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# EXERGETIC PERFORMANCE OF A DOMESTIC REFRIGERATOR USING R12 AND ITS ALTERNATIVE REFRIGERANTS

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**Abstract**

Production and use of R12 and other chlorofluorocarbon refrigerants will be prohibited completely all over the world in the year 2010 due to their harmful effects on the earth's protective ozone layer. Therefore, in this study, the exergetic performance of a domestic refrigerator using two environment-friendly refrigerants (R134a and R152a) was investigated and compared with the performance of the system when R12 (an ozone depleting refrigerant) was used. The effects of evaporator temperature on the coefficient of performance (COP), exergy flow destruction, exergetic efficiency and efficiency defect in the four major components of the cycle for R12, R134a and R152a were experimentally investigated. The results obtained showed that the average COP of R152a was very close to that of R12 with only 1.4% reduction, while 18.2% reduction was obtained for R134a in comparison with that of R12. The highest average exergetic efficiency of the system (41.5%) was obtained using R152a at evaporator temperature of -3.0°C. The overall efficiency defect in the refrigeration cycle working with R152a is consistently better (lower) than those of R12 and R134a. Generally, R152a performed better than R134a in terms of COP, exergetic efficiency and efficiency defect as R12 substitute in domestic refrigeration system.

Keywords: Domestic, Exergetic, Performance, Refrigeration, R12 alternatives.

## 1. Introduction

Chlorofluorocarbons (CFCs) have been used extensively over the last seven decades in refrigeration due to their favourable characteristics such as non-flammability, non-toxicity, non-explosiveness, and chemically stable behaviour with other materials. These characteristics are the primary requirements of the ideal

## Nomenclatures

| | |
|---|---|
| COP | Coefficient of performance |
| $h$ | Specific enthalpy, kJ/kg |
| $\dot{m}_r$ | Mass flow rate of refrigerant, kg/s |
| $Q_e$ | Refrigerating capacity, W |
| $R$ | Refrigerant |
| $s$ | Specific entropy of refrigerant, kJ/kg.K |
| $T$ | Temperature, K |
| $W_c$ | Compressor work input, W |
| $X$ | Exergy, W |

*Greek Symbols*

| | |
|---|---|
| $\delta$ | Efficiency defect |
| $\eta_x$ | Exergetic efficiency (%) |

*Subscripts*

| | |
|---|---|
| *comp* | Compressor |
| *cond* | Condenser |
| *evap* | Evaporator |
| *exp* | Expansion device or capillary tube |
| *i* | Particular component in a refrigeration system |
| *in* | Inlet or input |
| *o* | Environmental state |
| *out* | Outlet or output |
| *r* | Refrigerant |
| *t* | Total |
| *1* | Outlet of evaporator |
| *2* | Outlet of compressor |
| *3* | Outlet of condenser |
| *4* | Inlet of evaporator |

## Abbreviations

| | |
|---|---|
| GWP | Global warming potential |
| ODP | Ozone depletion potential |

ideal refrigerant. Unfortunately, in recent years it has been recognised that the chlorine released from CFCs migrates to the stratosphere and destroys the earth's stratospheric ozone layer causing health hazards [1, 2].

International concern regarding the potential destruction of the earth's protection layer led to twenty-four nations and the European Community signing the Montreal Protocol in 1987, which regulates the production and trade of ozone depleting substances. The CFCs have been banned in developed countries since 1996, and in 2010, producing and using of CFCs will be prohibited completely all over the world. Also, the partially halogenated HCFCs are bound to be prohibited in the near future [3-6]. Hydro-fluorocarbons (HFCs) are candidates for the definite substitution of both CFCs and HCFCs, as they do not contain chlorine and hence have zero ozone depletion potential [7]. In addition to zero ozone depletion potential (ODP), the working fluids in refrigeration systems must also have low global warming potential (GWP) and high energy efficiency [8].

Thermodynamic processes in refrigeration system release large amounts of heat to the environment. 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. Irreversibility causes the system performance to degrade. The losses in the cycle need to be evaluated considering individual thermodynamic processes that make up the cycle. Energy analysis is still the most commonly used method in the analysis of thermal systems. The first law is concerned only with the conservation of energy, and it gives no information on how, where, and how much the system performance is degraded. Exergy analysis is a powerful tool in the design, optimization, and performance evaluation of energy systems [9].

