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Research Article

Enhancement of household refrigerator energy efficiency by studying the effect of refrigerant charge and capillary tube length

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ABSTRACT

Experiments were carried out initially by considering pure R134a and hydrocarbon refrigerant mixtures such as (R290/R600a) (HCM1 44/56, HCM2 50/50, HCM3 54/46, HCM4 64/36, and HCM5 74/26 wt %). Tests are conducted at the atmospheric temperature of 30°C in a domestic refrigerator system. The performance parameters such as pull-down time, desired effect, power consumption, and running cost of the system are to be analysed at different evaporator temperatures, the mass of refrigerant and varying length of capillary tubes. To evaluate the refrigeration effect, Power consumption and COP of the domestic refrigerator at various freezer temperatures (–9°C, –12°C & –15°C) were selected. Results report that out of all the alternative mixtures the amount of energy input was less consumed in the case of HCM1 at a minimal expansion length of 6.3 mm. In the case of HCM5, the least energy was consumed at a capillary length of 5.24 mm whereas in the case of R134a it was at 3.3 mm.

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INTRODUCTION

Refrigerant R134a is considered environmentally safe and is virtually non-toxic (1993), but its Global Warming Potential is very high (1300). The Kyoto Protocol of the United Nations Framework Convention on Climate Change (UNFCCC) asked for a reduction in the emission of six categories of greenhouse gases, including R134a, used as a refrigerant in domestic refrigerators (2005). According to the Kyoto protocol (1997), the consumption of R134a must be phased out before January 1, 2024. From the environmental, ecological, and health point of view, there is an urgent need to find better substitutes for high global warming refrigerants. Suresh Bhakta Shrestha et al. [1] have analysed the working of refrigerants R290 and R600a mixtures in vapour compression refrigeration system for simultaneous cooling and heating applications when compared with R134a, R114, and R236ea. Results reported that pure R290 offers a significant pressure difference between condensation and evaporation usually more than 18 bar, is challenging to achieve in reciprocating compressors, though its outlet temperature is the same as that of R134a. It is also reported that the refrigeration system designed for R134a and R326a is not

