system along with a cylindrical design for the reactors for hydrogen flow. The results showed that with the increase in the heat source temperature, the coefficient of performance and the cooling power of the system increased while the minimum refrigeration temperature decreased. Hence the heat transfer properties of the hydride pair requires improvement for the efficient performance of the system.

The concept of "integrated thermal management controller (ITMC) for internal combustion engine based cogeneration system application was presented by Huangfu et al. [36] based on the principle of VCHP (Variable Conductance Heat Pipe-special pipe to control the temperature of a heat source). The ITMC experimental prototype was established which verified the working principle of ITMC by keeping the vapour temperature variation to only 5.3 ^{O}C between two designed extreme experimental conditions. The prototype thus can control the temperature in variable working conditions.

A novel micro CCHP (Combined Cooling, Heating and Power) system is constructed by Li and Wu [37] employing two-bed silica gel-water adsorption chiller. It is found from the simulation results that the average value and variation rate of electric load significantly influence the cooling capacity and the coefficient of performance of the chiller. The authors recommended the use of a 500 L water tank for better performance and reasonable start-up time and the inclusion of a cold accumulator for higher security of the system.

Tao Cao et al. [38] modelled and simulated a waste heat powered absorption cycle cooling system for shipboard applications and compared its performance with a vapour compression cycle cooling system. The results showed that in comparison to baseline COP of 3.6, the waste heat powered cooling system has an electricity-input based COP of 9.4 (in case of "free" waste heat energy) and a cooling COP of 0.6. The carbon dioxide emission and fuel consumption of waste heat powered cooling systems are 62% less than VCC cooling system and it can save the total energy by 8.23%.

An experimental investigation of the effects of three different heat inputs to the generator on the energy performance of the diffusion absorption refrigeration system was conducted by Mustafa [39] where a conventional diffusion absorption refrigeration system is used under heat inputs of 62, 80 and 115 W. The results of the analysis showed that the lowest energy performance was calculated as 0.30 for DAR-115W while the highest energy performance was 0.36 for DAR-62W system.

The efficient and simultaneous recovery of jacket water and exhaust gas waste heat from an internal combustion engine for refrigeration was studied by Wang and Wu [40] by using a mixed-effect absorption chiller coupling the single and double effect processes. The performance was compared to the CCHP systems and the simulation results showed that a 16 kW internal combustion engine has a COP of 0.96, exergy efficiency of 0.186 and cooling output of 34.4 kW. Hence a CCHP system utilized the waste heat efficiently as compared to a single or double-effect absorption chiller.

Hilali et al. [41] performed an experimental application of an engine exhaust-gas driven absorption refrigeration system. A 2 kW air-cooled absorption refrigeration system was integrated with the

1.4 L internal combustion engine to be used as a waste heat source. The results indicated the cooling capacity of 0.5 kW and 2.8 kW at idle and full load conditions respectively when the waste heat from the exhaust gas was varied between 3 kW and 16 kW from idle to full load respectively. The system showed an overall performance coefficient of 0.18 which can be increased by improving the generator design to have less pressure drop and better heat transfer efficiency.

An alternate refrigeration system based on the vapour absorption refrigeration cycle was suggested by Horuz [42] which utilizes the waste heat in the exhaust gases from the main propulsion unit of the vehicle. The experimental results proved that the 6 L turbo diesel engine was able to power a vapour absorption refrigeration system that uses aqueous ammonia solution as a working fluid pair and produced enough waste heat energy to achieve 10 kW of rated cooling effect. In another study by Horuz, the performance of a vapour absorption system fired by natural gas is compared with that of the same system driven by exhaust gases of an internal combustion engine [43] and it is concluded that the performance of both configurations yielded the same coefficient of performance and by careful designing, the coupling of the generator of a vapour absorption refrigeration system with the main engine exhaust system would not impede the performance of the vehicle propulsion unit.

Bux and Tiwari [44] analyzed single-stage vapour absorption refrigeration system using three different working fluid pairs (i.e. Lithium Bromide-water, Ammonia-water and Lithium Chloride-water solutions) driven by the exhaust heat or low-grade heat energy of four cylinder-four stroke diesel engine. The cooling capacity of $LiBr-H_2O$, NH_3-H_2O and $LiCl-H_2O$ was found to be 8.2, 7.0 and 8.3 respectively. The air conditioning systems were eco-friendly and used natural refrigerants which increased the mileage of the vehicle as well.

Another interesting research was done by Lethwala et al. [45] in which a single-effect ammonia-water vapour absorption refrigeration system was developed for automotive truck cold storage. The system worked on the exhaust gas of the vehicle and managed to reduce the temperature of cold storage up to -11 ^{O}C as compared to a vapour compression refrigeration system which reduced the temperature up to 1 ^{O}C. The results also indicated that the COP of the system varies from 0.3528 to 0.3113 under different operating conditions. Moreover, for a minimum assumed COP of 0.2, the input energy required by the generator is 17.5 kW where-as the waste heat available is 34 kW.

Manzela et al. [46] presented an experimental analysis of an ammonia-water absorption refrigeration system which was using the energy from the exhaust gas of an internal combustion engine and then from the original energy source i.e. Liquefied Petroleum Gas (LPG). The results showed that available energy for the refrigerator was higher when the engine exhaust gas was used as an energy source with 25% throttle valve opening thus the highest COP was recorded as compared to the case with LPG and wide-open-throttle valve.

A regenerator operating on open-cycle absorption air-conditioning system using exhaust gas of a petrol or diesel engine was conceptualized by Kaudinya et al. [47] who discussed that the cooling capacity in the exhaust gas is far in excess as compared to that required for cooling the internal space of the transport vans particularly transporting the food to distant markets. The author claimed that the operating cost of such system would be negligible but it would slightly increase the operating cost of the van.

Fernandez et al. [48] designed, modelled and analyzed the gas-to-thermal fluid heat recovery system in a trawler chiller fishing vessel from engine exhaust to drive an ammonia-water absorption refrigeration plant for onboard cooling application. These fishing vessels use four-stroke diesel engine with power ranging from 700 to 1200 kW thus the high and constant engine load during the fishing period permitted the heat recovery from the exhaust of the engine at a thermal level greater than that required for driving the absorption system. The simulation results from the real data showed that this type of heat recovery system is feasible for trawler chiller fishing vessels.

A thermodynamic analysis to determine the properties of exhaust gases of a diesel engine as a function of temperature and pressure ratio of the cycle was carried out by Mostafavi et al. [49] where the available and recoverable thermal energy from the exhaust gases were calculated to drive the absorption refrigeration unit. The results indicated that the cooling capacity of the exhaust gases is highly dependent on the operation conditions of diesel cycle. For a fixed maximum cycle temperature, the cooling capacity decreases upon increasing the cycle pressure ratio (due to high expansion ratio as compared to the low-pressure ratio cycles). Similarly, the cooling capacity is increased upon increasing the cycle temperature ratio at a fixed expansion ratio (because a high-temperature ratio causes the exhaust gas temperature to rise). Hence a low cycle temperature ratio was able to improve the performance of the system.

The feasibility of using waste heat from marine diesel engines to power an NH_3-H_2O absorption refrigeration system is examined by Ouadha et al. [50] explained that the quantity of waste heat provided is sufficient. The simulation results indicated that the COP of the system varies from 0.6057 to 0.6937 when the effectiveness of the solution heat exchanger is varied from 70 to 100 respectively (at the generator, condenser, absorber and evaporator temperatures of 90, 25, 25 and -5 ^{O}C). The results also showed that the cycle performance is increased upon increasing the generator and evaporator temperature or decreasing the condenser and absorber temperatures.

Kumar et al. [51] studied experimentally the working of an ammonia-water absorption refrigeration system driven by the exhaust of a turbo-charged diesel engine. The maximum COP obtained from the results was 0.136 and it was possible to produce the refrigeration effect of 2.2 kW without any input energy loss. In this study, the generator was not fed directly with the waste gas and only a fraction of heat energy was used to maintain the temperature of 95 ^{O}C if not done so, the temperature would reach higher than 400 ^{O}C eliminating the requirement of the electric heater as an energy input.

Waste heat utilization from the exhaust of a diesel engine by a cogeneration system was studied by Samanta et al. [52] in which an integrated cogeneration system comprising of a double stage vapour absorption refrigeration cycle and an organic Rankine cycle is driven by the exhaust gas from the diesel-engine produced power and cooling effect. The results showed that the maximum power was achieved at a higher evaporator pressure of organic Rankine cycle and a lower condenser pressures where-as the maximum cooling effect was achieved at an elevated evaporator temperature of the absorption refrigeration cycle and a minimal lower condenser temperature. An electrical power and cooling capacity of 20.80 kW and 69.12 kW was recorded from the simulation results as well.

An integrated refrigerating system comprising of a reciprocating engine, a vapour compression chiller and an absorption unit was analyzed by Longo et al. [53] and the COP and PER (Primary Energy Ratio: the ratio between the whole refrigerating capacity and thermal power of the fuel supplied) was assessed. The simulation results showed the value of COP around 1 where-as the PER was found to be around 1.6 (25% higher than the traditional vapour compression unit) due to the absorption unit which supplied 1050 kW out of 4100 kW of whole refrigeration capacity.

CHAPTER-4

Methodology

4.1 Brief Overview of Methodology

In this chapter, the theoretical analysis of an absorption chiller is done along with the numerical model of the system and the simulation results are discussed. Firstly, the equation used to develop the numerical model are shown with the assumptions taken into account are explained. The process starts with the modelling of a single effect ammonia-water absorption cycle which is constructed by engaging the fundamental components only. These are the evaporator, absorber, generator and condenser along with the solution pump and essential valves. The simulation is performed to analyze the variation of the coefficient of performance by changing some parameters of the system.

The modelling process is then proceeded with this basic design to create further configurations of a single-effect absorption cycle. In the second step, a solution heat exchanger is added in the basic single-effect cycle to analyze the COP of the system further. The solution heat exchanger is placed between the absorber and the generator.

In the third step, a sub-cooler is added to the absorption cycle with the solution heat exchanger hence another configuration is achieved. The simulation is performed for the analysis of the performance index. The sub-cooler is added between the condenser and the evaporator.

In the last step, a rectifier is modelled in the previous configuration with the sub-cooler and solution heat exchanger. The role of this rectifier will be discussed ahead. The rectifier is placed between the generator and the condenser.

4.2 Assumptions for Thermodynamic Analysis

All the equations, mass balances and heat relations that are coded for the simulation of the absorption cycles are as per the fundamental laws of physics. For the thermodynamic analysis the assumptions taken are as follows:

1) Heat losses to the surroundings are negligible.
2) The efficiency of the solution pump is 0.8

3) Frictional losses are negligible.
4) At the outlet of the absorber, the strong solution is saturated.
5) In the absorber, the mixing of vapour and the weak solution is considered adiabatic.
6) There is a thermodynamic equilibrium between the refrigerant vapour at the inlet of the condenser and the liquid at the rectifier for the separation of liquid and vapour.

4.3 Determining State Properties of Working Fluids

In the designing process of an absorption cycle, the main thermodynamic properties to be taken in to account are pressure, temperature, enthalpy, mass fraction, entropy and specific volume. During the simulation process, these properties are calculated by the EES software but the orthodox way to determine the properties is by using the property charts. These charts can be of three-dimensional type but for simplicity, generally two-dimensional charts are used where some of the properties are constant and these are presented as constant curves or lines.

In the case of pure fluids, the charts that are widely used are temperature-entropy (T-s), pressure-enthalpy ($ln(p)$-h) or enthalpy-entropy charts (h-s) but in the case of mixtures of fluids, an additional parameter is required to fully define a state point i.e. the "mass fraction" which is simply a ratio of 'mass of one component' and the 'total mass of both components'. The commonly used diagrams for the thermodynamic properties of mixtures are the Temperature-Mass fraction diagram, Pressure-Temperature diagram and Pressure-Enthalpy diagram.

4.4 Model Development

The numerical modelling of the vapour absorption cycle is described for the fundamental and the advanced configurations. The input heat energy required for the generator is taken employed from the exhaust of the diesel engine of the industrial vehicle.

An overview of the processes occurring in a vapour absorption cycle is as follows. First of all the the strong solution present in the generator is heated by using the heat energy from an external source (which in our case is the diesel enginer exhaust). This heat causes the separation of ammonia from the water. As the boiling point of ammonia is less than water, it will start converting into vapours first. Eventually vapours of both ammonia and water from in the generator leave behind a poor solution. These vapours are then segregated using the rectifier which only allows the ammonia vapours to pass and the water is condensed back into the generator. These ammonia vapours are then transported to the condenser where it loses heat in condenser tubes. The cooling is supplied through an external source here (air, water etc.). After passing through the condenser the ammonia vapours are liquified but the pressure is still high. Before entering back into the evaporator, this high-pressure liquid undergoes the expansion process in a throttling valve. The process is iso-entropic in nature. This low-pressure, low temperature liquid ammonia is now all set to produce the required cooling effect in the evaporator. In the evaporator, it absorbs the heat from the cooling space thus performing its essential function. After the evaporator, the ammonia is passed on to the absorber where there is a poor solution of ammonia and water. This poor solution starts absorbing the incoming ammonia vapour from the evaporator thus gradually forming the strong solution. To keep the mixing going on, it is necessary to remove the heat produced during the mixing process, hence an external supply of cool water or air is given. The strong solution produced is then pumped into the generator where again the ammonia is separated from the strong solution using the auxilliary heat. The poor solution collected at the bottom of the generator is sent back to the absorber after passing through the pressure reducing valve. This poor solution from the generator is mixed with the incoming ammonia vapour from the evaporator.

