International Journal of Heat and Technology
Vol. 44, No. 3, June, 2026, pp. 1241-1252
Journal homepage: http://iieta.org/journals/ijht

Energy-Based Performance Enhancement of a Domestic Refrigerator Retrofitted with Low-GWP Hydrocarbon Refrigerants

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https://doi.org/10.18280/ijht.440329

Received: 10 April 2026
Revised: 3 June 2026
Accepted: 16 June 2026
Available online: 30 June 2026

Keywords:
hydrocarbon refrigerants, retrofitting, charge optimization, low-GWP refrigerants, coefficient of performance, energy efficiency

ABSTRACT

Increased regulatory constraints and the high global warming potential of R134a have made the use of low-GWP refrigerants in domestic refrigeration systems quite popular. The current study explores the thermodynamic performance and energy efficiency of a retrofitted domestic vapor compression refrigerator using selected hydrocarbon refrigerants with a constant capillary tube length of 3 m. Realizing the importance of mass charge as the critical variable in the performance of small-capacity systems, a range of experiments was conducted by optimizing the charge and keeping the geometry constant to investigate the effect of the refrigerants. The refrigerant mass charge affects the superheating in the evaporator, the sub-cooling in the condenser, suction and discharge pressure, cooling power, and the life of the compressor. Any deviation in the charge quantity can lead to significant changes in performance and the energy requirement. The experimental results have been supported with thermophysical evaluation using REFPROP and MATLAB optimization for refrigerating effect (RE), mass flow rate (MFR), compressor work, actual cc, isentropic efficiency, and energy requirement. In general, the hydrocarbon refrigerants that were optimized in this study displayed better thermodynamic efficiency compared to R134a in the constant 3 m length of capillary tube setup. Among the refrigerants used, propylene was found to have the highest coefficient of performance (COP) obtained through MATLAB interpolation (8.8), while the experimental value at 20 g charge was 8.7. Meanwhile, Dimethyl-ether, propane, and isobutane also showed good energy efficiency in their optimum charge quantities. Propylene showed the maximum optimized COP (8.8) with lower compressor work and energy consumption. For certain refrigerants, optimizing the amount of refrigerant decreased the energy consumption by the compressors without affecting the cooling efficiency. But when the amount of refrigerant was excessively reduced for certain refrigerants, the cooling efficiency was greatly reduced. These results demonstrate that optimized charge hydrocarbons are more energy efficient, environmentally friendly, and technically feasible for domestic refrigerators.

1. INTRODUCTION

Domestic refrigeration technology continues to be one of the most significant engineering inventions made in contemporary times. This technology not only contributes to the conservation of food and medicine but also ensures the proper functioning of industrial activities and the maintenance of thermal comfort within buildings [1, 2]. For instance, in regions like Africa, which experience higher temperatures with limited access to power supply, domestic refrigeration technology assumes great socio-economic importance [3]. It helps secure food sources, minimizes post-harvest losses, and ensures public health safety. However, the same refrigerators that bring about such positive impacts also cause increased electricity usage and emission of greenhouse gases. Consequently, optimizing the thermodynamic performance and environmental friendliness of domestic refrigerators becomes an urgent need for both engineers and development practitioners. Over eighty percent of cooling applications across the globe utilize vapor-compression refrigeration technology, according to the International Institute of Refrigeration [4]. These are the four major processes in the vapor-compression process: vaporization, compression, condensation, and expansion. The efficiency of this process depends highly on the thermophysical characteristics of the refrigerant, the design of system components, and interactions between refrigerant charge and flow control devices. In residential refrigeration systems, the capillary tube is used as the expansion device. The small changes in the mass charge of the refrigerant may have an impact on the refrigerating capacity, work of the compressor, pull-down time, and coefficient of performance (COP) of the refrigerating unit [5].

The efficiency of a vapor compression refrigerator in the domestic sector depends to a great extent on two parameters which are interrelated: the thermodynamic properties of the refrigerant and the amount of the refrigerant flowing through the cycle. In the case of refrigerators using capillary tube expansion devices, the refrigerant charge influences the level of evaporator superheat, condenser subcooling, pressure ratio, compressor loading, cooling effect, and overall energy consumption. A slight deviation from the optimal refrigerant charge may greatly decrease the efficiency of the system and increase energy consumption. R134a was used extensively in refrigerators due to its good thermodynamic properties and zero ozone depletion potential. Its relatively high global warming potential made scientists seek refrigerants with lower GWP values, especially hydrocarbon refrigerants like propane (R290), propylene (R1270), isobutane (R600a), butane (R600) and Dimethyl-ether (RE170). Previous studies indicated that these refrigerants may be more efficient than R134a or at least demonstrate comparable efficiency while being environmentally friendly. Nevertheless, the performance of hydrocarbon refrigerants differs greatly from each other due to the great sensitivity of refrigeration systems to refrigerant charge and expansion device design [6, 7]. Though a number of studies comparing the performance of hydrocarbon refrigerants against R134a exist, very few studies have taken into account the comparison between hydrocarbons and R134a, along with the optimization of the refrigerant charge under a fixed capillary tube arrangement. This is because it is common practice when retrofitting existing household refrigerators to leave the existing capillary tube unchanged, instead of reconfiguring the whole system. Thus, it becomes very crucial to understand the performance of different refrigerants for different refrigerant charge rates for a fixed capillary tube [8]. The present study addresses this gap by experimentally investigating the performance of selected hydrocarbon refrigerants in a domestic refrigerator operating with a fixed capillary tube length of 3 m. The study evaluates the influence of refrigerant charge on refrigerating effect (RE), compressor work, cooling capacity, COP, system efficiency (SE), and energy consumption. Furthermore, an optimization approach is employed to determine the charge level that maximises performance for each refrigerant. By isolating the effects of refrigerant type and charge while maintaining a constant expansion geometry, the study provides practical guidance for retrofitting existing domestic refrigeration systems with environmentally sustainable refrigerants [9-11].

Several studies have explored alternative refrigerants. However, there are few systematic investigations on the effect of optimization of refrigerant mass charge on the performance of pure hydrocarbon refrigerants in a retrofitted domestic refrigerator. However, these two parameters play a crucial role in thermodynamic optimization. Overcharging raises the amount of work required from the compressor and energy consumed, while undercharging diminishes the cooling effect. The establishment of optimal values of these parameters is vital in ensuring maximal exergy utilization and minimal irreversibility during the process. From the exergy standpoint, refrigeration does not only depend on the energy required and its cooling effect but also on the quality of energy conversion and the minimization of thermodynamic losses. Entropy production is a result of irreversibilities in compression, throttling, and heat transfer stages. Choosing refrigerants with desirable thermodynamic characteristics along with optimal charge and expansion devices will help lower entropy production. This strategy will have practical benefits for energy-poor regions with expensive electricity prices. It is financially unfeasible to replace all existing household refrigerators in some places. Hence, retrofitting is the most feasible alternative that allows technological changes in refrigeration. With the installation of environmentally friendly refrigerants in an existing vapor compression cycle and optimization of operating parameters, it is possible to prolong machine life, reduce greenhouse gas emissions, and increase the cooling effect without making unreasonable expenses on households.

