FIRST AND SECOND LAW EVALUATION OF REFRIGERATION SYSTEM OPERATING WITH REFRIGERANTS CFC-R12, HCFC-R22 AND HFC-R134a.

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ABSTRACT

Based on the thermodynamic properties, the first law performance COP and the second law efficiency η_{ex}, were employed to evaluate the system behavior for refrigerants CFC-R12, HCFC-R22, and HFC-R134a. Test results showed that, the system-cooling COP for R-134a is 8% and 22% higher than the COP of R-12 and R-22, respectively. Based on the second law analysis, the components irreversibility varies considerably, in such way that, main source of exergy loss is due to the heat transfer in evaporator, condenser, and suction line for which, effective measures should be taken. However, this analysis indicates that R134a appears to be meritorious, its exergetic efficiency is 20% and 8% higher than that of R12 and R22, respectively.

Keywords: First and Second Law, CFC-R12, HCFC-R22, HFC, -R134a

INTRODUCTION

The role of chlorine in destruction of stratospheric ozone and the resulting health and environmental risks, have led to the Montreal protocol 1987. This protocol necessitates the eventual global phaseout of cholorofluorocarbon (CFC-R12) by the beginning of 1992 and hydrochlorofluorocarbon (HCFC-R22) by the end of 2010 year. This statement postulated the selection of alternatives and compounds with no chlorine refrigerants having higher or at least equal energy efficiency for heating, air conditioning and refrigeration equipment.

Considerable work has been done with regard to replacement of the heat pump refrigerants. Only some of relevant studies are mentioned here. These studies showed that, HFC-R134a has become the leading candidate to replace the current working fluids in refrigeration and air-conditioning systems. This is because the HFCs address the concern about stratospheric ozone depletion, as they contain no chlorine [1]. Linton [2], conducted work to rank CFC-R12 alternative refrigerants HFC-R134a and HFC-R152, using water to water heat pump. Comparison study of energetic characteristics between R12 and R134a has been done by Corr *et al.* [3]. The study indicated that, R12 systems are being successfully retrofit with R134a. Several compounds proposed a near-term or longer range substitutes for the regulated chlorofluorocarbon (CFC) refrigerants were tested in a vapor compression circuit by Sand *et al.* [4]. Their performance was evaluated relative to more commonly used refrigerants CFC-R12 and HCFC-R22. Concerning HCFC-R22, the mixture of R32 and one or both of HFCs (R152a and R134a), is considered as a promising alternative to replace HCFC-R22. In addition, R134a, as a pure fluid had been proved as a leading candidate [5-8]. Wide programs were performed to evaluate HCFC-R22 alternatives [9, 10]. The objective of these programs was to provide performance data on replacement refrigerants in compressors and system components. Throughout the evaluation process, it appeared that, HFC-R134a is typically considered as the long term replacement of CFC-R12, however, HFC-R134a is also being considered in some HCFC-R22 applications. The foremost reason for this choice is that, it has zero ozone depletion potential. In addition, R134a is non-flammable, it has extremely low toxicity, high thermal stability and it is now available in commercial quantities.

Most present day energy analysis, including the abovementioned work proceed beyond the first law of thermodynamics. This results in determination of the coefficient of performance, COP, which serves as a useful screening tool for alternatives quantitative evaluation. However, with this type of analysis, factors that play important role on system performance such as, heat transfer and thermo-physical properties are ignored. Therefore, further analysis and testing need to be taken place before an accurate and qualitative evaluation of alternative system efficiency can be made. In this paper, an attempt has been done to overcome this drawback by evaluating R134a as alternative to R12 and R22 based on the first law coefficient of performance COP and the second law exergetic efficiency η_{ex} as figures of merit with special emphasis on the effects of thermophysical properties on the components behavior and the system as a whole.

EXPERIMENTAL

Test Facility

Figure 1, is a schematic flow diagram of the test facility, which was specially built in the Refrigeration and Air-Conditioning Laboratory of the Faculty of Engineering Zagazig University. The refrigeration circuit uses a Prestcold MAKO-A3 air-cooled condensing unit. This unit is designed to operate with conventionally used refrigerants R12, R22, R502 and R134a. The condensing unit uses a semi-closed two-cylinder reciprocating compressor driven by a 1.5 hp electric motor. In addition, an air-cooled condenser is bolted to the unit baseframe. The air is drawn through the condenser and discharged over the compressor by multibladed fan, which is direct driven by an electric motor. The condensing unit is equipped with a filter drier and a liquid receiver fitted with a liquid outlet shut-off valve.

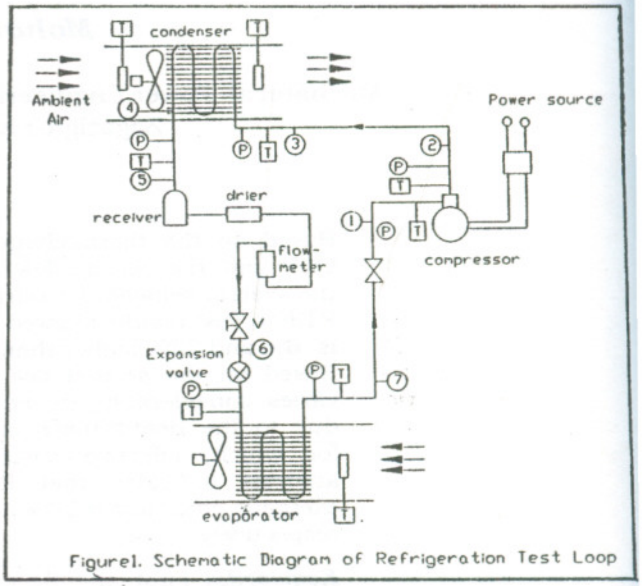

Figure 1 Schematic diagram of refrigeration test loop

The evaporator is an air-cooler Frigabon Muc type in the form of a finned tube, refrigerant to air heat exchanger. The air cooler is fixed in a specially built insulated room of length, width and height of 365,154 and 310 cm, respectively. A controlled electric heater was located inside the insulated-cooled- room to simulate the heat load imposed on the evaporator. (taking into account the heat transmitted through the room wall).

