A COMPARATIVE STUDY ON THE PERFORMANCE OF THE DOMESTIC REFRIGERATOR USING R600A AND LPG WITH VARYING REFRIGERANT CHARGE AND CAPILLARY TUBE LENGTH

S.O. Oyedepo,[1,*] R.O. Fagbenle,[2] T.O. Babarinde,[1] K.M. Odunfa,[2] R.O. Leramo,[1] O.S. Ohunakin,[1] O.O. Ajayi,[1] P.O. Babalola,[1] O. Kilanko,[1] A.D. Oyegbile,[1] & D. Lawson-Jack[1]

[1]*Mechanical Engineering Department, Covenant University, Ota, Nigeria*

[2]*Mechanical Engineering Department, University of Ibadan, Nigeria*

*Address all correspondence to: S.O. Oyedepo, Mechanical Engineering Department, Covenant University, Ota, Nigeria, E-mail: Sunday.oyedepo@covenantuniversity.edu.ng

A comparative experimental study on the performance of a domestic refrigerator using R600A and LPG with a varying refrigerant charge (w_r) and capillary tube length (L) was carried out. The enthalpy of the refrigerants R600A and LPG for each data set for the experimental conditions were obtained by using REFPROP software (version 9.0). The results show that the design temperature and pull-down time set by ISO for a small refrigerator are achieved earlier using refrigerant charge 60 g of LPG with a 1.5 m capillary tube length. The highest COP (4.8) was obtained using 60-g charge of LPG with L of 1.5-m. The average COP obtained using LPG was 1.14% higher than that of R600A. Based on the result of electric power consumption, R600A offered lowest power consumption. The compressor consumed 20% less power compared to LPG in the system. The system performed best with LPG in terms of COP and cooling capacity, while in terms of power consumption R600A performed best.

KEY WORDS: *cooling capacity, refrigeration system, COP, refrigerant charge, capillary tube length, power consumption*

1. INTRODUCTION

Refrigeration plays a significant role in domestic, industrial, commercial, and health sectors for comfort, food storage, medical applications, etc. There are innumerable applications of such systems, and they are the major consumer of electricity around the world (Anand and Tyagi, 2012; Rasti et al., 2012). The high energy consumption by refrigeration systems is due to the large amounts of units being used and also to

NOMENCLATURE

COP	coefficient of performance	w_r	refrigerant charge
h	specific enthalpy of refrigerant, kJ/kg	**Subscripts**	
L	capillary tube length	1	evaporator inlet
LPG	Liquidified Petroleum Gas	2	evaporator outlet
$\dot{m}$	refrigerant mass flow rate, kg/s	3	compressor outlet
P	pressure	c	compressor
PR	compressor pressure ratio	dis	discharge
Q_{ev}	cooling capacity, kW	r	refrigerant
W_c	compressor work, kW	suc	suction

their low thermodynamic efficiencies (Boeng and Melo, 2012). Due to the increasing demand of energy primarily for refrigeration and air conditioning applications (around 26–30%), this leads to degradation of environment, global warming, and depletion of the ozone layer, etc. To overcome these aspects, there is an urgent need in efficient energy utilization besides waste heat recovery for useful applications especially after the Kyoto and Montreal protocols. In addition to these, better designs of cooling systems are required to minimize the energy consumption (Alzoubi and Zhang, 2015).

The energy consumption by household refrigerators depends on its components, the refrigerant charge, and ambient conditions. It is well known that domestic refrigerators have the highest efficiency when operating with certain combinations of capillary tube and refrigerant charge (Gonçalves and Melo, 2004; Naer et al., 2001). Capillary tubes are used as expansion devices in low-capacity refrigeration machines such as domestic refrigerators, freezers, and window-type air conditioners. Usually, they have inner diameter (d) ranging from 0.5 mm to 2 mm and length (L) from 2 m to 6 m. Compared to other expansion devices, the capillary tubes are simple and cheap, as well as cause the compressor to start at a low torque as the pressure across the capillary tube equalizes during the off-cycle (Boeng and Melo, 2012). In order to enhance the system cooling capacity, the capillary tube and the suction line are usually placed together forming a counterflow heat exchanger (Dincer and Kanoglu, 2010). The heat exchanger may be of lateral or concentric type (Park et al., 2007). The flow inside the capillary tube is complex, and pressure drop through the capillary tube has a strong influence on the performance of the whole system.

Several authors have conducted experimental and numerical studies on the flow characteristics of a refrigerant in a capillary tube and the effects of the capillary tube dimensions and geometry on the performance of a vapor compression refrigeration system (Fiorelli and Silvares, 2004; Guobing and Yufeng, 2006; Salim, 2012; Matani and Agrawal, 2013; Pathak et al., 2014).

