

Analysis of Single Stage Vapour Absorption Refrigeration System

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Abstract- In this study, the first and second law thermodynamic analysis of a single-stage compression-absorption system with ammonia-water as working fluid pair is performed. Thermodynamic properties of each point (the inlet and outlet of each component) in the cycle are calculated using related equations of state. Heat transfer rate of each component in the cycle and some performance parameters (circulation ratio CR, coefficient of performance COP) are calculated from the first law analysis. From the second law analysis, the exergy destruction of each component and the total exergy destruction of all the system components are obtained. Variation of the performance and exergy destruction of the system are examined at various operating conditions. Simulation results are presented in tabular and graphical form. The results of the first and second law analysis of the system, done with the help of a program developed in Engineering Equation Solver software.

Keywords- Energy, Exergy Analysis, COP, EES

I. INTRODUCTION

With a view to search for a system which can be used for high temperature lift at lower pressure ratios without using the harmful ozone depleting refrigerants such as the chlorofluorocarbons (CFCs) as used in the conventional vapour compression systems, a novel system called the compression-absorption system was developed in 1895 by Osenbrueck, which became a German patent. Later in 1950, the idea of combining a compressor to an absorption system was presented by Altenkirch.

Among the heat pump technologies, absorption/compression cycles are suitable for large temperature lifts and high temperature applications. Furthermore, non-conventional sources of energy such as solar, waste heat, and geothermal can be used as their primary energy input. This system is a hybrid of the conventional vapour compression and absorption systems which possess best features of both the types.

Many industrial processes have heating demands in the temperature range of 75-100°C. At the same time, waste heat holding typically a temperature of 30-50°C is available. The limitation in heat pump alternatives for temperatures around 100°C after the high temperature refrigerant CFC-114 was abandoned has turned the focus on alternative heat pump processes using natural working fluids. Efficient heat pumping technologies like compression-absorption systems are therefore attractive in order to reduce the specific energy consumption.

Such systems are gaining popularity because they operate on environment friendly refrigerants conforming Montreal

and Kyoto protocols. The compression-absorption systems can be operated over a wide range of temperatures, between -10 and 160°C, using ammonia-water as the working fluid and with pressures not exceeding 20 bar (Stokar and Trepp, 1986). These systems utilize the available low temperature waste heat sources thus providing high temperature lifts required in many industrial applications such as pasteurization, dairy technology and steam production etc.

This system operates on a refrigerant-absorbent mixture unlike the vapour compression system which uses a pure refrigerant fluid. A key advantage of the compression-absorption cycle is the extended range of temperatures available for a mixture compared to pure refrigerants. This is the effect of reduced vapour pressure obtained for a refrigerant in a mixture with a less volatile component. Therefore, refrigerants which, as pure components, are restricted by too high pressures at the required temperatures can be used over new temperature ranges. One example is NH_3 which has a saturation pressure of 25 bar at 58.2°C. When mixed with H_2O it could be possibly be used upto 150-160°C at similar pressures. Thus, by using mixtures, it is possible to cover temperature regimes where it is difficult to find a suitable pure refrigerant [1].

An ammonia-water mixture is a non-ideal mixture which possesses more than Raoult's law of stability. Mixing of ammonia and water is accompanied by the absorption of heat. Ammonia-water mixture is suitable for high temperature heat pump applications because of the reduced pressure when working at higher temperatures. This non-azeotropic mixture has therefore been chosen for the present study. Another advantage of this cycle is the

gliding temperature profiles obtained in the absorber and desorber. Gliding temperature here refers to the temperature change obtained as the volatile component is absorbed into or evaporated out of the solution. Within the given temperature limits of the heat source and sink there is a considerable freedom of placing the solution field within the corresponding saturation pressures of the pure solute and solvent.

This characteristic feature provides greater design flexibility than is possible for single fluid heat pumps. By a suitable choice of the concentration range external conditions can be optimally matched to the properties of the working pair and the heat pump components. Desorption and absorption processes follow non-isothermal paths at constant, but different pressures due to the fact that bubble points of the desorbing and the resorbing solution change as the concentration of the refrigerant lessens or increases respectively.

