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THERMODYNAMIC PERFORMANCE ANALYSIS OF DEDICATED MECHANICALLY SUBCOOLED VAPOUR COMPRESSION REFRIGERATION SYSTEM

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ABSTRACT

In this work, the thermodynamic analysis of dedicated mechanically subcooled vapour compression refrigeration system is presented. A software based computer program in EES has been formulated for computation of the performance results. The effect of varation of degree of subcooling (5-30ºC), evaporator temperature (-20 to 10ºC) and condenser temperature (30-50ºC) has been investigated for energetic and exergetic performance of the system. The analysis of the system has been carried out using zero ODP and very low GWP (1 to 4) refrigerants viz.HFO-R1234ze and R1234yf to compare the performance of HFC-R134a. The results depicts that the COP and exergetic efficiency of dedicated subcooled VCR cycle are better than that of simple VCR cycle. Refrigerant R1234ze performs better than R1234yf and comparable to R134a.

Keywords: *VCR, Subcooling, COP, R1234ze, R1234yf, Exergetic Efficiency, Dedicated, R134a*

INTRODUCTION

Demand of energy, economy and eco-sustainability are the major considerations of a refrigeration system design. The demand of high grade energy is growing continuously in the developing countries viz. India. The cost of electricity is also increasing with its demand. The distortion of surrounding ecosystem with the use of high GWP refrigerants (HFCs) in the refrigeration and air-conditioning systems is a current issue. The high GWP HFCs have to be phased out as per the European Union (EU) regulation (Regulation (EU) No 517/2014 [1]). Many researchers have proved that the mechanically sub-cooling of vapour compression refrigeration (VCR) cycle improves its performance. Zubair [2, 3] and Zubair et al. [4] have investigated a dedicated and an integrated mechanical sub-cooling system from the thermodynamic standpoint. In these studies, it was found that the system performance had improved when operating in situations where the difference between the condensing and evaporating temperatures is large. Couvillion et al. [5] developed a mathematical model of a dedicated mechanical sub-cooling system by considering the individual component models of the equipment involved in the system. They found an improvement of 6-8% in the coefficient of performance (COP) and 20-170% in the capacity over a conventional simple cycle. Qureshi et al. [6] analysed that the load carrying capacity of the evaporator increased by approximately 0.5kW when R22 was subcooled in the main cycle by 5-8^{0}C. They noted that using the sub-cooling, the second-law efficiency of the cycle increased by an average 21% and this percentage increase is direcltly proportional to the ambient temperature. Khan and Zubair [7] have investigated finite time thermodynamic models of simple vapour compression refrigeration, two stage VCR system (TSS), dedicated mechanical sub-cool VCR system (DMSS), integrated mechanical sub-cooling VCR system (IMSS). They predicted that the optimum value of COP is obtained by varying the parameters of heat exchanger and reduced sub-cooler saturation temperature. Thornton et al. [8] have used dedicated mechanical sub-cooling VCR cycle to sub-cool the condenser exit of main VCR cycle. They predicted the optimum value of sub-cooling evaporator temperature which established the design rule for optimum distribution of heat exchanger area. Arora et al. [9, 10, 11], Arora and Kaushik [12] and Arora [13] have analysed vapour compression and absorption refrigeration systems for energetic and exergetic performance improvement. Llopis et al. [14, 15] have theoretically analysed the energy performance enhancement of CO_2 transcritical refrigeration system using dedicated mechanical sub-cooling cycle. They observed 20% increase in COP and 28.8% increases in cooling capacity of combined and reviewed sub-cooling method viz. internal heat exchanger, economizers, thermoelectric systems, dedicated sub-cooling methods of CO_2 refrigeration cycle. They concluded that the sub-cooling is a worth method to increase the performance of refrigeration cycle and deserve future development. Many researchers have suggested the replacement of R134a with R1234ze and R1234yf in refrigeration

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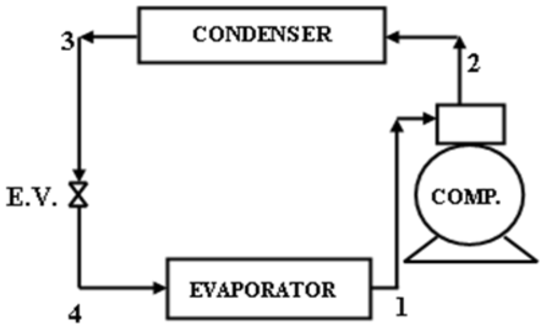
systems. Mota-Babiloni et al. [16] reviewed the recent developments in commercial refrigeration and focused on system modifications (as dedicated sub-cooling or the implementation of ejectors), tri-generation technologies (electrical, heating and cooling demand) and better evaporation conditions control. They concluded that the hydrocarbon, HFO and CO_2 are HFC replacements. Kalla et al. [17, 18] reviewed alternative refrigerants for vapour compression refrigeration system and investigated the performance of R22 and its substitutes in air-conditioners. Yataganbaba et al. [19] performed exergy analysis on a two evaporator vapour compression refrigeration system using R1234yf, R1234ze and R134a as refrigerants. The two refrigerants R1234yf and R1234ze were good alternatives to R134a regarding their environment friendly properties. Dixit et al. [24-26] carried out energy and exergy analysis of absorption-compression cascade and waste heat driven triple effect refrigeration cycles.

The literature survey gathered from different established data sources viz. Google Scholar, Research gate, Science direct etc. reveals that the extensive research have been done on simple VCR cycle. Out of these, some studies based on dedicated mechanical sub-cool cycle have been considered to enhance the performance of the cycle. It is observed that limited work has been carried out to explore the performance of dedicated mechanical subcool cycle using HFO refrigerants viz. R1234yf and R1234ze.



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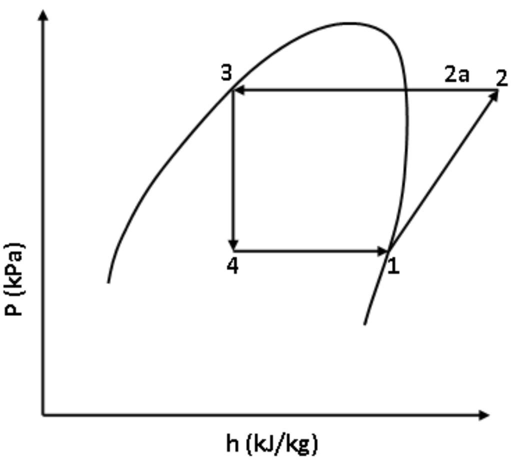

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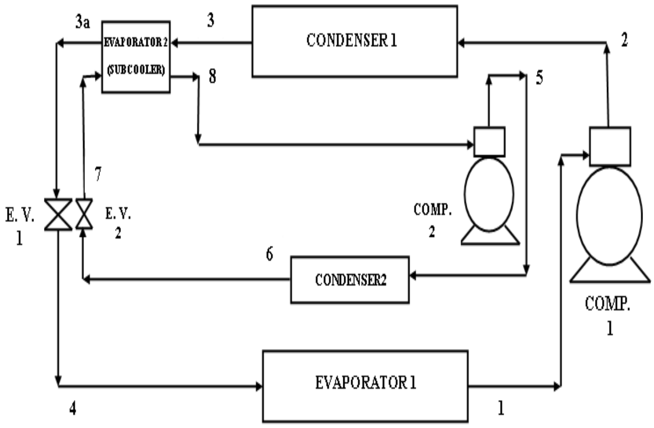
Figure 1. (a) Saturated Vapour Compression Refrigeration Cycle, (b) P-h diagram of saturated vapour compression refrigeration cycle

The present work investigates the thermodynamic performance of dedicated mechanically subcooled VCR cycle using R1234yf, R1234ze and R134a. The comparison of considered refrigerant has also been explored for drop-in replacement of R134a to R1234ze and R1234yf. The first and second law based performance parameters viz. COP and exergetic efficiency have been computed with the variation in degree of sub-cooling (DOS), evaporator temperature and condenser temperature.

