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Designing of cascade vapour compression refrigeration systems or ultralow temperature applications using new HFO ultra low GWP refrigerants

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Abstract

Due to flammable nature of R290, R600, R600a and R152a, for domestic applications., the use of ecofriendly HFO refrigerants is well demonstrated due to ultra-low global warming potential and lowest ozone depletion potential. This paper highlights the use of HFO refrigerants are the best alternatives and second alternatives are R245fa. R-152a and R32. Due to flammable nature of these ecofriendly refrigerants, these refrigerants can only be used by using safety measures, otherwise R134a and R410a and R404a are easily available in the markets which has higher GWP. It is found that the thermodynamic performances of R-1225ye(Z), R-1233z(E), and HFO - 1336mzz(Z) better than usingR1234yf. However, R1234ze(Z), and R1234ze(E) and R-1243zf gives good performances up to lowest evaporator temperature -30ºC. as compare to use R134a an R410a, Similarly R1224y(Z) has excellent thermodynamic performances can be used up to lowest evaporator temperature -10ºC as compare to R1234yf. It was found that the best exergetic efficiency was by using R-1225ye(Z) and R-1234ze(Z).

Keywords: : HFO refrigerants, Thermodynamic performances, Energy-Exergy, Vapour Compression Refrigeration System

1. Introduction

An eminent philosopher has defined Science as “TAPASAYA” and Technology as “Bhog” whereas science is pure selfless and open to all search for truth whereas technology is the exploitation of scientific knowledge for selfish purposes and profit motive and hence it tends to be secretive in this competitive world of today with a throat cutting environment. Sustainable Development is a process in which development can be sustained for generations. It also focuses attention on inter-generational fairness in the exploitation of development opportunities while social development is a function of technological advancement, and also the technological advancement, in turn is a function of scientific know how for a streamlined development of the society. Technology is one of the crucial determinants of sustainable development. Technological import through collaborations has been one of the most important sources of technological inputs for Indian conditions. The use of technologies originating in rich countries often ten to create many social, ecological and resource problems in poor countries. The exploitation of the vast natural resources through progressive development of science, engineering and technology that has brought about the vast changes in the civilization and society from the stone age to the present high technology era. In facts, the mad race for industrialization and economic development has resulted in over exploitation of natural resources, leading to a situation where the two worlds of mankind- the biosphere, lithosphere and hydrosphere of his inheritance and the techno sphere of his creation, are out of balance with each other, indeed on a collision path. To facilitate optimal utilization of finite natural resources for ensuring a sustainable benefit steam for better quality of life on the one hand and to simultaneously keep in mind the conservation of natural resources on the other hand, it is essential that the technology conservation process must be made as efficient as possible. Therefore, sustainable economic development depends on the careful choice of technologies and judicious management of resources for productive activities to satisfy the changing human needs without degrading the environment or depleting the natural resources base.

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2. Application of Ecofriendly Refrigerants in the Vapour compression Refrigeration Systems

The efforts under the Montreal protocol to protest the ozone layer, the alternative refrigerants have been proposed as a substitutes for ozone depleting substances. HFCs (Hydrofluoro carbons) PFCs (Perfluro carbons) have zero ODP potential but they are producer of greenhouse gases and are subjected to limitation and reduction commitments under UNFCC (United Nations Framework Convention on Climate change). With the entry into force of Kyoto protocol on 16th February 2005 developed countries have already planning and implementing rational measures intended to contribute towards meeting greenhouse gas reduction targets during the first commitment period of Kyoto protocol (2008-2012). The countries have also started together with developing countries to size up projects that qualify under Kyoto clean development mechanism. As what lies beyond 20-12 all Governments will work together over next few years to decide on future intergovernmental action on the climate change. In this light, it is vital that there should be continuous work on the replacement options for ozone depleting substances in way that serve the aim of the Montreal protocol and UNFCC alike. In the developing countries the conversion of CFCs to alternate is still a major issue. In this paper the first law and second law analysis of various ecofriendly refrigerants have been carried out which will help in deciding about the path to be followed to satisfy Montreal and Kyoto protocol. On the basis of theoretical analysis, it was observed thatR1234yf and R-1234ze refrigerant is the best alternatives for replacing R134a after 2030 due to its low ODP and GWP.