This behaviour can be utilized to reduce exergy losses by counter current heat exchange when non-isothermal heat sources and sinks are available such as ground water and district heating water. By using a compression-absorption cycle instead of a single fluid compression cycle at identical temperature levels, the process pressure levels are reduced. Depending on the thermodynamic properties of the refrigerant-solution the pressure ratio may become smaller which favourably affects compressor lifetime [1].

Compared with a single-fluid compression cycle, the performance of the compression/absorption cycle is improved as the external temperature gradients increase. Although the compression/absorption cycle is usually only put forward as an alternative to compression cycles for applications involving large external temperature gradients, this study has shown that this cycle is also interesting for applications where the external temperature gradients are small.

The more isothermal the conditions are, however, the greater is the need to optimize the parameters within the cycle in order to maximize its performance. From the results presented in this thesis it is clear that the compression/absorption heat pump cycle offers several advantages over other types of heat pumps. In many cases it gives a higher or equally high COP compared with the single-fluid compression heat pump but also the smaller SCD (i.e. size of the compressor) found for this cycle is an advantage.

II. THERMODYNAMIC ANALYSIS OF A COMPRESSION- ABSORPTION SYSTEM

A single stage compression-absorption heat pump under study has been described in this section to know about the various components comprising the system and the flow of the fluid across them.

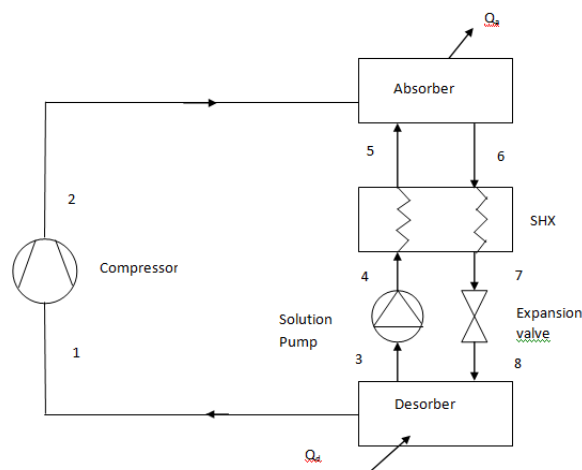


Fig 1. Schematic diagram of a compression-absorption system.

The compression absorption heat pump cycle combines two well-known heat pump concepts viz. the compression heat pump and the absorption heat pump. To create the hybrid cycle, the condenser and the evaporator in a conventional vapour compression heat pump are replaced by an absorber and a desorber. The various components forming a compression-absorption system are an absorber, a desorber, a solution heat exchanger, solution pump, expansion valve which form the solution circuit, where the weak and the strong solution of the ammonia-water mixture circulates.

A compressor is connected to the solution circuit part of the cycle to raise the pressure of the ammonia vapours from the desorber pressure to the absorber pressure. In a compression-absorption cycle, the ammonia vapours enters the compressor at state point 1, where they are compressed to the higher absorber pressure requiring the compressor work. At the state point 2, the compressed ammonia vapours leaves the compressor in a superheated state. Both the temperature and pressure are increased with the concentration remaining constant. The high pressure ammonia vapours from the compressor enter the absorber at point 2. The weak solution of ammonia and water from the solution heat exchanger enters the absorber at point 5. The absorption of ammonia vapours into the weak solution leads to the formation of strong solution which has the higher concentration of NH_3 . This process is accompanied by a rejection of heat. The absorption process takes place over a range of temperatures.

In the absorber, as the concentration of the ammonia increases the temperature decreases at a constant pressure. When the concentration reaches a maximum value, the strong solution leaving the absorber at state point 6 becomes saturated. The pressure of the weak solution leaving the desorber at state point 3 is raised to the absorber pressure that is the higher pressure level of the cycle, by means of a solution pump. A regenerative solution heat exchanger is placed to transfer heat from the

hot, strong solution leaving the absorber to the cold, weak solution entering the absorber.

III. ASSUMPTIONS TAKEN INTO CONSIDERATION FOR THE ANALYSIS

The thermodynamic analysis of the compression-absorption heat pump is carried out by making the following assumptions:

- System is at steady state condition.
- The absorber and the desorber outlets are assumed to be at saturated states.
- Isenthalpic expansion occurs across the expansion valve.
- The undesirable heat losses and pressure drops in various components and pipe lines are neglected.
- Pump efficiency is assumed to be 70%.

