Engineering e-Transaction (ISSN 1823-6379)
Vol. 5, No. 1, June 2010, pp 11-18
Online at http://ejum.fsktm.um.edu.my
Received 15 June 2010; accepted 23 July 2010

THERMODYNAMIC PERFORMANCE ANALYSIS OF R-600 AND R-600A AS REFRIGERANT

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ABSTRACT

In the developing countries like Malaysia, most of the vapor compression refrigeration system continues to run on halogenated refrigerants due to its excellent thermodynamic and thermo-physical properties. However, the halogenated refrigerants have adverse impacts such as ozone depletion potential and global warming potential. Hydrocarbons being natural fluid have drawn much attention of the scientists and researchers for the application as refrigerants. This paper presents a comparison of the energetic and exergetic performances of a domestic refrigerator using pure butane and isobutene as the refrigerant. The second law analysis such as exergy efficiency and exergy destruction, and coefficient of performance (COP) are investigated and were then compared with those of different refrigerants at varied operating conditions. Exergy efficiency of isobutene is found to be 50% higher than that of R-134a as a refrigerant mixture. The analyses show that the performances of butane and isobutene as refrigerants are comparable with HFC134a. It has also been found that at higher evaporating temperatures, the exergy losses are minimal for the refrigerants in the four components; the maximum exergy loss is occurred about 60% in the compressor.

Keywords: Hydrocarbons; Exergy loss; Exergy efficiency.

NOMENCLATURE

I = Exergy loss (kJ/sec)
COP = Coefficient of performance
h = Enthalpy (kJ/kg)
s = Entropy (kJ/kg.K)
T = Temperature (^{0}C)
Q = Heat transfer rate (kJ/sec)
W = Compressor work (kW)

Greek letters

Ψ = Specific Exergy (kJ/kg)
μ = Exergy Efficiency

1. INTRODUCTION

It is reported that Green house gas (GHG) from fossil fuel combustion for power generation and emission of halogenated refrigerants from vapor compression refrigeration system contribute significantly to the global warming. A reduction in GHG can only be achieved by using environment friendly and energy efficient refrigerant. Halogenated refrigerants have dominating the refrigeration and air conditioning industries over many decades due to its excellent thermodynamic and thermo physical properties. As per Montreal protocol 1987, developing countries are required to phase out all chlorofluorocarbons (CFC) by 2010 and all hydroflourocarbons (HCFCs) by 2040. Hydrocarbons can be used in the existing refrigeration equipment as a replacement for the conventional refrigerants and are compatible with mineral or synthetic oil. In some cases, no changes in the hardware configuration of the equipment are needed (UNDP, 1999). Hydrocarbon refrigerants were accepted before the introduction of CFC and HCFC-fluids came into picture (Granryd, 2001). Hydrocarbons are environmentally friendly and found to have zero Ozone Depletion Potential (ODP) and lower Global Warming Potential (GWP) too. Hydrocarbons are cheaper than R-134a and easily available. Most of the hydrocarbons offer good

miscibility with mineral oils and good compatibility with common materials employed in the refrigeration equipment. The thermo-physical properties of hydrocarbons are very similar to those of CFC refrigerants and also non-toxic and environmentally safe. Tashtoush *et al.,* (2002) and Sekhar and Lal, (2005) reported that ozone depletion potentials (ODPs) of hydrocarbons are very low ($<5.10^{-4}$), but the global warming potentials (GWPs) is quite high (GWP 1300). Thus still inconsistent reports on GWP of hydrocarbons are obtained.

There are many studies performed by many researchers about HCs mixture as refrigerants. Fatouh and Kafafy (2006) have evaluated the possibility of using HCs mixtures as working fluid to replace R134a in domestic refrigerators. In their simulation analysis, the performance characteristics of domestic refrigerators were predicted for various working fluids such as R134a, propane, commercial butane and propane/isobutene /n-butane mixtures with various propane mass fractions. They found that pure butane have low COP and high operating pressures. Wongwises *et al.* (2006) presented an experimental study on the application of HCs mixtures (propane, butane and isobutene) to replace HFC-134a in automotive air conditioners. They found that propane/butane/isobutene: 50/40/10% was the best alternative refrigerant to replace HFC -134a having the best performance of all other mixtures being investigated. Arcaklioglu *et al.* (2005), Jung *et al.* (2000) and Arcaklioglu (2004) reported that in case of using mixtures, there is a change in evaporator and condenser temperature during phase changing at constant pressure. This temperature gliding can be assumed as disadvantages for vapor compression system. However, this problem can be solved by using pure hydrocarbons.

