

Theoretical Analysis of New Regenerative Vapour Compression System Using Ejector as Second Stage of Compression

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Abstract: This paper presents a detailed theoretical analysis of regenerative vapour compression refrigeration cycle with ejector as second stage of compression. The basic purpose of using ejector, is to utilize the regenerative use of potential energy of ejector two phase expansion flow which would otherwise be lost in expansion valve. First stage of compression is achieved by compressor, in which only vapour compresses to 50-60% of the final pressure, while second stage of compression is achieved by a jet device (ejector) using internal potential energy of the working fluid flow. By this arrangement work input to compressor is reduced significantly, resulting an increase in COP of the system as compared to traditional cycle. A mathematical computational model is developed in the equation window of engineering equation solver(EES) for calculating different parameters such as, compressor work, work of pump, refrigerating capacity and COP of the new regenerative cycle using new generation refrigerant HFO-1234yf as compared to R-134a.

Keywords: Ejector, COP, HFO-1234yf, R-134a, New Regenerative Cycle

1. Introduction

Refrigeration and air-conditioning is a science of producing low temperature as compared to surroundings, since low temperature are maintained continuously it must run on a cycle. Vapour compression refrigeration system found wide application in MAC(mobile air conditioning) and stationary refrigeration applications. there are various means to increase the performance of the cycle and to increase COP such as to utilize the liquid vapour heat exchanger in vapour refrigeration system[]. but the main drawback is that the refrigerant at the compressor outlet is at high temperature (usually 80-110°C for R-134a); thus a large amount of energy must be rejected by the condenser to the environment. this waste heat can be utilized to increase the refrigeration performance of the system because improved system performance will reduce energy consumption as well as green house gases emissions. An ejector cooling system driven by low-grade heat energy can effectively use the waste heat to improve the system COP. An ejector based cooling system offers several advantages, such as no moving parts in the ejector, efficient utilization of the waste heat from the condenser of VC system and low cost. The study of refrigerant (CFCs, HCFCs and HFCs) ejectors for air-conditioning or refrigeration applications started in the mid-1950s for utilizing low-grade energy such as solar or waste heat energy as the heat source. The operation of a gas-to-gas or vapor-to-vapor ejector results mainly from the gas-dynamic effect and the momentum exchange of two gaseous streams (primary and secondary or entrained streams) inside the ejector. Two choking phenomena exist in the ejector performance one in the primary flow through the nozzle and the other in the entrained or suction flow. The entrained flow rate or the entrainment ratio (entrained-to-primary flow ratio $v = m_s/m_p$) of an ejector is affected by many factors. The physical phenomena involve supersonic flow, shock interactions, and turbulent mixing of two streams inside the ejector enclosure. It is so complicated that the design of an ejector to date still heavily relies on trials-and-errors

methods although a number of gas-dynamic theories for ejector analysis were developed by several researchers.

2. Literature Review

Many theories and experiments have been done to reduce the power consumption and increase the COP of the vapor compression refrigeration system. There are so many methods to increase the COP of the vapor compression system introducing the ejector is one of them. There are some theories and experiments which I reviewed for my study as follows:

Mark J. Bergander [1] studied a new regenerative refrigeration vapour compression system using R-22 as a refrigerant then he finds that there is an increase in COP of system and this increase is about 18% as compared with conventional system. Kairouni L. et al. [2] developed a improved cooling cycle for a conventional multi-evaporator simple compression system utilizing ejector for vapour pre compression is analyzed. The ejector increase the refrigeration cycle consists of multi evaporators. The COP of novel cycle is better than the conventional system. Arbel and Sokolov [3] presented a theoretical study of a solar driven combined VCR-VER using R-142b as a working fluid. The study compared the performance of the system with previous studies developed by Sokolov, where R-113 was used. They showed out not only technical but also ecological improvements by using R-142b. At this time use of R-113 is prohibited.

