Military Technical College
Kobry El-Kobba,
Cairo, Egypt

11-th International Conference
on Aerospace Sciences &
Aviation Technology

ASSESSMENT OF PROPANE/COMMERCIAL BUTANE MIXTURES AS POSSIBLE ALTERNATIVES TO R134a IN DOMESTIC REFRIGERATORS

M. Fatouh* and M. El Kafafy**

ABSTRACT

The possibility of using hydrocarbon mixtures as working fluids to replace R134a in domestic refrigerators has been evaluated through a simulation analysis in the present work. Performance characteristics of domestic refrigerators were predicted over a wide range of evaporation temperatures (-35 to -10°C) and condensation temperatures (40 to 60°C) for various working fluids such as R134a, propane, commercial butane and propane/iso-butane/n-butane mixture with various propane mass fractions. Performance characteristics of the considered domestic refrigerator was identified by the coefficient of performance (COP), volumetric cooling capacity, cooling capacity, condenser capacity, input power to the compressor, discharge temperature, pressure ratio and the refrigerant mass flow rate.

Results showed that pure propane could not be used as a drop-in replacement for R134a in domestic refrigerators because of its high operating pressures and low COP. Commercial butane yields many desrirable characteristics, but requires compressor change. Coefficient of performance of the domestic refrigerator using a ternary hydrocarbon mixture with propane mass fraction from 0.5 to 0.7 is higher than that of R134a. Comparison among the considered working fluids confirmed that average refrigerant mass flow rate of propoane/commercial butane mixture is 50% lqwer than that of R134a. Also, results indicated that R134a and propoane/commercial butane mixture of 60% propane mass concentration have approximately the same values of saturation pressure, compressor discharge temperature, condenser heat load, input power, cooling capacity and volumetric cooling capacity. However, the pressure ratio of the hydrocarbon mixture with 60% propane is lower than that of R134a by about 11.1%. Finally, the reported results confirmed that the propane/iso-butane/n-butane mixture with 60% propane is the best drop-in replacement for R134a in domestic refrigerators under normal, subtropical and tropical operating conditions.

KEYWORDS; Hydrocarbon mixtures, alternative refrigerants, performance characteristics, dmoestic refrigerator, R134a.

* Assoc. Professor, Dpt. of Mechanical Power Engineering , Faculty of Engineering at El-Mattaria (Helwan University), Masaken El-Helmia P. O., Cairo 11718, Egypt
* Corresponding author; drmohfat@hotmail.com - Fax no.: (00202) 6332398.

** Graduate student, Dpt. of Mechanical Power Engineering , Faculty of Engineering at El-Mattaria (Helwan University).

NOMENCLATURE

c	Specific heat (kJ/kg K)
C	Clearance ratio (-)
COP	Coefficient of performance (-)
h	Specific enthalpy (kJ/kg)
m	Mass flow rate (kg/s)
n	Specific heat ratio (-)
N	Compressor speed (rpm)
p	Pressure (MPa)
P	Power (kW)
PMF	Propane mass fraction (-)
PR	Pressure ratio (-)
Q	Heat transfer rate (kW)
Q_{vol}	Volumetric cooling capacity (kJ/m^3)
T	Temperature (°C)
v	Specific volume (m^3/kg)
V	Stroke volume (m^3)
η	Efficiency (-)
ρ	Density (kg/m^3)

Subscripts

1-4	Refers to Fig. (1)
cl	Clearance
com	Compressor
con	Condenser
cr	Critical
dis	Discharge
eva	Evaporator
is	Isentropic
mix	Mixrure
p	Constant pressure
r	Refrigerant
rel	Relative
sl	Saturated liquid
st	Stroke
v	Constant volume
vol	Volumetric

1. INTRODUCTION

R12 has been used for many decades as a working fluid in domestic refrigerators. It was found that R12 destroys the stratospheric ozone layer and also contributes significantly to the world's greenhouse warming problem. Table 1 indicates that ozone depletion potential (ODP) of R12 is 1.0 while its global warming potential (GWP) is 8500 times that of carbon dioxide over one hundred years. As a result, many environmental issues related to alternative refrigerants and energy efficiency are reported in literature. Cavallini [1] stated that R12 alternatives include pure refrigerants without chlorine atoms such as hydroflorocarbons (HFCs), mixture of environmentally friendly refrigerants and hydrocarbons refrigerants.

Number of investigators reported that GWP of HFCs is significant, though less than that of chlorofluorocarbons (CFCs). Kyoto protocol [2], which approved by many nations called for the reduction in emissions of greenhouse gases including HFCs. Hence, the adoption of R134a for refrigeration equipment has been resisted by environmental groups. Although hydrocarbon flammability has caused some concern, interest in the use of pure (propane and butane) or mixed hydrocarbons as refrigerants in domestic and commercial refrigerators has increased in the last decade [3-4]. Table 1 reveals that hydrocarbons have neither ozone depletion effect nor global warming effect [4-6]. However, they are cheap, plentiful and compatible with both mineral and synthetic oils [7].

Table 1 Environmental effects of some refrigerants [4-6]

Data	Refrigerant				
	R12	R134a	Propane (R290)	n-butane (R600)	iso-butane (R600a)
Natural	No	No	Yes	Yes	Yes
ODP	1	0.0	0.0	0.0	0.0
GWP, 100 years	8500	1300	0.0	0.0	0.0
ρ_{sl} (kg/m^3) at 25°C	1310.8	1206.35	492.65	532.45	550.65
Flammability limits Vol. %	None	None	2.1-11.4	1.7-10.3	1.9-10.0
Molecular mass (kg/kmol)	120.93	102.03	44.1	58.1	58.1

Richardson and Butterworth [8] reported that the amount of charge associated with hydrocarbons was roughly half that of R12 in refrigerators. Also, tests conducted indicated that the hydrocarbons are quite safe in domestic refrigerators due to the very small amounts involved [9]. Fire and explosion data of hydrocarbon substances are given in Table 1.

