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Improved energy efficiency for CFC domestic refrigerators retrofitted with ozone-friendly HFC134a/HC refrigerant mixture

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Abstract

Conversion of CFC12 systems to eco-friendly ones will be a major thrust area for refrigeration sector in the near future. As and when an existing CFC (chlorofluorocarbon) system has to be recharged it is advisable to retrofit the system with an eco-friendly energy efficient refrigerant. Presently two potential substitutes, namely, HFC134a and HC blends are available as drop in substitutes for CFC12. HC (hydrocarbon) refrigerants do have inherent problems in respect of flammability. HFC134a is neither flammable nor toxic. But HFCs (hydrofluorocarbons) are not compatible with mineral oil and the oil change is a major issue while retrofitting. The above techno-economic feasibility issue to retrofit the existing CFC12 systems with ozone friendly refrigerant and the energy efficiency of the system are the challenges in the domestic refrigeration sector. In this present work an experimental analysis has been carried out in a 165 l CFC12 household refrigerator retrofitted with eco-friendly refrigerant mixture of HFC134a/HC290/HC600a without changing the mineral oil. Its performance, as well as energy consumption, is compared with the conventional one. As the system has been running successfully for more than 12 months it is also evident that the new mixture is compatible with mineral oil. It has been found that the new mixture could reduce the energy consumption by 4 to 11% and improve the actual COP by 3 to 8% from that of CFC12. The new mixture also showed 3 to 12% improvement in theoretical COP. The overall performance has proved that the new mixture could be an eco-friendly substitute to phase out CFC12.

Keywords: HFC134a/HC mixtures; Household refrigerator; Zeotrope; Retrofitting; POE oil problems; CFC phase out; COP improvement

1. Introduction

The refrigeration sector is in the midst of an unprecedented transition crisis catalysed by environmental concerns with the impact of refrigerant emissions. As per the Montreal protocol, CFC12 is being phased out following a stipulated time frame. The developed countries have already phased out this substance and the developing countries are to totally phase out the CFCs by 2010 as per the Montreal protocol. Most of the developing countries are drastically reducing their CFC production and consumption. This demands for a suitable substitute for CFC12 for possible retrofitting of existing systems as well as for new systems.

The refrigerator production and sales rate in developing countries made significant leaps only in the last decade and for another 15 years all the refrigerators sold would one way or other come for service. Due to the reduction in production of CFCs and the nonavailability and high cost of CFC refrigerants, the existing CFC based refrigerators (in India approximately 25 million) may have to be dispensed with as and when some service problems arise, which the economy of a developing country would not permit. To retrofit the existing CFC12 refrigerators with eco-friendly technology two matching refrigerants are available, namely, HFC134a and HC blend. HFC134a systems do have inherent service issues because of the POE oil in the compressor, as HFC134a is not miscible with conventional mineral oil. On the other hand, HC blends have the problem of flammability and limitation in the charge quantity due to safety (fire hazard) regulations. Hence, it would be a significant benefit for the RAC technicians in the service sector if HFC134a can be made to work with mineral oil. It has been reported that the oil miscibility problem for HFC134a in mineral oil could be solved by adding suitable quantity of HC additive [1]. This paper presents the experimental study conducted by charging an 165 l household refrigerator with HFC134a/HC

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blend additive to work with mineral oil. Already many research works have been done in the above two refrigerants including retrofitting options.

The power consumption of a HFC134a system would be 10 to 15% more than CFC12 system [2]. The theoretical COP of the HFC134a refrigeration system operating at—18 °C evaporator temperature and 45 °C condenser temperature was observed as 2.5% lesser than that of CFC12 system [3]. The performance study on single-evaporator domestic refrigerator charged with pure and mixed refrigerants indicated that the COP of HFC134a is 3% less than that of CFC12 [4]. The retrofitting experience showed that HFC134a has a very low solubility in mineral oil and due to the reactive nature of the residual mineral oil with the lubricant Polyol ester (POE) oil and HFC134a, a stringent flushing procedure should be adopted while retrofitting CFC12 systems with HFC134a [5]. Experimental studies on the retrofitted HFC134a system indicated 5 to 8% lesser COP than that of conventional CFC12 system [6].

It has been reported that the CFC12 based heat pumps could be retrofitted with HFC134a and HC mixture with mineral oil as compressor lubricant. From the experimental data it was found that the COP of the CFC12 system could be improved by 5%. Further, the COP of the CFC12 heat pumps improve by 3.5%, when they were retrofitted with a mixture of HFC134a/HC600a (80:20) by weight [7]. Experiments were conducted on domestic refrigerators with HC mixtures at various composition, HC290/HC600a mixture with 0.55 to 0.6 mass percentage of HC600a yielded 3 to 4% increase in energy efficiency as compared to CFC12 [8].