In this study, three 1.5 hp Danfoss type thermostatic expansion valves for R-12, R22, and R134a were used. Thus, each valve has been employed with the corresponding tested refrigerants.

Measurements

Refrigerant pressure measurements at the inlet and outlet of the main circuit components, are made with a set of Bourdon tube refrigerant gauges having range of (0 - 250) psi. The gauge accuracy was about ± 2% of the full scale value.

Temperature measurements are made by copper-constantan thermocouples. The beads of the thermocouples were embedded in specially cut V shaped notches at the inlet and outlet of the main refrigerant circuit components, and were fixed by steel epoxy material. Air temperature at the inlet and outlet of the evaporator and condenser, as well as the ambient and cooled room temperature were also measured. The temperature readings were received by means of Digi-Sense digital thermometer having an uncertainty of 0.1C°.

The electric power supplied to the heater, was read directly by a digital wattmeter Bri-5040 with an uncertainty of 1%, for the same concern, an AC/DC clamp on was calibrated for compressor power measurements of an accuracy of 3% of the full scale.

The refrigerant flow rate was measured by a variable area rotameter manufactured by Gilmont type NO44 - 40C, installed in the liquid line before the expansion valve. The maximum error is in order of ±4.5%. A vane anemometer type Sketch was used to measure the air velocities at the exit of the evaporator and condenser

Experimental Procedure

Having assembled the refrigerant circuit the, pressure test was processed using nitrogen then the system was evacuated and charged by an adequate amount of the tested refrigerant and lubricating oil, according to the manufacturer instructions. Based on the recommendation of Sanvordenker [8], in this study, the compressor with the same lubricant was used.

As a prelude to the collection of the data to be reported later, a number of preliminary check tests were made. These include, ensuring the opening of all refrigerant circuit valves, and to ensure that the controlling devices are adjusted at the predetermined values. The baseline tests were first run with R12, using the commercially available alkylbenzene refrigerant lubricating oil. The experimental test run can be briefly described in the following sequences: the condensing unit and air cooler fan were switched on and the power to the heater was set to a convenient value. The system was then allowed to reach the steady state condition which, was indicated by the constancy of the cooled room temperature. Tests indicated that, a time of approximately three hours was necessary for the system to reach steady state. During this time necessary adjustments in the input power to the heater were made. Once steady state has been reached, reading of temperature, pressure, power and refrigerant flow rate were simultaneously recorded. After the aforementioned measurements are recorded, the energy quantities Q_e ($m.q_e$) and W_{com} and hence, the actual performance COP_{ac} were determined. The above mentioned steps were repeated following a decrease in the input power to the heater, which in turn, resulting in a corresponding decrease in the cooled room temperature. On completion of the base line tests with CFC-R12, the compressor and the system were drained, and thermostatic R12 expansion valve has been replaced with R22 one. The system was then evacuated several times and recharged with HCFC-R22. The same series of tests were then repeated for R22 and R134a..

THEORETICAL ANALYSIS

First Law Analysis

It is postulated that, different refrigerants exhibit different performances. The reason lies in their different thermodynamic properties. Although all properties are important, some of them are most dominating than others. These are: the normal boiling temperature- or related to it critical point-, latent heat of vaporization, isobaric specific heats and adiabatic index.

To better understand the effects of different properties on the system performance, reference should be made to Figure 2. Analysis is performed assuming that, the evaporation and condensation temperatures are considered constant and the evaporation temperature is equal to the source of heat-cold room-temperature, whereas condensation temperature is equal to the sink- condenser coolant-temperature. Considering these assumptions, the irreversible losses for considered cycle are those due to the throttling and superheating processes. Hence, the theoretical coefficient of performance COP_{th} for the cycle enclosed by 71234567 is

$$COP_{th} = q_e / \sum w_i \tag{1}$$

The refrigeration effect q_e is given by:

$$q_e = h_{fg(e)} - (h_6 - h_o) \tag{2}$$

$$= h_{fg(e)} - c_f\,(T_5 - T_e) \tag{3}$$

The total compression work $\sum W_i$ is :

$$\sum w_i = w_1 + w_2 + w_3 \tag{4}$$

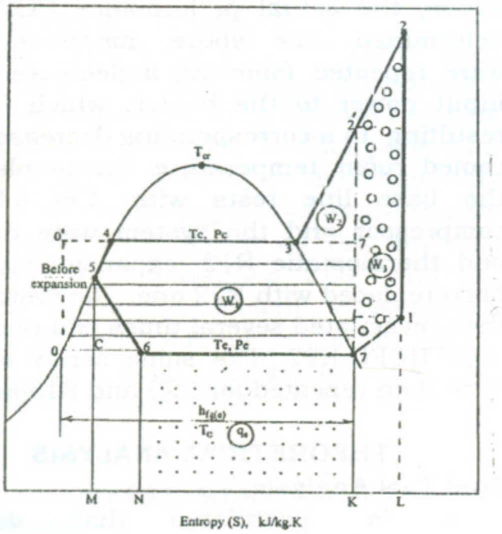

Figure2 T-S diagram for considered cooling cycle (First law analysis)

