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Exergy Analysis of a Vapor Absorption Refrigeration System Using Carbon Dioxide as Refrigerant

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ABSTRACT

Vapor absorption refrigeration systems can replace vapor compression systems in many applications as they can operate on a low-grade heat source and are environment friendly. Widely used refrigerants such as CFCs and HFCs cause significant global warming. Natural refrigerants can be an alternative to them, among which carbon dioxide is promising. Its inherent safety, ability to withstand high pressure and high heat transfer coefficient coupled with easy availability make it a likely choice for refrigerant. [Bmim][PF_6] is a room temperature ionic liquid (RTIL) composed of a large organic cation and an inorganic anion. Its properties such as non-toxicity, stability over a wide temperature range and ability to dissolve gases like carbon dioxide, make it a suitable absorbent for a vapor absorption refrigeration system. In this paper, an absorption chiller utilising the CO_2-[bmim][PF_6] working pair and consisting of a generator, condenser, evaporator and absorber has been proposed. A thermodynamic model was set up using the Redlich-Kwong cubic equations of state to predict the behavior of the refrigerant and absorbent pair at different points in the system. A MATLAB code was used to obtain the values of enthalpy and entropy at selected points in the system. The exergy destruction in each component and exergetic coefficient of performance (ECOP) of the system were calculated by performing an exergy analysis based on the second law of thermodynamics. The effect of every component on the ECOP was examined by calculating the amount of energy destructions in each of them. The exergetic coefficient of performance was calculated for five generator temperatures in the range of 60-100ºC for an evaporator temperature of 10ºC and was found to be in the range of 0.02-0.2. Such a chiller can be employed for refrigeration purposes, such as in the storage of food items.

1. INTRODUCTION

Artificial cooling is indispensable to modern life. Currently, more than 90% of refrigeration and air-conditioning equipment present within the atmosphere rely on the highly efficient vapor compression process, using electricity or heat energy to perform their intended operations. However, emission of pollutants such as nitrogen oxides, carbon dioxide and particulate matter from these systems has caused massive global warming due to greenhouse effect. The widespread use of refrigerants such as HFCs (hydrofluorocarbons) and CFCs (chlorofluorocarbons) has also been a matter of grave concern, owing to their ozone depletion potential and contribution to greenhouse effect. In the wake of these issues, the United Nations (UN) in 2015 formulated an ambitious plan to create a healthy, fair, sustainable and prosperous world by 2030. Achieving the goals outlined by the UN depends to a large extent on developing clean cooling technologies, as mentioned in a report published by the University of Birmingham Energy Institute. Therefore, phasing out of harmful refrigerants and switching to methods such as vapor absorption refrigeration are being undertaken to mitigate global warming. The ability of absorption refrigeration systems (ARS) to produce higher cooling capacity while being powered by energy sources other than electricity, such as low-grade waste heat from gas and steam turbines, biomass, solar energy and geothermal energy, is leading to a paradigm shift in cooling technology. By virtue of not depleting the ozone layer, the ARS is environment friendly. Many research and development efforts have been dedicated to this type of systems. The technology dates back to as early as the mid-nineteenth century, making it more than a hundred years old. Baltzar von Platen and Carl Munters had successfully developed the process at the KTH Royal Institute of Technology in Stockholm, Sweden in 1922. Commercial production of absorption refrigeration systems began in 1923 and the cycle was patented by Platen and Munters in 1928. The well-known refrigerant-absorber systems H_2O/LiBr and NH_3/H_2O are now being widely utilized, particularly in the field of industrial applications or

large-scale water chiller systems. Recently, more attention has been directed towards the recovery of waste heat using the NH_3/H_2O system. Besides these traditional binary-pairs, in the late 1950s, some pioneering studies were made to propose new refrigerant–absorbent pairs for the vapor absorption cycle, using fluoroalkane refrigerants with organic absorbents. Later in 1989, the Montreal Protocol came into effect, aiming to reduce the production and consumption of ozone-depleting substances. The Kigali Amendment of 2016 has extended the protocol and set a firm timeline for the withdrawal of HFCs. This has compelled researchers to lay emphasis on lower GWP synthetic alternatives and natural refrigerants, as demonstrated by Mota-Babiloni *et al.* (2017). For instance, the traditional refrigerant R134a is being replaced with R1234yf, R600a, and gases such as CO_2, ammonia and propane.