3. Thermodynamic Analysis of Vapour Compression Refrigeration systems

Thermodynamic processes in refrigeration system releases large amount of heat to the environment. Further, heat transfer between the system and the surrounding environment takes place at a finite temperature difference, which is a major source of irreversibility for the cycle and also responsible for the system performance degradation. The losses in the cycle need to be evaluated considering respective individual thermodynamic processes by applying first and second laws. Energy analysis (First Law) is still the most commonly used method in the analysis of thermal systems which only concern with the conservation of energy, and gives no information on how, where, and how much the system performance is degraded. On the other hand, second law is used to describe the quality of energy of materials. The first law optimization results in maximizing the coefficient of performance (COP) thus providing maximum heat removal from minimum power input; while the second law optimization is used for maximizing the exergy efficiency and minimizing entropy generation within the system, hence providing maximum cooling for the smallest distraction of available energy (exergy). The exergy method, known as the second law analysis calculates the exergy loss caused by irreversibility which is an important thermodynamic property that measures the useful work that can be produced by a substance or the amount of work needed to complete a process. Thus exergy analysis is powerful tool in the design, optimization, and performance evaluation of energy systems. The principles and methodologies of exergy analysis are well established [1-4]. An exergy analysis is usually aimed to determine the maximum performance of the system and identify the sites of exergy destruction. Exergy analysis of a complex system can be performed by analyzing the components of the system separately. Identifying the main sites of exergy destruction shows the direction for potential improvements. An important objective of exergy analysis for system that consume work such as refrigeration, liquefaction of gases, and distillation of water is to find the minimum work required for a certain desired results [5]. There are several studies on the exergy analysis of refrigeration system [6, 7]. Bejan [7] showed that the exergetic efficiency decreases as the refrigeration temperature decreases. He offered two simple models to explain this trend. In his model, thermodynamic imperfections are explained largely by the heat transfer irreversibility. The behaviour of two stage compound-cycle with flash intercooling, using refrigerant R22 has been investigated by Nikolaidis and Probert [8] using exergy method. A computational model based on the exergy analysis is presented by Yumrutas et. al [9] for the investigation of the effects of the evaporating and condensing temperatures on the pressure losses, exergy losses, second law of efficiency, and the COP of a vapour compression cycle. In the present research work, exergy analysis is performed on the operating data of vapour compression refrigeration cycle of an industrial ice plant. The expressions for the exergy losses (lost works) for the individual processes of the cycle as well as the coefficient of performance (COP) and second law efficiency for the entire cycle has been obtained. Effect of variation of condensing and evaporating temperatures on exergy losses, second law efficiency and COP has been investigated. The concept of structure coefficient (coefficient of structure bond) is used to explain the relation between irreversibilities of system & its component. The main components of a refrigeration are compressor, expansion valve, condenser and evaporator (brine chiller connected with ice box). Ice tank is made of mild steel plates and having heavy insulation of granulated corks around 300 mm in thickness. The tank is filled with recirculated brine in which cans filled with water are immersed. The tank is provided with piping for blowing air into ice cans to create agitation. In the present analysis, single stage vapour compression system has been used for ice production. Reverse osmosis water is being used in the system for production of ice cubes. The corresponding temperature versus entropy diagram for this system. At the starting point, the circulating refrigerant enters the compressor as a saturated vapour. From first point to second point, the refrigerant vapour is isentropically compressed and exits the compressor as a superheated vapour. From second point to point third', the superheated vapour travels through part of the condenser which removes the superheat by cooling the vapour between third point ' and fourth point, the vapour travels through the remainder of the condenser and is condensed into a saturated liquid. The condensation process occurs essentially at constant pressure. Between fourth points and fifth point, the saturated liquid refrigerant passes through the expansion valve and undergoes an abrupt decrease of pressure. This process results in the adiabatic flash evaporation and causes auto refrigeration of a portion of the liquid refrigerant (typically, less than half of the liquid flashes). The adiabatic flash evaporation process is isenthalpic (i.e., occurs at constant enthalpy) in nature. Between fifth points and first point, the cold and partially vaporized refrigerant travels through the coil or tubes in the evaporator where it is totally vaporized by the warm water (from the space being refrigerated) that circulates across the coil or tubes in the evaporator. The evaporator operates at essentially constant pressure. The resulting saturated refrigerant vapor returns to the compressor inlet at the first point to complete the thermodynamic cycle.

Modelling of vapour compression refrigeration system using ecofriendly refrigerants The major disadvantage of hydrocarbon refrigerants is their inflammability therefore system must be designed by consideration of flame proof regulations however with open compressors additional safety devices may be required. For practically mass flow rate of refrigerant may be low as 55% to 60% approximately as compared to R-22 and internal heat exchanger between the suction and liquid line is improved the refrigeration capacity along with COP. Due to good solubility with mineral oils, it may be necessary to use an oil it lower mixing characteristics or increasing viscosity for higher suction pressure, the use of internal heat exchanger is advantageous because it leads to higher oil temperatures to lower solubility improves its viscosity. For the thermodynamic analysis of the industrial ice plant working on vapour compression system, the principles of mass conservation, first and second laws of thermodynamics are applied to each component of the system.

Each component can be treated as control volume with inlet and outlet streams, heat transfer and work interactions. For the system, the mass conservation is governed by following equation:

$$\sum mi - \sum mo = 0 \tag{1}$$

Where m is the mass flow rate and suffix i and o represents inlet and outlet of the component. The first law of thermodynamics yields the energy balance of each component of the system as follows:

$$\sum(mh)i - \sum(mh)o + [\sum Qi - \sum Qo] + W = 0 \tag{2}$$

The overall energy balance of the system requires that the sum of the evaporator, condenser and compressor must be zero. The vapour compression system is assumed to be in steady state condition and further if the pump work for brine solution circulation and the environmental heat losses are neglected, the energy balance for the entire system can be written as

$$Q_{cond} = Q_{eva} + W_{comp} \tag{3}$$

The energy balance equations of various components of a vapour compression system are given below: Mass flow rate of circulated refrigerant can be calculated as

$$m_{Ref} = Q_{eva} / (h_1 - h_4) \tag{4}$$

For the compressor (by convention, the work done by compressor is presumed to be negative);