The literature reveals that most of the previous studies have focused on the independent variation of the refrigerant charge w_r or capillary tube geometries (L or d), while study of the effect of simultaneous variation of these parameters is still lacking. Accordingly, in the present study, the thermodynamic performance of a household refrigerator was experimentally studied by simultaneously varying w_r and L. The comparative study on the performance of a domestic refrigerator using LPG and R600A was also carried out.

The concerns over the global warming and ozone layer depletion have mandated replacing chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants with alternative (hydrocarbon) refrigerants in domestic refrigerators (UNEP, 2012). The efforts to explore eco-friendly alternative to R12 and R22 have brought out liquefied petroleum gas (LPG) and R600A (isobutane) as the best promising substitutes for R12 and R22 in the refrigeration system.

LPG is a mixture of commercial butane and commercial propane having both saturated and unsaturated hydrocarbons. At atmospheric pressure and temperature, it is a gas which is 1.5 to 2.0 times heavier than air. It is readily liquefied under moderate pressures. LPG emits less carbon per joule than butane but more carbon per joule than propane (Oyelami and Bolaji, 2015). The ozone depletion potential (ODP) of LPG is 0 and global warming potential (GWP) is 8 which is significantly negligible as compared to other refrigerants and exhibits properties similar to those of R12 (Punia and Singh, 2015). LPG does not form acids, hence eliminates the problem with blocked capillaries. From the environmental impacts point of view, R600A (isobutane) has low values (< 20) of GWP and null ODP (Copetti and Macagnan, 2005). Concerning the security, the hydrocarbon is flammable, with very low ignition concentration limits (lower and upper limits in the range of 1.5–2.1% and 8.5–11.4% per volume, and ignition temperatures in the range of $365^{o}C$ to $491^{o}C$). These make its use difficult in some refrigeration systems, and it is only recommended in systems with reduced cooling load or in cooling systems that use secondary refrigerants with efficiency losses.

The research efforts and development in the refrigeration and air conditioning sector applied to the use of hydrocarbon refrigerants is not associated only with the need to preserve the environment alone, but has great importance with regard to the latent need for energy efficient equipment. With this perspective, the present study deals with the thermodynamic evaluation of the use of hydrocarbon refrigerants (LPG and R600A) in a domestic refrigeration system. There are only few studies reported in the literature on the study of flow of LPG and R600A through a capillary tube. Therefore, in order to strengthen the work of LPG and R600A as refrigerants, the present study has been carried out to investigate the performance of a domestic refrigerator by simultaneously varying the refrigerant charges and capillary tube length using LPG and R600A as refrigerants.

2. MATERIALS AND METHODS

2.1 Thermodynamic Processes

Figure 1 shows the schematic diagram of a single-stage vapor compression refrigeration system.

The experimental refrigerator consists of a hermetically sealed compressor, wire mesh air-cooled condenser, a filter drier, a capillary tube, and an evaporator. A thermodynamic process in refrigeration systems takes place in the evaporator, by which heat is transferred to the refrigerant causing it to evaporate in the evaporator. The refrigerant vapor leaves the evaporator as a superheated refrigerant. It then passes through the heat exchanger suction line, gaining more heat before entering the compressor. In the compressor, it is compressed to a higher condensing pressure. Passing through the condenser at a temperature higher than that of the surroundings, heat is then rejected to the surroundings. At this time, the state of the refrigerant is subcooled in the condenser. It exchanges heat with the vapor leaving the evaporator in the suction line of the heat exchanger, and it is cooled even further. Finally, the refrigerant is expanded in an adiabatic process, causing a sharp drop in its temperature. It enters the evaporator as a saturated liquid at low temperature and pressure, and then the cycle is completed (Bilal and Salem, 2003).

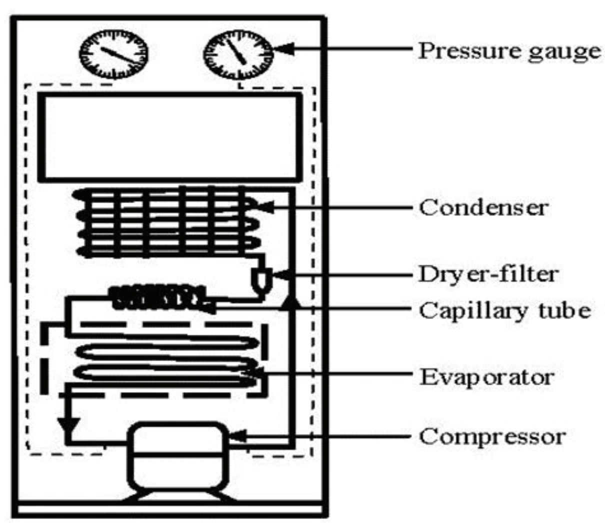

FIG. 1: Schematic diagram of experimental refrigerator

2.2 Experiment and Analysis

This study is an extension of work by Oyedepo et al. (2016). Hence, the experimental procedure in this study is similar to that of Oyedepo et al. (2016). The experimental setup (Fig. 2) consisted of a domestic VCRS of 1 ton of refrigerant (TR) capacity designed to work with R12, an evaporator of 79-L capacity, wire mesh air-cooled condenser, and a reciprocating compressor. The refrigerator was instrumented with two pressure gauges at the inlet and outlet of the compressor for measuring the suction and discharge pressure, and a power meter (with 0.01 kWh accuracy) for measuring the energy consumption. The test rig was thoroughly checked and commissioned before it was subjected to a series of tests under various conditions. The specifications of the domestic refrigerator used in this study are shown in Table 1. Experiments were

