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ENERGY AND EXERGY ANALYSIS OF ABSORPTION- COMPRESSION CASCADE REFRIGERATION SYSTEM

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ABSTRACT

In this study, an Absorption- Compression Cascade Refrigeration, comprising of a VCR system in low temperature stage and a VAR system at the high temperature stage, is analyzed. CO_2, NH_3 and R134a have been considered as refrigerants in the compression stage and the H_2O-LiBr refrigerant absorbent pair in the absorption stage. The analysis has been realized by means of a mathematical model of the refrigeration system. The study presents the results obtained regarding the performance of the refrigeration system based on energy and exergy analysis. The comparative study helps to find out the best refrigerant and appropriate operation parameters. It is found in the study that cascade condenser, compressor and refrigerant throttle valve are the major source of exergy destruction.

INTRODUCTION

There are various applications which require temperature in the range of - 30°C to - 100°C. It includes food preservation, rapid freezing, ice production, storage of medical products, drugs and so on. It is not economical to obtain temperature below -40°C, using single stage vapour compression refrigeration (VCR) system or single stage vapour absorption refrigeration (VAR) system [1]. Therefore, in order to obtain such low temperatures cascade refrigeration system is the best choice. The conventionally employed two stage cascade compression refrigeration systems consume a lot of electricity. About 56% of total electricity generation in India is done through fossil fuel burning [2] which causes emission of harmful gases like carbon dioxide and oxides of sulphur and nitrogen. Also, the refrigerants used in these conventional VCR systems are the major cause of ozone depletion and global warming. Therefore such a cooling technology is needed which can at least reduce the electricity consumption, if it cannot eliminate high grade energy consumption completely.

Vapour absorption refrigeration systems cascaded with vapour compression systems are capable of reducing electricity consumption together with maintaining high coefficient of performance (COP). This refrigeration system would decrease the electricity consumption compared to the two stages compression systems, since it is only required to operate the compression system at the low stage, whereas the absorption system is driven by heat. The heat driven VAR systems are of different configurations which can be operated by heat sources of temperature varying from 60°C to 200°C [3]. Generally, single effect VAR cycle is employed in cascade system and it requires generator temperature between 80-120°C. However, if heat is available at lower temperature then half effect configuration [4] can be used as it requires generator temperature between 60-80°C [5] and if heat is available at higher temperature then double effect [6], triple effect [7] and GAX [8] absorption refrigeration cycles can be used. The presence of flexibility of choosing the suitable configuration of VAR depending upon the temperature of available heat allows sustainable utilization of energy resources.

Fernández-Seara et al. [9] carried out a study on compression-absorption cascade system. Ammonia-water was the working substance in absorption refrigeration cycle whereas carbon dioxide was used as a refrigerant in compression cycle. The COP of 0.253 was reported by them. Kairouani and Nehdi [10] proposed a geothermal energy driven absorption-compression cascade refrigeration system and reported the COP of 5.4-6.2 (excluding pump work and generator heat load). Garimella et al. [11] developed a computational model of a waste heat driven single effect LiBr/H_2O absorption-subcritical CO_2 compression cycle for megawatt scale low temperature (-40 °C) cooling for high heat flux electronic application. They reported that this novel cascade cooling system consumed 31% less electricity than the equivalent VCR system. Cimsit and Ozturk [12] used different refrigerants and working pairs for the analysis of compression-absorption cascade systems. Ammonia, R134a, R-410A were used in compression section while NH_3/H_2O and LiBr/H_2O were used as working substances in absorption section of cascade cycle. It was concluded that electricity consumption in cascade systems is 48-51% lower than conventional VCR systems. They also reported that LiBr/H_2O based cascade system outperformed NH_3/H_2O based cascade refrigeration system by registering 33% higher COP. Wang et al. [13] studied the solar assisted R134a compression-LiBr/H_2O absorption cascade refrigeration system. Electric power consumption was reported to be lower by 50% in comparison with VCR system. Jain et al. [2] performed the first law and second law based thermodynamic analysis of cascaded vapour compression-absorption system (CVCAS) which consists of single effect VAR system coupled with VCR system. The electric power consumption in CVCAS was 61% lower than that in VCR system for same operating conditions. Colorado and Velazquez [14] carried out exergy based thermodynamic analysis of compression-absorption refrigeration cycle using NH_3, CO_2 and R134a in VCR section and H_2O-LiBr in VAR section so as to find out best working substance and suitable operating parameters. It was shown that highest irreversibility occurs in cascade condenser, accounting for around 19.96%, 19.31% and 13.28% of the total irreversibilities using NH_3, CO_2 and R134a respectively. A thermodynamic analysis of compression-absorption cascade refrigeration system using modified Gouy-Stodola equation was carried out by Jain et al. [15]. In their study they obtained the optimum temperature of cascade condenser which corresponds to minimum irreversibility and maximum COP of the system. Further, a comparative study of compression-absorption cascade refrigeration system and two stage vapour compression refrigeration system (TSVCS) reveals that primary energy consumption of compression absorption cascade system is 60.6% less and electrical COP is 153.6% more than that of TSVCS.

It is obvious from the literature review that though a lot of work on energy based analysis is reported, yet the exergy based thermodynamic analysis of absorption-compression cascade refrigeration system is limited. In this study attention is focussed on the components which are the major sites of exergy destruction. It is also endeavoured to find out the effects of various operating and design parameters on exergy destruction in different components, COP and exergetic efficiency.

SYSTEM DESCRIPTION

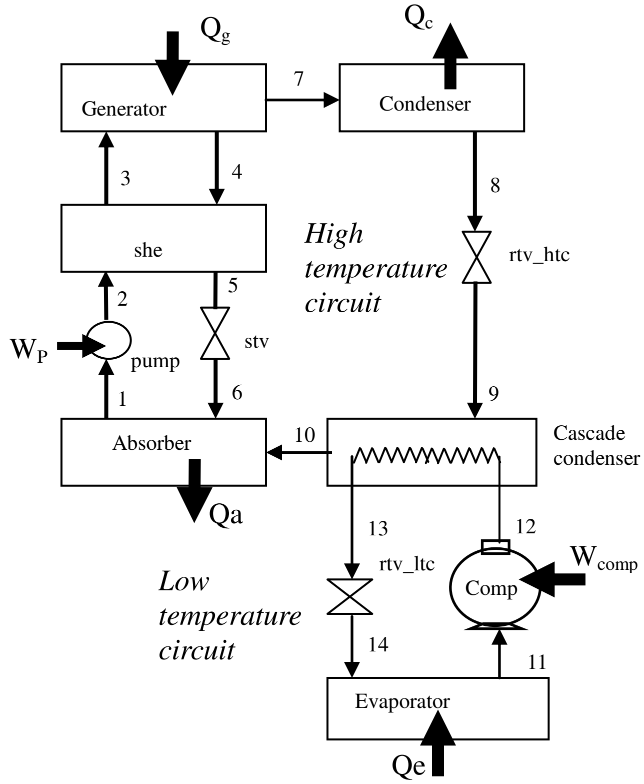


Figure 1. Absorption-compression cascade refrigeration system

A cascade refrigeration system in general comprises of a low temperature circuit (LTC) and a high temperature circuit (HTC). Actual cooling load is supplied to LTC while its heat of condensation acts as a cooling load to HTC which ultimately rejects heat to the surroundings. Thus LTC and HTC are coupled through a common heat exchanger referred to as cascade condenser, which acts as condenser for LTC refrigeration cycle and evaporator for HTC refrigeration cycle. In case of absorption-compression cascade refrigeration system, VAR cycle is utilized in HTC and VCR cycle is employed in LTC. In this study single effect LiBr-H_2O vapor absorption refrigeration cycle is used in HTC and R134a, CO_2 and NH_3 have been considered as refrigerants in the compression stage as shown in Fig.1. The compression system comprises of the evaporator, compressor, condenser and an expansion device. The major components of single effect VAR system are the absorber, generator, condenser, evaporator, solution heat exchanger, pump, solution throttle valve and a refrigerant throttle valve. The single effect absorption cycle is separately described by authors [16].