IV. PROBLEM FORMULATION

Analysis of an ammonia-water compression-absorption heat pump of 100 kW heating capacity has been carried out in this work. The input parameters taken are absorber pressure = 20 bar, desorber pressure = 4 bar, absorber load = 100 kW, absorber temperature = 50- 80°C, desorber temperature=30-40°C. By carrying out the thermodynamic analysis of the system for the conditions stated above the values of temperature, pressure and concentration at various state points of the cycle have been obtained.

The computer program developed in EES calculates the performance parameters of the system such as coefficient of performance, circulation ratio, solution heat exchanger effectiveness, and compressor work, pump work from the first law analysis and exergetic efficiency, exergy destruction ratio from the second law analysis.

V. RESULTS AND DISCUSSION

In this study, the first and second law thermodynamic analysis of a single-stage compression-absorption system with ammonia-water as working fluid pair is performed. Thermodynamic properties of each point (the inlet and outlet of each component) in the cycle are calculated using related equations of state. Heat transfer rate of each component in the cycle and some performance parameters (circulation ratio CR, coefficient of performance COP) are calculated from the first law analysis.

From the second law analysis, the exergy destruction of each component and the total exergy destruction of all the system components are obtained. Variation of the performance and exergy destruction of the system are examined at various operating conditions. Simulation results are presented in tabular and graphical form. The results of the first and second law analysis of the system, done with the help of a program developed in Engineering

Equation Solver software have been presented in this chapter. The computer program developed within this project calculates the performance for the compression-absorption heat pump. The performance of a system is strongly dependent on how the operating conditions are chosen.

The operating conditions selected for a compression-absorption heat pump of 100 KW heating capacity are as stated below:

- Absorber temperature (T_a) = 50-80°C.
- Desorber temperature (T_d) = 30-40°C in steps of 5.
- Desorber pressure (p_a) = 4 bar.
- Pressure ratio (R) = 5-6.
- Heat exchanger effectiveness = 0.50-1
- Compressor efficiency = 0.50-1
- Pump efficiency = 0.70

1. Variation of COP with Absorber Temperature

Variation of COP has been studied as a function of the absorber temperature for the various values of desorber temperatures varying in the range of 30-40°C. The results have been plotted and discussed in this section.

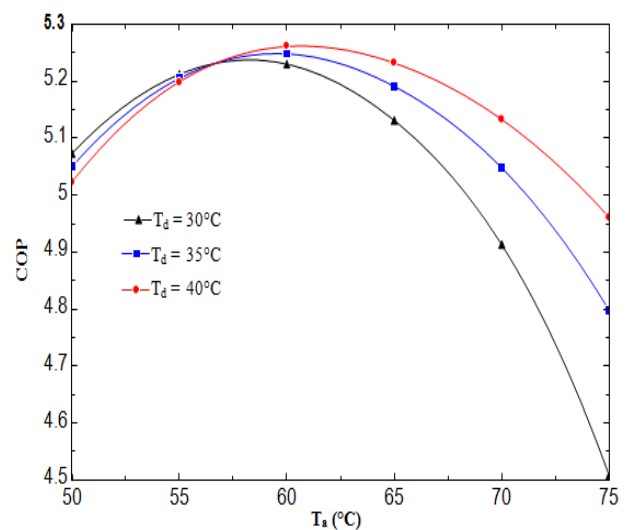


Fig 2. Variation of coefficient of performance with absorber temperature.

The variation of COP of the cycle as a function of absorber temperature for various values of desorber temperatures has been shown in Fig. 2. From Fig. 2, it is evident that as the absorber temperature increases, COP first increases upto an optimum value of the absorber temperature at which the COP obtained is maximum. With any further increase in the absorber temperature, COP starts decreasing.

It is so because as the absorber temperature increases, the mass flow rate of refrigerant decreases which causes a decrease in compressor work, which in turn increases the COP upto the absorber temperature of 60°C. If the

absorber temperature is increased further, COP starts decreasing because of an increase in compressor work.