The vapor compression refrigeration systems release large amount of heat to the surroundings. As a result of difference in temperatures between the system and the surrounding, irreversibility is taking place. This irreversibility degrades the performances of the system components. The losses in a component should be measured to improve the performance of the whole system. The losses in the cycle need to be evaluated considering individual thermodynamic processes that make up the cycle. In evaluating the efficiency of the vapor compression refrigeration system, the most commonly used term is the coefficient of performance (COP) which is known as first law of thermodynamics. But the first law of thermodynamics does not distinguish between heat and work. It cannot be used to identify the sources of thermodynamic losses in a thermodynamic cycle. First law gives no information on how, where and how much the system performance is degraded. On the other hand, second law of thermodynamics can be used to identify and quantify the thermodynamic losses in a cycle. Using the concept of irreversibility, thermodynamic losses (i.e. exergy losses in vapor compression refrigeration cycles) can be measured. Technically, the term ‘exergy’ can be defined according to thermodynamics principles as the maximum amount of work which can be produced by a system or a flow of matter or energy as it comes to equilibrium with a reference environment (Moran, 1989; Kotas, 1995; Moran and Sciubba, 1994; Szargut *et al.*, 1988; Szargut, 1980; Edgerton, 1992). Exergy is a measure of the potential of the system or flow to cause change, as a consequence of not being completely in equilibrium relative to the reference environment. Unlike energy, exergy is not subject to a conservation law (except for ideal processes). A system in complete equilibrium with its environment does not have any exergy. The exergy of a system increases the more it deviates from the environment. For instance, a specified quantity of hot water has higher exergy content during the winter than on a hot summer day. When energy loses its quality, exergy is destroyed. Exergy is the part of energy which is useful and therefore has its economic value. Exergy by definition depends not on the state of a system or flow but also on the state of the environment. The exergy analysis acknowledges that, although energy can not be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings. 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 (Kanoglu, 2002).

Therefore, exergy analysis identifies the margin available to design more efficient energy systems by

reducing inefficiencies. Exergy analysis permits many of the shortcomings of energy analysis to be overcome. Exergy analysis is useful in identifying the causes, locations and magnitudes of process inefficiencies. Exergy analysis acknowledges that, although energy cannot be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings and hence of no further use for performing tasks. The benefits of exergy analysis clearly go well beyond what many perceive to be the main application of the second law of thermodynamics, which forms the basis of exergy methods.

There has been little analysis about exergy for vapor compression refrigeration using pure hydrocarbons whereas in many investigations, hydrocarbons were found as acceptable refrigerant as an alternative to replace R-134a. Liedenfrost (1980) investigated performance of a refrigeration cycle using Freon as refrigerant on the exergy analysis. Bejan (1989) showed that the exergetic efficiencies decrease with the decrease of refrigeration temperature. He offered two models for explaining this trend. Thermodynamic imperfections were explained largely by the heat transfer irreversibility in those models.

Limited researches have been performed on exergy analysis of the individual components of a vapor compression refrigeration system using butane and isobutene as refrigerant. It is found that HCs have greater advantages on the basis of energy and other environmental impacts. Now it has become necessary to know the exergy performance as well as energy performance of butane and isobutene compared to existing refrigerant R134a. It is also found that a few analyses have been carried out about the effect of operating temperatures i.e. condenser and evaporator temperatures on exergetic performances but energetic performances of R-134a were examined in some experiments. In this study, energy and exergy performances of a refrigerator using R-134a were considered as a baseline and then the performances of a refrigerator using hydrocarbons specially butane and isobutene were estimated and compared with previous one. Exergy losses in the individual components were also investigated at different evaporating temperatures and condensing temperatures.