These items can be derived referring to Figure2, as follows:

w1 is equivalent to the area 77` 345 MN 67. Taking into account the equality of areas CMN6C and 5OC5 for equal enthalpy differences $(h_6 - h_C) = (h_5 - h_C)$. Thus, $w_1$ is equivalent to the area 77`34507. This, in turn is equal to the difference between area enclosed by rectangular 77` O`O and area enclosed by 0`40. Neglecting the nonelinearity of 04 and assuming constant isobaric specific heat c_f along 04, we have:

$$w_1 = \frac{h_{fg(e)}}{T_e}\left(T_c - T_e\right) - 0.5\left(T_c - T_e\right)c_f \ln\frac{T_c}{T_e} \tag{5}$$

$$= (T_c - T_e)\left[h_{fg(e)} / T_e - 0.5c_f \,.(\ln T_c / T_e)\right] \tag{6}$$

w_2 is given by the area enclosed by 2`37`. Thus ,

$$w_2 \cong 0.5\,(T_{2'} - T_3)\,.\,\Delta S_{3-7} \tag{7}$$

Since, $dS = c\, dT/T$, where c is the process isobaric specific heat. At the conditions of c_s = constant from T_c to $T_{2'}$, and c_g = constant from T_e to T_c and taking into account that $\Delta S_{3-7'} = \Delta S_{3-7}$, and $T_c/T_e = \Gamma$ thus :

$$\Delta S_{3-7'} = c_s \ln T_{2'} / T_c = c_g \ln \Gamma \tag{8}$$

Hence, $T_{2'} = T_c\,(\Gamma)^{c_g/c_s}$ (9)

Thus, Equation 7 can be re-written as:

$$W_2 \cong 0.5T_c\left[(\Gamma)^{c_g/c_s} - 1\right]c_g \ln\Gamma \tag{10}$$

In Equation 10, c_g and c_s are isobaric specific heats of vapor at saturation and superheating conditions, respectively.

The term w_3 is represented by area enclosed by 7`2` 217 , Thus :

$$W3 = \{0.5\,(T_1 - T_7) + (T_{2'} - T_1) + 0.5\,(T_2 - T_{2'})\}\,\Delta S_{71} \tag{11}$$

$$= 0.5\,\{(T_{2'} + T_2) - (T_1 + T_e)\}.c_{s7-1} \ln T_1/T_7 \tag{12}$$

Inserting Equations 3, 6, 10 and 12, into Equation 1 we have

$$COP_{th} = \frac{h_{fg(e)} - c_f(T_5 - T_e)}{(T_c - T_e)\left\{\frac{h_{fg(e)}}{T_e} - \frac{c_f}{2}\ln\Gamma\right\} + 0.5T_c\left[\Gamma^{c_g/c_s} - 1\right]c_g \ln\Gamma + 0.5c_{s71}\left\{(T_{2'} + T_2) - (T_1 - T_2)\right\}\ln\frac{T_1}{T_7}} \tag{13}$$

The differentiation of Equation. 13 with respect to thermodynamic properties c_g , c_f and h_{fg} does not give any extremum value for COP_{th}. However, the analysis of Equation 13 shows that, this coefficient decreases inversely with The ratio Γ, moreover the greater the latent heat $h_{fg(e)}$, the higher the specific heat c_s and the smaller the specific heats c_g and c_f, the greater is the coefficient of performance COP_{th}. It is worthnoting that this coefficient be compared with corresponding actual value which is determined from measured energy quantities $Q_e = (mq_e)$ and W_{com} as:

$$COP_{ac} = Q_e / W_{com} \tag{14}$$

Second Law Analysis

Principles form of the second law of the thermodynamic have been used in various ways to compare the actual efficiency of various processes to ideal processes operating under similar conditions. This analysis allows to determine the relative contribution of each of the components processes to the overall inefficiency of the system. Thus, the first law analysis is concerned with quantity performance analysis, while the second law can be considered as a quality analysis. The major analytical usefulness of the second law is that, it defines the property of entropy generation, which can be used quantitatively to determine possible direction and extent of the process. This method states that, each component irreversibility, (loss of availability or exergy),[12] can be calculated as a function of entropies of the refrigerant entering and leaving the component, the heat transfer rate, and the source (load) or sink temperature. For any given component, the availability or exergy loss is given by

$$L_i = T_a \cdot m \, \Delta S_i \tag{15}$$

The relative exergy loss ϕ_i can be expressed as :

$$\phi_i = L_i / E \tag{16}$$

Where E is the input exergy to the system. This exergy is supplied in the form of electrical power to the compressor (W_{com} = E) . Thus, W_{com} or E is spent to provide exergy demand for cooling and to compensate the irreversible losses of the system $\sum Li$. Computation of the system losses have been performed considering the following assumptions :

- Hydraulic losses as well as those due to the impurities in refrigerants can be ignored.
- Refrigerant superheat in the evaporator is negligible.
- Exergy destruction due to the heat transfer between the system components - with exception of the suction line - and the surrounding is negligible.

The exergy loss during the throttling process (5 - 6), Figure3, can be determined as :

$$L_{5-6} = m. \, T_a \, \Delta \, S_{5-6} \tag{17}$$

Where m, is the refrigerant mass flow rate, which can be determined by :

$$m = \frac{Q_e}{q_e} = \frac{COP_{ac} W_{com}}{q_e} = \frac{COP_{ac} \cdot E}{q_e} \tag{18}$$

The relative exergy loss during throttling process ϕ_{5-6} is given by:

$$\phi_{5-6} = \frac{L_{5-6}}{E} = \frac{COP_{ac}}{q_e} . T_a . \Delta S_{5-6} \tag{19}$$

The exergy loss in the evaporator, L_{6-7}, can be calculated as.

$$L_{6-7} = T_a \cdot Q_e \left(T_e^{-1} - T_r^{-1} \right) \tag{20}$$

and its relative exergy loss ϕ_{6-7}, is expressed as :-

$$\phi_{6-7} = \frac{L_{6-7}}{E} = T_a \, COP_{ac} \left(\frac{1}{T_e} - \frac{1}{T_r} \right) \tag{21}$$

