Vapour compression system analysis undergoing expansion in an ejector

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Abstract. This paper presents the analysis of a vapour compression system undergoing isentropic expansion in an ejector instead of irreversible isenthalpic expansion in conventional expanding devices i.e. capillary tube, thermostatic expansion valve etc. Comparative energy analysis of the optimized ejector expansion vapour compression system (EEVCS) has been done for commercially used refrigerants R134a, R407C and R410A and it is found that the COP enhancement over standard vapour compression system (VCS) is maximum for R410A (11.49%) followed by R407C (10.16%) and least in R134a (9.261%) at fixed condenser and evaporator temperature of 35°C and 5°C respectively. COP of both the systems decreases with the increase in condenser temperature, however the optimized ejector expansion vapour compression system exhibits less drop in its performance as compared to standard vapour compression system at the same condenser and evaporator temperature. The computational analysis has been carried out in Engineering Equation Solver (EES).

1. Introduction
Refrigeration is the process of maintaining an enclosed space temperature lower than the surrounding temperature by extracting heat from low temperature space and transferring it to the high-temperature source using thermodynamic work as input. The most commonly used refrigeration system at commercial and domestic level is a standard vapour compression system. It is also used in automobile air-conditioning and refrigeration [1].

The standard vapor compression system consumes huge amount of electrical energy in the compressor and electricity power generation using fossil fuels is catastrophic to the environment due to the production of the greenhouse gases [2]. Any enhancement possible in the vapor compression system that outcomes in higher productivity or lesser power input has been a constant topic of research.

In Standard VCS, throttling device e.g. thermostatic expansion valve, capillary tube is commonly used for the isenthalpic expansion process. The isenthalpic expansion is responsible for the decrease in cooling capacity of the system. This decrease in cooling capacity is also known as throttling loss. Ejector which comprises isentropic expansion can be used as an expansion device instead of conventional throttling devices to minimize the throttling loss or in other words to increase the cooling effect. Further, the ejector raises the compressor suction pressure leading to reduction in the compressor work. These two factors i.e. increase in cooling capacity and decrease in compressor work
lead to the COP enhancement of the optimized ejector expansion vapour compression system (EEVCS). The ejector is optimized for its maximum performance in optimized EEVCS.

2. Literature review
The patent by Gay [3] in 1931 was the first to explain how a two-phase ejector can be used as an expansion device to reduce throttling loss. In 1966, Kemper et al. [4] introduced a modification of this patent by raising the pressure and temperature of the primary flow before it enters motive primary nozzle by using a heater and a pump. Further modifications of the patent were proposed by Newton [5] in 1972. In 1990, Kornhauser [6] was the first to do numerical analysis of the ejector expansion refrigeration system (EERS) considering constant pressure mixing model. He studied the thermodynamic performance of EERS using refrigerant R-12. He obtained 21% more COP in EERS over standard vapour compression system at 30°C condenser temperature and -15°C evaporator temperature for the same refrigerant. Later, the experimental work on EERS using refrigerant R134a was done by Harrell and Kornhauser [7] in 1995. They claimed an enhancement in COP over the standard VCS ranging from 3.9 to 7.6 %. In 1995, Domanski [8] found that the theoretical COP of EERS was very much sensitive to the efficiency of an ejector. Numerical analysis of the performance of EERS was done by Nehdi et al. [9] in 2007 for 20 synthetic refrigerants. He introduced a geometric area ratio (i.e. ratio of the area of constant area section to the area of primary nozzle outlet) and found that the geometric parameters of the ejector have notable effects on the COP of the system. He obtained the maximum COP for the optimum area ratio \( \Phi_{op} \) equal to 10. In 2009, J Sarkar [10] carried out the thermodynamic analysis of EERS based on constant pressure mixing model for 3 natural refrigerants i.e. ammonia, propane and isobutane. He investigated the variation of COP, optimum area ratio and COP enhancement with the evaporator and condenser temperature. He found that the COP enhancement for ammonia is minimum (11.7%) followed by isobutane (22.8%) and then maximum for propane (26.1%) for the ranges of study. J. Sarkar [11] extended the analysis for a constant area mixing based EERS and obtained the variation of optimum area ratio, entrainment ratio and secondary pressure reduction with the evaporator temperature and condenser temperature. Sumeru et al. [12] provided a detailed review of two-phase ejector as an expansion device in standard vapour compression system. Energy and exergy analysis on modified EERS was done by Yari and Sirousazar [13] with several refrigerants. In the modified EERS, they worked on two-stage compressor with an intercooler and an internal heat exchanger to enhance its performance. They found that the modified EERS had 12 % higher COP as compared to the standard VCS and 8.6 % higher as compared to the standard EERS. Also the second law efficiency of the modified EERS was 20.7% and 8.1 % higher than those of the standard VCS and standard EERS respectively. Chen et al. [14] presented a comprehensive review of single phase and two phase ejectors used in various refrigeration systems. They also studied the effect of various parameters on the ejector performance.