FIG. 2: Experimental setup

TABLE 1: Specifications of the base line test unit

Item	Specification
Unit type	Freezer
Internal volume	69 L
Refrigerant/Lubricant	R12/Mineral oil
Compressor	Reciprocating compressor
Evaporator Diameter	Cross flow fin and heat exchanger 6.4 mm
Condenser Diameter	Natural cooling hot plate type heat exchanger 6.4 mm
Expansion device Diameter	Capillary tube 0.8 mm

conducted with LPG and R600A, by varying the *refrigerant charge* from 40 g to 60 g and L as 0.9 m, 1.2 m, and 1.5 m, with a dry bulb temperature of 32°C. The temperature (from –30°C to +90°C), pressure (from 100 to 1300 kPa), and compressor power (from 0 to 1100 W) were measured with an uncertainty of ± 0.1%.

The refrigerants were charged into the system by a digital charging system. Type K thermocouples were used to measure the temperature at inlet and outlet of the evaporator and outlet of the compressor. A temperature gauge was used for measuring the evaporator air temperature in order to obtain the pull-down time (the time required for changing the evaporator chamber air temperature from the ambient temperature to the desired final temperature). Readings were taken five times for each value of w_r with an accuracy of ± 0.05. The experiment was carried out at an average ambient temperature of 32°C in no load and closed door conditions. The REFPROP version 9.0 software was used to determine the enthalpy h of the refrigerant by using the temperatures from the readings as input data. The results were used to calculate the cooling capacity Q_{ev}, compressor pressure ratio (PR), the isentropic compressor work W_c, power per ton of refrigerant, and the COP of the refrigerator, as defined in the following fundamental equations

The cooling capacity Q_{ev} is given by

$$Q_{ev} = \dot{m}_r (h_2 - h_1)\ \mathrm{kW}\,. \tag{1}$$

The isentropic compression work in the compressor can be expressed as

$$W_c = \dot{m}_r (h_3 - h_2)\ \mathrm{kW}\,. \tag{2}$$

The refrigerant mass flow $\dot{m}_r$ can be estimated using the following equation (Almeida et al., 2010):

$$m_r = \frac{Q_{ev}}{q_{ev}}\,. \tag{3}$$

The power per ton of refrigerant (PPTR) is obtained by

$$\mathrm{PPTR} = 3.5\,\frac{W_c}{Q_{ev}}\,. \tag{4}$$

The pressure ratio (PR) is defined by

$$\mathrm{PR} = \frac{P_{dis}}{P_{suc}}\,, \tag{5}$$

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

$$\mathrm{COP} = \frac{Q_{ev}}{W_c}. \quad (6)$$

In the equations given above $\dot{m}_r$ is the refrigerant mass flow rate (kg/s), h_1, h_2, and h_3 are the specific enthalpies of the refrigerant (kJ/kg) at the evaporator inlet, evaporator outlet (compressor inlet), and compressor outlet, respectively, P_{suc} and P_{dis} are the compressor suction and discharge pressures (kPa), respectively, Q_{ev} is the cooling capacity (kW), and q_{ev} is the specific cooling effect (kJ/kg).

3. RESULTS AND DISCUSSION

The results of performance parameters of refrigeration system with varying refrigerants charges and capillary tube length using LPG and R600A are presented below.

3.1 Effect of Capillary Tube Length on the COP of the Refrigeration System

Figures 3 and 4 show the effect of L on the system COP with 40-g and 60-g charges, respectively. It can be seen that the COP increases with increase in L for all values of w_r. The highest COP of 4.8 was obtained with LPG at w_r = 60 g and L = 1.5 m, while for R600A the highest COP of 4.76 was obtained at w_r = 60 g and L = 1.5 m. Based on the results of this study, the average COPs of the two refrigerants are very close, but the average COP obtained using LPG is about 1.14% higher than that of R600A.