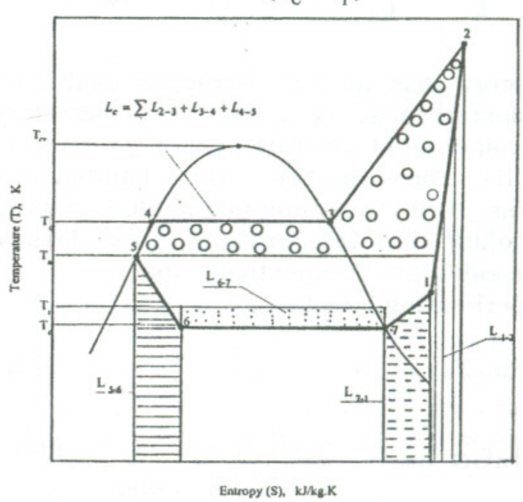

Figure 3 Individual exergy losses on T-S diagram (Second law Analysis)

The exergy loss in the suction line L_{7-1}, caused by the heat transfer with the surrounding (process 7-1), is computed as:

$$L_{7-1} = m.\ T_a\ .\ \Delta S_{7-1} \quad (22)$$

$$= \frac{COP_{ac}.E}{q_e}.T_a.c_{s(7-1)m}\left[\frac{(T_1 - T_e)}{T_{1-7(m)}}\right] \quad (23)$$

Where $c_{s(7-1)}$ is the isobaric specific heat of the superheated vapor, and is evaluated at the mean temperature $T_{(7-1)m}$. which is calculated as:

$$T_{(1-7)_m} = \left(T_1 - T_e\right) / \ln T_1 / T_e \quad (24)$$

The relative exergy loss during process (7-1) is :

$$\phi_{7-1} = \frac{L_{7-1}}{E} = \frac{COP_{ac}}{q_e} T_a.c_{s(7-1)m}\left(\frac{T_1 - T_e}{T_{(1-7)m}}\right) \quad (25)$$

The compressor exergy loss L_{1-2} (process 1-2), is calculated by :

$$L_{1-2} = m.\ T_a\ \Delta S_{1-2} \quad (26)$$

$$= \frac{Q_e}{q_e}.T_a.\Delta S_{(1-2)} \quad (27)$$

and the relative exergy compression loss ϕ_{12}, is ,

$$\phi_{1-2} = \frac{L_{1-2}}{E} = \frac{COP_{ac}}{q_e} T_a\left(S_2 - S_1\right) \quad (28)$$

Exergy loss in the condenser is due to the desuperheat of vapor (process 2-3) condensation of saturated vapor (process 3-4) and subcooling of the condensate (process 4-5). Assuming negligible variation in coolant (air) temperature, these losses can be calculated respectively, as:

Desuperheating loss L_{2-3} :

$$L_{2-3} = m\ .\ T_a\ .\ \Delta S_{2-3} \quad (29)$$

$$= \frac{Q_e}{q_e} T_a c_{s(2-3)m}(T_2 - T_3) / \left(\frac{1}{T_a} - \frac{1}{T_{(2-3)m}}\right) \quad (30)$$

and the relative loss ϕ_{2-3}, is :

$$\phi_{2-3} = \frac{L_{2-3}}{E} = \frac{COP_{ac}}{q_e} T_a c_{s(2-3)m}(T_2 - T_3)\left(\frac{1}{T_a} - \frac{1}{T_{(2-3)m}}\right) \quad (31)$$

Condensation exergy loss L_{3-4} :

$$L_{3-4} = m\ .\ T_a\ .\ \Delta S_{3-4} \quad (32)$$

$$= \frac{Q_e}{q_e}.T_a.h_{fg(c)}\left[\frac{1}{T_a} - \frac{1}{T_c}\right] \quad (33)$$

and the relative exergy loss of condensation, ϕ_{3-4} is :

$$\phi_{3-4} = \frac{COP_{ac}}{q_e}.T_a.h_{fg(c)}\left[\frac{1}{T_a} - \frac{1}{T_c}\right] \quad (34)$$

In Equation 34, the refrigerant latent heat $h_{fg(c)}$, is evaluated at the condensation temperature T_c.

The subcooling exergy loss L_{4-5} is given by :

$$L_{4-5} = m\ .\ T_a\ .\ \Delta S_{4-5} \quad (35)$$

$$= \frac{Q_e}{q_e}.T_a.c_{f(4-5)m}.(T_c - T_5)\left[\frac{1}{T_a} - \frac{1}{T_{(4-5)m}}\right] \quad (36)$$

and the relative exergy loss, ϕ_{4-5} is :

$$\phi_{4-5} = \frac{L_{4-5}}{E} = \frac{COP_{ac}}{q_e}.T_a c_{f(4-5)}(T_c - T_5)\left[\frac{1}{T_a} - \frac{1}{T_{(4-5)m}}\right] \quad (37)$$

Finally the exegetic efficiency of the system η_{ex} can be expressed as:

$$\eta_{ex} = \frac{E - \sum_{i=1}^{n} L_i}{E} = 1 - \sum_{i=1}^{n} \phi_i \quad (38)$$

In this study, the thermal properties of the tested refrigerants are evaluated at the process mean temperature. Properties of R12 and R22 were obtained from Reference 13, while those of R134a were taken from References 14 and 15. The values of these properties at, t = 0°C, are given in Table 1.

