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Experimental investigation of R290/R600a mixture as an alternative to R134a in a domestic refrigerator

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Abstract

R134a is the most widely used refrigerant in domestic refrigerators. It must be phased out soon according to Kyoto protocol due to its high global warming potential (GWP) of 1300. In the present work, an experimental investigation has been made with hydrocarbon refrigerant mixture (composed of R290 and R600a in the ratio of 45.2:54.8 by weight) as an alternative to R134a in a 200 l single evaporator domestic refrigerator. Continuous running tests were performed under different ambient temperatures (24, 28, 32, 38 and 43 °C), while cycling running (ON/OFF) tests were carried out only at 32 °C ambient temperature. The results showed that the hydrocarbon mixture has lower values of energy consumption; pull down time and ON time ratio by about 11.1%, 11.6% and 13.2%, respectively, with 3.25–3.6% higher coefficient of performance (COP). The discharge temperature of hydrocarbon mixture was found to be 8.5 to 13.4 K lower than that of R134a. The overall performance has proved that the above hydrocarbon refrigerant mixture could be the best long term alternative to phase out R134a.
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Keywords: Domestic refrigerator; R134a; R290/R600a; GWP

1. Introduction

In India, about 80% of the domestic refrigerators use R134a as refrigerant due to its excellent thermodynamic and thermo physical properties. But R134a has high GWP of 1300. The higher GWP due to R134a emissions from domestic refrigerators leads to identifying a long term alternative to meet the requirements of system performance, refrigerant-lubricant interaction, energy efficiency, environmental impacts, safety and service. The Kyoto Protocol of the United Nations Framework Convention on Climate Change (UNFCCC) calls for reductions in emissions of six categories of greenhouse gases, including hydrofluorocarbons (HFCs) used as refrigerants [1]. From the environmental, ecological and health point of view, it is urgent to find some better substitute for HFC refrigerants [2]. Many investigators have reported that GWP of HFC refrigerants is more significant even though it has less than that of chlorofluorocarbons (CFC) refrigerants [3–5].

Refrigerators are identified as major energy consuming domestic appliance in household environment [6]. Many researchers have reported that hydrocarbon mixed refrigerants is found to be an energy efficient and environment friendly alternative option in domestic refrigerators. Akash and Said [7] experimented with liquefied petroleum gas (LPG) (composed of R290, R600 and R600a, in the ratio of 30:55:15, by mass) as an alternative to R12 in domestic refrigerators at various mass charges 50, 80 and 100 g. The results reported that 80 g of LPG showed the best performance compared to that of R12. Jung et al. [8] experimented with R290/R600a (in the ratio of 60:40, by mass fraction) as an alternative to R12 in 299 and 465 l domestic refrigerators and reported that COP and energy efficiency were improved by 2.3 and 4%, respectively. Pure hydrocarbon refrigerants are not suitable as drop in substitutes for R134a due to its mismatch in volumetric cooling capacity and operating pressure [9]. Somchai and Nares [10] investigated with hydrocarbon mixtures and HC/HFC mixtures at different mass ratio in a 239 l domestic refrigerator at an ambient tem-

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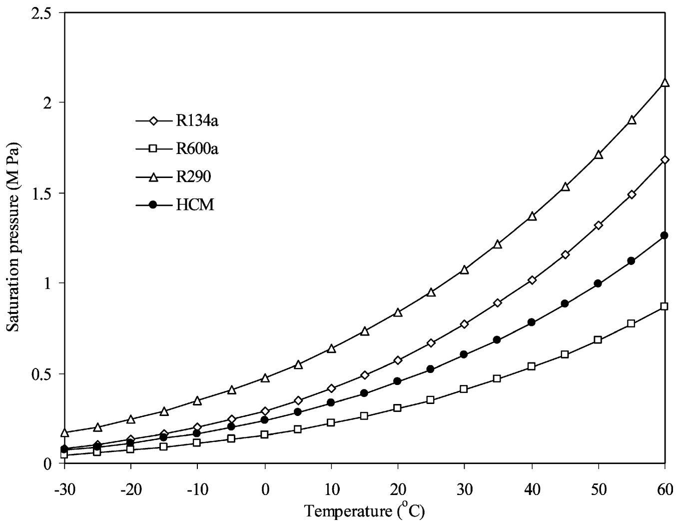

Fig. 1. Saturation pressure vs. temperature.

perature of 25 °C to replace R134a. It has been reported that R290/R600 mixture (in the ratio of 60:40, by mass fraction) is the most appropriate alternative. Fatouh and Kafafy [11] experimented with LPG (composed of 60% of R290 and 40% of commercial butane) as an alternative to R134a in a 280 l domestic refrigerator at 43 °C ambient temperature. Their results reported that COP of the refrigerator using LPG was higher than that of R134a by about 7.6% with lower values of energy consumption and ON time ratio by about 10.8 and 14.3%, respectively. Based on the theoretical investigation, it has been reported that capillary tube length using ternary hydrocarbon refrigerant mixture (composed of R290, R600 and R600a) with 60% R290 mass fraction requires 30% increase in capillary tube length compared to R134a [12].