Charge is considered to be one of the key operational parameters of a vapor-compression system. It defines evaporator superheat, condenser sub-cooling, suction and discharge pressures, capacity of a system, as well as general safety and efficiency [12, 13]. Even slight variations in charge quantity by 5-10% can result in noticeable changes. Undercharged systems have a tendency to show high values of superheat, low capacity, and compressor overheating, while an excessive charge results in high head pressure and even mechanical failure. In real-life conditions, improper charge is more often than improper expansion valves cause of the malfunction of the whole refrigeration system. Consequently, charge optimization is a crucial process that should be carried out properly.

This paper describes the experiment aimed at investigating the retrofitting of a domestic vapor-compression refrigerator initially designed for refrigerant R134a with selected pure hydrocarbons as substitutes. This research focuses on performance characteristics analysis with respect to refrigerant mass charge variation. Refreezing effect, compressor power, COP, efficiency, and energy consumption were chosen as key performance characteristics. Additionally, the optimization model was used to ascertain the mass charge that would maximise the actual COP and minimise the compressor work. The main contribution of this paper is the holistic nature of the research process involved. The approach used does not treat the change in the working fluid as merely swapping one fluid for another. Instead, the research acknowledges the interaction between the working fluid characteristics and the flow control geometry. Through this combination of design, construction, experimentation, and optimization, a systematic methodology for improving exergy efficiency in domestic refrigeration systems has been developed.

2. METHODOLOGY

2.1 System configuration

The experimental setup involved a locally designed and constructed domestic Vapor Compression Refrigeration System (VCRS) designed to simulate realistic conditions within a typical home. The schematic configuration of the system is presented in Figure 1, highlighting the various components along with the vapor line and high-pressure side of the refrigeration system. It describes the refrigeration system's operation, emphasizing the circulation of refrigerant through the liquid line and low-pressure side. It references a pressure-enthalpy cycle illustrated in Figure 2, which highlights the different state points utilized to determine transport properties sourced from REFPROP software. Temperature entropy and pressure enthalpy diagrams were used in identifying the state points required for the analysis of performance and exergy losses [14]. It comprised a reciprocating compressor, a naturally convective air cooled condenser, a bare-tube evaporator coil placed inside the cooling chamber, and a capillary tube as the expansion device. The piping for the refrigerant connections was made of copper tubing in order to provide adequate strength and compatibility with the refrigerants employed. Filters and dryers were installed to remove the presence of water, which could damage the compressor during operation. All other main components were chosen from available commercial equipment on the basis of estimated design capacity.

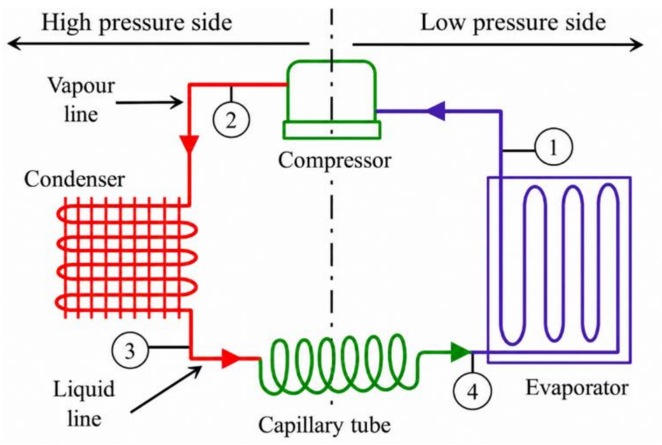


Figure 1. Schematic arrangement of components of Vapor Compression Refrigeration System (VCRS)

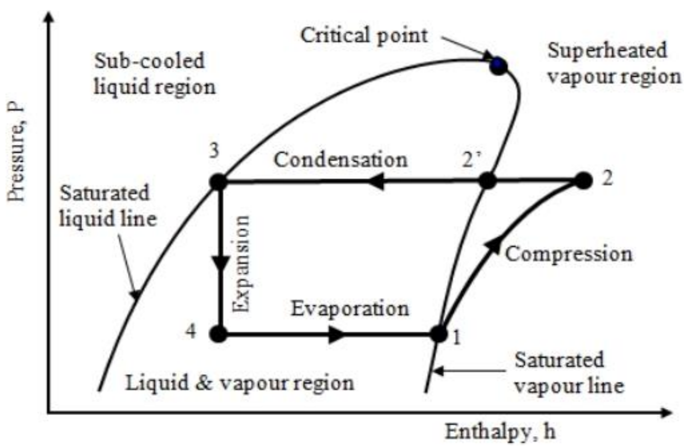
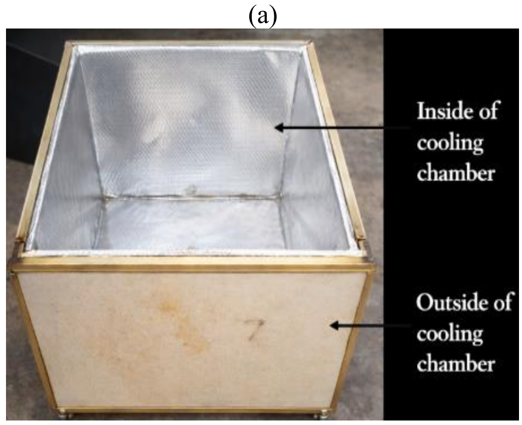


Figure 2. Pressure-enthalpy of Vapor Compression Refrigeration System (VCRS)