THERMODYNAMIC ANALYSIS

Assumptions

(a) The system operates under steady state conditions.

(b) Cascade condenser is perfectly isolated.

(c) Heat losses and pressure drops in connecting lines and various components are neglected.

(d) The subcooling and superheating in discharge and suction lines are neglected.

(e) The expansion process is isenthalpic.

(f) The solutions at the exit of generator and absorber are saturated in equilibrium at their respective concentrations and temperatures.

(g) Reference environmental temperature and pressure are 25°C and 101.3 kPa respectively.

The thermodynamic analysis of compression absorption system involves the principles of mass conservation, energy conservation and exergy balance.

Mass Balance

The mass flow rate through each component of low temperature circuit is $\dot{m}_{r,ltc}$. It is calculated using eqn. (1).

$$\dot{Q}_{evap} = \dot{m}_{r,ltc}\left(h_{11} - h_{14}\right) \tag{1}$$

Mass balance at absorber or generator

$$\dot{m}_s = \dot{m}_r + \dot{m}_w \tag{2}$$

Here, $\dot{m}_r$ is mass flow rate of refrigerant through condenser and evaporator.

Energy Balance

$$\dot{Q}_{abs} = \dot{m}_{10}h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1 \tag{3}$$

$$\dot{Q}_{gen} = \dot{m}_7 h_7 + \dot{m}_4 h_4 - \dot{m}_3 h_3 \tag{4}$$

$$\dot{Q}_{cond} = \dot{m}_7\left(h_7 - h_8\right) \tag{5}$$

$$\dot{Q}_{she} = \dot{m}_3\left(h_3 - h_2\right) = \dot{m}_4\left(h_4 - h_5\right) \tag{6}$$

$$\dot{W}_{pump} = \dot{m}_2\left(h_2 - h_1\right) \tag{7}$$

$$\dot{W}_{comp} = \dot{m}_{r,ltc}\left(h_{12} - h_{11}\right) \tag{8}$$

Exergy Balance

By the application of second law of thermodynamics, exergy destruction in each component of the absorption-compression cascade refrigeration system is obtained and furnished below:

$$\dot{E}D_{abs} = \dot{m}_{10}\left(h_{10} - T_0 s_{10}\right) + \dot{m}_6\left(h_6 - T_0 s_6\right) - \dot{m}_1\left(h_1 - T_0 s_1\right) \tag{9}$$

$$\dot{E}D_{gen} = \dot{m}_3\left(h_3 - T_0 s_3\right) - \dot{m}_4\left(h_4 - T_0 s_4\right) - \dot{m}_7\left(h_7 - T_0 s_7\right) + \dot{Q}_{gen}\left(1 - T_0 / T_{gen}\right) \tag{10}$$

$$\dot{E}D_{cond} = \dot{m}_7\left(h_7 - T_0 s_7\right) - \dot{m}_8\left(h_8 - T_0 s_8\right) \tag{11}$$

$$\dot{E}D_{cc} = \dot{m}_r\left((h_9 - h_{10}) - T_0(s_9 - s_{10})\right) + \dot{m}_{r,ltc}\left((h_{12} - h_{13}) - T_0(s_{12} - s_{13})\right) \tag{12}$$

$$\dot{E}D_{she} = \dot{m}_s\left((h_2 - h_3) - T_0(s_2 - s_3)\right) + \dot{m}_w\left((h_4 - h_5) - T_0(s_4 - s_5)\right) \tag{13}$$

$$\dot{E}D_{rtv,htc} = \dot{m}_r T_0\left(s_9 - s_8\right) \tag{14}$$

$$\dot{E}D_{stv} = \dot{m}_w T_0\left(s_6 - s_5\right) \tag{15}$$

$$\dot{E}D_{comp} = \dot{m}_{r,ltc} T_0\left(s_{12} - s_{11}\right) \tag{16}$$

$$\dot{E}D_{evap} = \dot{m}_{r,ltc}\left((h_{14} - h_{11}) - T_0(s_{14} - s_{11})\right) + \dot{Q}_{evap}\left(1 - T_0 / T_{evap}\right) \tag{17}$$

$$\dot{E}D_{rtv,ltc} = \dot{m}_{r,ltc} T_0\left(s_{14} - s_{13}\right) \tag{18}$$

$$\dot{E}D_{total} = \dot{E}D_{abs} + \dot{E}D_{gen} + \dot{E}D_{cond} + \dot{E}D_{cc} + \dot{E}D_{she} + \dot{E}D_{rtv,htc} + \dot{E}D_{stv} + \dot{E}D_{comp} + \dot{E}D_{evap} + \dot{E}D_{rtv,ltc} \tag{19}$$

Performance indices

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen} + \dot{W}_{pump} + \dot{W}_{comp}} \tag{20}$$

Exergetic efficiency

$$\eta_{ex} = \frac{\dot{Q}_{evap}\left|1 - {T_0}/{T_r}\right|}{\dot{Q}_{gen}\left(1 - {T_0}/{T_{gen}}\right) + \dot{W}_{pump} + \dot{W}_{comp}} \tag{21}$$

Exergy destruction ratio

In order to identify and compare the sites of thermodynamic inefficiencies from the point of view of exergy analysis, the exergy destruction ratio $(Y_{d,k})$ for each component is calculated. It is defined as the ratio of exergy destruction rate in a component to the total exergy destruction rate of the system [17] and it is expressed as:

$$Y_{d,k} = {\dot{E}D_k}/{\dot{E}D_{total}} \tag{22}$$

where 'k' denotes any component.

Initial/operating parameters

The parameters assumed for computation of results are mentioned in Table 1.