Jialin yu and Huazhao, (2007) [4] investigated a naval auto cascade refrigeration cycle with an ejector. The ejector is used to recover the some available work to increase the compression suction pressure this enables the compressor to operate at lower pressure ratio, which in turn improves the cycle efficiency. In this study they use the refrigerant as a mixture of R-23/R-134a. In this study they operated at condenser pressure of 40°C, the evaporator inlet temperature

-40 to 30°C, and mass fraction of R-23 is 0.15, the pressure ratio of the ejector reaches to 1.35, the pressure ratio of compressor is reduced by 25.8% and COP is increased by 19.1% over the conventional auto cascade refrigeration cycle.

A. khalil and E. Elgendy, (2011), [5] developed a mathematical model to design R-134a ejector and to predict the performance characteristics of vapor jet refrigeration system over a wide range of investigated parameters. Yinhai zhu and peixue jiang, (2012) [6] developed a model which combine the vapor compression system with ejector cooling cycle. The waste heat of condenser in vapor compression system is utilized to drive the ejector cooling cycle. In this system evaporator gets the additional cooling from ejector cooling cycle and this shows that there is an increase in refrigeration effect of combined cycle and finally increase in COP. This system shows the result for high compressor discharge temperature COP is improved by 9.1%.

The first theoretical principles of the ejector were elaborated by Parsons in 1900 while the first prototype was built by Leblanc (1910). Further improvements were introduced by Gay in 1931 [10]. Ejectors were first applied for refrigeration cycles by Heller in 1955 for absorption systems and by Badylkes in 1958 for vapor compression systems [9]. In the USA, the first application was reported by Kemper in 1966, but only patent is in existence while no experimental or theoretical background have been published. Following up on this early work, Kornhauser [11] has conducted a theoretical analysis and showed that the ideal ejector cycle resulted in 21% efficiency as compared with standard vapor compression cycle. The prototype unit was built, however its performance was much less than the ideal and reached at maximum only 5% using working fluids CFCs/ HCFCs/ HFCs. This was attributed to shortcomings in the design of the ejector, specifically too simplified two-phase flow model assumed in the design. Latest work on ejectors had concentrated on using them in transcritical CO₂ systems where high pressures allow for better recovery of the kinetic energy [12],[13], [14]. Detailed investigations were presented in [13], in particular a constant pressure mixing model for the superheated vapor ejector was established and the thermodynamic analysis of the ejector expansion for transcritical CO₂ was performed. It was found that the COP (Coefficient of Performance) of the transcritical CO₂ cycle with an ejector can be improved by as much as 16% over the basic transcritical CO₂ cycle for typical A/C operation conditions. However, only theoretical model is presented in the subject reference with no supporting practical experiments.

3. Ejector Working Principle

As outlined in Figure 1, a typical ejector consists of a motive nozzle, a suction chamber, a mixing section, and a diffuser. The working principle of the ejector is based on converting internal energy and pressure related flow work contained in the motive fluid stream into kinetic energy. The motive nozzle is typically of a converging-diverging design. This allows the high-speed jet exiting the nozzle to become supersonic. Depending on the state of the primary fluid, the flow at the exit of the motive nozzle might be two-phase.

Flashing of the primary flow inside the nozzle might be delayed due to thermodynamic and hydrodynamic non-equilibrium effects. The high-speed jet starts interacting with the secondary fluid inside the suction chamber. Momentum is transferred from the primary flow which results in an acceleration of the secondary flow. An additional suction nozzle can be used to pre-accelerate the relatively stagnant suction flow. This helps to reduce excessive shearing losses caused by large velocity differences between the two fluid streams. Depending on the operating conditions both the supersonic primary flow and the secondary flow might be choked inside the ejector. Due to static pressure differences it is possible for the primary flow core to fan out and to create a fictive throat in