Literature shows that limited work has been done on the performance analysis of a complete EEVCS while optimizing the ejector parameters for different refrigerants. Novelty of the work lies in the determination of the optimised area ratio of the ejector for the maximum achievable COP with the variation of condenser temperature but fixed evaporator temperature. Additionally, performance of the proposed system for three different refrigerants have been compared. A successful attempt has been made in this work to fill this gap.

3. System description
Figure 1 presents the schematic of ejector expansion vapour compression system (EEVCS). The refrigerant entering into the separator at state 1 gets separated into saturated vapour at state 2 and saturated liquid at state 6. The saturated vapour is sucked by the compressor and this mass flow of the refrigerant is known as primary flow, whereas the saturated liquid moves to the evaporator through expansion valve and is called secondary flow. Phase change of the compressed primary flow takes place in condenser and the saturated liquid refrigerant at state 4 is expanded in the primary motive
nozzle. At the exit of the primary nozzle the pressure is corresponding to state point 5 which is less than the pressure in the evaporator. This pressure difference entrains the secondary flow at state point 8 into the ejector and is called secondary pressure drop \((P_8-P_9)\). Now both the primary and secondary flow at state 5 and state 9 respectively are at the same pressure and constant pressure mixing of both the fluids take place in the constant area section up to state 10. Thereafter the diffuser section of the ejector increases the pressure of the refrigerant and comes out from the ejector at state 1 and gets separated into the separator. In this way the cycle continues.

The p-h diagram of both the EEVCS and SVCS is shown in figure 2. The SVCS is represented by 8-3s-4-11-8. In case of the EEVCS, primary flow circuit is shown by 1-2-3-4-5-10-1 which travels through separator, compressor, condenser, and ejector whereas cycle 1-6-7-8-9-10-1 represents the secondary flow movement into the separator, expansion valve, evaporator and ejector. The performance of an ejector is dependent on the pressure lift ratio (PLR) and the entrainment ratio.

![Figure 1. Schematic of EEVCS](image1)

![Figure 2. P-h diagram of SVCS and EEVCS](image2)

The pressure lift ratio is the ratio of pressure at the exit of the ejector i.e. state 1 to the pressure in the evaporator i.e. state 8 and the entrainment ratio \((\omega)\) is the ratio of the secondary mass flow rate to the primary mass flow rate of the working fluid of the system.

\[
PLR = \frac{P_{e,j,e,xit}}{P_{e,v,a,p}}, \quad \omega = \frac{m_{secondary}}{m_{primary}} = \frac{m_{evap}}{m_{com}}
\]

4. Thermodynamic analysis

The thermodynamic analysis of Optimized EEVCS has been done on the basis of conservation of mass, momentum and energy principles. The following assumptions are used to simplify this analysis:

- The primary and secondary flow are mixed at a constant pressure in the mixing section below evaporator pressure.
- There is no external heat transfer from or to the system except in evaporator and condenser.
- The pressure drop in the evaporator, condenser, connecting tubes and separator is neglected.
- The condition of the refrigerant at the exit of condenser and evaporator are saturated liquid and saturated vapour respectively.
- Steady-state conditions exist in all the processes.
- Isenthalpic process (6-7) occurs in the throttling valve.
- One-dimensional flow occurs inside the ejector.
Negligible kinetic energy of fluid at the inlet of the ejector nozzles and exit of the diffuser.

The efficiencies of primary nozzle, secondary nozzle, and diffuser are constant.

The mass, momentum and energy balance equations applied in the different components are shown below. The subscripts in the equations are corresponding to the state points mentioned in the schematic diagram of EEVCS as shown in figure 1.