Table 1. Parameters for the analysis

S. No.	Parameters	Value
1.	Cooling capacity ($\dot{Q}_{evap}$)	100 kW
2.	Isentropic efficiency of compressor (η_{comp})	60-80 %
3.	Evaporator temperature (T_{evap})	-35 to -55 °C
4.	Cascade condenser temperature (T_{cc})	2 to 16 °C
5.	Generator temperature (T_{gen})	80 to 110 °C
6.	Absorber temperature (T_{abs})	35 to 45 °C
7.	Condenser temperature (T_{cond})	35 to 45 °C
8.	Effectiveness of solution heat exchanger (ε_{she})	0.6-0.9
9.	Approach in cascade condenser (OT)	0 to10 °C
10.	Difference between evaporator and space temperature (DT)	7 °C

Model validation

A simple steady state simulation model based on sequential modular approach has been developed and implemented in a computer program using EES software [18]. The model equations are formulated from species, mass, energy and exergy balances. The thermodynamic model of absorption-compression cascade refrigeration system developed in this work is validated by the numerical data of Cimsit and Ozturk [12]. R134a and $LiBr-H_2O$ are considered as working substances in VCR and VAR cycles for validation.

Table 2 clearly indicates that there exist good agreements between the present data and those provided by Cimsit and Ozturk [12]. The maximum error is $\pm$ 2.03%, which may be attributed to the usage of different correlations for the calculation of thermophysical properties of $LiBr-H_2O$.

Table 2. Comparison of performance data of present model with that of Cimsit and Ozturk [12]

Parameters	Present model	Cimsit and Ozturk [12]	Difference (%)	Operating Parameters
$\dot{Q}_{abs}$ (kW)	74.27	72.76	2.03	T_{cond} = 40°C
$\dot{Q}_{gen}$ (kW)	77.98	76.45	1.96	T_{gen} = 90°C
$\dot{Q}_{cond}$ (kW)	62.09	61.06	1.66	T_{cc} = 10°C
$\dot{W}_{comp}$ (kW)	8.39	8.25	1.67	T_{evap} = -10°C
COP	0.579	0.590	-1.90	ε_{she} = 0.6
				η_{comp} =0.72
				$\dot{Q}_{evap}$ = 50 kW

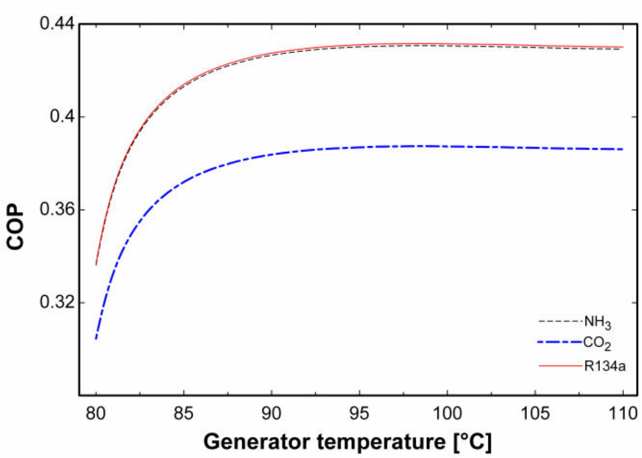
RESULTS AND DISCUSSION

For the base case the operating parameters considered are as follows: generator, condenser, evaporator, cascade condenser temperatures are 85°C, 40°C, -45°C and 7°C. The absorber temperature is same as the condenser temperature. The approach in cascade condenser is 7°C, Isentropic efficiency of compressor is 0.8 and effectiveness of solution heat exchanger is 0.7. In order to perform the parametric analysis one parameter is varied within the given limits while others are kept constant.

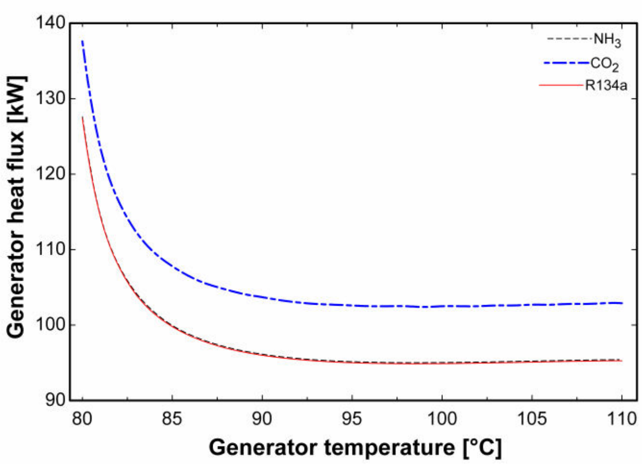
First law analyses

Figures 2(a), (b) and (c) respectively show the variation of COP, generator heat load and absorber heat load with the generator temperature. It is clear from Fig. 2(a) that absorption-compression cascade refrigeration system achieves maximum COP at a particular generator temperature. The maximum values of COP for NH_3, CO_2 and R134a are 0.4307, 0.3874 and 0.4316 respectively and the corresponding generator temperature is 98°C. The COP of the system operating with R134a is approximately 10% higher than that obtained with CO_2. The COP of NH_3 system is marginally lower than that of R134a based system. The variation of COP with generator temperature depends inversely on the variation of generator heat load for constant cooling capacity system. It is obvious from Fig 2(b) that there exists a generator temperature corresponding to which generator heat load is minimum. The value of generator temperature corresponding to minimum generator heat load is equal to that corresponding to maximum COP. Considering the absorber heat load, shown in Fig. 2(c), it can be stated that highest amount is delivered in case of CO_2 whereas lowest in case of R134a.

Fig. 3(a) shows the COP of absorption-compression cascade refrigeration system as a function of evaporator temperature. As evaporator temperature is increased from -55°C to -35°C, increase in COP is registered.



(a)



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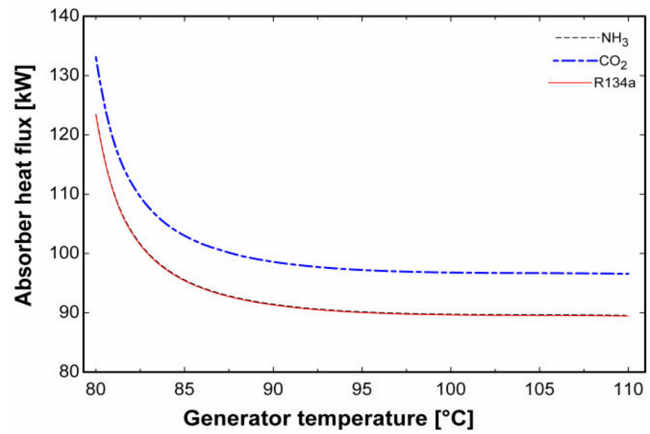
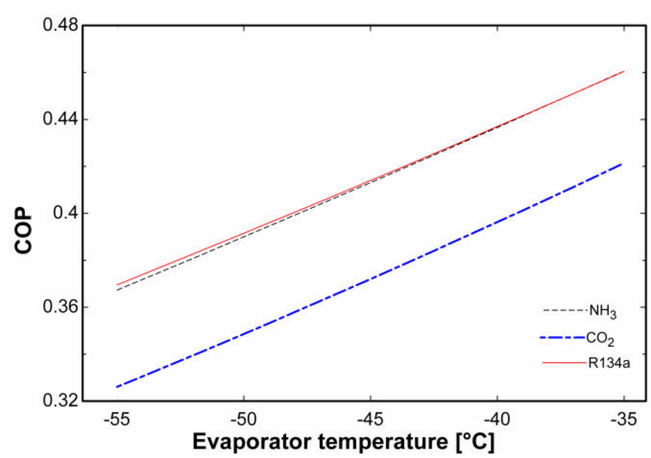
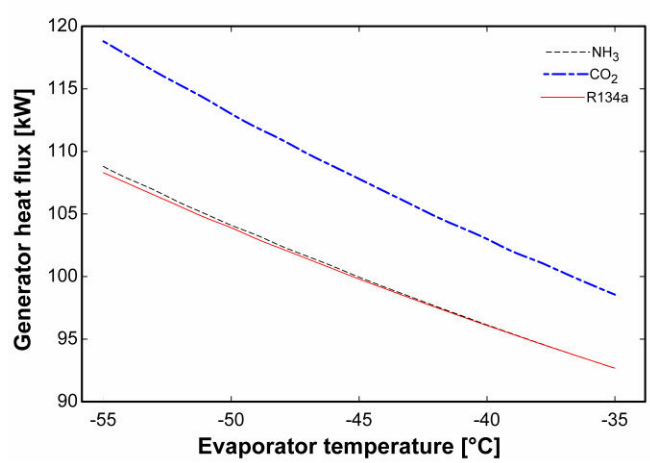


