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A comprehensive parametric and structural bond analysis of an actual vapor compression refrigeration system with dedicated mechanical subcooled system

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Abstract This study compares the performance of dedicated mechanical subcooled vapor compression refrigeration (DMS-VCR) system and actual vapor compression refrigeration (VCR) system of same capacity (100 kW), and evaluates them on the basis of energy, exergy, and coefficient of structural bond (CSB) configuration. Results indicate that DMS-VCR system outperforms VCR system at constant condenser temperature (T_{cond} = 40° C) with varying evaporator temperatures (T_{evap} = 0 °C, 5 °C and 10 °C).The most significant results occurs at 0 °C, resulting in a coefficient of performance (COP) 4.60 % higher than actual VCR system's COP and exergetic efficiency 4.38 % higher than actual VCR system's exergetic efficiency, making it a more favourable choice for water chilling applications. Additionally, the study investigates the potential of the CSB method in improving system efficiency by reducing irreversibility rate in a specific component. It finds that improving the efficiency of a component can significantly decrease the total irreversibility rate of the system. The CSB value of condenser-1 and evaporator are observerd to be highest (i.e. 1.96 and 2.27) and hence the performance of DMS-VCR system can be greatly improved by changing the efficiency parameter of evaporator and condenser-1 of the DMS-VCR system.

1. Introduction

The vapor compression refrigeration (VCR) system is widely used technology for heating and cooling purposes in residential, commercial and industrial applications. However, due to the high energy consumption of these systems and rising costs of energy, energy efficiency and its conservation have become a growing concern. The researchers are therefore exploring methods to enhance the performance of these systems so as to address the above issues.

One of the methods to enhance the performance of a VCR system is to subcool the liquid refrigerant leaving the condenser. This can be achieved by incorporating a dedicated mechanical subcooled (DMS) system (which is basically a VCR system only) used in conjunction with the existing VCR system. The sole purpose of DMS system is to subcool the liquid refrigerant leaving the condenser-1 of the VCR system used for cooling or heating application. A number of studies have been conducted on the topic of mechanical and dedicated subcooling of VCR systems as discussed in following paras.

Qureshi and Zubair [1, 2] investigated the mechanical subcooling of VCR systems and explored the most efficient methods of operation and management. They analysed the system at 0 °C of evaporator temperature and 45 °C of condenser temperature. Qureshi et al. [3] conducted experiments to demonstrate the effectiveness of the dedicated subcooling cycle concept and investigated how the efficacy of the DMS-VCR system is affected by considering different combinations of various refrigerants. They also reported that subcooling R22 refrigerant by 5 °C-8 °C in the VCR cycle led to a 0.5 kW increase in the load-bearing capacity of the evaporator. According to Agaro et. al. [4], DMS-VCR system is the most thoroughly studied system and is effectively used to improve the efficiency of the transcritical VCR system. Yang and Yeh [5] examined VCR system that uses various refrigerants (R22, R134a, R410A and R717) to identify ways to enhance their performance and minimize exergy destruction. Wang and Zhang [6] optimised and compared a single-stage transcritical CO_2 VCR cycle with different subcooling methods for district heating and cooling and investigated it on the basis of energy, exergy and economic view point. Agarwal et al. [7] examined the mechanical subcooled VCR system's energy and exergy aspects in order to improve its efficiency. However, they did not take into account the coefficient of structural bonds (CSB) factors, which are crucial in making accurate decisions regarding energy resource utilization. Boer et al. [8] conducted an energy, exergy, and structural analysis of a single effect vapor absorption cooling cycle that utilized ammonia and water. Through exergy analysis, they were able to assess the level of irreversibility within each component of the cycle as well as the overall system. The CSB analysis is also performed to access the impact that how any change in irreversibility within one component of the system would affect the rest of the components and the system as a whole. To assess the actual thermodynamic efficiency of chemical processes and the distribution of irreversibility in a plant, Tekin and Bayramoglu [9] carried out an exergy and CSB analysis. They reported the decrease in total exergy loss is equal to 4.21 % in sugar manufacturing plants. Nikolaidis and Probert [10] analyzed a refrigeration plant that uses a two-stage VCR system and demonstrated that the exergy method is effective for studying its behavior. Misra et al. [11] analyzed an absorption chiller system that is used for air-conditioning purposes and used the CSB technique to assess the economic cost of the product. Their study revealed that the capital cost of the optimal system configuration increased by approximately 33.3 % compared to the base-case, but this additional cost is offset by the reduced fuel cost, making it a cost-effective option in the long run. Solanki et al. [12] theoretically examined the performance of simple vapor compression refrigeration system and DMS-VCR system by CSB analysis and advanced exergy analysis methodologies. Their results reveal that the DMS system can avoid 42.9 % of the overall irreversibility rate using advanced exergy analysis, but they have not considered actual VCR cycle while conducting the analysis. They showed better CSB results for condenser-1, but the current study uncovers a contrasting finding. These outcome differences highlight the unique and distinct research approaches in both studies.

It is thus observed that sustainable and efficient VCR systems should be developed in light of increasing energy costs and energy conservation. The DMS-VCR system is capable of enhancing the energetic and exergetic performance of the system as is evident from the investigations performed by various researchers. Studies have been carried out to evaluate efficacy of DMS-VCR systems. Further, the CSB approach has been observed as a useful tool for assessing the impact of change in irreversibility of a component on the other components and the system. It can thus help the designers to wisely make decisions for changing the design of the component which affects the performance of other components critically.

Previous studies have attempted to improve the performance of mechanical subcooled systems through energy and exergy analyses, but they have overlooked the importance of analyzing the CSB in a system. Most studies on DMS-VCR systems have focused on introductory energy and exergy analyses using typical refrigerants for low cooling capacity systems. In addition to this, the performance analysis studies of DMS-VCR system using specific refrigerants such as R134a for high cooling capacity are observed to be scant in literature. To operate a system efficiently and to reduce energy consumption, it is essential to conduct a CSB analysis so as to find appropriate thermodynamic parameters and improve the design of the component so as to reduce the irreversibility rate. However, the CSB studies of DMS-VCR system are also lacking in literature. In the present study the DMS system is incorporated in a commercial water chiller of 100 kW capacity (which is based on actual vapor compression cycle) and comprehensive energy, exergy and CSB analysis of the combined DMS-VCR system is carried out under different operating conditions. In contrast to Solanki et al. [12], the novelty of current work is that it takes a more practical approach, analyzing an actual DMS-VCR system, providing a comprehensive and applicable perspective of refrigeration systems. The system simulation is carried out for a wide range of various operating parameters using the code developed for the DMS-VCR system using engineering equation solver software (EES) [13]. This research endeavor aims to establish a novel model that stands apart from Solanki et al. [12]. It achieves this by incorporating pivotal factors critical for system analysis, including subcooling within the condenser and dedicated subcooler, evaluation of pressure drops within the evaporator, suction line, condenser and liquid line, examination of superheating within the evaporator and suction line, and a thorough assessment of desuperheating within the discharge line. It's important to note that the effect of these crucial parameters was not analysed in the previous work of Solanki et al. [12]. This unique approach aims to determine the optimal subcooling degree for maximizing coefficient of performance (COP) and exergetic efficiency. The model also investigates the impact of subcooling on the overall irreversibility rate, providing a comprehensive understanding of its impact on system performance. The inclusion of varying evaporator temperatures (0 °C, 5 °C, and 10 °C) in our study significantly enhances its value. By exploring different evaporator temperature scenarios, our research provides a valuable and comprehensive perspective, allowing us to gain deeper insights into the system's behavior under diverse thermal conditions. This research contributes significant insights into enhancing system performance, making it an essential advancement in the practical field.