$$W_{comp} = - m_{Re} (h_2 - h_1)/ \eta_{mech}\, \eta_{elect} \tag{5}$$

Where η_{mech} and η_{elect} are mechanical and electric motor efficiencies respectively. Heat transfer rate in the condenser (zone B) is given by

$$Q_{cond} = m_{Ref} (h_3 - h_2) \tag{6}$$

Coefficient of performance of refrigeration is

$$COP = Q_{eva} / W_{comp} \tag{7}$$

For state point 4 dryness fraction and specific entropy can be represented as:

$$x_4 = (h_3 - h_f) / (h_1 - h_f) \tag{8}$$

Where h_f is the enthalpy of saturated liquid refrigerant at evaporator pressure. Specific entropy is:

$$s_4 = s_f + x_4 (s_1 - s_f) \tag{9}$$

Where sf is the entropy of saturated liquid refrigerant at evaporator pressure. Second law analysis is a relatively new concept, which has been used for understanding the irreversible nature of real thermal processes and defining the maximum available energy. The second law analysis is based on the concept of exergy, which can be defined as a measure of work potential or quality of different forms of available energy relative to the environmental conditions. In other words, exergy can be defined as the maximum theoretical work derivable by the interaction for an energy resource with the environment. Exergy analysis is applied to a system describes loses both in the components of the system and for the system as a whole. With the help of exergy analysis, the magnitude of these losses or irreversibility's and their order of importance can be understood respectively. With the use of irreversibility, which is a measure of process imperfection, the optimum operating conditions can be easily determined. It is possible to say that exergy analysis throws an insight to indicate the possibilities of thermodynamic improvement for the process under consideration. The formulation for exergy analysis is described below: The difference of the flow availability of a stream and that of the same stream at its restricted dead state is called flow exergy (ε) and by ignoring chemical exergy terms, flow exergy (ε) is given by

$$\varepsilon = (h - T_0 s) + V^2/2 + gZ - (h_0 - T_0 s_0) \quad (10)$$

Ignoring the potential and kinematic energy terms, Eq. (11) becomes

$$\varepsilon = (h - T_0 s) - (h_0 - T_0 s_0) \quad (11)$$

The exergy balance equation is given by

$$Ew = \sum EQ + \sum(m\varepsilon)\ i - \sum(m\varepsilon)\ o + T_0 S_{gen} \quad (12)$$

In equation (12) the term T_0S_{gen} is defined as the irreversibility (I) and can be written as:

$$I = T_0 S_{gen} \quad (13)$$

The above exergy analysis formulation has been performed on each component of the vapour compression and corresponding irreversibility of each component is calculated.

By carrying out an exergy-rate balance for the compressor, the irreversibility rate:

$$I_{comp} = W + E_1 - E_2 \quad (14)$$

The exergy-rate balance for the evaporator:

$$I_{Evaporator} = E_4 - E_1 - E_{Evaporator} \quad (15)$$

where, for the evaporator

$$E_{Evaporator} = Q_{Evaporator}\ (T_0 - T_{Eva})/T_{Eva} \quad (16)$$

The exergy rate balance for the condenser, is given by

$$I_{Cond} = E_2 - E_3 \quad (17)$$

The exergy-rate balance in the throttling valve, is given by

$$I_{throttling\ valve} = E_3 - E_4 \quad (18)$$

Efficiency defect (δk) of kth component of the system may be expressed as fractions of input which are lost through irreversibility

$$\delta k = I_k/W \quad (19)$$

Where Ik is the irreversibility rate of the kth component of the system under consideration. The total irreversiblitities of the system components is expressed as

$$I_{total} = I_{comp} + I_{Cond} + I_{Eva} + I_{throttling\ valve} \quad (20)$$

The relative irreversibility of the kth component of plant is
$= Ik/I_{total}$ (21)

Structural coefficients are used in the study of the system structure, optimization of plant components and product pricing in multi-product plants. The change of local irreversibility rates and exergy fluxes in relation to the overall plant's irreversibility rate is effectively expressed by the coefficient of structural bonds (CSB) which is defined by

$$\sigma_{k,\ i} = [\partial It/\partial xi]/[\partial Ik/\partial xi] \quad (21)$$

Where xi is the ith parameter of the system which produces the changes in kth component. The effect of a change in xi on the system would be to alter the rate of exergy input while leaving the output constant. This acceptance confirms to the usual practice of specifying a plant in terms of its output rather than its input. From the exergy balance of the system

$$E_{IN} = E_{OUT} + It \quad (22)$$

But E_{OUT} = constant, thus $\Delta E_{IN} = \Delta It$ As seen from above equation changes in the irreversibility of the system are equivalent to changes in the exergy input. In general, the ratio of the rates of exergy output to exergy input is less than unity. This ratio denotes the degree of thermodynamic perfection of the process and is called the rational efficiency($\eta_{Rational}$).

$$\eta_{Rational} = E_{OUT}/E_{IN} \quad (23)$$

$$\text{Plant rational efficiency } \eta_{Rational,\ plant} = 1 - \sum \delta k \quad (24)$$