2.2 Cabinet construction and flow control design

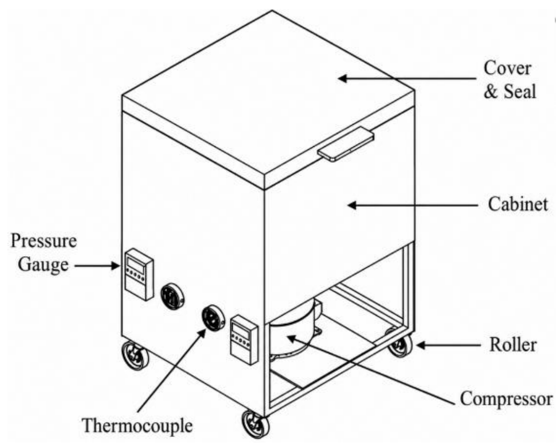
The cooling chamber was made up of a rigid insulated box measuring 480 mm by 470 mm by 380 mm internally, with dimensions of 530 mm by 500 mm by 410 mm externally. It was fabricated using mild steel, aluminium sheet metal, coated aluminium sheet metal, and styrofoam insulation material. The joints were well sealed with rivets to prevent heat influx and infiltration of air from the environment. Figure 3 shows the fabricated frame and chamber. The refrigerating capacity was calculated based on a complete cooling load analysis that took into account the wall heat gains, infiltration load, product load, and service load, allowing for some safety factor. The compressor was chosen based on its refrigeration capacity and pressure ratio. The evaporator and condenser were also selected based on their heat absorption and heat rejection capabilities, respectively [15]. The chosen expansion valve was a copper capillary tube having an internal diameter of 0.78 mm and a length of 3 meters. This setup was used during all experimental tests to ensure that the impact of different types of refrigerants and the amount of refrigerant charge could be studied without causing any further changes due to differences in the expansion valve design. The chosen length of the capillary guarantees sufficient pressure drop for reliable performance of the tested hydrocarbon refrigerants.

(a)



(b)

Figure 3. Construction of the refrigerator, (a) fabricated Frame, (b) refrigerating compartment



(a)

(b)

Figure 4. Assembled domestic refrigerator, (a) isometric view, (b) final view

All the parts have been fitted suitably by means of welding and riveting processes to the equipment. Figure 4(a) gives an isometric view of the domestic refrigerator system. Two temperature measuring devices called thermocouples have been installed on the evaporator and condenser to measure their respective temperatures at all times. Pressure measuring devices called pressure gauges have been installed at the compressor's inlet and outlets to measure suction and discharge pressures. The vacuum pump was used to expel the refrigerant out of the system. Figure 4(b) gives an illustration of the assembled system with its various components labeled.

2.3 Experimental analysis

The experimental study was performed on the designed domestic refrigerator to determine the performance of the refrigerator in terms of conventional and hydrocarbon refrigerants. A capillary tube of 3 m length was first fitted, and 90 g of the conventional refrigerant R134a was loaded in the compressor. The time taken by the refrigerator to attain thermal stability was 70 minutes. The transport properties of the refrigerants were measured through REFPROP software, and various performance parameters were determined as follows: RE, MFR, ECC, CC, COP ideal, COP actual, and SE, according to the following Eqs. (1)-(9) [16].

In order to have an equal comparison between the different refrigerants, all tests were carried out under identical boundary conditions. Prior to each test run, the refrigeration system was completely evacuated using a vacuum pump and brought back to ambient equilibrium. Then each refrigerant was added to the refrigeration system without changing anything in terms of system components except the refrigerant itself and its charge weight. The initial cabinet temperature, laboratory ambient temperature (27 ± 1 °C), circulating air through the condenser, the configuration of the cabinet, and operating conditions of the compressor were kept constant throughout the experiment program. No thermal load was placed in the cabinet during the experiment, so that the differences in performance could be attributed to the refrigerant properties and refrigerant charge only. Each test run was carried out till the steady-state condition was attained, when the variation between successive temperature and pressure readings was less than ±1%. In the charging process of the refrigerants, each was introduced in a sequence of three steps, utilizing quantities of 60 g, 40 g, and 20 g. However, exceptions were made for propylene and Dimethyl-ether, which were charged using 40 g, 30 g, and 20 g. This adjustment was necessary due to their higher density compared to the other refrigerants, as excessive amounts could potentially harm the compressor. This test was then repeated for several hydrocarbon refrigerants, including isobutane, butane, propane, propylene, and Dimethyl-ether.

After each test cycle, in which all the sensor outputs were logged, the refrigerant was evacuated using the vacuum pump, and the system was permitted to resume its original thermodynamic state before charging the next refrigerant charge. This testing method, where a constant 3 m capillary tube length with variations in both the refrigerant type and charge mass was used, allowed for a thorough evaluation of the fridge's thermodynamic and energetic performance under various operational scenarios. A 3 m capillary tube length was used because this ensured that there would be sufficient pressure drop within the refrigeration system. Alternative capillary tube lengths were not used in any experiments, as the main goal of this experiment was to observe the results of different refrigerants and their charges in constant expansion geometry. In a capillary expansion valve, tube length dictates pressure drop, refrigerant mass flux, and evaporator feed stability. Due to the low molecular weight and high latent heat of the hydrocarbon refrigerants, the use of a slightly longer capillary tube enhances refrigerant mass flow control and minimizes excess refrigerant collection in the evaporator. From experimental investigations conducted on refrigeration systems for home use that employ R134a and hydrocarbons as an alternative, optimum capillary lengths have been found to be in the range of 2.5 m to 3.5 m, when the internal diameter is between 0.6 mm and 0.9 mm [17]. At these lengths, adequate pressure losses are obtained to keep the evaporating temperature within required limits without compressor flooding.

2.4 Optimization of performance parameters of the retrofitted refrigerants

The performance of the cooling system is measured in terms of the COP. It is calculated based on the ratio of REs and compressor work. In the current experiment, the length of the capillary tube used remained constant at 3m throughout the tests. Hence, the optimization was purely done to find out the optimum charge of the refrigerant in relation to the geometry of the capillary tubes. The difference in COP obtained is an outcome of the combination between the properties of the refrigerants and their charges. Longer lengths increase the resistance and thereby lower the flow rate and REs while also increasing compressor work, which will reduce the COP value. On the other hand, increased refrigerant mass increases the flow rate and hence RE but may also increase compressor work. The optimum combination of refrigerant mass and capillary tube length provides maximum COP.

COP values were computed using experimental results of various parameters like pressures, temperatures, and enthalpy. However, the data is discontinuous; therefore, interpolation was used to compute values between two known points.

The formulation of the optimization problem is:

•Variables: Refrigerant mass and types.

•Objective: Maximization of actual COP.

•Constraints: Limits imposed by experiment on refrigerant mass and capillary tube length.

This method guarantees the selection of refrigerant charge having the greatest cooling capacity, least work done by the compressor, and maximum SE. Key characteristics of the refrigeration system, such as RE, MFR, ECC, W, COP, CC, SE, Pr, and η isentropic were computed using the conventional refrigerants and selected hydrocarbon refrigerants. Refprop software version 10.0 [18] was used to estimate the transport properties of the refrigerants, and the calculation was performed based on thermodynamic principles [19].