2. Variation of COP with compressor isentropic efficiency:

Isentropic efficiency of the compressor has also a considerable effect on the coefficient of performance of the system. The variation of COP as a function of compressor efficiencies has been shown in this section. The absorber temperature of 60°C is selected at which the COP of the system is found to be maximum.

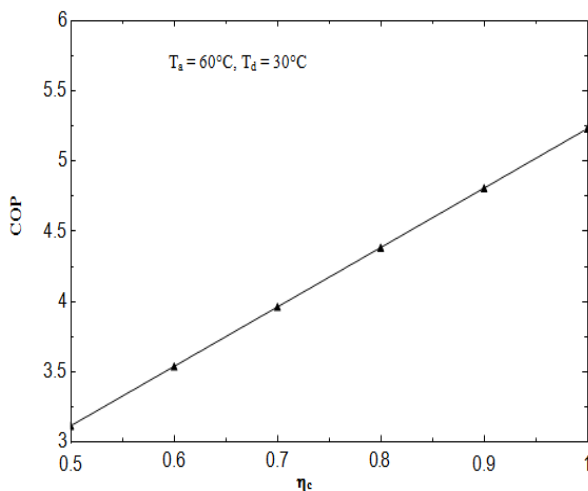


Fig 3. Variation of coefficient of performance with compressor efficiency.

It can be seen from Fig. 3, as the efficiency of the compressor increases the COP also increases due to decrease in the compressor work. Though the pump work has increased but the amount is not significant. It can be seen from Fig. 3, on increasing the pressure ratio the COP decreases if all other parameters are fixed.

This is due to the fact that as the pressure ratio increases, there is an increase in compressor work which decreases the COP. At a pressure ratio of 5, the COP value corresponding to a compressor efficiency of 0.7, the COP is found to be 3.961 whereas when the pressure ratio has been increased to 6, the COP reduces to 3.471.

3. Variation of COP with effectiveness of solution heat exchanger:

Fig. 4 shows the variation of COP with heat exchanger effectiveness at the absorber temperature of 60°C for the desorber temperature varying in the range of 30-40°C. It can be seen that as the heat exchanger effectiveness increases, the COP also increases for a constant value of desorber temperature.

Upto the effectiveness of 0.8, the COP is more for a higher value of desorber temperature. At heat exchanger effectiveness of 0.8, COP is found to be 5.3 for all the desorber temperatures.

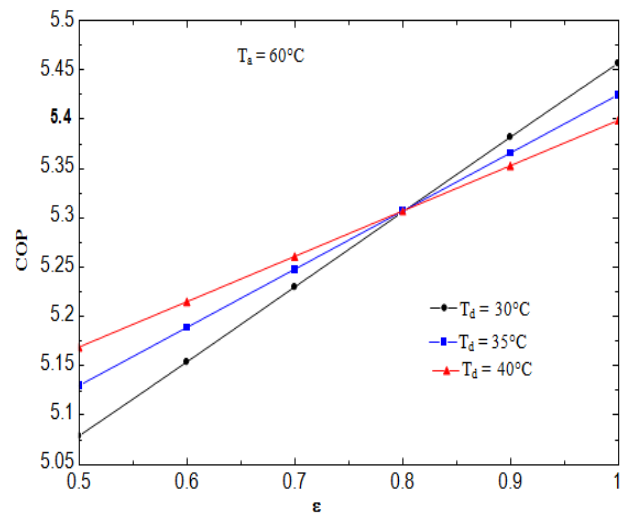


Fig 4. Variation of coefficient of performance with heat exchanger effectiveness.

For the heat exchanger effectiveness more than 0.8, COP is more at a lower value of desorber temperature. At low values of heat exchanger effectiveness COP is less. This is due to increase in weak solution concentration and mass flow rate of the refrigerant which increases the net work input and as a result of which the COP decreases. At higher values of heat exchanger effectiveness, both the compressor and the pump work decrease hence COP increases. For a high solution flow rate, corresponding to a small absorber glide, the solution heat exchanger becomes very important as the amount of energy to be transferred between the strong and the weak solution is large.

4. Variation of compressor work with absorber temperature:

The variation of the work input to the compressor with the absorber temperature has been presented in this section. The curve has been plotted at the desorber temperature of 30°C.