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suitable for R290. Somchai Wongwises et al. [2] have carried out an experimental analysis on vapour compression refrigeration systems with different hydrocarbon substitutes as a replacement of R134a. Initially, 239-litre capacity refrigeration with R134a as the refrigerant is chosen. At a surrounding temperature of 25°C experiments are conducted with various refrigerants at no-load condition. Fatouh et al. [3] has carried out a theoretical analysis to study the usage of hydrocarbon substitutes in domestic refrigerators. By considering various working fluids such as R134a, propane, commercial butane etc. with different ranges of evaporator and condenser temperatures are studied to evaluate the performance characteristics of the refrigerator. The findings from his study helped to identify the replacement of R134a with hydrocarbon refrigerant mixtures. M.A. Sattar et al. [4] has carried out the analysis to study the performance of refrigeration system with different working fluids and compare it with R134a. The investigation has been carried out to study the impact of condenser and evaporator temperature on the desired effect, running cost, heat rejection ratio etc. From the results it has been reported that the work input to the compressor is 25–30% less than the power consumed by the refrigerator with R134a as the refrigerant. Mani et al. [5–6] have done the experimental study with saturated hydrocarbons as the refrigerant on vapour compression refrigeration system. Experiments were carried out with R12 and R134a as the refrigerants. The results reported that the saturated hydrocarbons have a higher cooling capacity, with 50.1% greater than R12 and 87.2% than R134a. However, saturated hydrocarbon mixture has consumed 6.8 to 17.4% more energy than R134a. The refrigerant mixture 68% R290 and 32% R600a has been considered as a replacement for R134a. Mohanraj et al. [7–8] have done the experimental analysis on a 200-liter domestic refrigeration system to study the effects of R290 and R600a refrigerants. The experiments were carried out at different operating conditions. The results reported that the refrigierants considered for the analysis has been consumed lesser energy input, optimal pull-down time with higher COP than compared to R134a. Ching-Song Jwo et al. [9] have done an experimental investigation to study the impact of a zeotropic mixture comprising of 50% R290/50% R600a on the refrigeration system in comparison with R134a. Tests were carried out with different mass compositions of the zeotropic mixture. The results reported that zeotropic mixture shows better-desired effect with lesser energy input and higher COP. Deepak Paliwal and S.P.S. Rajput [10] has conducted a theoretical simulation work with pure hydrocarbon refrigerants R290 and R600a. The results show that R600a/R290 (60/40 by wt %) mixture can be used as a drop-in replacement refrigerant for HFC134a, for the given operating conditions. The overall performance has proved that the 100 g of R600a/R290 (60/40 by wt%) is the best long-term alternative refrigerant to phase out R134a. Saravana Kumar and Selladurai [11, 12], have proposed a new method to increase the available energy of saturated hydrocarbons mixture of R290/R600a on a domestic refrigerator. By involving exergy analysis the performance of different refrigerants is analysed. The impact of sink temperature with R134a and R290/R600a as the refrigerants were experimentally studied to evaluate energy efficiency ratio, COP, and efficiency defect in different components of the refrigeration system. The results reported that the COP of saturated hydrocarbon mixture has improved substantially when compared with R134a. Dongsoo Jung et al. [13] have analysed the theoretical analysis of a saturated hydrocarbon mixture R290/R600a in a refrigeration system. The analysis reported that the composition of 0.2–0.6 mass fraction of R290 gives better COP when compared to R12. Results reported that in the case of R290/R600a mixture 3–4% higher energy efficiency, a faster cooling rate than R12 has been observed. Bilen et al. [14] had theoretically investigated the possible alternative replacement of R134a with R152a (GWP=120) in the air conditioning system (AC) in the automobile. Tiwari et al. [15] have selected and studied experimentally four ozone-friendly hydro-fluorocarbon refrigerants (R125, R134a, R143a, and R152a) to replace R12 in a vapour compression refrigeration system. It is observed that the refrigerant R152a has a higher coefficient of performance (COP) than R12, while R134a has a slightly lower COP and higher refrigerating capacity than R12. Due to its high global warming potential (GWP) of R134a, the R152a is preferred as a working fluid in a vapour compression refrigeration system. Dalkilic & Wongwises [16] have made a theoretical performance study on a vapour compression refrigeration system using various alternative refrigerants such as R134a, R152a, R32, R290, R1270, R600, and R600a. The results were compared with R12, CFC22, and HFC134a as possible alternative replacements. Results reported that HFC152a gives a better alternative to HFC134a refrigerants. A. Baskaran and P. Koshy Mathews [17] have made theoretical performance analysis on a vapour compression refrigeration system with various eco-friendly refrigerants such as R152a, R32, R290, R1270, R600a, and R170. The results reported that the alternative refrigerants R170, R152a, and R600a have a slightly higher COP than R134a for the condensation temperature of 50°C and evaporating temperatures ranging between 30°C and −10°C. R.K. Roy [18] has applied the Taguchi method to find an optimum condition. The process provides not only the standard sets of orthogonal arrays but also a plan to analyse results according to the Signal-to-Noise (S/N) ratio. Significant factors are identified by analysis of variance (ANOVA). Finally, the prediction of the optimum combination of factors is made. Lu et al. [19] have applied the Taguchi method to determine the optimum design for a solar water heater with natural circulation. The performance characteristics of three ozone-friendly refrigerants like R32, R134a, and R152a theoretically were studied [20, 21]. Comparison among the investigated refrigerants confirmed that R152a and R134a have approximately the same performance, but the best performance was obtained from the use of R152a in the system.