Table1 Thermo-physical properties of R12,R22, and R134a at OC°, [13, 14, and 15]

Item	R12	R22	R134a
c_f, kJ/kg.K	0.944	1.167	1.25
c_g, kJ/kg.K	0.644	0.754	0.75
c_s, kJ/kg.K	0.60	0.62	0.818
t_{cr}, C°	112	96.13	101.3
h_{fg}, kJ/kg	152.06	204.4	198.43
ρ, kg/m^3	1396	1284	1292
k, W/m.K	0.0783	0.0977	-
μ, kg/m.s	2.446×10^{-4}	2.67×10^{-4}	$\approx 2.584 \times 10^{-4}$
σ, N/m	12.4×10^{-3}	11.7×10^{-3}	11.7×10^{-3}
$\gamma = c_p/c_v$	1.14	1.16	1.132

Table 2 Test Measurements (sample at $\Gamma \cong 1.148$)

Item / Ref	T_c K	T_e K	T_1 K	T_2 K	T_3 K	T_4 K	T_5 K	T_6 K	T_7 K	m kg/s	Q_e kW	W_{com} kW	COP_{ac} Eq.14	COP_{th} Eq.13
R12 $\Gamma=1.145$	**310**	**271**	**284.5**	**352**	**345**	**310**	**298**	**267**	**275**	**0.016**	**1.970**	**0.680**	**2.9**	**3.62**
R22 $\Gamma=1.148$	**307**	**267.5**	**282**	**376**	**368**	**307**	**291**	**265**	**275.5**	**0.011**	**2.137**	**0.825**	**2.59**	**3.4**
R134a $\Gamma=1.149$	**309**	**268.7**	**282.1**	**350**	**343**	**309**	**295**	**267**	**270.9**	**0.012**	**2.032**	**0.616**	**3.33**	**3.8**

RESULTS AND DISCUSSION

The system tests serve two objectives: (i) evaluation of the refrigeration system performance, based on the thermophysical properties of tested refrigerants, (ii) identification of the individual losses which may occur through the individual system components.

In order to verify the reliability of the test facility and accuracy of measuring instrumentations, a sample of obtained data for R12 and R134a is plotted together with results reported in by Linton *et al.* [2]. For the purpose of comparison, results of Linton *et al.* [2] have been modified to be presented in the form of COP as a function of Tc/Te , as shown in Figure 4. An inspection on these plots reveals that, a reasonable agreement between the present data and those of Linton *et al.* [2] is observed. However, small difference which can be observed is attributed to the different subcooling effects experienced by both systems. Thus, based on this comparison, the reliability of the test facility was approved.

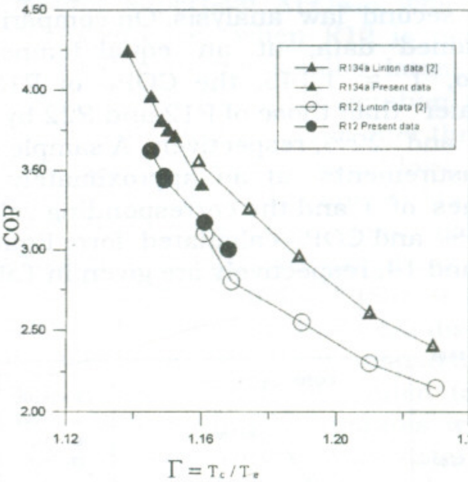

Figure 4 Comparison between the obtained data and results of Linton [2]for R12 and R134a

The First Law Analysis

The coefficient of performance COP_{th} predicted by Equation 13. As well as the coefficient COPac calculated from Equation 14 for three tested refrigerants is plotted in Figure 5 as a function of the temperature ratio Γ = Tc/Te. It is noticed that the trends of the data are similar and decreases with the increase of the ratio Γ = Tc/Te. From the inspection of Figure 5, one comes to the conclusion that, R134a is the most promising of the tested refrigerants as it exhibits the highest COP, specially at the low temperature ratio. This is due mainly to the relatively small adiabatic index ($\gamma = c_p/c_v$), which tends to reduce the discharge temperature and superheat loss. This in turn, results in reduction of the compression, work and accordingly higher coefficient of performance is resulted. In spite of relatively high value of R134a specific heat c_f tending to increase the throttling loss, it is believed that the effect of relatively high latent heat h_{fg134a} seems to be dominant in determining the refrigeration effect q_e. It is noteworthy that the coefficients COP_{ac} are 20 to 40 percent below those of COP_{th} . This discrepancy is due to the irreversibilities associated with the heat transfer processes in the evaporator, condenser, suction line, as well as, due to the entropy generation in compression and throttling processes. The nature of these losses will be highlighted by the second law analysis. On comparing the obtained data, at an equal temperature ratio, $\Gamma \cong 1.145$, the COP_{ac} of R134a is greater than those of R12 and R22 by about 8% and 22%, respectively. A sample of test measurements at an approximately equal values of Γ and the corresponding values of COP_{Th} and COP_{ac} calculated form Equations 13 and 14, respectively are given in Table 2.

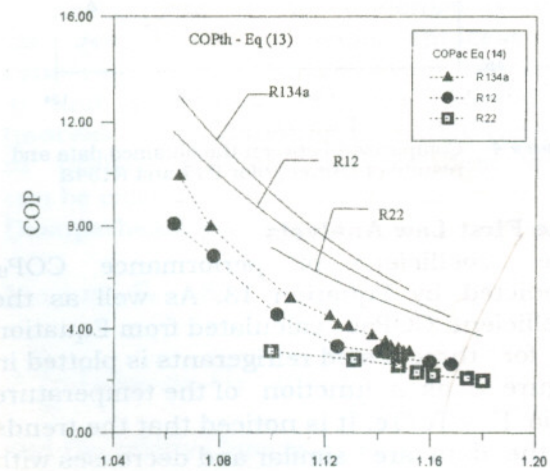

Figure 5 The coefficient of performance COP as a function of the temperature ratio Γ

Correlation

In view of what followed, it was decided to obtain correlation for all the experimental data for COP_{ac}. This coefficient is an interact complex consisting of two terms: referigeration capacity depending of the latent heat or related to it critical temperature T_{cr} and subcooling temperature $T_{s.c.}$. This effect is denoted by symbol $\Pi = T_{cr}/T_{sc}$. The second term is the compression work functioned on condensing to evaporation temperature ratio Γ. Thus, the coefficient COP_{ac} is well correlated by the least square curve fitting as:

$$COP_{ac} = C\,\Gamma^{n}\,.\,\Pi^{m} \qquad (39)$$

where the constant C and exponents (m and n) are listed in Table 3 for tested refrigerants. The maximum deviation between the correlated and measured values of COP_{ac} is 9,10 and 6% for R134a , R12 and R22, respectively