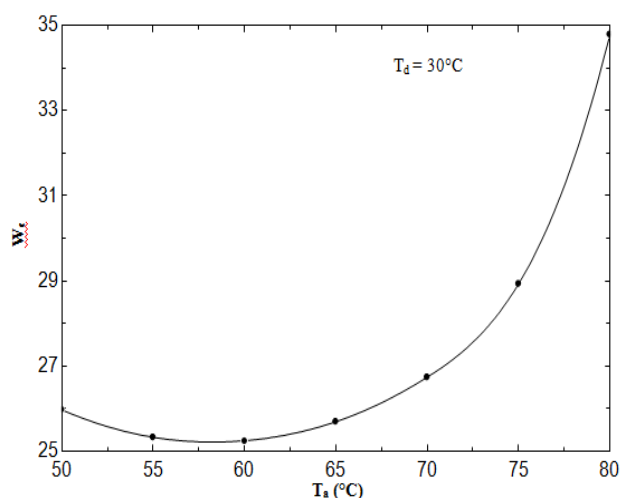


Fig 5. Variation of compressor input power with absorber temperature.

As shown in Fig. 5, the compressor power first decreases with an increase in absorber temperature, becomes minimum at a particular point which for the operating conditions selected for the present study is at an absorber temperature of 60°C for a desorber temperature of 30°C.

With any further increase in the absorber temperature, the compressor work starts increasing. At a fixed value of absorber temperature, the compressor work is more for low weak solution concentration.

5. Variation of Circulation Ratio with absorber temperature:

The circulation ratio (CR) is very important factor for the performance of the cycle as it governs the losses in the solution circuit and the pump work. At high weak solution concentration, the circulation ratio is low.

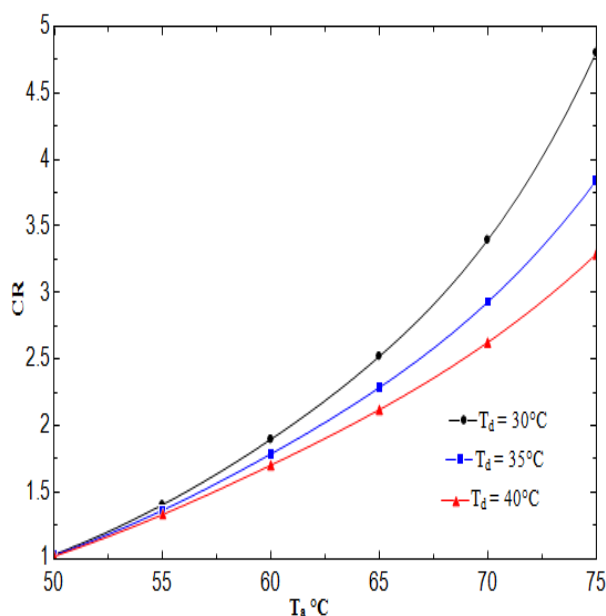


Fig. 6. Variation of circulation ratio with absorber temperature.

Fig. 6 shows that as the absorber temperature increases, circulation ratio increases at constant values of compression ratio and the desorber temperature. As the absorber temperature is increased, the mass flow rate of strong solution increases which results in an increase in the circulation ratio. It can also be explained as that on increasing the absorber temperature, the equilibrium concentration of strong solution reduces.

This results in lesser concentration width and consequently a higher circulation ratio. It can be seen from the fig. that at a constant value of absorber temperature, the circulation ratio is less at a higher value of desorber temperature. The reason for this is that as the desorber temperature is increased, weak solution concentration decreases which results in an increase in concentration width, hence the circulation ratio decreases.

6. Variation of Concentration Width with Absorber Temperature:

The variation of the concentration width with absorber temperature corresponding to three values of desorber temperatures i.e. at 30°C, 35°C and 40°C has been shown in this section. Concentration width is the difference between the strong and the weak solution concentrations.