The open literature review provides sufficient information on the vapour compression refrigeration system using R152a as a refrigerant. However, R152a has low GWP (120), and hence research on R152a as a refrigerant, most of the investigations were confined to mobile air-conditioning and theoretical studies. However, a detailed performance study is not available in the determination of maximum charge with a change in ambient temperatures and comparison of various performance characteristics such as pull-down time, refrigeration effect, power consumption, and COP. Therefore, in this research work, the study on domestic refrigeration systems using R152a as one of the refrigerant blends has been selected. To improve the performance of the refrigeration system it is important to study and apply the design of experimental techniques to analyse the process design. By implementing the experimental techniques, the performance of the system can be improved by optimising the process. By considering multiple input parameters the performance of the system output can be achieved by applying the statistical techniques in the design of experiments by Philip & Ross. The remaining of the paper is organised as follows: Experimental setup and methodology is discussed in section 2 followed by results and discussion in section 3 and 4.

EXPERIMENTAL SETUP

A test rig is the modified design of a domestic refrigerator. The arrangement of a system is shown in Fig. 1. The major components of a test rig work with R134a as a working refrigerant with a total capacity of 175 L. It consists of a deep freezer, a hermetically sealed reciprocating compressor, air-cooled condenser, strainer, and five different capillary tube lengths are used in this study. Capillary tubes are installed with the help of a brazing method. The capillary tubes have 5 different lengths i.e., 3.96 mm, 4.26 mm, 4.57 mm, 4.87 mm, and 5.18 mm of capillary tubes at a constant diameter of 0.78 mm are used in the experimental work.

The performance factors such as, refrigeration effect, energy input, and maintenance cost for different expansion valve lengths with 0.78 mm of capillary diameter are shown in Tables 1 and 2. The tests were carried out based on ISO 8187 methodology. Initially to carry out the analysis optimal length of capillary tube is required. As per design the capacity of the refrigerator working with R134a and refrigerant charge 180 g was filled in the system for performing a reference test. However, the performance tests are conducted with R134a with different capillary tube lengths such as 3.96 mm, 4.26 mm, 4.57 mm, 4.87 mm, and 5.18 mm at a different mass ratio i.e. 140 g, 160 g and 180 g. The tests were carried out continuously by maintaining the above conditions until the evaporator temperature of −15°C is achieved as the reference temperature. For the period of testing, the ambient temperature is maintained at 30°C, for changing capillary tube length and varying weight of refrigerant charge. Therefore, in the experimental observations, the possibility of hydrocarbon refrigerant mixtures (HCM) used with polyester (POE) oil as lubricant oil. In this system, blended hydrocarbon refrigerants are used in the form of a liquid state and these blends are calibrated with the help of an electronic balancing machine, having a precision is ±0.01 g. In order to minimize the investigational work, to take average values to be measured. The dissimilarity in investigational values from the regular value is maintained within ±5% error.

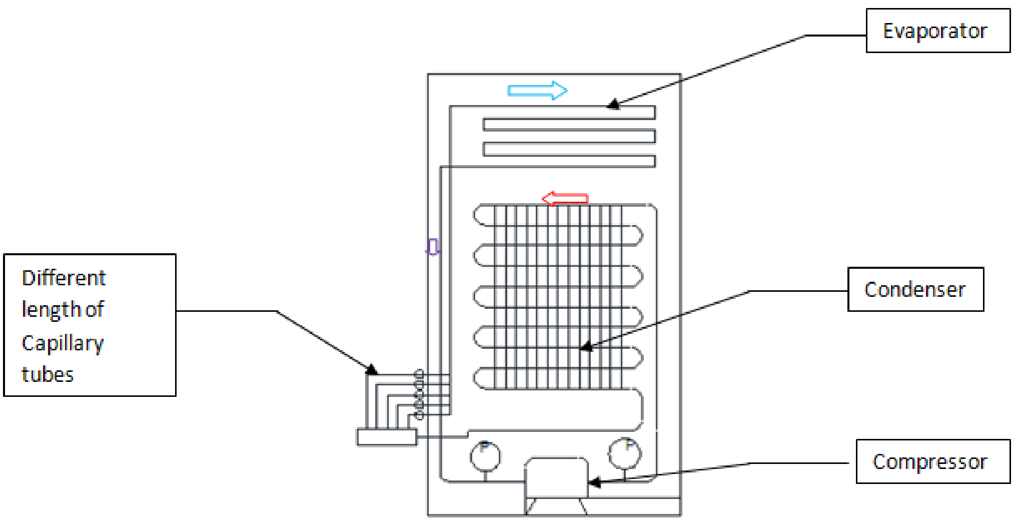

Figure 1. Schematic diagram of measurement system in experimental setup.

