J Therm Anal Calorim (2012) 110:961–971
DOI 10.1007/s10973-011-1904-z

Exergy analysis and experimental study of a vapor compression refrigeration cycle

A technical note

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Received: 28 July 2011 / Accepted: 1 September 2011 / Published online: 28 September 2011

Abstract This article presents a detailed experimental analysis of 2TR (ton of refrigeration) vapor compression refrigeration cycle for different percentage of refrigerant charge using exergy analysis. An experimental setup has been developed and evaluated on different operating conditions using a test rig having R22 as working fluid. The coefficient of performance, exergy destruction, and exergetic efficiency for variable quantity of refrigerant has been calculated. The present investigation has been done by using 2TR window air conditioner and the results indicate that the losses in the compressor are more pronounced, while the losses in the condenser are less pronounced as compared to other components, i.e., evaporator and expansion device. The total exergy destruction is highest when the system is 100% charged, whereas it is found to be least when the system is 25% charged.

Keywords Vapor compression system · Exergy analysis · COP · Exergy efficiency · Irreversibility · Exergy destruction

List of symbols

TR	Ton of refrigeration
BTU	British thermal units
COP	Coefficient of performance
Q_e	Refrigerating effect (W)
W_c	Compressor work (W)
$EX_{Devap.}$	Exergy destruction in evaporator (W)
$EX_{Dcomp.}$	Exergy destruction in compressor (W)
$EX_{Dcond.}$	Exergy destruction in condenser (W)
$EX_{Dexp.}$	Exergy destruction in expansion device (W)
EX_{Dtotal}	Total exergy destruction (W)
E_x	Exergy (W)
m	Mass flow rate (kg/s)
s	Entropy (kJ kg^{-1} K^{-1})
h	Enthalpy (kJ kg^{-1})
T_o	Reference temperature (°C)
T_r	Evaporator temperature (°C)
COP_c	Carnot coefficient of performance
$P_{disch.}$	Discharge pressure (kg/cm^2)
$P_{suc.}$	Suction pressure (kg/cm^2)
Cond. In	Condenser inlet temperature (°C)
Evap. Inlet	Evaporator inlet temperature (°C)
η_{exergy}	Exergy efficiency

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Introduction

Refrigeration plays a very important role in industrial, domestic, and commercial sectors for cooling, heating, and food preserving applications. There are innumerable applications of such systems and they are the major consumer of electricity around the world. Energy consumption is directly proportional to the economic development of any nation, however, this area is in great interest now because of increase in the cost of conventional fuels and environmental concerns globally. The scientists are looking for new and renewable sources of energy so as to minimize the costs. Due to the increasing energy demand, degradation of environment, global warming, depletion of ozone layer, etc., there is urgent need of efficient energy utilization and waste heat recovery for useful applications. The researchers are concentrating on the alternate and environment friendly refrigerants, especially after the Kyoto and the Montreal protocols. However, in a quest to find out alternate and environment friendly refrigerants, the energy efficiency of the equipment having conventional refrigerants is also very important in the present age of competitive business community. The aim of the scientific community all over the world is to switch to new and renewable energy sources besides, efficient utilization of all conventional sources.

Air conditioning bears a huge cost because thermal comfort is very essential as far as domestic and industrial sectors are concerned. The big challenge is to use less energy for air conditioning applications in order to reduce the associated power consumption so as to make them more efficient and environmental friendly. The quantitative information is required to be obtained that will show the irreversibility of a process in all the components of any plant. For effective use and proper optimization, the detailed understanding of different thermodynamic processes in any conversion system is very important. In order to optimize their design, a thorough thermodynamic analysis is required. The analysis based on first law of thermodynamic is most commonly used in engineering applications, however, it is concerned only with law of conservation of energy and therefore, it cannot show how and where irreversibility in the system or a process occurs. On the other hand, the analysis based on second law analysis is well-known method being used to analyze all the thermodynamic cycles for better understanding and evaluation of irreversibility associated with any process. Unlike the first law (energy), the analysis based on second law analysis (exergy) determines the magnitude of irreversibility associated in a process qualitatively and thereby, provides an indication to point out the directions in which the engineers should concentrate more in order to improve the performance of these thermodynamic systems [1–3]. Thus, the aim of second law-based analysis is to determine the exergy losses and to enhance the performance by changing the design parameters and hence, to reduce the cost of the refrigeration cycle [4].

A lot of studies on the performance evaluation and optimization have been carried out experimentally and theoretically are available in the literature [5–10]. Most of the studies carried out so far on the refrigeration systems shows that the performance analysis of refrigeration systems were investigated based on first law of thermodynamics. However, this approach is of limited use in view of the fact that the actual energetic losses are difficult to make out because the first law deals with the quantity of energy and not the quality of energy. In order to calculate the actual losses due to irreversibility in the process, exergy analysis based on second law of thermodynamics is the proper tool. Exergy analysis utilizes exergetic efficiency criterion taking into account all the losses appearing in a system, for measuring the actual performance.