2. System description and modeling of DMS-VCR system

2.1 Description of the dedicated subcooled refrigeration system

Fig. 1 shows the schematic diagram of the DMS-VCR system which comprises of two compressors, two condensers, two expansion valves, one evaporator and one sub-cooler. The VCR sub system, of DMS-VCR system, is used for cooling or heating for which DMS sub system is used to subcool the liquid refrigerant leaving the condenser-1 of VCR sub system. The heat exchange between the two sub systems occurs by using the evaporator of DMS sub system as a subcooler heat exchanger in which the liquid refrigerant leaving the condenser-1 of VCR sub system loses heat and the same is utilized to evaporate the refrigerant entering in the evaporator (used as sub-cooler) of VCR sub system.

Fig. 2 displays the pressure-enthalpy (P-h) diagram for DMS-VCR cycle. The P-h diagram is drawn considering that the refrigerant in both cycles is same. Hence the condenser pressure and temperature will be same in both VCR sub system and DMS sub system. However, the evaporator temperature of VCR cycle is lower than that of DMS cycle.

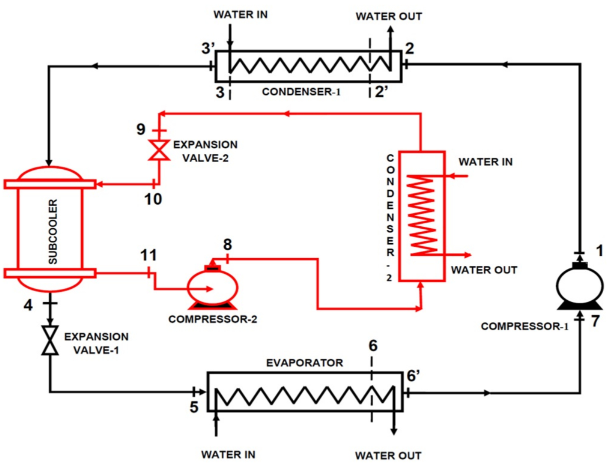


Fig. 1. Schematic diagram of DMS-VCR system.

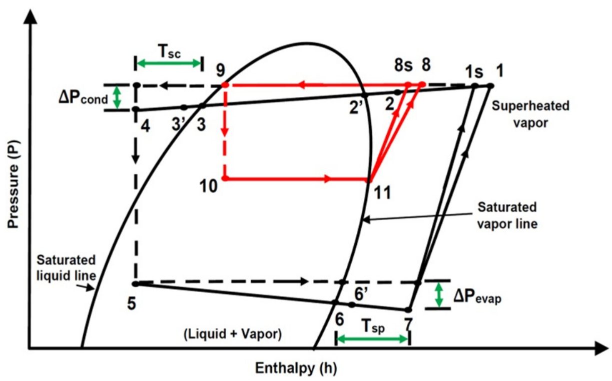


Fig. 2. Pressure - enthalpy diagram of DMS-VCR cycle.

The DMS cycle is shown by state points 8-9-10-11-8 and VCR cycle is depicted by state points 1-2-2'-3-3'-4-5-6-6'-7-1. In VCR cycle, a high-temperature, high-pressure superheated refrigerant vapor at state 1 leaves the compressor and loses heat to surroundings in the desuperheating process 1-2 in the discharge line. Further, desuperheating of the superheated refrigerant occurs in the condenser-1 during the process 2-2' followed by conversion of saturated refrigerant vapour into saturated liquid refrigerant during the process 2'-3 in the condenser due to heat transfer to the surroundings. The process 3-3' shows the subcooling of the refrigerant in the condenser and remaining subcooling takes place in the dedicated subcooler in the process 3'-4. The subcooler section is essentially evaporator of the DMS sub system which operates at a temperature below the condenser outlet temperature (T3'). The corresponding process in the evaporator of DMS sub system is process 10-11 during which the refrigerant in DMS evaporator becomes saturated vapor at state point 11. The refrigerant which leaves the subcooler at state point 4 is expanded to state 5 in the expansion valve-1 to evaporator pressure and temperature of the VCR sub system. The desired cooling goal of the DMS-VCR system is achieved by absorbing heat from the space that needs to be chilled during the process 5-6' in the evaporator. The refrigerant at state point 6 (within the evaporator) is saturated and at state point 6' the refrigerant leaves the evaporator in superheated state. Further superheating takes place in the suction line till state point 7 is achieved just before entry to compressor. The superheated refrigerant is then compressed in the compressor-1 (process 7-1). The VCR sub system cycle includes both subcooling in condenser-1, subcooler, superheating in evaporator and suction line and desuperheating in discharge line. Simultaneously pressure drops in condenser and evaporator are also considered as depicted in Fig. 2 i.e. P-h diagram representing various processes of both the cycles taking place in VCR sub system and DMS sub system of DMS-VCR system. Thus the present work is focused on the analysis of an actual VCR sub system subcooled by DMS sub system. The DMS sub system cycle is assumed to be ideal except compression process which is not assumed reversible adiabatic. Thus the considered DMS-VCR system reasonably follows a real system.

3. Mathematical modeling of system

The current system model in the EES software is created while taking into consideration the thermophysical properties of the refrigerant R134a. Table 1 shows the governing equations for the DMS-VCR system's parts.

The following assumptions apply to the system's thermodynamic modeling:

Table 1. Governing equations for various components of the DMS-VCR system.