Fig. 1 indicates that pure propane has the highest saturation pressure, while pure isobutane has the lowest. Hence, pure propane cannot be used as a drop in substitute for R134a from operation pressure point of view whereas pure isobutane require modifications in the compressor. Therefore, the mixture composed of R290 and R600a was considered as an alternative to R134a. A hydrocarbon mixture composed of 45.2% of R290 and 54.8% of R600a is the popular hydrocarbon mixture available in the Indian market. This mixture is further referred in this paper as HCM. Owing to these points, the above mentioned HCM can be considered to be a viable alternative to replace R134a.

The R290/R600a refrigerant mixture is a zeotrope, which does not behave like a single substance when it changes its state. Unlike pure refrigerants, the phase change process of zeotropic mixtures is non-isothermal and the compositions do not remain constant, which leads to composition shift and temperature glide [13]. When zeotrope evaporates inside the tubes, more volatile component (R290) in the mixture evaporates first and the liquid becomes rich in less volatile component (R600a). Because of increase in less volatile component (R600a) in liquid, the saturation temperature gets decreased; as a consequence the pressure drop is compensated [14]. The above hydrocarbon mixture is found to be chemically stable and non-reactive with non-metallic components used in hermetically sealed compressors [15]. Since the quantity of hydrocarbon mixture used in the system is about half of the R134a, even in case of any leakage, the full hydrocarbon quantity does not exceed the lower flammable limit during normal working conditions.

The literature review brings out the fact that many researchers [6–12] have studied with different hydrocarbon refrigerant mixtures as alternative to R12 and R134a in domestic refrigerators. However, the possibility of using HCM (composed of 45.2% of R290 and 54.8% of R600a) as R134a alternative at different ambient temperatures needs further investigation. The objective of the present study is to explore the possibility of using above mentioned HCM in a 200 l domestic refrigerator with different mass charges (40, 50, 60 and 70 g). The influence of ambient temperatures on the performance characteristics of the refrigerator under continuous and cycling running operating mode at different freezer air temperatures with 32 °C ambient temperature have been studied.

2. Experimental setup

The schematic diagram of a single evaporator R134a domestic refrigerator experimental setup with the total volume of 200 l is as shown in Fig. 2. It consists of a refrigerator cabin, a hermetically sealed reciprocating compressor, a wire mesh natural convection air cooled condenser, a capillary strainer, five capillary tubes of different length with ball valves, an evaporator and a sight glass. To optimize the capillary tube length for hydrocarbon refrigerant mixture, five capillary tubes of 0.78 mm diameter with different lengths (4, 4.5, 5, 5.5 and 6 m) were provided [11]. Domanski and Didion [16] reported that hydrocarbon refrigerants showed an improved COP due to the presence of suction line/liquid line heat exchanger. Hence, all the capillary tubes of the refrigerator were attached with the suction line to improve the performance. To estimate the actual COP and refrigeration capacity of the domestic refrigerator, the evaporator similar to one used in refrigerator was kept in a calorimeter filled with ethylene glycol as secondary refrigerant. The calorimeter was fitted with an electric heater connected with a dimmerstat to maintain constant temperature. The calorimeter was insulated to reduce the ambient heat infiltration. A manually operated stirrer was provided in the calorimeter to maintain uniform temperature inside. Four ball valves were fixed in the circuit between the capillary tube outlet and compressor suction to divert the refrigerant flow to either one of the evaporator (in the refrigerator cabin or calorimeter). Another five ball valves were fixed at the capillary tube inlet for choosing different capillary tube length. The refrigerator was instrumented with two compound pressure gauges with an accuracy of ±0.25% at the inlet and outlet of the compressor for measuring the suction and discharge pressures. Twelve calibrated RTD (Pt100) temperature sensors with an accuracy of ±0.25 K were placed inside the freezer, refrigerator cabin, compressor outlet, condenser outlet, capillary inlet, evaporator inlet and outlet, accumulator outlet, inside the calorimeter, entry and outlet of the evaporator fitted in calorimeter and suction of the compressor. During the experimentation, the experimental setup was placed in an

