

Research Article

Energetic and Exergetic Analysis of Two-Phase Constant Pressure Ejector Used in Ejector Expansion Refrigeration System (EERS) Using Alternative Refrigerants

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Abstract:

The study explores an ejector expansion refrigeration system in which a constant pressure ejector is incorporated as expansion device in place of throttle valve used in conventional vapour compression refrigeration system. To identify the potential performance enhancement, an energy and exergy numerical analysis were conducted using five different refrigerant R134a, R1234yf, R1234ze, R410a and R152a as these are used in various domestic and industrial application. The novelty of this study lies in the application of energetic and exergetic analysis to a two-phase constant-pressure ejector in an ejector expansion refrigeration system using alternative refrigerants, which has not been adequately explored in previous literature. Coefficient of Performance, Exergy Destruction, Exergy Efficiency is evaluated from the mathematical model developed using Engineering Equation Solver (EES). R1234ze gives better performance as improvement in COP of about 10% to 28% is observed in Ejector Expansion Cycle (EEC) as compared to (BC). Also, Total Exergy Destruction decrement from 52% to 26% is estimated due to which exergy efficiency improvement decreases to 8% from 23% in an EEC as compared to BC, if evaporator temperature changes from 248K to 288K at constant condenser temperature of 318K.

Keywords: Exergy, Ejector, COP, Entrainment Ratio, EERS

1. Introduction

In conventional vapor compression refrigeration systems, one of the primary sources of energy loss is from throttling losses in the expansion device. These losses occur due to the dissipation of kinetic energy as the refrigerant undergoes isenthalpic expansion through the throttle valve, primarily caused by friction. An effective way to minimize these throttling irreversibilities is by replacing the traditional throttle valve with a two-phase ejector. This modification not only recovers part of the energy typically lost during expansion but also increases the compressor inlet pressure. As a result, the compressor operates under improved conditions and consumes less power compared to the basic vapor compression cycle (BC), ultimately enhancing the overall system efficiency [1].

A one-dimensional model for a refrigeration cycle utilizing a two-phase ejector and projected a potential coefficient of performance (COP) enhancement of up to 21% compared to the conventional cycle employing an expansion valve is used [2]. Subsequent authors deployed this method to investigate the performance of Ejector Expansion Refrigeration System (EERS) [3] [4].

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Various investigations have confirmed these theoretical advantages that system using R134a in ejector expansion cycle and observed a COP improvement of up to 7.6% [5]. Integrating a two-phase ejector increased both *COP* and cooling capacity relative to a standard R134a expansion valve setup. Two-phase ejectors have gained particular attention in R744 (CO₂) transcritical systems, where throttling losses are notably high [6]. A 20% *COP* boost in an automotive R744 air conditioning system using this approach [7]. Similarly, an experimental study on a transcritical R744 cycle, finding an 8% *COP* increase and a 7% gain in cooling capacity when compared to a baseline expansion valve system [4]. Comparative analysis shows on systems equipped with ejectors and internal heat exchangers against traditional setups and found around 20% improvement in *COP*. The study also cited ejector efficiencies reaching up to 14.5% [8]. In another comparative analysis, a 15% improvement in *COP* when replacing the expansion valve with a two-phase ejector were recorded [9].

Naveen Kumar et al. analyzed the efficiency of a basic Vapor Compression Refrigeration System utilizing R134a and a Hydrocarbon blend (R290 & R600a). Their findings indicated that Hydrocarbons offer a highly efficient and environmentally friendly alternative, demonstrating superior exergy efficiency and significantly lower Global Warming Potential (*GWP*) compared to R134a [10].

Sattar et al. conducted an experimental study on a domestic refrigerator initially charged with R134a, later replaced with Hydrocarbons like isobutane and butane. They measured condenser and evaporator temperatures, *COP*, compressor work, heat rejection ratio, and cooling effect. The research showed energy consumption dropped by approximately 3% and 2% with isobutane and butane, respectively. Moreover, the system's *COP* and energy efficiency increased with Hydrocarbon usage compared to R134a [11].

While earlier studies have examined ejector refrigeration systems mainly with conventional working fluids and single-phase ejector models, the present work introduces a novel two-phase constant-pressure ejector configuration with alternative refrigerants. This approach provides a new perspective on improving system sustainability and efficiency, which has not been reported in detail in existing research. In this work, numerical analysis was carried out to evaluate the performance of the two-phase constant pressure ejector in an ejector expansion refrigeration system using alternative refrigerants.