In order to overcome difficulties of HFC refrigerants, massive work has been conducted to evaluate theoretical performance characteristics of domestic refrigerator working with pure or mixed hydrocarbon refrigerants. Agarwal [10] compared R134a with pure iso-butane and mixture of propane and isobutane using the theoretical refrigeration cycle for evaporation temperature of –25°C and condensation temperature of 55°C. The reported results indicated that iso-butane offers slightly lower discharge temperature and higher COP but lower volumetric capacity than R134a. Macline-cross and Leonardi [11] showed that R600a refrigerators have electricity savings up to 20% over that of R134a and R12 refrigerators. They also confirmed that hydrocarbon refrigerants have environmental advantages and still safe for small quantities. Thus, hydrocarbon mixtures can replace R12 and R134a in applications using positive displacement compressors. Maclaine-cross [12] proposed a hydrocarbon blend of propane (R290) and iso-butane (R600a) to avoid stratospheric ozone depletion and a typical 15% increase in Total Equivalent Warmming Impact (TEWI) from R134a leakage and service emissions. His measurements suggested that propane (R290) and iso-butane (R600a) mixture with mass concentration ratio of 55/45 matches the performance of R134a.

It is worth to mention that liquefied petroleum gas is cheap and available in many countries over the world [13]. It offers many advantages such as its local availability and low price compared to other refrigerants. It is a hydrocarbon mixture and its constituents differ from a country to another. Alsaad and Hammad [14] examined a possipility of using locally available liquefied pertrollum gas, which comprises 24.4% propane, 56.4% n-butane and 17.2% iso-butane, as a working fluid in domestic refrigerators. Akash and Said [15] evaluated performance characteristics of a domestic refigerators using liquefied petroleum gas of 30% propane, 55% n-butane and 15% iso-butane as a drop-in replacement for R12. According to Egyptian standard [16], Egyptian liquefied petroleum gas is mainly a mixture of propane/iso-butane/n-butane in which iso-butane to n-butane mass ratio is about two. It can be produced with various propane concentration ratio. Properties of the liquefied petroleum gas are given in Table 2. Since each element of this mixture has zero ozone depletion potential and global warming potential, as listed in Table 1, the mixture appears to be promising as a replacement for R134a with respect to environmental concern. Although liquefied petroleum gas was investigated at a fixed propane mass fraction and fixed condensation and evaporation temperatures, this hydrocarbon mixture needs to be examined over a wide range of propane mass fractions and operating temperatures.

Literature survey revealed that many investigations were carried out on domestic refrigerators working with either pure hydrocarbon or propane/iso-butane mixture as a drop-in replacement for R134a. However, the possibility of using propane/iso-butane/n-butane mixture to replace R134a in domestic refrigerators need to be investigated under different operating conditions. Thus, the present work aimed at evaluating and comparing the performance characteristics of a vapor compression domestic refrigerator working with R134a and a ternary (propane/iso-butane/n-butane) mixture with different propane mass fractions over a wide range of evaporation and condensation temperatures. In order to achieve these objectives, the domestic refrigerator has been simulated and a computer programe has been developed, in which the input data are evaporation temperatures, condensation temperatures, compressor specifications and various working fluids with different mass fractions of propane in the considered ternary hydrocarbon mixture. The program outputs are performance characteristics of the system, refrigerant mass flow rates and refrigerant properties at all state points of the cycle.

Table 2 Properties of propane/commercial butane mixture [17]

Propane (Vol. %)	100	80	60	50	40	20	0
Commercial butane (Vol. %)	0	20	40	50	60	80	100
Boiling point (°C) at 1 atm	-42	- 34	-26	- 22	- 18	- 10	- 2
p_{cr} (bar)	43	42	40	40	39	38	37
T_{cr} (°C)	96	106	116	121	126	136	143
P (bar) at 50°C	17.5	15.1	12.7	11.5	10.3	7.9	5.6
ρ (kg /m³) at 50°C	452	469	486	494	503	520	536
Latent heat (kJ/kg):							
at 1atm	427	423	410	406	402	393	381
at 15°C	352	352	356	356	356	360	360
Specific heat (kJ/kg °C):							
c_p at sea level	1.607	1.612	١,٦٢٠	1.628	1.624	1.633	١,٦٣٧
c_v at sea level	1.419	1.432	1448	1.453	1.461	1.473	1.4٩٠
c_{sl} (kJ/kg °C) at 15°C	2.84	2.67	2.6٤	2.6٤	2.59	2.55	2.51

2. System Analysis

Figure (1.a) shows a schematic diagram of a vapor compression refrigerator, which consists essentially of a hermetic resciprocating compressor, an evaporator, an air cooled condenser and a capillary tube. These components are connected together by pipelines in which a refrigerant with suitable thermodynamic properties circulates. The corresponding pressure–enthalpy (p-h) diagram is shown in Fig. (1.b).

In order to simulate the vapor compression refrigerator, a number of assumptions are made. These are (a) steady state operation, (b) no pressure loss through pipelines, i.e. pressure changes only through the compressor and capillary tube; (c) heat losses or heat gains from or to the system are neglected; (d) no superheating or subcooling takes place and (e) the compressor has ideal volumetric efficiency and isentropic efficiency of 75% [18].

In order to accept a drop-in replacement for a working fluid in an existing domestic refrigerator, some important performance characteristics have to be considered. These are; operating pressures, volumetric cooling capacity, coefficient of performance (COP) and compressor discharge temperature.

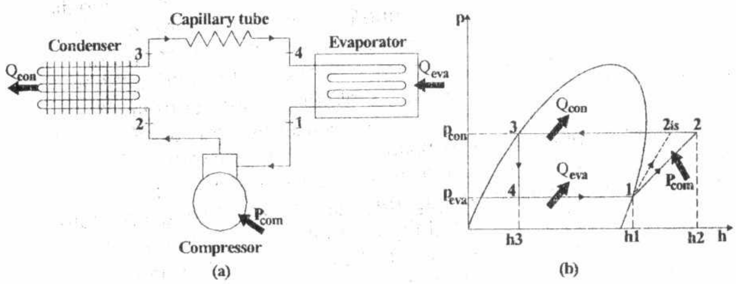

Fig. (1) Single-stage vapor compression refrigerator; (a) Schematic diagram, (b) p-h diagram