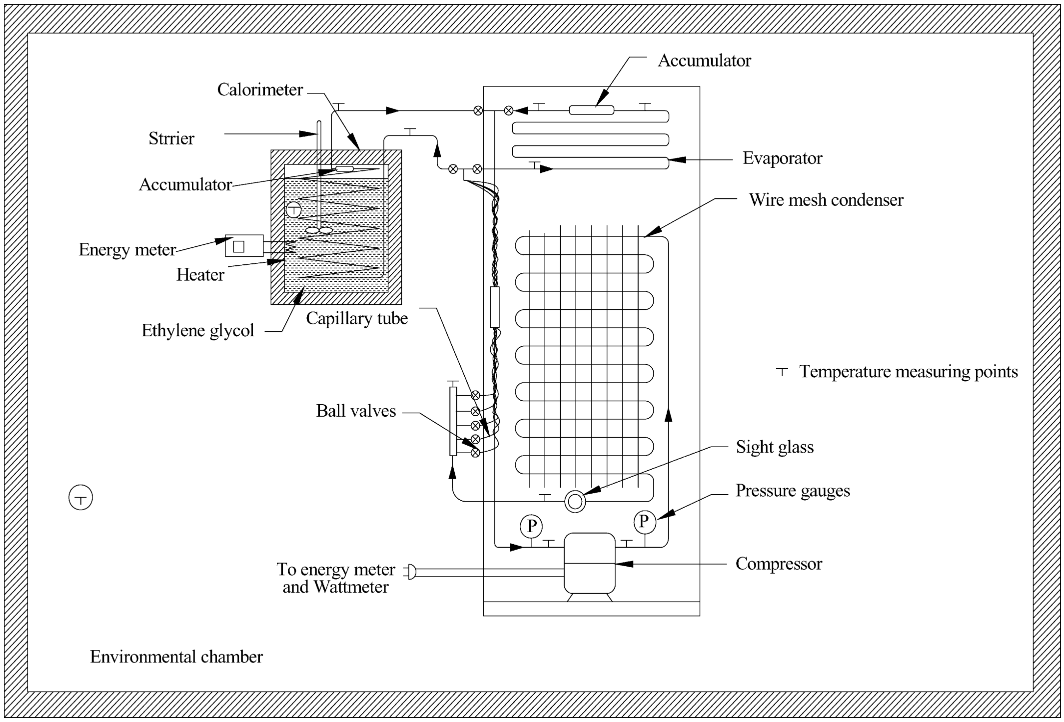

Fig. 2. Schematic diagram of domestic refrigerator experimental setup.

environmental chamber of dimension 2000 mm × 2000 mm × 2400 mm. During experimentation, temperature inside the chamber was maintained at 24, 28, 32, 38 and 43 °C, which was monitored by a digital thermometer with an accuracy ±0.5 K. A 2 kW air heater was placed inside the chamber to maintain the higher temperatures. The air heater was connected to a dimmerstat to maintain a constant temperature. The energy consumption of the refrigerator was measured under no load condition using a digital energy meter of ±0.5% accuracy and instantaneous compressor power consumption was measured by a digital Wattmeter with an accuracy of ±0.5%.

3. Experimental procedure

Initially, the system was flushed with nitrogen gas to eliminate impurities, moisture and other foreign materials inside the system, which may affect the accuracy of the experimental results. The experiments were conducted according to ISO 8187 [17]. To conduct no load pull down test, the door was kept open until temperature inside the refrigerator has reached the steady state condition with ambient. As per manufacturer's recommendation, 110 g of R134a was charged in the refrigerator for conducting baseline tests. During experimentation with R134a, 4 m capillary tube length was used. The continuous running tests were carried out by connecting the evaporator inside the calorimeter with the system. The pull down characteristics and cycling running tests were carried out by connecting the evaporator inside the refrigerator with the system. The actual refrigeration capacity and COP of the refrigerator were calculated as per the procedure followed by Sekhar and Lal [18]. The heater load was adjusted by a dimmerstat to maintain a temperature of $-12 \pm 0.5\,°C$ inside the calorimeter. The energy consumption