DESCRIPTION OF SYSTEM AND MODELLING

Description of System

The current work presents first and second law based thermodynamic analysis of dedicated mechanically subcooled vapour compression refrigeration system (DSC) using R1234ze, R1234yf and R134a. The mechanical sub-cooling of main VCR cycle has been carried out through the subcooler or evaporator of another simple VCR cycle designated as subcooler VCR cycle.



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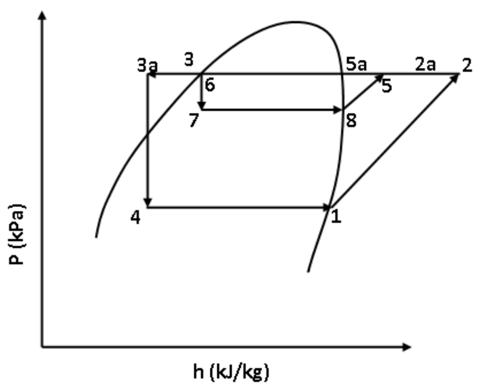


(b)

Figure 2. (a) Schematic diagram of dedicated mechanically subcooled vapour compression refrigeration cycle, (b) P-h diagram of dedicated sub-cooled vapour compression cycle

Figure 2(a) and 2(b) illustrate the schematic and P-h diagrams of dedicated mechanically subcooled vapour compression refrigeration system respectively. It comprises of two sections i.e. subcooled and subcooler vapour compression refrigeration cycles. The evaporator of subcooler VCR cycle receives the heat rejected by the condensate liquid refrigerant of subcooled VCR cycle and acts as subcooler. In this way, the two cycles are coupled together through subcooler or evaporator 2 in order to achieve low temperature in evaporator1. The refrigerants considered for the energy and exergy analysis of the system are R134a, R1234yf and R1234ze. The same refrigerant has been considered in the two sections simultaneously.

The subcooled VCR cycle is in lower section which consists of compressor1, evaporator1, expansion valve1, subcooler and condenser1 and the subcooler VCR cycle is upper section which consists of compressor2, condenser2, expansion valve2 and evaporator2 or subcooler. The vapour refrigerant leaving the evaporator1 of subcooled VCR cycle at state point1 enters into the compressor1 and is being compressed. The refrigerant at high temperature and pressure leaving the compressor1 at state point 2 and enters in to the condenser1. The liquid condensate has been subcooled from state point 3-3a by the evaporator2 of the subcooler cycle. The subcooled liquid refrigerant is throttled by the expansion valve1 from state point 3a-4. The low temperature liquid-vapour mixture of refrigerant enters in to the evaporator1 and low temperature is achieved in the evaporator1.

Similarly, the saturated vapour of the refrigerant leaving the evaporator2 or subcooler at state point 5 enters in to the compressor2. The compressed high pressure, high temperature vapour refrigerant enters into the condenser2 at state point 6. The vapour phase of the refrigerant is converted in to liquid in the condenser2. The liquid condensate refrigerant leaving the condenser2 at state point7 is expanded freely by the expansion valve2 from state 7-8. The low temperature liquid-vapour mixture of refrigerant enters into the evaporator2 or subcooler and absorbs heat from the condensate of condenser1 from state 8-5 and transformed into saturated vapour.

In this way, the mechanical sub-cooling is produced by the subcooler VCR cycle in the subcooler from states 3-3a. The evaporator temperature ($T_{e,SC}$) of the subcooler cycle is more than the evaporator temperature of subcooled cycle. The sub-cooling of liquid condensate refrigerant of subcooled cycle enhances the net specific refrigerating effect as shown in Figure 2 from states 4-1. The increase in refrigerating effect enhances the coefficient of performance (COP) of the subcooled cycle. However, the compressor work of the subcooler cycle reduces the COP of the overall cycle.

Thermodynamic Modelling

In the present work, the energy and exery analysis of dedicated mechanically subcooled vapour compression cycle have been carried out. In which the subcooler VCR cycle produces mechanical sub-cooling in the subcooled VCR cycle through subcooler using considered refrigerants R134a, R1234yf and R1234ze. The various performance parameters viz. compressor work, COP, exergetic efficiency, total exergy destruction rate and exergy destruction ratio have been explored for different variables viz. degree of sub-cooling, evaporator temperature, effectiveness of subcooler, isentropic efficiency of compressor and condenser temperature.

A set of computer code has been formulated in Engineering Equation Solver (EES) (Klein and Alvarado [22]) to perform the analysis. The steady state governing equations, based on first and second law of thermodynamics have been developed.

Energy Analysis

The first law based work, energy and mass conservation principles are presented in equations (1)-(2).

$$\sum \dot{Q} - \sum \dot{W} = \sum \dot{m}_o h_o - \sum \dot{m}_i h_i \tag{1}$$

$$\sum \dot{m}_i - \sum \dot{m}_o = 0 \tag{2}$$

where $\dot{Q}$, $\dot{W}$ and $\dot{m}$ are the rate of heat, work and mass transfer crossing the boundary of the system respectively.

The coefficient of performance (COP) of simple, dedicated mechanically subcooled, subcooler and overall VCR cycles are given by the equation (3)-(9).

$$COP_{SVCR} = \frac{\dot{Q}_e}{\dot{W}_{comp}} \tag{3}$$

where $\dot{Q}_e$ is the net refrigeration effect produced and $\dot{W}_{comp}$ is the actual work of the compressor.

$$COP_{\,DSC} = \frac{\dot{Q}_{e,DSC}}{\dot{W}_{comp1}} \tag{4}$$

$$\dot{Q}_{e,\,DSC} = \dot{m}_{r,\,DSC}(h_1 - h_4) \tag{5}$$

$$\dot{W}_{comp1} = \dot{m}_{r,\,DSC}(h_2 - h_1) \tag{6}$$

$$COP_{\,SC} = \frac{\dot{Q}_{e,SC}}{\dot{W}_{comp2}} \tag{7}$$

$$\dot{W}_{comp2} = \dot{m}_{r,\,SC}(h_6 - h_5) \tag{8}$$

$$COP_{\,OC} = \frac{\dot{Q}_{e,DSC}}{\dot{W}_{comp1}+\dot{W}_{comp2}} \tag{9}$$

where $\dot{Q}_{e,DSC}$, $\dot{Q}_{e,SC}$, $\dot{W}_{comp1}$, $\dot{W}_{comp2}$, $\dot{m}_{r,\,DSC}$ and $\dot{m}_{r,\,SC}$ are net refrigerant effect, net compressor work and mass flow rate of dedicated mechanically subcooled and subcooler VCR cycles respectively.

The effectiveness of subcooler ε_{SC} is the ratio of minimum heat transfer to the maximum heat transfer.

$$\varepsilon_{SC} = \frac{T_5\text{-}T_8}{T_3\text{-}T_8} \tag{10}$$

where ε_{SC} is the effectiveness of subcooler and T_3,T_5 and T_8 are temperatures at respective state points.

The energy balance in the subcooler is given by equation (11)

$$\dot{m}_{r,DSC}\,(h_3 - h_{3a}) = \dot{m}_{r,SC}\,(h_5 - h_8) \tag{11}$$

Exergy Analysis

The second law of thermodynamics expresses the concept of exergy. The exergy is defined as the degree measure of quality or usefulness of energy to impact the environment. The maximum useful work which can be extracted from a system as it reversibly comes into equilibrium with its environment. (Bejan et al. (1996), Arora and Kaushik [12]).