2. LITERATURE REVIEW

The increasing demand for clean refrigerants makes carbon dioxide an attractive alternative to the existing traditional refrigerants. From an environmental perspective, carbon dioxide (R744) is desirable because it has zero ODP (ozone depletion potential) and minimal GWP (global warming potential). Being the by-product of numerous industrial processes, CO_2 is widely available and inexpensive. Although CO_2 is not favorable thermodynamically, it has many favorable thermo-physical properties such as good heat transfer coefficient, insensitivity to pressure losses and very low viscosity. Being a high-pressure refrigerant it is used in systems typically designed to withstand pressures up to 90 bar. Due to low compression pressure ratio, it improves volumetric efficiency. CO_2 does not react with common metals and substances like Teflon. Owing to its high density (about 1.5 times that of ammonia) there is higher mass charge in the evaporators. Large CO_2 systems can offer low initial and life-cycle costs.

Ionic liquids (ILs) can be suitable absorbents for running a vapor absorption refrigeration unit. An ionic liquid is a salt in which the organic cation and inorganic anion are poorly coordinated, resulting in these substances being liquid at room temperature. The absence of volatility and toxicity are the most beneficial properties of ILs. Room temperature ionic liquids (RTILs) such as [bmim][PF_6] (1-butyl-3-methylimidazolium hexafluorophosphate) are thermally stable due to a large mass of the anion. They are highly incompressible and have a negligible vapor pressure.

Research at TU Berlin and TU Graz by Kühn *et al.* (2013) showed that ILs operate with same COP (coefficient of performance) as the water/LiBr system at a cooling capacity which was 80% that of the water/LiBr system. Due to high viscosity, ionic liquids need a more powerful solution pump. They investigated a single-stage absorption heat pump and by means of thermodynamic simulations on ASPEN Plus found that the mixture of NH_3/[bmim][PF_6] was more efficient than the conventional NH_3/H_2O pair. Preißinger *et al.* (2013) calculated that for the working pair water/[emim][DMP] (1-ethyl-3-methylimidazolium dimethylphosphate), the coefficient of performance was comparable or even higher to that of the LiBr system. They concluded that absorption chillers can be used for utilizing low-grade waste heat in industry. Yokozeki *et al.* (2008) measured the gaseous solubilities of carbon dioxide in 18 room-temperature ionic liquids (RTILs) at 298K and observed two types of chemical associations - AB and AB_2 where A = CO_2 and B = RTIL. Huang *et al.* (2005) found that most of the space occupied by CO_2 in the IL phase consists of localized cavities formed by small angular rearrangements of the anions. CO_2 is able to fit above and below the imidazolium ring and is found close to its long alkyl tail. Kamps *et al.* (2003) conducted experiments to conclude that from 293K to 393K and pressures up to 9.7MPa, carbon dioxide showed purely physical solubility in [bmim][PF_6]. Barghi *et al.* (2015) used a gravimetric method to demonstrate that between 25ºC and 55ºC and for pressures up to 10 bar, carbon dioxide's diffusivity in ionic liquids showed very little dependence on pressure.

As vapor absorption systems are emerging as a futuristic way of cooling, it is important to gather information on the efficiency of their working mechanism. This can be done through exergy analysis. As defined by Sahin (2014), exergy is the maximum amount of work that can be obtained from a stream of matter, heat or work as it comes to equilibrium with a reference environment and exergy analysis may be considered as an accounting of the use of energy and material resources. According to Dincer (2013), if we aim to achieve better use of resources, we must transition from energy management to exergy management. Therefore, research has been underway to investigate the feasibility of absorption systems using exergy analysis. Kim *et al.* (2012) performed the exergy analysis of an ARS using an ionic liquid (IL) as a working fluid in the chemical compressor. They developed a mathematical model to characterize the performance for specific refrigerant/IL pairs. A non-random two liquid (NRTL) model was built to predict the solubility of the mixtures. The trend of exergetic coefficient of performance (ECOP) observed in this study was close to that of coefficient of performance (COP) of the absorption system. It was concluded that ionic liquid can be a promising working fluid. Yokozeki (2004) evaluated the theoretical performance for a variety of refrigerants, such as R-22 and ammonia, and absorbents, such as DMA (*N,N*-di-methyl acetamide) and DMF (*N,N*-di-methyl formamide), using equations of state