S. No.	Component	Energy analysis equations	Exergy analysis equations
1	Evaporator	$\dot{Q}_{evap} = (\in C)_{evap}(T_{in,evap} - T_{evap})$ $= (\dot{m}_{ef}c_p)_{evap}(T_{in,evap} - T_{out,evap}) = \dot{m}_1(h_{6'} - h_5)$ $T_{evap} = T_5$ $\dot{Q}_{gain,s1} = \dot{m}_1(h_7 - h_{6'})$	$\dot{I}_{evap} = T_0(\dot{m}_1(S_{6'} - S_5) + (\dot{m}_{ef}\cdot c_p)_{evap}\cdot \ln((T_{out,evap}+273)/(T_{in,evap}+273)))$
2	Condenser-1	$\dot{Q}_{cond1} = (\in C)_{cond1}(T_{cond1} - T_{in,cond1}) = \dot{m}_1(h_2 - h_{3'})$ $= (\dot{m}_{ef}c_p)_{cond1}(T_{out,cond1} - T_{in,cond1})$ $T_{cond1} = T_3$ $\dot{Q}_{loss,dl} = \dot{m}_1(h_2 - h_1)$	$\dot{I}_{cond1} = T_0(\dot{m}_1(S_3 - S_2) + (\dot{m}_{ef}\cdot c_p)_{cond1}\cdot \ln((T_{out,cond1}+273)/(T_{in,cond1}+273)))$
3	Condenser-2	$\dot{Q}_{cond2} = (\in C)_{cond2}(T_{cond2} - T_{in,cond2}) = \dot{m}_2(h_8 - h_9)$ $= (\dot{m}_{ef}c_p)_{cond2}(T_{out,cond2} - T_{in,cond2})$ $T_{cond2} = T_9$	$\dot{I}_{cond2} = T_0(\dot{m}_2(S_9 - S_8) + (\dot{m}_{ef}\cdot c_p)_{cond2}\cdot \ln((T_{out,cond2}+273)/(T_{in,cond2}+273)))$
4	Compressor-1 [15, 16]	$\dot{W}_{comp1} = (\dot{W}_{sc,isen})/\eta_{isen1}$ $\dot{W}_{sc,isen} = \dot{m}_1(h_1 - h_7)$ $\eta_{isen1} = 0.85\text{-}0.046667*(\frac{P_{cond1}}{P_{evap}})$	$\dot{I}_{comp1} = T_0(\dot{m}_1(S_1 - S_7))$
5	Compressor-2 [15, 16]	$\dot{W}_{comp2} = (\dot{W}_{sc,isen})/\eta_{isen2}$ $\dot{W}_{sc,isen} = \dot{m}_2(h_8 - h_{11})$ $\eta_{isen2} = 0.85\text{-}0.046667*(\frac{P_{cond2}}{P_{sc}})$	$\dot{I}_{comp2} = T_0(\dot{m}_2(S_8 - S_{11}))$
6	Subcooler	$\dot{Q}_{SC} = \in_{sc}\cdot \dot{Q}_{SC,max}$ $\dot{m}_2(h_{11} - h_{10}) = \dot{m}_1(h_{3'} - h_4)$ $T_{11} = T_4 - \Delta T_{overlap}$	$\dot{I}_{sc} = T_0(\dot{m}_2(S_{11} - S_{10}) + \dot{m}_1(S_4 - S_{3'}))$
7	Expansion valve-1	$h_4 = h_5$	$\dot{I}_{ev1} = T_0(\dot{m}_1(S_5 - S_4))$
8	Expansion valve-2	$h_9 = h_{10}$	$\dot{I}_{ev2} = T_0(\dot{m}_2(S_{10} - S_9))$

1) The components of the system are operating at a steady-state condition.
2) The subcooling is assumed to take place in condenser-1 and dedicated subcooler.
3) The superheating takes place in evaporator as well as in suction line.
4) The desuperheating takes place in discharge line.
5) The compression processes in compressors are assumed to be irreversible adiabatic.
6) Water is used as heat exchange fluid in evaporator and condensers.

The components of the DMS-VCR system are analysed by using principles of conservation of mass and energy, and entropy generation [14] and Table 1 displays the governing equations for the DMS-VCR System.

$$\Sigma\dot{m}_{in} - \Sigma\dot{m}_{out} = 0 \quad (1)$$

(Principle of conservation of mass)

$$\Sigma\dot{Q} - \Sigma\dot{W} = \Sigma\dot{m}_{out}h_{out} - \Sigma\dot{m}_{in}h_{in} \quad (2)$$

(Principle of conservation of energy)

$$\dot{S}_{gen} = \Sigma\dot{m}_{out}s_{out} - \Sigma\dot{m}_{in}s_{in} - \Sigma\frac{\dot{Q}}{T} \geq 0\ . \quad (3)$$

(Principle of increase of entropy)

The following Eqs. (4) and (5) indicate the overall COP of actual VCR system and the DMS-VCR system:

$$COP_{VCR} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp1}} \quad (4)$$

$$COP_{DMS-VCR} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp1} + \dot{W}_{comp2}}\ . \quad (5)$$

A key point for evaluating the cooling impact and energy intake needed in a system is the coefficient of performance (COP). It does not, however, offer information regarding the efficiency of the energy conversion method or point out the elements that are responsible for irreversibility in a process occurring in a particular component and system. In order to assess the exergetic efficiency of the DMS-VCR system and identify the factors that lead to inefficiency, exergetic analysis is crucial. Exergy analysis can be used to assess the system's exergetic efficiency and encourage energy saving to overcome above restriction. It also pinpoints system's component responsible for irreversibility. The equations are provided in Table 1 for calculating the irreversibility in various components of DMS-VCR system. It is calculated by using the Gouy-Stodola theorem as shown in the Eq. (6).

$$\dot{I} = T_0 \dot{S}_{gen} \; . \tag{6}$$

Where, T_o is the ambient temperature.

The exergetic efficiency (η_{ex}) of DMS-VCR system is given by [17]:

For DMS-VCR system

$$\dot{I}_t = \dot{I}_{evap} + \dot{I}_{cond1} + \dot{I}_{cond2} + \dot{I}_{comp1} + \dot{I}_{comp2} + \dot{I}_{sc} + \dot{I}_{ev1} + \dot{I}_{ev2} \tag{7}$$

$$\eta_{ex} = \left(\dot{Q}_{evap} \left| 1 - \frac{T_0}{T_e} \right| \right) / \dot{W}_t \; . \tag{8}$$

where, $\dot{W}_t = \dot{W}_{comp1} + \dot{W}_{comp2}$.

The CSB approach has now been used after determining the irreversibility rate of each system component [11, 18].

$$CSB_n = \left| \frac{\delta \dot{I}_t}{\delta z_i} \right| / \left| \frac{\delta \dot{I}_n}{\delta z_i} \right| \tag{9}$$

$$\text{DMS}, \forall \in \text{EoS} \cap \{\text{evap, cond1, cond2, comp1, comp2, subcooler}\}$$

$$\forall z_i \in \text{system parameter}$$

According to the CSB principle, lowering the irreversibility rate of one component gradually (by changing the system efficiency parameter Z_i) lowers the irreversibility rate of the entire system. Z_i (the system efficiency parameter) in this context may be associated with the system temperature (i.e., evaporator or condenser temperature) or the degree of subcooling or the degree of overlap or the isentropic efficiency of the compressor.

Following are the effects of various CSB values:

In case-1, when $CSB_n > 1$, the system input decreases faster than the irreversibility rate of the n^{th} element. The n^{th} element has a favorable effect on the system's exergetic efficiency, so strengthening it, will increase the system's exergetic efficiency.

In case-2, when $CSB_n < 1$, the system input decreases slower than the n^{th} element's irreversibility rate. In this instance, lowering the irreversibility rate of the n^{th} element has a detrimental effect on the operation of other system components, which in turn has an adverse effect on the performance of the entire system.

In case-3, when $CSB_n = 1$, the performance of the entire system remains unchanged. This is because any enhancement in the n^{th} element's performance is balanced by a decrease in the performance of other components, resulting in a rigid system structure that offers no scope for overall performance enhancement.

Therefore, it is important to consider the value of CSB_n when designing the DMS-VCR system to ensure the overall performance enhancement of the system.