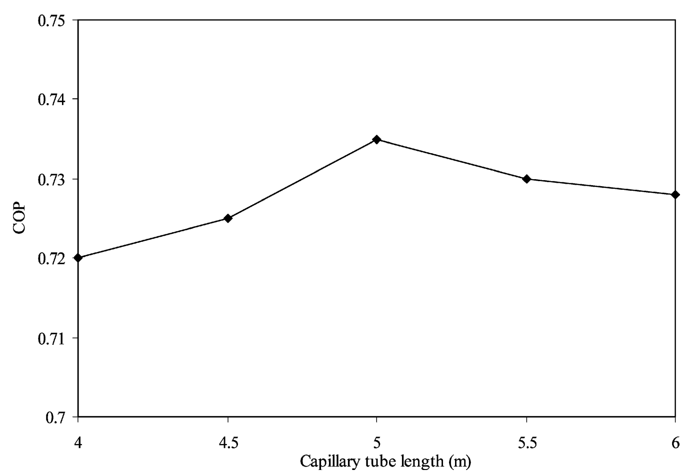

Fig. 3. Variation of COP with capillary tube length for 60 g of HCM at 32 °C ambient temperature.

of the compressor and heater were measured by separate energy meters. During continuous running tests, ambient temperature was maintained around 24, 28, 32, 38 and 43 °C. All the experimental observations were made after attaining the steady state conditions (4 h). The cycling running tests were carried out at 32 °C ambient temperature with different freezer air temperature settings (−12, −10, −8, −6 and −4 °C). After completing the base line reference test with R134a, the refrigerant was recovered from the system. Before experimenting with HCM, the length of the capillary tube was optimized for maximum COP. During capillary tube optimization, the refrigerator was charged with 60 g of HCM and ambient temperature was maintained at 32 °C. As shown in Fig. 3, the maximum COP was observed with 5 m capillary tube length. Hence, 5 m capillary tube was taken for the experimentation with HCM. Then, the refrigerator was charged with 40, 50, 60 and 70 g of HCM and these tests were repeated. Replacement of polyolester (POE) with mineral oil requires stringent flushing of compressor and refrigeration system, so that compressor and the refrigeration system should have less than 1% residue of POE. This needs a major modification in the refrigeration system. Hence, in the present study, the possibility of using HCM by retaining the same lubricant oil (POE) has been made. Since the mixture is zeotrope, the refrigerant is charged in liquid state and the charge quantity was ensured with the help of electronic balance having an accuracy of ±0.01 g. In order to reduce the experimental uncertainties, experiments were repeated for five times and average values were considered. The variation in experimental values from the average value is within ±5%. Temperatures at different locations were recorded every ten minutes intervals. Pressure at compressor suction and discharge was measured every twenty minutes intervals. The instantaneous power consumption of the refrigerator during continuous running tests was measured after attaining the steady state condition. The energy consumption per day during cycling running tests was measured after 24 h by using a digital energy meter. The measured values were used to study the performance characteristics of the refrigerator.

4. Results and discussion

Experimental results obtained for continuous running mode at different ambient air temperatures and cycling operating modes with different freezer air temperature settings at 32 °C ambient temperature are discussed in this section.

4.1. Continuous running tests

4.1.1. Pull down characteristics

Pull-down time is the time required to reduce the air temperature inside the refrigerator from ambient condition to the desired freezer and cabin air temperatures of −12 and 6 °C in the freezer and cabin, respectively, according to ISO8187 [16]. Pull down tests were carried out at 32 °C ambient temperature. The pull-down time of about 112 min was required to reach the desired freezer air temperature (−12 °C) for R134a (baseline test) as indicated in Fig. 4. It was observed that HCM40 yield higher steady state air temperatures in both freezer and cabin. Hence, HCM40 has not been considered for further discussion. The time required for HCM50, HCM60 and HCM70 was about 115, 99 and 90 min, respectively. An increase in pull down time by about 2.6% was observed for HCM50 due to insufficient refrigerant quantity. The pull down time was reduced by about 11.6 and 20.53% for HCM60 and HCM70, respectively compared to R134a due to its high latent heat of vaporization.

About 114 min was required to reach the required cabin temperature (6 °C) for R134a (base line test). The time required for HCM50, HCM60 and HCM70 was about 116, 102 and 94 min, respectively as shown in Fig. 5. An increase in pull down time by about 1.72% was observed for HCM50 due to insufficient refrigerant quantity. A reduction in pull down time in the refrigerator cabin was observed to be about 10.5 and 17.5% for HCM60 and HCM70, respectively.