Volumetric cooling capacity (Q_{vol}) is a measure of the size of the compressor required for particular operating conditions. Similar volumetric cooling capacities require no change in compressor size. It should be noted that as the volumetric capacity of the refrigerant increases, the size of the compressor reduces. The volumetric cooling capacity (Q_{vol}) can be estimated as follows;

$$Q_{vol} = \frac{(h_1 - h_4)\eta_{vol,cl}}{v_1} \tag{1}$$

h_1 and v_1 are the specific enthalpy and specific volume of the saturated vapor refrigerant at the compressor inlet and h_4 is the specific enthalpy of the refrigerant at the evaporator inlet. Clearance or ideal volumetric efficiency ($\eta_{vol,cl}$) is given by

$$\eta_{vol,cl} = 1 - C(PR^{1/n} - 1) \tag{2}$$

C is the compressor clearance ratio (V_c/V_{st}) and n is the polytropic index. V_c and V_{st} are the clearance and stroke volumes, respectively. It should be stated that for the considered doemstic refrigerator, C and V_{st} are 0.04 and 6.64 cm^3, respectively, according to compressor manfacturer data [19]. Pressure ratio (PR) is defined as the ratio of the condensing pressure (p_{con})to the evaporating pressure (p_{eva}), i.e.

$$PR = \frac{p_{con}}{p_{eva}} \tag{3}$$

The condensing and evaporating pressures are determined corresponding to the condensation and evaporation temperatures, respectively. The condensation temperature is decided by the temperature of the ambient air temperature according to tropical, subtropical and normal operating conditions whereas the evaporation temperature is determined by the load temperature based on the required freezer air temperature.

The coefficient of performance (COP) relates the cooling capacity to the required power and indicates the overall power consumption for a desired load. High COP means low energy consumption to absorb the same cooling capacity from the space to be cooled. It can be expressed as:

$$COP = \frac{Q_{eva}}{P_{com}} \tag{4}$$

Where Q_{eva} is the cooling capacity and P_{com} is the input power required to drive the compressor. Energy balance of the evaporator gives:

$$Q_{eva} = m_r(h_1 - h_4) \tag{5}$$

Input power required to drive the compressor can be written as;

$$P_{com} = m_r(h_2 - h_1) \tag{6}$$

Actual specific enthalpy of the superheated vapor refrigerant at the compressor exit (h_2) can be calculated as follows:

$$h_2 = h_1 + \frac{(h_{2,is} - h_1)}{\eta_{is,com}} \tag{7}$$

Where $\eta_{is,com}$ is the isentropic compressor efficiency and $h_{2,is}$ is the superheated vapor enthalpy at the compressor exit for isentropic compression process. The refrigerant mass flow rate (m_r) can be estimated using the following equation:

$$m_r = \frac{V_{st}\ N\ \eta_{vol,cl}}{v_1} \tag{8}$$

N is the compressor speed. Compressor discharge temperature ($T_{dis}=T_2$) is an important parameter because of its effect on compressor components and stability of lubricants. This temperature can be determined using both condensation pressure and the actual specific enthalpy at the compressor outlet. In order to compare various refrigerants as working fluids in domestic refrigerators, knowledge of their thermodynamic properties is required. In the present work, thermodynamic data from SUPERTRAPP [20] for all refrigerants under consideration were used.

3. Results and Discussions

The following operating conditions were considered in the present work for R134a and a blend of propane (R290), iso-butane (R600a) and n-butane (R600) with propane mass concentration ratio from 0.0 to 1.0;

- Condensation temperature of 40, 50 and 60°C which simualtes normal, subtropical and tropical conditions, respectively, according to ISO standard [21], and
- Evaporation temperatures from -35 to -10°C which covers the required freezer air temperature in single, double and triple stars refrigerators.

Effects of refrigerant type, propane mass fraction, condensation temperature and evaopration temperature on performance characteristics of the domestic refrigerator are discussed in the next sections.

3.1. Effect of Propane Mass Fraction

Variation of thermodynamic properties for the ternary hydrocarbon blend with propane mass fraction (PMF) is given in Table 3 which reveals that the condensation and evaporation pressures increase with propane mass fraction while the specific volume of saturated vapor at the compressor inlet deacreases as propane mass fraction increases. This is because the specific volume of pure propane (PMF=1) is lower than that of pure commercial butane (PMF=0) at the same operating temperature. Table 3 confirms that as propane mass fraction increases, h_1 and h_2 increase then decrease while h_3 decreases then increases. Thus, cooling effect (h_1-h_4), enthalpy difference through the condenser (h_2-h_3) and specific work (h_2-h_1) increase then decrease when propane masss fraction increases.

Table 3 Variation of thermodynamic properties for the ternary hydrocarbon mixture with propane mass fraction

PMF	T_{con} (°C)	P_{con} (kPa)	T_{eva} (°C)	P_{eva} (kPa)	h_1 (kJ/kg)	h_2 (kJ/kg)	h_4=h_3 (kJ/kg)	v_1 (m^3/kg)
0	40	426.1	-25	42	430	564	185	0.828
0.2	40	571.5	-25	57.1	437	580.1	178	0.650
0.4	40	728.1	-25	78.2	442.4	589.6	177.5	0.504
0.6	40	908.7	-25	107.2	444.8	593	182	0.386
0.8	40	1113.2	-25	144.6	445.4	592.3	190.3	0.298
1	40	1369	-25	203.3	441	579.9	203.7	0.216

Influence of propane mass fraction on performance characteristics of the domestic refrigerators can be predicted from Fig. (2). The effect of PMF on pressure ratio is shown in Fig. (2.A) and can be explained with reference to Table 3, when propane mass fraction increases, both evaporation pressure (at -25°C) and condensation pressure (at 40°C) increase. But the former increases largely than the latter, thereby pressure ratio decreases when propane mass fraction increases. Thus, volumteric efficiency increases with propane mass fraction, Eq. (2). The refrigerant mass flow rate increases as propane mass fraction increases as indicated in Fig. (2.B). This is due to the combined effect of decrease in specific volume of vapor refrigerant at the compressor inlet (v_1) and increase in volumtric efficiency. Clearly, as propane mass fraction varies from zero to one, average pressure ratio decreases by about 35% while average refrigerant mass flow rate increases by four folds or higher.