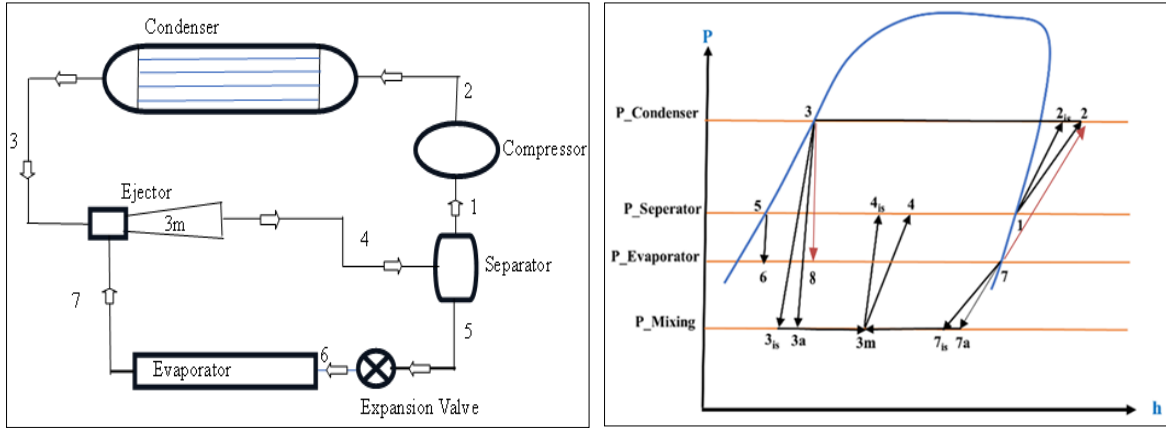
2. Mathematical Model

The ejector is a basic equipment that entrains and compresses a secondary flow from a lower pressure to an intermediate point of pressure using a primary flow from a higher pressure. There is a rising interest in incorporating ejectors into heat pumping and refrigeration systems, among other processes and applications, to increase energy efficiency [10-11]. Particularly, two-phase ejectors with liquid as the main and vapour as the secondary flows are the subject of this study. One may create effective systems with two-phase ejectors for the cooling process, refrigeration, and heat pumping purposes [12].

Figure 1a and b illustrates the schematic and pressure-enthalpy (P-h) diagram of *EEC* alongside the *BC*. The standard *BC* follows the path 7-2-3-8. In contrast, the *EEC* incorporates two fluids: the primary fluid moves through the separator, compressor, condenser, and ejector tracing the path 1-2-3-3a-3m-4-1. Meanwhile, the secondary fluid flows through the separator, expansion valve, evaporator, and ejector following the route 5-6-7-7a-3m-4-5. Unlike the two-pressure configuration of the *BC*, the *EEC* operates as a three-pressure system as depicted in Fig. 1a.

This section outlines the investigation of a 2-phase constant pressure ejector in an Ejector Expansion Refrigeration System (*EERS*) using refrigerant R134a, R123yf, R1234ze, R410a and R152a refrigerants. A numerical model for analysis is developed to analyse the effects of operational and geometric parameter. Figure 1 illustrates the ejector used in this study. The high-pressure motive flow from the condenser enters the ejector at zero velocity, representing a stagnation condition at state (3), and expands to a lower pressure at state (3a). This expansion creates a low-pressure region that draws in the secondary flow from the evaporator. The two flows merge within the ejector's mixing chamber. By the time the combined flow reaches the end of the mixing section, it reaches state (3m), after which it passes through a diffuser. The diffuser decelerates the flow, converting kinetic energy into pressure energy, and the resulting diffused stream exits the ejector at state (4) before entering the separator [13]. The mathematical models for the ejector expanding cycle were developed based on these presumptions [14, 15].

1. The refrigerant remains in thermodynamic quasi-equilibrium at all times.
2. Velocities are considered constant across the cross-section, following a one-dimensional model.
3. All fluid properties are uniform across the cross-section after complete mixing at the exit of the mixing section.
4. There is no external heat transfer.
5. Wall friction is neglected.
6. The throttling process is isenthalpic



(a) Schematic diagram of *BC* and *EEC*

(b) P-h Diagram for combined *BC* and *EEC*

Fig. 1. The schematic diagram and P-h curve of *BC* and *EEC* used in the present work.

2.1 Energy Analysis of Basic Cycle (*BC*) and Ejector Expansion Cycle (*EEC*)

Eqs (1) – (5) describe the operational dynamics of the ejector's motive nozzle. In parallel, Eqs (6) – (11) characterize the flow behavior within the secondary nozzle. The constant pressure mixing process inside the ejector is modelled using Eqs (12) – (18), which outline the essential thermodynamic relationships during this phase. Finally, Eqs (19) – (20) define the performance of the diffuser section at the ejector's outlet, where velocity reduction and pressure recovery occur [16].

Velocity at the exit of motive nozzle can be calculated by equation:

$$u_{3a} = \sqrt{2 * (h_3 - h_{3a}) * \eta_{mn}} \quad (1)$$

At inlet and the exit of the motive nozzle, the energy conservation equation can be applied as:

$$h_{3a} - h_3 = \frac{u_{3a}^2}{2} \quad (2)$$

Given that the pressure prior to the inlet of the uniform mixing area (3m) of an ejector is P_b and the value of the entrainment ratio of the ejection is ω .