The principles and methodologies of exergy analysis are well established [10-13]. An exergy analysis is usually aimed to determine the maximum performance of the system and identify the sites of exergy destruction. Analyzing the components of the system separately can perform exergy analysis of a complex system. Identifying the main sites of exergy destruction shows the direction for potential improvements.

There have been several studies on the performance of alternative environment-friendly refrigerants on the basis of energy and exergy analysis of refrigeration systems. Said and Ismail [14] assessed the theoretical performances of R123, R134a, R11 and R12 as coolants. It was established that for a specific amount of desired exergy, more compression work is required for R123 and R134a than R11 and R12. The differences are not very significant at high evaporation temperatures and hence R123 and R134a should not be excluded as alternative coolants. Also, in their study they obtained an optimum evaporation temperature for each condensation temperature, which yields the highest exergetic efficiency.

Aprea and Greco [15] compared the performance between R22 and R407C (a zeotropic blend) and suggested that R407C is a promising drop-in substitute for R22. Experimental tests were performed in a vapour compression plant with a reciprocating compressor to evaluate the compressor performance using R407C in comparison to R22. The plant overall exergetic performance was also evaluated and revealed that R22 performance is consistently better than that of its candidate substitute (R407C).

Aprea and Renno [8] studied experimentally, the performance of a commercial vapour compression refrigeration plant, generally adopted for preservation of foodstuff, using R22 and its candidate substitute (R417A) as working fluids. The working of the plant was regulated by on/off cycles of the compressor, operating at the nominal frequency of 50 Hz, imposed by the classical thermostatic control. The reported result indicated that the substitute refrigerant (R417A), which is a non-azeotropic mixture and non-ozone depleting, can serve as a long term replacement for R22; it can be used in new and existing direct expansion R22 systems using traditional R22 lubricants. Also in their analysis, the best exergetic performances of R22 in comparison with those of R417A were determined in terms of the coefficient of performance, exergetic efficiency and exergy destroyed in the plant components.

Khalid [16] studied the performance analysis of R22 and its substitute refrigerant mixtures R407C, R410A and R417A on the basis of first law. It was found that the COP of R417A is 12% higher than R22, but for R407C and R410A, COP is 5% lowered as compared to R22, and R417A can be used in existing system without any modification.

Various studies reviewed above focused mostly on the exergetic analysis of R22 and its alternative refrigerants. R12 is used solely in the majority of conventional household refrigerators, and there is currently little information on the exergetic performance of R12 alternatives.

Therefore, in this paper, exergetic performances of a domestic refrigeration system using R12 and its environment-friendly alternative refrigerants are experimentally studied and compared.

## 2. Materials and Methods

### 2.1. Exergetic analysis of vapour compression refrigeration system

A reversible thermodynamic process can be reversed without leaving any trace on the surroundings. This is possible only if the net heat and net work exchange between the system and the surrounding is zero [8]. All real processes are irreversible. Some factors causing irreversibility in a refrigeration cycle include friction and heat transfer across a finite temperature difference in the evaporator, compressor, condenser, and refrigerant lines, sub-cooling to ensure pure liquid at capillary tube inlet, super heating to ensure pure vapour at compressor inlet, pressure drops, and heat gains in refrigerant lines [17]. Accurate analysis of the system is obtained by evaluating the exergy used in the system components. The *p*-*h* diagram of the vapour compression refrigeration cycle is presented in Fig. 1. Exergy flow destroyed in each of the components is evaluated as follows [8, 18]:

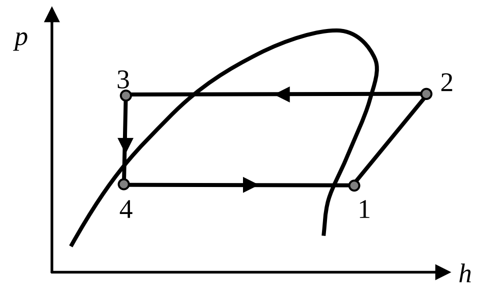


**Fig. 1. Vapour Compression Refrigeration System on *p*-*h* Diagram.**

#### 2.1.1. Exergy of the evaporator

Exergies at the evaporator inlet ($X_{evap,in}$) and outlet ($X_{evap,out}$) are calculated using Eqs. (1) and (2)

$$X_{evap,in} = \dot{m}_r\left(h_4 - T_o s_4\right) + Q_e\left(1 - \frac{T_o}{T_r}\right) \tag{1}$$

$$X_{evap,out} = \dot{m}_r\left(h_1 - T_o s_1\right) \tag{2}$$

Therefore,

$$X_{evap} = X_{evap,in} - X_{evap,out} \tag{3}$$

Substitution of Eqs. (1) and (2) into Eq. (3) gives

$$X_{evap} = \dot{m}_r\left(h_4 - T_o s_4\right) + Q_e\left(1 - \frac{T_o}{T_r}\right) - \dot{m}_r\left(h_1 - T_o s_1\right) \tag{4}$$

### 2.1.2. Exergy of the compressor

Exergies at the compressor inlet ($X_{comp,in}$) and outlet ($X_{comp,out}$) are calculated using Eqs. (5) and (6).

$$X_{comp,in} = \dot{m}_r\left(h_1 - T_o s_1\right) + W_c \tag{5}$$

$$X_{comp,out} = \dot{m}_r\left(h_2 - T_o s_2\right) \tag{6}$$

Therefore,

$$X_{comp} = X_{comp,in} - X_{comp,out} \tag{7}$$

or

$$X_{comp} = \dot{m}_r\left(h_1 - T_o s_1\right) + W_c - \dot{m}_r\left(h_2 - T_o s_2\right) \tag{8}$$

### 2.1.3. Exergy of the condenser

Exergies at the condenser inlet ($X_{cond,in}$) and outlet ($X_{cond,out}$) are calculated using Eqs. (9) and (10).

$$X_{cond,in} = \dot{m}_r\left(h_2 - T_o s_2\right) \tag{9}$$

$$X_{cond,out} = \dot{m}_r\left(h_3 - T_o s_3\right) \tag{10}$$

Therefore,

$$X_{cond} = X_{cond,in} - X_{cond,out} \tag{11}$$

or

$$X_{cond} = \dot{m}_r\left(h_2 - T_o s_2\right) - \dot{m}_r\left(h_3 - T_o s_3\right) \tag{12}$$

### 2.1.4. Exergy of the expansion device (capillary tube)

Exergies at the capillary tube inlet ($X_{exp,\mathrm{in}}$) and outlet ($X_{exp,out}$) are calculated using Eqs. (13) and (14).

$$X_{\exp,in} = \dot{m}_r\left(h_3 - T_o s_3\right) \tag{13}$$

$$X_{\exp,out} = \dot{m}_r\left(h_4 - T_o s_4\right) \tag{14}$$

Therefore,

$$X_{exp} = X_{exp,in} - X_{exp,out} \tag{15}$$

or

$$X_{\exp} = \dot{m}_r\left(h_3 - T_o s_3\right) - \dot{m}_r\left(h_4 - T_o s_4\right) \tag{16}$$

The enthalpy across the capillary tube remains constant ($h_3$ = $h_4$), since expansion process is an isenthalpy process, therefore, Eq. (16) can be expressed as

$$X_{\exp} = \dot{m}_r T_o\left(s_4 - s_3\right) \tag{17}$$

### 2.1.5. Total exergy of the system

The total exergy used in the system ($X_t$) is the total sum of exergy used in each component ($X_i$), where '$i$' stands for particular component:

$$X_t = \sum X_i \tag{18}$$

Therefore,

$$X_t = X_{evap} + X_{comp} + X_{cond} + X_{exp} \tag{19}$$

## 2.2. Exergetic efficiency

The overall system exergetic efficiency ($\eta_x$) is the ratio of the exergy output ($X_{out}$) to exergy input ($X_{in}$) [19]

$$\eta_x = \left(\frac{X_{out}}{X_{in}}\right) \times 100\% \tag{20}$$

Exergy output ($X_{out}$) is the difference between exergy input ($X_{in}$) and the total exergy used in the system ($X_t$), that is