4.1. Ejector
Primary nozzle exit enthalpy

\[ h_4 = h_5 + \eta_n (h_4 - h_{5,is}) = h_5 + \frac{v_5^2}{2} \]  (3)

Secondary nozzle exit enthalpy

\[ h_8 = h_9 + \eta_n (h_8 - h_{9,is}) = h_9 + \frac{v_9^2}{2} \]  (4)

Ejector area ratio (\( \phi \)) is defined as the ratio of the sum of the exit areas of the primary and secondary nozzle to the exit area of the primary nozzle

\[ \phi = \frac{(a_5 + a_9)}{a_5} \]  (5)

For a total mass flow rate of 1 kg/s, the primary and secondary mass flow rates can be computed as follows.

Primary mass flow rate

\[ \frac{1}{1+\omega} = a_5 \rho_5 V_5 \]  (6)

Secondary mass flow rate

\[ \frac{\omega}{1+\omega} = a_9 \rho_9 V_9 \]  (7)

Velocity and enthalpy of the mixed fluid after mixing

\[ V_{10} = \frac{1}{1+\omega} V_5 + \frac{\omega}{1+\omega} V_9 \]  (8)

\[ h_{10} + \frac{v_{10}^2}{2} = \frac{1}{1+\omega} \left( h_5 + \frac{v_5^2}{2} \right) + \frac{\omega}{1+\omega} \left( h_9 + \frac{v_9^2}{2} \right) \]  (9)

Enthalpy at the exit of the ejector

\[ h_1 = h_{10} + \frac{v_{10}^2}{2} = h_{10} + \frac{(h_{1is} - h_{10})}{\eta_d} \]  (10)

4.2. Compressor
The compression process is non-isentropic and the isentropic efficiency of compressor is based on the following empirical relation [15]

\[ \eta_{is,com} = 0.874 - 0.0135 \left( \frac{p_{cond}}{p_{evap}} \right) \]  (11)

Other relation of the isentropic compressor efficiency is

\[ \eta_{is,com} = \frac{(h_{is} - h_2)}{(h_3 - h_2)} \]  (12)

The specific work of the compressor is

\[ W_{com} = \frac{1}{1+\omega} (h_3 - h_2) \]  (13)
4.3. Evaporator
The cooling effect in EEVCS is
\[ Q_{\text{evap}} = \frac{\omega}{1+\omega}(h_8 - h_7) \] (14)

4.4. COP and COP enhancement
The COP of standard VCS
\[ COP_{\text{std}} = \frac{(h_8-h_{11})}{(h_3s-h_8)} \] (15)

The COP of EEVCS
\[ COP_{\text{eevcs}} = \omega \frac{(h_8-h_7)}{(h_3s-h_2)} \] (16)

The COP enhancement over standard VCS
\[ COP_e = \frac{COP_{\text{eevcs}} - COP_{\text{std}}}{COP_{\text{std}}} \] (17)

The mathematical modelling has been done in Engineering Equation Solver (EES) [16]. Initially the secondary pressure drop is varied for arbitrary value of entrainment ratio and the value of the pressure drop corresponding to maximum COP is found out. This value of pressure drop is given as input to the EES program and satisfying the mass balance equation at state 1, the optimum values of all other parameters are obtained for a given set of condenser and evaporator temperatures.

5. Model validation
The proposed system is validated with J Sarkar [10] for all the three refrigerants considered by him at the various operating conditions. Table 1 shows that the results obtained by the present analysis for ammonia at \( T_{\text{evap}} = -5^\circ C \) and different condenser temperatures \( T_{\text{cond}} \) are in good agreement.

Table 1. Comparison of COP and COP enhancement at \( T_{\text{evap}} = -5^\circ C \).

| \( T_{\text{cond}} \) (˚C) | COP | COP enhancement (%) |
|---------------------------|-----|---------------------|
| J Sarkar [10]             | Present analysis | J Sarkar [10] | Present analysis |
| 35                        | 4.669 | 4.97582 | 6.2724 | 6.27 |
| 40                        | 4.3298 | 4.33317 | 7.3636 | 7.38 |
| 45                        | 3.7743 | 3.81804 | 8.6101 | 8.62 |
| 50                        | 3.3770 | 3.39553 | 9.9734 | 10.0 |
| 55                        | 2.9795 | 3.04251 | 11.532 | 11.6 |

6. Results and discussions
The thermodynamic analysis of the proposed EEVCS has been performed for three refrigerants i.e. R134a, R407C and R410A. These refrigerants have zero ozone depletion potential (ODP) and medium global warming potential (GWP). Performance is determined for \( T_{\text{evap}} = 5^\circ C \) whereas \( T_{\text{cond}} \) is varied from 35˚C to 55˚C. The efficiency of nozzles and diffuser of an ejector are assumed to be 0.85.