According to the definition of isentropic efficiency for a motive nozzle, the enthalpy of the fluid at the nozzle exit can be determined using the following equation:

$$\eta_{mn} = \frac{h_3 - h_{3a}}{h_3 - h_{3a, is}} \quad (3)$$

A property relationship may be used to determine the specific volume of the mainstream at the entry point of the constant area mixing section using the equation given below:

$$v_{3a} = f(h_{3a}, P_b) \quad (4)$$

The area covered by the motive nozzle at the beginning of the uniform area mixing section based on the total ejector flow rate may be determined by applying the conservation of mass principle as follows:

$$a_{3a} = \frac{v_{3a}}{u_{3a}(1+\omega)} \quad (5)$$

The estimation process for the suction (secondary) nozzle is similar to that of the motive nozzle, given as follows:

$$s_{7a,is} = s_7 \quad (6)$$

$$h_{7a} = f(s_{7a,is}, P_b) \quad (7)$$

$$\eta_{sn} = \frac{h_7 - h_{7a}}{h_7 - h_{7a,is}} \quad (8)$$

$$u_{7a} = \sqrt{2 * (h_7 - h_{7a}) * \eta_{sn}} \quad (9)$$

$$v_{7a} = f(h_{7a}, P_b) \quad (10)$$

$$a_{7a} = \frac{\omega * v_{7a}}{u_{7a}(1 + \omega)} \quad (11)$$

A sequential method is used to determine the outlet properties of the mixing section. Initially, an estimated value for the output pressure P_m is selected. To determine the flow rate of the mixing stream at the outflow of A, assumptions are made that momentum conservation has been achieved for the mixing process. This is done using the equation:

$$u_{3m} = \frac{u_{3a} + \omega u_{7a}}{(1 + \omega)} \quad (12)$$

The enthalpy of the mixer stream at the outlet of the constant area mixing region may be determined by using the principle of energy conservation as follows:

$$h_{3m} = \omega h_3 + (1 - \omega)h_7 - \frac{u_{3m}^2}{2} \quad (13)$$

The particular volume of a mixed stream may be determined based on the property relationship using the following equation:

$$v_{3m} = f(h_{3m}, P_b) \quad (14)$$

The below condition must be satisfied in order to maintain mass conservation in the uniform area mixing region during the final phase.

$$\frac{(a_{3a} + a_{7a}) * u_{3m}}{v_{3m}} = 1 \quad (15)$$

The entropy for the mixed stream at the output of the diffuser section is:

$$s_{3m} = f(h_{3m}, P_b) \quad (16)$$

$$s_{4,is} = s_3 \quad (17)$$

Through the utilization of the principle of energy conservation throughout the ejector, the stream's enthalpy at the diffuser outlet can be determined:

$$h_4 = \frac{h_3 + \omega h_7}{(1 + \omega)} \quad (18)$$

At the diffuser's outlet, the isentropic enthalpy outlet may be determined based on the diffuser's efficiency as follows:

$$\eta_{ds} = \frac{h_{4,is} - h_{3m}}{h_4 - h_{3m}} \quad (19)$$

$$x_4 = f(h_{4,is}, P_4) \quad (20)$$

The fluid streams escape from the separator at a state where they are fully saturated and the enthalpies of both the gas and liquid phases are determined using property relationships as follows:

$$h_{f4} = f(P_4, x = 0) \quad (21)$$

$$h_{g4} = f(P_4, x = 1) \quad (22)$$

The feedback to the vapour stream flow rate may be calculated through a mass balance equation by:

$$m_p = (1 + \omega) * x_4 - 1 \quad (23)$$

The flow rate of the liquid exiting from the separator in a saturated state is determined by:

$$m_s = (1 + \omega) * (1 - x_4) \quad (24)$$

The enthalpy corresponds to a certain superheat value at the exit of the evaporator is given as:

$$h_7 = f(P_7, T_7) \quad (25)$$

The capacity of the evaporator may be determined by the given equation:

$$Q_{re,EEC} = m_s(h_7 - h_6) \quad (26)$$

The enthalpy at the input of the suction streams is proportional to the enthalpy at the output of an evaporator:

$$h_{7a} = h_7 \quad (27)$$

In order to determine the power of the compressor, the initial phase is to assess the output parameters of the isentropic compressor by:

$$s_1 = f(P_1, x = 1) \quad (28)$$

$$s_{2,is} = s_1 \quad (29)$$

$$h_2 = f(s_{2,is}, P_2) \quad (30)$$

Exact enthalpy at the compression output may be determined based on the isentropic efficiency achieved by the compressor: [17]

$$\eta_{comp} = \frac{h_{2,is} - h_1}{h_2 - h_1} \quad (31)$$

The amount of work performed by the compressor is:

$$W_{comp,EEC} = m_p(h_2 - h_1) \quad (32)$$

The coefficient of performance (COP) of the ejector expansion cycle (EEC) could be calculated by:

$$COP_{EEC} = \frac{Q_{e,EEC}}{W_{Comp,EEC}} \quad (33)$$

The capacity of evaporator for the basic cycle is determined by:

$$Q_{re,BC} = m (h_7 - h_8) \quad (34)$$

The outlet parameters of the isentropic compressors for the basic cycle may be determined in the following manner:

$$s_1 = f(P_7, T_7) \quad (35)$$

$$s_{2,is} = s_7 \quad (36)$$

$$h_{2,is} = f(s_{2,is}, P_2) \quad (37)$$

The exact enthalpy at the compressor outlet of the primary cycle could be determined by employing the concept of compressor isentropic efficiency as:

$$\eta_{comp,BC} = \frac{(h_{2,is} - h_7)}{(h_2 - h_7)} \quad (38)$$

Next, the precise calculation of the compressor performance for the fundamental cycle is determined by:

$$W_{comp,BC} = h_2 - h_7 \quad (39)$$

Eventually, the functioning of the fundamental cycle is determined by

$$COP_{BC} = \frac{Q_{e,BC}}{W_{comp,BC}} \quad (40)$$

In comparison to the *BC*, an *EEC* is characterized by its relative efficiency i.e, Performance Ratio (R) given by:

$$COP_{imp} = \frac{COP_{EEC} - COP_{BC}}{COP_{BC}} \quad (41)$$

2.2 Exergy Analysis in Basic Cycle (BC)

Exergy destruction in each component of the *BC* is calculated as:

Exergy Destruction in compressor of *BC* is given by [18]

$$E_{xd,comp,BC} = m * T_0 (s_2 - s_7) \quad (42)$$

Exergy Destruction in condenser for *BC* is given by: [18]

$$E_{xd,cond,BC} = m * [(h_2 - h_3) - T_0 (s_2 - s_1)] \quad (43)$$

Exergy Destruction in throttle valve of *BC* is given by: [18]

$$E_{xd,ev,BC} = m * T_0 (s_3 - s_8) \quad (44)$$

Exergy Destruction in evaporator of *BC* is given by: [18]

$$E_{xd,evap,BC} = m * [(h_7 - h_8) - T_0 (s_7 - s_8)] + Q_{re,BC} \left(1 - \frac{T_0}{T_r}\right) \quad (45)$$

Total Exergy Destruction in *BC* is given by: [18]

$$E_{xd,Total,BC} = E_{xd,comp,BC} + E_{xd,cond,BC} + E_{xd,ev,BC} + E_{xd,evap,BC} \quad (46)$$

Exergy Efficiency of BC is given by: [17]

$$\eta_{exergy,BC} = 1 - \frac{E_{xd,Total,BC}}{W_{comp,BC}} \quad (47)$$

2.3 Exergy Calculation in Ejector Expansion Cycle (EEC)

Exergy analysis offers valuable insights into potential efficiency enhancements for both systems and their individual components. However, there is a noticeable gap in published research specifically focusing on the exergy analysis of $EEERS$, which forms the basis of this study. The novelty of this work lies in its comparative second law investigation of a two-phase $EEERS$ using R134a, R1234yf, R1234ze, R410a and R152a refrigerants.

In EEC , exergy analysis is very important property to reduce the irreversibility for every component and also the whole cycle. In addition, exergy analysis can also play a vital role to calculate the improvement degree in the exergy efficiency in comparison with the BC . En. (52) to eqn. (63) shows the exergy analysis of EEC .

The expression for specific exergy of the refrigerant is given by:

$$E_{xd} = (h - h_0) - T_0(s_2 - s_0) \quad (48)$$

Exergy destruction in compressor is given by:

$$E_{xd,comp,EEC} = m_p * T_0(s_2 - s_1) \quad (49)$$

Exergy destruction in the condenser is:

$$E_{xd,cond,EEC} = m_p [(h_2 - h_3) - T_0(s_2 - s_3)] \quad (50)$$

Exergy destruction in the ejector can be calculated as:

$$E_{xd,ejec,EEC} = T_0[(1 + \omega)s_3 - s_7 - \omega s_4] \quad (51)$$

Exergy destruction in the expansion valve is:

$$E_{xd,ev,EEC} = m_s * T_0(s_5 - s_6) \quad (52)$$

Exergy destruction in the evaporator:

$$E_{xd,evap,EEC} = T_0(s_7 - s_6)\omega + \frac{T_0}{T_r}(h_7 - h_6)\omega \quad (53)$$

The total exergy losses of the cycle are:

$$E_{xd,Total,EEC} = E_{xd,comp,EEC} + E_{xd,cond,EEC} + E_{xd,ejec,EEC} + E_{xd,ev,EEC} + E_{xd,evap,EEC} \quad (54)$$

The exergy efficiency of the cycle is given by:

$$\eta_{exergy,EEC} = \frac{E_{xd,Total,EEC}}{W_{comp,EEC}} \quad (55)$$

The decrease in Total Exergy Destruction of EEC over BC is given by:

$$E_{xd,imp} = \frac{E_{xd,Total,EEC} - E_{xd,Total,BC}}{E_{xd,Total,BC}} \quad (56)$$

The improvement in exergy efficiency of *EEC* over *BC* is given by:

$$\eta_{exergy,imp} = 1 - \frac{\eta_{exergy,EEC} - \eta_{exergy,BC}}{\eta_{exergy,BC}} \quad (57)$$

Eqn. (58) and eqn. (59) define two critical performance metrics for evaluating ejector functionality: the mass entrainment ratio and the pressure lift ratio. The mass entrainment ratio reflects the ejector's capacity to draw in and transport secondary fluid mass—essentially quantifying how effectively the device can entrain refrigerant from the suction stream. Meanwhile, the pressure lift ratio represents the level of pressure augmentation the ejector imparts to the entrained suction flow, indicating its ability to boost pressure within the system [17].

$$\omega = \frac{\text{Secondary Mass Flow Rate}}{\text{Primary Mass Flow Rate}} = \frac{m_s}{m_p} \quad (58)$$

$$P_L = \frac{\text{Pressure at Diffuser Exit}}{\text{Pressure at Evaporator Outlet}} = \frac{P_4}{P_7} \quad (59)$$