For the refrigerant flowing in a refrigerating system, the exergy is defined as follows (Bejan et al. [20]):

$$\dot{E} = \dot{m}_r[(h - h_0) - T_0(s - s_0)] \tag{12}$$

where h_0 and s_0 are the enthalpy and entropy values of the refrigerant at dead state pressure P_0 and temperature T_0.

Exergy destruction($\dot{E}_D$) or internal exergy destruction losses which are caused by irreversibilites of the system is the algebric sum of total exergy at the inlet and outlet of the system. General exergy balance is given by (Dincer and Kanoglu [21]):

$$\dot{E}_D = \dot{E}_{in} - \dot{E}_{out} \tag{13}$$

where $\dot{E}_D$ is the rate of exergy destruction and $\dot{E}_{in}$ and $\dot{E}_{out}$ are the total exergy transferred by heat, work and mass.

Exergy Destruction ($\dot{E}_D$) in the Components of Dedicated Mechanically Subcooled Vapour Compression Refrigeration Cycle (DSC)

The exergy destruction in each component of the dedicated mechanically subcooled vapour compression refrigeration cycle is given by the following equations:

Evaporator 1

$$\left(\dot{E}_D\right)_{e,DSc} = \dot{E}_{X_4} + \dot{Q}_{e,DSC}\left(1 - \frac{T_0}{T_{b,DSC}}\right) - \dot{E}_{X_1} = \dot{m}_{r,DSC}(h_4 - T_0 s_4)$$
$$+\dot{Q}_{e,DSC}\left(1 - \frac{T_0}{T_b}\right) - \dot{m}_{r,DSC}(h_1 - T_0 s_1) \tag{14}$$

Compressor 1

$$\left(\dot{E}_D\right)_{comp1} = \dot{E}_{X_1} + \dot{W}_{comp1} - \dot{E}_{X_2} = \dot{m}_{r,DSC}(\mathrm{h}_1 - T_0 s_1) + \dot{W}_{comp1} - \dot{m}_{r,DSC}(\mathrm{h}_2 - T_0 s_2) \quad (15)$$

Condenser 1

$$\left(\dot{E}_D\right)_{cond1} = \dot{E}_{X_2} - \dot{E}_{X_3} = \dot{m}_{r,DSC}(h_2 - T_0 s_2) - \dot{m}_{r,DSC}(h_3 - T_0 s_3) \quad (16)$$

Expansion Valve 1

$$\left(\dot{E}_D\right)_{Ex.\ Valve1} = \dot{E}_{X_3} - \dot{E}_{X_{3a}} = \dot{m}_{r,DSC}(h_{3a} - T_0 s_{3a}) - \dot{m}_{r,DSC}(h_4 - T_0 s_4) = \dot{m}_{r,DSC} T_0 (s_4 - s_{3a}) \quad (17)$$

Subcooler

$$\left(\dot{E}_D\right)_{SC} = (\dot{E}_{X_8} - \dot{E}_{X_5}) + (\dot{E}_{X_3} - \dot{E}_{X_{3a}}) = \dot{m}_{r,SC}\{(h_8 - h_5) - T_0(s_8 - s_5)\} + \dot{m}_{r,DSC}\{(h_3 - h_{3a}) - T_0(s_3 - s_{3a})\} \quad (18)$$

Compressor 2

$$\left(\dot{E}_D\right)_{comp2} = \dot{E}_{X_5} + \dot{W}_{comp2} - \dot{E}_{X_6} = \dot{m}_{r,SC}(h_5 - T_0 s_5) + \dot{W}_{comp2} - \dot{m}_{r,SC}(h_6 - T_0 s_6) \quad (19)$$

Condenser 2

$$\left(\dot{E}_D\right)_{cond2} = \dot{E}_{X_6} - \dot{E}_{X_7} = \dot{m}_{r,SC}(h_6 - T_0 s_6) - \dot{m}_{r,SC}(h_7 - T_0 s_7) \quad (20)$$

Expansion Valve 2

$$\left(\dot{E}_D\right)_{Ex.\ valve2} = \dot{E}_{X_7} - \dot{E}_{X_8} = \dot{m}_{r,SC}(h_7 - T_0 s_7) - \dot{m}_{r,SC}(h_8 - T_0 s_8) = \dot{m}_{r,SC} T_0 (s_8 - s_7) \quad (21)$$

Evaporator 2

$$\left(\dot{E}_D\right)_{e,Sc} = \dot{E}_{X_8} + \dot{Q}_{e,SC}\left(1 - \frac{T_0}{T_{b,SC}}\right) - \dot{E}_{X_5} = \dot{m}_{r,SC}(h_8 - T_0 s_8) + \dot{Q}_{e,SC}\left(1 - \frac{T_0}{T_{b,SC}}\right) - \dot{m}_{r,SC}(h_5 - T_0 s_5) \quad (22)$$

where $\dot{E}_D$, $\dot{E}_X$, $\dot{m}_r$ and T_b are the exergy destruction rate, exergy rate, mass flow rate and temperature of space to be cooled for the dedicated mechanically subcooled and subcooler cycle respectively. T_0, h_0 and s_0 are the temperature, enthalpy and entropy of dead state.

Total Exergy Destruction

Total exergy destruction of the system is the sum of exergy destruction in each component of the system. The total exergy destruction in dedicated mechanically subcooled vapour compression refrigeration cycle (DSC) is given by:

$$\sum(\dot{E}_D)_{DSC} = (\dot{E}_D)_{e,DSc} + (\dot{E}_D)_{comp1} + (\dot{E}_D)_{cond1} + (\dot{E}_D)_{Ex.\ valve1} \quad (23)$$

$$\sum(\dot{E}_D)_{SC} = (\dot{E}_D)_{e,Sc} + (\dot{E}_D)_{comp2} + (\dot{E}_D)_{cond2} + (\dot{E}_D)_{Ex.\ valve2} \quad (24)$$

$$\sum(\dot{E}_D)_{OC} = (\dot{E}_D)_{e,DSc} + (\dot{E}_D)_{comp1} + (\dot{E}_D)_{cond1} + (\dot{E}_D)_{Ex.\ valve1} + (\dot{E}_D)_{SC} + (\dot{E}_D)_{comp2} + (\dot{E}_D)_{cond2} + (\dot{E}_D)_{Ex.\ valve2} \quad (25)$$

Exergetic Efficiency

The exergetic efficiency is the ratio of total exergy recovered to the total exery supplied and is given by (Dincer and Kanoglu (2010)):

$$\eta_{ex} = \frac{\sum \dot{E}_{out}}{\sum \dot{E}_{in}} = 1 - \frac{\sum \dot{E}_D}{\sum \dot{E}_{in}} \quad (26)$$

where η_{ex} is the exergetic efficiency of the cycle. $\sum \dot{E}_{out}$, $\sum \dot{E}_{in}$ and $\sum \dot{E}_D$ are the total exergy recovered, supplied and destructed respectively.

The exergetic efficiency of vapour compression refrigeration system is defined as the ratio of the exergy of heat absorbed in the evaporator from the space to be cooled at temperature T_b to the actual compressor work input ($\dot{W}_{comp}$) (Arora and Kaushik [12]). Exergy efficiency can also be defined as The exergetic efficiency is the ratio of exery in products to the exergy of fuel. The exergetic efficiency of dedicated mechanically subcooled, subcooler and overall VCR cycles are given by the equations (27)-(29).

$$\eta_{ex\,(DSC)} = \frac{\left|\dot{Q}_{e,DSC}\left(1 - \frac{T_0}{T_{b,DSC}}\right)\right|}{\dot{W}_{comp1}} \quad (27)$$

$$\eta_{ex\,(SC)} = \frac{\left|\dot{Q}_{e,SC}\left(1 - \frac{T_0}{T_{b,SC}}\right)\right|}{\dot{W}_{comp2}} \quad (28)$$

$$\eta_{ex\,(OC)} = \frac{\left|\dot{Q}_{e,OC}\left(1 - \frac{T_0}{T_{b,DSC}}\right)\right|}{\dot{W}_{comp1} + \dot{W}_{comp2}} \quad (29)$$

where T_0 is the ambient or dead state temperature.

