Exergy investigation of a vapor compression system comprising ejector for isentropic expansion

A Baruah, D K Saini and G Sachdeva
Department of Mechanical Engineering, National Institute of Technology Kurukshetra, Haryana-136119, India

Corresponding author’s e-mail: arunjoyb@gmail.com

Abstract. A modified vapor compression refrigeration system using ejector as an expansion device has been studied in this paper. The new system gives higher cooling effect and requires lesser compressor work than typical vapor compression system under similar operating conditions. Exergy analysis has been done to determine the scope of performance improvement in various components of the modified system. Three refrigerants namely R134a, R407C and R410A have been taken for the analysis due to their wide usage in the commercial refrigeration and air conditioning systems. The system is found to have highest 2nd law efficiency of 53.32% with R134a at 35°C condenser temperature and 5°C evaporator temperature. The increase in condenser temperature increases the total irreversibility of the system. The maximum exergy destruction has been observed in condenser for all the refrigerants. The analysis has been carried out in Engineering Equation Solver.

1. Introduction
Vapor compression (VC) system is the most commonly used system in refrigeration and air-conditioning devices. However, VC system consumes significant amount of electricity in the compressor to produce cooling effect. Many developing nations including India produce less electrical energy than the actual requirement, therefore millions of people have limited or no access to electricity. Moreover, the electricity generation using fossil fuels deteriorates the environment by producing harmful gases like CO₂, nitrogen oxides, sulfur oxides etc. [1]. So any improvement possible in the VC system that results in higher efficiency or lesser electricity input has always been tried by various researchers.

Vapor compression systems usually use throttling devices like capillary tube or expansion valve to expand the refrigerant from condenser to evaporator pressure. These throttling processes are thermodynamically isenthalpic (constant enthalpy) and are irreversible thus generate entropy. The increase in entropy in throttling process reduces the evaporating capacity of the VC system and the loss in refrigeration effect due to this is known as throttling loss. This loss can be minimized by replacing the isenthalpic expansion process by an isentropic expansion process in an ejector. Another advantage of using an ejector is that it raises the inlet pressure of the compressor to a value greater than that of the evaporator thus reducing compressor work input. Employing ejector for the expansion process in VC systems, the mentioned improvements lead to increase in the COP. Such systems in which ejector is used as an expansion device are referred as ejector-expansion refrigeration systems (EERS).
Numerical analysis on modern ejector expansion refrigeration cycle (EERC) was first done by Kornhauser in 1990 [2]. He found the COP of the ejector system 21 % higher than the COP of normal vapor compression refrigeration cycle for the refrigerant R12 at same operating conditions. Harrell and Kornhauser [3] further experimentally examined a two-phase ejector using R134a as the working fluid and obtained COP improvement ranging from 3.9 to 7.6% over the standard VC cycle. Domanski [4] analytically determined that the ejector efficiency has major effect on the COP of EERC. Nehdi et al. [5] performed a computer simulation of the EERC based on mass, momentum and energy balance and found that the area ratio Φ (area of constant area section divided by the area of primary nozzle outlet) has substantial impact on COP of the system. They evaluated the performance of several refrigerants at similar working conditions and the best COP was obtained for R141b (4.9) and R408a (4.6). Bilir and Ersoy [6] made a similar thermodynamic analysis of a constant area ejector based EERC and studied relation between various operating parameters. Sarkar [7] analyzed three natural refrigerants i.e. ammonia, propane and isobutane considering a constant pressure mixing model of EERC and obtained variation of COP, optimum area ratio and COP enhancement with evaporator and condenser temperatures for all the three refrigerants. Enhancement in COP was found to be highest for propane (26.1 %) followed by isobutane (22.8 %) and ammonia (11.7 %). In another study by Sarkar [8], a constant area mixing model of EERC using the same natural refrigerants was analyzed to get the COP and COP improvement with area ratio. Yari [9] did the exergy analysis of the EERC with R134a as the refrigerant and obtained that total exergy destruction was 24 % more for standard VC cycle than EERC. Yari and Sirosazlar [10] further performed first law and second law analysis for various refrigerants by modifying the standard EERC with an intercooler and internal heat exchanger and obtained that COP and 2nd law efficiency of their modified EERC were higher than both the standard EERC and VC cycle. Sumeru et al. [11] have given a comprehensive review of two-phase ejectors used in VC systems. Chen et al. [12] reviewed various refrigeration systems comprising both single and two phase ejectors and presented the important elements that determine optimum performance of ejector based refrigeration systems. Wang and Yu [13] carried out experimental investigation of a novel ejector expansion refrigeration system consisting two-phase ejector with R600a as refrigerant. They obtained that pressure lift ratio, entrainment ratio and ejector efficiency are strongly dependent on parameters like primary and secondary fluid pressures, nozzle exit position and nozzle throat diameter.