$$X_{out} = X_{in} - X_t \tag{21}$$

The only source of exergy input to the system is through the electrical power supplied to the compressor ($W_c$), that is, $X_{in} = W_c$ and Eq. (20) can be expressed as:

$$\eta_x = \left(\frac{W_c - X_t}{W_c}\right) \times 100\% \quad \text{or}$$

$$\eta_x = \left(1 - \frac{X_t}{W_c}\right) \times 100\% \tag{22}$$

The efficiency defect ($\delta$) is evaluated for each device of the system, considering the ratio of exergy used in each component ($X_i$) to the exergy required to sustain the process (exergy input through the compressor, $W_c$).

Therefore,

$$\delta_i = \frac{X_i}{W_c} \tag{23}$$

and

$$\sum \delta_i = \frac{\sum X_i}{W_c} = \frac{X_t}{W_c} \tag{24}$$

Substitution of Eq. (24) into Eq. (22) gives an expression Eq. (25), which shows the link between the efficiency defects of the components and the exergetic efficiency of the whole system.

$$\eta_x = \left(1 - \sum \delta_i\right) \times 100\% \tag{25}$$

## 2.3. Energetic performance

The overall energetic performance of refrigeration system is determined by evaluating its coefficient of performance (COP), and is calculated as the ratio between the refrigeration capacity ($Q_e$) and the electrical power supplied to the compressor ($W_c$)

$$\mathrm{COP} = \frac{Q_e}{W_c} \tag{26}$$

## 3. Experimental Set-Up

The test rig used for the experiment is a complete vapour compression refrigeration system developed in the form of a single temperature domestic refrigerator designed to work with R12. The schematic diagram of the experimental domestic refrigerator is shown in Fig. 2. The experimental refrigerator consists of an evaporator, wire mesh air cooled condenser and hermetically sealed reciprocating compressor. The refrigerator was instrumented with two pressure gauges at the inlet and outlet of the compressor for measuring the suction and discharge pressure, while the energy consumption of the refrigerator was measured with watt-hour meter.

The rig was thoroughly checked and commissioned before it was subjected to series of tests at various conditions. The evacuation was carried out with the help of vacuum pump and refrigerant was charged into the refrigerator with the help of charging system. The refrigerator was first charged with R12 and tested at the intended various conditions. The experiment was repeated for R134a and R152a refrigerants.

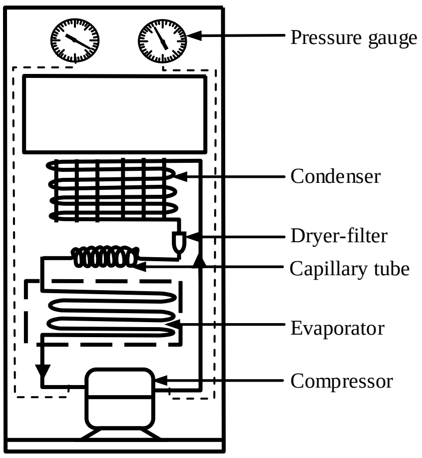


**Fig. 2. Schematic Diagram of Experimental Refrigerator.**

## 4. Results and Discussion

Figure 3 shows the variation of coefficient of performance (COP) with varying evaporator temperature for R12, R134a and R152a. The figure shows that the COP increases with increase in evaporator temperature. The trend is similar for all the investigated refrigerants. The results obtained showed that the average COP for R134a and R152a are 18.2 and 1.4% lower in comparison to R12. R152a has nearly the same COP with R12. Refrigerator working with R134a, which has significant lower COP requires higher electric power consumption in order to provide the same refrigerating load. Apart from direct costs, this is disadvantageous in terms of overall environmental pollution, since more fuel must be burned and higher amount of carbon dioxide are discharged into the atmosphere.