RE was calculated as per Eq. (1).

$$RE = h_{inlet} - h_{outlet} \tag{1}$$

where, h_{inlet} and h_{outlet} are the enthalpies at the evaporator and condenser, respectively. For the system, the evaporator temperature T_e = -10 °C, condenser temperature T_c = 40 °C, and ambient temperature T_a = 27 °C.

The mass flow rate (MFR) was calculated as:

$$m = \frac{Qe}{h_{inlet} - h_{outlet}} \tag{2}$$

Evaporator cooling capacity (ECC), compressor work (W) and actual COP were obtained using:

$$ECC = m \cdot RE \tag{3}$$

$$COP_{actual} = \frac{ECC}{W} \tag{4}$$

Compressor capacity (CC) and ideal COP were evaluated via:

$$CC = W + System\ Losses \tag{5}$$

$$COP_{ideal} = \frac{T_e}{T_c - T_e} \tag{6}$$

The SE, pressure ratio, and isentropic efficiency were calculated as:

This is calculated using Eq. (7):

$$\text{SE} = \frac{COP_{actual}}{COP_{ideal}} \times 100 \tag{7}$$

$$\text{Pr} = \frac{P_{discharge}}{P_{suction}} \tag{8}$$

$$h_{isentropic} = \frac{T_a - T_e}{T_a} \times 100 \tag{9}$$

The cooling capacity of the evaporator (CCE) used in this experiment is the calculated cooling capacity of the refrigeration system according to the measured MFR of the refrigerant and the RE. The cooling capacity was kept nearly constant since all experiments were conducted with similar settings in terms of the refrigerator cabinet, evaporator shape, compressor, and temperatures involved. The cooling capacity was therefore near the design capacity of around 0.54 kW. This means that the different performances of various refrigerants can be seen mostly in the compressor work, COP, RE, and energy consumption.

3. RESULTS AND DISCUSSION

3.1 Comparative performance of retrofitted refrigerants

The parameters of the thermodynamics and performance analysis using REFPROP are provided in Table 1. These parameters are related to the inherent thermophysical characteristics of the refrigerant regardless of the configuration of the experimental system. Hydrocarbon refrigerants produced greater RE compared to R134a, and both propane and propylene ran on a lower pressure ratio, thus requiring lower compression work and achieving better thermodynamic efficiency. The performance of the system in the 3 m capillary-tube configuration was measured and reported in Table 2. While the ECC remained nearly the same as the design value (about 0.54 kW) since the experimental system and operational conditions did not change during the experiments, there were some variations in the compressor work, actual COP, and the efficiency of the system. These variations resulted mainly from different refrigerant thermophysical characteristics and the pressure ratio of the refrigerant charge requirement.

Table 1. Thermodynamic performance parameters of refrigerants based on REFPROP

Refrigerant	Refrigerating Effect (kJ/kg)	MFR (kg/s)	Pressure Ratio	Isentropic Efficiency (%)
R134a	136.25	0.00396	5.0	71
Butane	274.26	0.00196	5.9	54
Dimethyl-ether	327.00	0.00165	4.8	93
Isobutane	244.60	0.00222	4.9	51
Propane	256.50	0.00210	3.9	70
Propylene	260.00	0.00205	3.8	91

Note: Mass Flow Rate (MFR).

Table 2. Experimental performance of refrigerants

Refrigerant	ECC (kW)	Compressor Work (kW)	Actual COP	SE (%)
R134a	0.542	0.102	4.20	79
Butane	0.537	0.137	3.90	74
Dimethyl-ether	0.539	0.079	4.70	88
Isobutane	0.538	0.147	3.70	69
Propane	0.538	0.106	4.30	81
Propylene	0.536	0.061	4.80	90

Note: Evaporator Cooling Capacity (ECC); Coefficient of Performance (COP); System Efficiency (SE).

Table 3. Thermophysical properties of selected refrigerants

Refrigerant	Molar Mass (kg/kmol)	Triple Point (°C)	Normal Boiling Point (°C)	Gas Phase Dipole (Debye)
R134a CF_3CH_2F	102.03	-103.30	-26.07	2.068
Butane $CH_3(CH_2)2CH_3$	58.12	-138.26	-0.49	0.05
Dimethyl-ether CH_3OCH_3	46.07	-141.49	-24.78	1.301
Isobutane $CH(CH_3)_3$	58.12	-159.42	-11.75	0.132
Propane $CH_3CH_2CH_3$	44.10	-187.63	-42.11	0.084
Propylene CH_2CHCH_3	42.08	-185.20	-47.62	0.366

Table 4. Critical properties of selected refrigerants

Refrigerant	Critical Temp. (°C)	Critical Pressure (MPa)	Critical Density (kg/m³)	Acentric Factor
R134a CF_3CH_2F	101.06	4.0593	511.9	0.327
Butane R600	151.98	3.796	228.0	0.201
Dimethyl-ether RE170	127.23	5.336	273.65	0.196
Isobutane R600a	134.66	3.629	225.5	0.184
Propane R290	96.74	4.251	220.48	0.152
Propylene R1270	91.06	4.555	229.63	0.146

R134a:
RE = 136.25 kJ/kg, COP_{actual} = 4.8, SE = 79%, Pr = 5.0, $\eta_{isentropic}$ = 71%

Butane:
COP_{actual} = 3.9, SE = 74%, Pr = 5.9, $\eta_{isentropic}$ = 54%

Dimethyl-ether:
COP_{actual} = 5.0, SE = 88%, Pr = 4.8, $\eta_{isentropic}$ = 93%

Isobutane:
COP_{actual} = 3.7, SE = 69%, Pr = 4.9, $\eta_{isentropic}$ = 51%

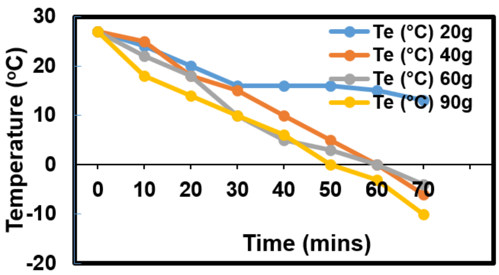
Propane:
COP_{actual} = 4.2, SE = 81%, Pr = 3.9, $\eta_{isentropic}$ = 70%