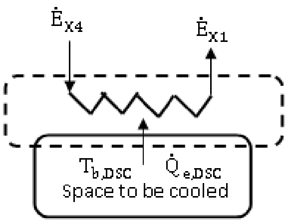


Figure 3. Exergy balance in evaporator of DSC and the space to be cooled [23]

Exergy Destruction Ratio (EDR)

The EDR is the total exergy destruction in the system to the exergy in the products (Arora and Kaushik [12]) and is given by equation (30).

$$EDR = \frac{\dot{E}_{D,total}}{\dot{E}_P} = \frac{1}{\eta_{ex}} - 1 \tag{30}$$

The exergy destruction ratio for the dedicated mechanically subcooled, subcooler and overall VCR cycles are given in following (31)-(33).

$$EDR_{DSC} = \frac{\sum(\dot{E}_D)_{DSC}}{\dot{E}_{P,DSC}} = \frac{\sum(\dot{E}_D)_{DSC}}{\left|\dot{Q}_{e,DSC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|} \tag{31}$$

$$EDR_{SC} = \frac{\sum(\dot{E}_D)_{SC}}{\dot{E}_{P,SC}} = \frac{\sum(\dot{E}_D)_{SC}}{\left|\dot{Q}_{e,SC}\left(1-\frac{T_0}{T_{b,SC}}\right)\right|} \tag{32}$$

$$EDR_{OC} = \frac{\sum(\dot{E}_D)_{OC}}{\dot{E}_{P,DSC}} = \frac{\sum(\dot{E}_D)_{OC}}{\left|\dot{Q}_{e,DSC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|} \tag{33}$$

Mathematical Modelling

The present work investigates the performance of dedicated mechanically subcooled vapour compression refrigeration cycle (DSC) on the bassis of first and second law of thermodynamics. The mathematical fomulation of thermodynamic relations have been carried out considering mass, energy and work conservation principles for system and system components.

Table 1. Input variable considered except where the variation of these system variables involved (Arora and Kaushik [12])

S.No.	Input variables	Values
1.	Evaporator1 temperature ($T_{e,\,DSC}$)	-10^{0}C
2.	Degree of subcooling ($(\Delta T)_{sc}$)(3-3a)	5^{0}C
3.	Condenser1,2 temperature ($T_{c,DSC}$ and $T_{c,SC}$), assuming summer conditions in a country near the equator line.	50^{0}C
4.	Isentropic efficiency of compressor ($\eta_{comp1}, \eta_{comp2}$)	80%
5.	Effectiveness of subcooler (ε_{SC})	0.8
6.	Net refrigerating effect ($\dot{Q}_{e,DSC}$)	3.5167 kW
7.	Ambient or dead state temperature (T_0) and pressure (P_0)	25^{0}C and 101.325 kPa

The following assumptions have been considered to formulate the complex thermodynamic model of dedicated mechanically subcooled vapour compression refrigeration cycle (DSC) except where the variation of physical parameters involved.

- It is assumed that the state of refrigerant is dry and saturated at the entry of compressor at state points 1 and 5.
- Difference between evaporator temperature and the temperature of space to be cooled ($T_{b,\,DSC}$ -$T_{e,\,DSC}$, $T_{b,SC}$ -$T_{e,SC}$) is 5 ^{0}C.
- The range of variation of evaportor1 temperature ($T_{e,\,DSC}$) is -20 ^{0}C to 10 ^{0}C.
- The range of variation of condenser1 temperature ($T_{e,\,SC}$) is 30 ^{0}C to 50 ^{0}C.

- The heat losses and pressure losses from the system and system components are negligible. The whole system operates in steady state condition.
- The range of degree of subcooling $(\Delta T)_{SC}$ is 5 to 30 ^{0}C.
- The pressure drop in evaporator (δ_e) and condenser (δ_c) is assumed to be zero.

RESULTS AND DISCUSSION

Energy and exergy analysis of dedicated mechanically subcooled vapour compression refrigeration cycle (DSC) have been performed using a computer software based program in Engineering Equation Solver (EES) (Klein and Alvarado [22]) which has been shown in Figure 1(a) and 2(b). The considered refrigerants for the analysis are R134a, R1234yf and R1234ze. The computation of the various performance parameters viz. compressor work, COP, exergetic efficiency, total exergy destruction rate and exergy destruction ratio (EDR) have been done by calling builtin functions (i.e. specific entropy, specific enthalpy, temperature and pressure etc.) of the EES library.

The results from the present analysis have been compared with the results of Arora et al. [12]. It is observed that the difference in the value of COP and exergetic efficiency is within a range of ±1%. The difference in results may be due to the pressure drop in evaporator and condenser is assumed to be zero.

Effect of Subcooling

The effect of degree of sub-cooling $(\Delta T)_{sc}$ on various performance parameters viz. compressor work $(\dot{W}_C)$, COP, exergetic efficiency (η_{ex}), total exergy destruction rate $(\dot{E}_{D,\ Total})$ and exergy destruction ratio (EDR) has been illustrated in Figure 4(a) to 4(d).