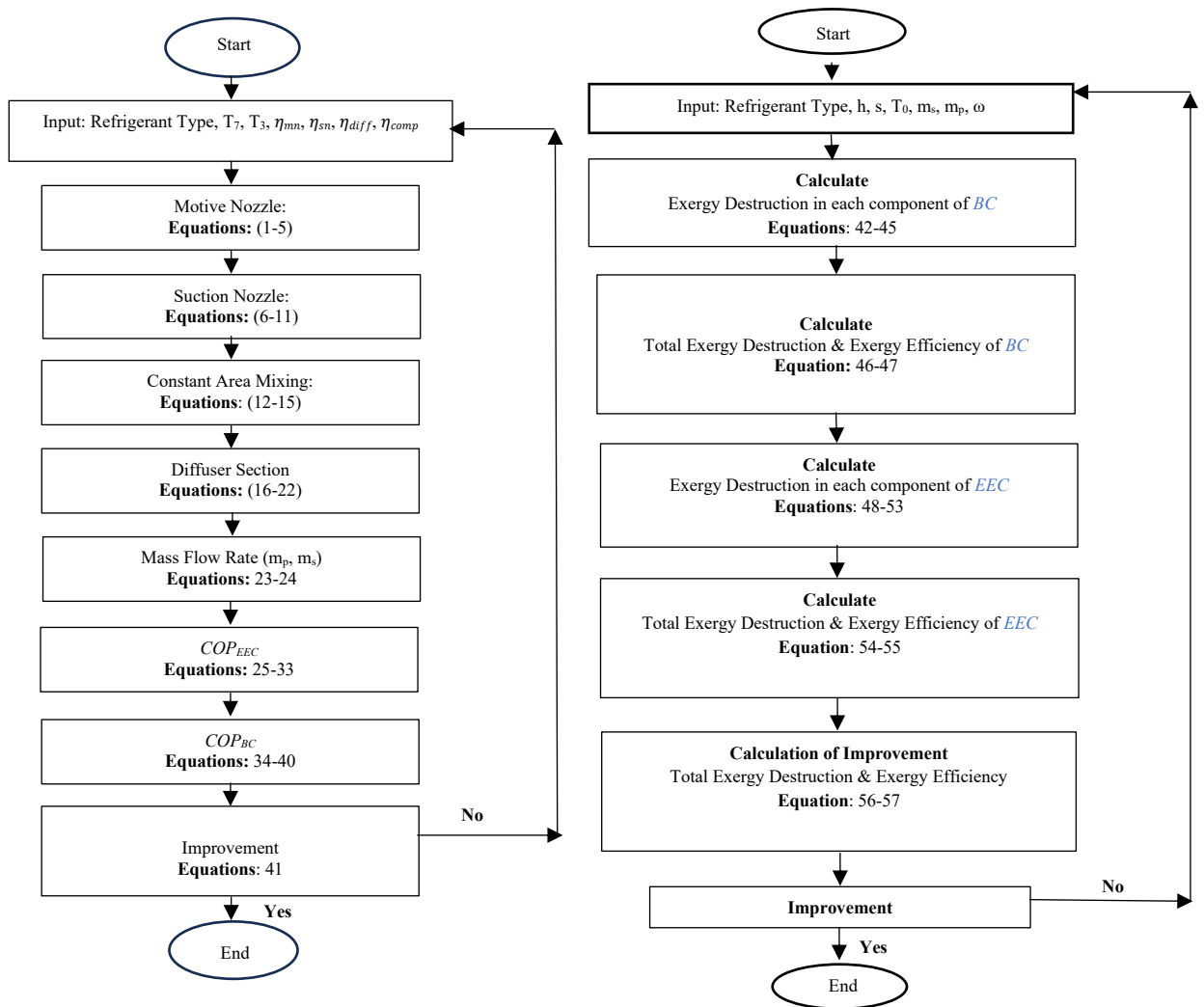


Fig. 2. Flow Chart for Energy and Exergy Analysis

3. Results and Discussion

Figures 3–6 are derived from the numerical analysis conducted during this study. The discussion provided is therefore based on analysis, supplemented with theoretical analysis for validation and explanation.

3.1 Energy Analysis of EEC

Figure 3 shows the variations of the system *COP* refrigerants R134a, R1234yf, R1234ze, R410a and R152a with the operating conditions of $T_e=248\text{K}$ to 288K and $T_c=313\text{K}$. The minimum increase in *COP* is obtained with R410a refrigerant. The *COP* of the all refrigerants increases in which maximum variation of *COP* for R134a with 80% whereas minimum with R410a around 33% when T_e varies from 248K to 288K . *COP* with all refrigerants increasing with increasing evaporator temperature, minimum variation can be seen in R410a while maximum is with R134a in *EEC*. Which indicates that at lower evaporator temperature, *COP* is low this is due to the reason at higher pressure difference compressor work will be higher. Whereas at higher evaporator temperature the pressure ratio across compressor decreases which leads to reduce the compressor work and higher *COP* can be achieved.