These processes occur in the basic single effect ammonia-water absorption cycle. For enhancing the performance of this cycle, often another heat exchanger is placed between the absorber and the generator which preheats the strong solution entering from the absorber into the generator and precools the poor solution coming from the generator in the absorber. A sub-cooler is also placed to further increase the performance especially when the evaporator temperature is low. It is placed between the condenser and evaporator thus heat is exchanged between the liquid and vapour phases of ammonia. Superheated vapour in the absorption cycle does not affect the COP negatively as in the case of the vapour compression cycle.

The condensation and evaporation temperatures are used to detemine the two pressure levels of the absorption cycle.

4.4.1 About EES

Engineering Equation Solver (EES) is a software capable of solving thousands of differential equations and coupled non-linear algebraic equations. It comes with a built-in library of thermodynamic and transport properties of numerous substances. EES is developed by F-Chart Software which is a project of Professor Sanford A. Klein (from University of Wisconsin-

Madison) who, under the banner of F-Chart Software, has provided innovative analysis tools for both commercial and educational applications. In this study, a professional version (V10.561) of EES is used to solve a system of equations for the modelling of ammonia-water absorption system.

4.4.2 Ammonia-Water Mixture Functionality in EES

The Engineering Equation Solver (EES) uses the equations of mass and energy balance to simulate the absorption cycle. The thermodynamic properties of an ammonia-water mixture are calculated from the built-in library of EES. There are two ways to find the properties; one is to use the CALL NH_3H_2O function in EES which is composed of three inputs and eight outputs. The structure of CALL NH_3H_2O is shown below:

$$CALL\ NH_3H_2O\ (Code, In1, In2, In3{:}\, T, P, x, h, s, u, v, q)$$

The first four parameters on the left are the inputs required by the function. The function needs these properties with their respective units; T= [K], P= [bar], x= [ammonia mass fraction], h= [KJ/kg], s= [KJ/kg], u= [KJ/kg], v= [m^3/kg], q= [vapour mass fraction].

The value of ‘q’ indicates the region in which the mixture lies e.g. $0 \leq q \leq 1$ indicates the saturated states, $q = -0.001$ indicates the sub-cooled region and $q = 1.001$ indicates the superheated states.

The code is a 3 digit integer that represents the three out of eight properties that are going to be used as inputs (In1, In2, In3). So the code ‘123’ will represent that input of ‘Temperature’, ‘Pressure’, and ‘Ammonia mass fraction’ is required for the calculation of the remaining thermodynamic properties. The correlations in NH_3H_2O are from:

Ibrahim, O.M., Klein, S.A., ‘‘Thermodynamic Properties of Ammonia-Water Mixtures’’, ASHRAE Trans.: Symposia, 21, 2, 1495 (1993).

There is another way of getting the thermodynamic properties of ammonia-water mixture which is relatively simple. It is:

$$a = out(NH3H2O, In1, In2, In3)$$

Whereas ‘a’ is any variable from the eight outputs in the function and the three inputs are required to calculate that respective variable. E.g. h=Enthalpy (NH3H2O, T=T[1], P=P[1], X=X[1]) will return the value of enthalpy of the ammonia-water mixture at the required temperature, pressure and mass fraction.

As also discussed earlier, the solver of EES does not consider the pressure drop inside the pipes and heat exchangers due to friction. Therefore the values of heat exchanged in the absorber, generator, condenser and evaporator are the theoretical values. For the real values, a value of efficiency (based on the real-time scenario) must be used.

4.5 Energy and Mass balances of a Fundamental Single Effect Ammonia-Water Absorption System

The fundamental single effect ammonia-water absorption chiller consists of only the basic components i.e. condenser, evaporator, absorber and generator along with the necessary solution pump and valves. The mass and energy balance across these components are as follows:

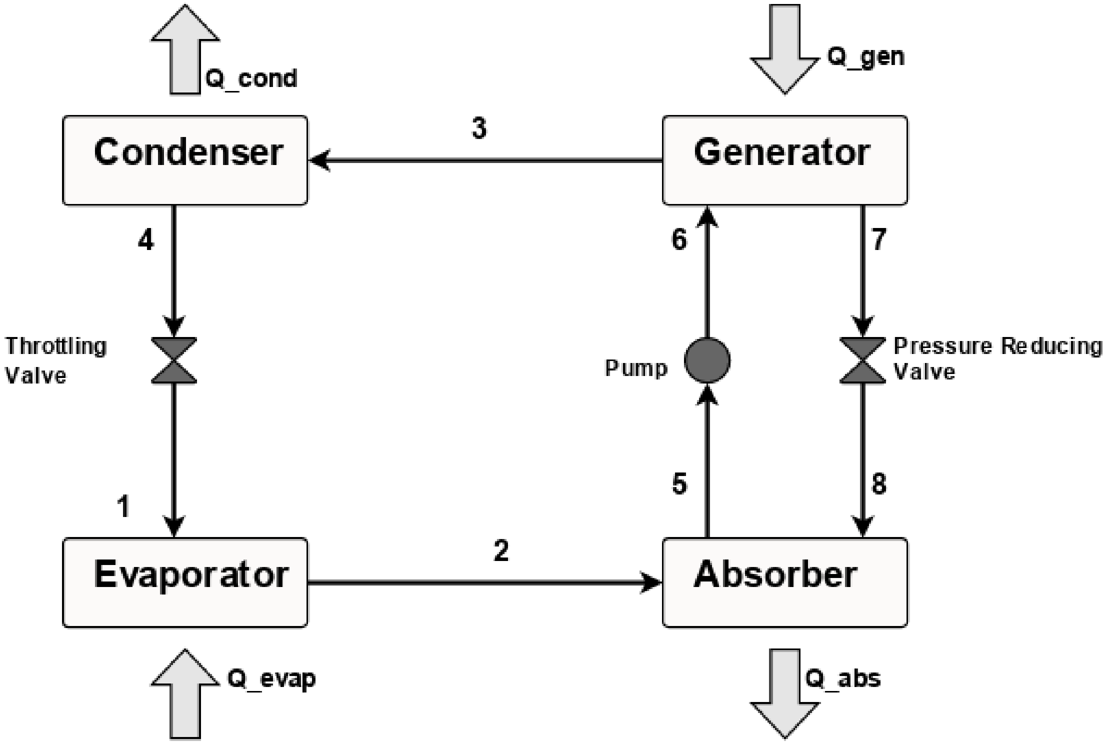

Figure 5 Basic Single Effect Ammonia-Water Absorption Refrigeration Cycle

4.5.1 Generator

$$m[6] = m[3] + m[7] \tag{vii}$$

$$m[6] * h[6] \; + \dot{Q}_{gen} = m[3] * h[3] + m[7] * h[7] \tag{viii}$$

$$m[6] * X[6] = m[3] * X[3] + m[7] * X[7] \tag{ix}$$

4.5.2 Absorber

$$m[2] + m[8] = m[5] \tag{x}$$

$$m[2] * h[2] + m[8] * h[8] = \dot{Q}_{abs} + m[5] * h[5] \tag{xi}$$

$$m[2] * X[2] + m[8] * X[8] = m[5] * X[5] \tag{xii}$$

4.5.3 Condenser

$$\dot{Q}_{cond} = m[3] * (h[3] - h[4]) \tag{xiii}$$

4.5.4 Evaporator

$$\dot{Q}_{evap} = m[2] * (h[2] - h[1]) \tag{xiv}$$

4.5.5 Pump

$$w_{pump} = \; m[5] * ([h[6] - h[5]) \tag{xv}$$

The enthalpy at point 6 is calculated using the enthalpy at point 5, i.e.:

$$h[6] = h[5] + \frac{P[6] - P[5]}{\rho[5] * \eta_p} \tag{xvi}$$

4.5.6 Throttling Valve

$$h[1] = h[4] \tag{xvii}$$

$$m[1] = m[4] \tag{xviii}$$

4.5.7 Pressure Reducing Valve

$$h[7] = h[8] \tag{xix}$$

$$m[7] = m[8] \tag{xx}$$

4.6 Mass and Energy Balance of Complete Model

The final model of the Single-effect ammonia-water absorption cycle consists of all the components from the basic single effect absorption cycle along with the solution heat exchanger

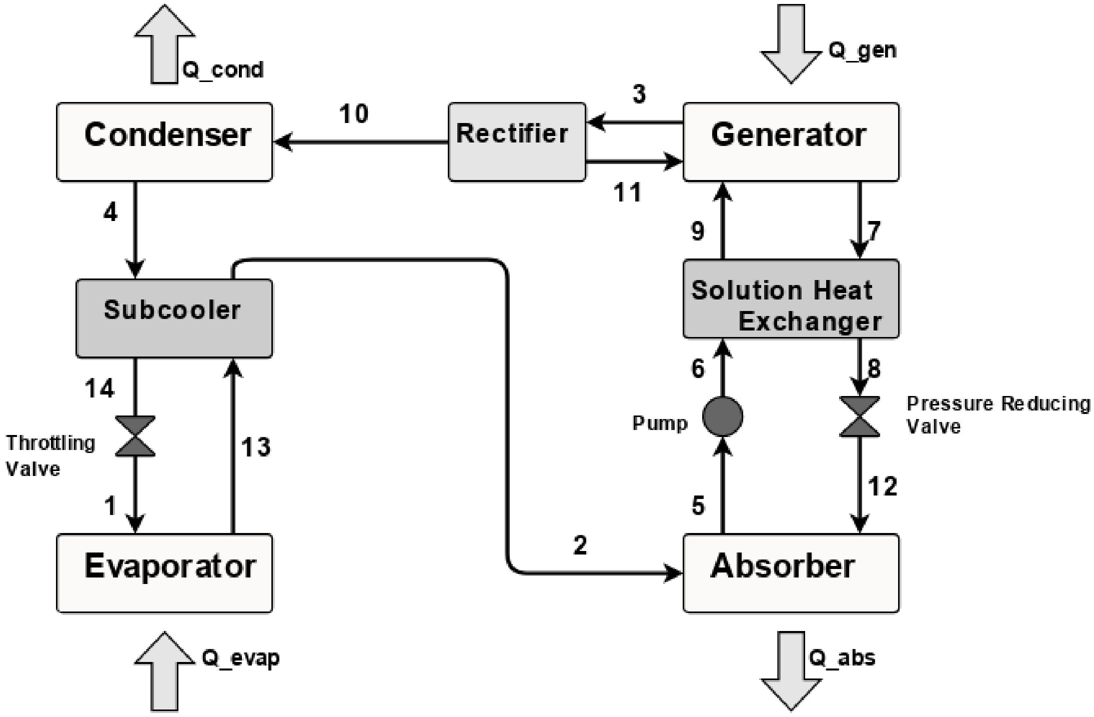

Figure 6: Final Model of Single Effect Ammonia-Water Absorption Refrigeration Cycle

between the absorber and generator and a sub-cooler heat exchanger between the condenser and the evaporator.

4.6.1 Generator

$$m[9] * h[9] + Q_{gen} + m[11] * h[11] = \dot{m}[3] * h[3] + m[7] * h[7] \tag{xxi}$$

$$m[9] + m[11] = m[7] + m[3] \tag{xxii}$$

$$m[9] * X[9] + m[11] * X[11] = m[3] * X[3] + m[7] * X[7] \tag{xxiii}$$

4.6.2 Absorber

$$m[2] * h[2] + m[12] * h[12] = \dot{Q}_{abs} + m[5] * h[5] \tag{xxiv}$$

$$m[2] * X[2] + m[12] * X[12] = m[5] * X[5] \tag{xxv}$$

$$m[2] + m[12] = m[5] \tag{xxvi}$$

4.6.3 Condenser

$$\dot{Q}_{cond} = m[4] * (h[10] - h[4]) \tag{xxvii}$$

4.6.4 Evaporator

$$\dot{Q}_{evap} = m[1] * (h[13] - h[1]) \tag{xxviii}$$

4.6.5 Rectifier

$$m[10] * h[10] + m[11] * h[11] + \dot{Q}_{rect} = m[3] * h[3] \tag{xxix}$$

$$m[10] * X[10] + m[11] * X[11] = m[3] * X[3] \tag{xxx}$$

$$m[10] + m[11] = m[3] \tag{xxxi}$$

4.6.6 Sub-cooler Heat Exchanger

In the sub-cooler heat exchanger, a method is adopted to calculate the heats exchanged between the two streams (rich and poor). For each stream, the heat rate is evaluated and then the minimum of these values is used to calculate the enthalpies of the streams leaving the sub-cooler from each side.