Propylene:
COP_{actual} = 4.8, SE = 90%, Pr = 3.8, $\eta_{isentropic}$ = 91%

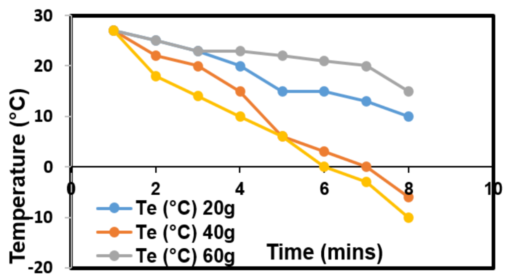
The variations found in COP and SE are strongly related to the difference in pressure ratio and RE of the refrigerants. It is evident from the results that R134a showed the highest value of pressure ratio (Pr = 5.0), which means a large amount of work had to be done in compressing it between the evaporator and the condenser. A higher pressure ratio means more work by the compressor and hence less efficient operation. On the other hand, the pressure ratios of propane (Pr = 3.9) and propylene (Pr = 3.8) were quite low. This is because of the higher latent heat of vaporisation and favourable thermodynamic characteristics that propylene and Dimethyl-ether performed better than other refrigerants. These refrigerants not only produced higher REs but were able to do so with low MFR. This means that per unit mass of refrigerant, more cooling was achieved, which contributed to a better COP value. In terms of thermodynamics, as observed in Table 3, low pressure ratios usually translate to low temperature levels and low entropy production during compression. Consequently, the isentropic efficiencies are increased, and there are fewer energy losses in the compressor. Consequently, it is understandable that the isentropic efficiencies of propylene (91%) and Dimethyl-ether (93%) have been observed to be higher than those of R134a. The critical properties of some refrigerants are presented in Table 4.

3.2 Pull down time of retrofitted refrigerants

Figure 5(a-e) depicts the pull-down time of the chosen hydrocarbon refrigerants compared to 90 grams of R134a using a 3-meter-long capillary tube. Pull-down time refers to the time taken for the cooling system to reach the lowest evaporator temperature, which is a good representation of the cooling capacity of the refrigerant. Less pull-down time implies that the cooling rate is quick and indicates the thermodynamic compatibility of the refrigerant with the compressor and expansion valve combination [20].



(a)



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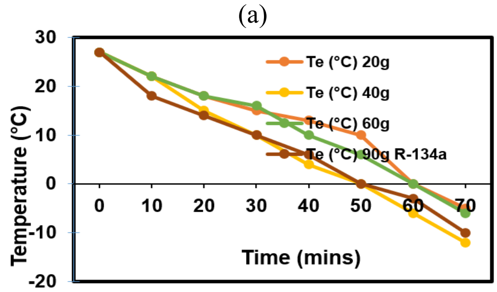

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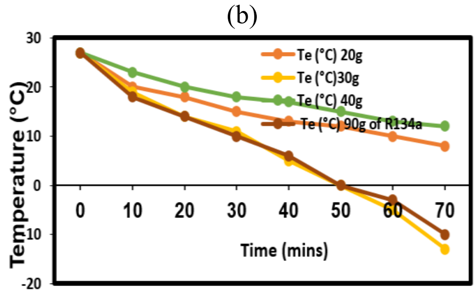

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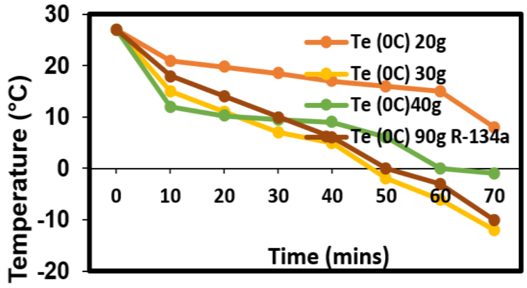

(e)

Figure 5. Pull-down time for different mass charges of various refrigerants compared with 90 g of R134a, (a) Isobutane, (b) butane, (c) propane, (d) propylene, (e) Dimethyl-ether

In the case of Isobutane presented in Figure 5(a), charges of 40 g and 60 g were able to meet the required evaporator temperature in around 70 minutes, similar to R134a, but the charge of 20 g failed to meet the minimum temperature requirement due to insufficient mass flow and refrigeration efficiency. It indicates that hydrocarbons would require a lesser amount of charge compared to R134a, but below a certain limit, performance will decrease due to insufficient mass flow through the evaporator.

In the case of butane presented in Figure 5(b), charges of 40 g were able to attain the minimum temperature in the test period. Over and under charging caused slower cooling because there was a direct effect of MFR and compressor work on the transient response. The cooling response time of R134a was slightly quicker under the given conditions. Propane in the test presented in Figure 5(c) showed steady performance regardless of the quantity of mass used in each run, reaching the minimum temperature in less than 70 minutes.

The case of propylene (Figure 5(d)) shows that the charge of 30 g had a faster pull-down capability compared to that of R134a as it recorded a lower evaporator temperature in the same operational period. Its excellent transient response implies better heat transfer properties and pressure ratio characteristics. For Dimethyl-ether (Figure 5(e)), there was a similar pattern. The charge of 30 g recorded the minimum temperature after 70 minutes, even when compared to 90 grams of R134a. The fast cooling ability is attributed to its higher RE and effective utilization of the evaporator at the tested conditions. Generally, the two refrigerants (propylene and Dimethyl-ether) had a shorter pull-down time and more stable cooling trends compared to propane. From the findings, it can be noted that the chosen hydrocarbon refrigerants were able to achieve the same or even better pull down efficiency when compared to R134a refrigerant at their optimal loading ratios. Pull-down rate is also influenced by differences in evaporating temperature and refrigerant mass flow characteristics. For example, the pull down rate for refrigerants like propylene and Dimethyl-ether was faster due to their thermodynamic properties that made them able to absorb heat efficiently from the evaporator. The quicker reduction in evaporating temperature increased the difference in temperature between the refrigerated space and the evaporating surfaces, hence increasing heat transfer. The charge level had an effect on the amount of superheat that occurred within the evaporator. This is because, with low charge levels, the amount of refrigerant reaching the evaporator was too little, hence there were high levels of superheat and decreased cooling capacity. On the other hand, higher charges led to increased amounts of condensation and loading in the condenser, which raised the condenser pressure and compressor load.

