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Mathematical Simulation of Fourth Generation Refrigerant R1243zf in Single Stage and Double Stage Vapour Compression Refrigeration System

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Abstract: In this paper a new fourth generation, environment friendly refrigerant R1243zf has been used in performance analysis between single stage and double stage vapour compression refrigeration system working between the same minimum and maximum pressure limits. The analysis has been done by mathematical simulation in Engineering Equation Solve (EES 10.439D, 2019). Results show that in low temperature cooling applications, a single stage vapour compression system does not perform well and a cascade system is the best optimum solution. In the given pressure limit, with the use of double stage vapour compression system, the increase in COP is 14% and decrease in compression work is 11.49%. The effects of evaporator temperature and condenser temperature have been seen on COP for both the systems and results show that with decrease in evaporator temperature, COP is decreasing, where as with decrease in condenser temperature COP is increasing.

Keywords : COP, R1243zf, EES, fourth generation

I. INTRODUCTION

Many industrial applications requires relatively very low temperature, and it is very difficult to achieve this temperature by the help of single vapour compression refrigeration system. A large temperature range also means a large pressure range and it is very difficult to work with a single stage cycle at this large pressure range. Cascade refrigeration system is one of the way to deal this type of situations.

The use of refrigerant was started in early 1800 century with use of natural refrigerant, which were replaced by CFCs in 1928 with increase in thermal performance, which were banned under Montreal Protocol (1987) due to their high ozone layers depletion (ODP) properties and then they are substituted by hydrocholoro carbon (HCFCs) and hydrocarbons in 1980s.

Thereafter researchers noted that HCFCs are responsible for ozone depletion and bear high global warming potential. As per guidelines of Kyoto Protocol (1997), HCFCs have to be phased out by 2020-2030 and HFCs by 2025-2040 [1,2] HFO (hydrofluoroolefin) are going to be our future refrigerants with low ozone depletion potential (ODP) and low global warming potential (GWP).The basic properties of new future HFO refrigerants expected as R410a and R32 alternatives which are presently used in refrigerators and room air conditioners. R1243zf is expected to be a good alternative of R134a. Unfortunately the problem is with its flammability, which is A2 category in ASHRAE classification safety groups[3]. Triple point data of a refrigerant, which defines the lowest temperature range at which any refrigerant may circulate in liquid state is very important for refrigerating industry. The triple point of R1243zf is 122.8 K[4]. In another experimental analysis in laboratory it has been find that the normal boiling temperature and critical pressure of R1243zf are 247.73 K and 3630.6 kPa, respectively[5]. Gang et al. [6] proposed a modified vapour compression cycle for application in freezers. Phase change material (PCM) technique is used to increase the degree of subcooling and results show that this technique has great potential to reduce energy consumption. Ehsan et al.[7] analyzed a CO_2/NH_3 cascade refrigeration system and concluded that by advance exergy analysis the system efficiency can be improved about 42.12%. Zhili et al.[8] performed energy and exergy analysis of cascade refrigeration system working with different low GWP refrigerant and suggested that R161 is suitable to use in high temperature cycle and R41 and R170 are suitable to use in low temperature cycle. While selecting a refrigerant, it is very difficult to meet all the selection criteria for a specific application by a single component refrigerant, blends are becoming more and more important[9].

Sanchez et al.[10] Performed energy analysis of vapour compression refrigeration system using refrigerants R134a, R152a, R600a , R290 , R1234yf and R1234ze(E) (HFOs) and found that these can be a good alternative to R134a without compromising the performance of refrigeration system. Ranendra and Bijan[11] studied a cascade refrigeration system for low temperature application using refrigerant pair R41-R404a and R41-R161, and concluded that R161 can be a better option to R404a because of its thermo physical and environmental properties.

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Esbari et al.[12] experimentally evaluated the possibility of R1234yf as a drop in replacement to R134a and found that COP is lower about 19% with R1234yf and minor difference in higher condenser temperature.