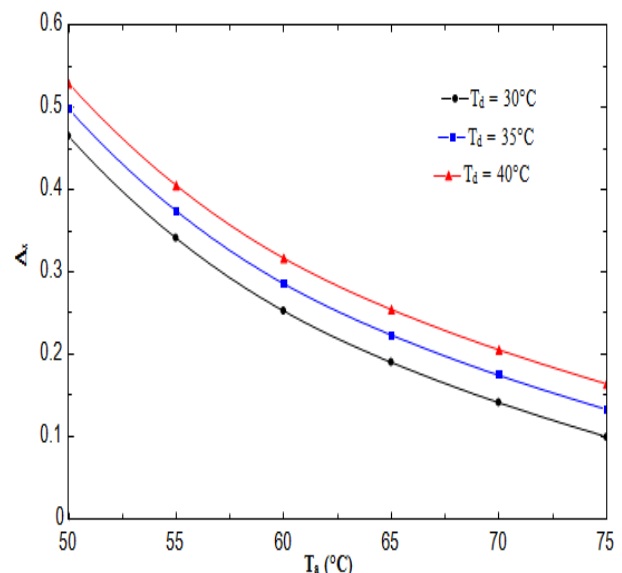


Fig. 7. Variation of concentration difference with absorber temperature.

In Fig. 7, it is shown that as the absorber temperature is increased, at a constant value of absorber pressure, the strong solution concentration increases with the weak solution concentration remaining constant. This causes the concentration difference between the strong and the weak solution to decrease. As the desorber temperature is increased, the concentration width increases due to decrease in the weak solution concentration. A sharper picture of the performance is provided by the results of the exergetic analysis. The exergy analysis emphasizes that both losses and irreversibility have an impact on system performance. The inlet and the outlet exergy of any process do not match. The difference represents the exergy lost during that process. An exergy analysis is performed to determine the potential areas of improvement in the cycle.

7. Results of Second Law Analysis:

The results of second law analysis of the system have been presented in this section. From the second law analysis of the system, both the individual component exergy loss and the non dimensional exergy loss have been calculated. The performance parameters used to evaluate the system from exergetic point of view are the exergetic efficiency and the exergy destruction ratio. Exergy destruction ratio (EDR) helps to predict the contribution of a component to the total exergy loss so that the component with higher values can be improved. It is found that for the same set of

operating conditions, the exergy loss in the absorber is the highest. At an absorber temperature of 60°C and desorber temperature of 30°C, 38.06% of the total exergy loss is contributed by the absorber. It can also be seen that the next highest exergy loss occurs in the compressor which is equal to 30.62% which is followed by the exergy loss in the desorber, expansion valve, pump and the solution heat exchanger.

7.1 Variation of exergetic efficiency with absorber temperature: The variation of exergetic efficiency with absorber temperature at various values of desorber temperatures i.e. at 30°C, 35°C and 40°C keeping all the other parameters as constant has been shown and discussed in this section.

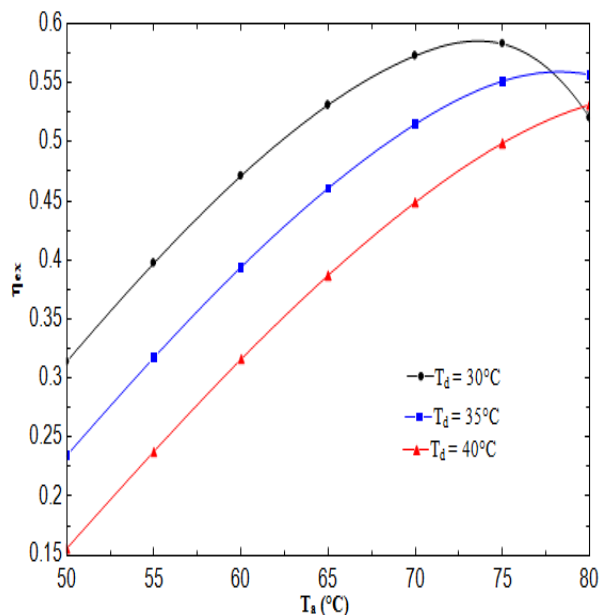


Fig. 8. Variation of exergetic efficiency with absorber temperature ($T_d = 30^\circ\text{C}$, 35°C , 40°C).