The exergy analysis is widely accepted as a useful tool in obtaining the improved understanding of the overall performance of any system and its components [11]. Exergy analysis also helps in taking account the important engineering decisions regarding design parameters of a system [12]. Many researchers have carried out exergy studies of different thermal energy conversion systems describing various approach for exergy analysis and its usefulness in a more simple and effective manner [13–23]. Padilla et al. [13] carried out the exergy analysis and the impact of direct replacement of R12 with zeotropic mixture R413A. The performance of a domestic vapor compression refrigeration (VCR) system originally designed to work with R12 was evaluated using a simulated modeling. They concluded that the overall energy and exergy performance of this system working with R413A is better than that of R12. Kumar et al. [14] derived a method to carry out the exergetic analysis of a VCR system using R11 and R12 as refrigerants. The procedure to calculate various losses as well as coefficient of performance (COP) and exergetic efficiency of the cycle has been explained by proper example. Arora and Kaushik [15] did a detailed exergy analysis of an actual VCR cycle. They developed a computational model to calculate the COP, exergy destruction, exergy efficiency, and the efficiency defects for R502, R404A, and R507A for temperature in the range of −50 to 0 °C and condenser temperature range of 40–55 °C. They concluded that R507A is a better substitute to R502A than that of R404A. Nikolaidis and Probert [16] studied the behavior of a two-stage compound compression cycle, with flash intercooling with R22 using the exergy method and gave some useful conclusions. Dincer [17] asserts that conventional energy analysis, based on the first law of thermodynamics, evaluates energy mainly on its quantity but analysis that are based on second law considers not only the quality of energy, but also quantity of energy and similar observation were given by other researchers as available in the literature [19–25].

In this study, the main objective is to investigate the performance of a VCR system based on exergy analysis. The experimental analysis has been done on a 2TR (ton of refrigeration) window air conditioning system using R-22 as refrigerant. With the objective to find out the losses at different operating conditions for vapor compression cycle, exergy analysis has been done by varying the quantity of refrigerant charge. The system has been modified for experimental study to find the possible design conditions with the minimum exergy destruction. Besides the effects of temperature changes in the condenser and evaporator on the plant's irreversibility rate was determined. It is observed that the greater is the temperature difference between the condenser and the environment or between the evaporator and the cold room, the higher is the irreversibility rate. The properties of refrigerant at each state point are calculated using Forane software [18] and the results are discussed in detail. The investigation will help to improve the understanding of vapor compression cycle and will enhance its efficiency. Further study on the design parameters (external and internal) operating conditions is in process to obtain the optimum performance of the system.

Experimental setup and procedure

The experimental setup used in this study is a 2TR window air conditioner which was further modified to incorporate the compound gages for temperature and pressure measurements at suction and discharge side. The main loop of the system under investigation is similar to that of a common VCR system and composed of four basic components only, i.e., compressor, an evaporator, condenser, and an expansion device. The vapors from low temperature evaporator are sucked into the compressor and compressed to increase the pressure and temperature. The compression is assumed to be polytropic and the vapors are condensed in the air cooled finned condenser in which the fan is driven by an electric motor which is also driving the evaporator fan at other end. The speed of the condenser and evaporator fan is kept constant. The compressor is hermetic type reciprocating compressor with 2,100 W nominal input power at 220 V (50 Hz). The evaporator was designed for a cooling capacity of 24K BTU (British thermal units). The refrigerant is charged in four steps and the performance is evaluated at each stage. The unit also comprises of other devices such as filters and compressor protection device. The refrigerant volume flow rate was measured using a rotameter which is specifically calibrated for R-22. The temperature and pressure were measured by using compound gages. The photographic view of the experimental setup is shown in Fig. 1 and the specifications of the VCR system are shown in Table 1.

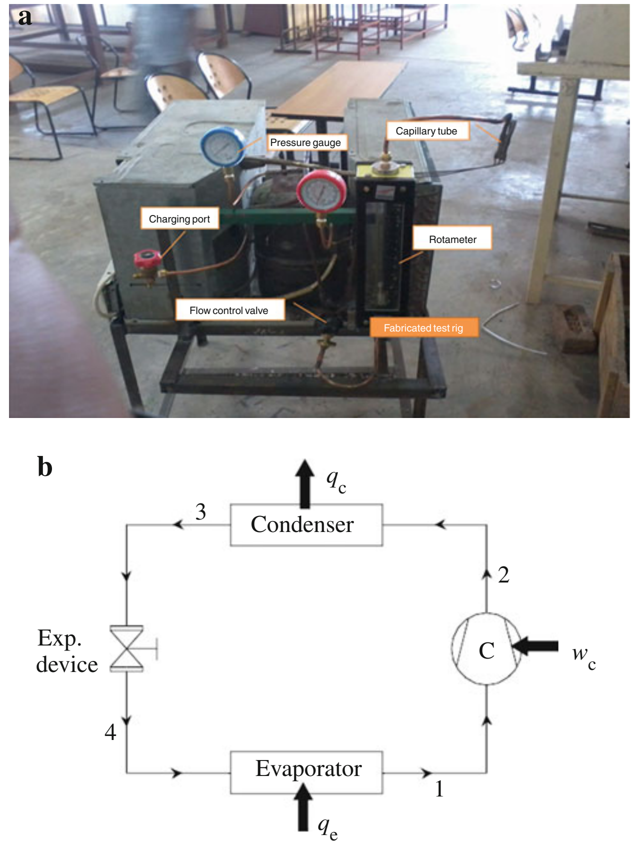


Fig. 1 **a** Experimental setup of 2 TR window air conditioner based on VCR cycle. **b** Line diagram of a typical VCR cycle