Table 3 Constant C and exponents m and n Equation 39

Refrigerant	C	n	m	Range of variables	
				Γ	π
R134a	11.6	-13.2	2.6	1.065- .152	1.27-1.3
R12	0.9	-14.8	12.5	1.063- .168	1.28-1.3
R22	2.5	-5.3	3.0	1.099- 1.18	1.25-.28

The Second Law Analysis

The relative throttling loss ϕ_{5-6} predicted by Equation 19, for the tested refrigerants are displayed in Figure 6, versus, $\Gamma = Tc/Te$. The predicted throttling loss is seemed to increase with the temperature ratio Γ. Comparing the values of ϕ_{5-6} for the three tested refrigerants, one can observe that the magnitude of ϕ_{5-6} for R134a is greater than that of R12 and R22. This can be explained solely by the effect of high value of $c_{f\ (R134a)}$ tending to increase the throttling loss. At an equal ratio of, $\Gamma = 1.145$, these relative losses constitute about 7, 6.1 and 6.4 % of the input exergy for R134a, R12 and R22, respectively.

The relative exergy loss in the evaporator ϕ_{6-7}, computed from Equation 21, is illustrated in Figure 7 versus the temperature ratio Γ. This figure shows that, the relative loss ϕ_{6-7} decreases with the increase of the ratio Γ up to about 1.14, then increases with further increase in Γ. The interpretation of this phenomenon is that, as the cooled room attained a lower temperature, resulting in small evaporator- cooled room temperature difference (T_r - T_e). Hence, the exergy loss predicted by Equation (20), is reduced. Further increase of loss ϕ_{6-7} can be attributed to the increased vapor fraction deteriorating the evaporation heat transfer coefficient, which leads to a higher evaporator-cooled room temperature difference (T_r-T_e), and subsequently increased exergy loss.

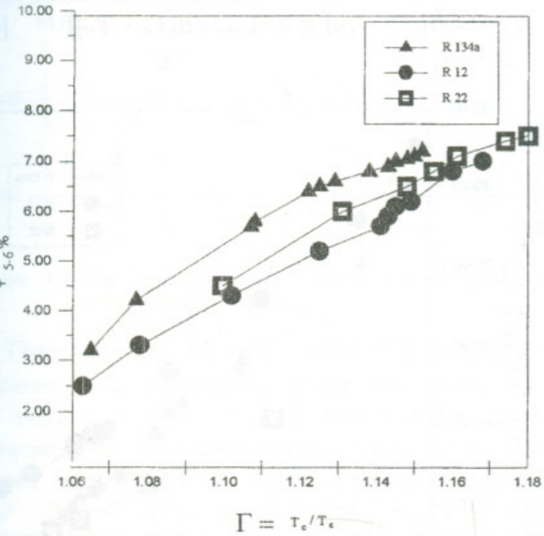

Figure 6 The relative exergy throttling loss ϕ_{5-6}vs.the temperature ratio Γ.

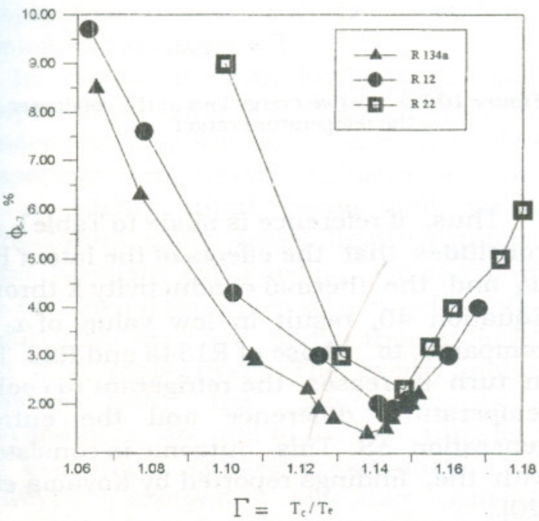

Figure 7 The relative exergy loss in the evaporator ϕ_{6-7} vs. the temperature Γ.

The influence of the thermal properties on the evaporator loss, appears through the effects of T_{cr} and σ on evaporation heat transfer coefficient. Low critical temperature refrigerant starts evaporation at relatively higher vapor fraction [16], deteriorating the evaporation heat transfer coefficient. The role of surface tension σ appears through its effect on the bubble radius which decreases with σ, resulting higher evaporation heat transfer coefficient. Based on the above-mentioned and referring to Table 1 one may expect that, T_{cr} and σ both contribute to the heights exergy loss for R22 and lower loss for R134a as shown in Figure 7.

The relative exergy loss of suction the line ϕ_{7-1} computed from Equation 25 is plotted in Figure 8, for the tested refrigerants. This figure shows that R12 exhibits highest values of ϕ_{7-1} compared with those of R134a and R22. This is due to the greater superheat $\Delta T_{7-1(R12)}$ which the system experiences when R12 is tested . Also, it is due to the smaller latent heat $h_{fg(R12)}$ or, related to it refrigeration effect $q_{e(R12)}$. The contribution of the later through Equation 25 yields a higher loss ϕ_{7-1} compared with those of R22 and R134a.

The relative exergy loss of the compression process ϕ_{12} calculated from Equation 28 is illustrated in Figure 9. From this figure, it is obvious that, R22 displays the highest compression loss compared with R12 and R134a. This is attributed to the following factors: (i) R22 operates with a lower critical temperature, thus it starts to evaporate with higher vapor quality and greater superheat, both tend in a larger superheat vapor horn. (ii) R22 operates with a higher adiabatic index γ_{R22}. This tends to raising the discharge temperature, and subsequently the entropy generation $\Delta S = S_2 - S_1$ is considerably increased. Unlike, refrigerants R12 and R134a, have relatively higher values of critical temperature and lower values of adiabatic index γ_{R12} and γ_{R134a}. (see Table 1). Both tend to reduce the superheat vapor horn, the discharge temperature, the entropy generation and finally the compression losses which are seemed to be competed for both refrigerants.