It can be seen from Fig. 8, as the absorber temperature is increased, the exergetic efficiency is increased for a constant value of desorber temperature because it increases the term

$$\left(1 - \frac{T_d}{T_a}\right)$$

in the expression for exergetic efficiency as stated by equation 3.49 in chapter 3. The compressor work decreases and the pump work increases but the rate of increase of the pump work is small, hence the exergetic efficiency increases. With an increase in the desorber temperature, the term

$$\left(1 - \frac{T_d}{T_a}\right)$$

and pump work decreases causing a decrease in the exergetic efficiency. From Fig. 8, it is seen that the exergetic efficiency is maximum at an absorber temperature of 75°C corresponding to a desorber temperature of 30°C which is equal to 0.4467. For higher values of desorber temperature, the peak of the curve shifts to the right.

7.2 Variation of exergetic efficiency with heat exchanger effectiveness: In this section, the variation of exergetic efficiency has been plotted as a function of heat exchanger effectiveness. The curve has been drawn for fixed values of absorber temperature, compressor efficiency and pressure ratio. The absorber temperature, compressor efficiency and pressure ratio are taken as 60°C, 0.7 and 5 respectively. The variation has been shown corresponding to the desorber temperatures of 30°C, 35°C and 40°C.

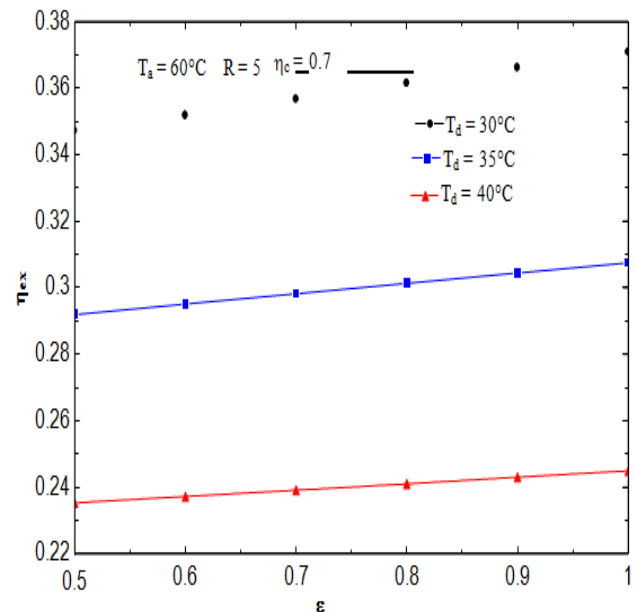


Fig. 9. Variation of exergetic efficiency with heat exchanger effectiveness.

Fig. 9 shows that on increasing the effectiveness of solution heat exchanger, the exergetic efficiency of the system also increases at a constant value of desorber temperature. For higher values of desorber temperatures, the exergetic efficiency is more at a fixed value of heat exchanger effectiveness.

7.3 Variation of exergetic efficiency with compressor efficiency: Compressor efficiency also affects the exergetic efficiency of this system. The variation of exergetic efficiency with the compressor efficiency has been shown in Fig. 10. From the fig, it can be seen that on increasing the compressor efficiency the exergetic efficiency also increases. The absorber and desorber temperatures are kept fixed at 75°C and 30°C respectively.

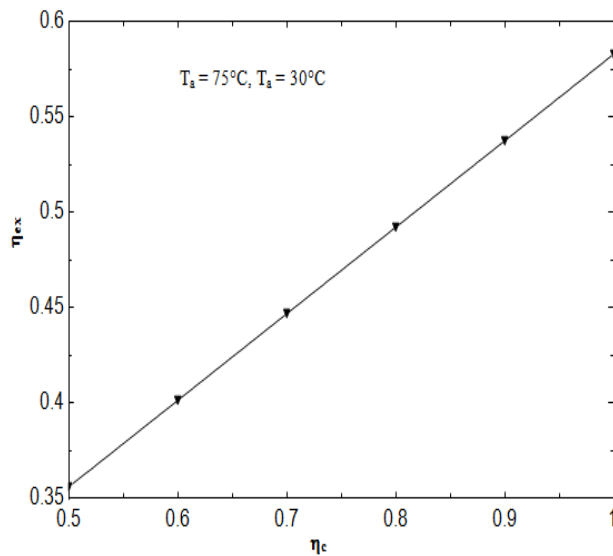


Fig 10. Variation of exergetic efficiency with compressor efficiency.