4. Model validation

Model validation of Solanki et al. [12] primarily involved theoretical data, which lacked real-world applicability. However, in the current study, we take a more comprehensive approach by employing a dual validation strategy. This approach combines the comparison of theoretical data of present work with experimental data, significantly enhancing the realism of our analysis. Consequently, our findings are aligned with both theoretical expectations and real-world observations, resulting in a more robust and reliable investigation. The current model is utilized for computation of results corresponding to the input data ($\dot{Q}_{evap}$ = 3.5167 kW, T_{evap}, = -10 °C, DOS = 5° C, T_{cond1}, T_{cond2} = 50 °C C, η_{comp1}, η_{comp2} = 80 %, $\in_{sc}$ = 0.8, T_o = 25 °C, P_o = 101.325 kPa) of Agarwal et al. [19]. The results obtained from the current programme are compared with the results of Agarwal et al. [19] and it is observed that the results obtained from the present model lie within 3.45 % of Agarwal et al. [19], but most of the errors are less than 2 % (see Fig. 3).

The numerical simulation data is also compared with the experimental data from Han et al. [20] in order to assess the accuracy of the existing model and to determine the effect of evaporation temperature on COP for R410A. Fig. 4 compares

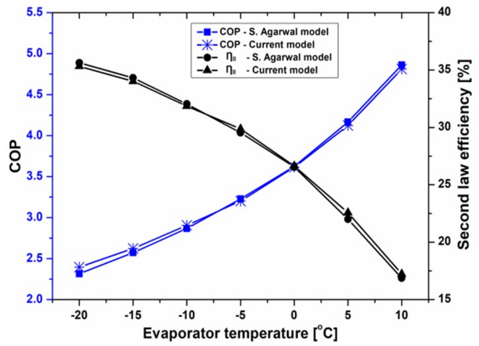


Fig. 3. Performance comparison for effect of evaporator temperature with refrigerant R134a (T_{cond1} = 50 °C).

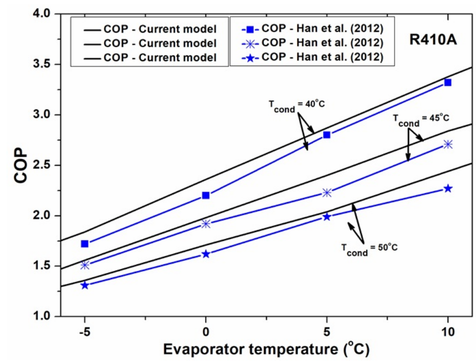


Fig. 4. COP comparison of current model with the experimental data of Han et al. [20] under different condenser temperature.

the coefficient of performance (COP) utilizing fixed superheating (T_{sup} = 8.3 °C) and subcooling (T_{sc} = 11.1 °C) degrees at varied condenser temperatures. The COP value provided by this study is marginally higher than that provided by Han et al. [20] experimental calculations. The experimental results and the theoretical results obtained from the present model corresponding to the data of Han et al. [20] are consistent and they lie between +1.5 % to +5 % of the experimental data reported by Han et al. [20] for coefficient of performance (COP). This level of consistency and accuracy in COP reaffirms the robustness and reliability of our model.

5. Results and discussion

This section presents the results and their discussion based on the process data presented in Table 2 of the DMS-VCR system for water cooling application.

A comparison of the DMS-VCR system with actual VCR system is carried out using the process data values listed in Table 2. As shown in Table 3, the DMS-VCR system's thermodynamic parameters (pressure, temperature, mass flow rate, entropy, and enthalpy) are calculated at the entry and exit of each component. The values of the performance parameters for the DMS-VCR system and actual VCR system are shown in Tables 4 and 5, respectively. For the same cooling capacity of 100 kW at T_{evap} = 5 °C, refrigerant (R134a) has mass flow rates of 0.59 kg/s (VCR sub cycle), 0.08 kg/s (DMS sub cycle), and 0.66 kg/s (actual VCR system). Additionally, the mass flow rates for external fluid (i. e. cooling water) for condensers-1, 2, and evaporator of the DMS-VCR system are 5.33 kg/s, 0.69 kg/s, and 4.77 kg/s, respectively, as compared with 5.84 kg/s in condenser-1 and 4.77 kg/s in evaporator of the actual VCR system. The DMS-VCR system performs better than an actual VCR system.

5.1 Energy analysis results

The results of the energy analysis comparing the use of the refrigerant HFC134a in a DMS-VCR system versus an actual VCR system are shown in Table 4. The primary goal of analysis is to demonstrate how the DMS-VCR system concept improved the COP and decreased compressor power. According to the results in Table 4, with variable evaporator temperatures (T_{evap} = 0 °C, 5 °C, and 10 °C) and constant condenser temperature of 40 °C, the DMS-VCR system reduces compressor-1 power usages by 8.65 %, 8.49 % and 8.31 %, respectively, while maintaining the same cooling capacity. This decrease is also helpful in reducing heat load on the condenser-1 by 8.75 %, 8.68 %, and 8.53 %, respectively. As a result, the COP of the DMS-VCR system increases by 4.60 %, 3.18 % and 1.75 %, respectively. Further, heat exchangers play crucial role in ensuring the overall performance and energy efficiency of the system by effectively transporting heat. It is essential to understand how well the evaporator, condensers, and subcooler work in order to maximize the DMS-VCR system's thermal performance and determine whether it is appropriate for use in real-world situations. The calculated effectiveness values for the evaporator, condenser, and subcooler are 0.80, 0.83, and 0.75, respectively.

5.2 Exergy analysis results

Table 5 displays the outcomes of the exergy analysis performed on the actual VCR system and the DMS-VCR system. The findings show that due to decrease in compressor work, cooling load, and refrigerant mass flow rate, the irreversibility

Table 2. Process data of DMS - VCR system.

	Parameters	Values (range)
Thermal parameters [21]	Evaporator cooling capacity (Q_{evap}, kW)	100 (-)
	Evaporator temperature (T_{evap}, °C)	5 (range 0 to 10)
	Condenser temperature (T_{cond}, °C)	40 (range 35 to 45)
	Evaporator coolant Inlet temperature ($T_{evap,in}$, °C)	15 (-)
	Evaporator coolant outlet temperature ($T_{evap,out}$, °C)	10 (-)
	Condenser coolant inlet temperature ($T_{cond,in}$, °C)	30 (-)
	Condenser coolant outlet temperature ($T_{cond,out}$, °C)	35 (-)
	Overlap degree ($T_{overlap}$, °C)	5 (-)
	Degree of subcooling in condenser (T_{sc}, °C)	5(-)
	Degree of subcooling in subcooler (T_{sc}, °C)	10 (-)
	Degree of superheating in evaporator (T_{sup}, °C)	5 (-)
	Degree of superheating in suction line (T_{sup}, °C)	10 (-)
	Desuperheating in discharge line (T_{dsup}, °C)	10 (-)
	Environment temperature (T_o, °C)	25 (-)
	Environment pressure (P_o, kPa)	101.325 (-)
	Refrigerants (for both cycles)	R134a (-)
	Pressure drop in the evaporator (kPa)	10 (-)
	Pressure drop in the suction line (kPa)	20 (-)
	Pressure drop in the condenser (kPa)	5 (-)
	Pressure drop in the liquid line and subcooler (kPa)	10 (-)

Table 3. Simulation condition data of thermodynamic properties of the DMS-VCR system.

