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Performance Enhancement of Constant Pressure Ejector Expansion Refrigeration System Using Environment Friendly Refrigerants

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Abstract

This study presents experimental findings comparing the performance of low-pressure refrigerants R134a, R1234yf and R1234ze in two different cycle configurations: a two-phase ejector cycle and traditional expansion valve cycles. A modified two-phase ejector setup was also tested, designed to leverage the pressure lift generated by the ejector to support multiple evaporation temperature levels. Analysis revealed that ejectors optimized for low-pressure refrigerants delivered work recovery efficiencies that were comparable, though slightly lower, than those observed in CO_2-based ejector systems. During analysis it is found that R1234yf delivers marginally lower COP values than R134a, yet it outperforms R1234ze in overall performance and also R1234yf shows a slightly lower refrigerating effect than R134a but consistently higher than R1234ze. Refrigerant R1234ze demonstrates the highest performance boost, achieving around a 26% increase in COP. R1234ze slightly surpasses R134a, delivering performance gains between 57% and 35% in VCC. R1234yf offers moderate enhancements, with improvements ranging from 52% to 27%. Significantly, the highest VCC efficiency boost—approximately 20%—is observed at the lowest evaporator temperature (248K), where the ejector achieves maximum pressure lift, optimizing energy recovery.

Keywords: *Ejector, Two-Phase Flow, Refrigeration, Throttle Valve, Entrainment*

Introduction

The idea of using ejectors in refrigeration systems dates back to the early 1900's. Originally, they were incorporated into steam-based setups to enhance efficiency by using refrigerant expansion to generate a vacuum or suction effect. By the mid-20th century, engineers began experimenting with integrating ejectors into refrigeration cycles, aiming to either supplement or replace traditional expansion valves.

This innovation was driven by the need to boost performance and energy efficiency, particularly in low-temperature applications. A key strategy in optimizing refrigeration systems involves reducing throttling losses. Replacing the standard expansion valve with an ejector can significantly cut these losses by recovering energy that would otherwise be wasted during the throttling phase [1].

Cycle Description

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The first computational investigation into the modern Ejector Expansion Refrigeration Cycle (EERC) was carried out by Kornhauser [2]. His study demonstrated that, under identical operational conditions, an ejector-based system utilizing R12 refrigerant achieved a 21% higher Coefficient of Performance (COP) compared to a conventional vapor compression refrigeration (VCR) system. Subsequent experimental research employed a two-phase ejector with R134a, reporting COP enhancements ranging from 3.9% to 7.6% over traditional VCR systems [3].

An ejector expansion refrigeration system (EERS) is an innovative refrigeration technology that replaces the expansion valve with an ejector to manage the flow and expansion of refrigerant. This system boasts several benefits over conventional refrigeration systems, particularly in energy efficiency and operational flexibility. Essentially, EERS operates on vapor-compression cycles, utilizing an ejector in place of a throttle valve [4].

As illustrated in Figure 1, the EERS comprises four main components: evaporators, vapor generators, condensers, and ejectors. The system functions by heating the vapor generator, which produces refrigerant at high pressures and temperatures, known as the 'primary fluid.' This primary fluid is then directed into the ejector, where it expands and creates a low-pressure area in the evaporator, referred to as the 'secondary fluid.' The primary fluid collects and compresses the secondary fluid and the resulting mixture is passed to the separator which then passes to the evaporator through expansion valve. Finally, the compressed refrigerant is cycled back to the condenser completing the refrigeration cycle [5].