Table 1. Technical specifications of domestic refrigerator test rig

Description	Range
Storage Volume	175 L
Current rating	1.1 amp max
emf	220-240 V
Frequency	50 Hz
No. of Doors	1
Refrigerant type R134a	R134a
Defrost System	Auto defrost
Refrigerant charged	180 g

Table 2. Measured quantities with their range and accuracy of instruments

Quantity	Range	Accuracy
Temperature	–40°C to 110°C	±0.1°C
Power consumption	0 to 1000 W	1 W
Voltage	0 to 240 V	0.1 V
Current	0 to 10 A	0.1 A
Pressure	0 to 150 MPa	±0.7 kPa
Refrigerant flow meter	0 to 100 cc/s	0.1 cc/s

Table 3. Test rig experimental ranges

Freezer compartment	–15°C to –9°C
Food compartment	3°C to 5°C
Steady ambient temperature	29°C to 35°C

TEST PROCEDURE

As per the guide lines given by ASHRAE hand book 2010, the energy consumption test and no-load pull-down test were conducted with the following system parameters as shown in the table 3.

RESULTS AND DISCUSSION

Performance Parameters of a Domestic Refrigeration system on Refrigeration Effect, Compressor Power and COP

The analysis was carried out as per the process followed in experimental procedure at an atmospheric temperature of 30°C with different capillary tube lengths and simultaneously varying the flow rate of refrigerant to determine the performance of a refrigeration system. Results reported that out of all the alternative mixtures the energy input was less consumed in the case of HCM1 at a minimal expansion length of 6.3 mm. In the case of HCM5, the least energy was consumed at a capillary length of 5.24 mm whereas in the case of R134a it was at 3.3 mm. The results concluded that the performance test for the wide range of mixtures of HCM1 to HCM5 from 3.96 mm to 5.18 mm.

Out of all the variety of mixtures, larger quantity of energy input is observed in the case of HCM1 and the least is observed in the case of HCM5. The desired effect is observed to be greater in the case of HCM1 to HCM4 and then decrease due to the length of the capillary is more for HCM5. The running cost of the refrigeration system is based on the energy input to the compressor, and the refrigeration effect. The tests were carried out continuously with similar parameters and the mean values are evaluated for the comparison of results. The desired effect, energy input to the compressor and maintenance cost are evaluated by maintaining different evaporator temperatures and changes in capillary tube length as shown in Figures 2 to 10.

From Figures 2 to 4, it is observed that the RE at –9°C in case of HCM5 initially increases and then decrease whereas HCM4 gives maximum RE at 106 g. RE at –12°C in case of HCM4 provides maximum RE at 96 g. RE at –15°C HCM4 continually improve from 4.57 mm of the capillary tube, and it reaches to 220 kW at 106 g. Finally, the RE gives the

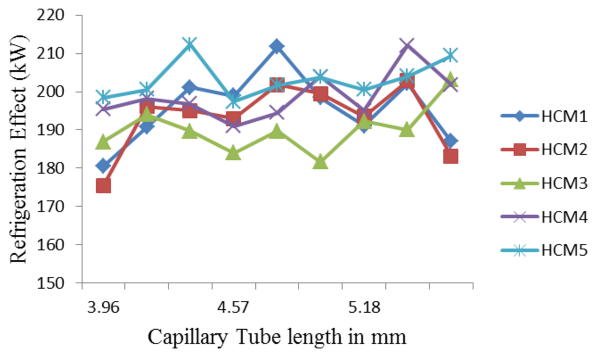

Figure 2. Variation of Refrigeration Effect with alternative mixtures with varying capillary lengths at –9°C evaporator temperature.