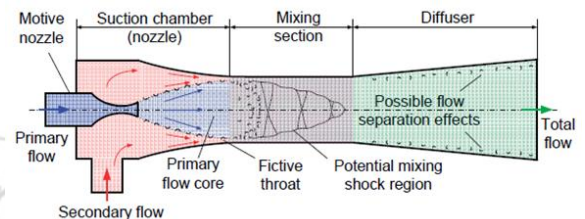


Figure1: Shows the Design of Ejector

which the secondary flow reaches sonic condition before both streams thoroughly mix in the subsequent mixing section. The mixing section can be designed as a segment having a constant cross-sectional area but often has a tapered inlet section. Most simulation models either assume mixing at constant area associated with pressure changes or mixing at constant pressure as a result of changes in cross-sectional area of the mixing section. The mixing process is frequently accompanied by shock wave phenomena resulting in a considerable pressure rise. The total flow at the exit of the mixing section can still have high flow velocities. Thus, a diffuser is used to recover the remainder of the kinetic energy and to convert it into potential energy, thereby increasing the static pressure. Typically, the total flow exiting the diffuser has a pressure in between that of the primary and the secondary streams entering the ejector. Therefore, the ejector acts as a motive-flow driven fluid pump used to elevate the pressure of the entrained fluid.

The two major characteristics which can be used to determine the performance of an ejector are the suction pressure ratio and the mass entrainment ratio. The suction pressure ratio is defined as the ratio of diffuser exit pressure to the pressure of the suction flow entering the ejector. The mass entrainment ratio is defined as the ratio of suction mass flow rate to motive mass flow rate. A well-designed ejector is able to provide large suction pressure ratios and large mass entrainment ratios at the same time

4. New Regenerative Cycle

In new generative cycle we compress the 50 -60 % part of desired pressure and the remaining pressure rise is achieved in ejector as a second step compression. The name regenerative is so called because it is using the heat from the the condenser there is no any extra source of energy. By using that energy we are achieving the final pressure.

The principle of the proposed system as shown in Figure 2 includes the main piping circuit (1), containing the evaporator (2), a compressor (3), an ejector device (4), a condenser (5), a separator tank (6), an intermediate heat exchanger (7) and an expansion valve (8). The circulation of a liquid phase of the working medium is provided by the additional liquid line (10 and 11), and a pump (9). The evaporator (2) absorbs the heat from source (12), while the condenser (5) is connected to the heat sink – high temperature heat receiver (13). It needs to note that the device as above can be used also for heating and in this capacity it can operate as a heat pump.

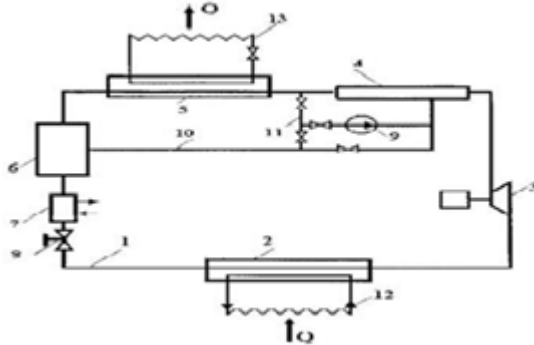


Figure 2: New Regenerative Cycle

Following processes of change in the state of a working fluid are depicted:

- 1-2 - evaporation of a part of the working fluid;
- 2-3 – compression of vapor in the compressor (the first step);
- 3-4-8 – mixing of vapor and liquid parts of the working medium in the ejector;
- 4-5 - compression of the working medium in the ejector (the second step);
- 5-6 - isobaric cooling of the liquid working medium;
- 6-7- compression of a part of the cooled liquid working medium by the pump;
- 7-8- expansion of this part of the cooled liquid working medium in the ejector;
- 6-1 – throttling of the evaporating part of the working fluid.