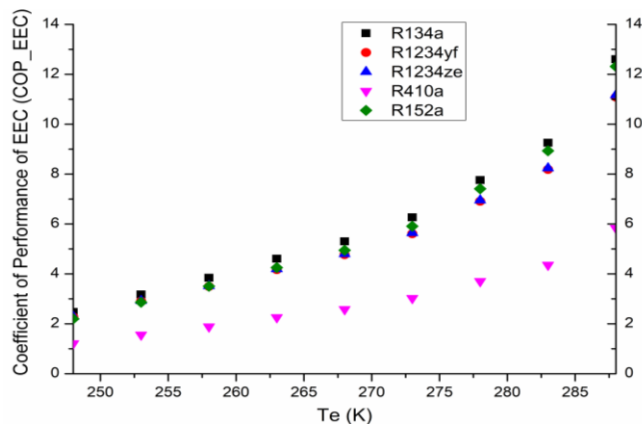


Fig. 3. Variation of *COP* with Evaporator Temperature in *EEC*

3.2 Exergy Destruction in each component of BC and EEC

Table 1: Exergy Destruction and Exergy Efficiency of the *BC* and *EEC* for R134a, R1234yf, R1234ze, R410a and R152a

Component	R134a		R1234yf		R1234ze		R410a		R152a	
	<i>BC</i>	<i>EEC</i>	<i>BC</i>	<i>EEC</i>	<i>BC</i>	<i>EEC</i>	<i>BC</i>	<i>EEC</i>	<i>BC</i>	<i>EEC</i>
$E_{xd, comp}$	4.15	3.65	2.01	2.78	13.59	5.66	32.29	9.92	32.29	8.41
$E_{xd, cond}$	8.52	7.29	5.66	5.19	7.62	9.09	1.49	10.59	1.49	9.51
$E_{xd, ejec}$	0.00	3.65	0.00	1.89	0.00	3.91	0.00	4.97	0.00	1.73
$E_{xd, ev}$	9.17	0.07	9.02	0.62	2.59	0.24	1.11	0.15	1.11	1.40
$E_{xd, evap}$	2.71	2.99	1.97	2.29	1.91	4.76	1.63	6.13	1.63	4.90
$E_{xd, Total}$	24.55	17.66	18.66	12.77	25.71	23.66	36.52	31.76	36.52	25.96
η_{exergy}	0.32	0.40	0.38	0.42	0.45	0.54	0.39	0.38	0.38	0.35

Table 1 presents the breakdown of exergy destruction across different components for both the *BC* and *EEC* using refrigerants R134a, R1234yf, R1234ze, R410a and R152a under operating conditions of an evaporator temperature of 273K and a condenser temperature of 318K . In the *EEC* system, exergy destruction values are based on a unit mass flow rate through the compressor. According to the data in Table 1, the condenser exhibits the highest level of irreversibility, while the smallest amount of exergy loss is observed in the expansion valve within *EEC*.

For the refrigerant R134a, the exergy destruction the minimum and maximum exergy destruction in evaporator and condenser for both *BC* and *ECC*, which are 2.71 kJ/kg , 2.99 kJ/kg and 8.51 kJ/kg , 7.29 kJ/kg respectively. For R1234yf, maximum exergy destruction for *BC* is in evaporator 9.02 kJ/kg and. When an ejector is employed, this

value drops to just 0.68 kJ/kg. For refrigerant R1234ze, maximum exergy destruction is in compressor 13.59 kJ/kg and decreases to 5.66 kJ/kg by incorporating an ejector and ejector itself have an exergy destruction of 3.92 kJ/kg. For refrigerant R410a, minimum exergy destruction is in expansion valve of 1.11 kJ/kg in *BC* which get reduced to 0.15 kJ/kg in *EEC*. R152a has an increased exergy destruction in evaporator of around 4.90 kJ/kg in *EEC* as compared to *BC* which is 1.63kJ/kg.

In the *EEC* system, evaporation occurs at a slightly lower temperature compared to the *BC*, which leads to a marginal increase in exergy destruction within the evaporator for all refrigerants. However, this increase is minimal and has a negligible impact on the total exergy destruction of the cycle. The combined effects of system modifications contribute to improved overall performance. As a result, the *EEC* exhibits lower total exergy destruction and higher exergy efficiency than the *BC*. Specifically, the *EEC* reduces total exergy destruction by 28% for R134a, 31.56% for R1234yf, 7.97% for R1234ze, 13.03% for R410a and 28.91% for R152a. Moreover, due to the recovered expansion work facilitated by the ejector, the exergy efficiency of the *EEC* improves by 25% for R134a, 10.5% for R1234yf, 20% for R1234ze, 2.6% for R410a and 7.89% for R152a when compared to the *BC*.

3.3 Total Exergy Destruction

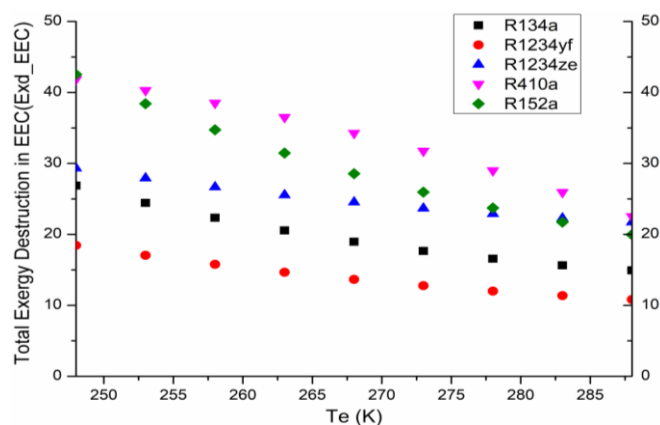


Fig. 4. Variation of Total Exergy Destruction with Evaporator Temperature for R134a, R1234yf, R1234ze, R410a and R152a in *EEC*