2. THEORITICAL ANALYSIS AND FORMULATIONS

Energy and exergy analyses need some mathematical formulations for the simple vapor compression refrigeration cycle. In the vapor compression system, there are four major components: evaporator, compressor, condenser and expansion valve. External energy (power) is supplied to the compressor and heat is added to the system in the evaporator whereas in the condenser heat rejection is occurred from the system. Heat rejection and heat addition are changed for different refrigerants which causes a change in energy efficiency for the refrigerants. Energy efficiency will be changed. Exergy losses in the different components of the system are not the same. Surrounding ambient temperature and pressure are denoted by T_0 and P_0 respectively. Exergy is consumed or destroyed due to the entropy created depending on the associated processes (Sahin *et al.,* 2005). To specify the exergy losses and destructions in the system, thermodynamic analysis is to be made. In this study, the following assumptions are made:

1. Steady state conditions are remaining in all the components.
2. Pressure loses in the pipelines are neglected.
3. Heat gains and heat loses from the system or to the system are not considered.
4. Kinetic and potential energy and exergy losses are not considered.

Schematic diagram of vapor compression refrigeration system is shown in Figure 1 and the relevant T-S diagram of the system is shown in Figure 2. States 1-2 represent isentropic compression in the compressor, states 2-3 represent the condensation i.e. heat rejection in the condenser, states 3-4 represent the throttling in the expansion valve and states 4-1 represent the evaporation in evaporator i.e. heat addition.

Mathematical formulation for exergy analysis in different components can be arranged in the following way (Bayrakci and Ozgur, 2009):

Specific Exergy in any state,

$$\Psi = (h - h_0) - T_0(s - s_0) \qquad (1)$$

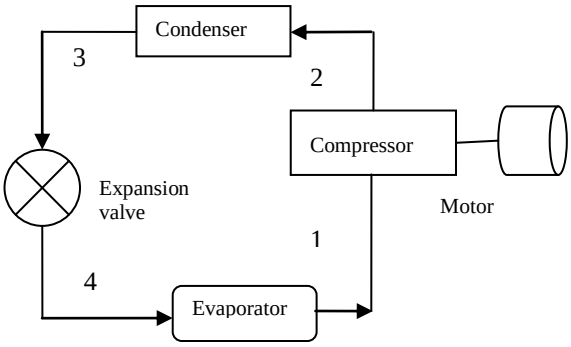

Fig. 1 Schematic diagram of simple vapor compression refrigeration system.

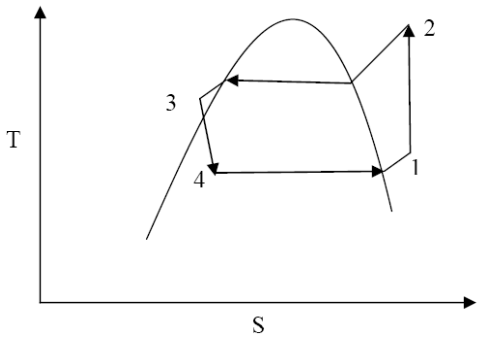

Fig. 2 T-S diagram of simple vapor compression refrigeration system.

For Evaporator:
Heat addition in evaporator,

$$Q = \dot{m}(h_1 - h_4) \quad (2)$$

Exergy losses,

$$I_{ev} = \dot{m}(\psi_4 - \psi_1) + Q\left(1 - \frac{T_0}{T_{ev}}\right)$$

$$= \dot{m}[(h_4 - h_1) - T_0(s_4 - s_1)] + Q\left(1 - \frac{T_0}{T_{ev}}\right) \quad (3)$$

For Compressor:
Compressor work, $W_c = \dot{m}(h_2 - h_1)$ (4)

For non-isentropic compression,

$$h_c = \frac{h_{2s} - h_2}{\eta_c} \quad (5)$$

Electrical Power, $W_{el} = \dfrac{W_c}{\eta_{mech} \times \eta_{el}}$ (6)

So, exergy loss,

$$I_{comp} = \dot{m}(\psi_1 - \psi_2) + W_{el}$$

$$= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)] + W_{el} \quad (7)$$

For Condenser:

$$Q_{cond} = \dot{m}(h_2 - h_3) \quad (8)$$

Exergy loss,

$$I_{cond} = \dot{m}(\psi_2 - \psi_3) - Q_{cond}(1 - \frac{T_o}{T_{cond}})$$

$$= \dot{m}(h_2 - h_4) - T_0(s_2 - s_3) - Q_{cond}(1 - \frac{T_o}{T_{cond}}) \quad (9)$$

For Expansion Valve:
Exergy destruction,

$$I_{\exp} = \dot{m}(\psi_4 - \psi_3)$$

$$= \dot{m}(s_4 - s_3) \text{ [Throttling, } h_4 = h_1\text{]} \quad (10)$$

Coefficient of performance,

$$COP = \frac{Q_{ev}}{W_{el}} \quad (11)$$

Total destruction,

$$I_{total} = I_{cond} + I_{\exp} + I_{comp} + I_{evap} \quad (12)$$

Exergy Efficiency,

$$\eta_x = \frac{\psi_1 - \psi_4}{W_{el}} \quad (13)$$

With reference to cited literatures, it is assumed that mechanical efficiency of the compressor is 90% and the electrical efficiency of the motor is 90%.