Table 1 Specification of experimental setup

System specifications	
Type	Window air conditioner
Capacity	2 TR (24000 BTU)
Condenser	Finned coils, air cooled
Eator	Finned coils
Expansion device	Capillary tube
Compressor	Hermetically sealed, reciprocating

Exergy analysis

Exergy analysis has two advantages over the conventional heat balance method for design and performance analysis of energy-related systems. It provides a more accurate measurement of the actual inefficiencies in the system and the true location of these in efficiencies. In refrigeration cycle, with the heat balance analysis, it is not possible to find out the true losses. Exergy analysis is based on the assumption that there is an infinite equilibrium environment that ultimately surrounds all systems that are to be analyzed. The exergy or available energy of a system is the maximum work that could be derived if the system were allowed to come to equilibrium with the environment. It is a consequence of the second law of thermodynamics that the combined exergy of all systems can only decrease or remain unchanged. Unlike energy, exergy is not conserved, once it is lost, it is lost forever. In other words, exergy (quality) is degradable, while energy (quantity) is conserved. Exergy can be exchanged between systems, but if there are thermodynamic irreversibility's such as, friction or heat transfer with finite temperature differences, some of the potential for the production of work is destroyed. In all real processes, therefore, the total exergy of the system decreases [19–23].

For a specified system boundary a clear distinction can be made between exergy destruction and exergy loss. Exergy loss is exergy that is passed on to some other system often the environment and which cannot be considered useful in the context of the purpose of the system. The term exergy destruction is used when the potential for the production of work is destroyed within the system boundary. The exergy of a system is a co-property of the system and the environment. In exergy analysis of compressors the environment consists of the local surroundings of the compressor. These local surroundings are modeled as being in equilibrium and infinite. Given sufficient information, the exergy of all the systems can be determined at any time. On the basis of first law, the performance of refrigeration cycle is based on the COP, which is defined as the ratio of net refrigerating effect (cooling/heating load) obtained per unit of power consumed. It is expressed as

$$\mathrm{COP} = \frac{Q_e}{W_c} \tag{1}$$

exergy balance for a control volume can be expressed as [2]:

$$\mathrm{EX_D} = \sum (me_x)_{\mathrm{in}} - \sum (me_x)_{\mathrm{out}} + \left[\sum\left(Q\left(1-\frac{T_o}{T}\right)\right)_{\mathrm{in}} - \sum\left(Q\left(1-\frac{T_o}{T}\right)\right)_{\mathrm{out}}\right] \pm \sum W \tag{2}$$

For the present system shown in Fig. 1b, the component wise exergy balance equation can be written as below:

(a) Compressor:

$$(\mathrm{EX_D})_{\mathrm{comp.}} = E_{x1} + W_c - E_{x2} = m_r(T_o \cdot (s_2 - s_1)) \tag{3}$$

(b) Condenser:

$$(\mathrm{EX_D})_{\mathrm{cond.}} = E_{x2} - E_{x3} = m_r(h_2 - T_o \cdot s_2) - m_r(h_3 - T_o \cdot s_3) \tag{4}$$

(c) Expansion device:

$$(\mathrm{EX_D})_{\mathrm{exp.}} = E_{x3} - E_{x4} = m_r(h_3 - T_o \cdot s_3) - m_r(h_4 - T_o \cdot s_4) = m_r(T_o \cdot (s_2 - s_1)) \tag{5}$$

(d) Evaporator:

$$(\mathrm{EX_D})_{\mathrm{evap.}} = E_{x4} + Q_e\left(1-\frac{T_o}{T_r}\right) - E_{x1} = m_r(h_4 - T_o \cdot s_4) + Q_e\left(1-\frac{T_o}{T_r}\right) - m_r(h_1 - T_o \cdot s_1) \tag{6}$$

The total exergy destruction in the system is the sum of exergy destruction in different components of the system and is given by

(e) Total exergy destruction:

$$(\mathrm{EX_D})_{\mathrm{total}} = (\mathrm{EX_D})_{\mathrm{comp.}} + (\mathrm{EX_D})_{\mathrm{cond.}} + (\mathrm{EX_D})_{\mathrm{exp.}} + (\mathrm{EX_D})_{\mathrm{evap.}} \tag{7}$$

(f) Exergy efficiency:

$$\eta_{\mathrm{exergy}} = \frac{Q_e\left|\left(1-\frac{T_o}{T_r}\right)\right|}{W_c} \tag{8}$$

Results and discussion

In order to have a comparative study of vapor compression cycle by varying the quantity of refrigerant charged, the real time data was measured and the calculations were made using Forane software [18]. The temperature, pressure, and mass flow rate of the refrigerant was measured using compound gages and rotameter, respectively, at different state points. The basic properties such as entropy and enthalpy of the refrigerant at different state points were calculated using Forane software [18]. The COP, refrigerating effect, compressor work, exergy destruction, and exergy efficiency are calculated using simple EXCEL sheet.

A 2TR window air conditioner equipped with different pressure, temperature, and flow measuring devices has been studied experimentally using energy and exergy analysis. The unit is charged with refrigerant R-22 in four steps, i.e., 25, 50, 75, and 100%, respectively, and the system performance is analyzed in each case. The reference temperature is measured to be 25 °C. The COP, cooling load, exergy destruction, and exergy efficiency has been evaluated against running time and evaporator temperature and the discussion of results is given in following paragraph.