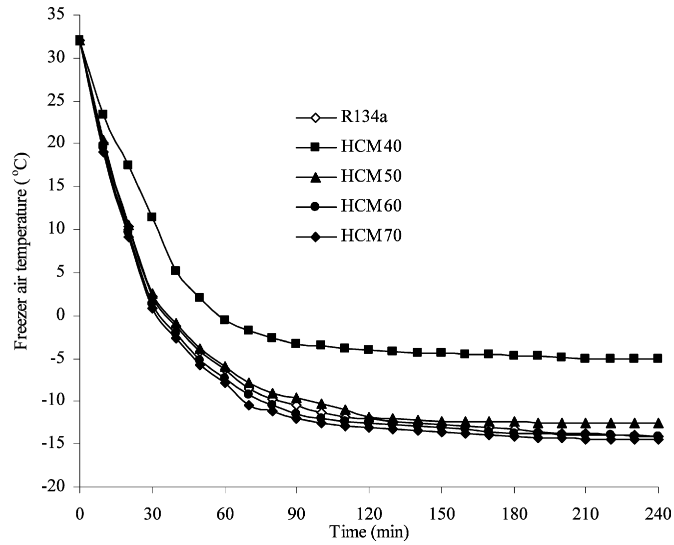

Fig. 4. Pull down time vs. Freezer air temperature at 32 °C ambient temperature (Continuous running test).

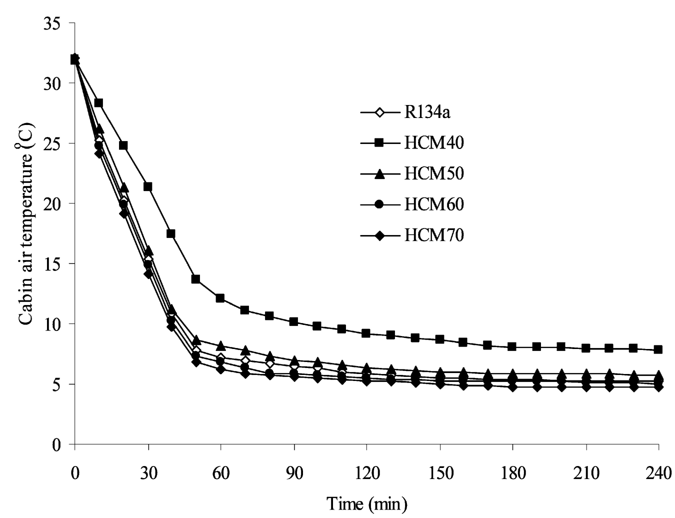

Fig. 5. Pull down time vs. Cabin air temperature at 32 °C ambient temperature (Continuous running test).

4.1.2. Performance characteristics

The discharge temperature is an important parameter considered for choosing an alternative. The discharge temperature influences the stability of the lubricants and compressor components. Fig. 6 reveals that discharge temperature of hydrocarbon mixtures was found to be lower than that of R134a by about 11.5 to 18.7 K, 8.5 to 13.4 K, and 5.5 to 8.7 K, respectively, for HCM50, HCM60 and HCM70 due to its lower specific heat ratio. HCM has lower impact on compressor components and stability of lubricants. Hence, longer compressor life time can be expected when HCM is used as an alternative. It is seen that as HCM charge increases, discharge temperature also gets increased due to increase in condensation pressure. The discharge temperature of R134a, HCM50, HCM60 and HCM70 gets in-

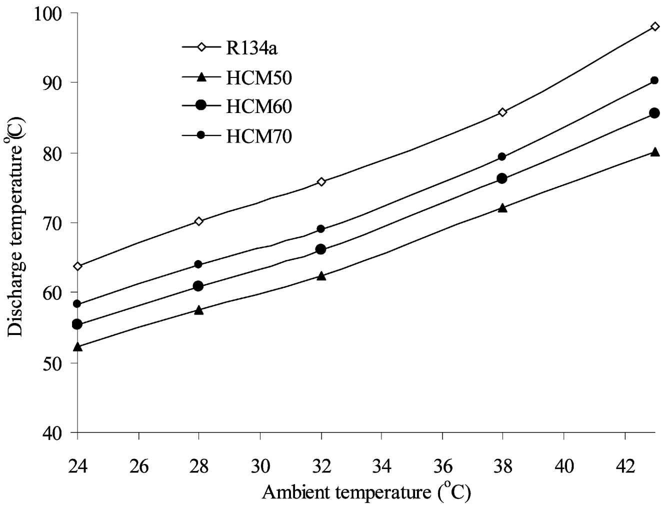

Fig. 6. Variation of compressor discharge temperature with ambient temperature at $-12 \pm 0.5\,^{\circ}\mathrm{C}$ evaporator temperature (Continuous running test).