Since the heating capacity is also fixed, the numerator of the expression for exergetic efficiency of the system as given by eqn. 3.49 remains constant. With an increase in compressor efficiency, compressor input power reduces which leads to an increase in the exergetic efficiency of the system.

7.4 Variation of Exergy destruction ratio in the absorber (EDRa) with absorber temperature: A comparative picture of exergy losses in various system components reveals that the highest exergy loss occurs in the absorber. This section presents the variation of exergy destruction ratio in the absorber as a function of absorber temperature at various values of desorber temperature. The heat exchanger effectiveness, compressor efficiency and pressure ratio are fixed at the values of 0.7, 0.7, 5 respectively.

For a particular set of operating conditions, absorber has the highest exergy destruction ratio. As can be seen from Fig. 11, as the absorber temperature is increased, the exergy destruction in the absorber reduces for a fixed value of desorber temperature. At a fixed absorber temperature, EDRa is more at a higher value of desorber temperature. For a fixed value of reference temperature and the absorber heat load, the thermal exergy loss component as stated in Eqn. 3.39 increases with an increase in absorber temperature which in turn increases the exergy loss in the absorber.

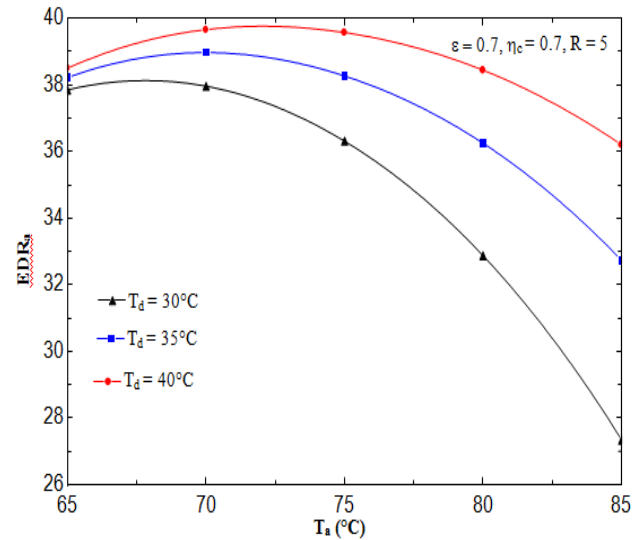


Fig 11. Variation of Exergy Destruction Ratio in the absorber with absorber temperature.

VI. CONCLUSION

Through a comprehensive energy and exergy analysis of the compression-absorption heat pump of 100 kW heating capacity considered in the present study, following conclusions can be drawn:

Computer simulations of the compression-absorption cycle showed that for every application, a value of the absorber temperature which maximizes the COP of the cycle can be determined. In the present work, at an absorber pressure of 20 bar and desorber pressure of 4 bar, the maximum value of COP is obtained corresponding to an absorber temperature of 60°C which is equal to 3.988.

On increasing the absorber temperature, keeping other parameters as constant, the COP of the heat pump first increases till it reaches an optimum value, which then starts decreasing with a further increase in the absorber temperature. Exergy efficiency values of the system were found to range from 35.67% to 40.68% at the absorber temperatures varying from 60 to 80 °C.

Exergetic efficiency of the system first increases with an increase in absorber temperature, reaches a maximum value which then starts decreasing with a further increase in the absorber temperature. On increasing the compressor efficiency and the heat exchanger effectiveness, the exergetic efficiency of the system increases. The highest exergy loss occurred in the absorber followed by compressor, desorber, expansion valve, and pump and solution heat exchanger.

Based on the comparative analysis of the exergy destruction ratio of all the components of the system, it can be concluded that the contribution of the exergy loss in the absorber to the total exergy loss is the highest. At an

absorber temperature of 60°C and a desorber temperature of 30°C, approximately 38.06% of the total exergy loss has been contributed by the absorber.

The second highest exergy loss equal to 30.62% for the similar set of operating conditions has occurred in the compressor which first increases upto a maximum value with an increase in absorber temperature and then starts decreasing with a further increase in the absorber temperature. At a higher value of desorber temperature, lower is the exergy destruction in the compressor. On increasing the isentropic efficiency of the compressor, the exergy destruction ratio of the compressor is reduced.

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