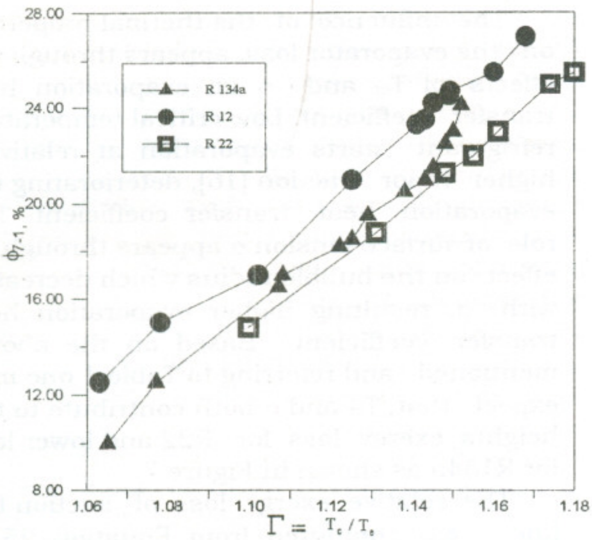

Figure 8 The relative exergy loss of suction line ϕ_{7-1} vs. the temperature ratio Γ

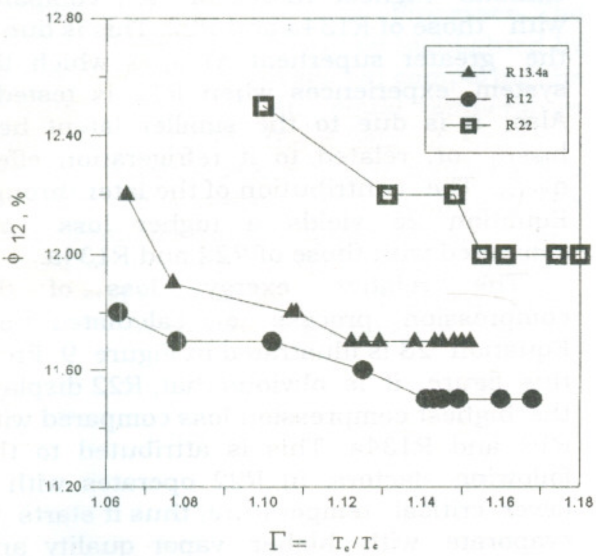

Figure 9 The relative exergy loss of compression process ϕ_{12} vs. the temperature ratio Γ

The relative exergy loss of the condenser is the sum of individual losses $\phi_{2-3} + \phi_{3-4} + \phi_{4-5}$ and is presented in Figure 10, for the three refrigerants under investigation. However, the calculation analysis indicates that, condensation loss ϕ_{3-4} calculated by Equation 34 is believed to be the most influential. The analysis of the obtained data indicates that, the relative condenser loss decreases inversely with the temperature ratio T_c/T_e, i.e. condenser temperature T_c. (at T_e = constant). This can be attributed to the contribution of the latent heat $h_{fg(c)}$, through Equation 34 which, decreases with the increase of condensation temperature. On comparing the obtained data, it is obvious that the highest exergy loss is belonging to R12. This is due to the poor condensation heat transfer coefficient $\alpha_{c(R12)}$. This coefficient is given in Reference 19, for condensation of refrigerants vapor at low velocities inside horizontal tubes as:

$$\alpha_c = 0.55[\rho(\rho - \rho_v)\, g\, k^3\, h_{fg}/\mu d(T_g - T_w)]^{1/4} \quad (40)$$

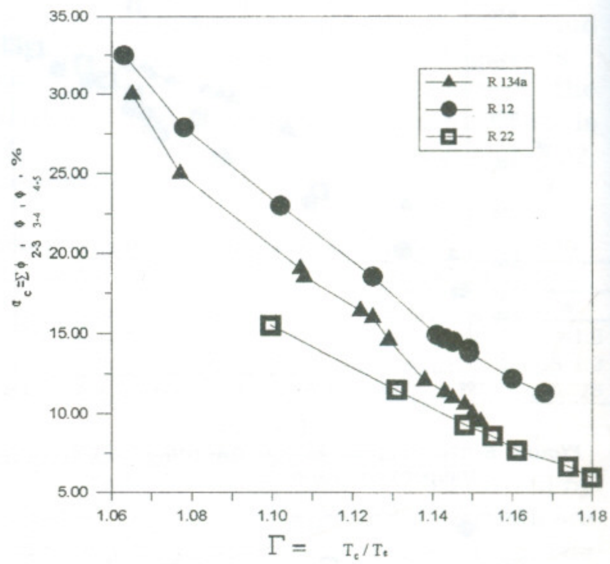

Figure 10 The relative exergy loss in the condenser ϕ_c vs. the temperature ratio Γ

Thus, if reference is made to Table1, one concludes that the effects of the latent heat h_{fg} and the thermal conductivity k through Equation 40, result in low value of $\alpha_{c\,(R12)}$ compared to those of R134a and R22. This in turn increases the refrigerant to coolant temperature difference and the entropy generation ΔS. This outcome is consistent, with the, findings reported by Koyama *et al.* [20].

The second law or exergetic efficiency η_{ex}; which is calculated from Equation 38, for the system employing R12 , R22 and

R134a is plotted versus the temperature ratio Γ in Figure 11.