4. Results and Discussion

The computation modeling of vapor compression refrigeration systems was carried out with the help of engineering equation solver of Hon'ble Dr. S.A. Klein (2002) for first and second law analysis in terms of energetic analysis i.e. COP (First law analysis) and exegetic analysis in terms of exergetic efficiency, exergy destruction ratio (EDR) and percentage exergetic destruction in each component (second law analysis). In this analysis we assumed negligible pressure losses and heat losses. The comparative performance of 3.5167 KW window air conditioner is evaluated for condenser temperature varying between 303K to 328K with increment of 5 and evaporator temperature is varying from 243K to 283 K with increment of 5. The energy and exergy change in vapour compression refrigeration cycle have been calculated for various ecofriendly refrigerants such as R-1234ze(Z), R-1234ze(E), R-1234yf, R-12343zf, R1224yd(E), R-1225ye(Z) for environmental temperature of 299K. The variation of first law efficiency (COP)and second law efficiency in terms of exergetic are shown in Table-1 to Table 8 respectively. As evaporator temperature increases the first law efficiency increases while exergetic efficiency decreases. The variation of condenser temperature with thermodynamic performances of vapour compression refrigeration systems using HFO 1336mzz(z) are shown in Table-

9 respectively. It was found that by increasing condenser temperature, the thermodynamic energy an exergy performances decreases. The effect of various ecofriendly HFO refrigerants for condenser temperature of 323 (K) and Evaporator temperature of 243 K is also shown in table 10. respectively.

Table 1(a) : Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using HFO-1336 mzz(Z) at condenser temperature =50^{o}C

Evaporator Temperature (^{o}C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"
-30	1.547	1.805	0.3565	2.273	0.8104
-25	1.748	1.782	0.3595	2.012	0.7232
-23	1.837	1.777	0.360	1.914	0.6893
-22(optimum)	1.883	1.777	0.3602	1.867	0.6725
-21(optimum)	1.931	1.777	0.3602	1.821	0.6559
-20	1.547	1.777	0.3600	1.776	0.6394
-15	1.980	1.796	0.3576	1.563	0.5589
-10	2.260	1.845	0.3515	1.369	0.4814
-5	2.568	1.934	0.3408	1.194	0.4068
0	2.949	2.086	0.3240	1.034	0.3349
5	3.402	2.341	0.2993	0.8875	0.2657
10	4.665	2.791	0.2638	0.7539	0.1988
15	5.572	3.699	0.2128	0.6311	0.1343
20	6.785	6.198	0.1389	0.5183	0.0720
25	8.486	4.12	0.02848	0.4144	0.0118

Table 1(b) : Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using HFO-1336 mzz(Z) at condenser temperature =50^{o}C

Evaporator Temperature (^{o}C)	Total Exergy Destruction (%)	Compressor Exergy Destruction (%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-30	64.35	18.51	19.04	26.92	35.65	2.273	0.8104
-25	64.05	18.51	20.58	26.1	35.95	2.012	0.7232
-23	63.98	18.51	21.2	24.42	36.0	1.914	0.6893
-22	63.98	18.51	21.55	24.07	36.02	1.867	0.6725
-21	64.0	18.51	21.91	23.71	36.02	1.821	0.6559
-20	64.0	18.51	22.27	23.36	36.00	1.776	0.6394
-15	64.24	18.51	24.29	21.59	35.76	1.563	0.5589
-10	64.85	18.51	26.67	19.84	35.15	1.369	0.4814
-5	65.92	18.51	29.49	18.11	34.08	1.194	0.4068
0	67.6	18.51	32.90	16.4	32.40	1.034	0.3349
5	70.07	18.51	37.09	14.71	29.93	0.8875	0.2657
10	73.62	18.51	42.34	13.04	26.38	0.7539	0.1988
15	78.72	18.51	49.12	11.40	21.28	0.6311	0.1343
20	86.11	18.51	58.18	9.778	13.89	0.5183	0.0720
25	97.15	18.51	70.90	8.185	02.85	0.4144	0.0118

Table 2(a): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1225ye(Z)using condenser temperature =50^{o}C

Evaporator Temperature (^{o}C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work)	Exergy-product
-30	1.427	2.041	0.3326	2.464	0.8104
-25	1.614	2.013	0.3326	2.179	0.7232
-23	1.696	2.006	0.3326	2.073	0.6893
-22	1.79	2.006	0.3326	2.022	0.6725
-21	1.784	2.006	0.3327	1.972	0.6559
-20	1.830	2.006	0.3326	1.922	0.6394
-19	1.877	2.006	0.3326	1.922	0.6394
-17	1.976	2.013	0.3319	1.78	0.5907
-15	2.081	2.023	0.3308	1.690	0.5589

-10	2.378	2.072	0.3255	1.479	0.4814
-5	2.732	2.164	0.3160	1.287	0.4068
0	3.160	2.323	0.3009	1.113	0.3349
5	3.689	2.591	0.2785	0.9539	0.2657
10	4.349	3.067	0.2459	0.8086	0.1988

Table 2(b): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1225ye(Z) using condenser temperature =50ºC