3. RESULTS AND DISCUSSIONS

In this section, thermodynamic performance parameters are calculated from the experimental data for the refrigerants using the different equations (1)-(13) and discussed. Here the results are compared and analyzed. The comparison of exergy destruction, exergy efficiency, energy consumption and performances are given below for each of the refrigerant used in the present study.

3.1 Sources of Data

Many researches are available for performance analysis of vapor compression refrigeration system. But a limited research is performed on exergy analysis of vapor compression refrigeration system. Some researchers use computer software for the second law analysis. Now a day, it is very important to know the thermodynamic performances of vapor compression system with the variation of evaporating temperatures and condensing temperatures. In this research necessary data has been taken from Sattar (2008). Equations (1) to (13) are used to calculate energy and exergy parameters using the data (Sattar, 2008). Here the mass flow rate of the working fluids is considered as unity. Properties of the refrigerants are obtained using software REPROF 7. The Reference temperature, T_0 is 25 ^{0}C and pressure, P_0 is 100 kPa were considered.

3.2 Variation of Work of Compression on Evaporating Temperature for Different Refrigerants

The energy uses by the compressor was measured and stored in a computer for 24 hours. The compressor consumes more energy when butane and isobutene are used as refrigerants instead of R-134a, shown in Figure 3. Conduction heat transfer plays an important role of electricity consumption in refrigerator. ASHRAE (1988) mentioned that most of the thermal load on a refrigerator is conduction through the refrigerator wall.

When the evaporating temperature increases the work of compression also decreases. Because the temperature difference between the internal compartment and the ambient is reduced. The higher the temperature difference, the higher the load imposed on a refrigerator. For this reason, the ambient temperature is a significant determinant of energy consumption. Since compressor efficiency also declines as the ambient temperature rises, a refrigerator's electricity use is very sensitive to the ambient temperature. Maclaine-Cross and Leonardi (1996) also found that refrigerator operated by hydrocarbon consumes less energy than refrigerator operated with R12 refrigerant.

The effect of evaporator temperature on work of compression is shown in Figure 3. The work of compression increases as the temperature of the evaporator decreases. This is due to the fact that when the temperature of the evaporator decreases the suction temperature also decreases. At low suction temperature, the vaporizing pressure is low and therefore the density of suction vapor entering the compressor is low. Hence the mass of refrigerant circulated through the compressor per unit time decreases with the decreases in suction temperature for a given piston displacement. Reduction of mass flow of refrigerant circulated increases the work of compression. The reason of high compressor work of this experiment is due to the low evaporating temperature.

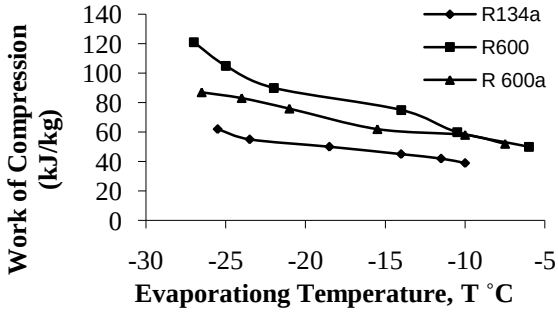

Fig. 3 Variation of power consumption at different evaporating temperatures for different refrigerants

3.3 Variation of Co-efficient of Performance on Evaporating Temperature for Different Refrigerants

The COP of the domestic refrigerator using R-134a as a refrigerant was considered as baseline and the COP of butane, and iso-butane were compared with it. The

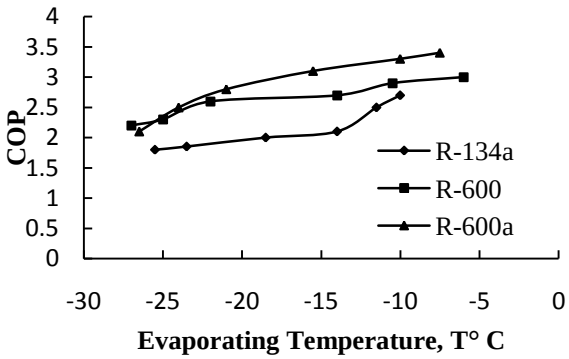

Fig. 4 Variation of coefficient of performance on different refrigerants at different evaporating temperatures.

effect of evaporator temperature on COP has been presented in Figure 4. The data represents a

progressive increase in COP with the increase of evaporating temperatures.