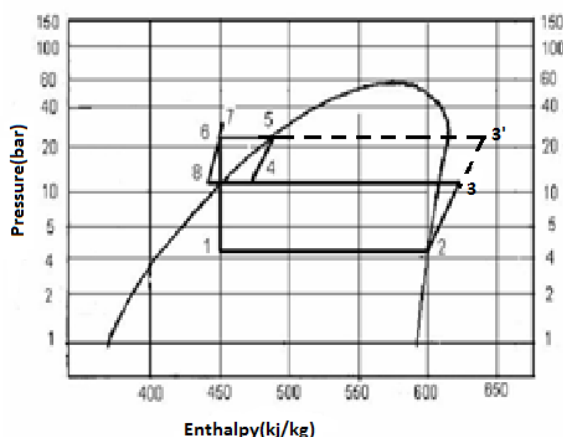


Figure 3: Comparison of P-h diagrams of the new refrigeration cycle with a two-phase ejector, Cycle 1 (points: 1-2-3-4-5-6-1 and 6-7-8-4) and the traditional cycle Cycle 2: (the point: 1-2-3'-6-1)

5. Mathematical Modelling and Analysis

To simplify the theoretical model of the refrigeration cycle with a vapor-liquid injector, the following assumptions that are analogous to the ones in the paper [12] are made:

- 1) Neglect the pressure drop in the condenser and evaporator and in the connection tubes. No heat losses to the environment from the system, except for the heat rejection in the condenser. The vapor stream from the separator is a saturated vapor and the liquid stream from the separator is a saturated liquid.
- 2) The flow across the expansion valve or the throttle valves is isenthalpic. The compressor has a given isentropic efficiency ($\eta_c = 0.8$). The evaporator has a given outlet superheat and the condenser has a given outlet temperature.
- 3) The flow in the ejector is considered a one-dimensional homogeneous equilibrium flow.
- 4) Both the motive stream and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section. The expansion efficiencies of the motive stream and suction stream are given constants. The diffuser of the ejector also has a given efficiency.

Using these assumptions, the equations of the ejector expansion R134a cycle have been stated. Assuming that the pressure before the inlet of the constant area mixing section of the ejector is P_b and the ejection ratio of the ejector (ratio of mass flows of vapor m_v and liquid m_f) is, $w = m_v / m_f$ 5.1 The motive stream follows an isentropic expansion process from pressure P_i to pressure P_b before it enters the constant area mixing section, or otherwise the value of entropy S_i for the moving stream in the point 7 and in the point 8 are equal:
 $S_7 = S_8$ 5.2

The corresponding enthalpy h_8 of the moving stream at the end of the isentropic expansion process can be determined using the P-h diagram for R22 or by equation
 $h_7 - h_8 = (P_7 - P_8) / \rho$ 5.3

Further, applying the conservation of energy across the expansion process, the velocity of the motive stream at the inlet of the constant area mixing section is given by equation:

$$U_{mb} = \mu \sqrt{2(H_8 - H_7)} = \mu \sqrt{2gH} \quad 5.4$$

where $g = 9.81 \text{ m/s}^2$, μ is a coefficient of discharge and H is pressure difference ($P_7 - P_8$) of the motive stream expressed in meters of a liquid column. With using a P-h diagram we can find the specific volume for both the motive stream in the point 8, V_8 , and the suction stream in the point 3, V_3 , as well as the same for their mixing in the point 4, V_4 .

In the case of two-phase flows, the cross-section area of the ejector mixing section per unit total ejector flow rate, a_m , can be determined by:

$$a_m = V_m / u_{mb} \quad 5.5$$

in which $V_m = 2 / (\rho_4 + \rho_5)$ is the mean specific volume of the vapor-liquid mixture at the ejector mixing section and ρ_4 , ρ_5 is the density of the vapor-liquid mixture in the states

corresponding to points 4 and 5 on the P-h diagram of figure3.