State point	T (°C)	h (kJ/kg)	P (kPa)	s (kJ/kg.K)	$\dot{m}$ (kg/s)	x
1	70.67	304.3	1024	1.016	0.599	1
2	60.67	293.8	1024	0.985	0.599	1
2'	39.82	271.3	1021	0.915	0.599	1
3	39.82	108	1016	0.395	0.599	0
3'	34.82	100.6	1012	0.394	0.599	0
4	24.82	86.15	1002	0.323	0.599	0
5	5	86.15	349.9	0.328	0.599	0.14
6	4.17	253.3	339.9	0.929	0.599	1
6'	9.17	252.9	329.9	0.946	0.599	1
7	19.17	266.9	309.9	0.983	0.599	1
8	45.22	277.1	1017	0.934	0.085	1
9	40	108.3	1017	0.395	0.085	0
10	19.82	108.3	568.8	0.399	0.085	0.16
11	19.82	261.5	568.8	0.922	0.085	1

Table 4. Energy analysis results of actual VCR system and DMS-VCR system.

Performance parameters		Actual VCR system			DMS-VCR system		
		T_{evap} = 0 °C	T_{evap} = 5 °C	T_{evap} = 10 °C	T_{evap} = 0 °C	T_{evap} = 5 °C	T_{evap} = 10 °C
First law parameters	COP_{VCR}	3.26	4.08	5.13	-	-	-
	$COP_{DMS-VCR}$	-	-	-	3.41	4.21	5.22
	$\dot{Q}_{cond1}$ (kW)	127.9	122	117.2	116.7	111.4	107.2
	$\dot{Q}_{cond2}$ (kW)	-	-	-	14.70	14.44	14.19
	$\dot{Q}_{evap}$ (kW)	100	100	100	100	100	100
	$\dot{Q}_{sc}$ (kW)	-	-	-	13.34	13.10	12.88
	$\dot{W}_{comp1}$ (kW)	30.64	24.49	19.48	27.99	22.41	17.86
	$\dot{W}_{comp2}$ (kW)	-	-	-	1.36	1.33	1.31
	$\dot{m}_{ref,1}$ (kg/s)	0.67	0.66	0.64	0.61	0.59	0.58
	$\dot{m}_{ref,2}$ (kg/s)	-	-	-	0.08	0.08	0.08
	$\dot{m}_{ef,ev}$ (kg/s)	4.77	4.77	4.77	4.77	4.77	4.77
	$\dot{m}_{ef,cond1}$ (kg/s)	6.12	5.84	5.61	5.58	5.33	5.13
	$\dot{m}_{ef,cond2}$ (kg/s)	-	-	-	0.70	0.69	0.68

rate in the DMS-VCR system is reduced by 4.68 %, 3.17 % and 0.87 %, respectively. In addition, the DMS-VCR system exhibits an increase in exergetic efficiency by 4.38 %, 3.09 % and 1.61 %, respectively (at T_{evap} = 0 °C, 5 °C, and 10 °C), which presents the significance of incorporating DMS sub system.

In Solanki et al. [12], the energy and exergy analysis focused solely on a fixed evaporator temperature of 5 °C. In contrast, the current study introduced a broader spectrum of evaporator temperatures, encompassing 0 °C, 5 °C, and 10 °C, coupled with varying condenser temperatures ranging from 35 °C to 40 °C. This expansive range of evaporator temperatures facilitates a comprehensive analysis across diverse operational conditions, providing deeper insights into the behavior of the system.

5.3 Results of structural analysis

To enhance the DMS-VCR system's performance, it is necessary to minimize the irreversibility rate of each system component. The VCR cycle compressor-1 has the greatest irreversibility rate, making up 41.10 % of the system's overall irreversibility rate. The DMS-VCR system's components are put in sequence of increasing irreversibility rate at T_{evap} = 5 °C, and the outcomes are: expansion valve-2 (0.12 kW i.e. 0.84 %), compressor-2 (0.29 kW i.e. 2.02 %), condenser-2 (0.35 kW i.e.

Table 5. Exergy analysis results of actual VCR system and DMS-VCR system.

Performance parameters		Actual VCR system			DMS-VCR system		
		T_{evap} = 0 °C	T_{evap} = 5 °C	T_{evap} = 10 °C	T_{evap} = 0 °C	T_{evap} = 5 °C	T_{evap} = 10 °C
Second law parameters	$\dot{I}_{cond1}$	3.74	3.36	3.10	3.41	3.07	2.83
	$\dot{I}_{cond2}$	-	-	-	0.36	0.35	0.34
	$\dot{I}_{comp1}$	8.84	6.44	4.72	8.08	5.89	4.33
	$\dot{I}_{comp2}$	-	-	-	0.30	0.29	0.29
	$\dot{I}_{evap}$	5.23	3.17	1.20	5.18	3.13	1.17
	$\dot{I}_{ev1}$	2.49	1.83	1.30	1.35	0.93	0.61
	$\dot{I}_{ev2}$	-	-	-	0.12	0.12	0.11
	$\dot{I}_{sc}$	-	-	-	0.55	0.55	0.54
	$\dot{I}_{t}$	20.30	14.80	10.32	19.35	14.33	10.23
	η_{II} (%)	29.87	29.36	27.19	31.18	30.27	27.63

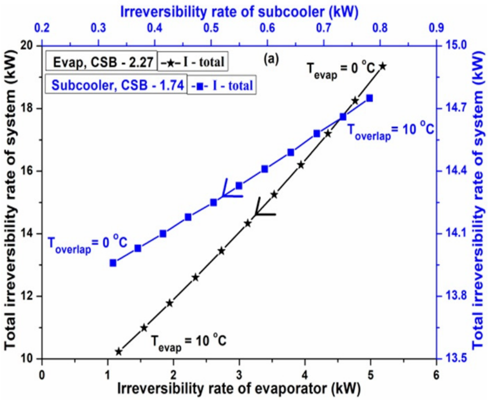
2.44 %), subcooler (0.55 kW i.e. 3.84 %), expansion valve-1 (0.93 kW i.e. 6.49 %), condenser-1 (3.07 kW i.e. 21.42 %), evaporator (3.13 kW i.e. 21.84 %), and compressor-1 (5.89 kW i.e. 41.10 %). Several researchers [11, 17, 18] have proposed the idea of CSB, and in this study, CSB values are calculated for various components of the DSM-VCR system, as presented in Figs. 5(a)-(c). The selected components for the CSB analysis include the subcooler, evaporator, condenser-1, condenser-2, compressor-1, and compressor-2, for which the CSB values are computed. The efficiency parameters (z_i) of the condenser temperature, evaporator temperature, isentropic efficiency, degree of subcooling and degree of overlap are considered for the purpose of CSB analysis.

Fig. 5(a) shows that increase in the evaporator's temperature from 0 °C to 10 °C causes a reduction in the irreversibility rate of evaporator by 77.41 % (from 5.18 kW to 1.17 kW), which reduces the system's overall irreversibility rate by 47.13 % (from 19.35 kW to 10.23 kW). Accordingly, it is appropriate to maintain the evaporator temperature high and the condenser temperature low for optimum performance of DMS-VCR system, depending on the needs of the particular application. However the condenser temperature depends on the surrounding temperature. As shown in Fig. 5(a), by decreasing the degree of overlap from 10 °C to 0 °C leads to a significant decrease of 57.69 % (from 0.78 kW to 0.33 kW) in the subcooler's irreversibility rate, which causes the DMS-VCR system's overall irreversibility rate to decline by 5.35 % (from 14.75 kW to 13.96 kW). Therefore, it is suggested to use a lower degree of overlap, although this would require a larger area of subcooler for effective heat transfer. In addition, if the temperature of the condenser-1 reduces by 10 °C (from 45 °C to 35 °C), it leads to a significant reduction in its irreversibility rate by 70.83 %, (from 5.76 kW to 1.68 kW) and condenser-2 by 79.31 % (from 0.58 kW to 0.12 kW), as illustrated in Fig. 5(b). The total irreversibility rate of the DMS-VCR system also reduces by 43 % (from 18.58 kW to 10.59 kW) when condenser-1 temperature is reduced by 10 °C and similar reduction in the irreversibility rate of DMS-VCR system is 5.23 % (from 14.73 kW to 13.96 kW) for condenser-2.