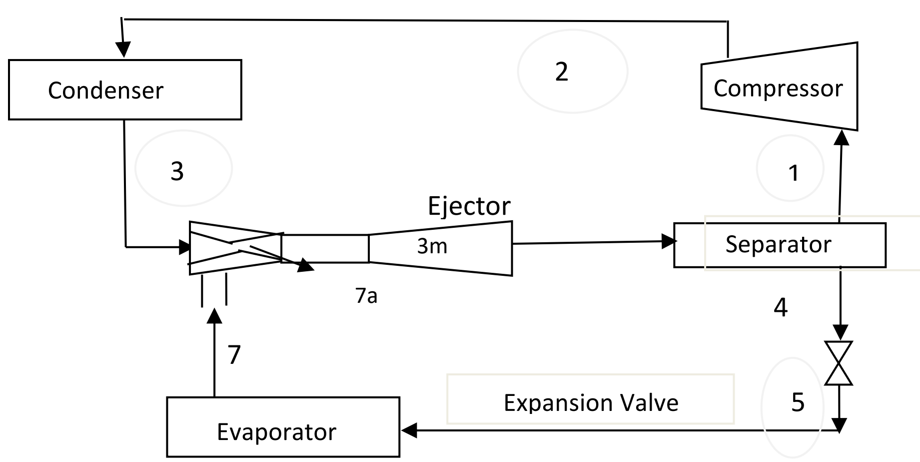

Figure 1: Layout of Ejector Expansion Refrigeration System

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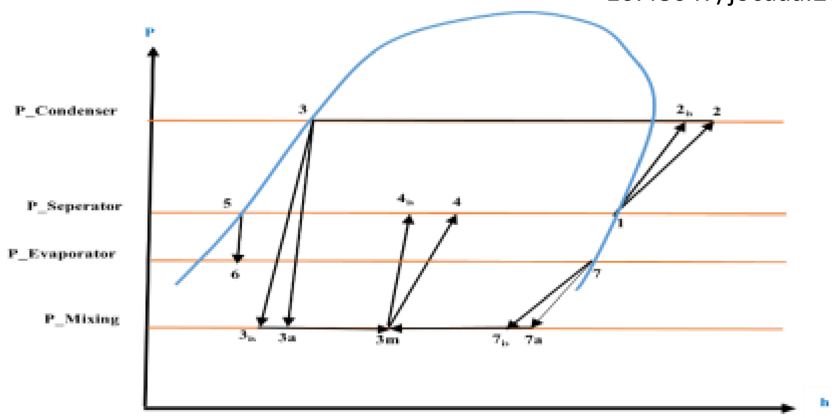

Figure 2: P-h Curve for Ejector Expansion Refrigeration Cycle

Refrigerant's

Refrigerants thermos-physical properties play a vital role in the selection criteria of any proper refrigerant. Table 1.1 shows the properties of different refrigerants used in the research

Table 1.1 Properties of Refrigerants Used in Analysis

Refrigerant Properties	R134a	R1234yf	R1234ze
Chemical Name	Norflurane	Hydrofluoro-olefin (HFO)	2,3,3,3 Tetraflourolefin
Chemical Formula	CH_2 FCF_3	CH2=CFCF3	$CF_3CH=CHF$
Molecular Weight (g/mol)	102.03	114.04	114
Critical Temperature (°C)	122	94.7	153.7
Critical Pressure (kPa)	4060	3382	3640
Boiling Point (at Atm. Pressure, °C)	-26.1	-29.4	-19
Latent Heat of Evaporation (KJ/kg)	217.2	180.25	162.9
Density Liquid (at 25°C in kg/m^3)	1206	1092	1170
Toxic	Low	Low	No
Flammability Rating	A1	A1	A2
Odor	No	No	No
Ozone Depletion Potential (ODP)	0	0	0
Global Warming Potential (GWP)	1300	<1	7

Thermodynamic Analysis

The procedure involved in the investigation of a two-phase constant pressure ejectors in an Ejector Expansion Refrigeration System (EERS) using refrigerant R134a is outlined below. The mathematical model is developed in Engineering Equation Solver (EES) to analyse the effects of operational and geometric parameters on Entrainment Ratio and Pressure Lift. To streamline the analysis process, the following assumptions have been adopted:

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.

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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. Steady-state conditions are considered.
7. The throttling process is isenthalpic.
8. The pressure drops across the condenser, evaporator, separator, and interconnecting pipelines are considered insignificant [6] [7] [8].

Based on these assumptions, a mathematical model for the Ejector Expansion Refrigeration Cycle (EERC) is developed, following the framework outlined by [9]. As illustrated in Fig. 1, the analysis begins by identifying the thermodynamic properties of the refrigerant at the condenser and evaporator outlets—corresponding to the motive and suction flows entering the ejector.

Equations (1–5) describe the operational dynamics of the ejector's motive nozzle. In parallel, equations (6–11) characterize the flow behavior within the secondary nozzle. The constant pressure mixing process inside the ejector is modelled using equations (12–18), which outline the essential thermodynamic relationships during this phase. Finally, equations (19–20) define the performance of the diffuser section at the ejector's outlet, where velocity reduction and pressure recovery occur [10].