(EOS) to determine their vapor-phase properties. By analysing the solubility data of these working pairs with the proposed EOS, he successfully demonstrated the usefulness of the EOS model for the vapor-absorption process. Fartaj *et al.* (2004) performed the second law analysis of the transcritical CO_2 refrigeration cycle. By theoretical modelling, the distribution of exergy losses was derived and the exergetic effectiveness of components discovered. It was revealed that the maximum non-idealities in the system are exhibited in the compressor and gas cooler. Rios (2015) used the effectiveness-NTU method to derive a finite heat capacity model for a four temperature level absorption chiller interacting with finite capacity heat reservoirs. Then, to accomplish the best compromise between efficiency and cooling load, a second law analysis along with a thermo-ecological criterion was applied. The COP along with exergy destruction across each heat exchanger interacting between different heat reservoirs and the internal components of the system was determined.

There are different types of absorption chillers ranging from direct-fired, indirect-fired to single, double or triple-effect. A single-effect absorption chiller, which was the main focus of this project, is a heat-driven technology employed to transfer heat from a low temperature to a high temperature (cooling/refrigeration cycle). A theoretical study of its operation at various temperatures, derived from a low-grade heat source, was conducted. CO_2 and [bmim][PF_6] were chosen to be the refrigerant and absorbent respectively. Analysis using Redlich-Kwong equations of state yielded thermodynamic properties of the mixture which were further utilised in the thermodynamic analyses to calculate the system's performance coefficients.

3. THERMODYNAMIC ANALYSIS

3.1 System Description

Fig.1 shows a schematic diagram of a vapor absorption system consisting of four major components- generator, condenser, evaporator and absorber, working in tandem with a heat exchanger, pump and expansion valve. Within the generator, a strong solution of the refrigerant-absorbent pair i.e. CO_2-[bmim][PF_6] is present. When heat is supplied to it, the refrigerant evaporates and its vapors move toward the condenser. The weak solution, containing majorly the absorbent and remaining refrigerant which does not evaporate, moves down to the absorber. After the condenser, liquid refrigerant flows to the evaporator, passing through a thermal regulation valve on its way. In the evaporator, as the liquid refrigerant evaporates, it extracts heat from the surroundings thereby producing the desired cooling effect. When this vapor further reaches the absorber, it encounters the weak solution that has arrived from the generator. It gets readily absorbed and hereby a strong solution of the refrigerant and absorbent gets pumped back to the generator. This completes the refrigeration cycle.
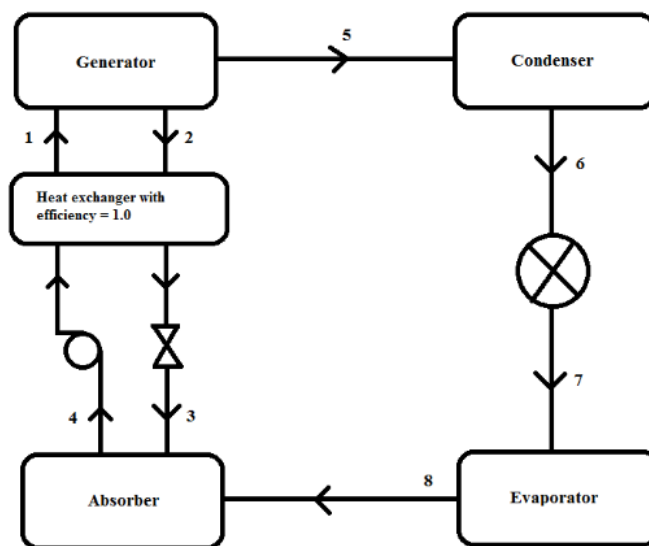
Figure 1: Schematic diagram of an absorption refrigeration system using [bmim][PF_6]/CO_2 as a working fluid pair