Table 2 presents the variation of the COP of standard VCS, maximum COP of EEVCS, COP improvement of the EEVCS over the COP of standard VCS, PLR, entrainment ratio, optimum ejector area ratio and secondary pressure drop for R134a. The optimized EEVCS exhibits less drop in its performance as compared to standard VCS with the increase in \( T_{\text{cond}} \), rather the COP enhancement increases with increase in \( T_{\text{cond}} \) as shown in column 4 of table 2. With the increase in \( T_{\text{cond}} \), optimum value of ejector area ratio decreases and the pressure of the primary fluid at the exit of the motive nozzle decreases which causes the increase in secondary pressure drop. Further, the vapour quality at the ejector exit increases which lowers the entrainment ratio. Thus more primary fluid comes out from
the primary nozzle at more kinetic energy and mixes with the comparatively less secondary fluid. Thereby the kinetic energy of the mixture is high at the end of the mixing. The diffuser section converts more amount of kinetic energy into the pressure and thus the pressure at the ejector exit increases with the increase in $T_{\text{cond}}$. In other words, pressure lift ratio increases with the increase in $T_{\text{cond}}$. As the $T_{\text{cond}}$ increases, PLR increases which reduces the compressor work and thus results in the increase in COP enhancement over the VCS.

**Table 2.** Variation of parameters with condenser temperature for R134a at $T_{\text{evap}} = 5^\circ\text{C}$.

| $T_{\text{cond}}$ (°C) | COP_{std} | COP_{eevcs} | COP_{e} (%) | PLR | $\Phi$ | $\omega$ | $P_8$-$P_9$ (kPa) |
|------------------------|-----------|-------------|-------------|-----|-------|---------|------------------|
| 35                     | 6.634     | 7.24809     | 9.261       | 1.072 | 8.218 | 0.80022 | 11.5             |
| 40                     | 5.468     | 6.08709     | 11.31       | 1.098 | 7.069 | 0.76749 | 15.0             |
| 45                     | 4.583     | 5.20835     | 13.65       | 1.128 | 6.306 | 0.73468 | 18.0             |
| 50                     | 3.883     | 4.51719     | 16.33       | 1.164 | 5.585 | 0.70159 | 22.0             |
| 55                     | 3.313     | 3.95691     | 19.44       | 1.205 | 5.056 | 0.66818 | 25.5             |

Variation of the COP of standard VCS, maximum COP of EEVCS, COP improvement, PLR, ejector area ratio, entrainment ratio and secondary pressure drop for R407C and R410A are provided in table 3 and 4 respectively. Similar trends have been obtained with these refrigerants as R134a.

**Table 3.** Variation of parameters with condenser temperature for R407C at $T_{\text{evap}} = 5^\circ\text{C}$.

| $T_{\text{cond}}$ (°C) | COP_{std} | COP_{eevcs} | COP_{e} (%) | PLR | $\Phi$ | $\omega$ | $P_8$-$P_9$ (kPa) |
|------------------------|-----------|-------------|-------------|-----|-------|---------|------------------|
| 35                     | 5.325     | 5.86589     | 10.16       | 1.086 | 7.018 | 0.76640 | 21.5             |
| 40                     | 4.483     | 5.04478     | 12.54       | 1.115 | 6.238 | 0.73394 | 26.5             |
| 45                     | 3.812     | 4.39260     | 15.25       | 1.150 | 5.549 | 0.70092 | 32.5             |
| 50                     | 3.260     | 3.85906     | 18.38       | 1.190 | 5.026 | 0.66700 | 38.0             |
| 55                     | 2.795     | 3.41128     | 22.06       | 1.238 | 4.602 | 0.63221 | 43.0             |