3.2 Effect of Capillary Tube Length on Power per Ton of Refrigerant

Instantaneous power consumption is the main criterion to choose a right quantity of mass charge. Figures 5 and 6 show variation of the electric power per ton of

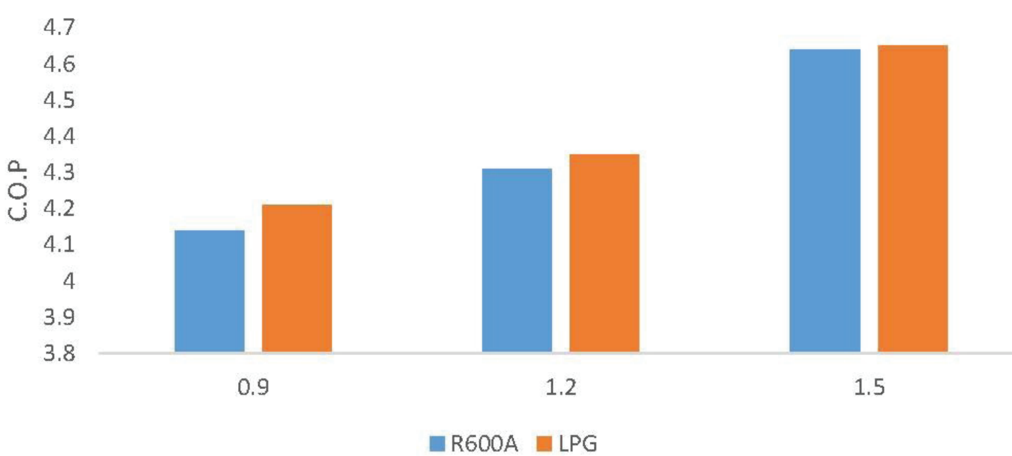

FIG. 3: Effect of capillary tube length on the COP of the system at a refrigerant charge of 40 g

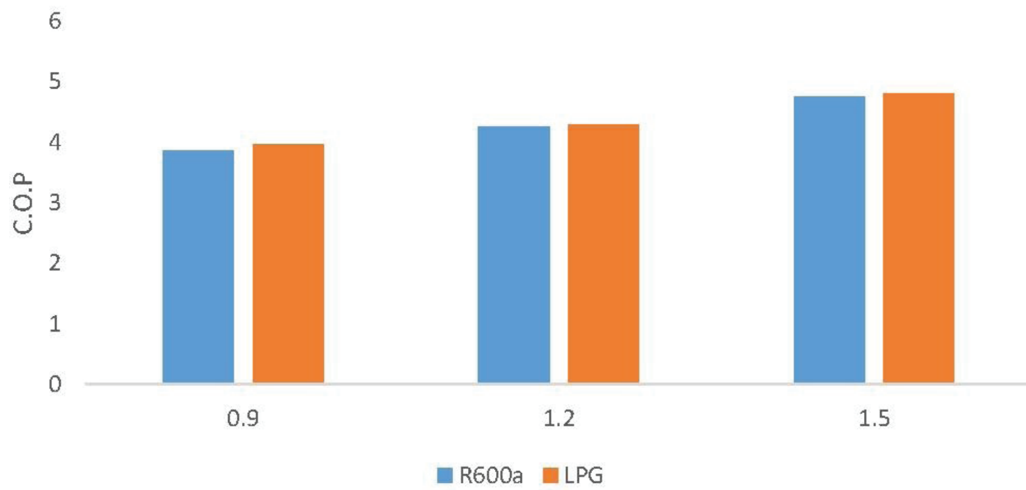

FIG. 4: Effect of capillary tube length on the COP of the system at a refrigerant charge of 60 g

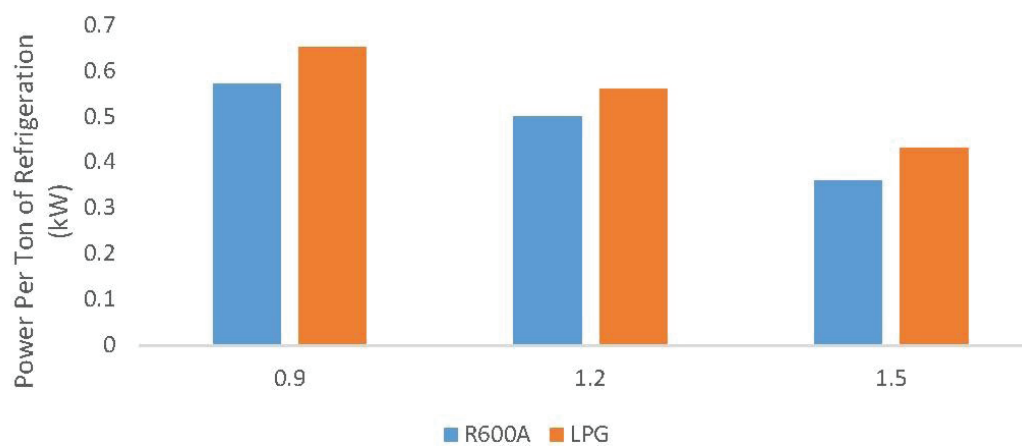

FIG. 5: Effect of capillary tube on power per ton of refrigerant at a refrigerant charge of 40 g

refrigerant (PPTR) with L and w_r. It is observed that PPTR decreases with increase in L, but increases with increase in w_r. This is mainly due to the increase in mass flow rate of the refrigerant through the compressor. The lowest PPTR of 0.36 kW was recorded at w_r = 40 g for R600A with L = 1.5 m, while the lowest PPTR of 0. 43 kW was recorded at w_r = 40 g with L = 1.5-m for LPG. The average power per ton of refrigerant for R600A is about 20% lower than that of LPG.