(c)

Figure 2. (a) COP, (b) generator heat load and (c) absorber heat load against generator temperature



(a)



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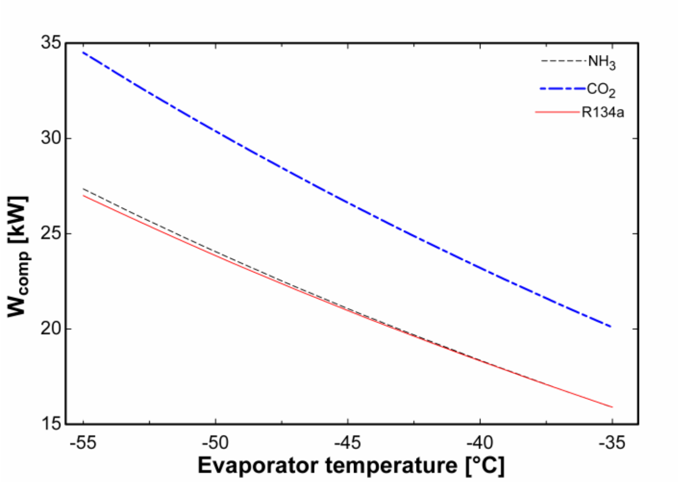


(c)

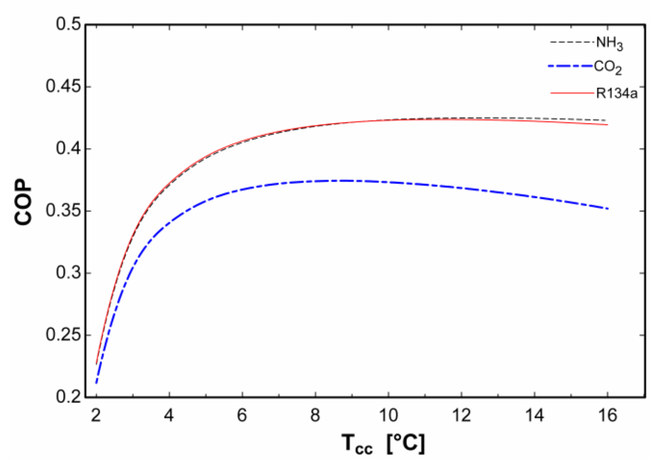
Figure 3. (a) COP, (b) generator load and (c) work of compression against evaporator temperature

The COP of R134a based system increases from 0.3695 to 0.4605 whereas the COP of CO_2 based system increases from 0.3261 to 0.4214. The COP of NH_3 based system is lower but closer to R134a based system. In Fig. 3(b) it can be seen that as evaporator temperature decreases from -35°C to -55°C, heat required in generator increases from 92 to 108 kW, 98 to 119 kW and 102 to 109 kW respectively for R134a, CO_2 and NH_3 based systems. The decrease in compressor work of absorption-compression cascade refrigeration system with increase in evaporator temperature is shown in Fig. 3(c). As evaporator temperature increases from -55°C to -35°C, compressor work decreases by almost 40% for all the refrigerants. It means rise in evaporator temperature by 1°C can reduce the electricity consumption by 2%. Carbon dioxide based system requires highest electrical input (20-35 kW) in the form of compressor work while R134a based system needs minimum electrical energy (16-27 kW).

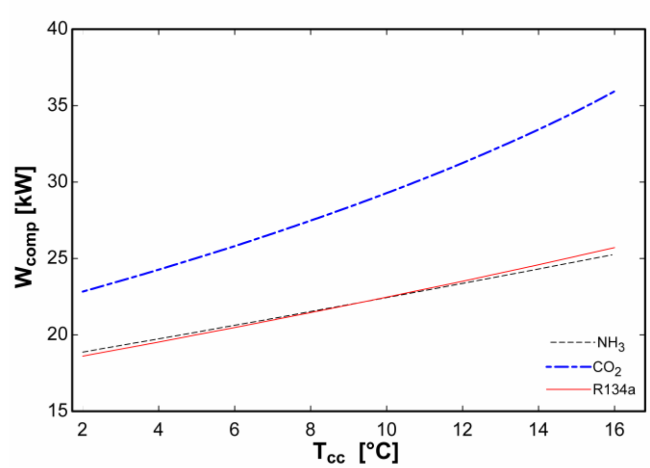
The cascade condenser temperature is one of the most important design parameters as it has intense effect on the performance of the absorption-compression cascade refrigeration system. Its lowest value is restricted to be above 0°C when LiBr-H_2O is the working substance in VAR subsystem because it depends on the freezing point of water. From Fig. 4(a) it can be seen that with the increase in cascade condenser temperature from 2 to 16°C, COP first increases, achieves a maximum and then reduces. The increase in cascade condenser temperature causes increase in absorber pressure, increase in pressure ratio across the compressor and increase in mass flow rate in VCR subsystem. The first two factors account for increase in compressor power as shown in Fig. 4(b) while the last factor i.e. increase in absorber pressure accounts for decrease in strong solution concentration (X_s). The weak solution concentration (X_w) remains constant. Hence, solution circulation ratio (=X_w/ (X_w-X_s)) decreases. The reduction in solution circulation ratio decreases the heat required in the generator as shown in Fig. 4(c). Thus, COP of cascade refrigeration system may increase or decrease depending upon the increase in compressor power requirement and reduction in generator heat duty.

It is clear from Figs. 4(a), (b) and (c) that maximum values of COP occur at different cascade condenser temperature for different refrigerants. In case of NH_3, maximum COP is 0.4249 and it occurs at 12.47°C whereas for R134a and CO_2 the maximum COP values are 0.4237 and 0.3743, occurring at 11.45°C and 8.64°C respectively.

For all values of cascade condenser temperature, the performance of CO_2 based system is inferior to the performance of the other two systems. However, the NH_3 based system and R134a based system perform differently at different cascade condenser temperature, though the difference is marginal. It can be observed that NH_3 based system outperforms R134a based system at higher cascade condenser temperature, particularly above 10°C for the conditions considered in this study.