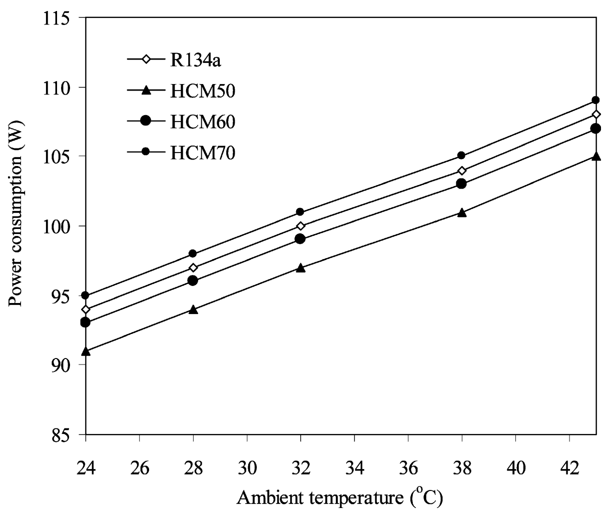

Fig. 7. Variation of compressor discharge temperature with ambient temperature at $-12 \pm 0.5\,^{\circ}\mathrm{C}$ evaporator temperature (Continuous running test).

creased by about 33 K with increase in ambient temperature from 24 to 43 °C.

Instantaneous power consumption is the main criterion to choose a right quantity of HCM mass charge. Power consumption of R134a and various mass charges of HCM were shown in Fig. 7. It is observed that power consumption increases with increase in refrigerant mass charge. This is mainly due to increase in mass flow rate of refrigerant through compressor. Power consumption of HCM50 and HCM60 were found to be lower than that of R134a by about 2.77–3.19% and 0.92–1.06%, respectively. But, HCM70 showed about 0.91–1.05% higher power consumption than that of R134a. The power consumption of the refrigerator increases with increase in ambient temperature due to increase in condensing temperature and pressure.

The COP of R134a and hydrocarbon mixtures is compared in Fig. 8. The COP of the HCM60 and HCM70 were higher than that of R134a by about 3.25–3.6% and 1.2–1.66%, respectively for the ambient temperatures between 24 and 43 °C. The COP

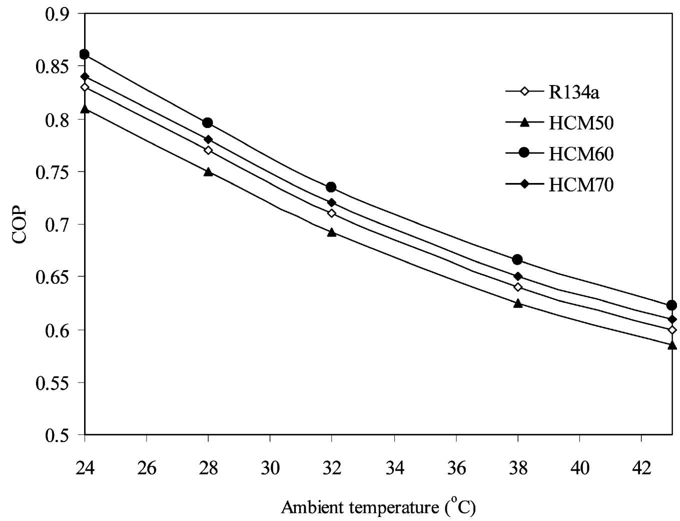

Fig. 8. Variation of compressor discharge temperature with ambient temperature at $-12 \pm 0.5\,^{\circ}\mathrm{C}$ evaporator temperature (Continuous running test).

of HCM50 was lower than that of R134a by about 2.4–2.59% due to lower refrigeration capacity. The COP of the refrigerator decreases with increase in ambient temperature from 24 to 43 °C due to increase in power consumption.

4.2. Cycling running (ON/OFF) tests

In order to study the actual operating conditions, the refrigerator was subjected to cycling running tests. ON time ratio and energy consumption per day were determined for five different freezer air temperatures settings. Based on the ON time ratio and energy consumption per day, an appropriate HCM mass charge could be determined. Cycling running test results of R134a and HCM were compared in Figs. 9 and 10. ON time ratio of refrigerator is the ratio of operating time to total time of the cycles. Fig. 9 shows the variation of ON time ratio as a function of freezer air temperature. The ON time ratio of HCM60 and HCM70 was found to be lower than that of R134a by about 13.2 and 15.7%, respectively due to high latent heat of vaporization. HCM50 has higher ON time ratio by about 2.5% compared to that of R134a due to lack of refrigerant quantity.