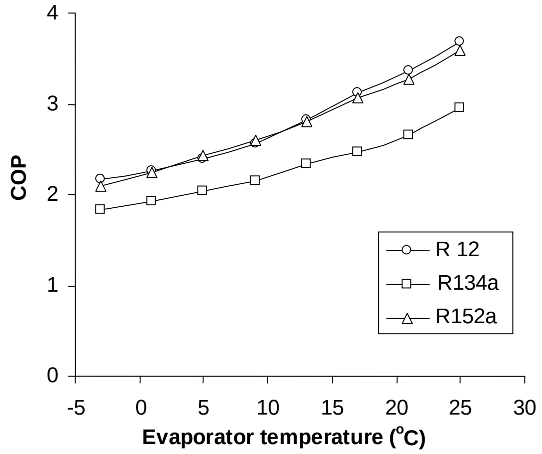


**Fig. 3. Variation of Coefficient of Performance (COP) with Varying Evaporator Temperature.**

Variation of exergetic efficiency with evaporator temperature for R134a and R152a compared with R12 is shown in Fig. 4. Exergetic efficiency decreases with increase in evaporator temperature. Average exergetic efficiencies for R134a and R152a are 13.6% lower and 4.4% higher in comparison to that of R12, respectively. Exergetic efficiency of 41.0, 37.3 and 41.5% were obtained at evaporator temperature of -3°C for R12, R134a, and R152a, respectively.

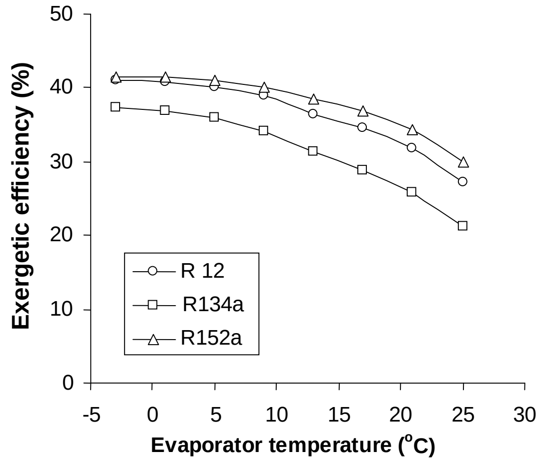


**Fig. 4. Variation of Exergetic Efficiency with Evaporator Temperature.**

Figure 5 shows the comparison of efficiency defect in compressor for R12, R134a and R152a with varying evaporator temperature. As shown in the figure, efficiency defect in compressor decreases with decrease in evaporator temperature. The result obtained showed that efficiency defect in compressor is 0.9% higher and lower for R134a and R152a respectively in comparison with that of R12.

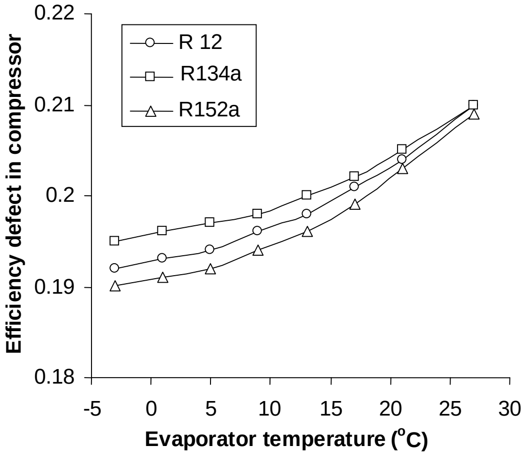


**Fig. 5. Variation of Efficiency Defect in Compressor with Evaporator Temperature.**

Figure 6 shows the variation of efficiency defect in condenser with evaporator temperature for R12, R134a and R152a. As shown in the figure, efficiency defect in condenser decreases with decrease in evaporator temperature. The result obtained showed that efficiency defect in condenser is 6.2 and 13.9% lower for R134a and R152a respectively in comparison with that of R12.