The relation between volumetric cooling capacity and propane mass fraction is presented in Fig. (2.C), which confirms that the volumetric cooling capacity increases with propane mass fraction. This is due to the decrease in the specific volume as indicated in Table 3 and the increase in the volumetric efficiency. It is evident from this figure that average volumteric cooling capacity at PMF=1 is 4.5 times higher than that at PMF=0. Figure (2.D) shows variations of coefficient of performance with propane mass fraction. It can be seen that coefficint of perormance slightly decreases as propane mass fraction increases. The main reason behined this trend is the rate of change in the specific work (h_2-h_1) which is higher than that of the cooling effect (h_1-h_4), Table 3. Thus, average COP reduces by approximately 10% when propane mass fraction rises from 0 to 100%.

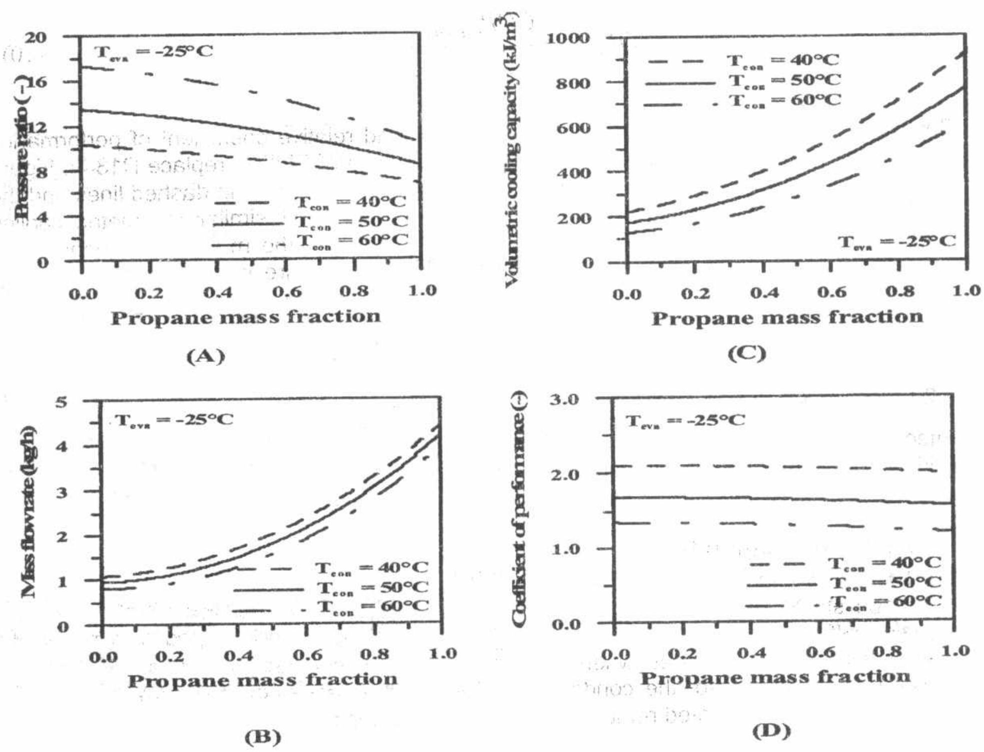

Fig. (2) Effect of propane mass fraction and condensation temperature on performance chatarteictics of domestic refigerators using propane/commercial butane mixtures

In conclusion, cooling effect (h_1-h_4), enthalpy difference through the condenser (h_2-h_3) and specific work (h_2-h_1) increase then decrease whereas specific volume at the compressor inlet (v_1) largely decreases as propane mass fraction increases. In average, pressure ratio and COP are reduced by 35% and 10%, respectively, as propane mass fraction increases from zero to one. Refrigerant mass flow rate and volumetric cooling capacity increase by more than four folds when mass fration of propane rises from 0 to 1.

3.1.1 Proper propane mass fraction for R134a replacement

In order to accept a working fluid as a drop-in replacement for existing refrigerant in a domestic refigerator, similar volumetric cooling capacity and similar or higher COP compared to those of replacing refrigerant are required. Similar volumetric cooling capacity requires no change in compressor size while high COP means low energy consumption to absorb the same cooling capacity from the space to be cooled. It is seen from previous section that the volumetric cooling capacity increases while the coefficient of performance decreases when propane mass concentration ratio in the hydrocarbon mixture increases. In order to determine the best propane mass fraction in a hydrocarbon mixture to be used as R134a substitute, relative volumteric cooling capacity and realtive coefficient of performance are introduced and given by Eqs. (8) and (9), respectively.

$$Q_{rel} = \frac{Q_{vol,mix}}{Q_{vol,R134a}} \tag{9}$$

$$COP_{rel} = \frac{COP_{vol,mix}}{COP_{vol,R134a}} \tag{10}$$

Consequentaly, relative volumteric cooling capacity and relative coefficient of performance can be plotted on the same graph to get the proper value of PMF to replace R134a. Figure (3) is such a plot with the relative volumetric cooling capacity shown in dashed lines and the relative COP shown in solid lines. Clearly, the right point of similar volumetric cooling capacity is located near to propane mass fraction of 0.6 in the mixture of propane, iso-butane and n-butane at which COP of the hydrocarbon mixture is approximatly equal or slightly higher than that of R134a over the whole range of condensation and evaporation temperatures. Hence, the ternary hydrocarbon mixture with propane mass fractions of 0.5, 0.6 and 0.7 are subjected to further investigation and comparison.

3.2 Effect of Operating Temperatures

Effects of operating temeratures on performance characteristics of the domestic refrigerator working with either R134a or a ternary hydrocarbon mixture were investigated over a wide range of condensation temperatures (40 to 60°C) and evaporation temperatures (-35 to -10°C).

3.2.1 Condensation temperature

It may be stated that as condensation temperature increases, cooling effect (h_1-h_4) and enthalpy difference through the condenser (h_2-h_3) decrease while specific work of the compressor (h_2-h_1) increases when propane mass fraction and evaporation temperature were kept constant. Also, the condensation temperature does not have any effect on the specific volume of saturated refrigerant at the compressor inlet.