The heat rate on the rich side (vapour):

$$Q_{rs} = m[13] * (h2e - h[13]) \tag{xxxii}$$

The heat rate on the poor side (liquid):

$$Q_{ps} = m[4] * (h[4] - h14e) \tag{xxxiii}$$

As per the procedure, the minimum of these two values are calculated as:

$$Qmin = \boldsymbol{min}(Q_{rs}, Q_{ps}) \tag{xxxiv}$$

The enthalpies of the streams leaving the sub-cooler are then calculated. The enthalpies h2e and h14e are evaluated at Temperatures T[4] and T[13] respectively. Also, the effectiveness of the sub-cooler heat exchanger is used for the final heat exchanged in the sub-cooler.

$$h[2] = h[13] + \left(\frac{Qmin * eshx}{m[13]}\right) \tag{xxxv}$$

$$h[14] = h[4] - \left(\frac{Qmin * eshx}{m[4]}\right) \tag{xxxvi}$$

4.6.7 Solution Heat Exchanger

The same method which is used for the calculations of sub-cooler is used for evaluating the final heat exchanged in the solution heat exchanger and the enthalpies of the streams leaving the solution heat exchanger.

The heat rate on the rich side of the solution:

$$Q_{rs} = m[6] * (h9e - h[6]) \tag{xxxvii}$$

The heat rate on the poor side of the solution:

$$Q_{ps} = m[7] * (h[7] - h8e) \tag{xxxviii}$$

The minimum of these two values are calculated as per the procedure:

$$Qmin = \boldsymbol{min}(Q_{rs}, Q_{ps})$$

The enthalpies of the streams leaving the solution heat exchanger are then calculated. The enthalpies h9e and h8e are evaluated at Temperatures T[7] and T[6] respectively. Also, the effectiveness of the solution heat exchanger is used for the final heat exchanged in the solution heat exchanger.

$$h[9] = h[6] + Qmin1 * \frac{eff}{m[6]} \tag{xxxix}$$

$$h[8] = h[7] - Qmin1 * \frac{eff}{m[7]} \tag{xl}$$

4.6.8 Pump

$$w_{pump} = m[5] * ([h[6] - h[5]) \tag{xli}$$

The enthalpy at point 6 is calculated using the enthalpy at point 5, i.e.:

$$h[6] = h[5] + \frac{P[6] - P[5]}{\rho[5] * \eta_p} \tag{xlii}$$

4.6.9 Solution Heat Exchanger Valve

$$h[8] = h[12] \tag{xliii}$$

$$m[8] = m[12] \tag{xliv}$$

4.6.10 Throttling Valve

$$m[14] = m[1] \tag{xlv}$$

$$h[14] = h[1] \tag{xlvi}$$

4.6.11 Coefficient of Performance

For the evaluation of the performance of our cycle, the COP is calculated as follows:

$$COP = \frac{Q_{evap}}{W_p + Q_{gen}} \tag{xlvii}$$

The convergence of the solution is also validated by calculating the energy balance of heat transfer rates.

$$ConvgQ = Q_{gen} + Q_{evap} + W_p - (Q_{rect} + Q_{cond} + Q_{abs}) \tag{xlviii}$$

4.7 Verification of the Final Model

After formulating all the equations of mass and energy transfer for the given problem statement, the validation of the final model was necessary. For this purpose, one of the standard handbooks for absorption chiller and heat pumps [54] was utilized in which an example problem's data was fed to the developed model and the results were compared which were found to be identical. Now the developed model was ready to be used for the given case problem. The initial conditions for the given problem are discussed in the next section.

4.8 Data for Given Problem

After the mass and energy and balances, the initial conditions are imposed according to the working conditions. The heat rate at the generator is determined concerning the temperature of heat energy entering from the exhaust of diesel engine of the vehicle. The rectifier attached next to the generator ensures that the refrigerant with specified purity will enter into the condenser so that the performance of the system is not affected. Furthermore, the efficiencies of the heat exchangers (solution heat exchanger and sub-cooler) are used to replicate the conditions as close to the real scenario as possible.

4.8.1 Efficiencies

Efficiency of Pump= 0.62

Solution heat exchanger effectiveness (eff) = 0.9

Effectiveness of sub-cooler (eshx) = 0.99

4.8.2 Known Mass fraction

Concentration of Pure mixture entering in condenser X[10]=0.99

X[10]=X[4]=X[14]=X[1]=X[13]=X[2]

4.8.3 Specified Cooling Load

Heat transfer rate at the evaporator ($\dot{Q}_{evap} = 4\ kW$)

4.8.4 Given Temperatures (^{O}C)

Absorber Temperature = 40

Ambient Temperature = 40

Generator Temperature = 150-160

Cabin temperature T_C = 20

Temperature difference between the condenser and ambient (DT_cond) = 10

Temperature difference between the evaporator and the cabin (DT_Evap) = 10

4.8.5 Qualities

Absorber exit Q[5]=0

Generator exit to Solution Heat Exchanger Q[7]=0

Generator exit to rectifier Q[3]=1

Rectifier exit to Condenser Q[10]=1

Rectifier exit to Generator Q[11]=0

By carefully applying all the initial conditions on the final model of a single-stage ammonia-water absorption system with solution heat exchanger and sub-cooler, the results were simulated and analyzed which will be discussed in the next chapter in detail.

CHAPTER-5

Results and Analysis

The simulation results of the model were developed for single-stage ammonia-water vapour absorption refrigeration system. The output at the project data is calculated and analyzed further at the ranges of operating parameters.

5.1 Comparison of All Modeled Configurations

Firstly the most suitable configuration was found out before proceeding further to complete analysis. A graph was plotted between the coefficient of performance and generator temperature for all the possible configurations of our model. The generator temperature was varied between 100 to 160 ^{O}C. As it is quite evident from the figure that the highest COP was obtained for the model with both the sub-cooler and solution heat exchanger.

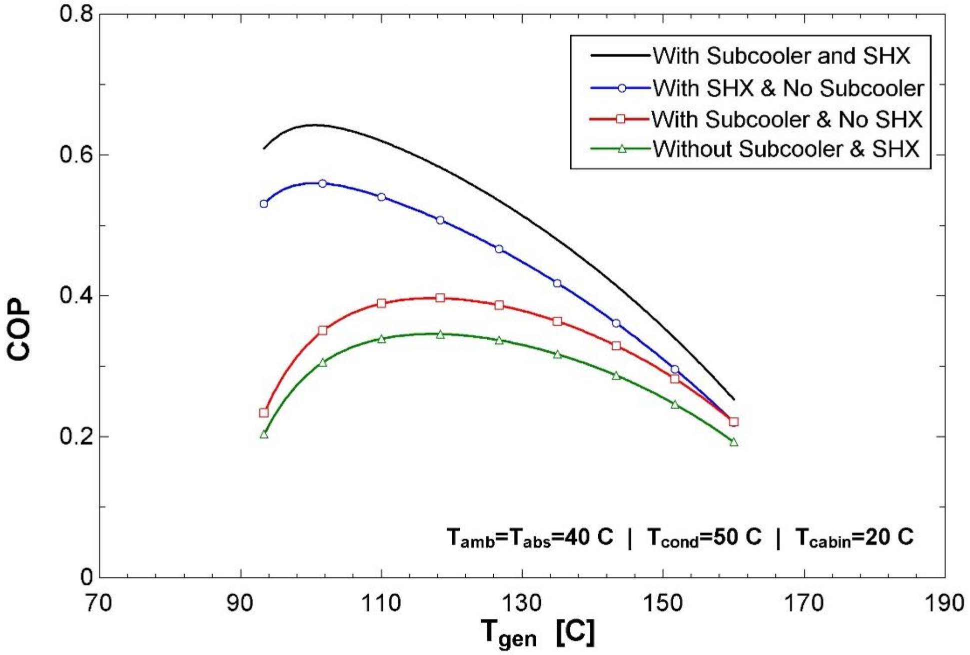

Figure 7: COP vs. Generator Temperature for all configurations of Single-Stage Ammonia-Water Absorption System

Table 1 shows the coefficient of performance for all configurations at $T_{gen}=150$ ^{O}C and effectiveness of 90% for sub-cooler and SHX.

Table 1: COP for all configurations of Single-Stage Ammonia Water Absorption System

Configuration	COP
With Sub-cooler and SHX	0.3556
With SHX and no Sub-cooler	0.3100
With Sub-Cooler and no SHX	0.2930
Without SHX and Sub-cooler	0.2555

The SHX and sub-cooler model was selected for the further process. As per the given conditions, the condenser temperature is fixed to 50 ^{O}C which is 10 ^{O}C higher than the ambient and the absorber temperature which is fixed to 40 ^{O}C and the cabin temperature (space to be cooled) is set to 20 ^{O}C (the evaporator temperature is 10 ^{O}C lower than this cabin temperature). Another point to ponder here is that including Solution Heat Exchanger and sub-cooler in the simple absorption cycle tends to increase the coefficient of performance which increases the effectiveness of the cycle by decreasing the required heat input to the generator and eventually decreasing the generator temperature. It indicates that even low energy waste heat can be utilized to run the absorption plant if sub-cooler and solution heat exchanger is incorporated in the cycle.

The analysis for the single-stage ammonia-water absorption system was performed for the model with both the sub-cooler and solution heat exchanger which will be discussed below.

For the analysis of the system, the performance parameters are evaluated by varying the generator, absorber and condenser temperature within a suitable range. The performance parameters considered are “Coefficient of Performance (COP)” and “Solution Circulation Ratio (f)”. The ranges of the operating parameters used for the analysis of the absorption system are shown in Table 1:

Table 2: Ranges of Operating Parameters

Operating Parameter	Temperature Ranges (^{O}C)
Generator Temperature	100-160
Absorber Temperature	20-40
Condenser Temperature	20-50
Evaporator Temperature	-10-10
Solution Heat Exchanger Effectiveness	0-1

The results calculated using the project data are shown below. The generator, absorber, ambient, condenser and cabin temperatures are 150 ^{O}C, 40 ^{O}C, 40 ^{O}C, 50 ^{O}C and 20 ^{O}C respectively whereas the pump efficiency is 0.8 and the effectiveness of the sub-cooler and solution heat exchanger is 0.9 each.

Table 3: Results of Modeled System at Project Data

State	Enthalpy (kJ/kg)	Mass Flow (kg/s)	Pressure (kPa)	Quality	Temperature (^{O}C)	Mass Fraction (X)
1	88.88	0.003533	600	0.04403	9.581	0.99
2	1366	0.003533	600	0.9913	46.48	0.99
3	1792	0.006032	2033	1.0000	150.0	0.80
4	233.9	0.003533	2033	-0.0010	50.00	0.99
5	-57.87	0.009014	600	0.0000	40.00	0.54
6	-55.63	0.009014	2033	-0.0010	40.25	0.54
7	498.5	0.005481	2033	0.0000	150.0	0.25
8	60.07	0.005481	2033	-0.0010	51.70	0.25
9	211	0.009014	2033	0.03001	89.92	0.54
10	1414	0.003533	2033	1.0000	84.05	0.99
11	164.2	0.002499	2033	0.0000	87.60	0.54
12	60.07	0.005481	600	-0.0010	51.98	0.25
13	1221	0.003533	600	0.9470	15.00	0.99
14	88.88	0.003533	2033	-0.0010	20.43	0.99

5.2 Variation of Performance Parameters with Generator Temperature

In this section, the performance parameters are varied with generator temperature at different absorber, condenser and cabin temperatures. The curves for the coefficient of performance and solution circulation ratio are plotted against a range of generator temperature. The variation of these curves is studied by changing the absorber, condenser and cabin temperature.

5.2.1 Variation with Absorber Temperature

Figure 8 shows the combined effect of generator and absorber temperature on the coefficient of performance. It can be seen that after a certain optimum generator temperature, the COP decreases with the increase in the generator temperature. Before the optimum temperature at which the COP is maximum, the trend is increasing. Absorber temperature also plays a crucial role in determining the COP of the system which decreases with the increase in the absorber temperature at a generator temperature lower than the optimum generator temperature and vice versa.

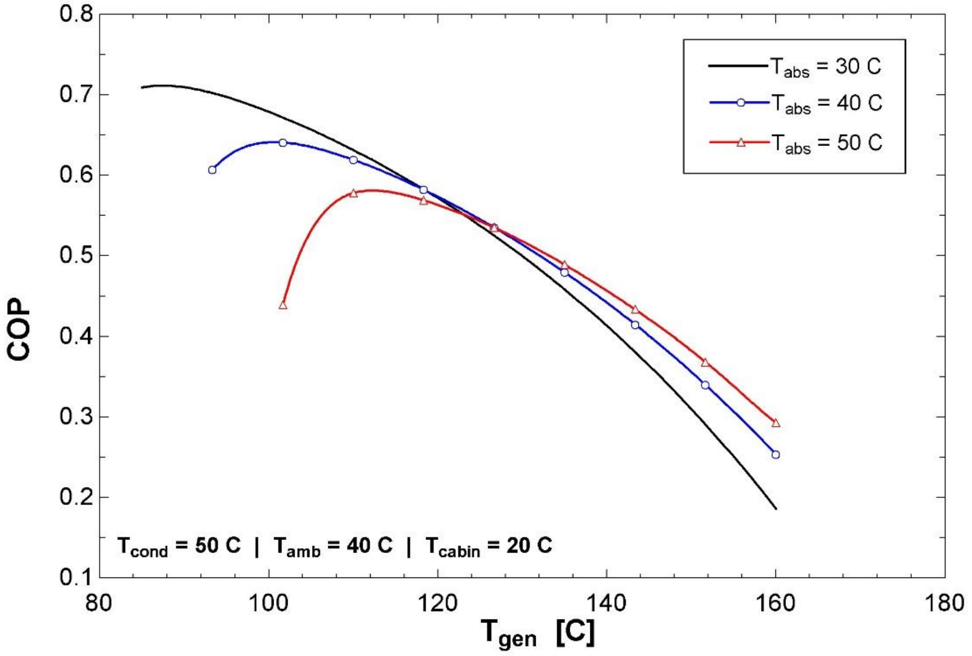

Figure 8: COP vs. Generator Temperature (at different absorber temperatures)

To determine the absorber and generator temperature at which the maximum COP occurs, a contour plot is created which determines the pairs of the absorber and generator temperature for the best performance of the cycle at constant ambient and condenser temperature. As shown in figure 9, the contour plot of COP explains that at constant ambient, condenser and cabin temperature, for higher generator temperatures, absorber temperature is kept low to yield maximum COP and for low generator temperature, high absorber temperature is required for the efficient coefficient of performance.