The method employed here for calculating the cross-section area of the mixing channel is characteristic of similar techniques of two-phase ejector calculation given by Fisenko [4]. From the known values of the velocity of the mixing stream and across mixing section area (cross-section area of the cylindrical channel at the mixing chamber outlet), it is possible to calculate the pressures of the working fluid flow at the mixing chamber outlet, P_{mix} , and at the ejector outlet after the diffuser, P_d .

In this event the following equations were applied:

$$P_b a_m + \left(\frac{1}{(1+w)}\right) * u_{mb} + \left(\frac{1}{(1+w)}\right) * u_{sb} = P_m a_m + u_{mix} \quad 5.6$$

$$P_d = P_{mix} + \rho (u_{mix}^2 - u_d^2) / 2 \quad 5.7$$

The former being the momentum conservation equation, whereas the latter is the energy conservation equation in the form of Bernoulli equation. In these equations u_{mb} , u_{sb} – velocities of the liquid and vapor flows (motive and suction) at the mixing section inlet, u_{mix} , u_d – mixture flow velocity at the diffuser inlet and outlet.

It needs to emphasize that in our case, the mixture velocity in the mixing chamber has to be somewhat higher than local sonic speed of this two-phase flow because in this case the efficiency of the vapor-liquid ejector increases [4]. In its turn, according to the known data [1], the speed of sound propagation α in a two-phase medium can be as low as only 20-50 m/s and for its estimate one can apply the equation:

$$\alpha^2 = kP / \rho_{mix} \text{ or } \alpha^2 = P / \rho_f \beta (1 - \beta) \quad 5.8$$

where k is isentropic coefficient, P , ρ_{mix} is pressure and density of the two-phase flow, ρ_f is density of the liquid phase and β is the volumetric content of vapor in the mixture.

5.1 Governing Equations

A. Compressor

The compressor is assumed to be non-isentropic. Process 1-2s is an isentropic compression process, while process 1-2 is the actual compression process. The actual enthalpy of state 2 is expressed by:

$$h_3 = h_2 + (h_{3s} - h_2) / \eta_c \quad 5.9$$

where η_c is the isentropic efficiency of the compression process.

The enthalpy and entropy of the refrigerant at state 1 are determined by the temperature and pressure at the compressor inlet as:

$$h_2, s_2 = f(T_2, P_2) \quad 5.10$$

The refrigerant enthalpy at state 2s for the isentropic process is:

$$h_{3s} = f(s_{3s}, P_3) \quad 5.11$$

Where $s_{3s} = s_2$

The quantity of energy needed for the compression of the vapor flow m_v by the compressor with the performance η_c is determined by the expression

$$\ell_c = m_v(h_3 - h_2) / \eta_c \quad 5.12$$

B. Condenser/Generator

In the regenerative system shown in Fig. 2, low temperature fluid, which generates vapor by absorbing heat from the high temperature compressor discharge, becomes the working fluid to drive the ejector. Note that the condensing temperature in the basic refrigeration cycle is lower than the evaporating temperature of the ejector cycle. Therefore, only part of the sensible heat can be used to vaporizing the refrigerant of the ejector cycle in the generator. The total energy balance in the vapor generator is:

$$m_3(h_3 - h_4) = m_{10}(h_{10} - h_9) \quad 5.13$$

For the heat exchanger design, there is a minimum temperature difference at the generator's two sides:

$$T_3 > T_{10} + \Delta T_g, T_4 = T_9 + \Delta T_g \quad 5.14$$

The fluid state at the generator exit is:

$$T_4, s_4 = f(P_4, h_4) \quad 5.15$$

$$T_{10}, h_{10}, s_{10} = f(P_3) \quad 5.16$$

$$Q_g = m(h_3 - h_4) \quad 5.17$$

C. Pump:

In the regenerative system the pump is also used to raise the pressure of mixing fluid. The total mass balance at the pump is:

$$m_4 + m_5 = m_{10} + m_{11} \quad 5.18$$

$$h_7, s_7 = f(T_7, P_7) \quad 5.19$$

The quantity of energy ℓ_p , consumed by a pump in compressing a working fluid is calculated with the formula $\ell_p = m_f \Delta P_{7-6} / (\rho_{mix} * \eta) = m_f(h_7 - h_6) / (\rho_{mix} * \eta)$ 5.20 where η is efficiency (coefficient of efficiency) of the pump.