Evaporator Temperature (ºC)	Total Exergy Destruction (%)	Compressor Exergy Destruction (%)	Condenser Exergy Destruction (%)	Throttle valve Exergy Destruction(%)	Exergetic - efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-30	67.11	18.19	18.45	30.58	0.33.26	2.464	0.8104
-25	66.82	18.26	19.78	28.89	0.33.26	2.179	0.7232
-23	66.75	18.19	20.37	26.22	0.33.26	2.073	0.6893
-22	66.74	18.19	20.69	27.88	0.3326	2.022	0.6725
-21	66.73	18.19	21.01	27.55	0.3327	1.972	0.6559
-20	66.74	18.31	21.34	27.21	0.3326	1.922	0.6394
-19	66.75	18.19	21.68	26.88	0.3326	1.922	0.6394
-17	66.81	18.19	22.41	26.02	0.3319	1.78	0.5907
-15	66.92	18.36	23.18	25.53	0.3308	1.690	0.5589
-10	67.45	18.39	26.36	23.85	0.3255	1.479	0.4814
-5	68.40	18.42	27.98	22.17	0.3160	1.287	0.4068
0	69.91	18.44	31.15	20.50	0.3009	1.113	0.3349
5	72.15	18.46	35.07	18.84	0.2785	0.9539	0.2657
10	75.41	18.47	40.01	17.17	0.2459	0.8086	0.1988

Table 3(a): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1243zf

Evaporator Temperature (ºC)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-30	1.472	1.948	0.3392	2.389	0.8104	0.03337
-25	1.659	1.931	0.3411	2.12	0.7232	0.03273
-24	1.70	1.930	0.3413	2.069	0.7062	0.03212
-23	1.741	1.930	0.3413	2.019	0.6893	0.03192
-22	1.785	1.930	0.3413	1.971	0.6725	0.03122
-21	1.829	1.931	0.3412	1.923	0.6559	0.03153
-20	1.875	1.933	0.3409	1.876	0.6394	0.03133
-15	2.127	1.959	0.3380	1.653	0.5589	0.03041
-10	2.424	2.014	0.3317	1.451	0.4814	0.02954
-5	2.777	2.113	0.3212	1.266	0.4068	0.02872
0	3.204	2.277	0.3052	1.098	0.3349	0.02795
5	3.73	2.549	0.2817	0.9429	0.2657	0.02722
10	4.39	3.029	0.2482	0.801	0.1988	0.02654

Table 3(b) : Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1243zf

Evaporator Temperature (ºC)	Total Exergy Destruction (%)	Compressor Exergy Destruction(%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency	Exergy_fuel kW"	Exergy-product kW"
-30	66.08	17.93	19.10	29.17	0.3392	2.389	0.8104
-27	65.94	17.98	19.85	28.23	0.3411	2.12	0.7232
-25	65.89	18.01	20.39	27.60	0.3413	2.069	0.7062
-24	65.87	18.03	20.68	27.29	0.3413	2.019	0.6893
-23	65.87	18.05	20.97	26.98	0.3413	1.971	0.6725
-22	65.87	18.06	21.28	26.66	0.3412	1.923	0.6559
-21	65.88	18.08	21.59	26.35	0.3409	1.876	0.6394
-20	65.91	18.09	21.92	27.29	0.3380	1.653	0.5589
-15	66.2	18.15	23.72	24.47	0.3317	1.451	0.4814
-10	66.83	18.21	25.88	22.90	0.3212	1.266	0.4068

-5	67.88	18.26	28.46	21.33	0.3052	1.098	0.3349
0	69.48	18.30	31.61	19.77	0.2817	0.9429	0.2657
5	71.83	18.34	35.5	18.21	0.2482	0.801	0.1988
10	75.18	18.37	40.4	16.65	0.3392	2.389	0.8104

Table 4(a): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R-1233zd (E) at condenser temperature =50°C

Evaporator Temperature (°C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-30	1.472	1.948	0.3392	2.389	0.8104	0.03337
-27	1.581	1.936	0.3406	2.225	0.7577	0.03273
-25	1.659	1.931	0.3411	2.11	0.7232	0.03232
-24	1.70	1.930	0.3413	2.019	0.6893	0.03212
-23	1.741	1.930	0.3413	2.069	0.7062	0.03192
-22(optimum)	1.785	1.930	0.3413	1.971	0.6725	0.03122
-21(optimum)	1.829	1.931	0.3412	1.923	0.6559	0.03153
-20	1.875	1.933	0.3409	1.876	0.6394	0.03133
-15	2.127	1.957	0.3380	1.653	0.5589	0.030414
-10	2.424	2.014	0.3317	1.451	0.4814	0.02954
-5	2.777	2.113	0.3212	1.266	0.4068	0.02872
0	3.204	2.277	0.3052	1.094	0.3349	0.02795
5	3.73	2.549	0.2817	0.9429	0.2657	0.02722
10	4.39	3.029	0.2482	0.8010	0.1988	0.02654

Table 4(b): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R-1233 zd (E) at condenser temperature =50°C