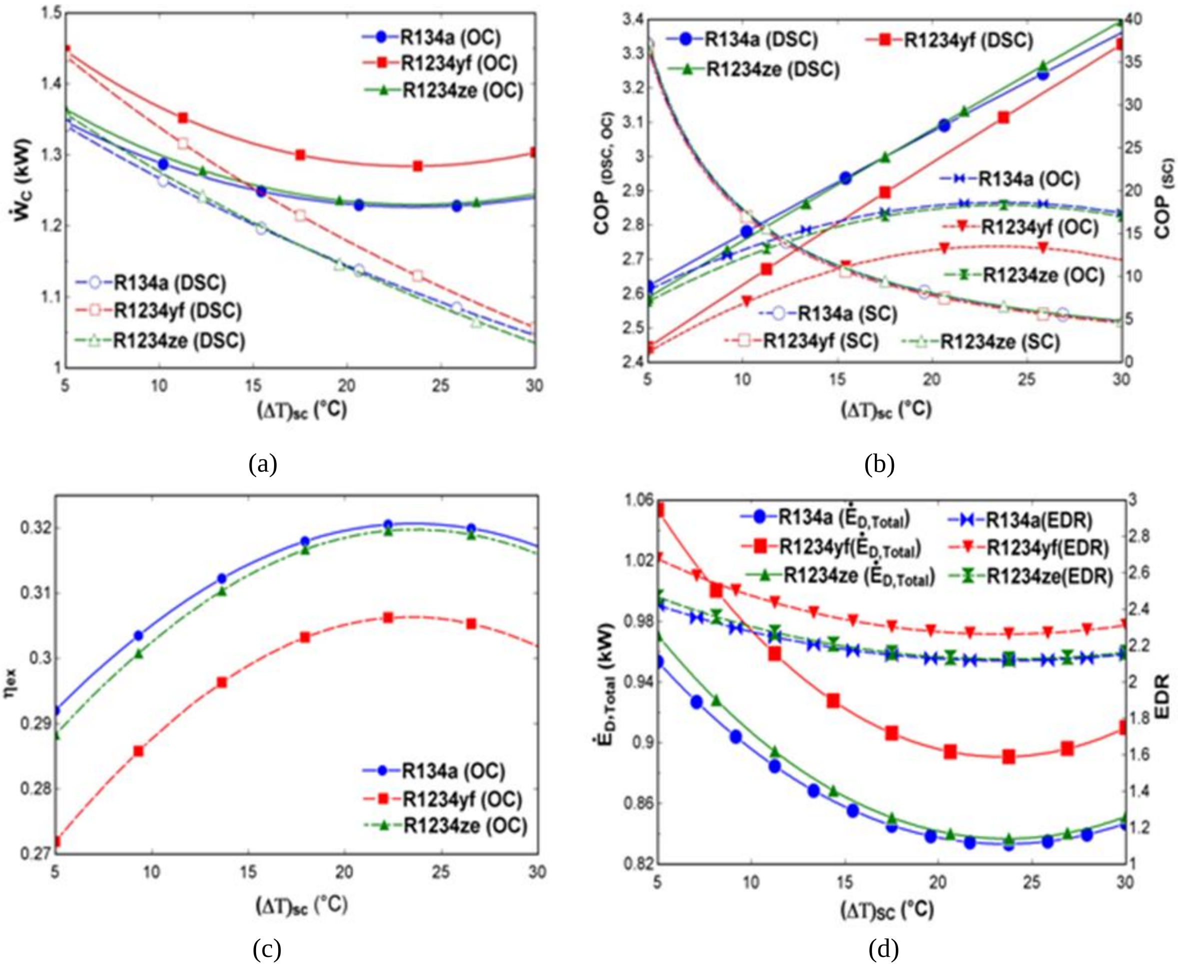


Figure 4. (a) Variation in compressor work $(\dot{W}_C)$ with degree of subcooling (ΔT_{sc}), (b) effect of degree of subcooling (ΔT_{sc}) on COP, (c) Effect of degree of subcooling (ΔT_{sc}) on exergetic efficiency (ηex), (d) variation in total exergy destruction rate (E ̇_(D,Total)) and exergy destruction ratio (EDR) with subcooling (ΔT_{sc}) of OC

Figure 4(a) represents the effect of sub-cooling $(\Delta T)_{sc}$ on compressor work $(\dot{W}_C)$. For 1 Ton refrigeration, compressor work of dedicated mechanically subcooled (DSC) $(\dot{W}_{Comp1})$ decreases with increase in degree of subcooling (DOS) because the mass flow rate of refrigerant decreases with increase in degree of subcooling for constant refrigeration load of 3.5167kW. However, the compressor work first decreases and then increases for overall cycle (OC) with increase in degree of sub-cooling for the considered refrigerants. This is due to that the compressor work of OC $(\dot{W}_{Comp1} + \dot{W}_{Comp2})$ decreases for low values of DOS as the value of $\dot{W}_{Comp1}$ decreases and the values of $\dot{W}_{Comp2}$ are very small. For the higher values of DOS, the compressor work of subcooler cycle $(\dot{W}_{Comp2})$ increases while $\dot{W}_{Comp1}$ decreases. However the compressor work for the simple VCR cycle is is higher than that of the DSC. The optimize value of degree of sub-cooling lies within 20-25^{0}C for overall cycle. It has been observed that the compressor work is minimum for R1234ze and maximum for R1234yf for DSC and OC.

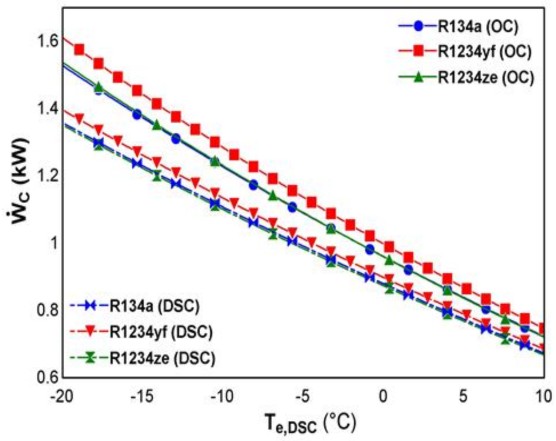
Figure 4(b) shows the variation of COP with degree of sub-cooling for the considered refrigerants R134a, R1234yf and R1234ze. The COP of the dedicated mechanically subcooled cycle increases and the COP of subcooler cycle decreases with increase in DOS for the considered refrigerants. However, the COP of the overall cycle first increases, maximise for optimum value of DOS which lies within 20-25^{0}C and then decreases with increase in DOS. In order to increase DOS, The temperature and pressure of evaporator of subcooler cycle has to achieved lower values which reduces the COP of the subcooler cycle while for the constant refrigerating load of $\dot{Q}_{e,DSC}$ = 3.5167kW, the compressor work $(\dot{W}_{Comp1})$ of DSC decreases linearly and the COP of DSC is $\frac{\dot{Q}_{e,DSC}}{\dot{W}_{comp1}}$ ("Eq.(4)"). Thus the COP of the DSC increases linearly with increase in DOS. The COP of OC is $\frac{\dot{Q}_{e,DSC}}{\dot{W}_{comp1}+\dot{W}_{comp2}}$ ("Eq.(9)") which increases with increase in COP of DSC and decreases with decrease in COP of subcooler cycle due to increase in compressor work of SC with increase in DOS. Thus the COP of OC first increases (as COP of DSC increases faster) reaches to maximum value and then decreases (as COP of SC cycle decreases faster) for the refrigerant considered. It is observed that the COP is maximum for R1234ze and minimum for R1234yf. The COP of simple VCR cycle is lower than the DSC and OC.

Figure 4(c) illustrates the effect of degree of sub-cooling (ΔT_{sc}) on exergetic efficiency $(\eta_{ex\,(OC)})$ of overall cycle. The exergetic efficiency of the overall cycle first increases reaches to a maximum value and then decreases with increase in DOS for the considered refrigerants. The optimum value of degree of sub-cooling lies between 20 to 25^{0}C for which the value of exergetic efficiency is maximum. The exergetic efficiency of overall cycle is given by $\eta_{ex\,(OC)} = \frac{\left|\dot{Q}_{e,OC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|}{\dot{W}_{comp1}+\dot{W}_{comp2}}$ (eq. (29)). In which the term $\frac{|\dot{Q}_{e,OC}|}{\dot{W}_{comp1}+\dot{W}_{comp2}}$ is the COP$_{(OC)}$ which first increases, reaches to a maximum value and then decreases with increase in DOS (Figure 4(b)) and the term $\left|\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|$remains constant with increase in DOS because the value of T_0 and $T_{b,DSC}$ remains constant with increase in DOS. Hence the value of $\eta_{ex\,(OC)}$ first increases, reaches to a maximum value and then decreases. The optimum value of DOS lies between 20 to 25^{0}C for the considered refrigerants. Hence the exergetic efficiency of the OC follows the trend of COP. It can be observed that $\eta_{ex\,(OC)}$ is maximum for R134a and minimum for R1234yf.