3.3 Experimental performance of varied mass charges

The experimental characteristics of various masses of selected hydrocarbon refrigerants were analyzed using a fixed length of 3 m of the capillary tube as compared to 90 grams of R134a (Tables 5-9). In the analysis, it is noted how the refrigerant charge affects refrigeration, MFR, work of the compressor, actual COP, and system effectiveness. In case of iso butane illustrated in Table 5, a charge of 40 grams gave the best performance as reflected by the highest refrigerating effectiveness (367.6 kJ/kg), actual COP (6.7), and efficiency (98 percent), outperforming R134a in all the mentioned parameters. An increase in the charge to 60 grams resulted in some reduction in efficiency, while a charge of 20 grams gave lower COP and SE as a result of poor feeding into the evaporator. This proves the existence of an optimum mass charge where MFR and compressor work are well matched. It is possible to explain the presence of an optimum refrigerant charge by looking at the effects of superheating and subcooling combined. When undercharged, there is too much superheat since there is not enough liquid refrigerant present to evaporate fully during its path through the evaporator. Conversely, overcharging leads to high subcooling and pressure inside the condenser, leading to higher compressor load and discharge pressure. When at the optimum charging condition, the evaporation process is maximized while the condenser pressure is kept at acceptable levels. It means that compression power would be minimized while the refrigeration effect would still be high enough, thus achieving better COP. Propylene at 20 g and Dimethyl-ether at 30g performed better because they managed to create a good balance between the two mentioned processes in comparison with R134a and other hydrocarbon refrigerants.

In the case of butane, as shown in Table 6, a 40 grams of mass charge provided maximum COP (4.8) and efficiency (84%), just outperforming R134a. Overcharging and undercharging affected the efficiencies of both 20 g and 60 g charges negatively because the SE and compressor loading were disturbed. Propane gave good performance in all tests as evident from the results shown in Table 7. The RE was highest in case of the 60 g charge. However, the actual COP was the highest in case of the 20 g charge (6.17). All tested masses provided COPs equal or higher than R134a, suggesting that propane is thermodynamically suitable for expansion geometry under the present operating conditions, although optimum performance depended on refrigerant charge. Propylene showed better performance as the amount of charge was decreased, as shown in Table 8. The highest COP (8.7) and highest SE (94.5 percent) were provided by a 20 g charge. In comparison, propylene showed higher COP and lower work done by the compressor than R134a under the existing experimental conditions. At 30 g and 40g charge amounts, propylene offered much higher efficiencies than the traditional refrigerant, thus showing its good compatibility with the expansion geometry.

In addition, Dimethyl-ether proved favourable performance in all cases, as evident from Table 9. With a 30 g charge, the highest SE (93%) and actual COP (6.1) were provided. Dimethyl-ether provided higher COP and SE than R134a in all tests. For all the refrigerants used at a given capillary length, maximum efficiency was attained at medium charge mass instead of the highest mass charge. Propylene and Dimethyl-ether had lower charge masses compared to R134a yet offered higher cooling capacities, higher COPs, and increased system efficiencies. The experimental results show that some hydrocarbon refrigerants have the potential to attain greater efficiency in terms of energy compared to R134a when running at optimum charge levels under the constant 3 m capillary tube setup. Performance has been shown to depend highly on the type of refrigerant used as well as its optimum level of charge.

Table 5. Performance of different mass charges of isobutane compared with 90 g of R134a

Mass (g)	RE (kJ/kg)	MFR (kg/s)	ECC (kW)	CC (kW)	COP Ideal	COP Actual	SE (%)
60	265.0	0.0020	0.530	0.10	6.9	5.3	76
40	367.6	0.0015	0.54	0.084	6.8	6.7	98
20	140.2	0.0038	0.5327	0.19	4.2	2.8	66
90 (R134a)	142.1	0.0038	0.5399	0.144	5.7	4.2	82

Note: Refrigerating Effect (RE); Mass Flow Rate (MFR); Evaporator Cooling Capacity (ECC); Compressor capacity (CC); Coefficient of Performance (COP); System Efficiency (SE).

Table 6. Performance of different mass charges of butane compared with 90 g of R134a

Mass (g)	RE (kJ/kg)	MFR (kg/s)	ECC (kW)	CC (kW)	COP Ideal	COP Actual	SE (%)
60	43.93	0.01229	0.539	0.2078	5.3	2.5	47
40	250.00	0.00216	0.540	0.1128	5.7	4.8	84
20	43.00	0.01255	0.540	0.1230	5.8	4.4	75
90 (R134a)	142.18	0.0038	0.5399	0.144	5.7	4.2	82

Note: Refrigerating Effect (RE); Mass Flow Rate (MFR); Evaporator Cooling Capacity (ECC); Compressor capacity (CC); Coefficient of Performance (COP); System Efficiency (SE).

Table 7. Performance of different mass charges of propane compared with 90 g of R134a

Mass (g)	RE (kJ/kg)	MFR (kg/s)	ECC (kW)	CC (kW)	COP Ideal	COP Actual	SE (%)
60	281.8	0.00191	0.5382	0.0944	6.8	5.7	83.0
40	279.2	0.00193	0.5370	0.1073	6.0	5.0	83.0
20	263.0	0.00205	0.5359	0.0868	7.0	6.17	73.9
90 (R134a)	142.1	0.00389	0.5399	0.144	5.7	4.2	82

Note: Refrigerating Effect (RE); Mass Flow Rate (MFR); Evaporator Cooling Capacity (ECC); Compressor capacity (CC); Coefficient of Performance (COP); System Efficiency (SE).

Table 8. Performance of different mass charges of propylene compared with 90 g of R134a

Mass (g)	RE (kJ/kg)	MFR (kg/s)	ECC (kW)	CC (kW)	COP Ideal	COP Actual	SE (%)
40	250.0	0.00216	0.54	0.086	6.8	6.2	91.0
30	279.2	0.00193	0.539	0.0891	6.4	6.0	94.0
20	263.0	0.00205	0.5359	0.0615	9.2	8.7	94.5
90 (R134a)	142.1	0.00389	0.5399	0.144	5.7	4.2	82

Note: Refrigerating Effect (RE); Mass Flow Rate (MFR); Evaporator Cooling Capacity (ECC); Compressor capacity (CC); Coefficient of Performance (COP); System Efficiency (SE).

Table 9. Performance of different mass charges of Dimethyl-ether compared with 90 g of R134a

Mass (g)	RE (kJ/kg)	MFR (kg/s)	ECC (kW)	CC (kW)	COP Ideal	COP Actual	SE (%)
40	250.3	0.00215	0.540	0.086	6.8	6.2	91
30	260.2	0.00207	0.539	0.087	6.5	6.1	93
20	220.6	0.00245	0.5392	0.0984	6.0	5.5	91
90 (R134a)	142.1	0.00389	0.5399	0.144	5.7	4.2	82

Note: Refrigerating Effect (RE); Mass Flow Rate (MFR); Evaporator Cooling Capacity (ECC); Compressor capacity (CC); Coefficient of Performance (COP); System Efficiency (SE).