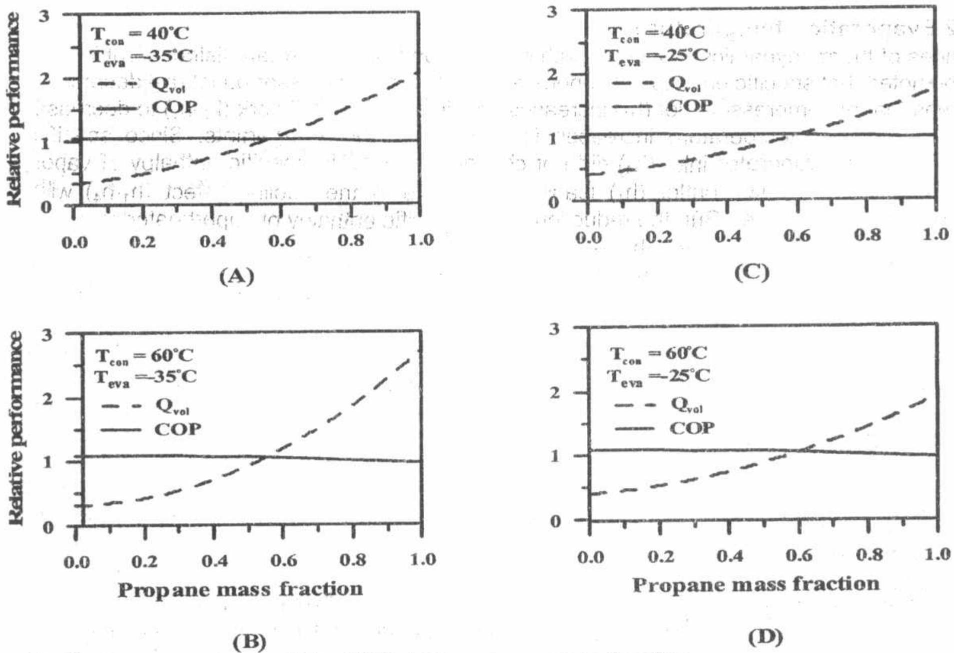

Fig. (3) Proper propane mass fraction

Effect of condensation tempertaure on the performance characteristics of domestic refrigerators working with propane/iso-butane/n-butane mixture can be investigated from Fig. (2). Increase in condensation pressure when evaporation pressure and mass farction of propane were kept constant leads to increase in pressure ratio with condensation temperature as shown in Fig. (2.A). This leads to a reduction in the volumetric efficiency (refer to Eq. (2)) and refrigerant mass flow rate as illustrated in Fig. (2.B). Consequentaly, this decrease in the refrigerant mass flow rate along with the decrease in the cooling effect as mentioned earlier cause the cooling capacity (Q_{eva}) to decrease as condensation temperature increases, see Eq. (5). Clearly, pressure ratio at condensation temperatures of 50°C and 60°C are higher than that at condensation temperature of 40°C by about 30.5% and 67.6%, respecively, when zero propane mass fraction was considered. These values become 25.1% and 54.8% at propane mass fraction of unity. At zero propane mass fraction, condensation temperatures of 50°C and 60°C yield refrigerant mass flow rates lower than those at condensation tempearture of 40°C by about 11.5% and 25.1%, respectively, while at propane mass fraction of unity, the corresponding reduction in refrigerant mass flow rate are about 5.8% and 12.1 %, respectively.

Figure (2.C) confirms that volumetric cooling capacities decrease as condensation temperature increases because of the decrease in both cooling effect (h_1-h_4) and volumeteric efficiency. Figure (2.D) indicates that high condensation temperature yields low coefficient of performance. This can be attributed to decrease in cooling effect and increase in specific work with condensation temperature as mentioned earlier. Both average volumetric cooling capacity and average coefficient of perfromance reduce by about 38% as condensation temperature rises from 40°C to 60°C and by about 20% when condensation temperature increases from 40°C to 50°C. In summary, pressure ratio increases while refrigerant mass flow rate, volumetric cooling capacity and COP reduce as condensation temperature increases when propane mass fraction and evaporation temperature are kept constant.

3.2.2 Evaporation temperature

Changes of thermodynamics properties with evaporation temperature are listed in Table 4. It can be noted that specific enthalpy of superheated vapor at compressor outlet (h_2) decreases whereas that at compressor inlet (h_1) increases causing the specific work (h_2-h_1) to decrease when evaporation temperature increases for the considered refrigerants. Since specific enthalpy at the evaporator inlet (h_4) did not change, increase in specific enthalpy of vapor refrigerant at evaporator outlet (h_1) leads to increase in the cooling effect (h_1-h_4) with evaporation temperature. But, the reduction in the specific enthalpy of superheated vapor at compressor outlet (h_2) causes the enthalpy difference through the condenser (h_2-h_3) to decrease as the evaporation temperature rises. It is evident from Table 4 that discharge temperature and the specific volume of vapor refrigerant at the compressor inlet are inversely proportional with evaporation temperature.

Influence of evaporation temperature on performance characteristics of the domestic refrigerator can be examined from Figs. (4-7). Figure (4) illustrates the variation of saturation pressure as a function of evaporation temperature for R134a and the ternary hydrocarbon mixture refrigerants for various propane mass fractions. Clearly, saturation pressure is directly proportional with saturation temperature. For constant condensation temperature, increase in evaporation temperature causes evaporation pressure to increase, thereby pressure ratio decreases as shown in Fig. (5A). This reduction in pressure ratio with evaporation temperature causes the volumetric efficiency to increase with evaporating temperature, Eq.(2).

With reference to Eq. (8), increase in the evaporating temperature causes both specific volume of vapor refrigerant at the compressor inlet (v_1) to decrease and volumetric efficiency to increase. As a result, the refrigerant mass flow rate increases with evaporating temperature as shown in Fig. (5B).

Variations of condenser heat load against evaporation temperature are presented in Fig. (5C). Condenser heat load was obtained by multiplying refrigerant mass flow rate into the enthalpy difference through the condenser. It is clear that condenser heat load increases as evaporation temperature increases. This is maily due to increase in the mass flow rate as stated earlier. Changes of compressor input power with evaporation temperature are illustrated in Fig. (5D), which reveals that compressor input power increases as evaporation temperature increases. This increase in compressor input power is mainly due to increase in mass flow rate when evaporating temperatures increase as shown in Fig. (5B).