And these differences can be reduced by using internal heat exchanger. Zhili et al.[13] studied the possibilities of low GWP refrigerants in three stage cascade refrigeration system and found that R1150 is alternative of R14 in low temperature cycle, and R23 can be replaced by R41 and R170 in medium temperature cycle and R152a, R717 and R161 are recommended refrigerants for high temperature cycle. Ngoc[14] evaluated a air conditioning refrigeration cycle with R1243zf and found the highest COP was achieved by R1243zy as refrigerant and it has a good capabilities to replace R134a in a refrigeration cycle.

II. SYSTEM DESCRIPTION

The schematic diagram for single stage vapour compression system and double stage stag vapour compression refrigeration system along with their T-s diagrams are given in figure 1 and figure 2 respectively. Both the refrigeration system, are working between the same maximum and minimum pressure limits. In double stage vapour compression system the refrigerant is same in both low temperature cycle and high temperature cycle.

In single stage system, the refrigerant is saturated vapour at the exit point of evaporator and leaves the condenser as saturated liquid. The system works between the pressure limits of 0.14 Mpa to 0.32 Mpa.

In double stage vapour compression system, refrigerant enters the low pressure compressor (LPC) as saturated vapour at 0.14 Mpa and leaves the compressor at intermediate pressure 0.32 MPa and then enters the heat exchanger and transfers the heat to the high pressure cycle refrigerant. In high pressure cycle, refrigerant leaves the heat exchanger as saturated vapour and enters the high pressure compressor at 0.32 MPa pressure and compresses the refrigerant to 0.8 MPa and then enters in to condenser.

Table:1. Thermo physical properties of refrigerant used [3]

Refrigerant	Normal boiling temperature (Kelvin)	Refrigerant molar mass (g.mol^{-1})	Critical Temperature (Kelvin)	P_c(Critical Pressure (kPa)	GWP (1TH=100)	Safety group (ASHRAE)	ODP
R1243zf	247.73	96.051	376.93	3518	<1	A_2	0

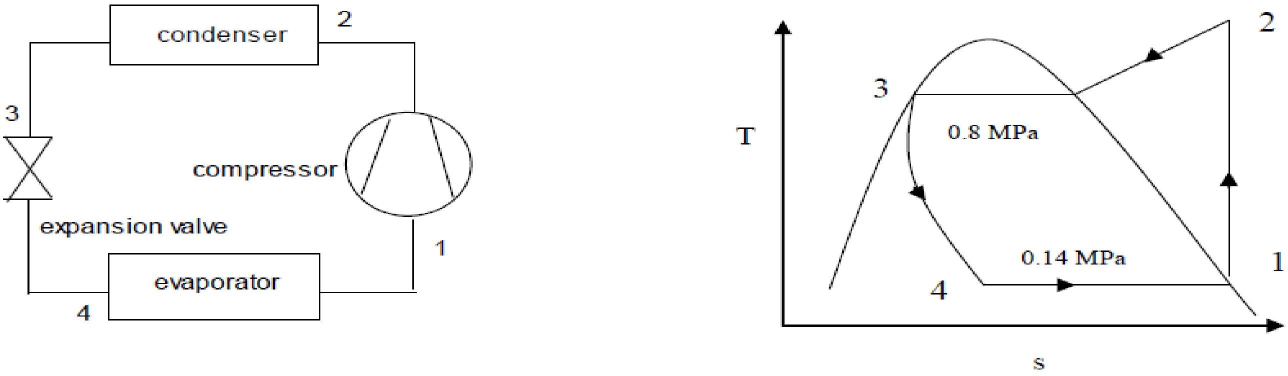

Figure: 1. vapour compression refrigeration single stage system (a) System Schematic (b) T-s diagram