3.4 Compressor work and energy consumption test

Tables 10-14 illustrate the compressor work, total energy usage and percentage energy savings of the selected hydrocarbons when compared against 90 g of R134a. It can be useful to note how the effect of refrigerant charge plays an important role in power usage and energy conservation for such systems. It can be noted from the result of Isobutane as seen in Table 10 that there was a decrease in energy consumption as the refrigerant charge decreased from 0.06 kg to 0.02 kg. There is 61% decrease in the energy usage which was 0.53 kJ at its lowest value compared to R134a. However, even though the compressor work decreases uniformly, the energy demand of the system was relatively low compared to conventional refrigerant.

The butane energy consumption of 0.10 kJ as illustrated in Table 11, showed that the refrigerant charge of 0.04 kg used more energy than other hydrocarbon charge but yet less than the energy demand by R134a. While, the energy consumption of 0.02 kg charge shows the least energy usage of 0.10 kJ and highest percent energy reduction of 95%. Propane showed the increased energy demand as charge mass increase as seen in Table 12. The 0.06 kg charge used 1.59 kJ, and the 0.02 kg charge consumed 0.45 kJ, marking a 75 percent reduction as compared to R134a. This result clearly shows that energy consumption is dependent on the refrigerant charge for a given expansion geometry [21]. The reduction in energy usage seen at reduced optimum charging conditions can be attributed to the reduced compressor work. Less refrigerant in the system lowers the suction density and MFR, which in turn means less work is needed to compress the refrigerant. But this decrease is advantageous only up to a point when there is enough refrigerant in the system to ensure efficient heat transfer in the evaporator. Thus, the values presented were derived from calculations based on measured data from the refrigerating system. These calculations adhered to relevant codes and utilized standard formulas and transport properties from REFPROP, which indicates the states of refrigerant at each state point throughout the experimentation of the refrigeration system.

For propylene, the amount of energy consumption decreases with decreasing refrigerant charges as shown in Table 13. For a refrigerant charge of 0.02 kg, energy consumption becomes 0.32 kJ, thus recording 83 percent energy consumption reduction compared to R134a. As can be seen from this result, less energy consumption results when there is optimum compressor load. For Dimethyl-ether, the amount of energy consumption decreases from 0.86 kJ at 0.04 kg to 0.43 kJ at 0.02 kg, showing a maximum 77 percent energy reduction as compared to R134a. Energy consumption is calculated over the total test duration. The declining trend indicates that energy consumption depends on compressor loading which in turn is dependent on refrigerant charge. From the results, it can be said that hydrocarbon refrigerants require less energy consumption as compared to R134a. Of the tested refrigerants, propylene and Dimethyl-ether demonstrated the greatest reduction in energy usage, with low charges needed for both, further establishing their feasibility as effective and environmentally friendly substitutes to R134a in household refrigerators.

Table 10. Energy consumption of varied masses of isobutane and percentage reductions when compared with 90 g of R134a

Mass (kg)	Compressor Work (kJ/kg)	EC (kJ)	Percentage Reduction (%)
0.06	26.5	1.59	12
0.04	30.8	1.23	32
0.02	26.6	0.53	61
0.09 of R134a	20.4	1.80	0

Note: Energy Consumption (EC).

Table 11. Energy consumption of varied masses of butane and percentage reductions when compared with 90 g of R134a

Mass (kg)	Compressor Work (kJ/kg)	EC (kJ)	Percentage Reduction (%)
0.06	8.9	0.53	61
0.04	28.2	1.12	38
0.02	5.2	0.10	95
0.09 of R134a	20.4	1.80	0

Note: Energy Consumption (EC).

Table 12. Energy consumption of varied masses of propane and percentage reductions when compared with 90 g of R134a

Mass (kg)	Compressor Work (kJ/kg)	EC (kJ)	Percentage Reduction (%)
0.06	26.6	1.59	12
0.04	29.8	1.19	32
0.02	22.8	0.45	75
0.09 of R134a	20.4	1.80	0

Note: Energy Consumption (EC).

Table 13. Energy consumption and percentage reduction for different propylene mass charges compared with 90 g of R134a

Mass (kg)	Compressor Work (kJ/kg)	EC (kJ)	Percentage Reduction (%)
0.04	21.5	0.86	53
0.03	24.8	0.74	59
0.02	16.1	0.32	83
0.09 of R134a	20.4	1.80	0

Note: Energy Consumption (EC).

Table 14. Energy consumption of varied masses of Dimethyl-ether" and percentage reductions when compared with 90 g of R134a

Mass (kg)	Compressor Work (kJ/kg)	EC (kJ)	Percentage Reduction (%)
0.04	21.5	0.86	53
0.03	22.6	0.68	63
0.02	21.7	0.43	77
0.09 of R134a	20.4	1.80	0

Note: Energy Consumption (EC).

3.5 Optimization results at constant capillary tube length

The optimization was performed in MATLAB, where the refrigerant mass charge is obtained such that the maximum COP can be achieved. In this case, the expansion geometry is kept constant, meaning that the variation in performance is purely due to the thermodynamic properties of the refrigerant itself. Figure 6(a) shows the optimized COP values of the chosen refrigerants. Propylene has the maximum optimized COP value of 8.8, showing excellent thermodynamic efficiency and thus great potential as an alternative to R134a. The COP values of isobutane, Dimethyl-ether, and propane are 6.7, 6.2, and 6.1, respectively. Butane and R134a have COP values of 5.8 and 4.8, respectively. All of the chosen hydrocarbon refrigerants possessed more optimized COPs than R134a in the optimization approach used in this study. However, the magnitude of difference was not similar for all refrigerants, and the results were achieved at different optimum refrigerant masses [22]. Propylene and propane each recorded approximately 260 kJ/kg, while Dimethyl-ether achieved 250 kJ/kg. Butane showed the lowest RE among the hydrocarbons Figure 6(b). Optimized compressor work is presented in Figure 6(c). R134a required the highest compressor work (0.14 kW), followed by butane (0.098 kW). Dimethyl-ether and propane each required 0.088 kW, isobutane required 0.086 kW, and propylene exhibited the lowest compression work. Because energy consumption is directly proportional to compressor work, propylene and isobutane offer the most energy efficient performance. The isobutane had the highest figure of 340 kJ/kg, which shows its ability to absorb heat effectively on a unit mass basis. The propylene and propane both registered a figure of about 260 kJ/kg, while the Dimethyl-ether managed 250 kJ/kg. Butane had the lowest refrigerating capacity among all the hydrocarbons tested.