Evaporator Temperature (°C)	Total Exergy Destruction (%)	Compressor Exergy Destruction (%)	Condenser Exergy Destruction (%)	Throttle valve Exergy Destruction (%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-30	66.08	17.93	19.10	29.17	33.92	2.389	0.8104
-25	64.05	18.51	20.58	26.1	34.06	2.225	0.7577
-23	63.98	18.51	21.2	24.42	34.11	2.11	0.7232
-22	63.98	18.51	21.55	24.07	34.13	2.019	0.6893
-21	64.0	18.51	21.91	23.71	34.13	2.069	0.7062
-20	65.91	18.09	21.92	26.04	34.13	1.971	0.6725
-15	66.2	18.15	23.72	24.47	34.12	1.923	0.6559
-10	66.8	18.21	25.88	22.90	34.09	1.876	0.6394
-5	67.88	18.26	28.46	21.33	33.80	1.653	0.5589
0	69.48	18.30	31.61	19.77	33.17	1.451	0.4814
5	71.83	18.4	35.5	18.21	32.12	1.266	0.4068
10	75.18	18.37	40.4	16.65	30.52	1.094	0.3349

Table 5(a): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1234yf

Evaporator Temperature (°C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-40	0.9988	2.536	0.2829	3.521	0.9961	0.05271
-35	1.137	2.431	0.2915	3.092	0.9013	0.05016
-30	1.295	2.35	0.2985	2.715	0.8104	0.04577
-25	1.477	2.292	0.3038	2.381	0.7232	0.04786
-20	1.688	2.259	0.3069	2.084	0.6394	0.04386
-19	1.734	2.256	0.3072	2.028	0.6230	0.04350
-18	1.781	2.263	0.3074	1.974	0.6068	0.04315
-17	1.830	2.253	0.3075	1.921	0.5907	0.04280
-16	1.881	2.253	0.3074	1.870	0.5748	0.04246
-15	1.933	2.255	0.3072	1.819	0.5589	0.04212
-10	2.223	2.286	0.3043	1.582	0.4814	0.04052
-5	2.586	2.366	0.2971	1.369	0.4068	0.03908
0	2.986	2.516	0.2844	1.178	0.3349	0.03768

5	3.501	2.782	0.2644	1.005	0.2657	0.03642
10	4.148	3.264	0.2345	0.8478	0.1988	0.03526

Table 5(b): Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1234yf

Evaporator Temperature (^oC)	Total Exergy Destruction (%)	Compressor Exergy Destruction(%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-40	71.71	18.28	15.16	38.35	28.29	3.521	0.9961
-35	70.86	18.33	16.14	36.46	29.15	3.092	0.9013
-30	70.15	18.37	17.29	34..58	29.85	2.715	0.8104
-25	69.62	18.41	18.61	32.71	30.38	2.381	0.7232
-20	69.31	18.43	20.16	30.84	30.69	2.084	0.6394
-19	69.28	18.44	20.50	30.47	30.72	2.028	0.6230
-18	69.26	18.44	20.85	30.09	30.74	1.974	0.6068
-17	69.25	18.44	21.22	29.72	30.75	1.921	0.5907
-16	69.26	18.45	21.59	29.35	30.74	1.870	0.5748
-15	69.28	18.45	21.98	28.98	30.72	1.819	0.5589
-10	69.57	18.48	24.13	27.12	30.43	1.582	0.4814
-5	70.29	18.48	26.70	25.28	29.71	1.369	0.4068
0	71.56	18.48	29.82	23.44	28.44	1.178	0.3349
5	73.56	18.49	33.66	21.62	26.44	1.005	0.2657
10	76.55	18.49	38.49	19.8	23.45	0.8478	0.1988

Table 6(a): Effect of Evaporator Temperature on the variation of thermal performance parameters of vapour compression refrigeration system using R-1234ze(Z)

Evaporator Temperature (^oC)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-32	1.640	1.534	0.3946	2.145	0.8463	0.02610
-31	1.676	1.534	0.3947	2.099	0.8283	0.02597
-30	1.713	1.534	0.3947	2.053	0.8104	0.02584
-27	1.83	1.537	0.3942	1.922	0.7577	0.02546
-25	1.913	1.541	0.3935	1.838	0.7232	0.02521
-22	2.048	1.553	0.3917	1.717	0.6725	0.02485
-20	2.145	1.537	0.3942	1.640	0.7577	0.02461
-18	2.247	1.579	0.3877	1.565	0.6068	0.02480
-15	2.413	1.608	0.3835	1.457	0.7577	0.02449
-10	2.728	1.678	0.3735	1.289	0.5589	0.02349
-5	3.103	1.786	0.3589	1.133	0.4068	0.02297
0	3.554	1.954	0.3385	0.9894	0.3349	0.02154
5	4.109	2.222	0.3104	0.8559	0.2657	0.02199
10	4.804	2.682	0.2716	0.7320	0.1988	0.02154

Table 6(b): Effect of evaporator temperature on the thermal performance parameters of vapour compression refrigeration system using R1234ze(Z)

Evaporator Temperature (^oC)	Total Exergy Destruction (%)	Compressor Exergy Destruction(%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-32	60.54	17.47	20.96	22.23	0.3946	2.145	0.8463
-31	60.53	17.50	21.19	21.97	0.3947	2.099	0.8283
-30	60.53	17.53	17.53	21.43	0.3947	2.053	0.8104
-27	60.58	17.61	22.20	20.91	0.3942	1.922	0.7577
-25	60.65	17.66	22.75	20.38	0.3935	1.838	0.7232
-22	60.83	17.73	23.06	19.59	0.3917	1.717	0.6725
-20	61.01	17.78	24.32	19.06	0.3942	1.640	0.7577
-18	61.23	17.82	25.03	18.53	0.3877	1.565	0.6068
-15	61.65	17.88	26.19	17.74	0.3835	1.457	0.7577
-10	62.65	17.98	28.43	16.42	0.3735	1.289	0.5589
-5	64.11	18.06	31..13	15.11	0.3589	1.133	0.4068