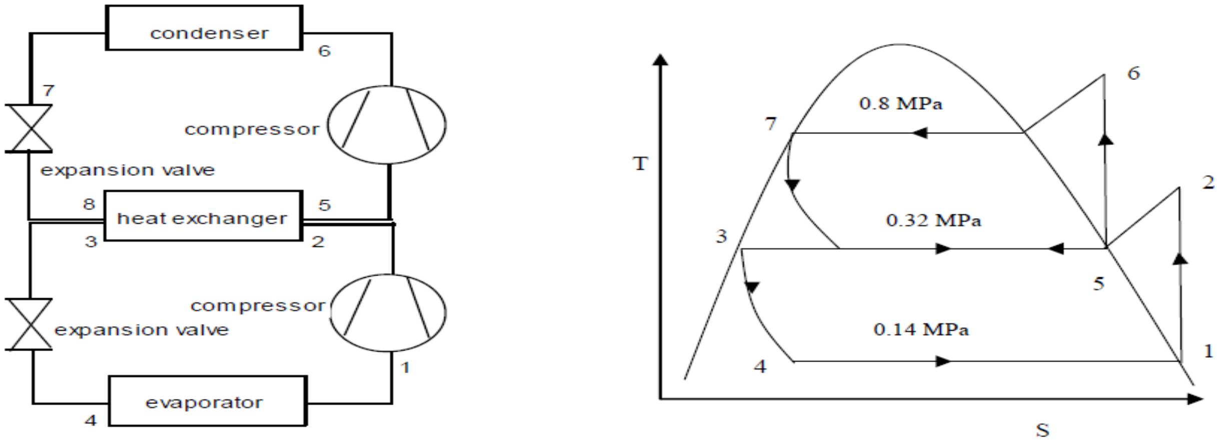

Figure: 2. Vapour compression double stage system (a) System Schematic (b) T-s diagram

III. MATHEMATICAL MODEL

The following assumptions have been made as per the first law analysis of thermodynamic model.

1. The processes are assumed to be steady flow in all the components.
2. Pressure drop and heat loss are negligible in all the system components and connecting pipes.
3. The compression process in compressor is assumed to be isentropic.
4. The enthalpy is constant in expansion process.
5. In both the system, the condenser pressure is assumed to be 0.8 MPa and evaporator pressure is 0.14 MPa.
6. Refrigerant Mass flow rate in single stage refrigeration system is assumed to be 0.05 kg/s. And in double stage system, the mass flow rate in high temperature cycle is assumed to be 0.05kg/s.

On the basis of above assumptions the following equations have been used. For single stage vapour compression system, the input work to compressor can be expressed as:

$$W_c = \dot{m}(h_2 - h_1) \qquad 1$$

And cooling capacity obtained by evaporator can be expressed as:

$$Q_L = \dot{m}(h_1 - h_4) \qquad 2$$

Where $\dot{m}$ flow rate of refrigerant (kg/s) in single stage system, h_1, h_2 and h_4 are enthalpy in kJ/kg.

The coefficient of performance can be calculated as:

$$COP = \frac{Q_L}{W_c} \qquad 3$$

The work of compression for double stage refrigeration system is given as:

$$W_{net} = W_{c1} + W_{c2} \qquad 4$$

Where W_{c1} and W_{c2} are the input work of compression of high pressure cycle and low pressure cycle respectively, calculated by the following equation:

$$W_{c1} = \dot{m}_1(h_6 - h_5) \qquad 5$$

$$W_{c2} = \dot{m}_2(h_2 - h_2) \qquad 6$$

Where $\dot{m}_1$,$\dot{m}_2$ are refrigerant mass flow rate of (kg/s) of high temperature cycle and low temperature cycle respectively.

Energy balance equation for heat exchanger is given as:

$$m_1(h_5 - h_8) = m_2(h_2 - h_{3)} \qquad 7$$

COP of double stage vapour compression system is calculated as:

$$COP = \frac{Q_L}{W_{net}} \qquad 8$$

IV. VALIDATION OF MATHEMATICAL MODEL

To check the accuracy of mathematical model in EES, a test has been performed between literature result and current model. The input variables have been taken from Ngoc A.L [14].