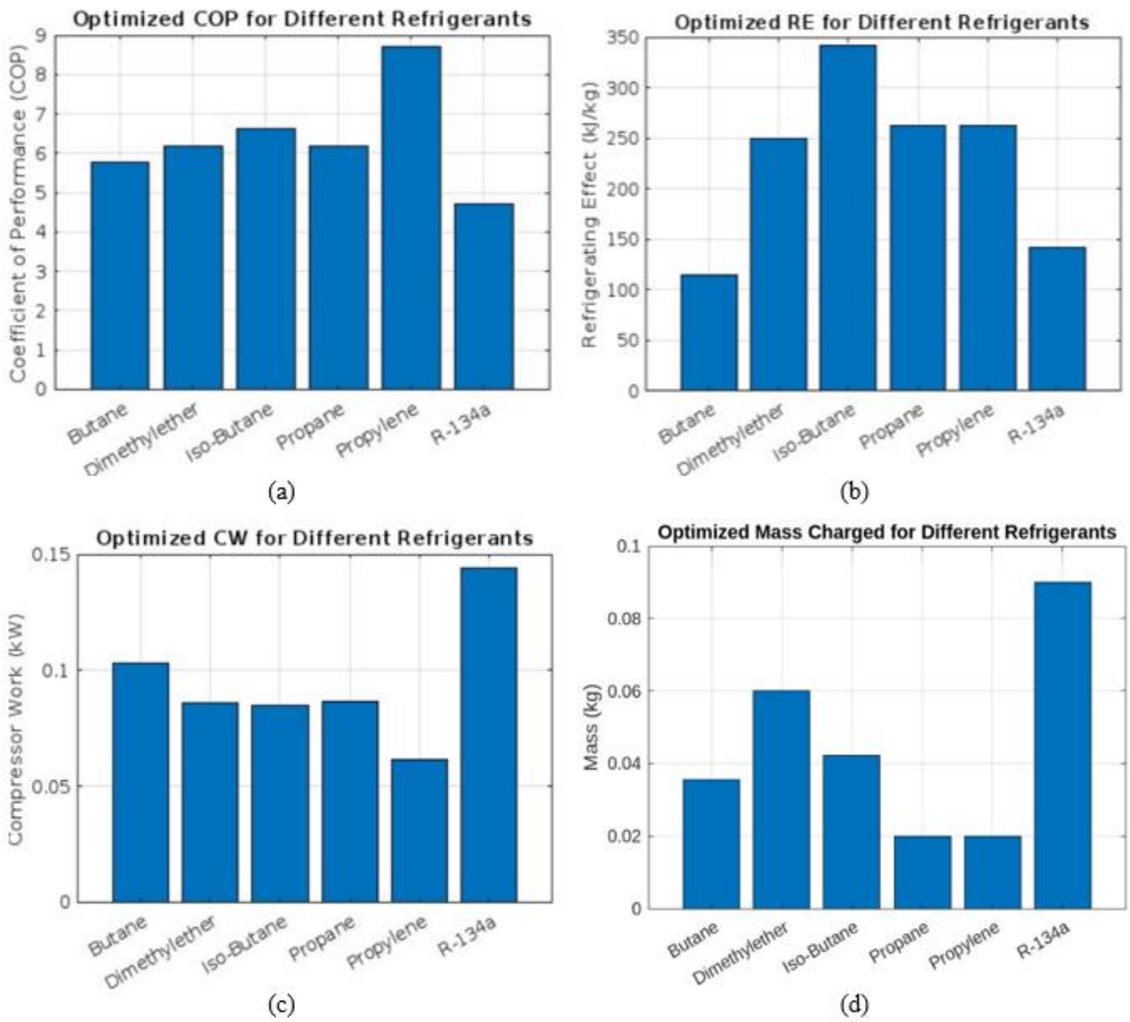

Figure 6. Optimized results: (a) coefficient of performance (COP), (b) refrigerating effect (RE), (c) compressor work, (d) mass charged for hydrocarbon refrigerants and R134a

The outcomes of optimization additionally illustrate the thermodynamic connection among pressure ratio, work input by the compressor, and COP. Propylene produced the maximum COP value since it had a comparatively high RE while having the minimum compressor work compared to all other test refrigerants. Propylene possessed a low pressure ratio, thus reducing the amount of energy spent on compressing the refrigerant for circulating it around the circuit. At the same time, propylene had high latent heat, hence enabling efficient cooling with low refrigerant charge. Likewise, Dimethyl-ether had the best COP performance owing to its high RE and isentropic efficiency. On the contrary, R134a needed higher refrigerant charge and compressor work to reach comparable cooling capability. It proves that the improvements in performance depend not only on the type of refrigerant used but also on the interaction between the refrigerant properties and charge optimization. Optimal mass charge per refrigerant is illustrated in Figure 6(d). The optimal mass charge for R134a was found to be 0.09 kg, 0.06 kg for Dimethyl-ether, and the least optimal mass charges were needed for propane and propylene. This confirms that hydrocarbons outperform fluorocarbons in terms of energy efficiency, even when minimal refrigerant charge is used. Thus, the optimization process results were based on interpolation-based estimates. These estimates were validated under optimal charge conditions and considered the generated data.

4. CONCLUSION

This study sought to analyse the performance of selected hydrocarbon refrigerants in a retrofitting exercise into a domestic vapor-compression refrigerator running with a constant 3 m capillary tube. The results reveal that refrigerant performance was determined by the nature of the refrigerant as well as its charge. Of all the refrigerants studied, the best optimized COP (COP = 8.8) was observed for the case of propylene. Other hydrocarbon refrigerants that showed good thermodynamic performance were isobutane, Dimethyl-ether, and propane. However, the performance dropped drastically in the cases where the refrigerant charge was not the optimal one.

These findings have application to the operating conditions studied in this research, which include a domestic refrigerator that works with a constant 3-meter-long capillary tube in a controlled ambient condition and without any load in the cabinet. Further studies under different ambient temperatures and cabinet loads could shed further light on the practical performance of the system. Overall, the research proves that propylene, isobutane, Dimethyl-ether, and propane can be used as effective alternatives to R134a for household refrigerators and may thus serve as replacements for hydrofluorocarbons.

ACKNOWLEDGMENT

The authors acknowledge the Department of Mechanical Engineering, Federal University of Oye Ekiti, Nigeria, and the Faculty of Engineering and Quantity Surveying, INTI International University, Malaysia, where the experiments were carried out.

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