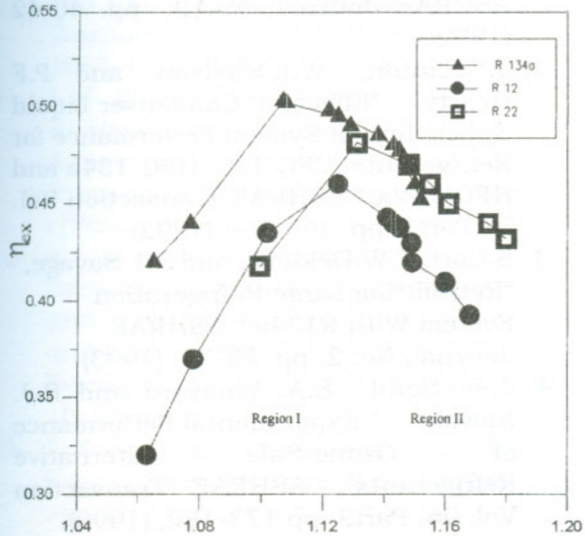

Figure 11 The second law or exegetic efficiency η_{ex}; vs. temperature ratio Γ

This figure shows that the exergetic efficiency η_{ex} increases with the ratio Γ in the region I. The reason lies in the faster increase of the reversible work than the actual input power. Thus, even though the COP drops with increasing ratio Γ, the system is still performing thermodynamically efficient relative to the overall temperature difference (T_c-T_e), across which it must transfer heat.

In region II with further increase in temperature ratio Γ i.e. increase in condenser temperature T_c and decrease in evaporator temperature T_e, both contribute to a higher input power and smaller refrigeration effect which finally resulted in lower exergetic efficiency η_{ex}. On comparing the obtained data, the system η_{ex} of R134a is about 20% and 8% higher than η_{ex} of R12 and R22, respectively. The exergy analysis showed that, the main sources of irreversibility are due to the heat transfer in the evaporator, condenser and suction line. However, exergetic efficiency can be increased for alternative refrigerants by improving the system design as for example, by employing cross-counter flow heat exchanger, liquid line to suction line heat exchanger and recirculated heat exchanger.

Based on the above given analysis HFC-R134a can be considered as the most suitable long term replacement for CFC-R12 and HCFC-R22, from point of view of the thermodynamic evaluation.

CONCLUSIONS

A typical cooling system was tested with R12, R22 and R134a, without any changes with exception of thermostatic expansion valves. The effects of the thermodynamic properties on the first law performance and the second law efficiency are the major concerns in this study. From the present results, the following conclusions can be drawn.

1. Evaluation based on the first law, indicated that the system COP_{ac} of R134a is 8% and 22% higher than the COP of R12 and R22 , respectively.
2. The second law analysis indicates that the components irreversibities vary considerably, such way that the maximum exergy losses for the expansion device, compressor condenser, evaporator and suction line are approximately,8% , 13%, 35%, 10 % and 35%, respectively. Thus, this analysis illuminates which measures should be taken, in order to improve the perfection of the system components.
3. Individual thermal property may determine the behavior of individual system component, but the global system perfection is determined by the integrated contributions of all properties involved in the system processes. The second law analysis indicates that, R134a appears to be meritorious, its exegetic efficiency is 20% and 8% higher than that of R12 and R22 respectively. Thus from standpoint of the thermodynamics HFC-R 134a can be considered as an appropriate substitute for CFC-R12 and HCFC R22.

NOMENCLATURE

C	constant, Equation 39
c	isobaric specific heat, [kJ/kg.K]
COP	coefficient of performance
d	diameter, [m].
E	exergy demand , [kW]
h	enthalpy, [kJ/kg]
h_{fg}	latent heat, [kJ/kg]
k	thermal conductivity, [W/m.K]
L	exergy loss, [kW]
m	refrigerant mass flow rate, [kg/s]
Q	refrigeration capacity ,[k.W]
q	heat per unit mass [kJ/kg]
S	specific entropy, [kJ/kg K].
t	temperature, [°C]
T	absolute temperature, [K]
w	specific work [kJ/kg]
Wcom	compression power, [kW]

Greek Symbols

α	heat transfer coefficient, [W/m^{2}K]
σ	liquid surface tension, [N/m]
ρ	density, [kg/m^3]
μ	dynamic viscosity, [kg/m.s]
γ	adiabatic index=c_p/c_v
ϕ	relative exergy loss, %
η_{ex}	exergetic efficiency, %
Γ	temperature ratio T_c/T_e
Π	temperature ratio T_{cr} / T_{sc}

Subscripts

a	ambient
ac	actual
be	before expansion
c	condensation, condenser
com	compressor, compression
cr	critical
ex	exergetic
f	saturated liquid
g	saturated vapor
m	mean
r	refrigerated room.
R12	refers to refrigerant-R12
R22	refers to refrigerant-R22
R134a	refers to refrigerant-R134a
s	super heated vapor.
sc	subcooling
th	theoretical
w	wall

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التقييم باستخدام القانونين الأول و الثانى للديناميكا الحرارية لنظام تبريد باستخدام الموائع CFC-R12., HCFC-R 22,HFC-R134a

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ملخص البحث

استنادا الى الخواص الثرموديناميكيه تم استخدام معامل الاداء للقانون الاول والفعاليه للقانون الثانى فى تقييم نظام تبريد باستخدام الموائع CFC-R12., HCFC-R 22,HFC-R134a . أظهر تحليل النتائج ان معامل الاداء للنظام بإستخدام مائع التبريد R134a اعلى من مثيله باستخدام الموائع R12، R-22 بحوالى ٨% و ٢٢% على التوالى .

أظهرت دراسة النتائج استنادا الى القانون الثانى ان فقد الفعاليه (الاكسيرجيا) تكون واضحه نتيجة انتقال الحراره فى المبخر ، المكثف ، خط السحب و كفاءة النظام باستخدام مائع R-134a أعلى بحوالى ٢٠% و ٨% عن مثيله باستخدام الموائع R-12،R-22 على التوالى.