Domestic refrigerators and freezers were experimentally tested with various alternative refrigerants including single substances, azeotropes and zeotropes [9]. The major and minor design modifications required including that in the compressor has been reported. Most of the alternatives proposed were HFCs and HCs. To overcome the flammability issue of HC, CF_3I (trifluoroiodomethane) has been proposed as a flame suppressant, however, this reduces the refrigeration capacity. The nonazeotropic mixture of 80%HFC134a/20%HFC143a with internal heat exchanger has yielded a performance lesser than that of CFC12 [10]. For good oil miscibility in HFC systems polyol ester oil is required. It was also observed from published literature that suitable additives could increase the solubility of mineral oil with HFC134a [11]. With this background of diverse reports on alternative refrigerants the present work was envisaged to study the feasibility of making HFC134a work in CFC12 systems but with mineral oil. The oil miscibility issue is tackled by incorporating a HC blend as an additive to the drop in substitute for CFC12.

2. Refrigerant selection

Generally, HC additives with HFC134a behave like a zeotrope. As in the case of pure substances during boiling there is a decrease in saturation temperature because of the pressure drop along the flow path. But in most of the refrigeration systems the pressure drop is not very large due to the sufficient tube diameter. When zeotropic mixtures evaporate inside tubes, the more volatile component evaporates first and the liquid becomes rich in less volatile component. Because of the increase in the less volatile component in the liquid, the decrease in saturation temperature as a consequence of the pressure drop is compensated [12]. It has been reported that HC600a could be used as an additive to tackle the oil return problem in the evaporator [1,3]. Since HC600a is less volatile than HFC134a the evaporator liquid becomes rich in HC600a. Due to this the circulation fluid is rich with the more volatile component [14] HFC134a and the oil might not be removed, as mineral oil is not miscible with HFC134a. Hence the requirement of a high volatile component with the above mixture is found to be essential to improve the oil carrying capacity of the refrigerant mixture. Propane being more volatile than HFC134a can be considered. But propane has lesser COP than HFC134a in domestic refrigeration systems [15], and so a reduction in COP is expected if that alone is added with HFC134a as an oil carry over additive. However, HFC134a/HC600a mixture could improve the COP of CFC12 systems by 5% [7]. To utilise the above advantages of HC600a and HC290, a blend of both was considered as the additive with HFC134a in this study.

The availability and thermodynamic properties were considered to select the right composition of HC blend to be used as additive. The COP of the domestic and commercial refrigeration systems have been reported to be increased by 10 to 20% with the use of HC blends that contain HC600a and HC290 [16–20]. HC blend that consists of 54.8% HC600a and 45.2% HC290 is one of the most popular HC blends available. Also it is found to be chemically stable and nonreactive with nonmetallic components used in hermetic compressors [21]. Hence it has been taken as a viable additive with HFC134a to retrofit existing CFC12 systems (using compressor with mineral oil) without incurring any additional cost that is needed to retrofit with pure HFC134a refrigerant or with HC blend. Further in this paper this HC mixture (54.8% HC600a and 45.2% HC290) is referred as HC blend. Since the quantity of the HC blend used in the system is less than 10% of the total charge, even if any leakage occurs the full HC quantity does not exceed the lower flammability limit in respect of flammability during normal operating conditions.

To confirm the properties of HC blend, the thermophysical properties of the selected blend as given by the manufacturer have been compared with the property values obtained in the REFPROP software for different compositions of HC290 and HC600a. It was found that the property values specified by the manufacturer were matching with the REFPROP values for the blend that contains 45.2% propane and 54.8% isobutane. With this composition for blend, various mass fractions of that with HFC134a have been taken

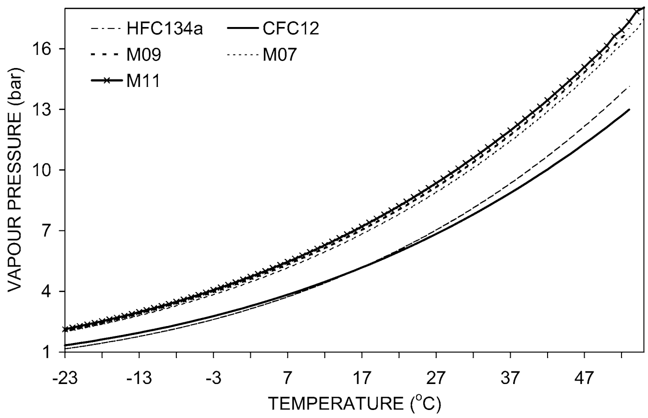

Fig. 1. Variation of vapour pressure with respect to temperature.

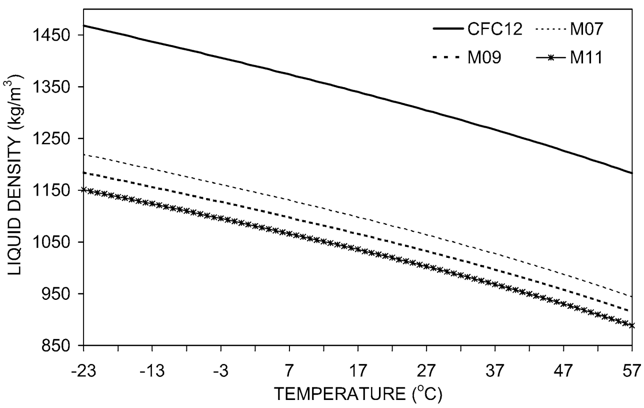

Fig. 2. Variation of liquid density with respect to temperature.