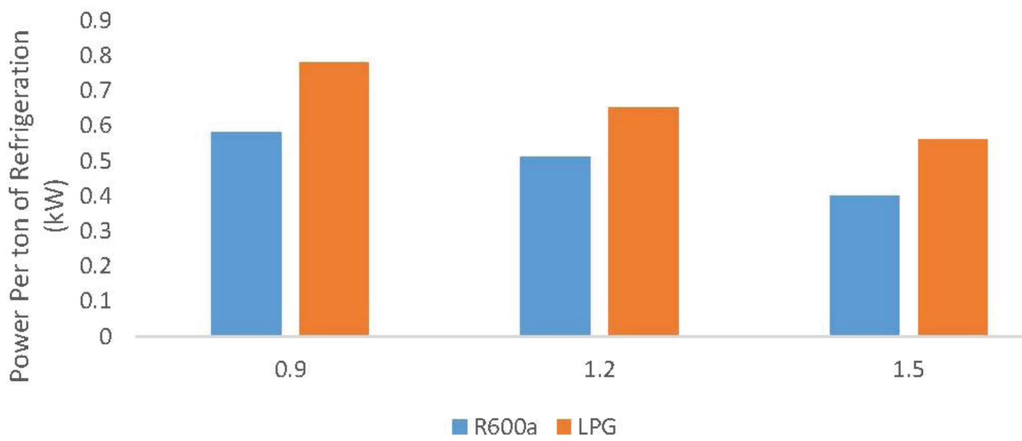

FIG. 6: Effect of capillary tube on power per ton of refrigerant at a refrigerant charge of 60 g

3.3 Effect of Capillary Tube Length and Refrigerant Charges on Cooling Capacity

Variations of cooling capacity of refrigerants for various time intervals are presented in Figs. 7–10. Figures 7 through 10 show a comparison of cooling capacity for 40-g and 60-g charges of R600A and LPG, respectively, with capillary tube lengths of 0.9 m, 1.2 m, and 1.5 m are made. At a 40-g charge, the cooling capacity of R600a varies from 3.095 kJ/s (L = 1.5 m) to 5.7752 kJ/s (L = 0.9 m), while that of LPG varies from 3.105 kJ/s (L = 1.5 m) to 5.8752 kJ/s (L = 0.9 m). At a 60-g charge, the cooling capacity of R600A varies from 2.005 kJ/s (L = 1.5 m) to 6.5779 kJ/s (L = 0.9 m) and that of LPG varies from 2.015 kJ/s (L = 1.5 m) to 6.6779 kJ/s (L = 0.9 m). Based on the results of this study, the cooling capacity of LPG is about 1.59% higher than that of R600A under the same environmental condition. This is because the density and latent heat of vaporization of LPG are higher than those of R600A. When the latent heat is high, the energy efficiency and capacity of the compressor would be lower.

3.4 Effect of Capillary Tube Length and Refrigerant Charges on Refrigerant Mass Flow Rate

The effect of capillary tube length on the refrigerant mass flow rate was investigated in this study with 0.9 m, 1.2 m, and 1.5 m capillary tube length using 40-g and 60-g refrigerant charges for R600A and LPG. The variations of refrigerant mass flow rate with capillary tube length are presented in Figs. 11 and 12. From Figs. 11 and 12, it can be seen that with increase in the capillary tube length, there is a decrease in refrigerant mass flow rate because of the increase in frictional resistance which is directly proportional to the length of capillary tube. As the capillary tube length in-