D. Evaporator:

The function of the evaporator in the basic refrigeration system differs from that in the hybrid system. For the evaporator, assume that the refrigerant at the exit (state 1) is super heated. The governing equations for the evaporator are then

$$P_1 = f_{sat}(T_{evp.}) \quad 5.21 \quad T_2 = T_{evp.} + \Delta T_e \quad 5.22$$

$$h_2, s_2 = f(T_2, P_2) \quad 5.23$$

Refrigerating capacity of the system Q_o

$$Q_o = m_v(h_2 - h_1) \quad 5.24$$

E. Ejector:

The ejector works as a compression device where the high pressure primary flow (state 13) entrains the low-pressure secondary flow (state 7) into the ejector. Previous studies have shown that the ejector performance is influenced by both the ejector geometry and the operating conditions. The ejector performance is usually evaluated based on the combined mass flow rates of the two flows. The mass flow rate of the primary flow of the ejector is determined by the ejector's structure and the thermodynamic properties of the primary flow. Assuming isentropic flow, the mass flow rate of the primary flow through the nozzle, m_{11} , when choked can be expressed by (Huang et al., 1999; Zhu et al., 2007).

$$m_{11} = A_t [\Psi_{ej} \gamma P_{11} \rho_{11}]^{1/2} (\rho / (1 + \gamma))^{(\gamma+1)/2(\gamma-1)} \quad 5.25$$

where Ψ_{ej} represents a coefficient related to the isentropic efficiency of the compressible flow in the nozzle and P_{11} and T_{11} are the pressure and temperature of the primary flow, respectively, at the ejector inlet.

The characteristics of the secondary flow in the ejector are more complex than those of the primary flow. In the critical mode, the secondary flow is choked in the ejector (hypothetical section 80) which determines the ejector performance. Zhu and Li (2009) derived the following to calculate the secondary flow mass flow rate:

$$m_6 = \frac{2\pi p_7}{R_m - R_8} \sqrt{\frac{p_8}{\rho_8}} (R_m^3/6 - R_m R_8^2/2 + R_8^3/3) 5.26 \text{ where } \rho_8, p_8 \text{ and } V_8 \text{ are the density, pressure and velocity of the primary flow, respectively, at a hypothetical section where the secondary flow is choked and } R_8 \text{ is the radius of the mixing layer in this hypothetical section}$$

5.2 Other energy characteristics of the cycle are defined as follows:

- Thermal performance (Q_h)
 $Q_h = (m_f + m_v) (h_5 - h_6) = (m_f + m_v) c_p (T_5 - T_6) 5.27$
- The compression work ℓ done by the compressor and the pump
 $\ell = \ell_c + \ell_p 5.28$
- The coefficient of performance (COP) of the two phase ejector cycle can be determined by:
 $COP = Q_h / \ell 5.29$
- For the basic one-step refrigeration cycle operating in the same temperature range, the evaporator heat capacity Q_{bo} and the condenser heat capacity Q_{bh} are given by:
 $Q_{bo} = m_v (h_{b2} - h_b) 5.30$
 $Q_{bh} = m_v (h_{b3} - h_{b6}) 5.31$
- The compressor work of the same basic cycle operating without using the ejector is found by:
 $\ell_b = m_v (h_{b3} - h_{b2}) \eta 5.32$

where h_{bi} are the enthalpies of the corresponding points in the P-h diagram cycle of Fig. 5 where a comparison is shown between one step compression conventional cycle and the new cycle. Then, the performance of the basic refrigerant cycle with the same temperature range is given by:

$$COP_b = Q_{bh} / \ell_b 5.33$$

In carrying out calculations it has been assumed that coefficient of efficiency of the hydraulic pump and compressor is equal to 0.8, and corresponding values (magnitudes) of the evaporator capacity for both cycles under consideration are identical.