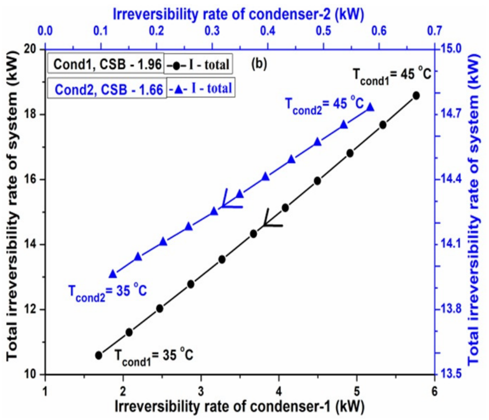
Additionally, the study reveals that achieving 100 % isentropic efficiency in compressor-1 and 2 can lead to a reduction of their irreversibility rates by 100 %, indicating an ideal condition. This would result in a significant decrease in the DMS-VCR system's overall irreversibility rate, with a reduction of 50.24 % in compressor-1 (from 16.02 kW to 7.97 kW) and compressor-2 seeing a reduction of 3.64 % (from 14.56 kW to 14.03 kW), as shown in Fig. 5(c). Compressor-1 has the greatest influence on the DMS-VCR system's total irreversibility rate.

Further, the study shows that, when compared to the other components, compressor-1 adds the most to the overall irreversibility of the DMS-VCR system, accounting for 5.89 kW, or 41.10 %. Therefore, improving the efficiency of compressor-1 is crucial for enhancing the DMS-VCR system's performance. The efficiency of the compressors depends on the pressure ratio as mentioned in equations shown under serial number 4 and 5 of the Table 1. The pressure ratio across compressors depends on the evaporator and condenser pressures and decides the efficiency of compressors. However, the evaporator pressure depends on the evaporator temperature which is dependent on application temperature for which the system is used for and can be varied only in a narrow range whereas condenser temperature depends on the ambient or surrounding conditions and hence it is not feasible to change it at will. Thus DMS-VCR system pressure ratio across compressor-1 can be changed only in a narrow range depending on the small range of the application for which it is being used.

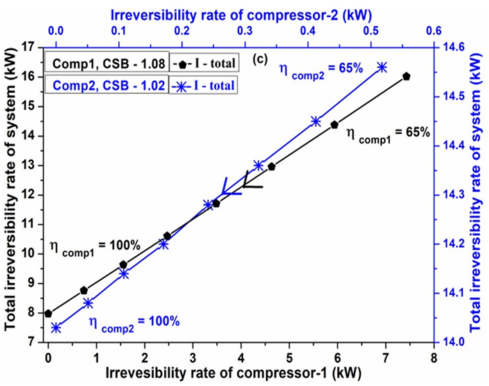
The computed CSB values for the components of DMS-VCR system are shown in Fig. 6. The components can be ranked in ascending order of their CSB values, as follows: compressor-2 (1.02), compressor-1 (1.08), condenser-2 (1.66), subcooler (1.74), condenser-1 (1.96), and evaporator (2.27). All the components have CSB values greater than one which means re-



(a)



(b)



(c)

Fig. 5. Figures showing how the system's overall irreversibility rate has changed in relation to its various components, including: (a) evaporator and subcooler; (b) condenser-1 and condenser-2; (c) compressor-1 and compressor-2.

ducing its value will reduce the component irreversibility rate as well as the system irreversibility rate.

Therefore, modifying these components' efficiency factors would enhance both the performance of the individual component as well as the performance of the entire system. Therefore, even a minor adjustment to the component's operating parameters with a high CSB number could have a big impact on how well the system performs. The component with the highest CSB value is evaporator (2.27), and its contribution to the total irreversibility rate is 21.84 %, followed by the condenser-1 (CSB-1.96) with a contribution of 21.42 %. Compressor-1 has the highest overall contribution to the total irreversibility rate, which is 41.10 %, but it is less sensitive to improvement in exergetic efficiency as its CSB is low i.e.1.08 compared to other components of the DMS-VCR system. The CSB analysis emphasizes the significance of raising the system's overall efficiency, particularly by focusing on components with higher CSB values, such as the evaporator and condenser-1. The overall performance of the DMS-VCR system can be significantly enhanced by increasing the efficiency of these components.

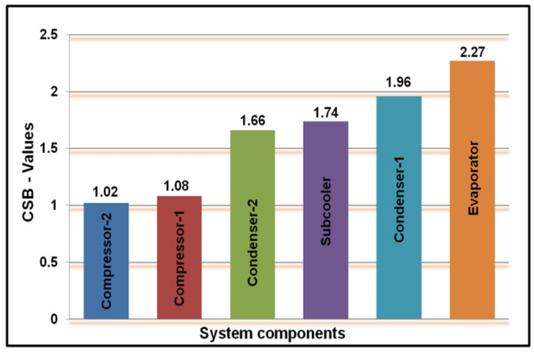


Fig. 6. Comparison of CSB values of DMS-VCR system components.

5.4 Effect of evaporator and condenser temperatures

Fig. 7 illustrates the effect of evaporator temperature on both the coefficient of performance and the total irreversibility rate on DMS-VCR system. Although the working range of the external fluid for the condenser and evaporator is only 5 °C (for example, the evaporator's external fluid intake temperature is 15 °C and its outlet temperature is 10 °C). The DMS-VCR system's evaporator temperature varies between 0 °C and 10 °C while maintaining a constant intake and output temperature for the external fluid and other design parameters. Consequently, the system's overall COP improves from 3.41 to 5.22 on increasing evaporator temperature. The system's total irreversibility rate reduces from 19.35 kW to 10.23 kW, and the exergetic efficiency decreases from 31.18 % to 27.63 %. This decline in irreversibility rate is exclusively because of the reduction in the temperature difference between the evaporator and external fluid, which is caused by an increase in the evaporator temperature. Hence, high evaporator temperature is necessary from an energy point of view and however from exergetic viewpoint, the evaporator temperature should be lower.

Fig. 8 displays the effect of condenser temperature on the DMS-VCR system's total irreversibility rate and coefficient of performance. The DMS-VCR system's condenser temperature is changed from 35 °C to 45 °C while maintaining the same temperatures for the external fluid's inlet and exit as well as

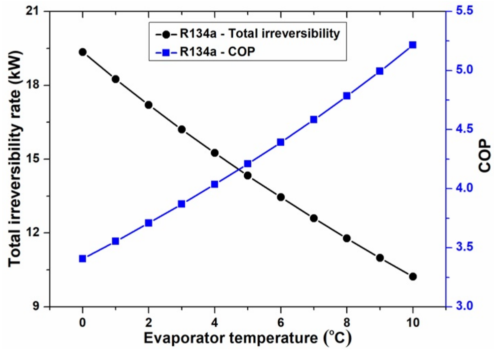


Fig. 7. Effect of evaporator temperature on total irreversibility rate and COP.