Figure 2 shows the comparison of COP against time in each case and it is observed from Tables 1, 2, 3, 4, 5, 6, and 7 that the evaporator temperature is least in case when the system is 100% charged but this increases pressure ratio of the compressor and hence compressor work also increases as a result the COP goes down as can be seen from Fig. 2. In all the cases, it is observed that the COP fluctuates with time having number of peaks after certain intervals except when the system is 75% charged. In the case when the system is 75% charged the COP has the highest peak in the middle of running time followed by small peaks on either sides. Except one case when the system is 25% charged the COP first increases along the highest peak in the first few minutes and then decreases gradually with small ups and downs before finally reaching steady state condition in an hour's time. After an hour of running the system, the steady

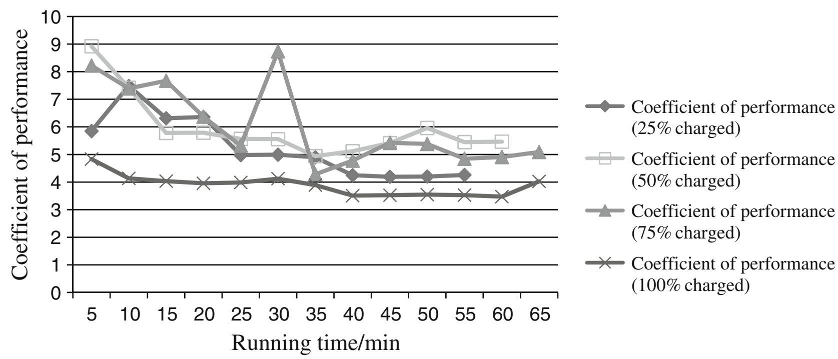


Fig. 2 Variation of COP with respect to running time

state COP was observed with very small fluctuations and hence, the data after a certain period was not plotted as can be seen in Fig. 2.

Figures 3, 4, 5, and 6 show the exergy efficiency variation with respect to evaporator temperature. It is observed that the highest exergy efficiency is found when the system is 100% charged. This is due to the fact that as observed the refrigerant temperature (evaporator) is the minimum when the system is 100% charged and therefore the term $Q_e\left|\left(1-\frac{T_o}{T_r}\right)\right|$ increases. The highest exergy efficiency is found to be 45.9% at an evaporator temperature of 7 °C when the system is 100% charged, while the least exergy efficiency was found to be 3.5% at an evaporator temperature of 24.5 °C and 25% system charging. The reason for reduced exergy efficiency when the refrigerant is only 25% charged is because the refrigerant temperature is almost same as the reference temperature and hence exergy content is very small in the later case as compared to the former case. Thus, the results are consistent with the hypothetical observations. However, as far as energy analysis is concerned. This particular result is reverse as can be seen by Carnot cycle efficiency.

$$\mathrm{COP_C} = \frac{T_L}{(T_H - T_L)} \tag{9}$$

From Tables 1, 2, 3, 4, 5, 6, and 7, it is observed that the exergy destruction in the compressor is found to be the highest as compared to other components, i.e., evaporator, condenser, and expansion devices. This is attributed to the reciprocating compressor being used in the

Table 2 Parameters of the VCR system when the unit is 25% charged

Time	$P_{disch.}$/kg/cm^2	$P_{suc.}$/kg/cm^2	m/kg/s	Cond. In/°C	Evap. Inlet/°C
5	4	−15 cm of Hg	0.0236	52	24.5
10	9.2	0.4	0.0283	52	24.5
15	9.4	0.45	0.0306	58	21
20	9.6	0.55	0.0318	58	21
25	9.7	0.6	0.033	68	20.1
30	9.8	0.61	0.0332	68	20.1
35	9.9	0.68	0.0334	75	20.8
40	9.95	0.7	0.0334	75	20.8
45	10	0.7	0.0334	76	21.1
50	10.1	0.71	0.0334	76	21.1
55	10	0.7	0.0332	75	22.1

Table 3 Exergetic performance results of the VCR system when the unit is 25% charged

RE/kJ	$W_{comp.}$/kJ	COP	$EX_{Devap.}$	$EX_{Dcomp.}$	$EX_{Dcond.}$	$EX_{Dexp.}$	EX_{Dtotal}	Exergy efficiency	Exergy efficiency/%
4.283	0.732	5.848	2.443	3.723	1.836	2.442	10.446	0.0359	3.5
5.140	0.686	7.485	2.932	3.654	2.826	2.930	12.343	0.04319	4.31
5.547	0.878	6.313	3.109	4.068	3.092	3.169	13.440	0.1132	11.3
5.760	0.906	6.357	3.229	4.196	3.233	3.291	13.950	0.1175	11.75
5.941	1.193	4.977	3.322	4.579	3.376	3.425	14.703	0.1398	13.9
5.983	1.198	4.991	3.345	4.592	3.416	3.449	14.804	0.1408	14
5.895	1.203	4.897	3.391	4.614	3.447	3.469	14.924	0.1244	12.44
5.895	1.387	4.248	3.391	4.765	3.481	3.469	15.107	0.1244	12.44
5.915	1.411	4.191	3.392	4.785	3.484	3.474	15.137	0.1186	11.86
5.915	1.407	4.201	3.392	4.775	3.491	3.474	15.133	0.1186	11.86
5.854	1.374	4.258	3.394	4.731	3.453	3.445	15.024	0.0972	9.72