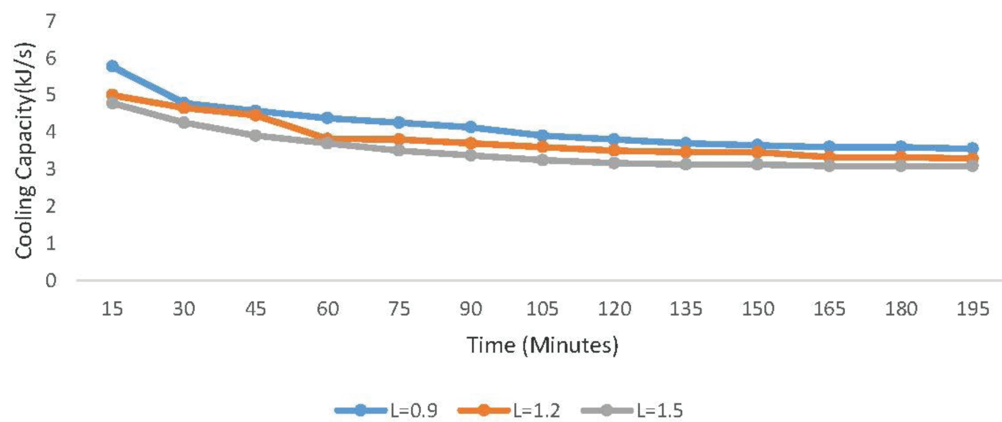

FIG. 7: Cooling capacity for a 40-g charge of R600a at capillary tube length 0.9 m, 1.2 m, and 1.5 m

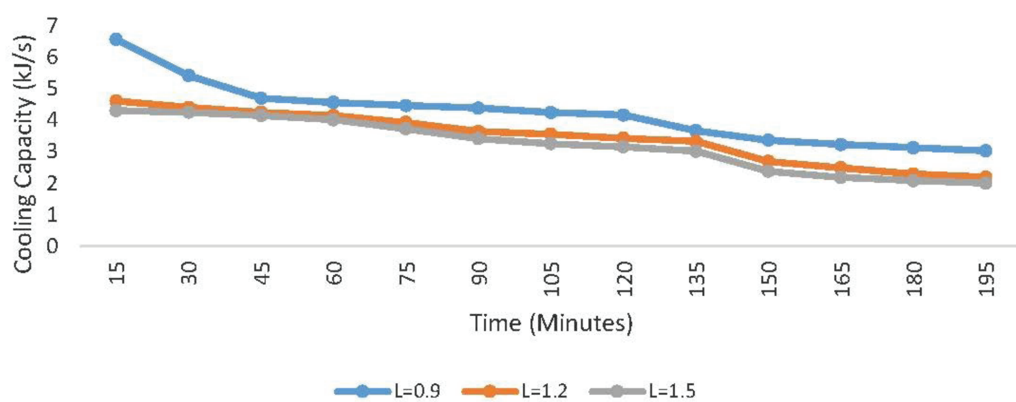

FIG. 8: Cooling capacity for a 60-g charge of R600a at capillary tube lengths 0.9 m, 1.2 m, and 1.5 m

creased from 0.9 m to 1.2 m and from 1.2 m to 1.5 m, the refrigerant mass flow rate decreased by about 4.89% and 12.8% (R600A at 60 g); 5.26% and 13.89% (LPG at 60 g); 4.92% and 15.52% (R600A at 40 g), and by about 5.17% and 16.36% (LPG at 40 g). The rate of decrease in the refrigerant mass flow rate increases with increase in the capillary tube length.

3.5 Effect of Capillary Tube Length and Refrigerant Charges on Pressure Ratio

Figures 13 and 14 present the variation of compressor pressure ratio with capillary tube length at refrigerant charges of 60 g and 40 g, respectively. Observation of Figs. 13

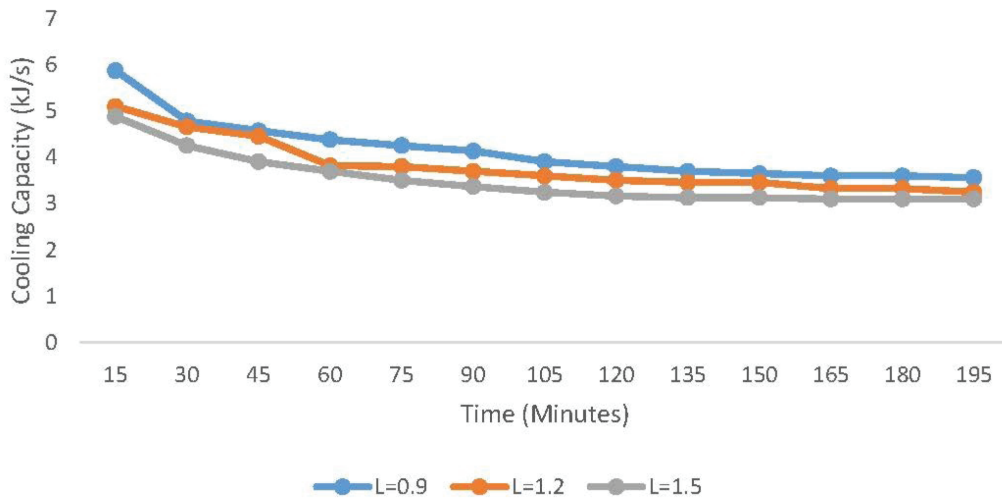

FIG. 9: Cooling capacity for a 40-g charge of LPG at capillary tube lengths 0.9 m, 1.2 m, and 1.5 m