Table:2. Input parameters for model testing[14]

Parameters	Values
Pressure at compressor inlet (Pa)	319000
Pressure at compressor outlet (Pa)	1277000
Discharge temperature (°C)	65.3
Mass flow rate (Kg/s)	30.71

Table:3. Performance parameters of Literature and current model

System	Work input to compressor (kW)	Cooling capacity (kW)	COP
Ngoc A.L. [14]	1.0	4.416	4.42
Current model	1.025	4.445	4.336
change	2.5%	0.6%	1.9%

V. RESULT AND DISCUSSION

A study to find the difference in performance of single stage and double stage vapour compression system working between same pressure limits has been performed using mathematical simulation in EES. The data of performance parameters of both the systems; single stage and double stage are given in Table 4. It is noteworthy that the change in cooling capacity is very small where as the change in compressor work input is considerable which is 12%. This is due to the reason that in single stage refrigeration system working on higher pressure ratio, the compression ratio is also high and at high compression ratio the volumetric efficiency decreases and work of compression increases.

Table: 4. Performance parameters of single stage and double stage vapour compression system

Type of Refrigeration System	Work input to compressor (kW)	Cooling capacity (kW)	COP
Single Stage system	1.914	6.86	3.583
Double Stage system	1.684	6.916	4.105
change	12% Decrease	0.8% Increase	14.5% Increase

The change in condenser temperature with COP for both the systems is shown in figure 3. It was observed that the increase in condenser temperature leads to decrease in COP of both the systems (see figure 3). The difference of COP between single stage and double stage refrigeration system is maximum at 31 °C which is 22% and minimum 1t 40 °C, which is 14%.The changes in condenser pressure with COP for both the system; single stage and double stage are shown in figure 4 and it can be observed form figure 4 that COP is decreasing with increase in condenser pressure. It has been also be observed from the figure 4 that the difference of COP between single stage and double stage system is almost constant in the given condenser pressure range between 0.76 Mpa to 0.85 Mpa. The changes in COP with evaporator temperature are given in figure 5 which indicates that the COP is decreasing with decrease in evaporator temperature.

The COP difference between Single stage and double stage refrigeration system is also increasing with decrease in evaporator temperature. At -10°C The COP difference is 11.7% and at -28°C the COP difference is 18.8%. The changes of COP with evaporator temperature are shown in Figure 6 and it can be concluded from the figure 6 that with increase in evaporator pressure the COP is increasing. The COP is increasing due to the reason that with increase in evaporator pressure, the pressure ratio decreases and as the pressure ratio decreases , the volumetric efficiency of compressor increases and hence COP increases.