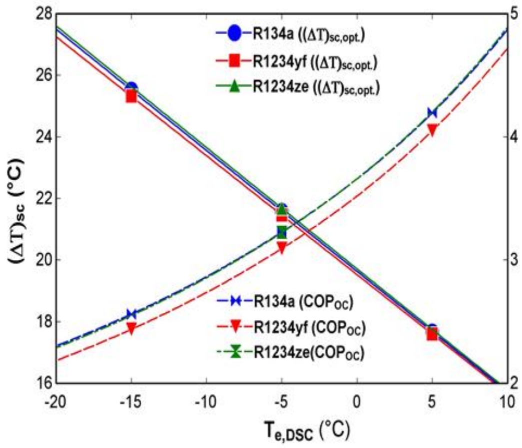
Figure 4(d) shows the variation in total exergy destruction rate $(\dot{E}_{D,\ Total})$ and exergy destruction ratio (EDR) with degree of subcooling (ΔT_{sc}) for overall cycle. The value of $\dot{E}_{D,\ Total}$ and EDR first decreases, reaches to a minimum value and then increases with increase in DOS for the refrigerants considered. The value of $\dot{Q}_{e,OC}$ first increases reaches to a maximum value and then decreases with increase in DOS (Figure 4(b)). Therefore, the exergy recovered from the system and system components first increases, reaches to a maximum value and then decreases with increase in DOS. Consequently, the $\dot{E}_{D,\ Total}$ first decreases, reaches to a minimum value and then increases. EDR is given by $\frac{\Sigma(\dot{E}_D)_{OC}}{\left|\dot{Q}_{e,DSC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|}$ (eq. (33)) which is proportional to $\dot{E}_{D,\ Total}$ and reciprocal to $\left|\dot{Q}_{e,DSC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|$. Hence EDR first decreases, reaches to a minimum value and then increases with increase in DOS (ΔT_{sc}). It can also be observed that the values of $\dot{E}_{D,\ Total}$ and EDR are minimum for R134a and maximum for R1234yf.

Effect of Evaporator Temperature

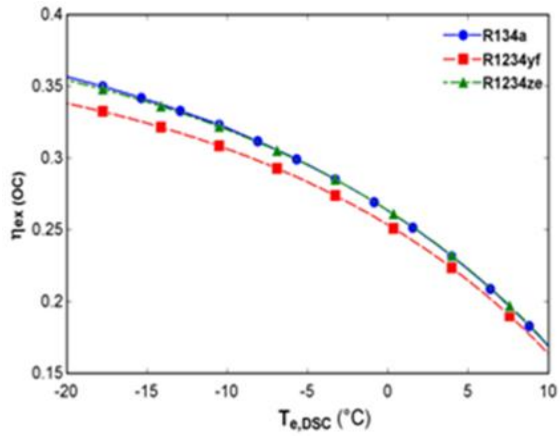
The effect of evaporator1 temperature ($T_{e,\ DSC}$) on various performance parameters viz. compressor work $(\dot{W}_C)$, degree of subcooling $(\Delta T)_{sc}$, COP, exergetic efficiency (η_{ex}), total exergy destruction rate $(\dot{E}_{D,\ Total})$ and exergy destruction ratio (EDR) has been shown in Figure 5(a) to 5(d).



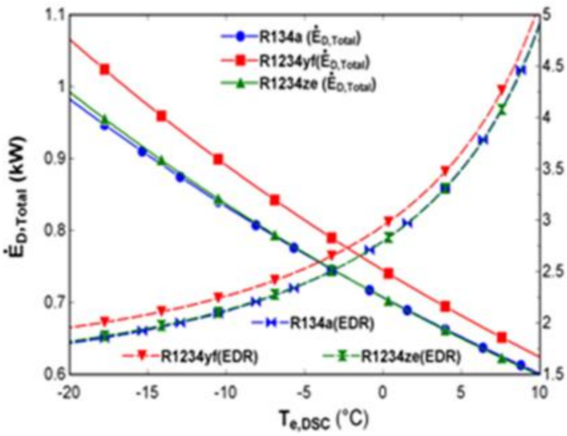
(a)



(b)



(c)



(d)

Figure 5. (a) Variation in compressor work$(\dot{W}_C)$ with evaporator1 Temperature ($T_{e,\ DSC}$), (b) effect of evaporator1 temperature ($T_{e,DSC}$) on degree of subcooling (ΔT_{sc}) and COP, (c) effect of evaporator1 temperature ($T_{e,DCS}$) on exergetic efficiency (η_{ex}) of overall cycle, (d) variation in total exergy destruction rate $(\dot{E}_{D,Total})$ and exergy destruction ratio (EDR) with evaporator1 temperature ($T_{e,DCS}$) of overall cycle

Figure 5(a) shows the varition in compressor work $(\dot{W}_C)$ with evaporator1 temperature ($T_{e,\ DSC}$) for dedicated mechanically subcooled and overall VCR cycle. The compressor work of DSC ($\dot{W}_{Comp1}$) and OC ($\dot{W}_{Comp1} + \dot{W}_{Comp2}$) decreases with increase in evaporator1 temperature ($T_{e,\ DSC}$) for considered refrigerants. As the evaporator1 temperature increases, the pressure ratio between evaporator and condenser decreases and for constant refrigeration load of 3.5167kW, the mass flow rate of refrigerant also decreases. Moreover, the evaporator2 temperature of sub-cooler cycle also increases. Hence the compressor work of DSC and OC decreases with increase in evaporator1 temperature. It is observed that the value of compressor work is minimum for R1234ze and maximum for R1234yf.

Figure 5(b) illustrates the effect of evaporator1 temperature ($T_{e,\ DSC}$) on DOS $(\Delta T)_{sc}$ and COP for OC. The value of DOS decreases and COP increases with increase in evaporator1 temperature for the refrigerant considered. For the constant refrigeration load of 3.5167kW, lower amount of sub-cooling is required in the subcooler to the refrigerant of DSC. Thus the DOS decreases with increase in $T_{e,\ DSC}$. However, the compressor work decreases as shown in Figure 5(a), the value of COP increases. It has been observed that the value of DOS and COP are maximum for R1234ze and minimum for R1234yf.

Figure 5(c) depicts the effect of evaporator1 temperature ($T_{e,\ DSC}$) on exergetic efficiency (η_{ex}) of OC for considered refrigerants. The exergetic efficiency is given by $\mathrm{COP}_{(OC)} \times \left|\left(1 - \frac{T_0}{T_{b,DSC}}\right)\right|$ in which the value of $\mathrm{COP}_{(OC)}$ increases (Figure 5(b)) and the value of term $\left|\left(1 - \frac{T_0}{T_{b,DSC}}\right)\right|$ decreases with increase in evaporator1 temperatue as ($T_{b,\ DSC}$ -$T_{e,\ DSC}$ =5^{0}C). `Hence, the value of $\eta_{ex\ (OC)}$ decreases with increase in evaporator1 temperature. It can be observed that the value of $\eta_{ex\ (OC)}$ is maximum for R134a and minimum for R1234yf.

Figure 5(d) illustrates the variation in total exergy destruction rate $\left(\dot{E}_{D,\ Total}\right)$ and exergy destruction ratio (EDR) with evaporator1 temperature ($T_{e,\ DSC}$) of OC. The decreases with increase in $T_{e,\ DSC}$. It is observed that the value of $\dot{E}_{D,\ Total}$ and EDR are minimum for R1234ze and maximum for R1234yf. $\dot{E}_{D,\ Total}$ increases and EDR decreases with increase in $T_{e,\ DSC}$ for refrigerant considered. As the exergetic efficiency decreases with increase in evaporator1 temperature (Figure 5(c)), the rate of exergy recovered of system component decreases. By which the $\dot{E}_{D,\ Total}$ decreases with increase in $T_{e,\ DSC}$. However, the exergy of product decreases with increase in $T_{e,\ DSC}$ because the term $\left|\left(1 - \frac{T_0}{T_{b,DSC}}\right)\right|$ decreases due to which the EDR of the overall cycle.

Effect of Condenser Temperature

The effect of degree of condenser1,2 temperature ($T_{c,\ DSC} = T_{c,\ SC}$) on various performance parameters viz. compressor work $\left(\dot{W}_C\right)$, COP, exergetic efficiency (η_{ex}), total exergy destruction rate $\left(\dot{E}_{D,\ Total}\right)$ and exergy destruction ratio (EDR) has been depicted in Figure 6(a) to 6(d).