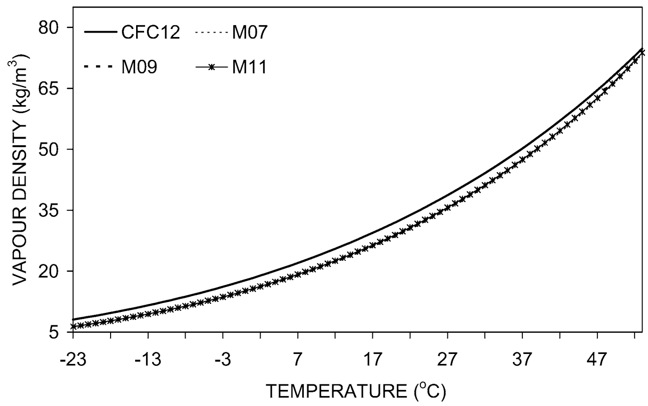

Fig. 3. Variation of vapour density with respect to temperature.

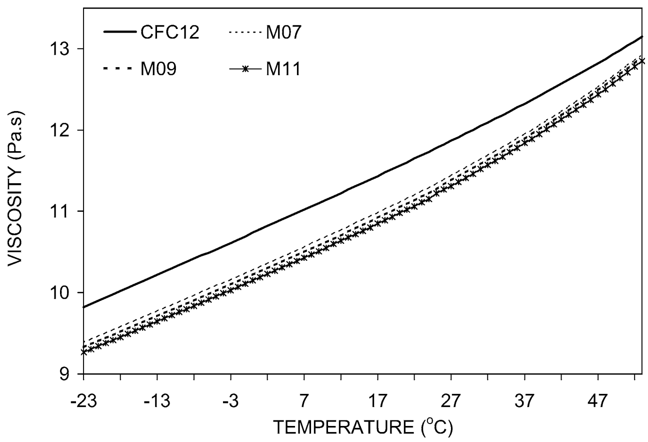

Fig. 4. Variation of viscosity with respect to temperature.

to find the thermophysical properties of the blend through REFPROP.

HFC134a with 12% of HC600a was identified as a promising alternative refrigerant in bottle coolers operating with mineral oil [13]. Experiments conducted on domestic refrigerators with HFC134a/HC600a mixture as refrigerant and mineral oil as compressor lubricant indicated that the performance was good while 8% of HC600a was used as additive. Based on the above observation, in the present work experiments were conducted for the mixture containing 7, 9 and 11% HC blend (by weight) in HFC134a. These mixtures are further referred in this paper as M07, M09 and M11 mixtures. The properties of the three blends obtained from REFPROP have been plotted for the operating temperature range as shown in Figs. 1–4. From Fig. 1 it was found that the more mass fraction of propane, the higher the condensing pressure and it also lowers the evaporator temperature, which may cause high frost formation on the evaporator. Hence the mass fraction of HC290 was limited to 0.04 and correspondingly a mixture which contains 0.09 mass fraction of HC blend (91% HFC134a and 9% HC blend—M09) was selected as refrigerant mixture to handle the oil return problem as well as improve the performance of the system. As shown in Fig. 2 the mixture has a lower liquid density and that could yield lower frictional losses. Since the vapour density of the new mixture is less than that of CFC12, a reduction in compressor work is expected.

3. Experimental setup and procedure

Two equally old (6 years) refrigerators R_M and R_C belonging to the same brand and having the same type of compressor were taken for this study. Refrigerator R_M was used to conduct all the tests with the mixtures and R_C that was running with CFC12 throughout the test period to compare the performance of R_M with the conventional system at various ambient conditions. The schematic of the experimental setup is shown in Fig. 5. Pressure gauges of $\pm 0.25\%$ accuracy were mounted in the inlet and outlet of both condenser and evaporator and film type PT100 RTD sensors of $\pm 0.1\,^{\circ}\mathrm{C}$ accuracy were used to measure the temperature at various points in the evaporator, inlet and outlet of compressor, condenser and at various positions inside the food/freezer compartment of the system. Twenty such sensors were fixed along the length of the evaporator coil. In that one sensor was placed at the inlet to accumulator, two were placed along the accumulator length and another one sensor was placed at the exit of the accumulator. Since clinching type plate evaporator was used the influence of