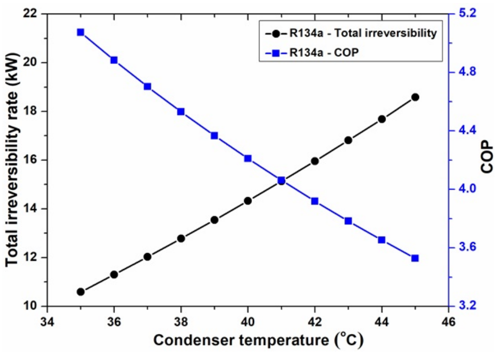


Fig. 8. Effect of condenser temperature on total irreversibility rate and COP.

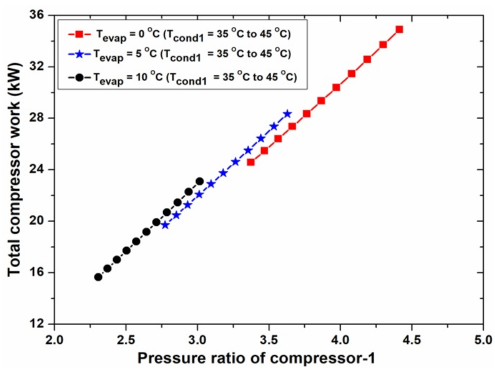


Fig. 9. Effect of pressure ratio of compressor-1 on total compressor work.

other design parameters. It causes the system's overall coefficient of performance to drop from 5.07 to 3.53, while the system's exergetic efficiency declines from 36.48 % to 25.37 % and its total irreversibility rate increases from 10.59 kW to 18.58 kW.

Fig. 9 displays the effect of pressure ratio of compressor-1 on total compressor work of the DMS-VCR system. The pressure ratio across compressor increases at lower evaporator and higher condenser temperatures. The compressor power is dependent on pressure ratio and inlet temperature to compressor. Hence the compressor power is observed to be higher for higher pressure ratio. The current paper explores the effect of compressor pressure ratio on total work at different evaporator temperatures, a critical aspect not addressed in Solanki et al. [12].

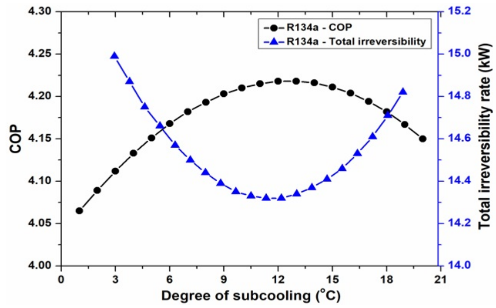


Fig. 10. Variation of COP and total irreversibility rate with respect to degree of subcooling in dedicated subcooler.

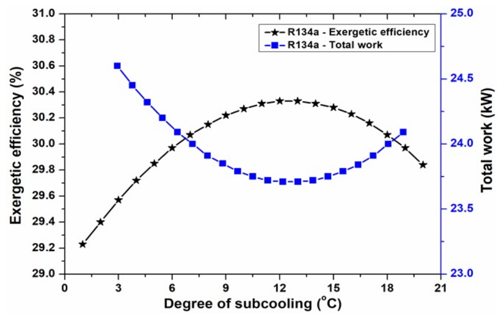


Fig. 11. Variation of exergetic efficiency and total work with respect to degree of subcooling.

The effect of temperature at inlet to compressor is not substantial. In nutshell both the evaporator and condenser temperatures affect pressure ratio and the total compressor work. The higher condenser temperature results in higher total compressor work due to higher pressure ratios. On the other hand, higher evaporator temperatures result in a drop in total compressor work due to reduction in pressure ratio.

5.5 Effect of degree of subcooling and superheating

The degree of subcooling has a significant effect on the effectiveness, efficiency, and overall cooling capacity of the system. Engineers and technicians can increase the cooling efficiency of the system and guarantee its optimal performance by optimizing the degree of subcooling. The current DMS sub system of DMS-VCR system is made to further cool the refrigerant liquid leaving the condenser below its saturation temperature. This additional cooling is accomplished by transferring heat from the refrigerant in VCR section of DMS-VCR system to another secondary refrigerant in DMS section of DMS-VCR system. The degree of subcooling is varied from 1 °C to 20 °C keeping the other design parameters constant (see Table 2). Through various simulations, it has been concluded that the ideal degree of subcooling is 12 °C. According to Figs. 10 and 11, at this subcooling level, the system's overall COP and exergetic efficiency increase, while its total irreversibility rate and total compressor work decreases. Below or above the optimal value of subcooling, the reverse patterns are observed.

Upto ideal degree of subcooling the VCR cycle compressor-1 work decreases from 24.3 kW to 22.04 kW, whereas the DMS cycle compressor work increases from 0.30 kW to 1.67 kW, resulting in a decrease in the total compressor work from 24.60 kW to 23.71 kW. It increases the system's overall COP from 4.06 to 4.22 (an improvement of 4 %). As a result, the system's entire irreversibility drops by 4.5 %.

The superheating is also an important parameter to improve the overall performance of the system, as it increases the temperature of the refrigerant leaving the evaporator by gaining heat from the products to be cooled or surrounding. This indirectly contributes to the protection and longevity of the compressor.

In the current system, the degree of superheat is varied from 1 °C to 20 °C in the evaporator and suction line of the system, keeping other design parameters constant (see Table 2). Figs. 12 and 13 show the effect of degree of superheating on the performance parameters of the DMS-VCR system. The EES simulation results show that the system's overall COP decreases from 4.40 to 4.04 and the system's entire irreversibility rate increases by 8.83 %. Further, the exergetic efficiency decreases from 31.62 % to 29.06 %, while, the total compressor work increases from 22.74 kW to 24.74 kW. Therefore, it is essential to note that superheating in the present case is not beneficial from the performance view point. In order to maximize performance and ensure safe operation, engineering systems must carefully assess the degree of subcooling and superheating.

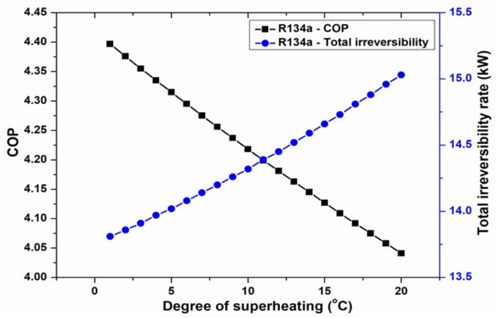


Fig. 12. Variation of COP and total irreversibility rate with respect to degree of superheating.

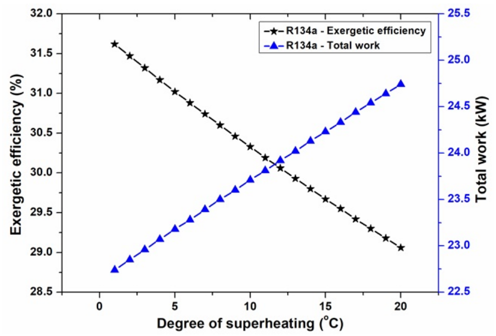


Fig. 13. Variation of exergetic efficiency and total work with respect to degree of superheating.