Table 4 Effect of evaporation temperature on thermodynamic properties of the investigated refrigerants at condensation temperature of 50°C

Refrigerant	T_{eva} (°C)	P_{eva} (kPa)	h_1 (kJ/kg)	h_2 (kJ/kg)	h_4=h_3 (kJ/kg)	v_1 (m^3/kg)	T_2 (°C)
PMF= 0.0	-35	25.7	416.4	570.2	210.9	1.30E+00	68.1
	-25	41.9	430	560.3	210.9	8.29E-01	63.2
	-15	65.3	443.8	552.5	210.9	5.49E-01	59.4
PMF=0.6	-35	66.9	431.1	602	210.2	6.00E-01	84.6
	-25	105.3	444.3	588	210.2	3.93E-01	78.1
	-15	158.9	457.4	576.5	210.2	2.67E-01	72.8
PMF=1.0	-35	137.1	429.2	588.6	233.1	3.12E-01	80.4
	-25	203.3	441	575	233.1	2.16E-01	74.4
	-15	291.5	452.6	563.5	233.1	1.54E-01	69.4
R134a	-35	66.1	229	312.8	123.5	2.84E-01	83
	-25	106.4	235.3	305.6	123.5	1.82E-01	76.4
	-15	163.9	241.5	299.6	123.5	1.21E-01	70.9

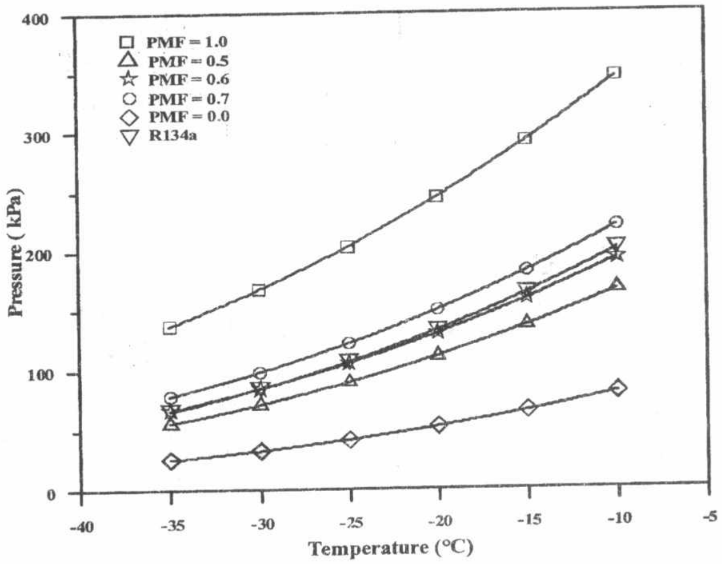

Fig. (4) Comparison of pressure for the considered refrigerants.

Volumetric cooling capacity as a function of evaporation temperatures is presented in Fig. (6). Clearly, volumetric cooling capacity increases with evaporating temperature. This is due to the increase in cooling effect and decrease in specific volume as shown in Table 4. Coefficient of performance (COP) as a function of evaporating temperature is illustrated in Fig. (7), which shows that coefficient of performance increases with evaporation temperature. It is mentioned earlier that as evaporation temperature increases, compressor input power (Fig. (5D)) and cooling capacity (Eq.(5)) increase. But the increase in cooling capacity is larger than that in compressor input power, thereby, COP increases with evaporation temperature.

3.3 Comparison Among Investigated Refrigerants

Comparion of thermodynaic properties for the considered working fluids can be made with refernce to Table 4 which reveals that specific volume of pure commercial butane (PMF=0) is the highest one over the entire range of evaporation temperatures. The specific volume of hydrocarbon blend with propane mass fraction of 0.6 is higher than that of R134a by a factor of 2.16. Among the considered refrigerants, commercial butane yields the lowest discharge temperature. Hydrocarbon blend with propane mass fractions of 0.6 produces higher discharge temperature than that of R134a by about 2°C over the considered range of operating conditions. Clearly, cooling effect (h_1-h_4) of hydrocarbon mixtures is higher than that of R134a. The cooling effect of hydrocarbon mixture with propane mass fractions of 0.6 is higher than that of R134a by a factor of 2 or higher . Enthalpy difference through the condenser (h_2-h_3) using R134a is lower than that of the hydrocarbon blend. The enthalpy difference through the condenser of the hydrocarbon mixtures with propane mass fractions of 60% is approximately double that of R134a.