Figure 6(a) illustrates the effect of condenser1,2 temperature ($T_{c,\ DSC} = T_{c,\ SC}$) on compressor work ($\dot{W}_{C,SVCR}, \dot{W}_{Comp1}, \dot{W}_{Comp2}$ and $(\dot{W}_{Comp1} + \dot{W}_{Comp2})$) of simple vapour compression refrigeration cycle (SVCR), DSC, SC and OC for refrigerants considered. The values of compressor work increases with increase in condenser temperature. As the condenser temperature increases, the pressure ratio between evaporator and condenser increases for which more work has been done by the compressor. Therefore, the value of $\dot{W}_C$ increases with increase in T_c. It is also observed that the value of $\dot{W}_C$ is minimum for R1234ze and maximum for R1234yf in DSC, OC and SVCR.

Figure 6(b) shows the effect of condenser1,2 temperature ($T_{c,\ DSC} = T_{c,\ SC}$) on COP of DSC, OC, SC and SVCR for considered refrigerants. The COP of DSC, OC and SVCR decreases with increase in $T_{c,\ DSC} = T_{c,\ SC}$. This is due to that for constant refrigeration load of 3.5167kW, the compressor work increases with increase in condenser temperature. Thus the COP of DSC, OC and SVCR decreases with increase in condenser temperature. However, the COP of sub-cooler VCR cycle (SC) increases with increase in condenser temperature. The rate of heat of heat transfer to evaporator2 increases in the sub-cooler, consequently the refrigeration load of sub-cooler cycle increases and hence COP of sub-cooler cycle increases with increase in condenser temperature. It has also been observed that the COP is maximum for R1234ze and minimum for R1234yf.

Figure 6(c) presents the variation in exergetic efficiency (η_{ex}) with condenser temperature ($T_{c,\ DSC} = T_{c,\ SC}$) of DSC, OC and SVCR cycles for considered refrigerants. The value of η_{ex} decreases with increase in $T_{c,\ DSC} = T_{c,\ SC}$. Exergetic efficiency is given by the relation, $\eta_{ex} = \mathrm{COP} \times \left|\left(1 - \frac{T_0}{T_b}\right)\right|$ in which the term $\left|\left(1 - \frac{T_0}{T_b}\right)\right|$ remains constant and the value of COP decreases with increase in condenser temperature. Hence the value of η_{ex} decreases with increase in $T_{c,\ DSC} = T_{c,\ SC}$. It can also be observed that the value of η_{ex} is maximum for R1234ze and minimum for R1234yf.

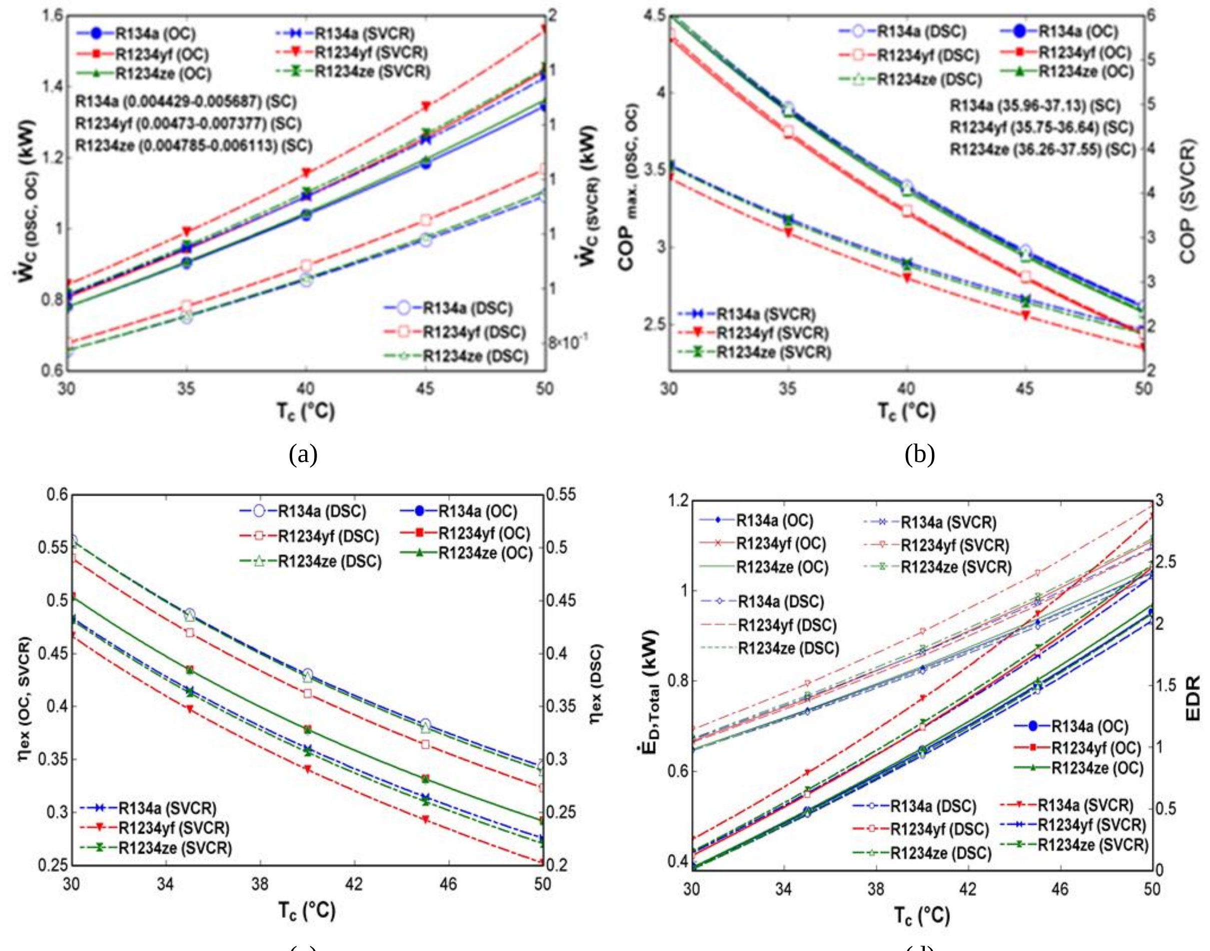


Figure 6. (a) Variation in compressor work $\left(\dot{W}_C\right)$ with condenser temperature (T_c), (b) effect of condenser temperature ($T_{e,DSC}$) on COP, (c) Effect of condenser temperature (T_c) on exergetic efficiency (η_{ex}), (d) Variation in total exergy destruction rate $\left(\dot{E}_{D,Total}\right)$ and exergy destruction ratio (EDR) with condenser temperature (T_c)

Figure 6(d) depicts the effect of condenser temperature ($T_{c,\,DSC} = T_{c,\,SC}$) on total exergy destruction rate $\left(\dot{E}_{D,Total}\right)$ and exergy destruction ratio (EDR) of DSC, OC and SVCR cycles. The value of $\dot{E}_{D,\ Total}$ and EDR increases with increase in $T_{c,\,DSC} = T_{c,\,SC}$. As the condenser temperature increases, the exergy destruction rate in system components increases as well as the exergy of products $\left(\left|\dot{Q}_{e,DSC}\left(1-\frac{T_0}{T_{b,DSC}}\right)\right|\right)$ decreases because the value of COP decreases (Figure 6(b)). Hence the total exergy destruction rate and EDR increase with increase in $T_{c,\,DSC} = T_{c,\,SC}$. It has been seen that the value of $\dot{E}_{D,\ Total}$ and EDR is minimum for R134a and maximum for R1234yf.