Table 4 Parameters of the VCR system when the unit is 50% charged

Time	$P_{disch.}$/kg/cm^2	$P_{suc.}$/kg/cm^2	m/kg/s	Cond. In/°C	Evap. Inlet/°C
5	9	0.4	0.028	47	23
10	9.4	0.5	0.0306	53	20.3
15	9.6	0.6	0.0306	62	21.5
20	9.8	0.65	0.0306	62.2	21.3
25	9.9	0.7	0.0317	64	19.6
30	10	0.71	0.03175	64.4	20.3
35	10	0.71	0.03179	69	22.5
40	10.05	0.715	0.03179	67.9	21.4
45	10.05	0.7	0.03179	65	19.4
50	10.05	0.71	0.0332	62	17.3
55	10.05	0.71	0.0341	65	19.5
60	10.05	0.71	0.0353	65	19.5

system and the losses are huge due to friction between the cylinder and piston rings. Although, lubricant is used to minimize these losses but still frictional losses are more prominent in reciprocating compressors. Another component of loss is due to the wire drawing effect at entry and exit through the valve plates which changes the entropy of the system and increases the irreversibility of the system. The exergy destruction in each case with respect to the percentage of refrigerant charged is given in the order as below:

For 25% refrigerant charged:

$$(\mathrm{EX_D})_{\mathrm{comp.}} > (\mathrm{EX_D})_{\mathrm{evap.}} > (\mathrm{EX_D})_{\mathrm{exp.}} > (\mathrm{EX_D})_{\mathrm{cond.}}$$

For 50% refrigerant charged:

$$(\mathrm{EX_D})_{\mathrm{comp.}} > (\mathrm{EX_D})_{\mathrm{exp.}} > (\mathrm{EX_D})_{\mathrm{evap.}} > (\mathrm{EX_D})_{\mathrm{cond.}}$$

For 75% refrigerant charged:

Table 6 Parameters of the VCR system when the unit is 75% charged

Time	$P_{disch.}$/kg/cm^2	$P_{suc.}$/kg/cm^2	m/kg/s	Cond. In/°C	Evap. Inlet/°C
5	15.16	3.2	0.03532	58	22.9
10	15.18	3.4	0.03532	60	24.3
15	15.27	3.5	0.03532	60	24
20	15.28	3.6	0.03532	64.5	23
25	15.28	3.7	0.03532	70.5	22.5
30	15.3	3.7	0.03532	56.1	22.3
35	15.3	3.7	0.03532	79.5	24.5
40	15.3	3.7	0.03532	74	22.6
45	15.3	3.7	0.03532	69.3	22.2
50	15.3	3.7	0.03532	70.1	24.8
55	15.32	3.7	0.03532	74	23
60	15.32	3.7	0.03532	74	24.7
65	15.32	3.7	0.03532	72.4	25.7

$$(\mathrm{EX_D})_{\mathrm{comp.}} > (\mathrm{EX_D})_{\mathrm{cond.}} > (\mathrm{EX_D})_{\mathrm{exp.}} > (\mathrm{EX_D})_{\mathrm{evap.}}$$

For 75% refrigerant charged:

$$(\mathrm{EX_D})_{\mathrm{comp.}} > (\mathrm{EX_D})_{\mathrm{exp.}} > (\mathrm{EX_D})_{\mathrm{evap.}} > (\mathrm{EX_D})_{\mathrm{cond.}}$$

The exergy destruction in the evaporator is found to be the least when the system is 75% charged. This is because of the higher evaporator temperature observed during the test and this reduces the term $Q_e\left|\left(1-\frac{T_o}{T_r}\right)\right|$ in Eq. 6.

The variation of total exergy destruction with respect to the evaporator temperature for four different cases viz. 25, 50, 75, and 100% charging is shown in Figs. 7, 8, 9, and 10, respectively. It is observed that the total exergy destruction is comparable when the system is 75 and 100% charged and it is least when the system is 25% charged.

Table 5 Exergetic performance results of the VCR system when the unit is 50% charged

RE/kJ	$W_{comp.}$/kJ	COP	$EX_{Devap.}$	$EX_{Dcomp.}$	$EX_{Dcond.}$	$EX_{Dexp.}$	EX_{Dtotal}	Exergy efficiency	Exergy efficiency/%
5.117	0.573	8.921	2.885	3.586	2.800	2.932	12.204	0.0691	6.91
5.559	0.750	7.412	3.076	3.967	3.076	3.176	13.297	0.1270	12.7
5.556	0.961	5.780	3.104	4.142	3.110	3.175	13.532	0.1037	10.3
5.513	0.952	5.790	3.079	4.133	3.131	3.175	13.519	0.1067	10.6
5.716	1.026	5.566	3.163	4.320	3.261	3.298	14.044	0.1445	14.4
5.722	1.030	5.555	3.165	4.320	3.280	3.294	14.061	0.1307	13
5.700	1.154	4.939	3.222	4.415	3.287	3.303	14.229	0.0868	8.68
5.729	1.119	5.119	3.199	4.387	3.290	3.294	14.172	0.1089	10.89
5.697	1.052	5.413	3.168	4.311	3.277	3.308	14.066	0.1480	14.8
5.996	1.006	5.960	3.229	4.452	3.416	3.453	14.552	0.2003	20.03
6.156	1.130	5.447	3.409	4.630	3.527	3.542	15.109	0.1578	15.78
6.390	1.169	5.465	3.526	4.790	3.649	3.664	15.629	0.1638	16.38