The refrigerating effect decreases with the decrease of evaporating temperature, whereas the compressor duty increases with the decrease of evaporating temperature. Therefore, the COP decreases with the decreases of evaporating temperature (ASHRAE, 1988). When R-134a is used, the calculated values of COP of the refrigerator are obtained between 1.8 and 2.40 at evaporating temperature ranges from -25.5 °C to -10 °C. The COP lies between 2.0 and 3.1 when isobutene is used at evaporating temperature ranges from -26 °C to -7.5 °C. When butane is used, the COP is obtained between 2.2 and 3.4 at evaporating temperature range of -26 °C to -6 °C. It is observed that the COP of the system is between 0.8 and 3.5 in freezing application at the evaporating temperature -16 °C to -6 °C. Among the refrigerants, R-600 has the highest performance.

3.4 Variation of Exergy Losses on Evaporating Temperature for Different Refrigerants

The exergy losses were measured for the vapor compression refrigeration system. The liquid refrigerant at low pressure side enters the evaporator. Exergy loss increases as the temperature of the evaporator decreases as shown in Figure 5. This can be explained that if the evaporating temperature increases the heat transfer between the refrigerant entered into the evaporator tubes and the medium being cooled also increases which ultimately increase the refrigerating effect thus the exergy loss decreases. Among the three refrigerants, isobutene exhibits minimum exergy loss.

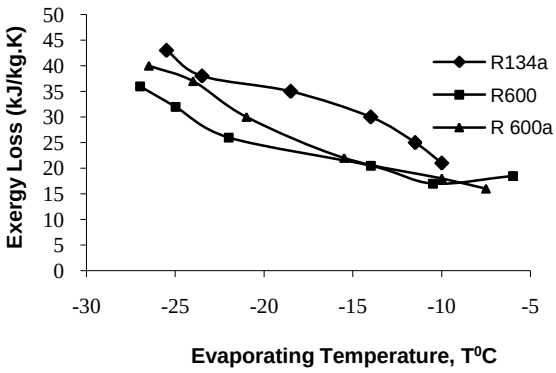

Fig. 5 Variation of exergy losses at different refrigerants at different evaporative temperatures.

At higher evaporating temperature, exergy loss is lower compared to that of at lower evaporating temperature. Vincent and Heun (2006) found that higher exergy destruction occurred in the compressor compared to condenser and other parts. They found that compressor has greater effect on the total exergy destruction.

3.5 Variation of Exergy Losses on Evaporating Temperature in Different Components

Exergy losses in the individual components for Refrigerant R-600a are shown in Figure 6 with the variation of evaporating temperatures. The trends of exergy losses in the different components of the vapor compression system for other refrigerants are also found similar. Greater portion of exergy losses take place in the compressor. Evaporator has lower exergy losses compared to the other components. Experimental results with other refrigerants also

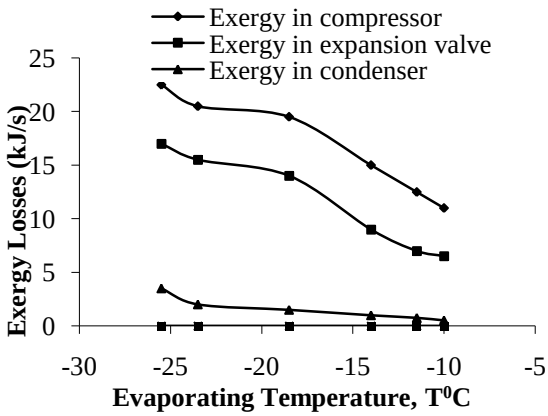

Fig. 6 Variation of Exergy losses at different evaporating temperatures using Refrigerant R-600a.

show the similar results i.e. compressor has the highest exergy losses compared to that of other components. The exergy losses in the components decrease with the increase of evaporating temperature. The higher the temperature differences in any component with the surroundings, the higher the exergy loss. Bayrakci and Ozgur (2009) studied about four different pure hydrocarbons (R290), butane (R600), isobutene (R600a) and isopentane (R1270) and also R22 and R134a. He found that R600 can be assumed as appropriate alternative to R22 and R134a. Yumrutas *et al.* (2002) studied the effect of evaporating and condenser temperature on exergy loss (los work). In the evaporator, the higher the temperature differences, the higher the exergy loss. Khan (1992) found that most of the irreversible losses are due to the low compressor efficiency and expansion process. These

losses increase with the increase in the difference between the condenser and evaporator temperatures. The performance results are provided for R-12, R-134a, R-22 and R-502 refrigeration systems.