Fig. 10 shows that the energy consumption is a function of freezer air temperature for each HCM mass charge. The energy consumption per day increases with change in freezer air temperature due to increase of ON time ratio to meet the required temperature in the freezer. Hydrocarbon refrigerant mixtures have higher latent heat compared to R134a. Therefore, the compressor running time can be reduced. Energy consumption of HCM60 and HCM70 was lower than that of R134a by about 11.1 and 5.8%, respectively, due to its lower ON time ratio. HCM50 has 4% higher energy consumption than that of R134a due to insufficient quantity and more ON time ratio. The annual energy consumption of refrigerator working with HCM was reduced by about 70 kW h, which reduces indirect global warming.

The results obtained in this study are similar to the previous work reported by Fatouh and Kafafy [11] for LPG mixture

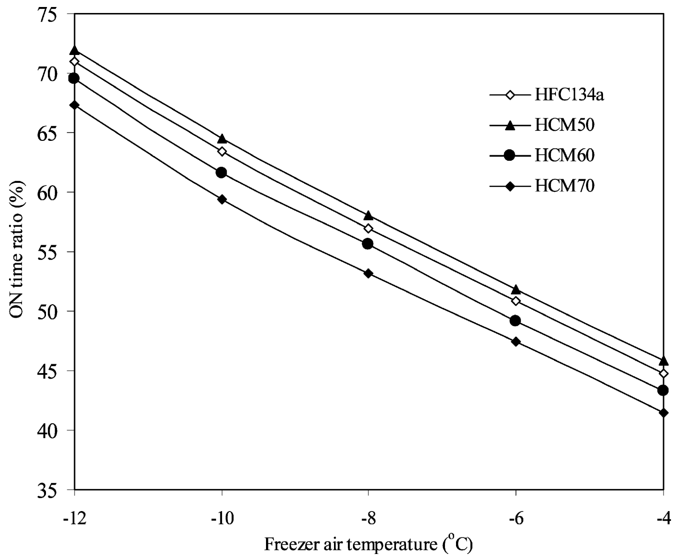

Fig. 9. ON time ratio with different freezer air temperature settings at 32 °C ambient temperature (Cycling running test).

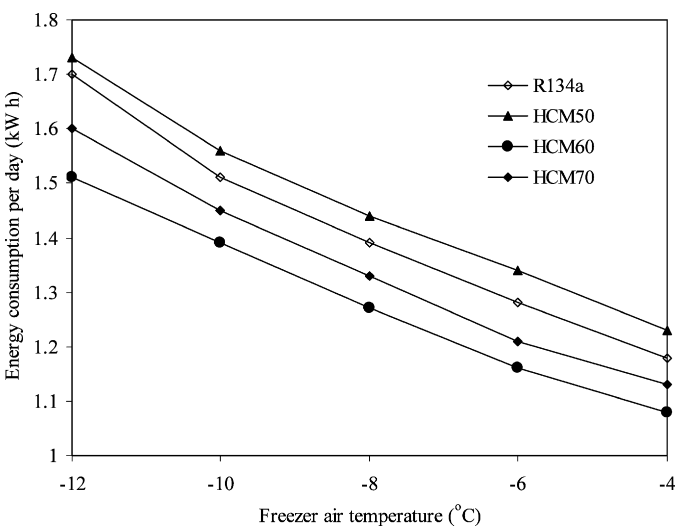

Fig. 10. Energy consumption per day with different freezer air temperature settings at 32 °C ambient temperature (Cycling running test).

composed of 60% of propane and 40% of commercial butane at 43 °C ambient temperature. An improved COP of about 7.6% with 10.8% reduction in energy consumption has been reported for 5 m capillary tube length and 60 g of LPG, where as in the present work the COP has been improved by about 3.25–3.6% for wide range of ambient temperatures between 24 and 43 °C. At 32 °C ambient temperature, a reduction in energy consumption of about 11.1% has been observed for 60 g of HCM.

4.3. Mixture behavior

Since the HCM is zeotropic in nature, which may cause large temperature variation in the evaporator resulting in uneven frost formation. As per refrigerator manufacturer's recommendation, a 3 K temperature variation in the evaporator (freezer) is permissible as mentioned by Sekar et al. [19]. To ensure this, temperature at inlet and outlet of the evaporator and outlet of accumulator was measured. It was observed that the temperature variation in the evaporator was found to be about 2.6 K, which confirmed that the HCM did not affect the evaporator performance. To conduct oil miscibility test, initially the compressor was charged with 500 ml of POE as per manufacturer's recommendation. After 1000 hours of operation, the compressor was removed from the system and oil was drained from the compressor. About 50 ml of lubricant loss was observed, which may occur during recovery of refrigerant from the system. Hence it is evidenced that HCM is miscible and returns the oil to the compressor. The performance of the compressor working with HCM was found to be good.