Table 7 Exergetic performance results of the VCR system when the unit is 75% charged

RE/kJ	$W_{comp.}$/kJ	COP	$EX_{Devap.}$	$EX_{Dcomp.}$	$EX_{Dcond.}$	$EX_{Dexp.}$	EX_{Dtotal}	Exergy efficiency	Exergy efficiency/%
6.012	0.7312	8.222	3.603	3.995	3.945	3.686	15.230	0.1247	12.4
5.772	0.7806	7.393	3.592	4.058	3.945	3.702	15.299	0.1058	10.5
5.962	0.7771	7.672	3.589	4.048	3.963	3.682	15.284	0.1175	11.7
5.916	0.9290	6.368	3.645	4.164	3.974	3.685	15.469	0.0840	8.4
5.941	1.1162	5.322	3.635	4.313	3.995	3.691	15.636	0.1048	10.4
5.832	0.6676	8.735	3.635	3.921	3.935	3.697	15.189	0.1028	10.2
5.853	1.3670	4.281	3.594	4.578	4.030	3.694	15.898	0.0931	9.3
5.839	1.2222	4.777	3.647	4.408	4.009	3.682	15.747	0.0989	9.8
5.863	1.0809	5.424	3.574	4.281	3.988	3.697	15.540	0.1647	16.47
5.818	1.0809	5.382	3.554	4.334	3.984	3.704	15.578	0.1268	12.68
5.888	1.2151	4.845	3.527	4.419	3.998	3.707	15.653	0.1862	18.62
5.878	1.1975	4.908	3.609	4.440	4.002	3.684	15.736	0.0854	8.5
5.807	1.1409	5.089	3.641	4.408	3.991	3.682	15.724	0.0409	4.09

Fig. 3 Variation of exergy efficiency with respect to evaporator temperature when 25% of the total refrigerant is charged

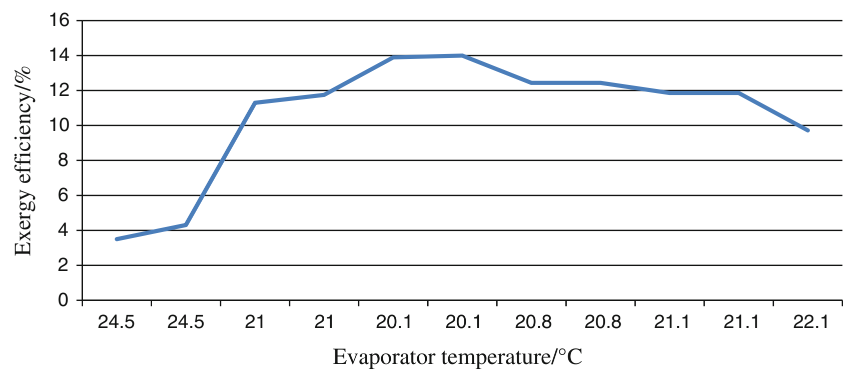


Fig. 4 Variation of exergy efficiency with respect to evaporator temperature when 50% of the total refrigerant is charged

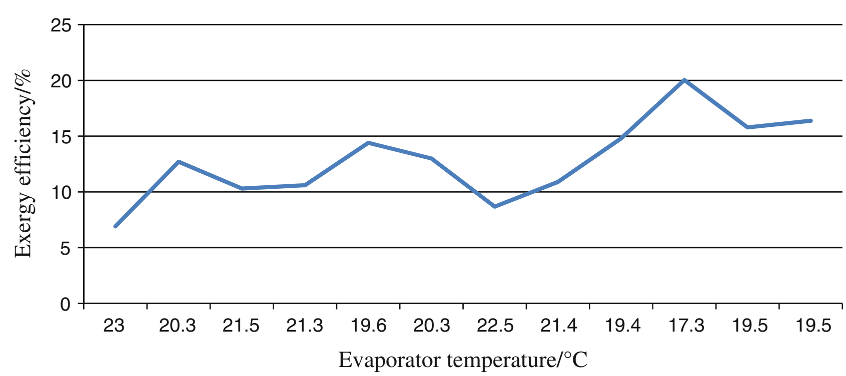
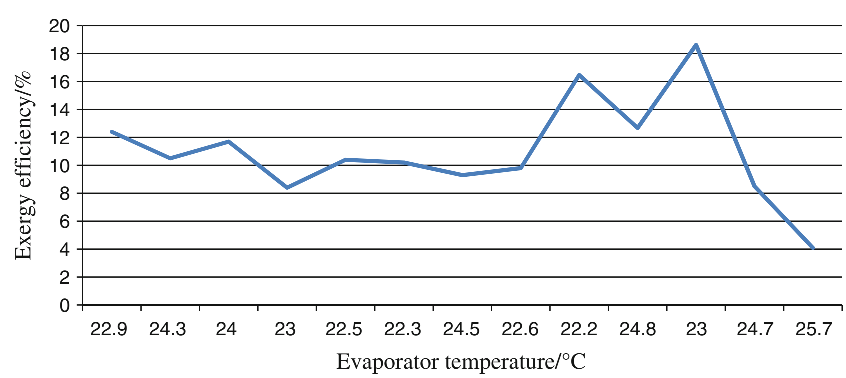


Fig. 5 Variation of exergy efficiency with respect to evaporator temperature when 75% of the total refrigerant is charged