3.2 Redlich-Kwong Equations of State Analysis

In order to determine the performance of the vapor absorption cycle, thermodynamic properties at various points in the system were calculated. The Redlich-Kwong type of cubic equations of state formulated by Yokozeki (2001, 2004) were used to evaluate the behavior of the refrigerant-absorbent mixture, i.e., CO_2-[bmim][PF_6] at various conditions.

$$P = \frac{RT}{V - b} - \frac{a(T)}{V(V + b)} \tag{1}$$

$$a(T) = 0.427480 \, \frac{R^2 \, T_c^2}{P_c} \, \alpha(T) \tag{2}$$

$$b = 0.08664 \, \frac{RT_c}{P_c} \tag{3}$$

$$\alpha(T) = \sum_{k=0}^{\leq 3} \beta_k \left(\frac{T_c}{T} - \frac{T}{T_c} \right)^k \tag{4}$$

P_c and T_c are the critical pressure and temperature of pure compounds. These values are 73.89 bar, 304.13K and 27.6 bar, 730.25 K for CO_2 and [bmim][PF_6], respectively.

The temperature dependent part of the 'a' parameter in the EOS equation was also determined empirically by solving the equation using a MATLAB code. The coefficients β_k were determined so as to reproduce the vapor pressure of each pure compound. For mixture equations, 'a' and 'b' were modified to consider N components in the mixture. Modified equations were as follows:

$$k_{ij} = \frac{l_{ij} l_{ji} \, (x_i + x_j)}{l_{ji} \, x_i \, + \, l_{ij} x_j} \tag{5}$$

where $k_{ii} = 0$

$$\alpha(T) = 1 + \beta_1 \left(\frac{T_c}{T} - \frac{T}{T_c} \right) \tag{6}$$

$$a(T) = \sum_{i,j=1}^{N} \sqrt{a_i a_j} \, \left(1 - f(T)k_{ij} \right) x_i x_j, \quad a_i = 0.427480 \, \frac{R^2 T_{ci}^2}{P_{ci}} \, \alpha_i \, (T) \tag{7}$$

$$f(T) = 1 + C_{ij}(T), \qquad \text{where } C_{ij} = C_{ji} \text{ and } C_{ii} = 0 \tag{8}$$

$$b = \frac{1}{2} \sum_{i,j=1}^{N} \left(b_i - b_j \right) \left(1 - m_{ij} \right) \left(1 - k_{ij} \right) x_i x_j, \quad b_i = 0.08664 \frac{RT_{ci}}{P_{ci}} \tag{9}$$

where $m_{ij} = m_{ji}$ and $m_{ii} = 0$

Here i and j represent the two components CO_2 and [bmim][PF6] respectively and x terms correspond to their respective mole fractions in the mixture. The enthalpy at a state point can be determined using the following relationship:

$$H = \int \sum_{i=1}^{N} C_{pi}^0 x_i dT + \left(\frac{a}{b} - \frac{T}{b} \frac{da}{dT} \right) \ln \frac{V}{V+b} + RT \left(\frac{PV}{RT} - 1 \right) - \cdots$$

$$\frac{RT^2}{V-b}\frac{db}{dT} + \frac{a}{b}\frac{db}{dT}\left[\frac{1}{V+b} - \frac{1}{b}\ln\left(1+\frac{b}{V}\right)\right] + C \tag{10}$$

In the above equation, C is an arbitrary constant which remains constant for the component mixture being studied. The ideal gas heat capacity for each compound is determined as follows:

$$C_p^0 = C_0 + C_1 T + C_2 T^2 + C_3 T^3 \tag{11}$$

The following assumptions were made to perform the theoretical analysis:

- Pumping power W_p is negligible.
- There are no pressure drops in the connecting lines.
- The condition at the exit of the evaporator is pure refrigerant at dew point with $T = T_e$.
- The condition at point 6 is refrigerant at bubble point and there is no subcooled liquid. The condition at point 5 is a superheated state of pure refrigerant with $P = P_c$ and $T = T_g$.
- Pressures in the condenser and generator (P_c and P_g) are the same and similarly, the evaporator and absorber pressures (P_e and P_a) are equal.
- The condition at solution inlet to the absorber is a solution's bubble point specified with the absorber pressure (P_a) and solution concentration of the generator (x_g).
- Temperatures in the generator (T_g), absorber (T_a), condenser (T_c), and evaporator (T_e) are specified as a given cycle condition.
- The refrigerant-gas flow rate (m_r) is set to be 1 kg/s.