Exergy analysis provides the scope of performance improvement in the system and its components. Very limited work on exergy analysis of ejector expansion refrigeration system is available in open literature which is the motivation for the present work. Novelty lies in the comparative 2nd law analysis of a two phase EERS for three commercially used refrigerants.

2. Ejector Expansion Refrigeration System
The schematic of an EERS is shown in figure 1. After exiting the ejector at state 5, the refrigerant gets divided into saturated vapor and saturated liquid in the separator. The vapor refrigerant flows towards the compressor and is called the primary flow, while the liquid refrigerant travels towards the expansion valve and is termed as the secondary flow. Both the primary and secondary fluids are of the same refrigerant moving in the system. The primary flow (also motive flow) at high pressure exiting from the condenser at state 3 gets expanded in the primary nozzle and comes out at state 4. The low pressure at state 4 induces refrigerant i.e. secondary fluid from the evaporator at state 8 and expansion of the secondary fluid takes it to the pressure corresponding to state 9. Both the fluids are now at same pressure and there the mixing of primary and secondary fluid initiates at constant pressure that continues in the constant area mixing section up to state 10. Thereafter, the diffuser part of the ejector decelerates the fluid flow and converts its kinetic energy into pressure energy. The mixed fluid is now at intermediate pressure corresponding to state 5 and again gets separated in the separator, thus the cycle continues.

Figure 2 shows p-h diagram of the EERS along with standard VC cycle. Cycle 8-2b-3-11-8 is the standard VC cycle. In the EERS, the primary flow travels through separator, compressor, condenser,
and ejector (cycle 1-2-3-4-10-5-1), while secondary flow goes through separator, expansion valve, evaporator and ejector (cycle 6-7-8-9-10-5-6). The standard VC system is a two-pressure system whereas the EERS is a three-pressure system as can be seen in figure 2.

The entrainment ratio $\omega$ for EERS is the ratio of mass flow rate of secondary fluid to mass flow rate of primary fluid.

$$\omega = \frac{m_6}{m_1} = \frac{m_{evap}}{m_{cond}} = \frac{m_{secondary}}{m_{primary}}$$

The COP of both standard VC system and EERS are shown in equations (2) and (3) respectively

$$COP_{VC} = \frac{h_8-h_{11}}{h_{2b}-h_8}$$

$$COP_{EERS} = \frac{m_{evap}(h_8-h_7)}{m_{cond}(h_2-h_2)} = \omega \frac{(h_8-h_7)}{(h_2-h_1)}$$

It can be seen that for a high COP the entrainment ratio must be as high as possible.

3. Methodology

The system analysis has been done considering the following assumptions-

- Pressure loss in the condenser, evaporator, separator and the connecting lines is negligible.
- No heat exchange occurs in any component except in condenser and evaporator.
- Refrigerant state at evaporator and condenser outlet are vapor and liquid respectively.
- Refrigerant kinetic energy at ejector nozzle inlets and diffuser outlet is negligible.
- All the processes are in steady state and expansion process 6 to 7 is isenthalpic.
- One-dimensional flow of refrigerant occurs inside the ejector.
- Further, the mixing of the two flows is assumed to occur at constant pressure in the suction chamber of the ejector (processes 4-10 and 9-10 in figure 2).

The prerequisite mass, momentum and energy analysis of the system has been done similar to that carried out in [7, 8]. To avoid repetition, it is not described here.

3.1. Exergy analysis

General exergy balance can be written as
I = E_{in} - E_{out} + E_Q + E_W \tag{4}

where I is irreversibility or exergy destroyed, E_{in} and E_{out} are exergy of fluid at entry and exit of the control volume, E_Q and E_W are net exergy associated with heat and work transfer respectively.