**Table 4.** Variation of parameters with condenser temperature for R410A at $T_{\text{evap}} = 5^\circ\text{C}$.

| $T_{\text{cond}}$ (°C) | COP_{std} | COP_{eevcs} | COP_{e} (%) | PLR | $\Phi$ | $\omega$ | $P_8$-$P_9$ (kPa) |
|------------------------|-----------|-------------|-------------|-----|-------|---------|------------------|
| 35                     | 6.262     | 6.98129     | 11.49       | 1.079 | 7.484 | 0.78686 | 33.0             |
| 40                     | 5.112     | 5.82722     | 13.99       | 1.107 | 6.554 | 0.74936 | 41.0             |
| 45                     | 4.23      | 4.94653     | 16.93       | 1.141 | 5.703 | 0.71039 | 51.5             |
| 50                     | 3.525     | 4.24518     | 20.44       | 1.182 | 5.055 | 0.66939 | 61.5             |
| 55                     | 2.937     | 3.66480     | 24.77       | 1.234 | 4.521 | 0.62537 | 71.0             |

**Figure 3.** Comparison of design parameters and performance of EERC at fixed $T_{\text{evap}} = 5^\circ\text{C}$ and $T_{\text{cond}} = 35^\circ\text{C}$. 
The design parameters i.e. ejector area ratio, secondary pressure drop and performance parameters i.e. COP of EEVCS are dependent on the operating conditions and properties of the refrigerants. The comparison of the performance of different parameters of EEVCS for refrigerant R134a, R407C, and R410A at \( T_{\text{evap}} = 5^\circ\text{C} \) and \( T_{\text{cond}} = 35^\circ\text{C} \) is shown in figure 3. The optimum value of ejector area ratio is maximum for R134a followed by R410A and R407C. The optimum value of entrainment ratio is minimum for R407C (0.76640) and maximum for R134a (0.80022), whereas PLR is minimum for R134a (1.072) followed by R410A (1.079) and R407C (1.086). R134a provides higher COP than that of R407C and R410A. The COP enhancement over standard VCS is maximum for R410A (11.49\%) and minimum for R134a (9.261\%).

7. Conclusions
This paper presents the thermodynamic analysis of EEVCS with refrigerants R134a, R407C and R410A using EES. For a given evaporator and condenser temperature and total mass flow rate of 1kg/s, the present analysis delivers optimum values of design parameter like ejector area ratio and performance parameters like COP, entrainment ratio and pressure lift ratio of the ejector system. The results drawn from this study are as follows

- The COP of the standard VCS and EEVCS decreases with the increase in \( T_{\text{cond}} \).
- The optimized EEVCS exhibits less drop in its performance as compared to standard VCS, thereby increase in COP enhancement is observed with the increase in \( T_{\text{cond}} \).
- The optimum value of entrainment ratio and ejector area ratio decreases with the increase in \( T_{\text{cond}} \).
- The optimum ejector area ratio is maximum for R134a, whereas the secondary pressure drop is maximum for R410A at a fixed \( T_{\text{evap}} = 5^\circ\text{C} \) and \( T_{\text{cond}} = 35^\circ\text{C} \).
- The optimum entrainment ratio is minimum for R407C and maximum for R134a, whereas PLR is minimum for R134a followed by R410A and R407C at a fixed \( T_{\text{evap}} = 5^\circ\text{C} \) and \( T_{\text{cond}} = 35^\circ\text{C} \).
- The maximum cooling COP is 6.08709 for R134a at a fixed \( T_{\text{evap}} = 5^\circ\text{C} \) and \( T_{\text{cond}} = 35^\circ\text{C} \).
- The COP enhancement over standard VCS is maximum for R410A (11.49\%) followed by R407C (10.16\%) and least in R134a (9.261\%) at \( T_{\text{evap}} = 5^\circ\text{C} \) and \( T_{\text{cond}} = 35^\circ\text{C} \).

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Nomenclature :
- \( a \) : area of the nozzle (m\(^2\))
- \( h \) : specific enthalpy (kJ/kg)
- \( \dot{m} \) : mass flow rate
- \( P \) : pressure (kPa)
- \( Q \) : cooling effect (kW)
- \( T \) : temperature (°C)
- \( V \) : velocity (m/s)
- \( W \) : specific work (kJ/kg)

Abbreviations
- COP : coefficient of performance
- EERC : ejector expansion refrigeration cycle
- EEVCS : ejector expansion vapour compression system
- PLR : pressure lift ratio
- VCS : vapour compression system
Subscripts
com compressor
cond condenser
d diffuser
e enhancement
eje ejector
evap evaporator
is isentropic
n nozzle
std standard
1,2,3... cycle locations

Greek symbols
Φ geometric area ratio
ρ density (kg/m³)
ω entrainment ratio
η efficiency

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