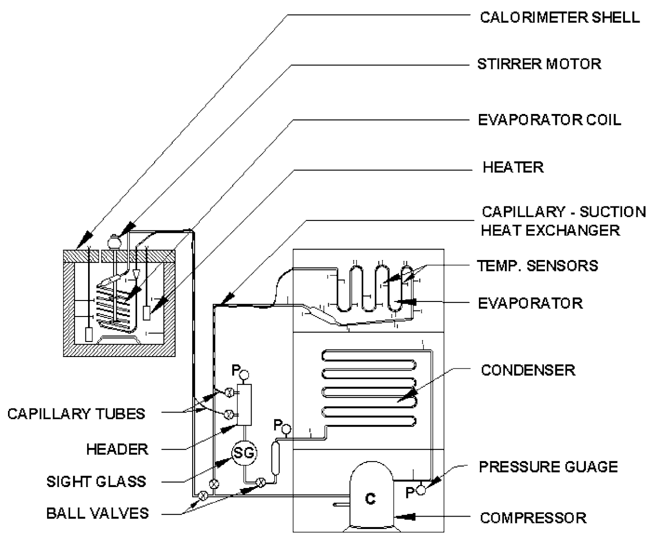

Fig. 5. Schematic diagram of the experimental setup.

plate heat transfer between adjacent pipes in measuring the temperature was observed. Hence to avoid the influence of plates in measuring the temperature the pipe was sufficiently separated from the plate wherever the RTDs were fixed. To estimate the actual COP the same type of evaporator coil used in the refrigerator was kept inside a secondary refrigerant (ethylene glycol) calorimeter in which the heat in-leak was ensured to be less than 5% of the evaporator capacity. To maintain constant temperature a stirrer was provided and five temperature sensors were provided inside the calorimeter to find the average temperature of the ethylene glycol. Suitable ball valves were fixed in the flow circuit between evaporator coils so that either one of the evaporators could be selected to be included in the circuit for study. To measure the compressor load and heater load a wattmeter with ± 0.2 W accuracy was used. Proper instrumentation was also made to measure the energy consumption, pressure, flow and temperature at various points in the refrigerator, R_C. All the temperature sensors were connected to a computer through a data logger. The measured values were used to study the system performance changes with respect to life. A sight glass was provided at the outlet of the condenser to check the quality of refrigerant.

As per the manufacturer of the appliances, the quantity of charge was 135 g. To optimise the charge both R_M and R_C were charged with 125, 130, 135, 140 and 145 g of CFC12 and their performance at 32 °C were studied. During the study, due to additional modification in the refrigerant circuit for experimentation, the R_M has shown a good performance at a charge of 140 g. R_C has shown good performance at the recommended charge of 135 g. Hence these charge quantities were taken as reference to evolve mixture quantities. Fig. 3 shows that the vapour densities of the selected mixtures at suction conditions were less than that of CFC12 and correspondingly a lower mass of new mixture has to be charged in an existing compressor which is designed to handle the flow rate of CFC12. An equivalent charge has been evolved at 116.5, 114 and 111.5 g for M07, M09 and M11, respectively. The heat capacities have been found to be matching with that of CFC12 for the selected mass due to the higher enthalpy values of the mixtures. The viscosity is also found to be less than that of CFC12 and due to that an improved heat transfer coefficient is expected [18,20].

It has been reported that the refrigerants HFC134a, HC600a and HC290 had an improved COP due to suction line/liquid line heat exchange [22]. Hence the capillary-suction heat exchanger used in domestic refrigerator has been considered as an important avenue for COP improvement and the same was maintained in the present setup. The dimension of the capillary tube was measured and it was found to be matching with the manufacturer's specifications.

To measure the energy consumption both refrigerators were kept inside a test room at no load. The temperature at various compartments inside the refrigerator viz.—15 °C freezer compartment, 0 to 8 °C food compartment and 8 to 14 °C crisper compartment were maintained (Bureau of Indian Standards (BIS) 1476). All the observations were taken at a steady state condition after 3 h. The energy consumption per day was also recorded with an energy meter with $\pm 0.25\%$ accuracy. The above said data were observed by running the system with CFC12 and various mixture compositions. To find out the actual COP the evaporator inside the refrigerator was disconnected and the system was connected to the calorimeter by suitably operating the ball valves. To find the evaporator capacity at a particular temperature the heater load was adjusted by a dimmerstat to match with evaporator capacity so that the temperature of ethylene glycol remained constant for a minimum of 5 h before the readings were taken. For various calorimeter temperatures ranging from -15 to -4 °C, the power consumption of the compressor and heater load were recorded by separate wattmeters. To calculate the actual evaporator capacity, the heat leak in the calorimeter shell at various temperatures has been added with the heater load. The above procedure was repeated five times and the average has been considered.

The three mixtures were prepared separately in three different cylinders. Each mixture component was weighed individually in an electronic balance with an accuracy of ± 0.1 g and filled in the respective cylinders. The HC additive was first sent into the mixture cylinder (as it had lower vapour pressure than HFC134a) which was followed by admitting HFC134a. Also the cylinders were maintained at low temperature to avoid backpressure development that obstructs flow. To avoid the concentration shift, excess quantity of refrigerant mixture was prepared so that the required charge (for all tests put together) will not be more than 90% [23] of the prepared quantity. A purity level of 99.5% was maintained for all the refrigerants. From the energy consumption test it was found that the M09 mixture is superior. Hence for analysis of actual COPs CFC12 and M09 only were considered.