Exergy at any point is denoted by \( E = \dot{m} e \), where \( \dot{m} \) is the mass flow rate of the fluid and \( e \) is the specific exergy at that state point, given by \( e = (h - h_o) - T_o(s - s_o) \). The exergy balance for all the components is given in table 1, where subscripts are corresponding to state points in the schematic diagram of EERS as shown in figure 1.

For a total mass flow rate of 1 kg/s, primary and secondary mass flow rates are \( \frac{1}{1+\omega} \) kg/s and \( \frac{\omega}{1+\omega} \) kg/s respectively.

### Table 1. Irreversibility equations for various components.

| Component       | Irreversibility                                      |
|-----------------|------------------------------------------------------|
| Evaporator      | \( I_{\text{evap}} = \frac{\omega}{1+\omega}(e_7 - e_8) + \dot{m}_{\text{ef, evap}}(e_{12} - e_{13}) \) |
| Condenser       | \( I_{\text{cond}} = \frac{1}{1+\omega}(e_2 - e_3) + \dot{m}_{\text{ef, cond}}(e_{14} - e_{15}) \) |
| Compressor      | \( I_{\text{comp}} = \frac{1}{1+\omega}(e_1 - e_2) + W_{\text{comp}} \) |
| Ejector         | \( I_{\text{ejector}} = \frac{1}{1+\omega}e_3 + \frac{\omega}{1+\omega}e_8 - e_5 \) |
| Expansion valve | \( I_{\text{ev}} = \frac{\omega}{1+\omega}(e_6 - e_7) \) |
| Separator       | \( I_{\text{separator}} = e_5 - \frac{1}{1+\omega}e_1 - \frac{\omega}{1+\omega}e_6 \) |

The total irreversibility is the sum of the irreversibility of all the components. 2\textsuperscript{nd} law efficiency is given by \( \eta_{II} = 1 - \left( \frac{I_{\text{total}}}{W_{\text{comp}}} \right) = \frac{W_{\text{rev}}}{W_{\text{act}}} \).

### 3.2 Model validation

Results obtained in the present analysis have been validated with those of Yari and Sirousazar [10]. At evaporator and condenser temperature of 5°C and 40°C respectively and the working fluid R125, the values of Exergy Destruction Ratio (EDR) have been compared in Table 2. EDR of a component is the ratio of the irreversibility of that component to the total irreversibility of the system. The present results have been determined using Engineering Equation Solver (EES) [14].

### Table 2. Model validation: EDR values comparison.

| Component       | Yari and Sirousazar [10] | Present analysis |
|-----------------|--------------------------|------------------|
| Evaporator      | 14.00                    | 14.92            |
| Condenser       | 38.20                    | 38.64            |
| Compressor      | 21.81                    | 22.05            |
| Ejector         | 25.50                    | 23.82            |
| Expansion valve | 0.35                     | 0.57             |

### 3.3 Selection of refrigerants

For the present analysis, three refrigerants R134a, R407C and R410A have been taken. All these refrigerants have zero ODP and medium GWP. These are some of the most widely used refrigerants currently in domestic refrigeration and air-conditioning, automobile ACs, industries as well commercial buildings etc.
4. Results and discussions
The analysis is carried out by increasing condenser temperature from 35°C to 55°C but fixed evaporator temperature of 5°C. Other input conditions are as follows.
- Evaporator external fluid (water) inlet temperature=11°C
- Evaporator external fluid (water) outlet temperature=7°C
- Condenser external fluid (air) inlet temperature=28°C
- Condenser external fluid (air) outlet temperature=32°C
- Dead state temperature= 25°C
- Dead state pressure=101.325 kPa

4.1 Evaporator
Figure 3 shows the variation of irreversibility in the evaporator with respect to condenser temperature for all the refrigerants. Increase in condenser temperature increases the vapor content of the refrigerant at the ejector outlet and thus separator has more amount of refrigerant vapor than the refrigerant liquid which results in decrease in ω. There is no significant change in the other parameters of the exergy equation for the evaporator. On account of decrease in ω, the exergy destruction or the irreversibility decreases with the rise in condenser temperature. Irreversibility in the evaporator decreases by 12.15% due to the rise in condenser temperature from 35˚C to 55˚C with refrigerant R134a.