Figure 4 depicts that as T_e increases from 248 K to 288K keeping $T_c = 313$ K the exergy destruction decreases for all refrigerants *EEC*. The maximum exergy destruction is observed in refrigerant R410 a ranging from 43kJ/kg to 23kJ/kg while minimum is observed in refrigerant R1234yf ranging from 18kJ/kg to 11 kJ/kg for given temperature range in *EEC*. The reason behind decreasing exergy destruction with increasing evaporator temperature is at low evaporator temperature, the temperature difference of refrigerant and environment is higher which results in more irreversibility's during heat absorption and compression and increases entropy generation especially in the ejector and evaporator. On the other hand, at high evaporator temperature, the temperature difference of refrigerant and environment is less which leads to lower entropy generation in the evaporator and compressor work requirement also decreases which in result reduces exergy destruction in the compressor and ejector.

3.4 Exergy Efficiency

Figure 5 depicts that as T_e increases from 248 K to 288K keeping $T_c = 313$ K the exergy efficiency decreases for all refrigerants. The maximum exergy efficiency is observed in refrigerant R1234ze a ranging from 56% to 42% while minimum is observed in refrigerant R134a ranging from 48% to 28% at T_e as 248K and 288K respectively in *EEC*. The is due to the fact that if evaporator temperature increases the refrigerants have to operates at higher pressure which reduces compressor work therefore irreversibilities in ejector, compressor and evaporator decreases as a result exergy efficiency increases. On the other hand, decreasing evaporator temperature its pressure also decreases which demands more pressure lift therefore more energy is consumed in compressor and entraining refrigerants, resulting in more exergy destruction in evaporator and ejector which ultimately reduces the exergy efficiency.

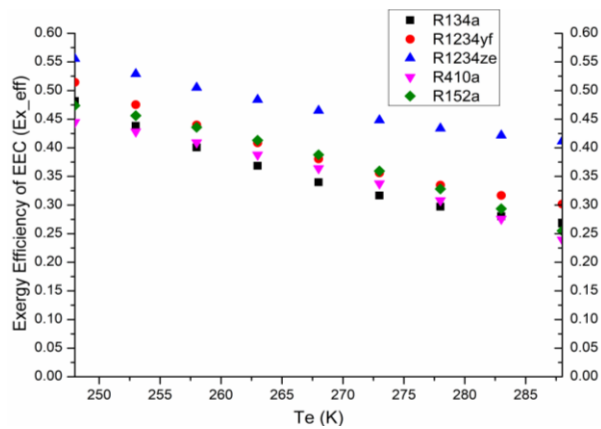
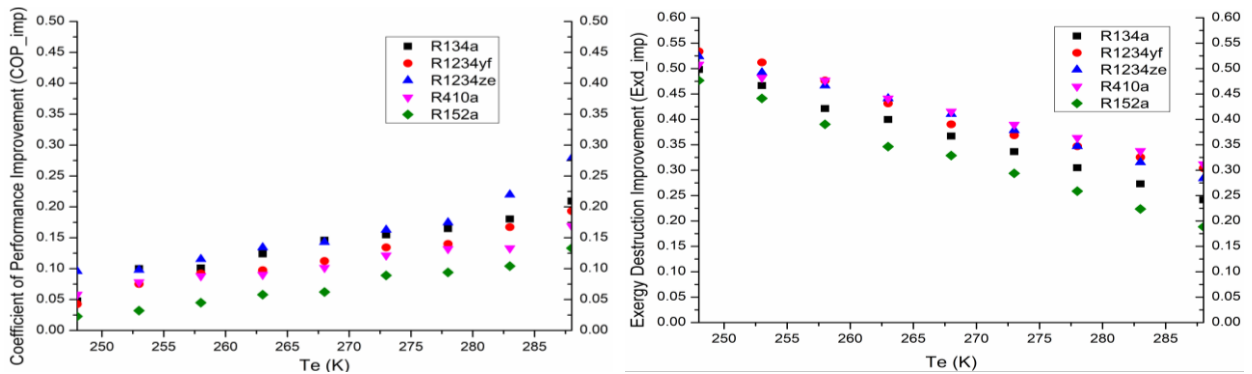


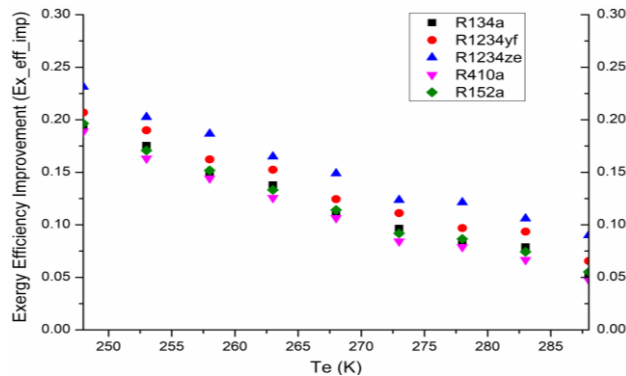
Fig. 5. Variation of Exergy Efficiency with Evaporator Temperature for R134a, R1234yf, R1234ze, R410a and R152a in *EEC*