4. Results and discussion

As per the procedure discussed earlier, tests were carried out in both systems at no load conditions. To compare the performance of the systems R_M and R_C, initially, they were charged with CFC12 and their energy consumption, temperature distribution, pressure and flow were measured inside a test room at a temperature of 32 °C. In the above condition refrigerators R_M and R_C consumed 1.58 and 1.576 kW·h per day, respectively. The other recorded values also showed less than 1% difference between the two systems for all the parameters. Hence both systems were assumed to be identical. During the twelve month long test the change in performance of the system R_C was also insignificant. Hence the values were considered to be comparable.

The mixtures prepared as per the procedure mentioned were charged into the system one after the other and all the required parameters were recorded at various ambient conditions of 24, 28, 32, 38 and 43 °C. Each test was repeated for minimum of five times to check the repeatability of the data and the average was considered. The variation in the test data from the average was within ±4%. The energy consumption data obtained for different mixtures at various ambient conditions are plotted in a graph as shown in Fig. 6. From that it has been found that the energy consumption of the three mixtures is less than that of CFC12 at various operating conditions. Among the three mixtures the M09 was characterised with 4.8 to 6.4% less energy consumption than that of CFC12. The theoretical COP of the system at various operating conditions was calculated by taking the enthalpy property values of the refrigerant from REFPROP software at various state points where the pressure and temperature were obtained from the experimentation and plotted as shown in Fig. 7. This also indicates that the COP of M09 mixture is 3 to 12% higher than that of CFC12. It is observed that the COP of M07 mixture falls below the COP of CFC12 as ambient temperature increases to 39 °C while M09 and M11 does not behave so. This can be attributed to the fact that the pressure ratio for M07 mixture is higher than that of M09 and M11 which is typically 8.65 for M07 while it is 8.48 and 8.45 for M09 and M11, respectively, at 55 °C condensing temperature and −23 °C evaporating temperature. The actual COP of M09 mixture from calorimeter test for brine temperature varying from −15 to −4 °C are plotted in Fig. 8. The readings were taken for 26, 32 and 43 °C ambient temperature. For all the temperatures M09 mixture shows 3 to 8% higher COP than that of CFC12. It is also found that the COP improvement of M09 is significantly high at higher brine temperatures.

The refrigerant mixture being zeotropic in nature may cause a temperature variation along the evaporator coil and due to that, uneven frost formation would occur in the freezer compartment. This may create problems in fixing the thermostatic switch. As per manufacturer's guidelines, in domestic systems 3 °C temperature variation along the length of the evaporator is permissible. To ensure this, the temperature along the evaporator coil was recorded for the various operating conditions and it has been plotted in graphs as shown in Figs. 9–13. It was observed that a steep gradient exists only in the accumulator, which is kept outside the freezer cabin, and the glide inside the cabin was found to be around 3 °C only for the different operating conditions. This shows that the proposed mixture does not affect the

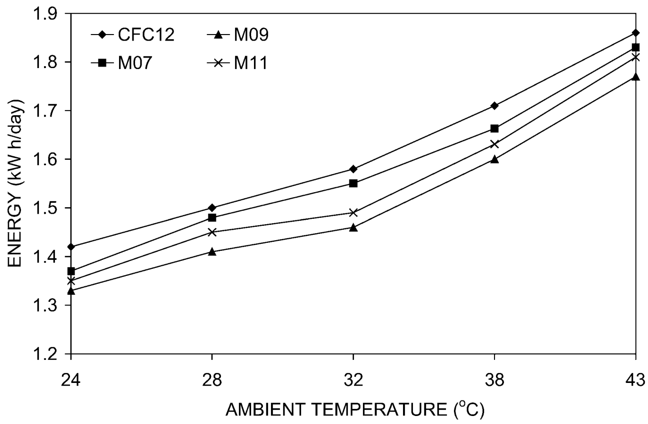

Fig. 6. Variation of energy consumption per day with respect to ambient temperatures.

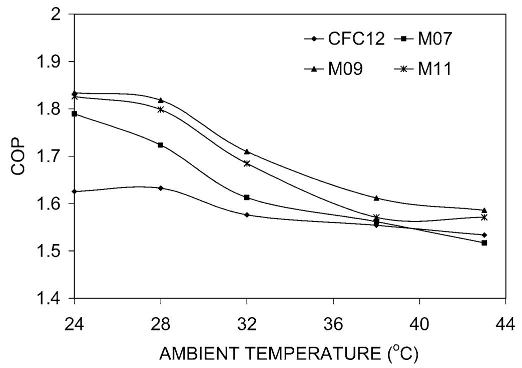

Fig. 7. Variation of COP with respect to ambient temperatures.