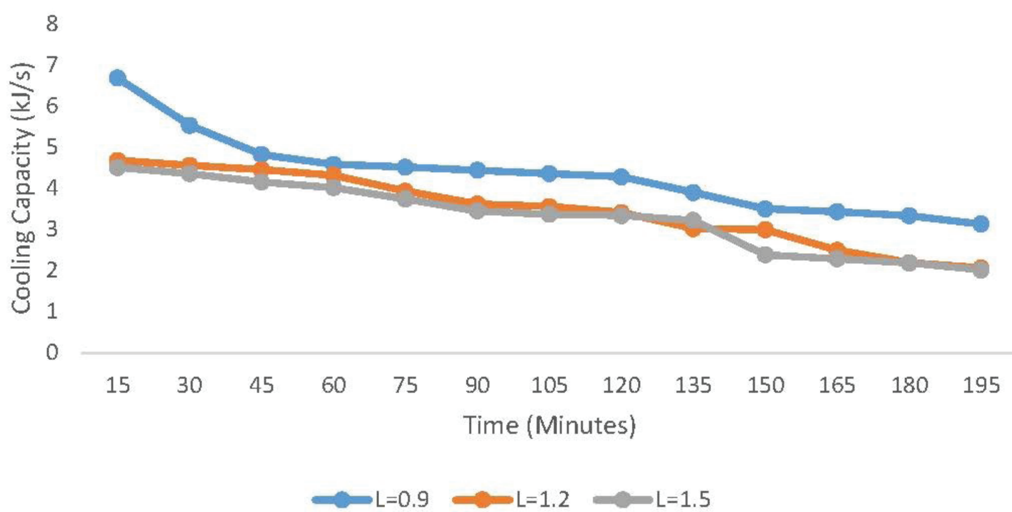

FIG. 10: Cooling capacity for a 60-g charge of LPG at capillary tube lengths 0.9 m, 1.2 m, and 1.5 m

and 14 shows that as the capillary tube length increases, the pressure ratio also increases. The figures also reveal a significant difference between the pressure ratio of LPG and R600A. The pressure ratio of R600A is higher than that of LPG, which is an indication that LPG operates at a relatively very low pressure ratio than R600A. A high pressure ratio will affect the systems performance and reliability of the com-

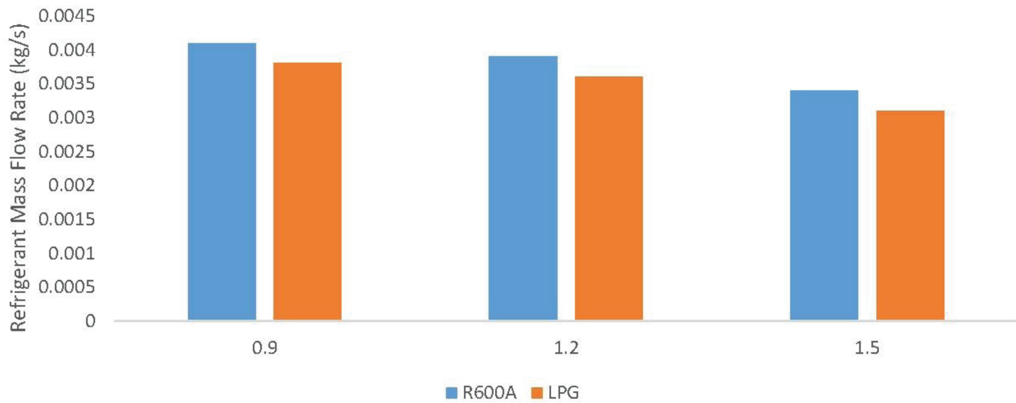

FIG. 11: Effect of capillary tube length on refrigerant mass flow rate at a refrigerant charge of 60 g

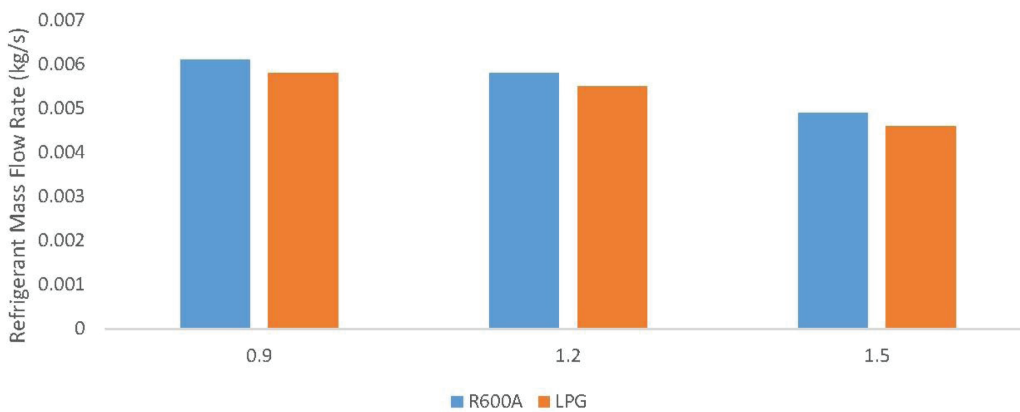

FIG. 12: Effect of capillary tube length on refrigerant mass flow rate at a refrigerant charge of 40g

ponents negatively, while a low pressure ratio will prolong the equipment life and greatly improve its performance.