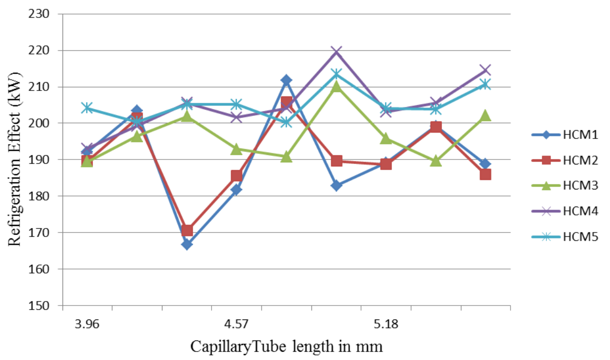

Figure 3. Variation of Refrigeration Effect with alternative mixtures with varying capillary lengths at –12^0 C evaporator temperature.

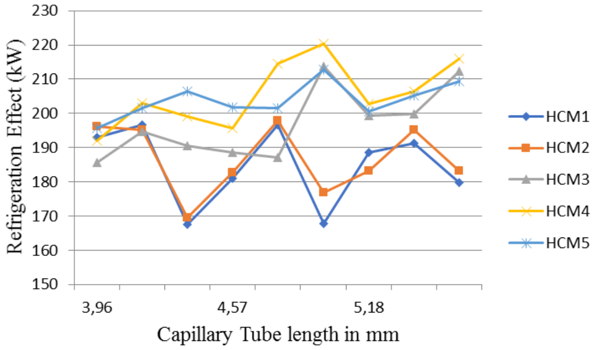

Figure 4. Variation of Refrigeration Effect with different mixtures with varying capillary lengths at –15°C evaporator temperature.

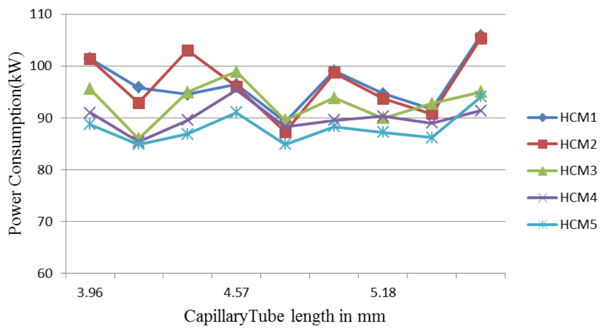

Figure 5. Variation of Power Consumption with different mixtures with varying capillary lengths at –9°C evaporator temperature.

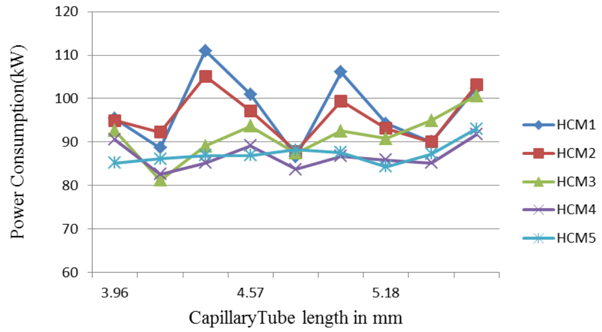

Figure 6. Variation of Power Consumption with different mixtures with varying capillary lengths at –12^{0} C evaporator temperature.

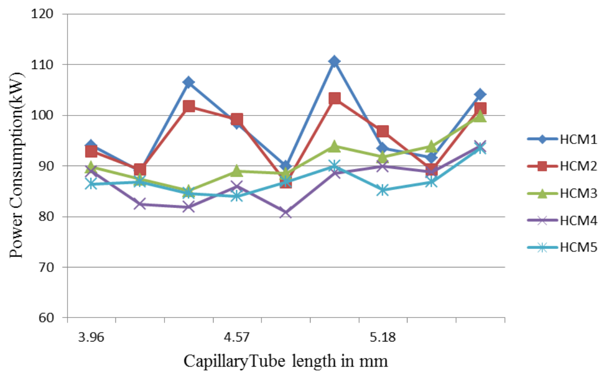

Figure 7. Variation of Power Consumption with different mixtures with varying capillary lengths at –15°C evaporator temperature.

maximum for HCM4 when compared to remaining HCM's due to above condition the RE increases and decreases.