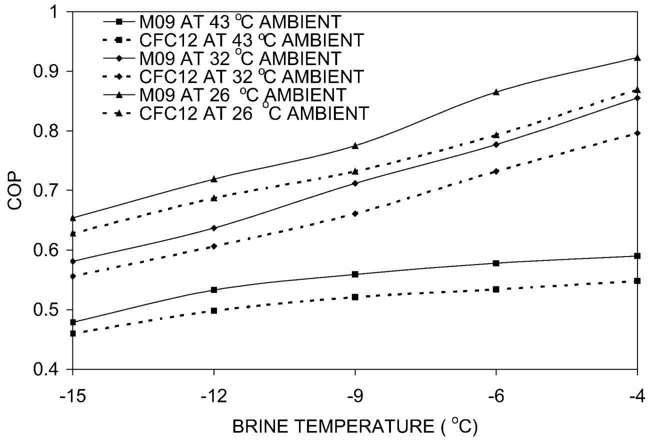

Fig. 8. Actual COP with respect to brine temperature for ambient temperatures 26, 32 and 43 °C.

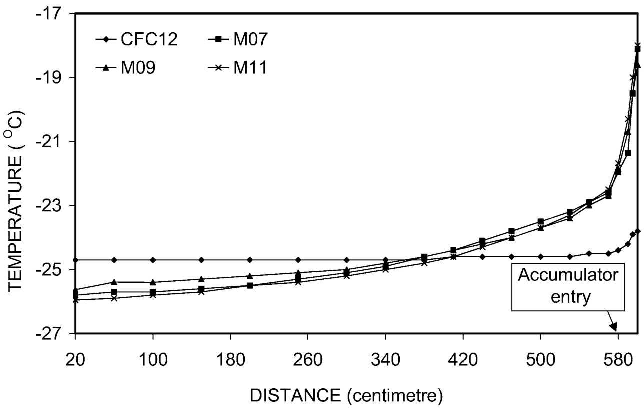

Fig. 9. Temperature distribution along the length of the evaporator coil (ambient = 24 °C).

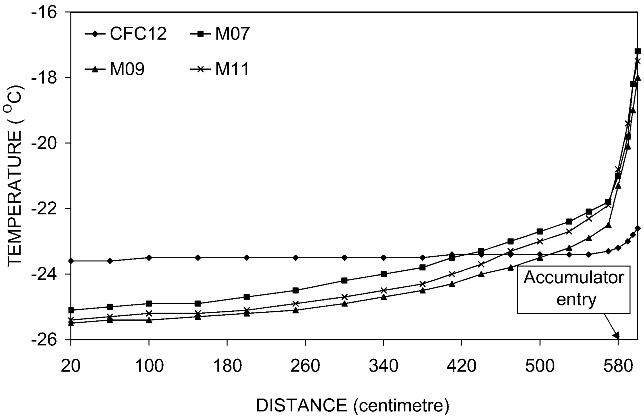

Fig. 10. Temperature distribution along the length of the evaporator coil (ambient = 28 °C).

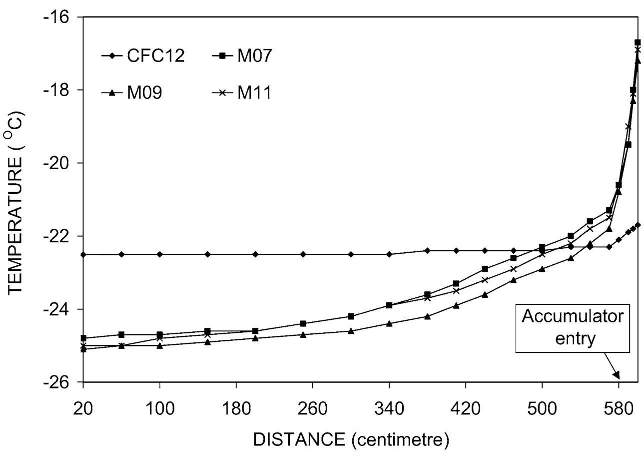

Fig. 11. Temperature distribution along the length of the evaporator coil (ambient = 32 °C).

performance of the freezer compartment. It is also observed that when the ambient temperature increases, the average temperature of evaporator coil was 1.5 to 6% lower than that of CFC12 systems. Hence better performance of the refrigerator could be attained.

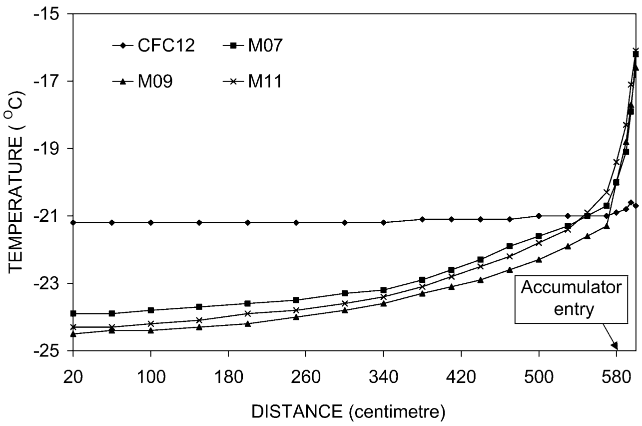

Fig. 12. Temperature distribution along the length of the evaporator coil (ambient = 38 °C).