Figure 10 shows the effect of generator and absorber temperature on solution circulation ratio. It is quite evident that the circulation ratio approaches high values if generator temperature falls below a certain threshold at given conditions. It shows asymptotic behaviour which decreases with the increase in the generator temperature. Higher the generator temperature, lower the circulation ratio is. At constant condenser, ambient and cabin temperature, with an increase in the absorber temperature, and the circulation ratio increases.

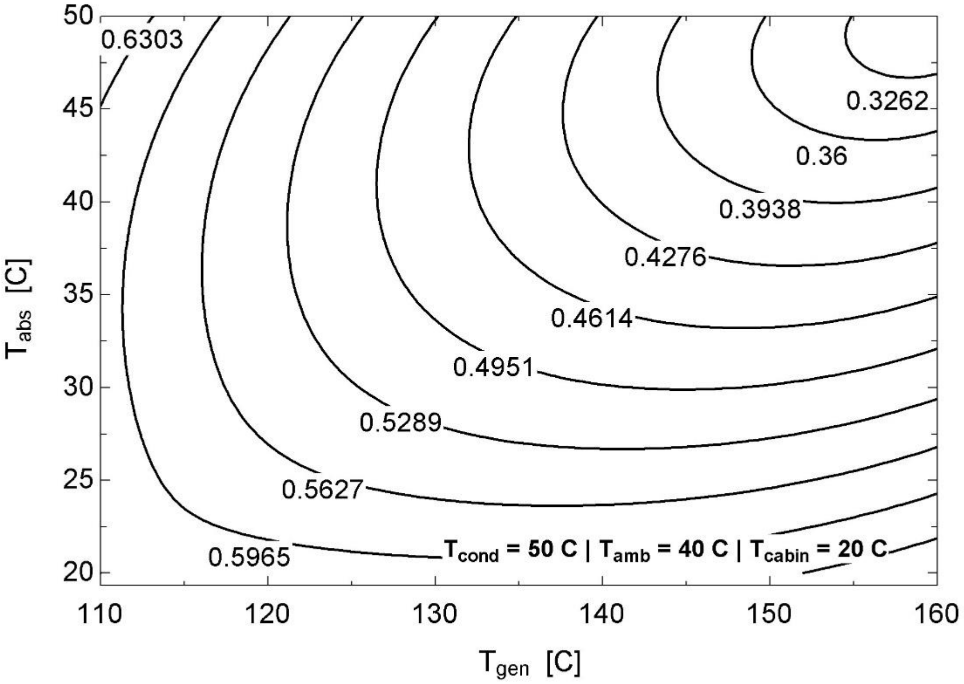

Figure 9: Absorber Temperature vs. Generator Temperature (Contour Plot of COP)

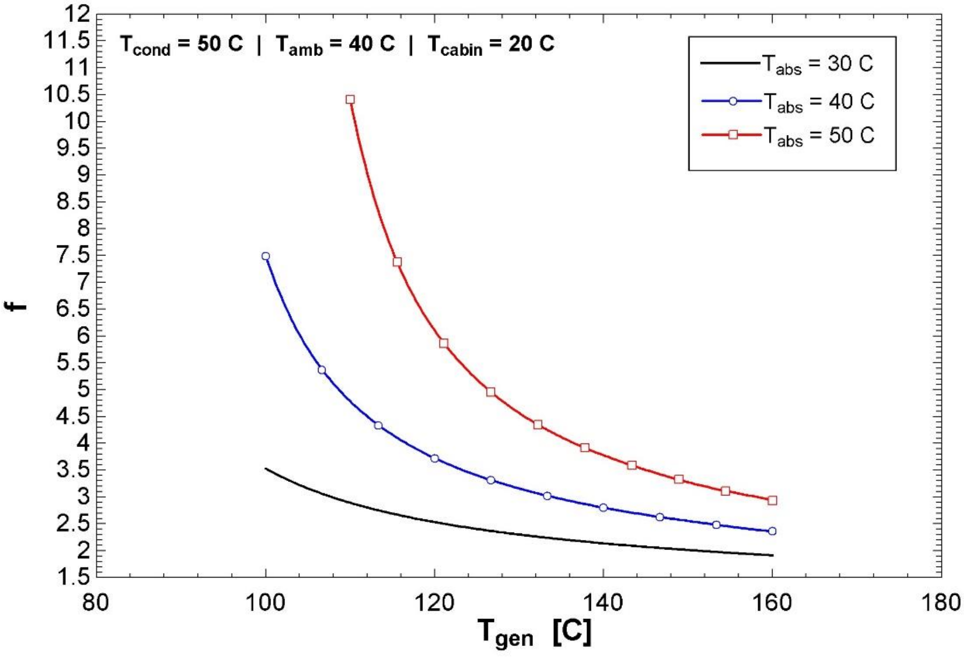

Figure 10: Solution Circulation Ratio vs. Generator Temperature (at different absorber temperatures)

5.2.2 Variation with Condenser Temperature

The variation of coefficient of performance against generator temperature can be seen in figure 11. The curves for different condenser temperature is plotted by keeping the ambient, absorber and cabin temperature constant. The trends depict that below a certain temperature of the generator, the COP increases for a given condenser temperature and drops afterwards. Similarly, for a given generator temperature, by increasing the condenser temperature, the value of COP increases up to a certain optimum point and after that, the COP decreases. Therefore, condenser temperature also affects the performance of the system in a way similar to the absorber temperature.

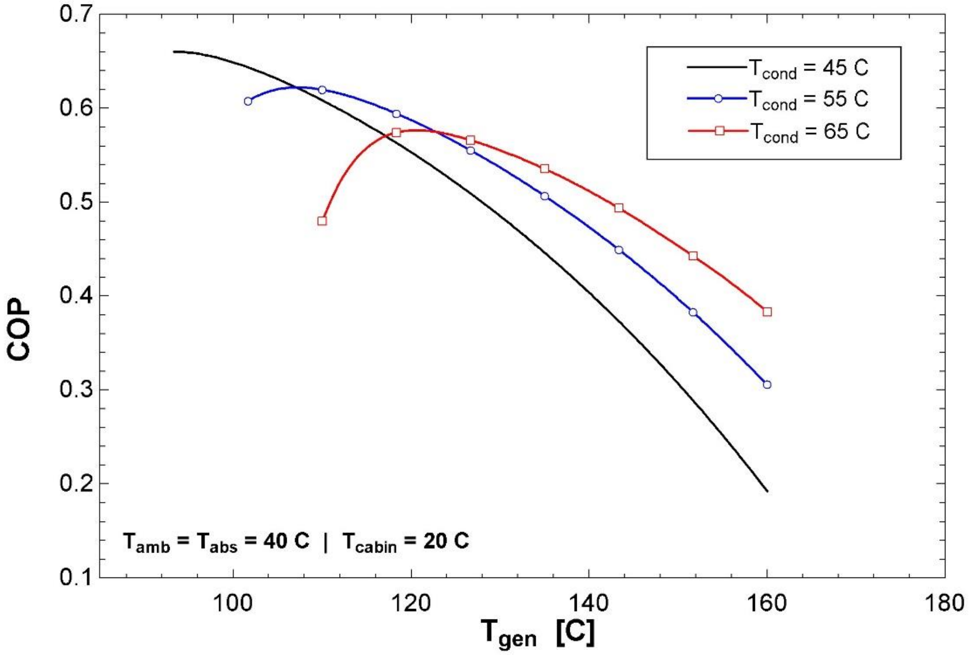

Figure 11: COP vs. Generator Temperature (at different condenser temperatures)

Figure 12 shows the variation of solution circulation ratio with generator temperature at different condenser temperatures. Below a certain temperature of the generator, the circulation ratio adopts a higher value. The same asymptotic behavior is shown by the curves as the one with varying absorber temperature, hence the value of circulation decreases with the increasing generator temperature beyond an optimum point. Therefore it can be said that upon increasing the generator temperature, the value of the solution circulation ratio decreases for a given condenser temperature.

Also, at a fixed generator, absorber, ambient and cabin temperature, the circulation ratio increases with the increase in condenser temperature.

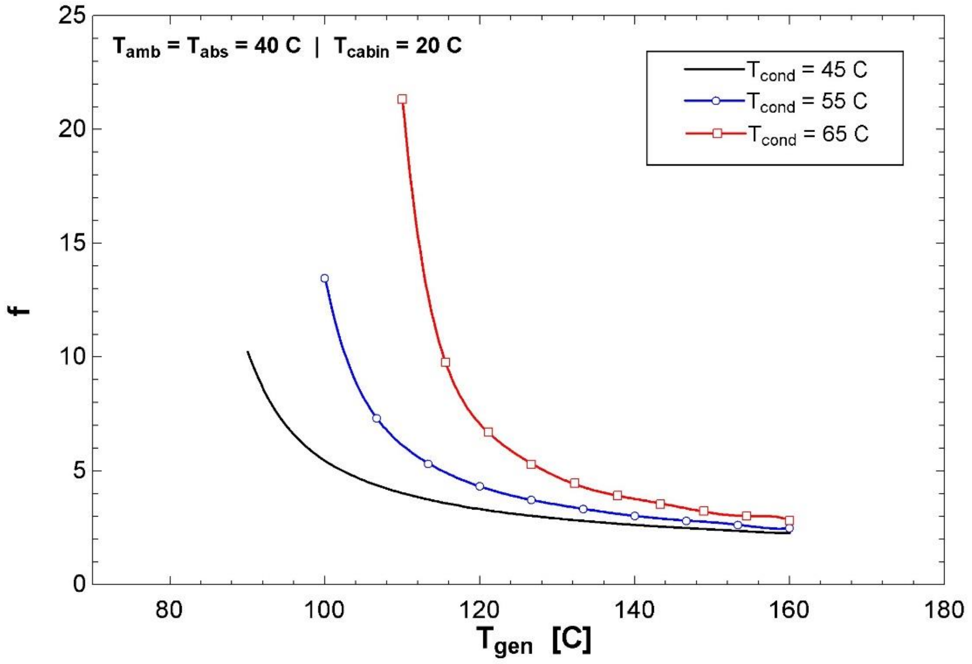

Figure 12: Solution Circulation Ratio vs. Generator Temperature (at different condenser temperatures)

5.2.3 Variation with Cabin Temperature

The variation of coefficient of performance is also plotted at different cabin temperatures. The cabin temperature decides the temperature of the evaporator (which is 10 ^{O}C lower than the cabin temperature). The curve is plotted by varying generator temperature at different cabin temperatures in figure 13. The COP shows an increasing trend with increasing generator temperature until a certain optimum point and decreases afterwards. For given conditions, at a fixed ambient, absorber, condenser and generator temperature (before the optimum point), the value of COP decreases with decreasing cabin temperature and vice versa after a certain generator temperature.

In the project data, cabin temperature is set at 20 ^{O}C for which the optimum generator temperature required is low, but the source of flue gases temperature is high which can be utilized to produce even low cabin temperature of up to 0 ^{O}C which is plotted in Figure 13 and it can be shown that increasing the generator temperature decreases the cabin environment temperature. But this may decrease the coefficient of performance because of high heat input required which is already being wasted if not consumed.