(a)



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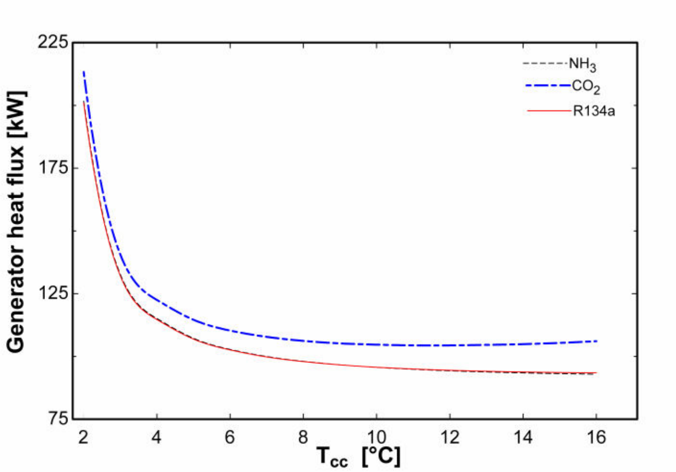


(c)

Figure 4. (a) COP, (b) generator load and (c) work of compression against cascade condenser temperature

Second law analyses

Fig. 5 shows the exergetic efficiency and exergy destruction rates of R134a, ammonia and carbon dioxide based systems versus generator temperature at design conditions. It indicates a maximum value of exergetic efficiency ($\eta_{ex,max}$) and a minimum value of exergy destruction rate as T_{gen} increases from 80°C to 110°C.The variation of exergetic efficiency with generator temperature can be explained in similar way as variation of COP with T_{gen}. However, temperature of the heat source is additional factor which govern the exergetic efficiency, resulting in higher slope of exergetic efficiency as compared to corresponding COP. The maximum values of exergetic efficiency for NH_3, CO_2 and R134a based absorption-compression cascade refrigeration system are 35.43%, 29.99% and 35.56% respectively, occurring at same generator temperature of 85°C. The corresponding minimum values of exergy destruction rates are 24.43 kW, 31.29 kW and 24.30 kW for NH_3, CO_2 and R134a based cascade systems.

The comparison of Fig. 5 and Fig. 2(a) reveals that maximum exergetic efficiency and maximum COP occur at different generator temperatures. Also, an increase of 11.70%, 10.66% and 11.72% in the second law efficiency of NH_3, CO_2 and R134a based systems is observed as generator temperature is increased from 80°C to 85°C.

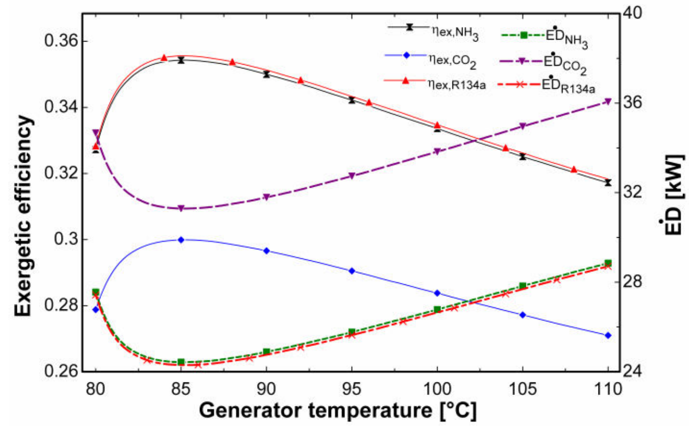
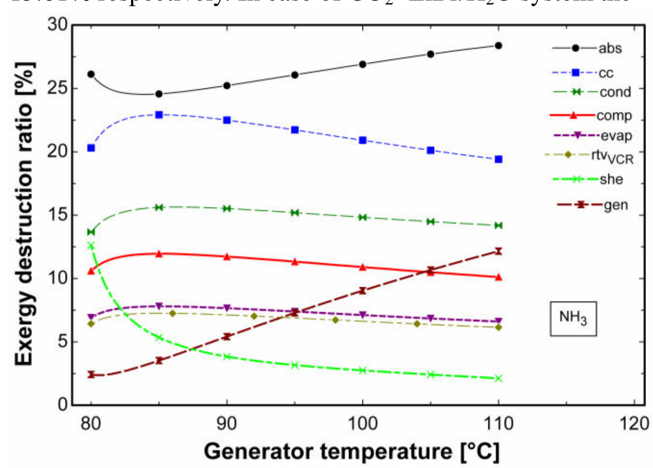
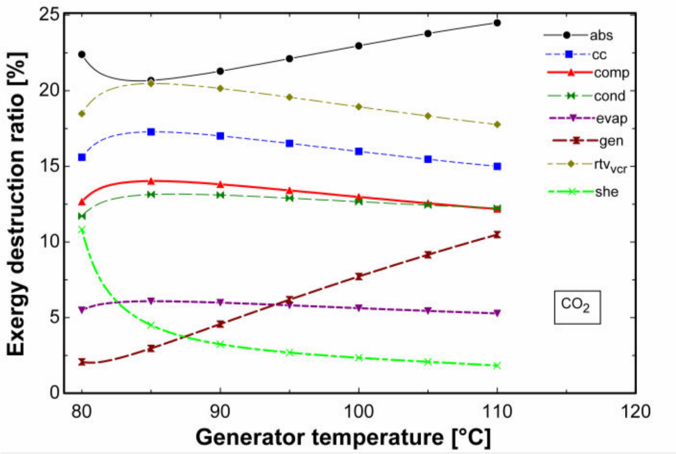


Figure 5. Exergetic efficiency and total exergy destruction rate against generator temperature

Figs. 6(a), (b) and (c) respectively show the variation of exergy destruction ratio of the main components of NH_3-LiBr/H_2O, CO_2- LiBr/H_2O and R134a- LiBr/H_2O absorption-compression cascade refrigeration systems with generator temperature. At the generator temperature corresponding to maximum exergetic efficiency, the sites of major irreversibilities are different for different working substances. The absorber, the cascade condenser and the condenser are the major contributors of exergy destruction for NH_3-LiBr/H_2O system with exergy destruction ratio of 24.57%, 22.92% and 15.61% respectively. In case of CO_2- LiBr/H_2O system the



(a)



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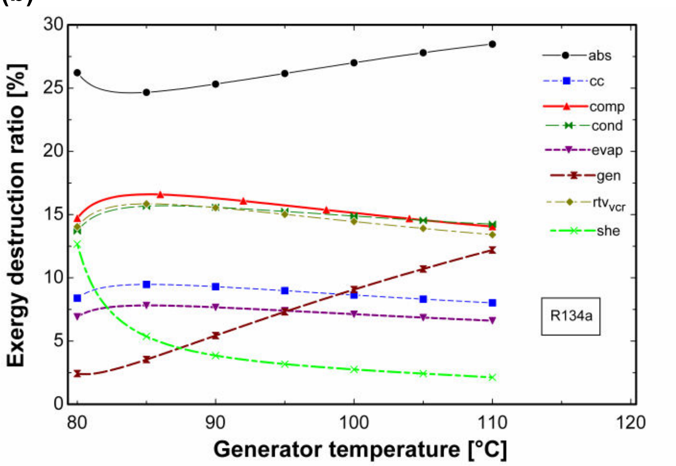


(c)

Figure 6. Exergy destruction ratio of main components against generator temperature for (a) Ammonia (b) Carbon dioxide and (c) R134a refrigerants

major sources of exergy destruction are absorber (20.68%), RTV_{vcr} (20.47%) and cascade condenser (17.28%) whereas for R134a- $LiBr/H_2O$ cascade system absorber (24.63%), compressor (16.56%) and RTV_{vcr} (15.82%) are the prominent locations of exergy destruction.