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

$$u_{3a} = \sqrt{2 * (\mathrm{h}_3 - \mathrm{h}_{3a}) * \eta_{mn}} \tag{1}$$

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

$$h_{3a} - h_3 = \frac{u_{3a}^2}{2} \tag{2}$$

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:

$$\eta_{mn} = \frac{h_3 - h_{3a}}{h_3 - h_{3a,is}} \tag{3}$$

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:

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

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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:

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

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

$$s_{7a,is} = s_7 \tag{6}$$

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

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

$$u_{7a} = \sqrt{2 * (\mathrm{h}_7 - \mathrm{h}_{7a}) * \ \eta_{\mathrm{sn}}} \tag{9}$$

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

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

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:

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

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:

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

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

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

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

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

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

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

$$s_{4,is}=s_3 \tag{17}$$

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

$$h_{4=}\frac{h_3+wh_7}{(1+\omega)} \tag{18}$$

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

$$\eta_{ds}=\frac{\mathrm{h}_{4,is}-\mathrm{h}_{3m}}{\mathrm{h}_4-\mathrm{h}_{3m}} \tag{19}$$

$$\mathrm{X}_4=f(\mathrm{h}_{4,is},\mathrm{P}_4) \tag{20}$$

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:

$$\mathrm{h}_{f4}=f(P_4,x=0) \tag{21}$$

$$\mathrm{h}_{g4}=f(\mathrm{P}_4,x=1) \tag{22}$$

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

$$\mathrm{m}_p=(1+\omega)*\mathrm{x}_4-1 \tag{23}$$

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

$$m_s=(1+\omega)*(1-x_4) \tag{24}$$

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

$$\mathrm{h}_7=f(\mathrm{P}_7,\mathrm{T}_7) \tag{25}$$

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

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

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

$$h_{7a} = h_7 \tag{27}$$

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

$$s_1 = f(P_1, x = 1) \tag{28}$$

$$s_{2,is} = s_1 \tag{29}$$

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

Exact enthalpy at the compression output may be determined based on the isentropic efficiency achieved by the compressor: [11]

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

The amount of work performed by the compressor is:

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

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

$$COP_{EEC} = \frac{Q_{e,EEC}}{W_{Comp,EEC}} \tag{33}$$

Volumetric Cooling Capacity of EEC is given by:

$$VCC_{EEC} = \frac{Q_{re,EEC}}{Specific\,Volume\,of\,vapor\,refrigerant\,(v_{g1})} \tag{34}$$

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

$$Q_{re,BC} = m_p\,(h_7 - h_8) \tag{35}$$

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

$$s_1 = f(P_7, t_7) \tag{36}$$

$$s_{2,is} = s_7 \tag{37}$$

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$$h_{2,is} = f(s_{2,is}, P_2) \tag{38}$$

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

$$\eta_{comp,BC} = \frac{(h_{2,is} - h_7)}{(h_2 - h_7)} \tag{39}$$

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

$$W_{comp,BC} = m_p\,(h_2 - h_7) \tag{40}$$

Eventually, the functioning of the fundamental cycle is determined by

$$COP_{BC} = \frac{Q_{e,BC}}{W_{comp,BC}} \tag{41}$$

Volumetric Cooling Capacity of BC is given by:

$$VCC_{BC} = \frac{Q_{re,BC}}{Specific\ Volume\ of\ vapor\ refrigerant\ (v_{g1})} \tag{42}$$

In comparison to the BC, an EEC is characterized by its relative efficiency i.e., COP Improvement given by:

$$COP_{imp} = \frac{COP_{EEC} - COP_{BC}}{COP_{BC}} \tag{43}$$

Volumetric Cooling Capacity Improvement (VCC_{imp}) is given by: [11]

$$VCC_{imp} = \frac{VCC_{EEC} - VCC_{BC}}{VCC_{BC}} \tag{44}$$

Results and Discussion

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Despite its slightly inferior performance, R1234yf is a more environmentally friendly option due to its low Global Warming Potential (GWP). R1234ze exhibits the lowest COP values among the three refrigerants across the observed temperature range. This may be attributed to its lower vapor pressure and enthalpy of vaporization, which reduce its thermodynamic efficiency in the EERS.

The data reinforces that R134a offers the highest thermodynamic potential, but due to environmental regulations, R1234yf serves as a promising alternative, balancing moderate performance with sustainability. R134a again leads in terms of cooling capacity, but R1234yf remains a viable alternative for systems prioritizing environmental compliance, especially in applications where slightly reduced performance is acceptable.

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