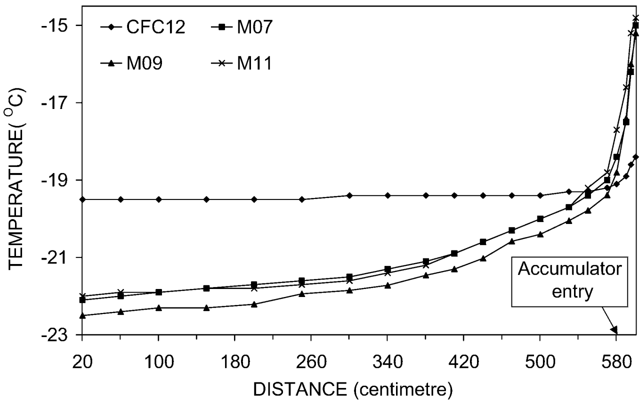

Fig. 13. Temperature distribution along the length of the evaporator coil (ambient = 43 °C).

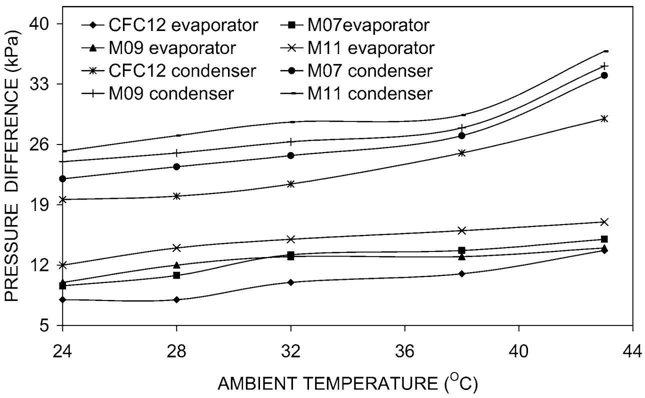

Fig. 14. Pressure difference across the condenser and evaporator with respect to ambient temperature.

The pressure difference across the condenser and evaporator is also plotted in Fig. 14. It shows higher values than that of CFC12 system due to the higher vapour pressure of the mixture as seen in Fig. 1 itself. The compressor dome temperature was also measured for all the tests carried out and the results are plotted in Fig. 15. It is found that the new mixture reduces the compressor operating temperature also and this can extend compressor life. It is inferred that

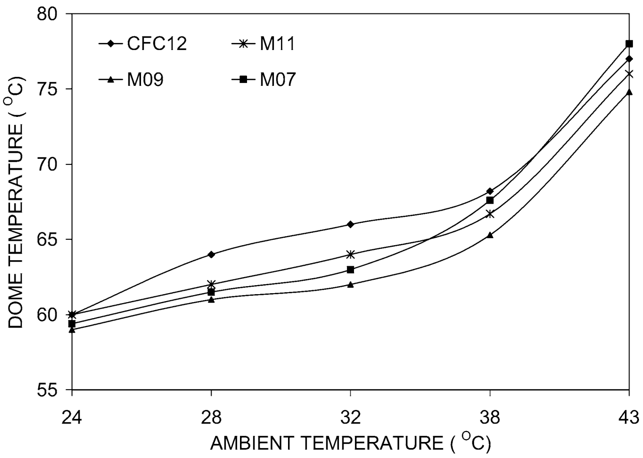

Fig. 15. Variation of compressor dome temperature with respect to ambient temperature.

even in respect of the dome temperature M09 evinces better performance characteristics. However, the rise in dome temperature for M07 is due to the higher pressure ratio of M07 mixture. It is recalled that the same characteristic has been reflected with a drop in COP shown in Fig. 7. These two observations are complementing each other. The higher pressure ratio of M07 can be attributed to the fact that it has a higher percentage of HFC134a which is obviously having a higher pressure ratio than CFC12.

Thus in all respects M09 proves to be a superior substitute for CFC12 and its compatibility with mineral oil is significantly beneficial in the context of CFC12 phase out.

5. Conclusions

The behaviour of HFC134a/HC blend refrigerant mixture with mineral oil as the lubricant to the compressor has been experimentally analysed in a domestic refrigerator and the following conclusions are made:

- The M09 mixture has been identified as a promising alternative to conventional CFC12 system.
- The M09 mixture could reduce energy consumption by 4.8 to 6.4% in a conventional CFC12 refrigerator operating at −15 °C evaporator cabin temperature.
- The improvement in theoretical COP for M09 mixture is 3 to 12%.
- The actual COP of the M09 is 3 to 8% higher than that of CFC12 for calorimeter temperature ranging from −15 to −4 °C.
- The overall performance of M09 is superior to CFC12 at all normal ambient conditions.
- It is also confirmed that the temperature glide due to zeotropic nature of the refrigerant mixture is well within the acceptable limit of 3 °C.
- Even though the pressure drop is more as compared to CFC12 since COP is better than CFC12 this refrigerant mixture is superior. The oil return aspect also has been ensured in this study by continuously running the system for 8,000 h.
- During the tests, none of the existing components were replaced/modified and hence for the small-scale service sector this simple retrofitting method would be a better choice to tackle the service issues in the near future while CFCs are to be phased out.
- The M09 mixture will also reduce the indirect global warming by reducing the CO_2 emissions in power plants because of its higher energy efficiency.