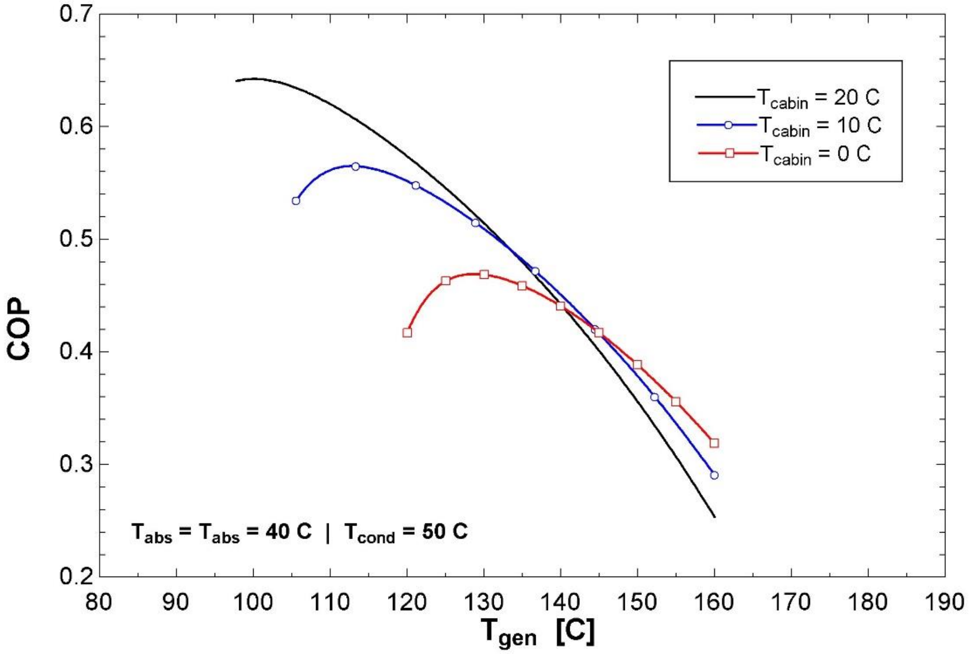

Figure 13: COP vs. Generator Temperature (at different cabin temperatures)

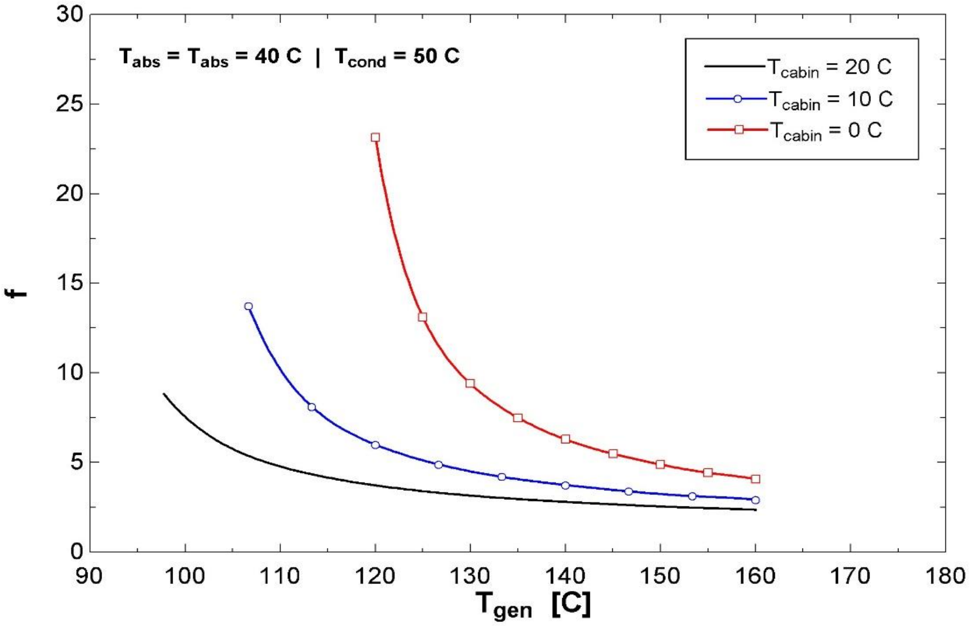

Figure 14: Solution Circulation Ratio vs. Generator Temperature (at different cabin temperatures)

The effect of changing generator temperature at different cabin temperatures for the solution circulation ratio is also plotted in figure 14. Solution circulation ratio decreases with the increasing generator temperature showing a similar asymptotic behaviour as in the previous cases. The increase in the cabin temperature at a fixed ambient, absorber and condenser temperature causes the circulation ratio to decrease, however, the rate of decrease is higher at lower cabin temperatures.

5.3 Variation of Performance Parameters with Absorber Temperature

The selection of absorber temperature is a critical choice as discussed in the previous section. The contour plot also facilitates the deciding an optimum value for absorber temperature. In this section, the performance parameters (COP and Solution Circulation Ratio) are plotted against a range of absorber temperature at different condenser temperatures. These curves are plotted at two fixed temperatures of generator to have a detailed insight into the variations.

5.3.1 Variation at a Higher Generator Temperature

The curves for the coefficient of performance and solution circulation ratio are plotted against a range of absorber temperature (20 ^{O}C-40 ^{O}C) at a fixed generator temperature of 150 ^{O}C and a fixed ambient and absorber temperature of 40 ^{O}C with different condenser temperatures as shown in figure 15 and figure 16.

In figure 15, the coefficient of performance increases with the increase in the absorber temperature at any condenser temperature (with the given conditions), the rate of increase, however, is higher in the case of higher condenser temperature. For a given absorber temperature, the value of COP increases with the increase in the condenser temperature.

Figure 16 shows the effect of these conditions on the solution circulation ratio. It is evident that for any condenser temperature, the solution circulation ratio increases with the increase in absorber temperature in the range (considering the other conditions). For a given absorber temperature, the value of circulation ratio is higher for the condenser working at an elevated temperature.

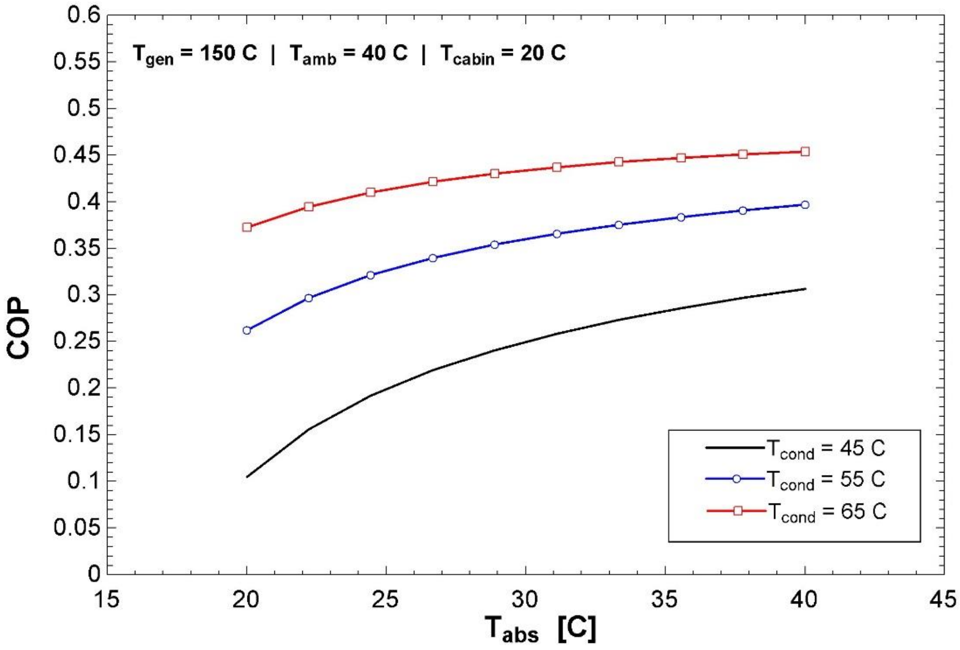

Figure 15: COP vs. Absorber Temperature (at T_{gen}=150 C and different condenser temperatures)

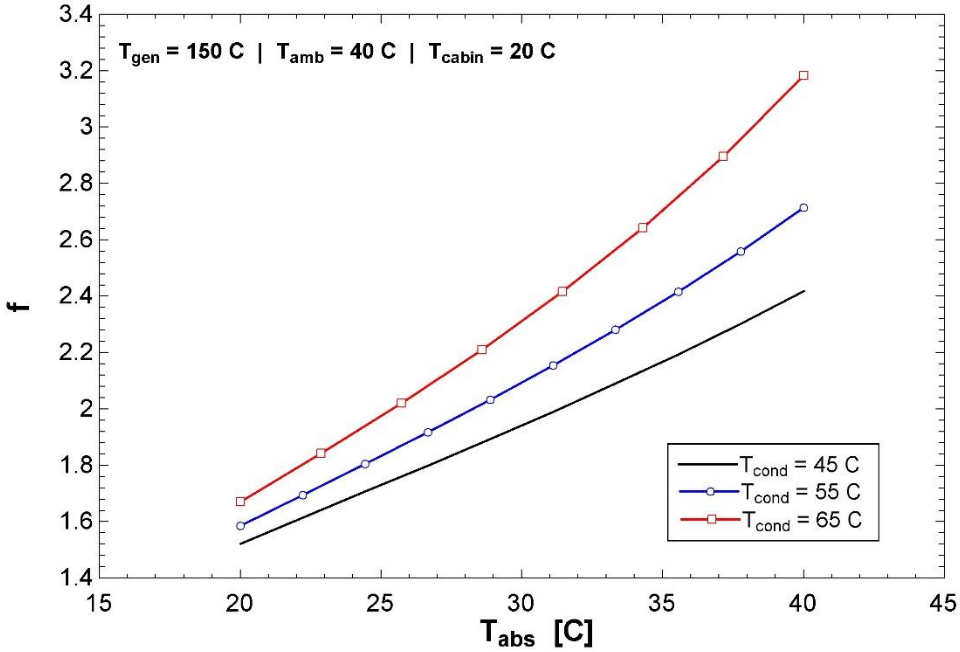

Figure 16: Solution Circulation Ratio vs. Absorber Temperature (at T_{gen} =150 C and different condenser temperatures)

5.3.2 Variation at a Lower Generator Temperature

The effect of absorber temperature on the coefficient of performance and solution circulation ratio is plotted at different condenser temperatures by keeping the generator temperature to a lower value i.e. 110 ^{O}C. The cabin and the ambient temperatures are set to 20 ^{O}C and 40 ^{O}C respectively as shown in figure 17 and figure 18.

In figure 17, the coefficient of performance is plotted against absorber temperature. It can be seen that at this generator temperature, the curve for condenser temperature 45 ^{O}C is almost a horizontal line showing no effect of absorber temperature. However, at higher condenser temperatures, this effect is quite visible as COP decreases with the increase in absorber temperature. The curve for condenser temperature 65 ^{O}C is a clear example of the fact that after a certain temperature for the given conditions, the COP decreases drastically. Therefore, at given conditions, there is a certain range of condenser temperature for better COP of the system.

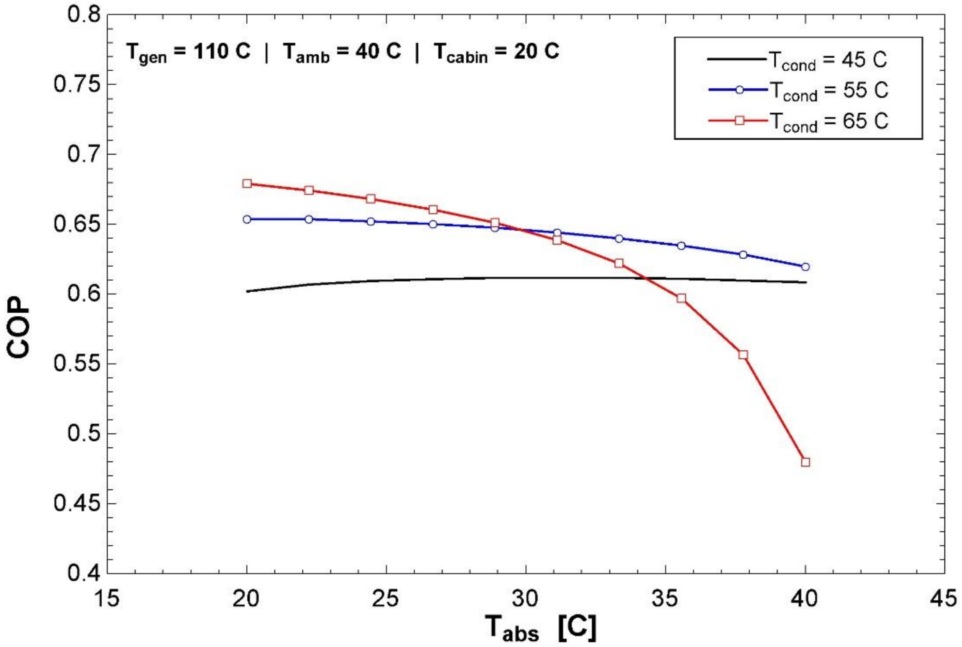

Figure 17: COP vs. Absorber Temperature (at T_{gen}=110 C and different Condenser Temperatures)

The behaviour of the curve plotted between the solution circulation ratio and the absorber temperature at a lower temperature in figure 18 is similar to the curve plotted for a higher temperature of the generator. With the increase in absorber temperature, the circulation ratio increases. However the values of circulation ratio are higher as compared to that with higher generator temperature. For a given absorber temperature in this range, the circulation ratio increases with the increase in condenser temperature. The rate of increase is higher for the system with a higher condenser temperature.