3.5 Improvement in *EEC* over *BC*



(a) Improvement in *COP*

(b) Improvement in Total Exergy Destruction



(c) Improvement in Exergy Efficiency

Fig. 6. Improvement in *COP*, Total Exergy Destruction and Exergy Efficiency of *EEC* over *BC* using R134a, R1234yf, R1234ze, R410a and R152a

Figure 6 (a) illustrates the improvement of *COP* from *BC* to *EEC* while varying T_e from 248K to 288K and keeping T_c constant at 313K. As the graph shows increasing the temperature of evaporator *EEC* cycle gives much higher *COP* as compared to *BC* for all refrigerants. But R1234ze give maximum improvement of around 26% which get decreases

to 13% for R152a at $T_e = 288\text{K}$. As evaporator temperature increases, both *BC* and *EEC* show improvement in *COP*, but *EEC* demonstrates a superior rate of *COP* improvement due to its ability to recover expansion work through the ejector. The observed trends illustrate the relationship between various performance indices and the evaporator temperature, demonstrating the dynamics of system efficiency.

Figure 6(b) summarises reduction in exergy destruction i.e., exergy destruction improvement of an *EEC* compared to a *BC*, plotted against evaporator temperature (248K to 288K) for five refrigerants. R134a improves from 48% at 248K down to 27% at 288K, R1234yf have slightly higher improvement (53% to 30%), R1234ze maximum improvement (52% to 26%) in comparison to all refrigerants. R410a performs better with (51% to 31%) and lowest among all being R152a (47% to 19%). All refrigerants exhibit decreasing exergy-destruction reduction as evaporator temperature rises as that of in Volumetric Cooling Capacity (*VCC*) improvement trend. The exergy-efficiency improvement of the *EEC* over a *BC* varies with evaporator temperature for different refrigerants. R1234ze leads with the highest gains, from 23 % at 248K down to 8% at 288K. Exergy efficiency improvement in refrigerant R1234yf follows 21% to 3%, then R134a and R152a have a close variation from 19% to 6% and lastly R410a from 18% to 4%. Declining trend shows at higher evaporator temperatures for all refrigerants P_L and thus exergy efficiency (second-law efficiency) decreases as shown in figure 6 (c).

4. Conclusion

In this study, a refrigeration system integrating both *EEC* and *BC* was analyzed. Performance results for the conventional and enhanced configurations were calculated using the *EES* software. This paper presents a theoretical analysis of an *EEC* using R134a, R1234yf, R1234ze, R410 and R152a as the working fluid. The study explores the performance behaviour of the mentioned refrigerants in *EEC* equipped with a constant-pressure mixing ejector. System characteristics were examined across a condensing temperature range of 313K and an evaporation temperature range from 248K to 288K.

The cycle *COP* of R134a at a range of 2.5 -12.6, R1234yf ranges 2.3-11.1, R1234ze ranges 2.3 – 11.2, R410a ranges 1.21–5.83 and for R152a ranges 2.2 -12.3. The refrigerants have improvement of *COP* in *EEC* as R134a increases from 4.77% - 20.93%, R1234yf increases as 4.27% - 19.32%, R1234ze increases as 9.58% - 27.86, R410a increases as 5.79% - 16.88% and R152a increases from 2.3% - 13.3% as compared to *BC*.

The findings indicate that the R1234ze based *EEC* outperforms the *BC*, with performance gains becoming especially significant under conditions of elevated evaporator temperatures. The comparative exergy analysis of five different refrigerants revealed that R1234ze delivered the most substantial reduction in total system exergy destruction. Specifically, when the condenser temperature was maintained at 313K, both the *EEC* and *BC* experienced a 46% decrease in exergy destruction with R1234ze. Additionally, among all refrigerants evaluated, R1234ze demonstrated the highest improvement in exergy efficiency. Based on the combined energy and exergy assessments, R1234ze emerges as the most optimal refrigerant choice for *EEC* applications.

Abbreviations

<i>EERS</i>	Ejector Expansion Refrigeration System
<i>EEC</i>	Ejector Expansion Cycle
<i>BC</i>	Basic Cycle
<i>COP</i>	Coefficient of Performance
<i>VCC</i>	Volumetric Cooling Capacity (kJ/m^3)
<i>EES</i>	Engineering Equation Solver
<i>GWP</i>	Global Warming Potential

Symbols

W	Compressor Work (kJ/kg)
P_L	Pressure Lift
Q_{re}	Refrigerating Effect (kJ/kg)
ω	Entrainment Ratio
T	Temperature (Kelvin)
P	Pressure (kPa)
m	Mass Flow Rate (kg/sec)

E_{xd}	Exergy Destruction (kJ/kg)
η	Efficiency
h	Enthalpy (kJ/kg)
s	Entropy (kJ/kg-K)
u	Velocity (m/s)
v	Specific Volume (m ³ /kg)
x	Dryness Fraction
a	Area (m ²)

Subscripts

mn	Motive Nozzle
sn	Suction Nozzle
$Total$	Total
imp	Improvement
$comp$	Compressor
$cond$	Condenser
ev	Evaporator
$ejec$	Ejector
$evap$	Evaporator
$exergy$	Exergy
is	Isentropic
b	Baseline
r	Refrigerant
0	At Environment Pressure and Temperature

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