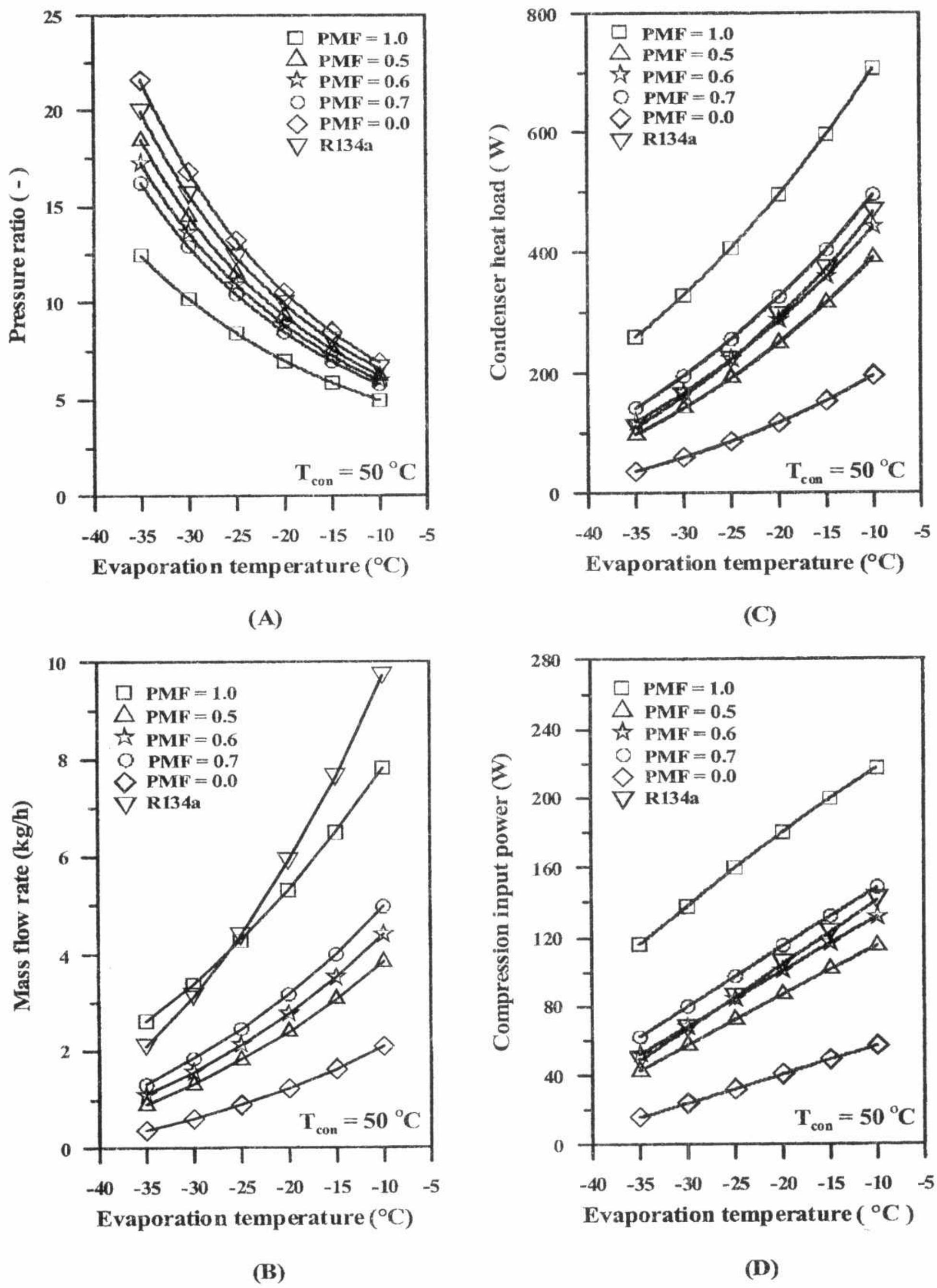

Fig. (5) Comparison of pressure ratio, mass flow rate, condenser heat loads and compression input power for the considered refrigerants

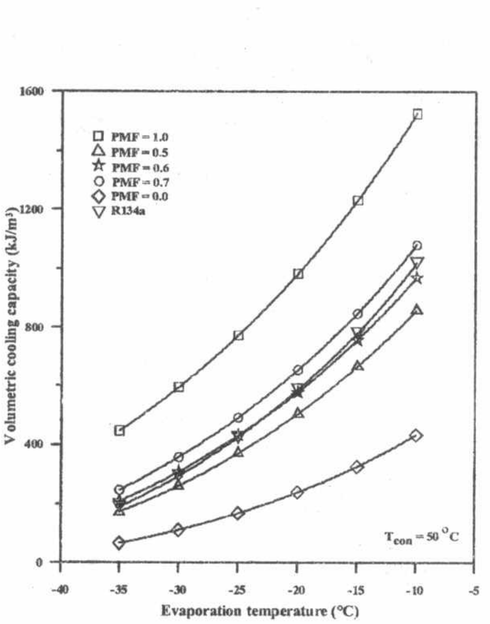

Fig. (6) Comparison of volumetric cooling capacity for the considered refrigerants

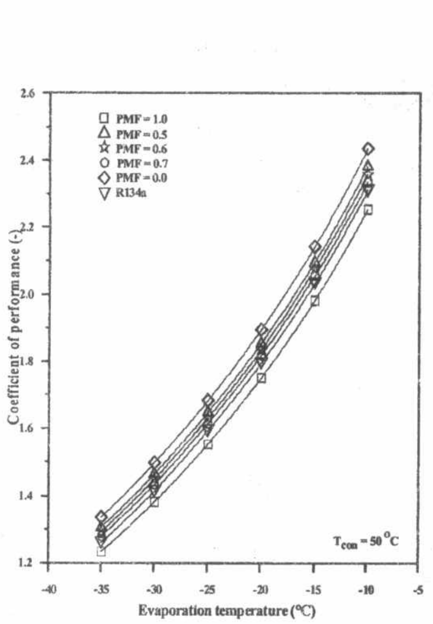

Fig. (7) Comparison of COP for the considered refrigerants

Comparison of performance characteristics of the domestic refrigerators working with various refrigerants namely; R134a, propane, commercial butane and the ternary hydrocarbon mixture with propane mass fractions of 0.5, 0.6 and 0.7 can be concluded from Figs. (4-7). Figure (4) indicates that pure propane (PMF=1) has the highest saturation pressure while pure commercial butane (PMF=0) has the lowest one. This means that pure propane and commercial butane need thick and thin wall tube, respectively, thereby increase in the capital cost for pure propane unit may be expected. Hence, pure propane could not be used as a drop-in replacement for R134a from operating pressure point of view. The ternary hydrocarbon mixture of 60% propane yields approximately the same saturation pressure of R134a over the considered range of evaporation temperature.

Figure (5A) reveals that pure commercial butane and propane yield the highest and lowest pressure ratio, respectively. Average pressure ratio of the hydrocarbon mixture with propane mass fraction of 50%, 60% and 70% is lower than that of R134a by about 6.3%, 11.1% and 15.3%, respectively. For condensation temperature of 50°C, pressure ratio at evaporation temperature of -10°C is approximately 0.32, 0.40, 0.36, 0.35, 0.32 and 0.32 of that at evaporation temperature of -35°C for R134a, pure propane, PMF=0.7, PMF=0.6, PMF=0.5 and commercial butane, respectively. This means that change in pressure ratio with evaporation temperature for R134a, PMF=0.5 and commercial butane is higher than that of pure propane, PMF=0.7 and PMF=0.6.