- Velocity of outflow of the motive fluid from the ejector nozzle
 $U_{mb} = \mu \sqrt{2(H_7 - H_8)} 5.34$
- Cross-section area of the mixing nozzle
 $A_m = 2 / (\rho_4 + \rho_5) u_{mb} 5.35$

6. Results and Discussions

Performance of regenerative cycle with R134a: For the basic cycle of vapor compression refrigeration system using R134a as a refrigerant COP is=2.935, and for the regenerative cycle using ejector as second step compression COP is=3.589 which is higher than basic cycle of vapour

compression system. COP is increased by 22%. Where as work of compression is also reduced.

Performance of regenerative cycle with R1234yf: For the basic cycle of vapor compression refrigeration system using R1234yf as a refrigerant COP=3.245, and for the regenerative cycle using ejector as second step compression COP=3.626, Which is higher than the basic cycle of vapor compression system. COP is increased by 11.7 %, Where as work of compression is also reduced.

Table 1: Qualities and quantity of R134a, R1234yf refrigerants in characteristics points are indicated on the diagram.

Points	P (bar)	Ti (°C)	Xi
1	4.2	-5	0.22
2	4.2	-5	1
3	11.9	35	1
4	11.9	30	0.1
5	24.2	55	~0.01
6	24.2	40	0
7	30	41	0
8	11.9	30	~0.02
3"	24.2	85	1

Table 2: Effect of evaporator temperature on COP of R-134a and R-1234yf

Te(°C)	COP of R-134a	COP of R-1234yf
-10	3.39	3.321
-8	3.42	3.479
-6	3.49	3.524
-4	3.541	3.262
-2	3.612	3.767
0	3.684	3.858
2	3.725	3.991
4	3.812	4.191
6	3.924	4.287
8	4.125	4.401
10	4.249	4.627

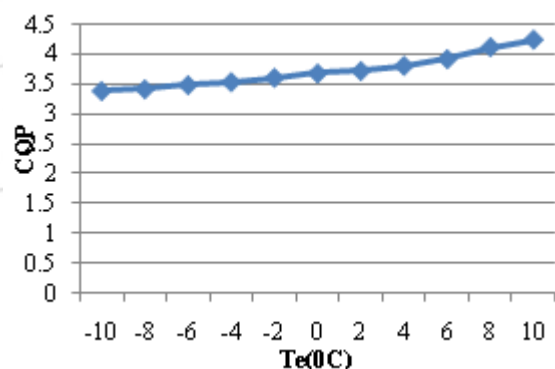


Figure 4: Effect of evaporator temperature on COP with R134a

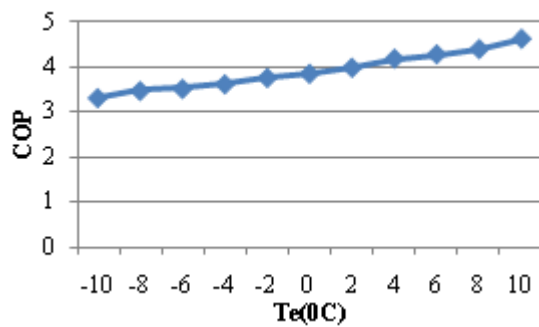


Figure 5: Effect of evaporator temperature on COP with R1234yf

7. Conclusions

In this paper performance analysis of a new regenerative vapor compression cycle is done with R134a, R1234yf as a refrigerant. Following things can be concluded,

- 1) COP of HFO-1234yf is 8.89% higher than R-134a at higher evaporator temperature.
- 2) COP of the regenerative cycle is higher than the conventional vapour compression cycle i.e. COP of R-134a is increased by 22% and COP of R-1234yf is increased by 11.7%.

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