The chiller was assumed to be equipped with an evaporator at 10°C, 45.02 bar while the generator was kept at atmospheric pressure and studied at five temperatures: 60, 70, 80, 90 and 100°C. For this work, a MATLAB program had been employed and all desired cycle conditions were evaluated. In the case of CO_2-[bmim][PF$_6$], the absorbent [bmim][PF$_6$] may have a non-negligible vapor pressure at the generator exit. In practical applications, a rectifier (distillation unit) may be required in order to separate the refrigerant from the absorbent. However, in this study, its utilization has been avoided.

Table 1: Enthalpy and entropy of the refrigerant at generator outlet for various operating temperatures

Operating temperature [⁰C]	Enthalpy [J/mol]	Entropy [J/mol. K]
60	-210180	322.9431
70	-204810	334.1110
80	643960	638.3387
90	199820	792.0069
100	-241900	296.9505

3.3 First Law of Thermodynamic Analysis

Fig. 1 shows a schematic diagram of a simple vapor absorption refrigeration cycle. Theoretical cycle performances are evaluated as follows. The overall energy balance gives

$$Q_g + Q_e + W_p = Q_c + Q_a \tag{12}$$

The material balance in the absorber or generator is given as follows:

$$m_s x_a = (m_s - m_r)x_g \tag{13}$$

This provides a mass flow ratio f, as defined by

$$f = \frac{m_s}{m_r} = \frac{x_g}{(x_g - x_a)} \tag{14}$$

where m_r and m_s are the mass flow rates of gaseous refrigerant and the absorber-exit solution (or solution pumping rate) respectively. Then, the system performance is defined by a heat ratio i.e. output power divided by input power;

$$COP = \frac{Q_e}{(Q_g + W_p)} \tag{15}$$

This can be expressed in terms of H and f:

$$COP = \frac{(H_8 - H_7)}{[H_5 + H_3(f - 1) - H_4 f]} \tag{16}$$

3.4 Second Law of Thermodynamic (Exergy) Analysis

The analysis based on the first law of thermodynamics determines the amount of energy entering and leaving from each one of the components as well as the entire system. However, neither does it give information about energy quality, nor the irreversibilities in the system. Therefore, the second law of thermodynamics has been implemented to provide a better understanding of the thermal performance characteristics of each component in the system. This law also helps to identify the component with high energy dissipation and irreversible losses, which in turn allows focusing more attention to such a component in order to investigate and reduce its irreversibility losses.

As it is known, energy can be transformed from one form to another during which its quality changes. According to the first law of thermodynamics, the magnitude of energy is conserved in a process. However, from experimental evidence, a loss in the quality of energy is observed, causing the outlet energies to have a lower value as compared to the inlet energies during any process. Our goal was to indicate how energy transforms and study how energy degrades due to entropy production and exergy destruction at various locations in the system. Exergy is defined as the amount of work available from an energy source, as follows:

$$Ex = \dot{m}(h - h_{ref}) - \dot{m}T_{ref}(s - s_{ref}) \tag{17}$$

Where T_{ref} is the reference environmental temperature and h_{ref} and s_{ref} are specific enthalpy and entropy evaluated at the reference environmental state. The magnitude of the exergy involved with the heat transfer process, Ex_Q, is the work that could be obtained using a reversible Carnot cycle operating between the temperature from which the heat is received and the ambient temperature.

$$Ex_Q = \left(1 - \frac{T_{ref}}{T}\right)Q \tag{18}$$

The bracketed term is exergy quality, which increases with the increasing temperature *T*. The exergetic coefficient of performance can be defined as the ratio of the exergy obtained at the evaporator for cooling to the exergies supplied to the generator and pump.