From Figures 5 to 7, it is observed that the compressor power consumption increases up to HCM3 and then decreases. The minimum power consumption maintained at HCM4 with a 4.57 mm capillary length and –15°C is observed to be 80.81 kW.

From Figures 8 to 10, COP starts increasing from HCM1 to HCM3 and reaching maximum at HCM4 are 2.65 with 4.57 mm and –15°C then COP starts decreasing from HCM4 to HCM5. From HCM1 to HCM3 it is observed that the reduction in the desired effect is smaller when compared to the reduction in energy input. Whereas for HCM4 refrigeration effect increases and reduction in the desired effect is observed in the case of HCM5.

Based on the series of experiments conducted of all the alternative refrigerant mixtures better COP with minimum energy input is observed in the case of HCM4 in comparison with domestic refrigerant R134a. The performance parameters of the system are desired effect, power consumption and COP are finding optimisation of the system with a help of a statistical technique by involving three variables at different levels by applying orthogonal array L27.

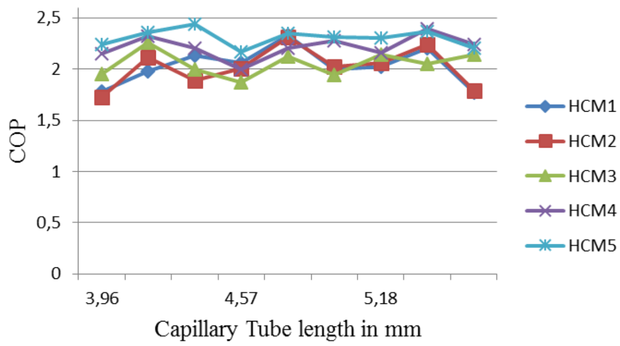

Figure 8. Variation of COP with different mixtures with varying capillary lengths at –9°C evaporator temperature.

Taguchi Method Based Design of Experiments (DOE)

To improve the performance of the refrigeration system by optimising the design parameters can be obtained by implementing the statistical tool by segregating the layout of the parameters. The purpose of the Taguchi method is to develop the set of experiments and give the best possible results with a minimum number of experiments as shown in Figure 11.

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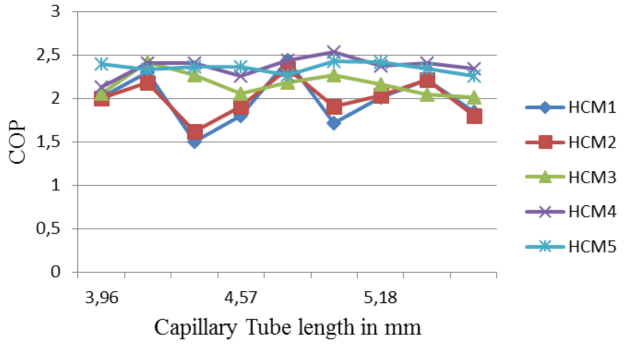

Figure 9. Variation of COP with different mixtures with varying capillary lengths at –12°C evaporator temperature.

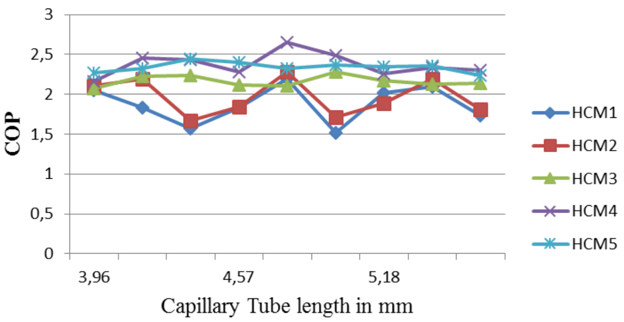

Figure 10. Variation of COP with different mixtures with varying capillary lengths at –15°C evaporator temperature.