0	66.15	18.14	34.42	13.81	0.3385	0.9894	0.3349
5	68.96	18.21	38.49	12.51	0.3104	0.8559	0.2657
10	72.84	18.26	43.62	11.22	0.2716	0.7320	0.1988

Table 7(a) : Effect of Evaporator Temperature on the variation of thermal performance parameters of vapour compression refrigeration system using R-1234ze(E)

Evaporator Temperature (°C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-31	1.412	2.006	0.3327	2.490	0.8283	0.0383
-30	1.448	1.997	0.3336	2.429	0.8104	0.0380
-28	1.521	1.983	0.3353	2.312	0.7751	0.0374
-25	1.639	1.967	0.3371	2.145	0.7232	0.0366
-23	1.724	1.96	0.3278	2.04	0.689	0.03606
-22	1.768	1.958	0.3381	1.989	0.6725	0.0358
-21(optimum)	1.813	1.957	0.3381	1.939	0.6559	0.03554
-20(optimum)	1.86	1.957	0.3382	1.891	0.6394	0.03529
-19	1.908	1.958	0.3381	1.843	0.6230	0.03504
-18	1.958	1.960	0.3379	1.7960	0.6068	0.03480
-15	2.117	2.019	0.3365	1.661	0.5589	0.03408
-10	2.420	2.019	0.3313	1.453	0.5589	0.03296
-5	2.780	2.11	0.3216	1.265	0.4068	0.03190
0	3.215	2.266	0.3062	1.094	0.3349	0.03093
5	3.750	2.531	0.2832	0.9379	0.2657	0.03002
10	4.421	3.001	0.2499	0.7955	0.1980	0.02917

Table 7(b) : Effect of Evaporator Temperature on the thermal performance parameters of vapour compression refrigeration system using R1234ze(Z)

Evaporator Temperature (°C)	Total Exergy Destruction (%)	Compressor Exergy Destruction (%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-31	66.73	18.27	18.27	30.57	0.3327	2.490	0.8283
-30	66.64	18.28	18.52	30.22	0.3336	2.429	0.8104
-28	66.47	18.30	19.05	29.51	0.3353	2.312	0.7751
-25	66.29	18.33	19.90	26.46	0.3371	2.145	0.7232
-23	66.22	18.35	20.51	27.76	0.3278	2.04	0.689
-22	66.19	18.36	20.84	27.41	0.3381	1.989	0.6725
-21	66.18	18.37	21.17	27.06	0.3381	1.939	0.6559
-20	66.18	18.38	21.51	26.71	0.3382	1.891	0.6394
-19	66.19	18.38	21.86	26.36	0.3381	1.843	0.6230
-18	66.21	18.39	22.23	26.02	0.3379	1.7960	0.6068
-15	66.35	18.41	23.40	24.98	0.3365	1.661	0.5589
-10	66.87	18.43	25.64	23.25	0.3313	1.453	0.5589
-5	67.84	18.45	28.31	21.54	0.3216	1.265	0.4068
0	69.38	18.47	31.55	19.85	0.3062	1.094	0.3349
5	71.68	18.48	35.53	18.17	0.2832	0.9379	0.2657
10	75.01	18.49	40.54	16.50	0.2499	0.7955	0.1980

Table 8(a) : Effect of Evaporator Temperature on thermal performance parameters of vapour compression refrigeration system using R-1224yd(Z)

Evaporator Temperature (°C)	COP	EDR	Second law Efficiency	Exergy_input (Compressor Work) kW"	Exergy-product kW"	Mass flow rate
-10	2.622	1.786	0.3589	1.341	0.4814	0.03140
-8	2.764	1.820	0.3546	1.272	0.4512	0.03105
-5	2.996	1.885	0.3466	1.174	0.4068	0.03052
0	3.448	2.046	0.3283	1.02	0.3349	0.02969
5	4.002	2.308	0.3023	0.8787	0.2657	0.02889
10	4.698	2.765	0.2656	0.7485	0.1988	0.02814

Table 8(b) : Effect of Evaporator Temperature on thermal performance parameters of vapour compression refrigeration system using R-1224yd(Z)

Evaporator Temperature (°C)	Total Exergy Destruction (%)	Compressor Exergy Destruction(%)	Condenser Exergy Destruction(%)	Throttle valve Exergy Destruction(%)	Exergetic efficiency (%)	Exergy_fuel kW"	Exergy-product kW"
-10	64.11	18.45	27.12	18.7	35.89	1.341	0.4814
-8	64.54	18.46	28.17	18.08	35.46	1.272	0.4512
-5	65.34	18.47	29.89	17.16	34.66	1.174	0.4068
0	67.17	18.49	33.25	15.63	32.83	1.02	0.3349
5	69.77	18.5	37.39	14.12	30.23	0.8787	0.2657
10	73.44	18.5	42.58	12.62	26.56	0.7485	0.1988

Table 9: Effect of variation of Condenser temperature on the thermal performance parameters of vapour compression refrigeration system using HFO-1336 mzz(Z) ecofriendly refrigerant ($T_{\_Eva}$=-30°C) Exergy-product 0.8104