Fig. 7 shows exergetic efficiency as a function of cascade condenser temperature. As cascade condenser temperature increases from 2°C to 16°C, exergetic efficiency attains a maximum value at a particular value. The NH_3-$LiBr/H_2O$ cascade system attains maximum exergetic efficiency (35.43%) at cascade condenser temperature of 7°C whereas CO_2- $LiBr/H_2O$ and R134a- $LiBr/H_2O$ systems attain maximum exergetic efficiencies at cascade temperature of 5°C and 6°C respectively. The corresponding maximum exergetic efficiencies of CO_2- $LiBr/H_2O$ and R134a- $LiBr/H_2O$ systems are 30.30% and 35.58%. Thus it can be said that exergetic efficiency attains maximum value at different cascade condenser temperatures, depending on the working substance considered. Furthermore, it is seen from the figure that the poorest exergetic performance is exhibited by CO_2- $LiBr/H_2O$, whereas the best performance may be shown by either of NH_3-$LiBr/H_2O$ and R134a- $LiBr/H_2O$ system depending upon the value of cascade condenser temperature. At higher values of cascade condenser temperature NH_3-$LiBr/H_2O$ outperform R134a- $LiBr/H_2O$ cascade system.

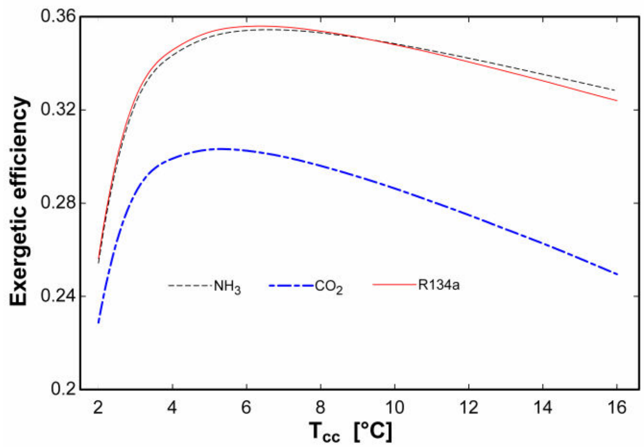
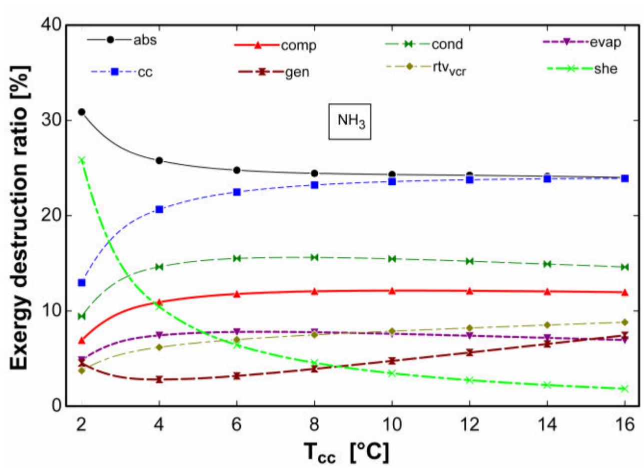
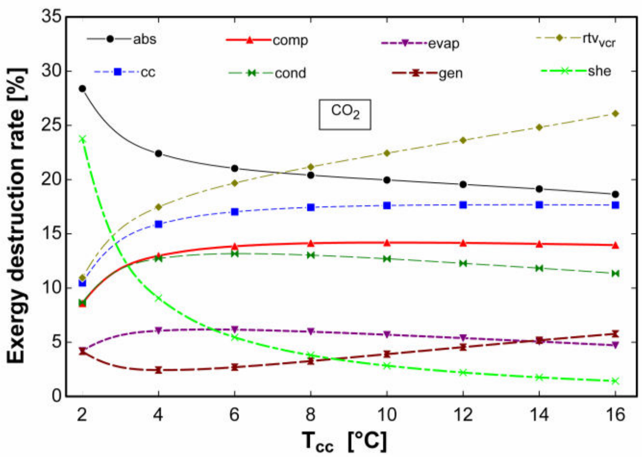


Figure 7. Exergetic efficiency against cascade condenser temperature

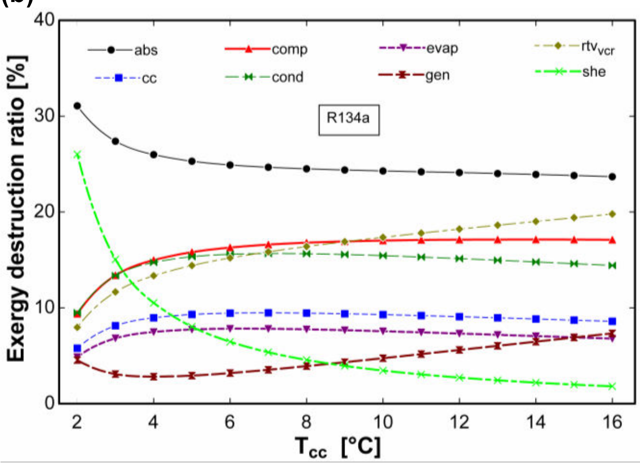
Figs. 8(a), (b) and (c) show the exergy destruction ratio of various components as a function of cascade condenser temperature. At cascade condenser temperature of 7°C (corresponding to maximum exergetic efficiency for NH_3-$LiBr/H_2O$ system), absorber accounts for maximum exergy destruction followed by cascade condenser, condenser and compressor. For CO_2-$LiBr/H_2O$ system, at cascade condenser temperature of 5°C (corresponding to maximum exergetic efficiency), the major sources of exergy destruction are identified as absorber, RTVvcr and cascade condenser. In case of R134- $LiBr/H_2O$ system the main sites of exergy destruction are found to be absorber, compressor and condenser.



(a)



(b)



(c)

Figure 8. Exergy destruction ratio of main components against cascade condenser temperature for (a) Ammonia (b) Carbon dioxide and (c) R134a refrigerants

Fig. 9 reveals the variation of exergetic efficiency with evaporator temperature. It is observed that with the increase in evaporator temperature the rate of decrease in exergetic efficiencies of NH_3-LiBr/H_2O and R134a-LiBr/H_2O are respectively 3.26% and 4.23%, whereas for CO_2-LiBr/H_2O cascade system it is 1.5% only.