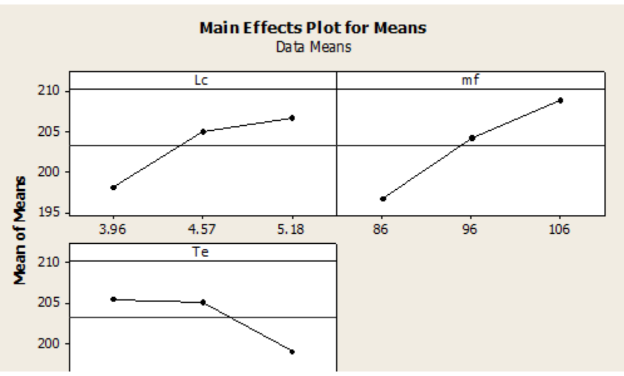

Figure 11. Effect of process parameters of RE for HCM4 Mixture.

From Tables 4–6, the maximum mean value of refrigerating effect was found at level 3 corresponding to the capillary length of 5.18 mm and at level 3 corresponding to charge a mass of 106 g and at level 1 corresponding to evaporator temperature of –15°C.

Expected Value of Refrigeration Effect

Based on the tests carried out, the minimum parameters are determined as shown in Figure 12. The mean values at each individual stage are taken from Table 7 and the expected values of the refrigerating capacity are taken below.

From Table 8, the minimum mean value of power consumption was found at level 1 corresponding to capillary length of 3.96 mm and at level 2 corresponding to charge the mass of 96 g and at level 3 corresponding to evaporator temperature of –12°C.

Predicted Value of Energy Input to the Compressor

Based on the test carried, the minimum parameters are determined as shown in Table 9 and Figure 13. The mean values at each individual stage are taken as shown in the table and the expected values of the energy input are taken below.

CONCLUSION

The analysis was carried out as per the process followed in experimental procedure at an atmospheric temperature of 30°C with different capillary tube lengths and simultaneously varying the flow rate of refrigerant to determine the performance of a refrigeration system. Results report that out of all the alternative mixtures the amount of energy input was less consumed in the case of HCM1 at a minimal expansion length of 6.3 m. In the case of HCM5, the least energy was consumed at a capillary length of 5.24 mm whereas in the case of R134a it was at 3.3 mm. The results concluded that the performance test for the wide range of mixtures of HCM1 to HCM5 from 3.96 mm to 5.18 mm. Out of all the variety of mixtures larger quantity of energy, input is observed in the case of HCM1 and the least is observed in the case of HCM5. The desired effect is observed to be greater in the case of HCM1 to HCM4 and then decrease due to the length of the capillary is more for HCM5. The running cost of the refrigeration system is based on the energy input to the compressor and the refrigeration effect.

Table 4. Process Parameters and their levels

Processes Parameters	Units	Range	Level 1	Level 2	Level 3
Capillary Length	Meters	3.96–5.18	3.96	4.87	5.18
Refrigerant Charge	Grams	86–106	86	96	106
Evaporator Temperature	Degree Celsius	–15––9	–15	–12	–9

Table 5. Experimental results of RE for HCM4 Mixture

Experimental Runs	Length of Capillary Tube (meter)	Mass of Refrigerant (grams)	Evaporator Temperature (°C)	RE (KW)	SN Ratio	Mean
1	3.96	86	–15	192.07	45.6692	192.07
2	3.96	86	–12	193.05	45.7134	193.05
3	3.96	86	–9	195.4	45.8185	195.4
4	3.96	96	–15	202.94	46.1474	202.94
5	3.96	96	–12	199.23	45.9871	199.23
6	3.96	96	–9	198.16	45.9403	198.16
7	3.96	106	–15	199.13	45.9827	199.13
8	3.96	106	–12	205.62	46.2613	205.62
9	3.96	106	–9	196.79	45.8801	196.79
10	4.57	86	–15	195.75	45.834	195.75
11	4.57	86	–12	201.6	46.0898	201.6
12	4.57	86	–9	190.95	45.6184	190.95
13	4.57	96	–15	214.45	46.6265	214.45
14	4.57	96	–12	204.1	46.1969	204.1
15	4.57	96	–9	194.47	45.7771	194.47
16	4.57	106	–15	220.35	46.8623	220.35
17	4.57	106	–12	219.48	46.8279	219.48
18	4.57	106	–9	203.96	46.1909	203.96
19	5.18	86	–15	202.69	46.1366	202.69
20	5.18	86	–12	203.19	46.158	203.19
21	5.18	86	–9	195.08	45.8043	195.08
22	5.18	96	–15	206.52	46.2992	206.52
23	5.18	96	–12	205.62	46.2613	205.62
24	5.18	96	–9	212	46.5267	212
25	5.18	106	–15	216	46.6891	216
26	5.18	106	–12	214.47	46.6273	214.47
27	5.18	106	–9	204.78	46.2258	204.78