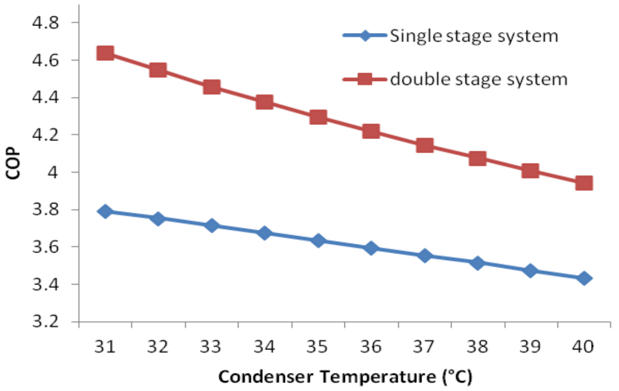

Fig.3. Change in COP with condenser temperature

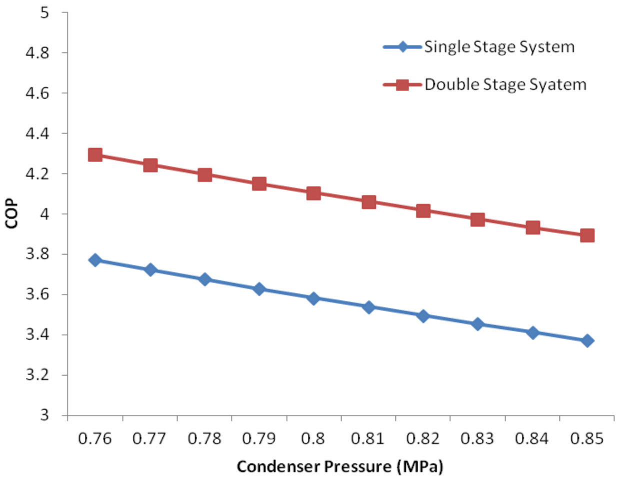

Fig.4. Change in COP with condenser pressure

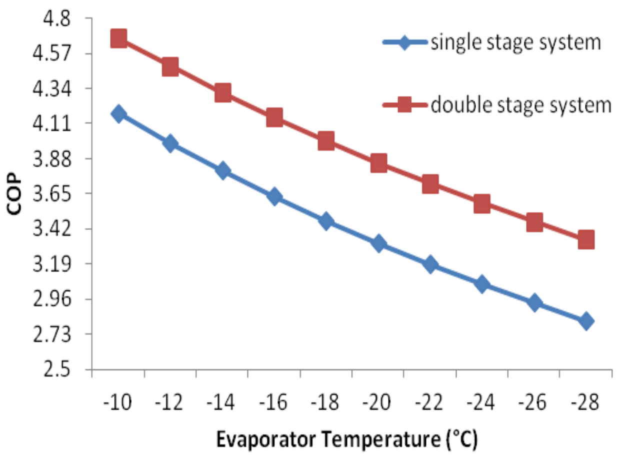

Fig.5. Change in COP with evaporator temperature

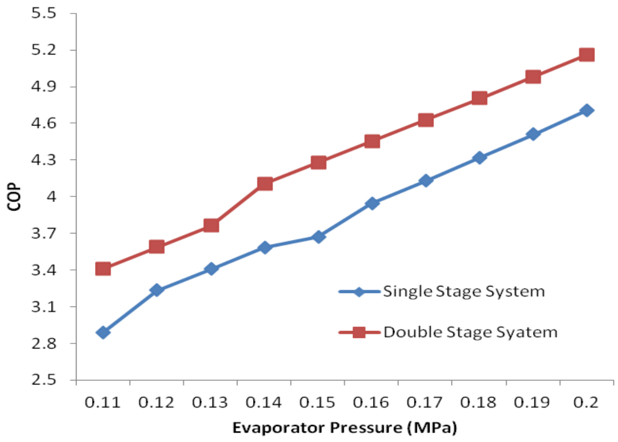

Fig.6. Change in COP condenser pressure

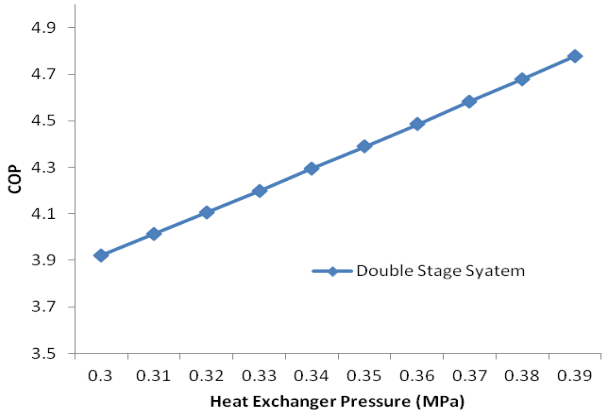

Fig.7. Change in COP with heat exchanger pressure

The changes in COP with heat exchanger temperature for double stage vapour compression system are shown in Figure 7. It can be observed from the figure 7 that the COP is increasing with increase in heat exchanger or intermediate pressure because at higher intermediate pressure, the net work input to compressor is minimum and COP is maximum.

VI. CONCLUSION

In this paper a thermodynamic comparison has been performed between single and double stage vapour compression refrigeration system using fourth generation, environment friendly refrigerant HFO-1243zf. The results obtained from this thermodynamic analysis are as follows:

1. Refrigerant R1243zf can be better alternative for low temperature cooling.
2. Double stage vapour compression system gives the better performance in comparison to single stage refrigeration system if the working pressure limit is moderate.
3. In double stage refrigeration system, working at higher intermediate pressure gives better COP.

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