![Figure 3. Irreversibility in evaporator](image)

![Figure 4. Irreversibility in condenser](image)

4.2 Condenser
Figure 4 shows that the irreversibility in the condenser increases with the condenser temperature. The entrainment ratio ω decreases similar to the reason already discussed in evaporator. Decrease in entrainment ratio results in more mass flow rate of primary fluid and thus more heat load on the condenser. Moreover, increasing condenser temperature while keeping the external fluid temperatures constant, the finite temperature difference for heat transfer increases in the condenser, which causes increase in irreversibility in the condenser. For R134a, it is observed that there is 313.74% increase in condenser irreversibility while raising the condenser temperature from 35°C to 55°C.

4.3 Compressor
Figure 5 shows the change in irreversibility in the compressor with respect to the condenser temperature. Increase in the mass flow rate of primary fluid and the work input are responsible for the increase in irreversibility with increase in condenser temperature. For R134a, there is 65.4% increase in compressor irreversibility as condenser temperature varies from 35°C to 55°C.
Figure 5. Irreversibility in compressor

4.4 Ejector
Figure 6 shows that the irreversibility in the ejector increases with the increase in condenser temperature. The variation in the mass flow rates of the primary and secondary fluid affects the efficiencies of the nozzle, mixing chamber and diffuser section which causes the increase in irreversibility. Irreversibility in ejector increases by 186.14% as condenser temperature is raised from 35°C to 55°C for R134a.

4.5 Total irreversibility
Figure 7 shows the variation of total irreversibility of the system with increase in condenser temperature. Significant increase in condenser irreversibility is the primary cause for increase in total irreversibility of the system.

Figure 7. Total irreversibility

Figure 8. 2nd law efficiency

Figure 8 shows that the 2nd law efficiency decreases with rise in condenser temperature because of the increase in total irreversibility of the system. R134a has highest 2nd law efficiency and R407C has the least for the complete range of condenser temperature considered for the analysis. Figure 9 compares the irreversibility in evaporator, condenser, compressor and ejector for all the three refrigerants considered in the analysis at T_{evap} = 5°C and T_{cond} = 40°C. Irreversibility is highest in the
condenser. So there is more scope for the thermodynamic performance improvement in the condenser. Irreversibility in the separator and expansion valve is negligible and has been neglected in this study.

![Figure 9. Irreversibility of various components at \(T_{\text{evap}} = 5^\circ\text{C}\) and \(T_{\text{cond}} = 40^\circ\text{C}\)](image)

5. Conclusions
The following observations can be drawn from the exergy analysis of the EERS-

- Total irreversibility increases and 2nd law efficiency decreases with the rise in condenser temperature while keeping the evaporator temperature constant.
- Irreversibility is highest in the condenser and also the increase in irreversibility is maximum in it as condenser temperature increases.
- For the complete range of condenser temperatures considered in the analysis, 2nd law efficiency of R134a is highest followed by R410A and R407C. In other words, R134a gives better performance than the other refrigerants. R134a is a high boiling point refrigerant in comparison to R407C and R410A which makes it suitable to run the entire system at low pressure. Total irreversibility of the system with R134a is less than the other refrigerants considered in the analysis, thereby the second law efficiency of the proposed system is highest.

Irreversibility in the condenser can be reduced by minimizing the finite temperature difference for the heat transfer. Further work needs to be carried out to develop means to reduce the irreversibility of the different components and the system as a whole.

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Appendix

| Notation | Subscripts |
|----------|------------|
| E ERC/EERS | ejector expansion refrigeration cycle/system | act | actual |
| E | exergy (kW) | cond | condenser |
| e | specific exergy (kJ/kg) | comp | compressor |
| h | specific enthalpy (kJ/kg) | ef | external fluid |
| I | irreversibility (kW) | ev | expansion valve |


\[ \dot{m} \quad \text{mass flow rate (kg/s)} \]

\[ s \quad \text{specific entropy (kJ/kg/K)} \]

\[ T \quad \text{temperature (°C)} \]

\[ \text{VC} \quad \text{vapor compression} \]

\[ W \quad \text{work (kW)} \]

\[ \omega \quad \text{entrainment ratio} \]

\[ \text{evap} \quad \text{evaporator} \]

\[ o \quad \text{dead state} \]

\[ \text{rev} \quad \text{reversible} \]

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