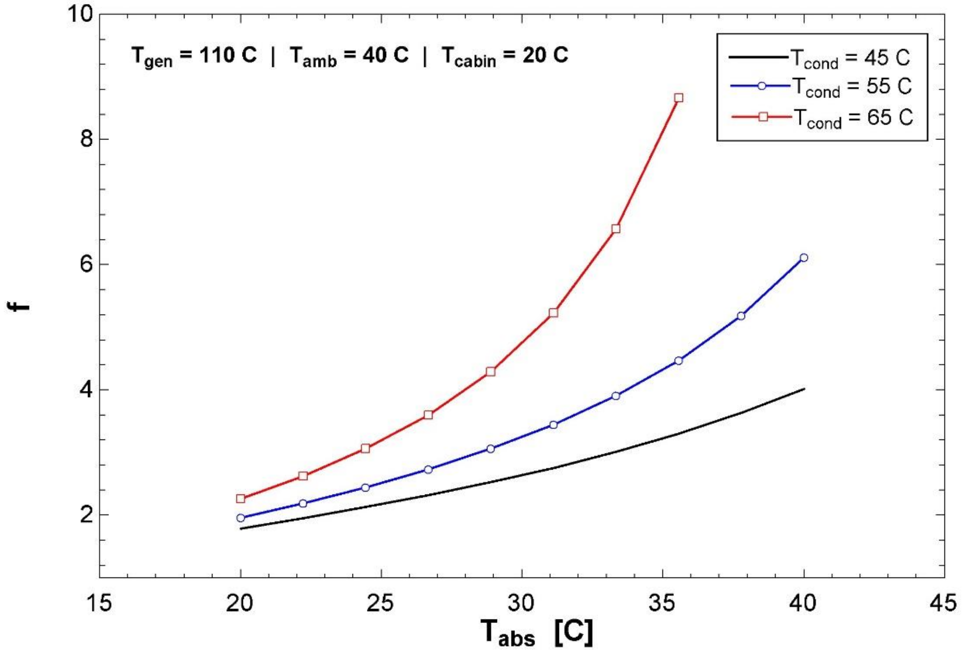

Figure 18: Solution Circulation Ratio vs. Absorber Temperature (at T_{gen}=110 C and different condenser temperatures)

Comparing the solution circulation ratio graph in figure 16 and figure 18, it is to be noted that at higher generator temperature, the circulation ratio increases at a lower rate concerning the absorber temperature than at lower generator temperature where circulation ratios are higher and even reach up to 10 for certain conditions.

5.4 Variation of COP with Effectiveness of SHX

The coefficient of performance also depends on the effectiveness of the solution heat exchanger and sub-cooler. The variation of COP versus the effectiveness of SHX is shown in figure 19 which shows that COP is improved by increasing the effectiveness of SHX and is most favourable at lower generator temperature at the given conditions of absorber, cabin and condenser. COP can also be improved by increasing the effectiveness of the sub-cooler. It is to be noted that when the effectiveness of SHX is varied, the effectiveness of sub-cooler is kept constant and vice versa. The effect of SHX is of more importance as the rate with which COP increases with variation in the effectiveness of SHX is more than in sub-cooler.

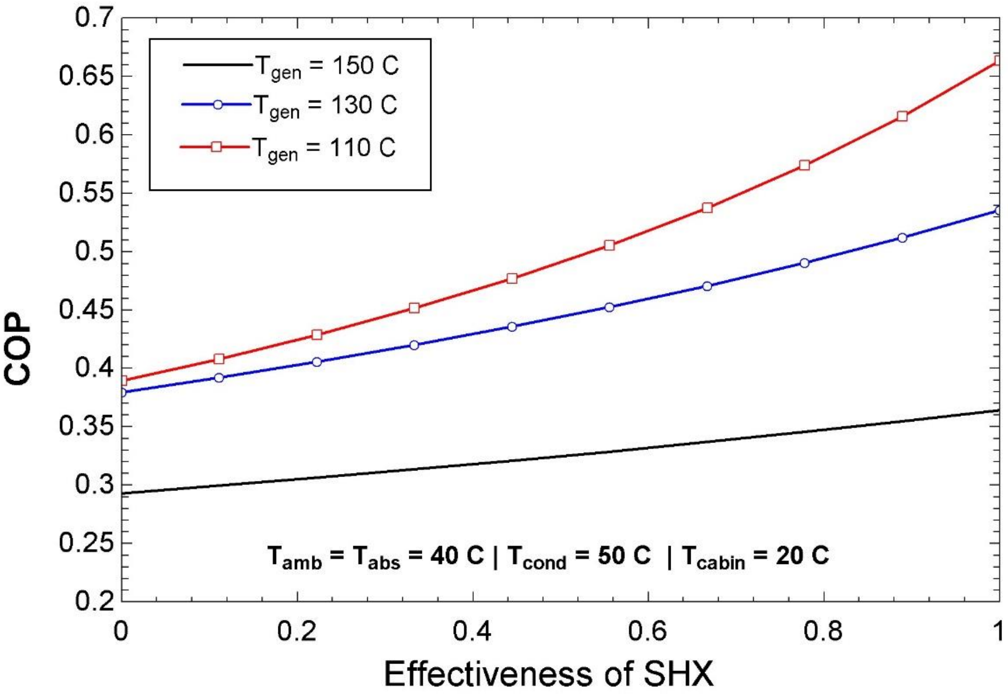

Figure 19: COP vs Effectiveness of SHX (at different generator temperatures)

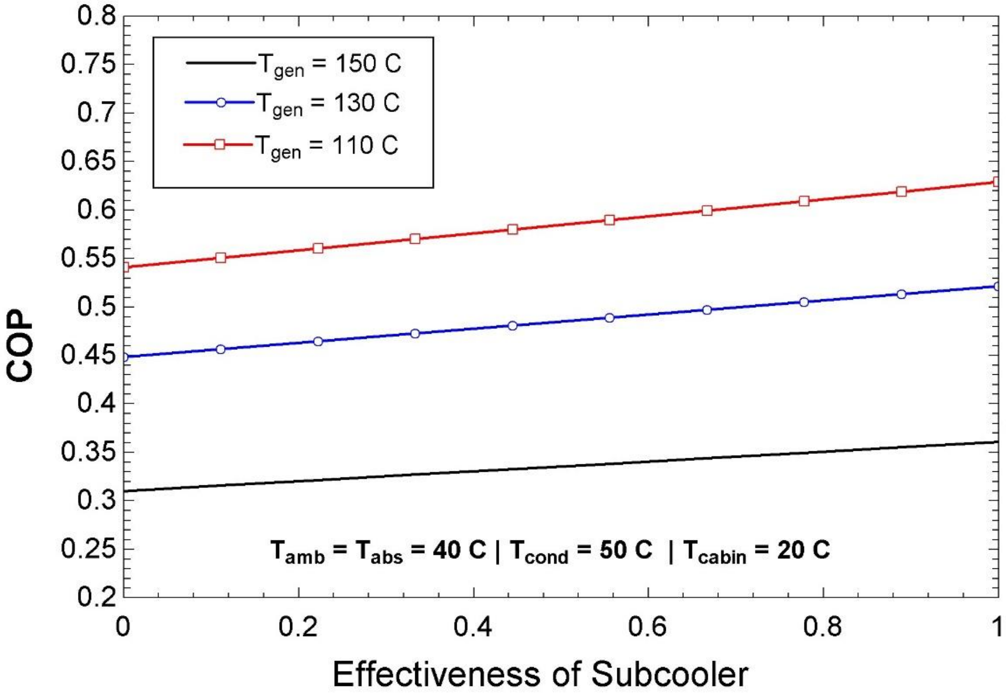

Figure 20: COP versus Effectiveness of Sub-cooler (at different generator temperatures)

5.5 Improvement of Operating Conditions

The simulation of the single-stage ammonia-water absorption system with sub-cooler and solution heat exchanger with the given operating conditions of generator temperature 150 ^{O}C, ambient and absorber temperature 40 ^{O}C, condenser temperature 50 ^{O}C, cabin temperature 20 ^{O}C, the effectiveness of sub-cooler and solution heat exchanger 0.9 and pump efficiency 0.8, the coefficient of performance of 0.36 and solution circulation ratio of 2.55 is achieved. These performance parameters can be improved further by adjusting the ranges of operating parameters. From the analysis of the results in the previous chapter, the following improvements can be made in the operating conditions:

- For the given conditions, the generator temperature of 100 ^{O}C (or close to this value) yields a better coefficient of performance (0.64) and solution circulation ratio (7.55)
- For the given conditions at a constant generator temperature of 150 ^{O}C, the absorber temperature higher than 40 ^{O}C would result in a higher coefficient of performance and solution circulation ratio. For example, at T_{abs}=65 ^{O}C, the COP and circulation ratio is 0.40 and 5.2 respectively.

- For the given conditions at a constant generator temperature of 150 ^{O}C, the condenser temperature higher than 50 ^{O}C would yield a higher coefficient of performance and solution circulation ratio. For example, at T_{cond}=70 ^{O}C, the COP and circulation ratio is 0.47 and 3.54 respectively.
- For the given conditions, the cabin temperature lower than 20 ^{O}C (evaporator temperature lower than 10 ^{O}C) would increase the coefficient of performance and solution circulation ratio. For example at T_{cabin}=0 ^{O}C (T_{evap}=-10 ^{O}C), the COP and circulation ratio is 0.38 and 4.89. However, the values only slightly deviate from the previous results.

An important point to be noted here is that to achieve maximum performance, there is a range of optimum operating conditions. If any of the operating parameters is shifted above or below this range, the new optimum range is created and then the rest of the parameters must be adjusted according to this new range to achieve the maximum performance of the system. As in our case, there is an optimum range of parameters of generator, absorber and condenser temperature for absorption cycle of the given fluid, but in practical operation, the operating conditions are varied, the generator temperature is higher due to which operating conditions are changed and a new range of absorber and condenser temperature is obtained to achieve the desired cabin temperature.

5.6 Sensitivity Analysis

In EES, the Min/Max command is provided to find the values of independent variables that will maximize or minimize the required output variable or objective function during the optimization process. Moreover the professional version of EES also enables the user to conduct sensitivity analysis with the optimization of objective function. In this analysis, not only the optimum values of independent variables are calculated but also the independent variables which are most sensitive for the variation of objective function are evaluated.

In our case, the objective function is coefficient of performance. The primary independent variables are temperature of generator, absorber and condenser. As explained above, these three parameters considerably affect the values of COP. The temperature of cabin is set to 20 ^{O}C as per the requirement and the ambient temperature is considered to be 40 ^{O}C. For the optimization of objective function, firstly the maximization of COP is selected from the Min/Max command and the temperatures of generator, absorber and change in condenser are selected to be the independent variables. After selecting the ‘Variable Metric Method’ for optimization, the bounds for the selected independent variables are entered. For sensitivity analysis, the independent variables are varied ±10% from their optimum value to evaluate the correlation between the COP and variation in independent variables.

Table 4 Sensitivity Analysis Table

Vary	**-10%**	**COP**	**Optimum**	**+10%**	**COP**
DT_cond	16.18	0.7584	17.68	19.18	0.7583
T_abs	20	0.7587	20	22	0.7477
T_gen	90	0.7587	90	97	0.7305

It can be seen in Table 4, all the independent variables are varied over the range of ±10% to check the effect in COP value. It was observed, COP was sensitive to generator temperature the most. The maximum value of COP at the given condition is 0.7587 at optimum values of independent variables.

CHAPTER-6

Conclusion and Recommendation

A vapour absorption refrigeration cycle using an ammonia-water mixture as the working fluid pair has been examined in this work. The system is integrated with a truck diesel engine which delivers waste heat at a constant temperature range. The system has been theoretically investigated by evaluating their performance using an energy balance model produced according to the first law of thermodynamics applied to each design component of the given system. The calculations are performed on the various sets of operating conditions to check the performance of the cycle and the results are in agreement with the literature. The mathematical model is dynamic which can be changed as per different operating and environmental conditions which yields different performance parameters from which most optimized parameters are selected to increase the effectiveness of the absorption system. The numerical model is iterative and is linked to a set of equations of state for each component delivering the consistent calculation of the thermodynamic properties of the ammonia-water solution used. The thermodynamic analysis of this absorption cycle is performed for several operating conditions by varying generator, condenser, absorber and cabin temperatures. It is found that the high performance of the cycle is obtained at low generator temperature and high absorber temperature or vice versa. Furthermore, it was found that there was an optimum generator temperature if other conditions are kept constant for which maximum COP is obtained. Also, the increase in the solution heat exchanger effectiveness increases the coefficient of performance with no effect on the circulation ratio.

6.1 Future Recommendations

The present numerical model of a single-stage ammonia-water absorption refrigeration system exhibits promising results. However, the performance of the system can be further analyzed and improved in the following ways:

1) The performance of the system can be analyzed by using Lithium Bromide-Water mixture as working fluid pair. Moreover, a variety of working fluids such as Ammonia-Sodium thiocyanate and Ammonia-Lithium Nitrate mixtures are available for absorption systems where the former is effective at higher generator temperatures and the latter improves the

COP at higher absorber temperature proving to be a possible alternate in terms of COP as compared to the ammonia-water solution [55]

2) Advanced designing of the system e.g. multi-stage systems and multi-effects system etc. can be done for enhanced performance. However, the design complexities and dimensional constraints must be taken into account.
3) The VCHPs (Variable Conductance Heat Pipes) can be incorporated in the designing of the system which will be able to bring down the heat-source temperature to achieve high performance.

References

[1] D. F. Lucien Borel, Thermodynamics and Energy Systems Analysis: From Energy to Exergy, Lausanne: EPFL Press, 2010.