Table 6. Response table for the signal to noise ratios for HCM4 Mixture

Level	Lc	MF	Te
1	198.0	196.6	**205.5**
2	205.0	204.2	205.2
3	206.7	209.0	199.1
Delta	8.7	12.3	6.5
Rank	2	1	3

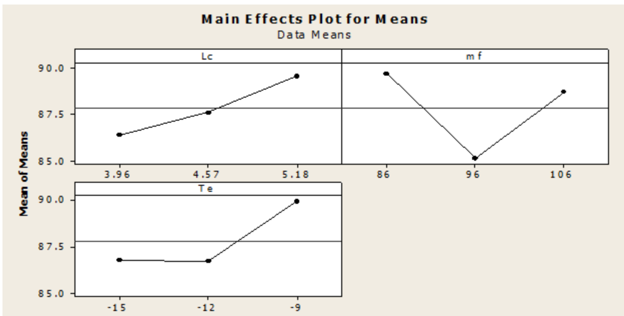

Figure 12. Effect of process parameters of POWER for HCM4 Mixture.

Table 7. Optimum conditions for RE

Factors	Level	Value
Capillary Length (m)	1	5.18 m
Charge mass (g)	2	106
Evaporator Temperature (°C)	3	−15

Table 8. Response table for the signal to noise ratios for HCM4 Mixture

Level	Lc	MF	Te
1	**86.39**	86.67	86.78
2	87.57	**85.12**	**86.75**
3	89.53	88.70	89.95
Delta	3.15	4.55	3.20
Rank	3	1	2

Table 9. Optimum conditions for Power Consumption

Factor	Level	Value
Capillary Length (m)	1	3.96 m
Charge mass (g)	2	96
Evaporator Temperature (°C)	3	−12

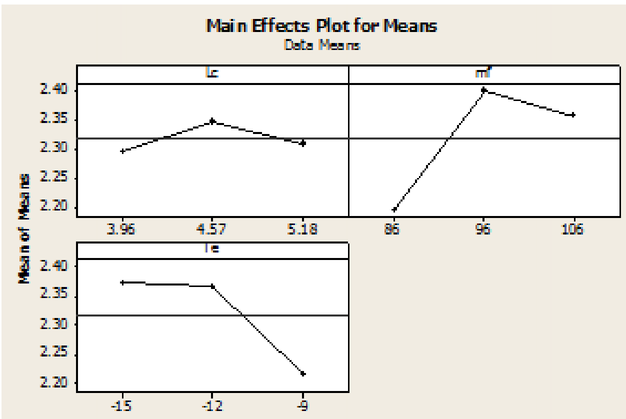

Figure 13. Main effects of the process parameters of means of COP for HCM4 Mixture.

NOMENCLATURE

L	Capillary tube length
RE	Refrigeration Effect
HCM	Hydrocarbon mixture
M_f	Mass of the refrigerant
T_e	Evaporator temperature
R	Refrigerant
COP	Coefficient of Performance
GWP	Global Warming potential
S/N	Signal to noise ratio

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This manuscript has not been published and is not under consideration for publication elsewhere. We have no conflicts of interest to disclose.

AUTHORSHIP CONTRIBUTIONS

Santhosh Kumar Gugulothu: Concept, design, analysis, drafting and writing.

DATA AVAILABILITY STATEMENT

No new data were created in this study. The published publication includes all graphics collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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