5. Conclusion

The performance of HCM composed of 45.2% of R290 and 54.8% of R600a under continuous running and cycling running tests has been experimentally investigated and following conclusions are drawn.

1. HCM demanded lengthening of capillary tube by about 25% to achieve maximum COP.
2. 60 g of HCM consumes about 11.1% lesser energy compared to that of R134a.
3. Pull down time and ON time ratio of the HCM are reduced by about 11.6 and 13.2%, respectively.
4. The discharge temperature of HCM is about 8.5 to 13.4 K lower than that of R134a. Hence, the life of the compressor can be improved.
5. COP of the domestic refrigerator working with HCM is improved by up to 3.6%.
6. Temperature variation in the evaporator is found to be within 3 K.
7. The miscibility of HCM with POE was found to be good.
8. HCM also reduce the indirect global warming due to its higher energy efficiency.

Thus, the reported results prove that the above HCM can be used as an alternative to phase out R134a in domestic refrigerators.

References

[1] W.-T. Tasi, An overview of environmental hazards and exposure and explosive risk of hydrofluorocarbon HFCs, Chemosphere 61 (2005) 1539–1547.
[2] E. Johnson, Global warming from HFC, Environ. Impact Assessment Rev. 18 (1998) 485–492.
[3] P. Ashford, D. Clodic, A. McCulloch, L. Kuijpers, Emission profiles from the foam and refrigeration sectors comparison with atmospheric concentrations, Part 2: Results and discussion, Int. J. of Refrig. 27 (2004) 701–716.
[4] A. McCulloch, P.M. Midgley, P. Ashford, Release of refrigerant gases (CFC12, HCFC22, HFC134a) to the atmosphere, Atmospheric Environment 37 (2003) 889–902.
[5] A. McCulloch, A.A. Lindley, From mine to refrigeration: A life cycle inventory analysis of the production of HFC134a, Int. J. Refrig. 26 (2003) 865–872.

[6] R. Radermacher, K. Kim, Domestic refrigerators: recent developments, Int. J. Refrig. 19 (1996) 61–69.
[7] B.A. Akash, S.A. Said, Assessment of LPG as a possible alternative to R-12 in domestic refrigerator, Energy Conversion Management 44 (2003) 381–388.
[8] D.S. Jung, C.-B. Kim, K. Song, B.J. Park, Testing of propane/isobutane mixture in domestic refrigerators, Int. J. Refrig. 23 (2000) 517–527.
[9] M. Fatouh, M. El Kafafy, Assessment of propane/commercial butane mixtures as possible alternatives to R134a in domestic refrigerators, Energy Conversion and Management 47 (2006) 2644–2658.
[10] S. Wongwises, N. Chimres, Experimental study of hydrocarbon mixtures to replace HFC134a in domestic refrigerators, Energy Conservation and Management 46 (2005) 85–100.
[11] M. Fatouh, M. El Kafafy, Experimental evaluation of a domestic refrigerator working with LPG, Applied Thermal Engineering 26 (2006) 1593–1603.
[12] M. Fatouh, Theoretical investigation of adiabatic capillary tubes working with propane/n-butane/isobutene blends, Energy Conversion and Management 48 (2007) 1338–1348.
[13] L. Rajapaksha, Influence of special attributes of zeotropic refrigerant mixtures on design and operation of vapour compression refrigeration and heat pump systems, Energy Conversion and Management 48 (2007) 539–545.
[14] K. Stephan, Two phase heat exchange for new refrigerants and their mixtures, Int. J. Refrig. 18 (1995) 198–209.
[15] D. Colbourne, T.J. Ritter, Compatibility of non-metallic materials with hydrocarbon refrigerants and lubricant mixtures, IIF–IIR Commission B1, B2, E1 and E2, Purdue University, USA, 2000.
[16] P.A. Domanski, D.A. Didion, Evaluation of suction-line/liquid-line heat exchange in the refrigeration cycle, Int. J. Refrig. 17 (1994) 487–493.
[17] ISO, International Standard Organization, International Standard-8187, Household refrigerating appliances (refrigerators/freezers) characteristics and test methods, 1991, International Organization for Standardization, Switzerland.
[18] S.J. Sekhar, D.M. Lal, R134a/R600a/R290 a retrofit mixture for CFC12 systems, Int. J. Refrig. 28 (2005) 735–743.
[19] S.J. Sekhar, D.M. Lal, S. Renganaraynan, Improved energy efficiency for CFC domestic refrigerators retrofitted with ozone friendly HFC134a/HC refrigerant mixture, Int. J. Thermal Sci. 43 (2004) 307–314.