$$ECOP = \frac{Q_e\left(1 - {}^{T_{ref}}/_{T_{inlet}}\right)}{Ex_{in} - Ex_{out} + W_p} \tag{19}$$

The exergy loss or irreversibility I can be written as

$$I = \sum Ex_{in} - \sum Ex_{out} + \sum Q\left(1 - \frac{T_{ref}}{T}\right) - W \tag{20}$$

The exergy losses of the components in Fig. 1 are as follows:

$$I_e = Ex_7 - Ex_8 + Q_e \left(1 - \frac{T_{ref}}{T_{inlet}} \right) \tag{21}$$

$$I_c = Ex_6 - Ex_8 \tag{22}$$

$$I_a = Ex_8 + Ex_3 - Ex_4 \tag{23}$$

$$I_g = Ex_1 - Ex_2 - Ex_6 \tag{24}$$

4. RESULTS AND DISCUSSION

The vapor absorption system is designed to operate using a low-grade heat source, with temperatures ranging from 60-100ºC. The Redlich-Kwong equations of state enable us to determine the enthalpy at each state point of the system as a function of temperature, pressure, volume, mole fractions of the components and the EOS parameters 'a' and 'b'. These state point enthalpies can be inserted in equation (16) to obtain an estimate of the COP of system using carbon dioxide and [bmim][PF$_6$] as the working pair. A second law analysis yields the enthalpies at state points 6, 7 and 8, as they are present in the condenser and evaporator regions of the cycle. The enthalpies at these points are assumed to be dependent only on the refrigerant. The enthalpies and entropies obtained are used to calculate the exergy using equation (17).
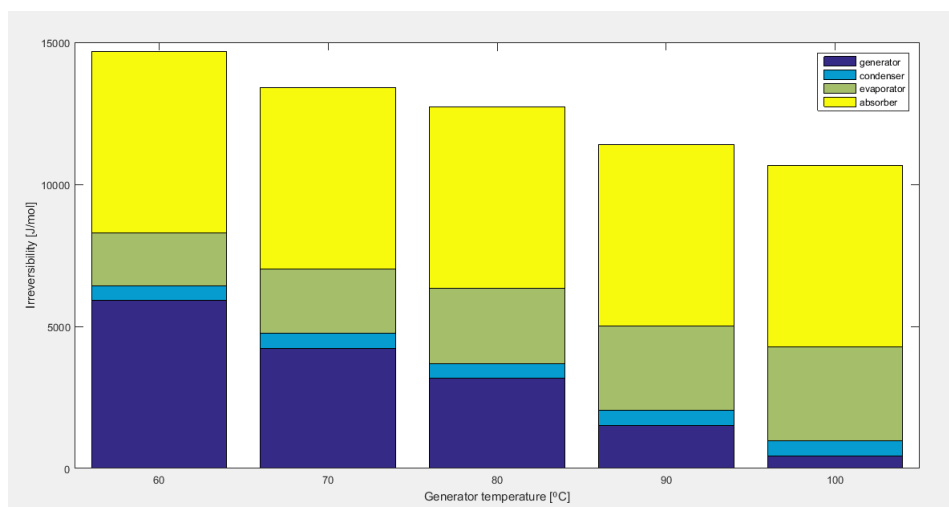
Figure 2: Irreversibilities in the absorption system components at various generator temperatures

Table 2: Percentage irreversibility in each component of the ARS at various generator temperatures

Generator temperature [ºC]	Generator	Condenser	Evaporator	Absorber
60	40.23	3.57	12.70	43.49
70	31.4	3.93	16.9	47.76
80	23.93	3.96	19.86	48.18
90	13.24	4.6	26.16	55.99
100	4.16	4.92	31.08	59.84

The irreversibility in each component of the absorption system with respect to generator temperature, i.e., operating temperature is depicted in Figure 2. At any given temperature, cycle irreversibility is mainly a consequence of large

exergy loss. Table 2 gives a detailed account of the percentage irreversibility in each component at the various operating temperatures. It can be observed that the largest irreversibility always occurs in the absorber while condenser operates with the least irreversibility. This indicates a potential for performance improvement in the absorber, which can be achieved by operating it at a low temperature. Since the irreversibility in the evaporator is increasing with generator temperature, it should also be operated at a low temperature. As the operating temperature increases, the cumulative irreversibility of the system is reducing, leading to increase in ECOP as shown in Fig. 4(a). This is accompanied by a significant drop in the irreversibility of the generator and rise of the same in the other three components, namely condenser, evaporator and absorber.