Thus the techno economic viability of retrofitting CFC12 domestic refrigerators with M09 mixture is proved.

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References

[1] M. Janssen, F. Engels, The use of HFC134a with mineral oil in hermetic cooling equipment, Report 95403, No. 07. Presented in the 19th International Congress of Refrigeration, The Hague, 1995.
[2] E.B. Muir, Commercial refrigeration and CFCs, Internat. J. Refrigeration 13 (1995) 106–112.
[3] A. Theodore, H.H. Michael, Refrigerants and energy efficiency, Internat. J. Refrigeration 13 (1990) 270–273.
[4] D.S. Jung, R. Radermacher, Performance simulation of single-evaporator domestic refrigeration charged with pure and mixed refrigerants, Internat. J. Refrigeration 14 (1991) 223–232.
[5] N.E. Carpenter, Retrofitting HFC134a into existing CFC12 systems, Internat. J. Refrigeration 15 (1992) 332–339.
[6] R. Camporese, G. Bigolaro, S. Bobbo, Experimental evaluation of refrigerant mixtures as substitutes for CFC12 and R502, Internat. J. Refrigeration 20 (1997) 22–31.
[7] M.S. Kim, W.J. Mulroy, D.A. Didion, Performance evaluation of two azeotropic refrigerant mixtures of HFC134a with HC290 and HC600a, Trans. ASME 116 (1994) 148–154.
[8] J. Dongsoo, B.K. Chong, H.L. Byong, W.L. Hong, Testing of a hydrocarbon mixture in domestic refrigerator, ASHRAE Trans. Symposia AT-96-19-3 (1996) 1077–1084.
[9] E. Baskin, Synopsis of residential refrigerator/freezer alternative refrigerants evaluation, ASHRAE Trans. (1998) 266–273.
[10] N. Kamal, R.J. Bandorawalla, M.S. High, K.A.M. Gasem, Thermodynamic performance of nonazeotropic HFC134a/HFC143a refrigerant mixtures, Heat Transfer Eng. 21 (2000) 46–54.
[11] S. Motoshi, Physical and chemical properties of refrigeration lubricants, ASHRAE Trans. (1999) 983–993.
[12] K. Stephan, Two-phase heat exchange for new refrigerants and their mixtures, Internat. J. Refrigeration 18 (1995) 198–209.
[13] Ch. Alok, R.S. Agarwal, Mixture of HFC134a and HC600a a promising substitute to CFC12, IIF–IIR Commission B1, B2, E1 and E2, New Delhi, India, 1998, pp. 182–190.
[14] H. Kruse, F. Wieschollek, Concentration shift when using refrigerant mixtures, ASHRAE Trans. (1997) 747–755.

[15] R.W. James, J.F. Missenden, The use propane in domestic refrigerator, Internat. J. Refrigeration 15 (1992) 92–100.
[16] S. Devotta, M. Murthy, N. Sawant, Performance of two door refrigerators retrofitted with a HC Blend, IIF–IIR Commission B1, B2, E1 and E2, New Delhi, India, 1998, pp. 210–220.
[17] D. Colbourne, An Overview of Hydrocarbons as Replacement Refrigerants in Commercial Refrigeration and Air Conditioning, Refrigeration 2000, Northern Ireland Centre for Energy Research and Technology, 2000, pp. 1–15.
[18] I.L. Maclaine-cross, E. Leonardi, Why hydrocarbons save energy, AIRAH J. 51 (1997) 33–37.
[19] D. Colbourne, K.O. Suen, Assessment of performance of hydrocarbon refrigerant. IIF–IIR Comision B1, B2, E1 and E2, 2000, Purdue University, USA.
[20] I.L. Maclaine-cross, E. Leonardi, Performance and safety of LPG refrigerants, in: Proceedings of the Fuel for Change Conference, Paradise Queensland, Australia, 1995, pp. 149–168.
[21] D. Colbourne, T.J. Ritter, Compatibility of non-metallic materials with hydrocarbon refrigerants and lubricant mixtures, IIF–IIR Commission B1, B2, E1 and E2, 2000, Purdue University, USA.
[22] P.A. Domanski, D.A. Didion, Evaluation of suction-line/liquid-line heat exchange in the refrigeration cycle, Internat. J. Refrigeration 17 (1994) 487–493.
[23] R.B. Frank, R.P. Dennis, Modelling and testing of fractionation effects with refrigerant blends in an actual residential heat pump system, ASHRAE Trans. (1997) 781–794.