Among the considered refrigerants, the lowest mass flow rates can be obtained using the commercial butane as illustrated in Fig. (5B). Change in refrigerant mass is of order 5.45, 4.51, 4.2, 3.95, 3.11, 5.0 for R134a, pure propane, PMF=0.7, PMF=0.6, PMF=0.5 and pure commercial butane, respectively as evaporation temperature varies from -35°C to -10°C. Thus, the effect of evaporation temperature on refrigerant mass flow rate is large for R134a and small for the ternary hudrocarbon of propane mass fraction of 0.5. Average value of refrigerant mass flow rate of the hydrocarbon blend with 50%, 60% and 70% propane are lower than that of R134a by about 58.9%, 51.7% and 44.2%, respectively.

Although the ternary hydrocarbon mixture has high enthalpy difference (h_2-h_3) through condenser (Table 4), there is a small diference in condenser heat load between the hydrocarbon mixture and R134a as shown in Fig. (5C), which reveals that there is a large diference in condenser heat load between the pure hydrocarbon refrigerants and R134a. When evaporation temperature varies from -35°C to -10°C, condenser heat load increases by a factor of 4.57, 2.87, 3.63, 3.87, 4.17 and 5.14 for R134a, pure propane, PMF=0.7, PMF=0.6, PMF=0.5 and pure commercial butane, respectively. Thus, influence of evaporation temperature on the condenser heat load is high and low for commercial butane and pure propane, respectively. Clearly, hydrocarbon blend with propane mass fraction of 0.6 yields about the same condenser heat loads of R134a. Hydrocarbon mixture with propane mass fraction of 70% and 50% yield higher and lower condenser heat loads, respectively, than that of R134a.

Comparison of compressor input power has been made in Fig. (5D). It is seen that among the considered refrigerants, the input power for pure propane (PMF=1) is the highest one while that of pure commercial butane is the lowest one. It is clear that hydrocarbon mixture of 60% propane yields approximately the same input power as R134a. Compressor input power at evaporation temperature of -35°C is higher than that of -10°C by a rank of 3.12, 1.97, 2.48, 2.64, 2.85 and 3.5 for R134a, pure propane, PMF=0.7, PMF=0.6, PMF=0.5 and pure commercial butane, respectively. Clearly, pure propane has the highest line slope compared to that of pure commercial butane. Thus, influence of the evaporation temperature on compressor input power of propane is higher than that of butane.

Volumteric cooling capcities of the considered working fluids are compared in Fig. (6), which confirms that pure commercial butane and pure propane yield the lowest and highest volumetric cooling capacity, respectively. This means that both pure propane and commercial butane require compressor design change to achive the same cooling capacity of R134a. Consequently, such refrigerants are not suitable to be used as a drop-in replacement for R134a in domestic refigerator. It is clear that hydrocarbon blend of 60% propane yields the same volumetric cooling capacity as R134a over the considered range of operating conditions. Volumetric cooling capacity for the hydrocarbon blend of 50% propane is lower than that of R134a by about 12.9% while volumetric cooling capacity for the ternary hydrocarbon blend of 70% propane is higher than that for R134a by about 15.6%. This confirms that same cooling capacity can be achieved using hydrocarbon mixture of 60% propane without change in the compressor size. Volumetric cooling capacity at evaporation temperature of -10°C is approximately 5.73, 3.60, 4.53, 4.82, 5.20 and 6.37 times that one at evaopration temperature of -35°C for R134a, pure propane, PMF=0.7, PMF=0.6, PMF=0.5 and commercial butane, respectively. With reference to lines slope, it can be seen that the senstivity of volumetric cooling capacity to the evaporator temperature is high and low for pure propane and pure commercial butane, respectively.

A comparison among the various refrigerants based on coefficient of performance is shown in Fig. (7), which indicates that pure commercial butane and pure propane yield the highest and lowest coefficient of performance, respectively. It is clear that COP for hydrocarbon mixture of 70% propane is very close to the coefficient of performance of R134a over the considered range of operating conditions while coefficient of performance for the ternary hydrocarbon blend of 60% propane is higher than that for R134a by about 2.3%. Thus, using this hydrocarbon mixture, same cooling capacity can be absorbed from space to be cooled with lower power consumption. Clearly, COP at evaporation temperature of -10°C is higher than that one at evaporation temperature of -35°C by about 82% for all working fluids.

4. CONCLUSIONS

In the present work, theoretical investigation was carried out to simulate the domestic refrigerator This simulation aimed at comparing performance characteristics of the domestic refrigerator working with R134a, commercial butane, propane and a propane/iso-butane/n-butane mixture with different concentrations under various operating conditions. Based on the simulation results, the following conclusions are drawn:

- Hydrocarbon refrigerants offer desirable environmental requirements, i.e. zero ozone depletion potential and approximately zero global warming potential.
- Pure commercial butane offers many desirable characteristics such as low operating preessure, discharge temperature and mass flow rates and high COP and specific volume, but requires compressor design change.
- Pure propane yields undesirable characteristics such as highe operating pressure and low coefficient of performance.
- The propane/iso-butane/n-butane mixture with 60% propane is the best drop-in replacement for R134a in domestic refrigerators under normal, subtropical and tropical operating conditions.
- Vapor pressures of R134a and the ternary hydrocarbon mixture with 60% propane are nearly same. However, this hydrocarbon mixture with 70% and 50% propane yields higher and lower vapor pressure, respectively than that of R134a over the considered range.
- Pressure ratio of the hydrocarbon mixture with 50%, 60% and 70% is lower than that of R134a by about 6.3%, 11.1% and 15.3%, respectively.
- Hydrocarbon blend with propane mass fraction of 0.6 produces higher discharge temperature than that of R134a by about 2°C only over the considered range of operating conditions.
- Mass flow rate of hydrocarbon mixture with different concentrations is lower than that of R134a by approximately 50% over the entire range of operating conditions.
- Condenser heat load of R134a is close to that of a propane/iso-butane/n-butane mixture with propane mass fraction of 0.6.
- A ternary hydrocarbon blend of 60% propane and R134a need nearly same input power for the compressor.
- Volumetric cooling capacities of hydrocarbon mixture with 70% propane are higher than that of R134a by nearly 15.5%. However, volumetric cooling capacities of R134a and the hydrocarbon blend with 60% propane are the same over the considered range of operating conditions.
- Coefficient of performance of the hydrocarbon mixture with 60% propane is higher than that of R134a by about 2.3% over the considered range of operating conditions.

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