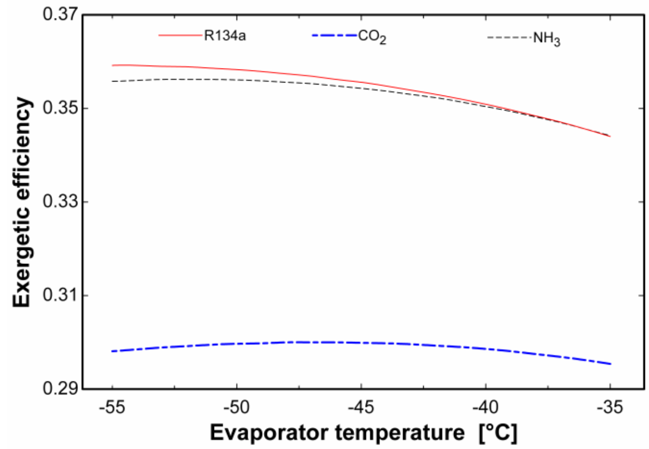
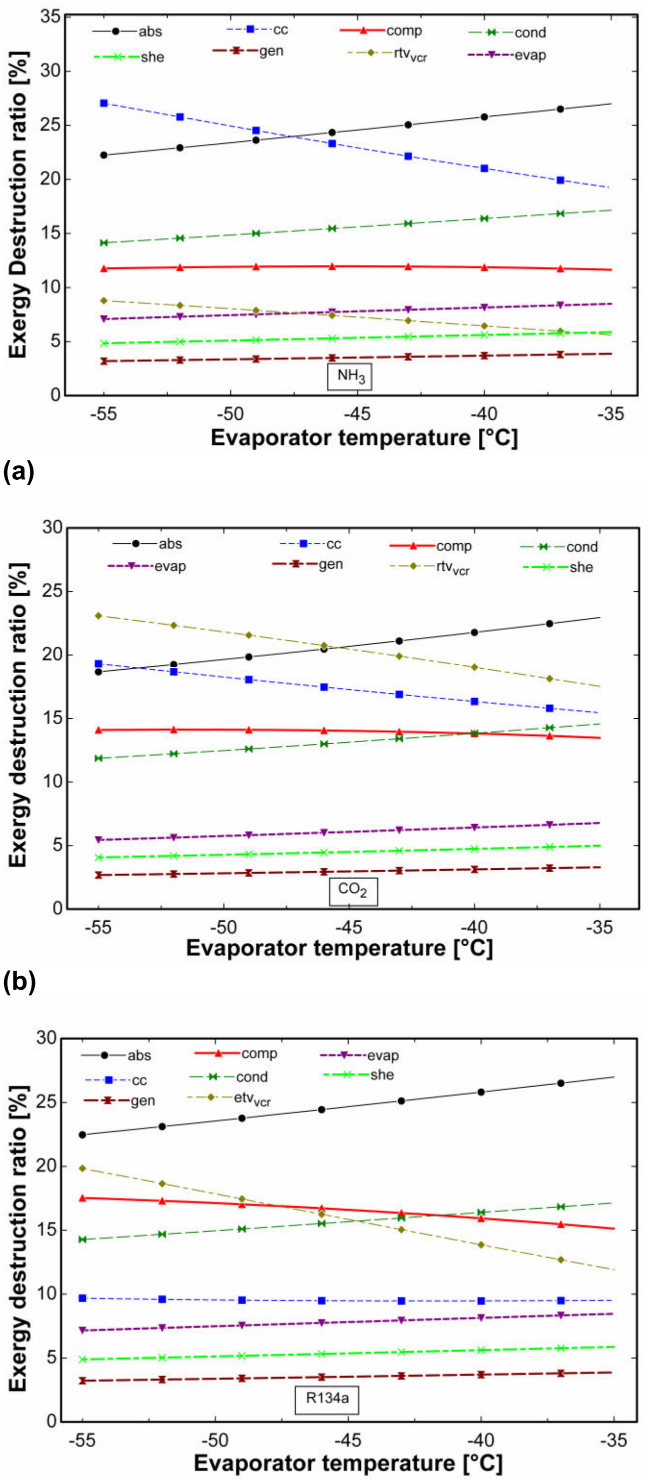


Figure 9. Exergetic efficiency against evaporator temperature

Figs. 10(a), (b) and (c) respectively show exergy destruction ratio of various components of NH_3, CO_2 and R134a based refrigeration systems as a function of evaporator temperature. As evaporator temperature increases from -55°C to -35°C, the exergy destruction ratio of absorber, condenser, generator, evaporator and solution heat exchanger increases while that of refrigerant throttle valve of VCR subsystem decreases for all the three refrigerants. However, the variation in exergy destruction ratio is of varying degree. In case of cascade condenser and compressor, the trends are quite diverse and depend on the refrigerant used. Exergy destruction ratio of cascade condenser decreases for NH_3 and CO_2 based systems whereas for R134a based system it first attains a minimum value and then begins to rise. Similarly, exergy destruction ratio of compressor first attains a maximum value and then falls for NH_3 and CO_2 based systems while it reduces continuously for R134a based system.

Exergy destruction ratio of various components for NH_3-LiBr/H_2O is shown in Fig. 10(a). It is observed that at low evaporator temperatures cascade condenser is the major source of irreversibility while at higher values of evaporator temperature it is absorber in which maximum exergy destruction takes place. Fig. 10(b) shows that refrigerant throttle valve of VCR and absorber are the main sites of irreversibilities at evaporator temperature of -55°C and -35°C respectively. Absorber remains the major source of exergy destruction throughout the evaporator change for R134a-LiBr/H_2O cascade refrigeration system.



(a)

(b)

(c)

Figure 10. Exergy destruction ratio of main components against evaporator temperature for (a) Ammonia (b) Carbon dioxide and (c) R134a refrigerants

Fig. 11 shows the variation of exergetic efficiency with compressor isentropic efficiency. As expected, both the COP and the exergetic efficiency improves with the increase in compressor isentropic efficiency. With the increase in compressor isentropic efficiency from 0.6 to 0.9, the COP (not shown in figure) of NH_3, CO_2 and R134a based systems increase from 0.3625 to 0.4333, from 0.3210 to 0.3928 and from 0.3634 to 0.4341 respectively. Correspondingly, exergetic efficiencies increase from 28.82% to 38.38% for NH_3, from 24.08% to 32.66% for CO_2 and from 28.93% to 38.50% for R134a systems as shown in Fig. 11. Thus, it is clear that effect of compressor isentropic efficiency on the performance of absorption-compression cascade system is quite significant.