Condenser temperature (°C)	COP	EDR	Exergetic Rational Efficiency	Exergy_ input Compressor Work)	Mass flow rate	Condenser heat	Total Exergy Destruction (%)	Compressor Exergy Destruction (%)	Condenser Exergy Destruction (%)	Throttle valve Exergy Destruction (%)
55	1.382	2.14	0.3185	2.545	0.03912	6.012	68.15	18.22	21.16	28.88
50	1.547	1.805	0.3565	2.273	0.03651	5.79	64.35	18.51	19.04	26.92
45	1.731	1.507	0.3988	2.032	0.03424	5.549	60.12	18.79	16.44	25.02
40	1.937	1.241	0.4463	1.816	0.03225	5.333	55.37	19.08	13.28	23.15
35	2.17	0.9995	0.500	1.62	0.03049	5.137	49.99	19.38	9.443	21.33
30	2.439	0.7793	0.562	1.442	0.02892	4.959	43.8	19.68	4.757	19.55

Table 10: Effect of ecofriendly refrigerants on the variation of thermal performance parameters of vapour compression refrigeration system using HFO refrigerants

Refrigerant	R1234ze(Z)	R1234ze(E)	R-1243zf	HFO-1336 mzz(Z)	R-1233zd(E)	R-1225 ye(Z)	R1234yf	R134a
COP_ACtual	1.713	1.448	1.472	1.547	1.669	1.427	1.307	1.508
COP_ Carnot	4.339	4.339	4.339	4.339	4.339	4.339	4.339	4.339
Exergetic Efficiency	0.3947	0.3336	0.3392	0.3565	0.3847	0.3289	0.3475	0.3475
Exergy Destruction Ratio (EDR)	1.534	1.998	1.948	1.805	1.534	1.5994	2.295	1.878
Second law Efficiency	0.5636	0.4766	0.4845	0.5053	0.5495	0.4698	0.4336	0.4964
Exergy input "kW"	2.053	2.429	2.389	2.273	2.107	2.464	2.670	2.332
Exergy product "kW"	0.8104	0.8104	0.8104	0.8104	0.8104	0.8104	0.8104	0.8104
compressor Exergy Destruction	0.3599	0.4441	0.4284	0.4206	0.3785	0.4483	0.4899	0.4081
condenser Exergy Destruction	0.4401	0.4499	0.4564	0.4327	0.4317	0.4547	0.4906	0.4689
Valve Exergy Destruction	0.4457	0.7340	0.6969	0.6120	0.4889	0.7535	0.8902	0.6479
Total Exergy Destruction	1.243	1.619	1.579	1.463	1.296	1.654	1.859	1.522
% compressor Exergy Destruction	17.53	18.23	17.93	18.51	17.96	18.19	18.30	17.5
% condenser Exergy Destruction	21.43	18.52	19.1	19.04	20.49	18.46	18.37	20.1
% Valve Exergy Destruction	20.7	29.22	28.17	25.92	22.21	29.54	32.34	26.78
% Evaporator Exergy Destruction	1.1301	1.3801	1.114	1.1187	1.1187	1.1187	1.1187	1.1244
% Total Exergy Destruction	60.53	66.44	66.08	64.35	61.53	67.11	69.65	65.25
% Exergetic Efficiency	39.47	33.36	33.92	35.65	38.47	32.89	30.35	34.75
Condenser Heat Rejected "kW"	5.57	5.946	5.906	5.79	5.624	5.984	6.187	5.849
Cooling load "kW"	3.5167	3.5167	3.5167	3.5167	3.5167	3.5167	3.5167	3.5167

The optimum thermodynamic exergetic performance of vapour compression refrigeration system using ecofriendly HFO refrigerants is shown in Table-11. It was found that the best exergetic efficiency was by using R-1225ye(Z) and R-1234ze(Z)

Table 11: Ecofriendly HFO refrigerants for optimum evaporator for optimum exergetic efficiency of vapour compression refrigeration system

Ecofriendly HFO Refrigerants	T_{EVA} (ºC)	Exergetic Efficiency (%)
R-1234ze(Z)	-30, -31	39.47
R-1234ze(E)	-20, -21	33.82
HFO-1336 mzz(Z)	-21, -22	36.02
R-1243zf	-22, -23,-24,	34.13
R1234yf	-17	30.75
R-1233zd(E)	-27,-28	38.5
R-1225ye(Z)	-21,	39.47

5. Conclusions

The analysis of the single stage vapour compression refrigeration system performance by the exergy method demonstrates how effective this method is for analyzing behaviour. Employing the concepts of efficiency defect and exergetic efficiency has enabled the proportions of input lost through irreversibility's, in various refrigeration systems sub systems, to be evaluated easily. Using the technique of the coefficient of structural bond has demonstrated that a change in any component variable in a system component significantly influences the other component in the system as a whole, and a reduction of irreversibility's rate in a plant component give a greater reduction in the irreversibly rate of system as whole. The greater the value of condenser temperature, the greater the irreversibly. Because T_{cond} and T_{evap} affect the system exergetic efficiency, they need to be optimized each particularly heat transfer area chosen for the two heat exchanger in the refrigerating system.

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