(a)   (b)
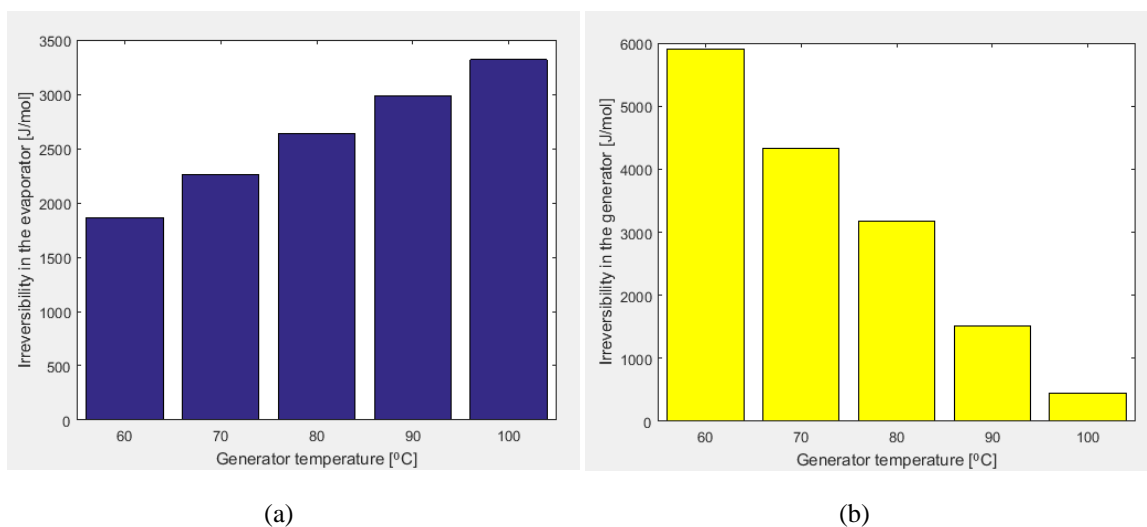
Figure 3: Irreversibility in the (a) evaporator and (b) generator at various generator temperatures

Figures 3 (a) and (b) show the irreversibilites in the evaporator and generator respectively, for five different generator temperatures between 60ºC and 100ºC. Irreversibility is an indicator of exergy destruction. It is observed that for the evaporator, the irreversibility is lowest for a generator temperature of 60ºC. Thereafter, it continues to rise and reaches its maximum value at 100ºC. On the contrary, in the generator it is observed that the maximum irreversibility or exergy destruction takes place at 60 ºC generator temperature, after which it steadily reduces and attains a minimum value at 100 ºC.