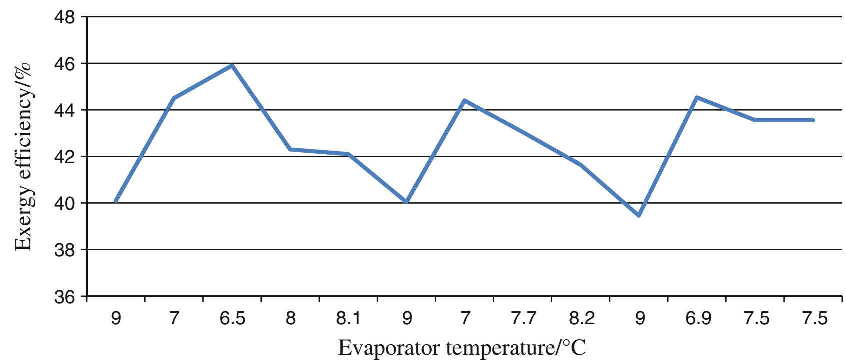


Fig. 6 Variation of exergy efficiency with respect to evaporator temperature when 100% refrigerant is charged

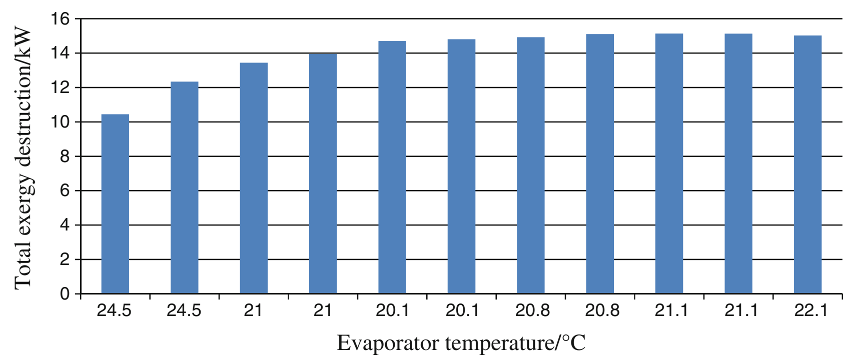


Fig. 7 Variation of total exergy destruction with respect to evaporator temperature when 25% of the total refrigerant is charged

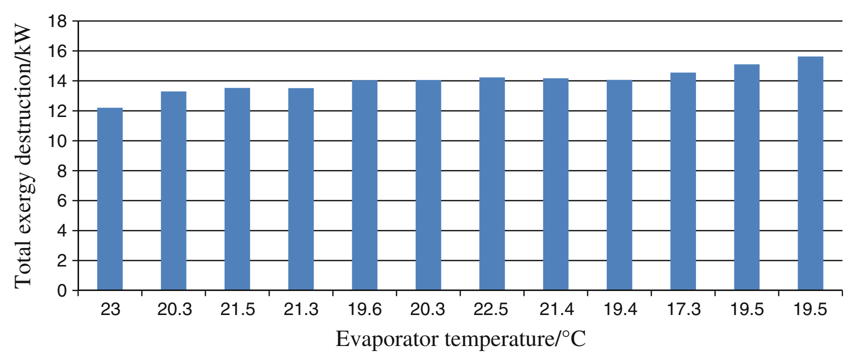


Fig. 8 Variation of total exergy destruction with respect to evaporator temperature when 50% of the total refrigerant is charged

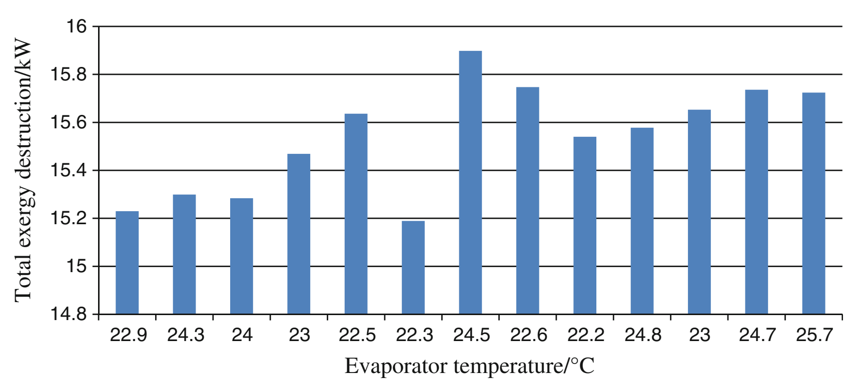


Fig. 9 Variation of total exergy destruction with respect to evaporator temperature when 75% of the total refrigerant is charged

This is because the evaporator temperature is higher due to insufficient amount of charging and is comparable to the reference temperature (Table 8, 9).