[2] W. Gosney, Principles of Refrigeration, Cambridge University Press, 1982.

[3] S. A. S. C. Pongsid Srikhirin, "A Review of Absorption Refrigeration Technologies," Renewable & Sustainable Energy Reviews, vol. 5, no. Elsevier Science Limited, pp. 343-372, 2001.

[4] H. Perez-Blanco, "Absorption Heat Pump Performance for Different Types of Solutions," International Journal of Refrigeration, vol. 7, no. Butterworth & Co Limited, pp. 115-122, 1984.

[5] T. B. P. Holmberg, "Alternative Working Fluid in Heat Transformers," ASHRAE Transactions, vol. 96, pp. 1582-9, 1990.

[6] G. J. Z. T. Marciss RA, "Absorption Fluid Data Survey: Final Report on Worldwide Data," Office of Scientific and Technical Information, chicago, United States, 1988.

[7] R. R. S. A. K. Keith E. Herold, Absorption Chillers and Heat Pumps, Florida, USA: CRC Press Inc., 1996.

[8] K. B. A.Sathiamourtty, "Thermodynamic Analysis of Vapour Absorption Refrigeration Cycle Using Different Working Fluids," International Journal of Applied Engineering Research, vol. 15, no. Research India Publications, pp. 911-917, 2020.

[9] D. A. Subhash Kumar, "Comparative Study on Performance Analysis of Vapour Absorption Refrigeration System Using Various Refrigerants," IPASJ International Journal of Mechanical Engineering (IIJME), vol. 3, 2015.

[10] R. Anirban Sur, "Review On Solar Adsorption Refrigeration Cycle," International Journal of Mechanical Engineering and Technology (IJMET), vol. 1, pp. 190-226, 2010.

[11] S. M.U.Siddiqui, "A Review of Solar Powered Absorption Systems," Renewable and Sustainable Energy Reviews, vol. 42, pp. 93-115, 2015.

[12] B. W. W. S. X. L. Wei Wu, "An Overview of Ammonia-based Absorption Chillers and Heat Pumps," Renewable and Sustainable Energy Reviews, vol. 31, pp. 681-707, 2014.

[13] Y. D. Y. J. D.C.Wang, "A Review on Adsorption Refrigeration Technology and Adsorption Deterioration in Physical Adsorption Systems," Renewable and Sustainable Energy Reviews, vol. 14, pp. 344-353, 2010.

[14] M. M. R. I. T. M.R.Anisur, "Curbing Global Warming with Phase Change Materials for Energy Storage," Renewable and Sustainable Energy Reviews, vol. 18, pp. 23-30, 2013.

[15] A. H.Z.Hassan, "A Review on Solar Cold Production Through Absorption Technology," Renewable and Sustainable Energy Reviews, vol. 16, pp. 5331-5348, 2012.

[16] P. J. Tarik Shaikh, "Review of Solar Absorption Refrigeration System Using LiBr-Water and Simulate the Performance of the System," International Journal of Advanced Engineering Research and Studies, pp. 57-60, 2013.

[17] X. W. A. K. H. A. I. Rasoul Nikbakhti, "Absorption Cooling Systems-Review of Various Techniques for Energy Performance Enhancement," Alexandria Engineering Journal, vol. 59, pp. 707-738, 2020.

[18] H.-M. Henning, "Solar Assisted Air Conditioning of Buildings-An Overview," Applied Thermal Engineering, vol. 27, pp. 1734-1749, 2007.

[19] R. A. N.Ben Ezzine, "A Numerical Investigation of a Diffusion-Absorption Refrigeration Cycle based on R124-DMAC Mixture for Solar Cooling," Energy, vol. 35, pp. 1874-1883, 2010.

[20] L. F. S. Z. Jian Sun, "A Review of Working Fluids of Absorption Cycles," Renewable and Sustainable Energy Reviews, vol. 16, pp. 1899-1906, 2012.

[21] S. N. Arshi Banu P.S, "Feasibility Studies of Single Effect H2O-LiBr+LiI+LiNO3+LiCl Vapour Absorption Cooling System for Solar Based Applications," Journal of Chemical and Pharmaceutical Sciences, no. 12, pp. 1-7, 2017.

[22] J. C. B. A. C. Jerko M.Labus, "Review on Absorption Technology with Emphasis on Small Capacity Absorption Machines," Thermal Science, vol. 17, no. 3, pp. 739-762, 2013.

[23] A. Y. Al-Zubaydi, "Solar Air Conditioning and Refrigeration with Absorption Chillers Technology in Australia-An Overview on Researches and Applications," Journal of Advanced Science and Engineering Research, vol. 1, pp. 23-41, 2011.

[24] J. A.Manrique, "Thermal Performance of an Ammonia-Water Refrigeration System," International Communications in Heat and Mass Transfer, vol. 18, pp. 779-789, 1991.

[25] S. O. N. M.I.Karamangil, "A Simulation Study of Performance Evaluation of Single-Stage Absorption Refrigeration System Using Conventional Working Fluids and Alternatives," Renewable and Sustainable Energy Reviews, vol. 14, pp. 1969-1978, 2010.

[26] M. P. A. M.Izquierdo, "Crystallization as a Limit to Develop Solar Air-Cooled LiBr-H2O Absorption Systems Using Low-grade Heat," Solar Energy Materials and Solar Cells, vol. 81, pp. 205-216, 2004.

[27] O. A. P. K. E. A. Kai Wang, "State-of-the-art Review on Crystallization Control Technology for Water/LiBr Absorption Heat Pumps," International Journal of Refrigeration, vol. 34, pp. 1325-1337, 2011.

[28] X. X. Y. J. Sheng Wang, "Experimental Investigation of Two-phase Flow Charateristics of LiBr/H2O Solution Through Orifice Plates in Vacuum Environment," Internation Journal of Refrigeration , vol. 38, pp. 267-274, 2014.

[29] I. C. C. T. Evangelos Bellos, "Yearly Investigation of a Solar-driven Absorption Refrigeration System with Ammonia-Water Absorption Pair," Thermal Science and Engineering Progress, vol. 23, pp. 1008-85, 2021.

[30] S. M. H. L. C.-G. J. R. S. Andre Aleixo Manzela, "Using Enginer Exhaust Gas as Energy Source for an Absorption Refrigeration System," Applied Energy, vol. 87, pp. 1141-1148, 2010.

[31] J. G. G Vicatos, "A Car Air-Conditioning System Based on an Absorption Refrigeration Cycle Using Energy from Exhaust Gas of an Internal Combustion Engine," Journal of Energy in Southern Africa, vol. 19, no. 4, pp. 6-11, 2008.

[32] W. D. M. J.Koehler, "Absorption Refrigeration System for Mobile Applications Utilizing Exhaust Gases," Heat and Mass Transfer, vol. 32, no. 5, pp. 333-340, 1997.

[33] F.Meunier, "Adsorptive Cooling: A Clean Technology," Clean Prod Processes, vol. 3, pp. 8-20, 2001.

[34] R. Y. Y. J. Z. S.Jiangzhou, "Locomotive Driver Cabin Adsorption Air-Conditioner," Renewable Energy, vol. 28, pp. 1659-1670, 2003.

[35] J. C. M. L. Z. C. Y. Z. K. Y. Feng Qin, "Developement o a Metal Hydride Refrigeration System as an Exhaust Gas-Driven Automobile Air Conditioner," Renewable Energy, vol. 32, pp. 2034-2052, 2007.

[36] J. R. Z. S. Y.Huangfu, "Development of an Experimental Prototype of an Integrated Thermal Management Controller for Internal-Combustion-Engine-Based Cogeneration Systems," Applied Energy, vol. 84, pp. 1356-1373, 2007.

[37] J. S.Li, "Theoretical Research of a Silica Gel-Water Adsorption Chiller in a Micro Combined Cooling, Heating and Power (CCHP) System," Applied Energy, vol. 86, pp. 958-967, 2009.

[38] H. L. Y. H. R. R. H.-H. C. Tao Cao, "Performance Investigation of Engine Waste Heat Powered Absorption Cycle Cooling System for Shipboard Applications," Applied Thermal Engineering, vol. 90, pp. 820-830, 2015.

[39] M. A. Ersoz, "Investigation the Effects of Different Heat Inputs Supplied to the Generator on the Energy Performance in Diffusion Absorption Refrigeration Systems," International Journal of Refrigeration, vol. 54, pp. 10-21, 2015.

[40] J. W. Jialong Wang, "Investigation of a Mixed Effect Absorption Chiller Powered by Jacket Water and Exhaust Gas Waste Heat of Internal Combustion Engine," International Journal of Refrigeration, vol. 50, pp. 193-206, 2015.

[41] M. S. S. Ismail Hilali, "An Application of Enginer Exhaust Gas Driven Cooling System in Automobile Air-Conditioning System," Journal of Thermal Science and Technology, vol. 35, no. 1, pp. 27-34, 2015.

[42] I. Horuz, "An Alternative Road Transport Refrigeration," Journal of Engineering and Environmental Science, vol. 22, pp. 211-222, 1998.

[43] I.Horuz, "Vapor Absorption Refrigeration in Road Transport Vehicles," Journal of Energy Engineering, vol. 125, no. 2, pp. 48-58, 1999.

[44] A. Sohail Bux, "Natural Refrigerants Based Automobile Air Conditioning System," International Journal of Emerging Science and Engineering (IJESE), vol. 2, no. 7, pp. 32-38, 2014.

[45] J. V. Y.Lethwala, "Development of Vapour Absorption Refrigeration System in Automotive Trucks for Cold Storage," International Journal of Creative Research Thoughts (IJCRT), vol. 6, no. 1, pp. 637-645, 2018.

[46] S. d. M. H. L. C.-G. C. B. M. J. R. S. Andre Aleixo Manzela, "An Experimental Comparison Between LPG and Engine Exhaust Gas as Energy Source for an Absorption Refrigeration System," International Journal of Energy Research, vol. 36, pp. 820-828, 2012.

[47] S. R. J.V.Kaudinya, "The Feasibility of Air-Conditioning By Exhaust Gas," International Journal of Ambient Energy, vol. 9, no. 4, pp. 185-190, 1988.

[48] A. V. M. V. Jose Fernandez-Seara, "Heat Recovery System to Power an Onboard NH3-H2O Absorption Refrigeration Plant in Trawler Chiller Fishing Vessels," Applied Thermal Engineering, vol. 18, pp. 1189-1205, 1998.

[49] B. M.Mostafavi, "Thermodyanmic Analysis of Combined Diesel Engine and Absorption Refrigeration Unit-Naturally Aspirated Diesel Engine," Applied Thermal Engineering, vol. 17, no. 5, pp. 471-478, 1997.

[50] Y. E.-G. Ahmed Ouadha, "Integration of an Ammonia-Water Absorption Refrigeration System with A Marine Diesel Engine: A Thermodynamic Study," The 3rd International Conference on Sustainability Energy Information Technology (SEIT), vol. 19, pp. 754-761, 2013.

[51] R. K. D. Mukul Kumar, "Experimental Analysis of Absorption Refrigeration System Driven by Waste Heat of Diesel Engine Exhaust," Thermal Science, vol. 23, no. 1, pp. 149-157, 2019.

[52] S. S. Auritro Samanta, "Thermodynamic Analysis of a Cogeneration System Integrating Organic Rankine Cycle and Absorption Chiller Using a Diesel Engine Exhaust," in AIP Conference Proceedings 2341,030015, 2021.

[53] A. C. G.A.Longo, "Simulation of an Absorption Chiller Driven By The Heat Recovery on an Internal Combustion Engine," in International Refrigeration and Air Conditioning Conference, 2002.

[54] R. R. S. A. K.E.Herold, "Single-Stage Ammonia/Water Systems," in Absorption Chillers and Heat Pumps, New York, USA, CRC Press, 1996, pp. 185-187.

[55] D. J. Z.Crespinsek, "Comparison of the Performances of Absorption Refrigeration Cycles," Wseas Transactions on Heat and Mass Transfer, vol. 4, no. 3, pp. 65-76, 2009.