(a)   (b)
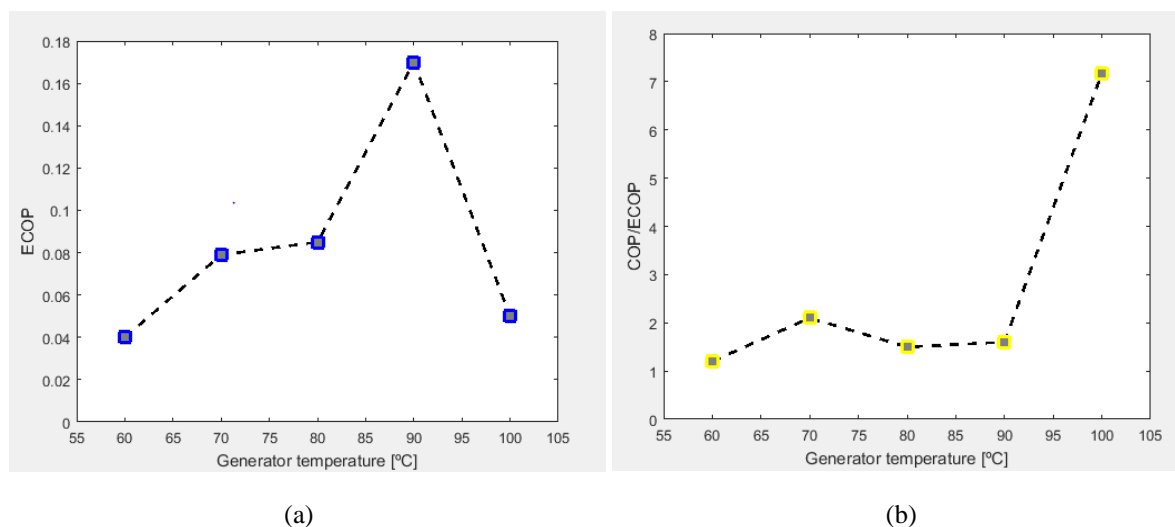
Figure 4: (a) ECOP (b) the ratio of COP to ECOP of the absorption system at various generator temperatures

Ultimately, the ECOP is calculated using equation (18). Fig. 4(a) shows the variations of ECOP with respect to five operating temperatures i.e. generator inlet temperatures: 60ºC, 70ºC, 80ºC, 90ºC and 100ºC. The condenser was maintained at room temperature, i.e., 25 ℃ and the evaporator at 10 ℃. It can be seen that the ECOP rises, reaching up to a maximum of 0.17 at 90ºC, and thereafter starts reducing. The ECOP trend observed is a result of the exergy losses. Fig. 4(b) shows the variation of the COP/ ECOP ratio for five operating temperatures from 60ºC to 100 ℃. A sharp rise is observed at 100ºC.

5. CONCLUSION

This work shows that carbon dioxide as refrigerant and the ionic liquid [bmim][PF$_6$] as absorbent can make a viable working pair for a vapor absorption refrigeration system. Such a system can make use of low-grade heat energy for its operation and is therefore, environment-friendly. Second law thermodynamic analysis was performed on the system for a better understanding of its efficiency, as the exergy value of heat is much lower than its energy value. It was very effective in determining the exergy destruction across different components of the system and calculating the exergetic coefficient of performance (ECOP). The analysis was carried out by calculating thermodynamic state parameters at different stages of the absorption cycle with the help of the Redlich-Kwong EOS model. A mathematical model representing the system was developed in MATLAB and the desired results were obtained. The system's operation was studied at five operating temperatures from 60°C to 100°C. It was observed that total irreversibility of all components in the system decreased with increase in generator temperature. At each operating temperature, the highest irreversibility occurred in the absorber, while lowest in the condenser. The highest ECOP was found to be 0.17 at a generator temperature of 90°C. It was observed that the COP/ECOP ratio was greater than one for all operating temperatures, indicating that the value of ECOP was always less than that of COP, owing to exergy destruction in all components of the refrigeration system. Such a chiller can achieve an evaporator temperature of 10°C and can be employed for refrigeration purposes, such as in the storage of food items.

NOMENCLATURE

a	independent mixing parameter	
b	independent mixing parameter	
COP	coefficient of performance	
ECOP	exergetic coefficient of performance	
Ex	exergy	(W)
f	flow ratio	
h	specific enthalpy	(J/kg)
H	enthalpy	(J)
I	irreversibility	(W)
ṁ	mass flow rate	(kg/s)
P	pressure	(Pa)
P$_c$	critical pressure	(Pa)
Q	heat transfer	(W)
R	universal gas constant, 8.314	(J/mol·K)
s	specific entropy	(J/kg.K)
T	state temperature	(K)
T$_c$	critical temperature	(K)
V	molar volume	(m^3/mol)
W	work done	(W)
x	mass fraction	

Subscripts

a	absorber
c	condenser
e	evaporator
g	generator

in   inlet
out   outlet
p   pump
r   refrigerant
ref   reference
s   solution

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