Tables 1, 2, 3, 4, 5, 6, and 7 also show that the average COP is highest when the system is 50% charged. This is because the refrigerating effect is higher in this case and

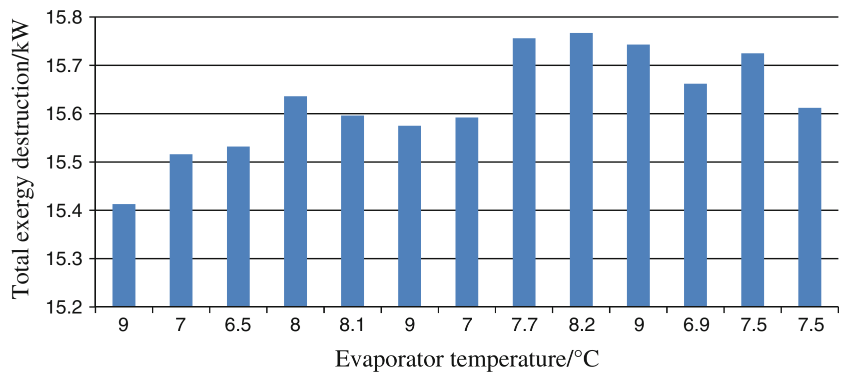


Fig. 10 Variation of total exergy destruction with respect to evaporator temperature when 100% of the total refrigerant is charged

Table 8 Parameters of the VCR system when the unit is 100% charged

Time	$P_{disch.}$/kg/cm^2	$P_{suc.}$/kg/cm^2	m/kg/s	Cond. In/°C	Evap. Inlet/°C
5	18.6	4.1	0.03768	74	9
10	18.8	4.2	0.03768	80	7
15	18.8	4.2	0.03768	81.5	6.5
20	18.4	4.15	0.03768	82	8
25	18.6	4.15	0.03768	82	8.1
30	18.7	4.2	0.03768	81	9
35	18.7	4.2	0.03768	83	7
40	18.5	4.2	0.03768	88	7.7
45	18.5	4.2	0.03768	88	8.2
50	18.8	4.2	0.03768	88	9
55	19	4.2	0.03768	88	6.9
60	18.6	4.2	0.03768	89	7.5
65	18.3	4.15	0.03768	81	7.5

Table 9 Exergetic performance results of the VCR system when the unit is 100% charged

RE/kJ	$W_{comp.}$/kJ	COP	$EX_{Devap.}$	$EX_{Dcomp.}$	$EX_{Dcond.}$	$EX_{Dexp.}$	EX_{Dtotal}	Exergy efficiency	Exergy efficiency/%
6.285	1.299	4.834	3.915	4.171	3.357	3.968	15.413	0.4011	40.1
6.232	1.507	4.135	3.876	4.318	3.327	3.994	15.516	0.4451	44.5
6.262	1.552	4.033	3.855	4.363	3.319	3.993	15.532	0.4593	45.9
6.258	1.582	3.954	3.883	4.431	3.338	3.982	15.636	0.4231	42.3
6.262	1.571	3.985	3.889	4.408	3.323	3.9752	15.596	0.4210	42.1
6.273	1.522	4.121	3.892	4.385	3.327	3.970	15.575	0.4004	40.04
6.217	1.597	3.891	3.849	4.431	3.319	3.9921	15.592	0.4440	44.4
6.262	1.786	3.506	3.890	4.578	3.300	3.986	15.756	0.4305	43.05
6.228	1.767	3.524	3.888	4.589	3.308	3.981	15.767	0.4164	41.64
6.183	1.744	3.544	3.892	4.566	3.297	3.987	15.743	0.3946	39.46
6.202	1.759	3.524	3.847	4.544	3.274	3.996	15.662	0.4453	44.53
6.266	1.804	3.471	3.864	4.600	3.278	3.981	15.725	0.4356	43.56
6.266	1.556	4.026	3.884	4.397	3.345	3.9843	15.612	0.4356	43.56

the compressor consumers less work. It is also observed from the tables that the exergy destruction is almost comparable when the system is 75 and 100% charged while it is least when the system is 25% charged, this is due to the fact that when the system is 25% charged, the evaporator temperature is higher and therefore the term $Q_e\left|\left(1-\frac{T_o}{T_r}\right)\right|$ is significantly low.

Conclusions

Exergy analysis is a technique to present the process and this further aid in reducing the thermodynamic losses occurring in the process. This is an important tool in explaining the various energy flows in a process and in the final run helps to reduce losses occurring in the system. In this experimental study, a window air conditioning system based on vapor compression cycle is modified for experimental analysis. The system comprises of four components, i.e., compressor, a capillary tube (expansion device), a condenser, and an evaporator and is having a cooling capacity of 24K BTU. Based on the experiment testing following conclusions are drawn:

(a) Although the quantity of refrigerant charged do affect the exergy losses but the maximum losses in all the cases are in the compressor. This is attributed to the frictional losses and losses due to wire drawing effect during suction and delivery of the refrigerant. This will augment the study of tribology to exactly study the friction characteristics and also the design aspects needs to be improved to reduce the wire drawing effect to have efficient compressor.

(b) It is observed that the total exergy destruction is comparable when the system is 75 and 100% charged and it is least when the system is 25% charged because the evaporator temperature is very close to the reference temperature.

(c) The average COP is highest when the system is 50% charged and this is because of higher refrigerating effect and reduced compressor work.

(d) The exergy efficiency of the system varies from 3.5 to 45.9% which is mainly due to the variation of evaporator temperature.

(e) The average values of the system exergy efficiency are more when the system is 100% charged. These values show that the overall exergy performance is better when the system is fully charged, but the compressor work is the highest in this case and the COP is also less as compared to other situations. When the actual requirements are less, the system should be operated with variable refrigerant flow so as to achieve optimum balance between the exergy efficiency and energy saving.

Acknowledgements The necessary facilities provided by SMVD University, Katra (J&K) are highly appreciated. One of the authors (SA) also expresses thanks to Mr. Raman Kumar, workshop operator and Mr. Pardeep Kumar, technician for their help in modifying the existing air conditioning unit.

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