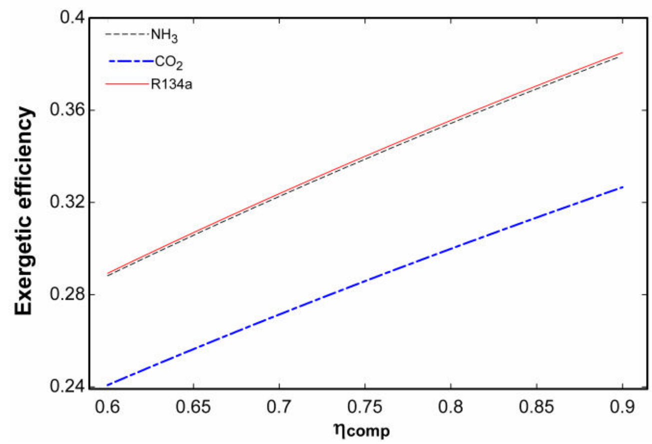
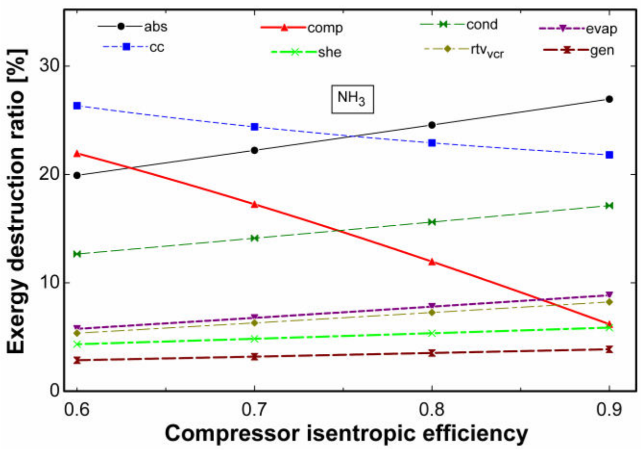
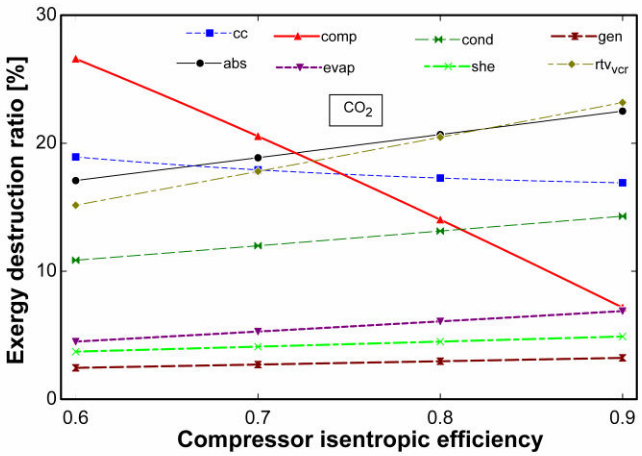


Figure 11. Exergetic efficiency versus compressor isentropic efficiency

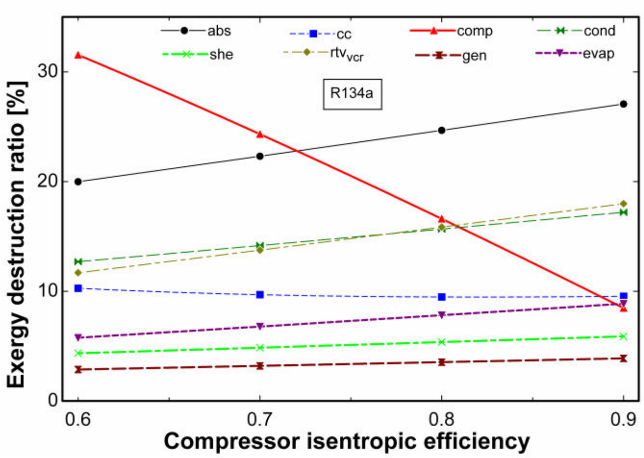
Fig. 12(a), (b) and (c) respectively show the variation of exergy destruction ratio of various components of NH_3-LiBr/H_2O, CO_2--LiBr/H_2O and R134a--LiBr/H_2O cascade systems with compressor isentropic efficiency. It is found in the study that as isentropic efficiency of the compressor drops from 0.9 to 0.6 exergy destruction rate of compressor in particular and the exergy destruction rate of the absorption-compression cascade refrigeration system as a whole y increase substantially. For NH_3-LiBr/H_2O system, compressor irreversibility increases from 1.331 kW to 7.267 kW while total irreversibility increases from 21.53 kW to 33.11 kW. For CO_2--LiBr/H_2O system, compressor and total irreversibilities increase from 1.978 kW to 11.24 kW and 27.63 kW to 42.26 kW respectively. Similarly, in case of R134a--LiBr/H_2O cascade system, exergy destruction rate of the compressor increases from 1.814 kW to 10.39 kW and total exergy destruction rate increases from 21.42 kW to 32.93 kW as isentropic efficiency of compressor reduces to 0.6 from 0.9.



(a)



(b)



(c)

Figure 12. Exergy destruction ratio of main components against compressor isentropic efficiency for (a) Ammonia (b) Carbon dioxide and (c) R134a refrigerants

CONCLUSIONS

The following conclusions are drawn from the present study:

(a) The maximum value of COP and exergetic efficiency occur corresponding to different generator temperatures. The maximum exergetic efficiency occurs at a temperature lower than the generator temperature corresponding to maximum COP.

(b) The generator temperature has greater impact on COP while exergetic efficiency is comparatively less affected by it. An increase of around 28% in COP is found with the increase in generator temperature whereas the corresponding increase in exergetic efficiency is about 11%.

(c) Maximum COP and maximum exergetic efficiency occur at different cascade condenser temperature for different refrigerants in VCR subsystem. Maximum COP occurs at higher cascade condenser temperature than at which maximum exergetic efficiency is obtained.

(d) The sites of highest irreversibilities are different for different refrigerants. For NH_3-LiBr/H_2O system it is cascade condenser, for CO_2-LiBr/H_2O system it is refrigerant throttle valve of VCR subsystem and in case of R134a-LiBr/H_2O system it is compressor.

(e) At design point, R134a-LiBr/H_2O absorption-compression cascade refrigeration system is the best performer from the view point of both first law and second law of thermodynamics.

NOMENCLATURE

COP	Coefficient of performance
$\dot{ED}$	Exergy destruction rate (kW)
h	Specific enthalpy ($kJ.kg^{-1}$)
HTC	High temperature circuit
LTC	Low temperature circuit
$\dot{m}$	Mass flow rate ($kg.s^{-1}$)
$\dot{Q}$	Heat transfer rate (kW)
s	Specific entropy($kJ.kg^{-1}.K^{-1}$)
T	Temperature (°C or K)
VAR	Vapour absorption refrigeration
VCR	Vapour compression refrigeration
Y_d	Exergy destruction ratio
$\dot{W}$	work transfer rate (kW)
Greek letters	
ϵ	Effectiveness of heat exchanger
η	efficiency
η_{ex}	Exergetic efficiency
Subscripts	
0	Reference state
1, 2…	State points
a, abs	absorber
c,cond	Condenser

cc	Cascade condenser
comp	Compressor
e, evap	Evaporator
g, gen	Generator
htc	High temperature circuit
ltc	Low temperature circuit
p	pump
r	Refrigerant, room
rtv	Refrigerant throttle valve
s	Strong solution
she	Solution heat exchanger
stv	Solution throttle valve
w	Weak solution

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