5.6 Effect of pressure drop in evaporator and condenser

The overall pressure drop in an evaporator is influenced by both the momentum pressure drop and the friction pressure drop. As the vapor flows through the evaporator tubes, it experiences frictional resistance with the interior surfaces, leading to a decrease in pressure. Further, as the vapor velocity increases, the momentum pressure increases in the evaporator. But in a condenser, the vapor condenses into a liquid form, resulting in a decrease in vapor velocity and a decrease in momentum pressure drop. However, friction pressure drop continues to be positive, just like in the evaporator, because the fluid faces resistance as it passes through the evaporator and condenser's interior surfaces. The condenser experiences a marginal overall reduction in pressure as a result of the combination of these pressure drops. The pressure ratio across the compressor rises as the evaporator pressure drop increases, increasing the work required by the compressor; nevertheless, the pressure ratio across the condenser does not vary significantly as discussed above. The COP and exergetic efficiency of the DMS-VCR system decreases as a result of the increased compressor work.

Figs. 14 and 15 depict how a pressure drop in an evaporator affects the COP, total irreversibility rate, exergetic efficiency and total compressor work with variation of pressure drop in evapora-

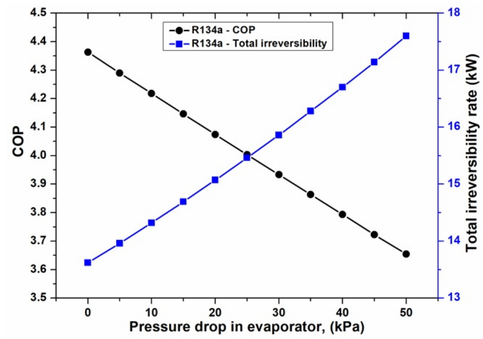


Fig. 14. Variation of COP and total irreversibility rate with respect to pressure drop in evaporator.

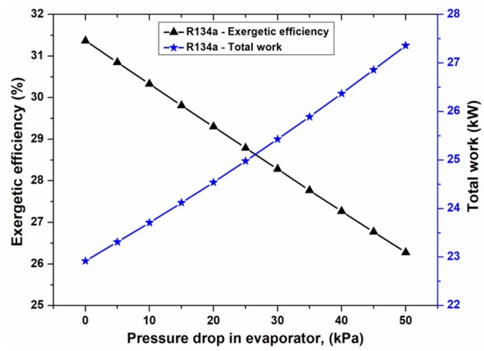


Fig. 15. Variation of exergetic efficiency and total work with respect to pressure drop in evaporator.

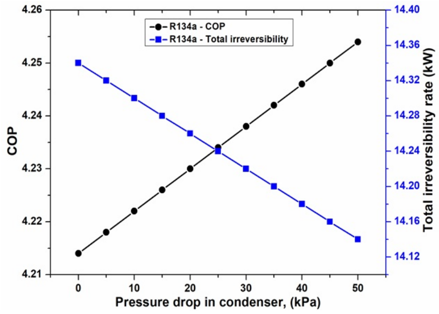


Fig. 16. Variation of COP and total irreversibility rate with respect to pressure drop in condenser.

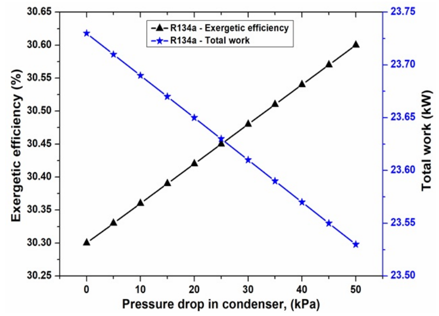


Fig. 17. Variation of exergetic efficiency and total work with respect to pressure drop in condenser.

tor from 0 kPa to 50 kPa. Due to a decrease in the specific refrigerating effect, the cooling capacity decreases as the pressure drop in the evaporator increases, and compressor work increases as the pressure ratio across the compressor rises. Both of these impacts contribute to decrease the COP from 4.36 to 3.65 and an increase in total compressor work by 19.37 %. The total irreversibility rate of the system increases by 29.22 % with a decrease in exergetic efficiency from 31.37 % to 26.28 %.

Contrary to the above the condenser pressure drop exhibits very marginal changes in the values of performance parameters i.e. COP and exergetic efficiency increases from 4.21 to 4.25 and from 30.30 % to 30.60 %, respectively, while total irreversibility rate and total compressor work decreases from 14.34 kW to 14.14 kW, and from 23.73 kW to 23.53 kW, respectively, with variation of pressure drop from 0 kPa to 50 kPa (see Figs. 16 and 17). This therefore implies that the evaporator pressure drop has a more significant effect on the overall performance of the DMS-VCR system.

6. Conclusion

The current study focuses on energy, exergy and CSB analysis of DMS-VCR system. At constant condenser temperature (T_{cond} = 40 °C) and variable evaporator temperatures (T_{evap} = 0 °C, 5 °C, and 10 °C), the results show that the DMS-VCR system works better than an actual VCR system. At 0 °C, the work input for the compressor-1 of the DMS-VCR system is reduced by 8.65 %. The COP and exergetic efficiency of DMS-VCR system both increase by 4.60 % and 4.38 %, respectively. Due to less work being required to provide the same cooling capacity, the DMS-VCR system's operating cost is also reduced. The irreversibility rate of each component is also calculated by exergy analysis, as well as the total irreversibility rate of the DMS-VCR system, which is revealed to be 4.68 % lower than the actual VCR system.

The CSB analysis is then employed to investigate how changes in irreversibility rate of individual components affect the system's overall performance. The values of CSB of evaporator and condenser-1 are most suitable for reducing the irreversibility rate of the overall system and increasing the exergetic efficiency.

Nomenclature

CSB : Coefficient of structural bonds
COP : Coefficient of performance
Cp : Specific-heat (kJ/kg.K)
C : Thermal capacitance rate (kW/K)
DMS : Dedicated mechanical subcooling
DOS : Degree of subcooling
DOO : Degree of overlap
H : Enthalpy (specific, kJ/kg)
I : Irreversibility (rate, kW)
$\dot{m}$: Mass (flow rate, kg/s)
P : State point pressure (kPa)
PR : Pressure ratio
Q : Rate of heat transfer (kW)
s : Entropy (specific, kJ/kg.K)
S_gen : Entropy generation rate (kW/K)
T : Temperature (°C)
VCR : Vapor compression refrigeration
W : Power input rate (kW)

Greek symbols

ϵ : Effectiveness
η : Efficiency

Subscripts

o : Environment condition
dl : Discharge line
comp : Compressor
cond : Condenser
EoS : Element or subsystem
ef : External fluid
ev : Expansion valve
ex : Exergetic
evap : Evaporator
in : Inlet condition
isen : Isentropic
max : Maximum
out : Outlet condition
ol : Overlap
ref : Refrigerant
sc : Subcooling

sup : Superheating
sl : Suction line
t : Total
Z_i : System efficiency parameter
n : Nth element of system
x : Dryness fraction (quality)
1,2,3… : State points

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