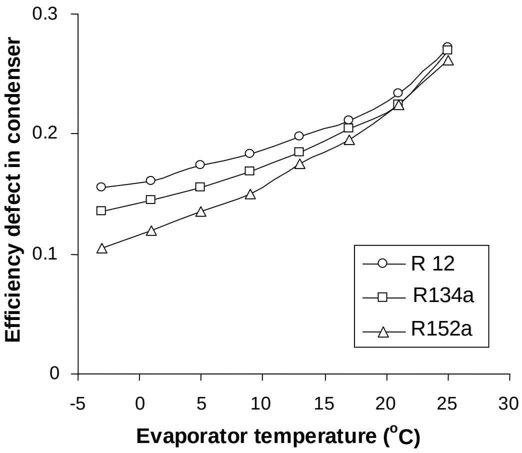


**Fig. 6. Variation of Efficiency Defect in Condenser with Evaporator Temperature.**

Figure 7 shows the variation of efficiency defect in capillary tube with evaporator temperature for R12, R134a and R152a. As revealed in the figure, efficiency defect in capillary tube decreases with increase in evaporator temperature. The result obtained showed that efficiency defect in capillary tube is 19.1% higher and 20.4% lower for R134a and R152a respectively in comparison with that of R12.

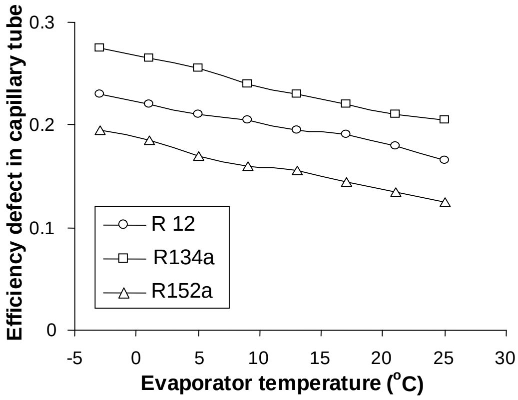


**Fig. 7. Variation of Efficiency Defect in Capillary Tube with Evaporator Temperature.**

Figure 8 shows the variation of efficiency defect in evaporator with evaporator temperature for R12, R134a and R152a. This figure revealed that the efficiency defect in evaporator decreases with decrease in evaporator temperature. The results obtained showed that efficiency defects in evaporator are 24.4% higher and 18.5% lower for R134a and R152a respectively in comparison with that of R12. As shown in Fig. 8, the overall efficiency defect in evaporator is marginal in comparison with those of other components in the system (Figs. 5, 6 and 7). Transferring heat at lower temperature difference can further reduce the efficiency defect in the evaporator.

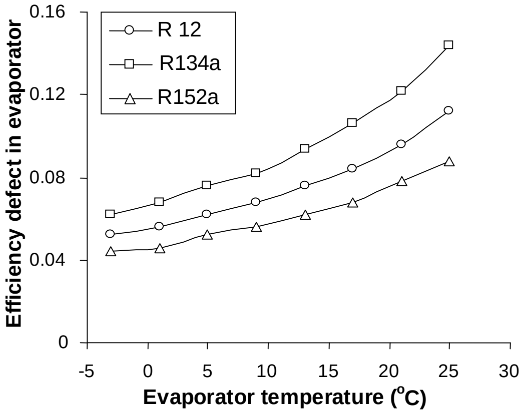


**Fig. 8. Variation of Efficiency Defect in Evaporator with Evaporator Temperature.**

## 5. Conclusion

The exergetic performance of a domestic refrigeration system is experimentally investigated using two environment-friendly alternative refrigerants. After the successful investigation on the exergetic performance of R12 and its substitutes (R134a and R152a) in the experimental refrigerator, the following conclusions can be drawn based on the results obtained:

i. The coefficient of performance (COP) of the domestic refrigeration system using R12 as a refrigerant was considered as benchmark and the COPs of the system using R134a and R152a were compared. The COP obtained using R152a was very close to that of R12 with only 1.4% reduction, while that of R134a was significantly low with 18.2% reduction. Refrigerant with lower COP will consume more energy, which will have great adverse effect on the environment.

ii. The highest exergetic efficiency was obtained using R152a in the system. The average exergetic efficiencies of the system using R134a and R152a are 13.6% lower and 4.4% higher than that of R12, respectively.

iii. The overall efficiency defect in the cycle working with R152a is consistently better (lower) than those of R12 and R134a.

iv. The highest efficiency defects in three of the four components in the refrigeration system (compressor, evaporator and capillary tube) were obtained using R134a as refrigerant.

v. Generally, the experimental domestic refrigeration system performed better using R152a than using R12 and R134a as working fluids.

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