3.6 Variation of Exergy Loss on Condensing Temperature for Different Refrigerants

It is shown in the Figure 8 that exergy losses are increased with the increase of condensing temperature for all the refrigerants. It is obvious because higher the temperature difference between the ambient and the component the higher the exergy losses. Availability of work also increased. In the low temperature region, exergy loss for each refrigerant remains same but in the high temperature region the loss for R-600 is increased. Because the more the difference in temperatures between the ambient (air) and the system (working fluid), the more the exergy losses. Chance of irreversibility increases for temperature rises.

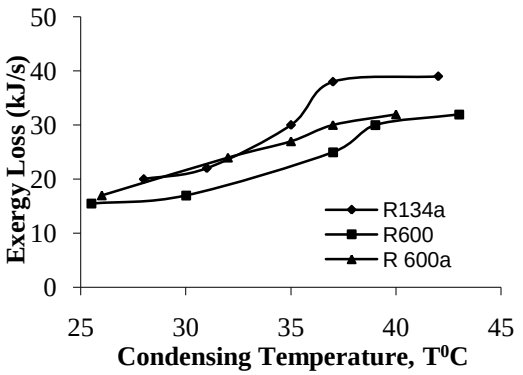

Fig. 7 Variation of exergy loss at different condensing temperature with different refrigerants.

3.7 Variation of Exergy Efficiency on Evaporating Temperature for Different Refrigerants

It is observed that with the increase of evaporating temperature, irreversibility also decreases for all the refrigerants. Among the considered working fluids, R-600 shows the best exergetic performance (Figure 8). Refrigerant R-134a has lower performances among the refrigerants. The function of the condenser is to remove the discharged heat of compressor and evaporator carried by the refrigerant Heat is discharged to the hot refrigerant during evaporation in the evaporator and by the compressor during work of compression. The heat from the hot refrigerant is removed by transferring heat to the wall of thee condenser tubes and then from the tubes to the condensing medium.

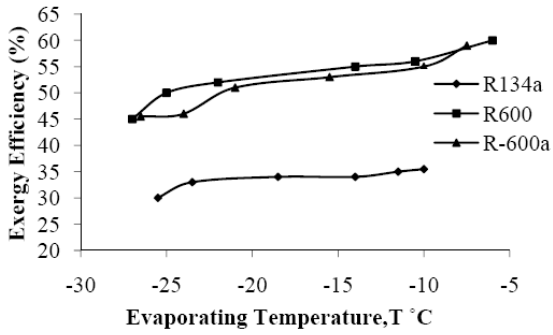

Fig. 8 Exergy efficiency with different refrigerant at different evaporating temperatures.

6. CONCLUSIONS

After the successful investigation of the HC as a refrigerant on the basis of performance, the following conclusions can be drawn based on the results obtained:

- Exergy loss for butane and isobutene are less than that of the refrigerant R134a in the present test unit. In the higher evaporating temperature exergy loss is decreased for all refrigerants.
- Exergy efficiency is also higher for butane compared to that of isobutene and R-134a as refrigerant.
- Exergy loss in the compressor is higher than that in the other parts of the system i.e. upto 60% of the total exergy loss occurs in the compressor.
- For higher temperature differences between evaporator and condenser the exergy losses are higher.
- The co-efficient of performances for the butane and isobutene are comparable with that of performance of R134a.

It can be inferred that, to improve the overall performance of the vapor compression system it is necessary to minimize the temperature difference between the evaporator and condenser. To improve the performance of the compressor, it is necessary to improve the motor efficiency and the lubricating system as well. For getting high COP and low exergy loss it is necessary to operate the system at low condensing and high evaporating temperature ranges.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the Fundamental Research Grant Scheme (FRGS), Malaysia for funding the project. The research has been carried out under the project FP 060/2008C.

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