Appendix

Table 5 Thermodynamic Properties of Ammonia (EES generated)

T	P	h_f	h_g	s_f	s_g	u_f	u_g	v_f	v_g
[C]	kPa	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[m^3/kg]	[m^3/kg]
-20	190.1	108.6	1438	0.654	5.904	108.3	1319	0.001504	0.6237
-18	207.6	117.6	1440	0.6894	5.873	117.3	1321	0.001509	0.5742
-16	226.3	126.7	1443	0.7246	5.843	126.3	1323	0.001515	0.5294
-14	246.4	135.8	1446	0.7596	5.814	135.4	1325	0.001521	0.4887
-12	267.9	144.8	1448	0.7944	5.785	144.4	1327	0.001527	0.4518
-10	290.8	154	1451	0.8291	5.757	153.5	1329	0.001534	0.4182
-8	315.2	163.1	1453	0.8635	5.729	162.6	1331	0.00154	0.3875
-6	341.2	172.2	1455	0.8977	5.701	171.7	1333	0.001546	0.3596
-4	368.9	181.4	1458	0.9318	5.674	180.9	1335	0.001553	0.334
-2	398.3	190.6	1460	0.9657	5.647	190	1336	0.001559	0.3106
0	429.6	199.9	1462	0.9995	5.621	199.2	1338	0.001566	0.2892
2	462.6	209.1	1464	1.033	5.595	208.4	1340	0.001572	0.2695
4	497.7	218.4	1466	1.066	5.569	217.6	1341	0.001579	0.2514
6	534.8	227.7	1468	1.1	5.544	226.8	1343	0.001586	0.2348
8	573.9	237	1470	1.133	5.519	236.1	1344	0.001593	0.2195
10	615.3	246.4	1472	1.166	5.494	245.4	1346	0.001601	0.2053
12	658.9	255.8	1474	1.198	5.47	254.7	1347	0.001608	0.1923
14	704.9	265.2	1476	1.231	5.446	264	1348	0.001615	0.1802
16	753.3	274.6	1477	1.264	5.422	273.4	1350	0.001623	0.1691
18	804.2	284.1	1479	1.296	5.399	282.8	1351	0.001631	0.1587
20	857.8	293.6	1480	1.328	5.376	292.2	1352	0.001638	0.1491
22	914	303.1	1481	1.36	5.353	301.6	1353	0.001646	0.1402
24	972.9	312.7	1483	1.392	5.33	311.1	1354	0.001655	0.132
26	1035	322.3	1484	1.424	5.307	320.6	1355	0.001663	0.1243
28	1100	331.9	1485	1.456	5.285	330.1	1356	0.001671	0.1171
30	1167	341.6	1486	1.488	5.263	339.7	1357	0.00168	0.1104
32	1238	351.3	1487	1.519	5.241	349.2	1358	0.001689	0.1042
34	1312	361.1	1488	1.551	5.219	358.8	1359	0.001698	0.09839
36	1390	370.9	1489	1.582	5.198	368.5	1359	0.001707	0.09295
38	1471	380.7	1489	1.613	5.176	378.2	1360	0.001716	0.08787
40	1555	390.6	1490	1.644	5.155	387.9	1361	0.001726	0.08311
42	1643	400.5	1490	1.675	5.134	397.6	1361	0.001735	0.07864
44	1735	410.5	1491	1.706	5.113	407.4	1361	0.001745	0.07446
46	1831	420.5	1491	1.737	5.092	417.3	1362	0.001756	0.07053

48	1930	430.5	1491	1.768	5.071	427.1	1362	0.001766	0.06684
50	2033	440.7	1491	1.799	5.05	437.1	1362	0.001777	0.06337
52	2141	450.8	1491	1.83	5.029	447	1362	0.001788	0.06011
54	2253	461.1	1491	1.861	5.008	457	1362	0.001799	0.05703
56	2369	471.4	1490	1.891	4.988	467.1	1362	0.00181	0.05414
58	2489	481.7	1490	1.922	4.967	477.2	1362	0.001822	0.0514
60	2614	492.1	1489	1.953	4.946	487.3	1362	0.001834	0.04882
62	2744	502.6	1489	1.983	4.925	497.5	1361	0.001847	0.04639
64	2879	513.1	1488	2.014	4.905	507.8	1361	0.00186	0.04409
66	3018	523.8	1487	2.045	4.884	518.1	1360	0.001873	0.04191
68	3162	534.5	1485	2.075	4.863	528.5	1359	0.001887	0.03985
70	3312	545.2	1484	2.106	4.842	538.9	1359	0.001901	0.03789
72	3467	556.1	1482	2.137	4.821	549.5	1358	0.001915	0.03604
74	3627	567.1	1481	2.167	4.8	560.1	1356	0.00193	0.03428
76	3792	578.1	1479	2.198	4.778	570.7	1355	0.001946	0.03262
78	3964	589.3	1477	2.229	4.756	581.5	1354	0.001962	0.03103
80	4141	600.5	1474	2.26	4.735	592.3	1352	0.001978	0.02952
82	4323	611.9	1472	2.291	4.713	603.3	1350	0.001995	0.02809
84	4512	623.4	1469	2.322	4.69	614.3	1348	0.002013	0.02673
86	4707	635	1466	2.354	4.668	625.4	1346	0.002032	0.02542
88	4909	646.8	1463	2.385	4.645	636.7	1344	0.002051	0.02418
90	5116	658.7	1459	2.417	4.621	648.1	1342	0.002072	0.023
92	5331	670.7	1455	2.449	4.597	659.6	1339	0.002093	0.02187
94	5552	683	1451	2.481	4.573	671.2	1336	0.002115	0.02079
96	5780	695.4	1447	2.513	4.548	683	1332	0.002139	0.01975
98	6015	708	1442	2.546	4.523	695	1329	0.002163	0.01876
100	6257	720.8	1436	2.579	4.497	707.1	1325	0.002189	0.01781
102	6507	733.9	1431	2.613	4.47	719.5	1321	0.002217	0.0169

EES Code (Professional Version V10.561)

```
$UnitSystem SI C kPa

"Given"
eshx = 0.9                          "Subcooler effectiveness"
eff = 0.9                            "SHX effectiveness"
T_gen = 150 [C]                       "Generator Temperature"
T_amb=40 [C]                          "Ambient temperature"
T_C= 20 [C]                           "Cabin temperature"
T_abs=40 [C]                          "Absorber temperature"
eta_p=0.8                           "Pump effeciency"
P_gen=pressure(Ammonia,T=T[4],x=1)        "Generator pressure"
P_abs=15.45*T_C+291                           "Absorber pressure"
DT_cond=10 [C]                         "Approach temperature of condenser"
DT_evap=5 [C]                          "Approach temperature of evaporator"
Q_dot_evap = 4 [kW]                     "Coolig Load Given"
X[10] = 0.99                          "Rectifier Ammonia Mass Fraction"
 f = (X[10]-X[7])/(X[9]-X[7])

"Known Pressure conditions"
P[1]=P_abs
P[2]=P_abs
P[3]=P_gen
P[4]=P_gen
P[5]=P_abs
P[6]=P_gen
P[7]=P_gen
P[8]=P_gen
P[9]=P_gen
P[10]=P_gen
P[11]=P_gen
P[12]=P_abs
P[13]=P_abs
P[14]=P_gen

"Known Mass fractions"
X[4]=X[10]
X[14]=X[4]
X[1]=X[14]
X[13]=X[1]
X[2]=X[13]
X[5]=X[6]
X[6]=X[9]
X[9]=X[11]
X[7]=X[8]
X[8]=X[12]

"known mass flow "
m_dot_r=m[10]
m[5]=m[6]
m[6]=m[9]
m[7]=m[8]
m[8]=m[12]
m[10]=m[4]
m[14]=m[4]
```

```
m[1]=m[14]
m[13]=m[1]
m[2]=m[13]

"known quality"
Q[5]=0
Q[7]=0
Q[3]=1
Q[10]=1
Q[11]=0

"ABSORBER"
m[2]*h[2]+m[12]*h[12]=Q_dot_abs+m[5]*h[5]
m[2]*X[2]+m[12]*X[12]=m[5]*X[5]
m[2]+m[12]=m[5]

"GENERATOR"
m[9]*h[9]+Q_dot_gen+m[11]*h[11]=m[3]*h[3]+m[7]*h[7]
ZZ=m[9]+m[11]-m[7]-m[3]
XX=m[9]*X[9]+m[11]*X[11]-m[3]*X[3]-m[7]*X[7]

"RECTIFIER"
m[10]*h[10]+m[11]*h[11]+Q_rect=m[3]*h[3]
m[10]*X[10]+m[11]*X[11]=m[3]*X[3]
m[10]+m[11]=m[3]
h[10] = enthalpy(NH3H2O, P = P[10], X = X[10], Q = Q[10])
h[11]=enthalpy(NH3H2O, X=X[11], P=P[11],Q=Q[11])
T[10] = temperature(NH3H2O, h = h[10],X=X[10], P=P[10])
T[11]  = temperature(NH3H2O, h = h[11], X= X[11], P=P[11])


"Sub-cooler Exchanger"

Q_rs = m[13]*(h2e - h[13])
Q_ps = m[4]*(h[4] - h14e)
h2e = enthalpy(NH3H2O, T = T[4], P = P[13], X=X[10])
h14e = enthalpy(NH3H2O, T = T[13] , P = P[14], X=X[10])

Qmin = min((Q_rs),(Q_ps))

h[2] = h[13] + ((Qmin*eshx)/m[13])
h[14] = h[4] - ((Qmin*eshx)/m[4])
T[2] = temperature(NH3H2O , h = h[2], X=X[2], P=P[2])
T[14] = temperature(NH3H2O, h = h[14],X=X[14], P =P[14])

"Solution Exchanger"

Q_rs1 = m[6]*(h9e - h[6])
Q_ps1 = m[7]*(h[7] - h8e)
h9e = enthalpy(NH3H2O, T = T[7], P = P[9], X=X[9])
h8e = enthalpy(NH3H2O, T = T[6] , P = P[8], X=X[8])

Qmin1 = min(Q_rs1,Q_ps1)

h[9] = h[6] + Qmin1*eff/m[6]
h[8] = h[7] - Qmin1*eff/m[7]
```

```
T[8] = temperature(NH3H2O , h = h[8], X=X[8], P=P[8])
T[9] = temperature(NH3H2O, h = h[9],X=X[9], P = P[9])

"Across Pump"

T[5]=T_abs
h[5]=enthalpy(NH3H2O,T=T[5],P=P[5],Q=Q[5])
X[5]=massfraction(NH3H2O, Q=Q[5], T=T[5], P=P[5])
rho[5]=density(NH3H2O,T=T[5],P=P[5],Q=Q[5])
h[6]=h[5]+((P[6]-P[5])/(rho[5]*eta_p))
T[6]=temperature(NH3H2O,X=X[6],P=P[6],h=h[6])
W_dot_p=m[5]*(h[6]-h[5])

"Liquid leaving generator/entering SHX"

T[7] = T[3]
h[7]=enthalpy(NH3H2O,T=T[7],P=P[7],Q=Q[7])
X[7]=massfraction(NH3H2O,T=T[7],P=P[7],Q=Q[7])


"Leaving evaporator"
T[13]=T_C-DT_evap
h[13]=enthalpy(NH3H2O,T=T[13],P=P[13],X=X[10])

"Leaving valve/entering evaporator"

h[1]=h[14]
T[1] = temperature(NH3H2O, h=h[1], P=P[1],X=X[1])

"Leaving condenser/subcooler 1"

T[4]=T_amb+DT_cond
h[4]=enthalpy(NH3H2O,T=T[4],P=P[4],X=X[4])
Q[4]=quality(NH3H2O,T=T[4],P=P[4],X=X[4])
"Vapor leaving generator/entering rectifier"


T[3] = T_gen
h[3]=  enthalpy(NH3H2O,T=T[3],P=P[3],Q=Q[3])
X[3] = massfraction(NH3H2O, T=T[3],P=P[3],Q=Q[3])
"Energy balances on condenser and evaporator"
Q_dot_cond=m[4]*(h[10]-h[4])
Q_dot_evap=m[1]*(h[13]-h[1])

 "SHX Valve"
h[12] = h[8]
T[12] = temperature(NH3H2O, P=P[12], h=h[12],X=X[12])

"OUTPUT Q"
Q[13] = quality(NH3H2O, X = X[10], T=T[13], P=P[13])
Q[6] = quality(NH3H2O, X = X[6], T=T[6], P=P[6])
Q[8] = quality(NH3H2O, X = X[8], T=T[8], P=P[8])
Q[9] = quality(NH3H2O, X = X[9], T=T[9], P=P[9])


Q[12] = quality(NH3H2O, X = X[12], T=T[12], P=P[12])
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```
Q[1]= quality(NH3H2O, X=X[10], T=T[1], P=P_abs)
Q[2]=quality(NH3H2O, X=X[10], T=T[2], P=P_abs)
Q[14]=quality(NH3H2O, X=X[10], T=T[14], P=P_gen)

"Density Output"
{rho[1]=density(NH3H2O,T=T[1],P=P[1],Q=Q[1])
rho[2]=density(NH3H2O,T=T[2],P=P[2],Q=Q[2])
rho[3]=density(NH3H2O,T=T[3],P=P[3],Q=Q[3])
rho[4]=density(NH3H2O,T=T[4],P=P[4],X=X[4])

rho[6]=density(NH3H2O,T=T[6],P=P[6],X=X[6])
rho[7]=density(NH3H2O,T=T[7],P=P[7],Q=Q[7])
rho[8]=density(NH3H2O,T=T[8],P=P[8],X=X[8])
rho[9]=density(NH3H2O,T=T[9],P=P[9],X=X[9])
rho[10]=density(NH3H2O,T=T[10],P=P[10],Q=Q[10])
rho[11]=density(NH3H2O,T=T[11],P=P[11],Q=Q[11])
rho[12]=density(NH3H2O,T=T[12],P=P[12],X=X[12])
rho[13]=density(NH3H2O,T=T[13],P=P[13],Q=Q[13])
rho[14]=density(NH3H2O,T=T[14],P=P[14],X=X[14])}


 COP=Q_dot_evap/(Q_dot_gen+W_dot_p)
ConvgQ= Q_dot_gen + Q_dot_evap + W_dot_p - (Q_rect + Q_dot_cond + Q_dot_abs)
```