CONCLUSIONS

The current work presents the energy and exergy analysis of dedicated mechanically sub-cooled vapour compression refrigeration cycle (DSC). The comparison of DSC, overall cycle (OC), sub-cooler cycle (SC) and simple vapour compression refrigeration cycle (SVCR) has also been considered for R134a, R1234ya and R1234ze using a computer software based program in EES. The theoretical performance analysis involves the variation of DOS, evaporator temperature and condenser temperature to check the performance parameters viz. compressor work, COP, exergetic efficiency, and total exergy destruction rate and exergy destruction ratio. The main concluding points of the current work are as follows:

- The COP of dedicated mechanically sub-cooled VCR cycle (DSC) and overall cycle (OC) is higher than the simple VCR cycle.

- The optimum value of DOS lies between 20^0C to 25^0C for which $\dot{W}_C$, COP and η_{ex} are maximum in OC.
- The COP of OC improves while η_{ex} goes down with increase in evaporator temperature.
- The high condenser temperature reduces the performance (COP andη_{ex}) of the cycle.
- Higher condenser temperature promotes exergy destruction while exergy destruction reduces for lower values of evaporator temperature. The optimum value of DOS lies between 20^0C to 25^0C.
- The performance of R1234ze is better than R1234yf and comparable to R134a.
- Finally, it is inferred that the dedicated mechanically sub-cooled VCR cycle performs better than that of simple VCR cycle and the performance of HFO R1234ze is better than R1234yf.

NOMENCLATURE

COMP	Compressor
Cond	Condenser
COP	Coefficient of performance
DOS	Degree of sub-cooling
DSC	Dedicated mechanically sub-cooled VCR cycle
$(\Delta T)_{sc}$	Degree of subcooling
$\dot{E}_D$	Exergy destruction rate (kW)
ED	Exergy destruction
EDR	Exergy destruction ratio
EES	Energy equation solver
$\dot{E}$	Rate of exergy (kW)
E.V.	Expansion valve
GWP	Global warming potential
h	Specific enthalpy (kJ/kg)
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
$\dot{m}_r$	Mass flow rate of refrigerant (kg/s)
ODP	Ozone depleting potential
P	Pressure (kappa)
SVCR	Simple vapour compression refrigeration
$\dot{Q}_e$	Rate of net refrigerating effect (kW)
s	Specific entropy (kJ/kg^{0}C)
T	Temperature (^{0}C)
T_b	Boundary temperature (^{0}C)
T_e	Evaporator temperature (^{0}C)
T_0	Dead state temperature (^{0}C)
VCR	Vapour compression refrigeration
$\dot{W}$	Work rate (kW)
ε	Effectiveness
η	Efficiency
Σ	represents summation
0	Dead state
C	Condenser
Comp	Compressor
Cond	Condenser
e	Evaporator
ex	Exergetic
SC	Su-cooling cycle
OC	Overall cycle
i	Input
o	Output

REFERENCES

[1] Parliament, E., and CotE, U. (2014). Regulation (EU) No 517/2014 of The European Parliament and of the council of 16 April 2014 on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006. Official J. Eur. Union, 57, 195-230.

[2] Zubair, S. M. (1990). Improvement of refrigeration/air-conditioning performance with mechanical sub-cooling. Energy, 15(5), 427-433.
[3] Zubair, S. M. (1994). Thermodynamics of a vapor-compression refrigeration cycle with mechanical subcooling. Energy, 19(6), 707-715.
[4] Zubair, S. M., Yaqub, M., and Khan, S. H. (1996). Second-law-based thermodynamic analysis of two-stage and mechanical-subcooling refrigeration cycles. International Journal of Refrigeration, 19(8), 506-516.
[5] Couvillion, R. J. (1988). Analysis of a Vapour-compression Refrigeration System with mechanical subcooling. ASHRAE Trans., 94(2), 641-660.
[6] Qureshi, B. A., Inam, M., Antar, M. A., and Zubair, S. M. (2013). Experimental energetic analysis of a vapor compression refrigeration system with dedicated mechanical sub-cooling. Applied Energy, 102, 1035-1041.
[7] Zubair, S. M. (2001). Thermodynamic optimization of finite time vapor compression refrigeration systems. Energy conversion and management, 42(12), 1457-1475.
[8] Thornton, J. W., Klein, S. A., and Mitchell, J. W. (1994). Dedicated mechanical subcooling design strategies for supermarket applications. International Journal of Refrigeration, 17(8), 508-515.
[9] Arora, A., Arora, B. B., Pathak, B. D., and Sachdev, H. L. (2007). Exergy analysis of a vapour compression refrigeration system with R-22, R-407C and R-410A. International journal of Exergy, 4(4), 441.
[10] Arora, A., Dixit, M., and Kaushik, S. C. (2016). Computation of optimum parameters of a half effect water lithium bromide vapour absorption refrigeration system. Journal of Thermal Engineering, 2(2), 683-692.
[11] Arora, A., Dixit, M., and Kaushik, S. C. (2016). Energy and exergy analysis of a double effect parallel flow LiBr/H2O absorption refrigeration system. Journal of Thermal Engineering, 2(1), 541-549.
[12] Arora, A., and Kaushik, S. C. (2008). Theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A. International journal of refrigeration, 31(6), 998-1005.
[13] Arora, A., Arora, B. B., Pathak, B. D., and Sachdev, H. L. (2007). Exergy analysis of a vapour compression refrigeration system with R-22, R-407C and R-410A. International journal of Exergy, 4(4), 441.
[14] Llopis, R., Cabello, R., Sánchez, D., and Torrella, E. (2015). Energy improvements of CO2 transcritical refrigeration cycles using dedicated mechanical subcooling. International Journal of Refrigeration, 55, 129-141.
[15] Llopis, R., Nebot-Andrés, L., Sánchez, D., Catalán-Gil, J., and Cabello, R. (2018). Subcooling methods for CO 2 refrigeration cycles. A Review. International Journal of Refrigeration.
[16] Mota-Babiloni, A., Navarro-Esbrí, J., Barragán-Cervera, Á., Molés, F., Peris, B., and Verdú, G. (2015). Commercial refrigeration–an overview of current status. International journal of refrigeration, 57, 186-196.
[17] Kalla, S. K., Arora, B. B., and Usmani, J. A. (2018). Alternative refrigerants for HCFC 22 - A review. Journal of Thermal Engineering, 4(3), 1998-2017.
[18] Kalla, S. K., Arora, B. B., and Usmani, J. A. (2018). Performance analysis of R22 and its substitutes in air conditioners. Journal of Thermal Engineering, 4(1), 1724-1736.
[19] Yataganbaba, A., Kilicarslan, A., and Kurtbaş, İ. (2015). Exergy analysis of R1234yf and R1234ze as R134a replacements in a two evaporator vapour compression refrigeration system. International journal of refrigeration, 60, 26-37.
[20] Bejan, A., Tsatsaronis, G., Moran, M., and Moran, M. J. (1996). Thermal design and optimization. John Wiley and Sons.
[21] Dincer, I., and Kanoglu, M. (2010). Refrigeration systems and applications (Vol. 2). New York: Wiley.
[22] Klein, S. A., and Alvarado, F. (2012) Engineering Equation Solver. F-Chart Software, Middleton, WI. Version 9, 224-3D.
[23] Kotas, T. J. (1995) The exergy method of thermal plant analysis, Reprint ed., Krieger Publishing Company, Krieger Drive, Malabar, Florida-32950.
[24] Dixit, M., Kaushik, S. C., and Arora, A. (2017) Energy and exergy analysis of absoption-compression cascade refrigeration system. Journal of Thermal Engineering, 3(5), 1466-1477.
[25] Dixit, M., Arora, A., and Kaushik, S. C. (2016) Energy and exergy analysis of a waste heat driven cycle for triple effect refrigeration. Journal of Thermal Engineering, 2(5), 954-961.
[26] Dixit, M., Kaushik, S. C., and Arora, A. (2017) Energy and exergy analysis of absoption-compression cascade refrigeration system. Journal of Thermal Engineering, 3(5), 1466-1477.