3.6 Variation of Evaporator Temperature with Pull-Down Time

According to Oyedepo et al. (2016) and Fatouh and El Kafafy (2006), the pull-down time is the time required for changing the evaporator chamber air temperature from ambient condition (32^{o}C) to the desired final temperature (–12^{o}C) based on ISO-8187 standard for the considered refrigerator class (Bolaji, 2010; Fatouh and El Kafafi; 2006; Mohanrajet et al., 2007). Figures 15 and 16 show the comparison of pull-down time of R600A and LPG in the refrigerator for 40-g and 60-g charges, respectively.

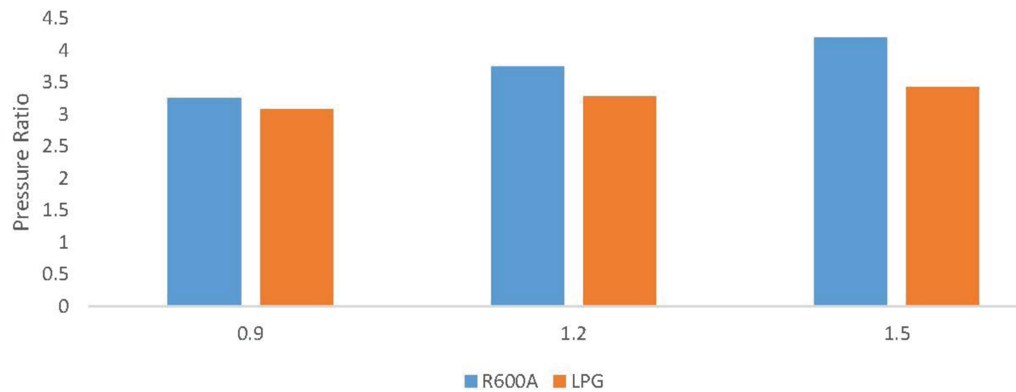

FIG. 13: Effect of capillary tube length on pressure ratio at a refrigerant charge of 60 g

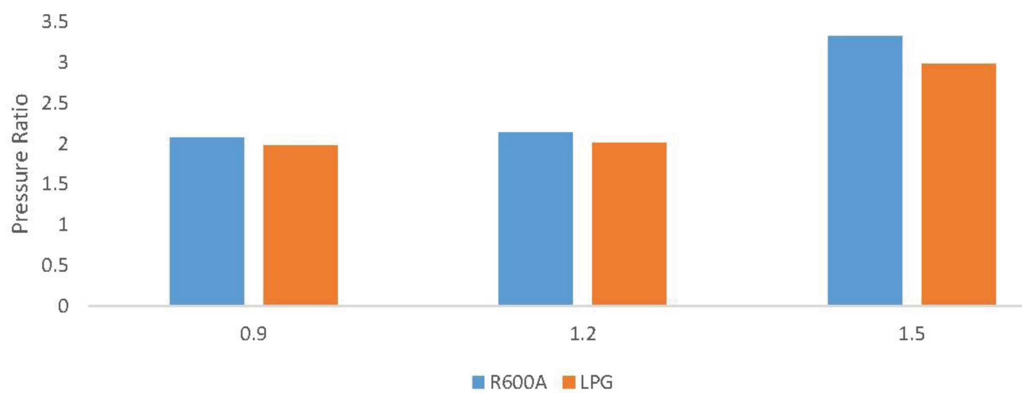

FIG. 14: Effect of capillary tube length on pressure ratio at a refrigerant charge of 40 g

According to the ISO standard, the design temperature (–12^{o}C) and pull-down time of 135 min were achieved in the refrigerator system using 60 g of R600a with capillary tube lengths 1.2 m and 1.5 m (Fig. 16). Using 60 g of LPG, the design temperature (–12^{o}C) was achieved at pull-down time of 105 min with a capillary tube length of 1.5 m. These results show that the design standard set by ISO for the refrigeration system was achieved with refrigerant charge of 60 g and capillary tube length of 1.5 m using LPG at a lower time (105 min) compared to R600A.

In order to accept a refrigerant as a drop-in replacement, similar or better cooling capacity and power consumption should be achieved (Bolaji, 2012). Based on pull-down time and COP, the appropriate combination of capillary tube length and refrigerant charge as a drop in refrigerant for chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants is LPG with L = 1.5 m and w_r = 60 g, on the basis of

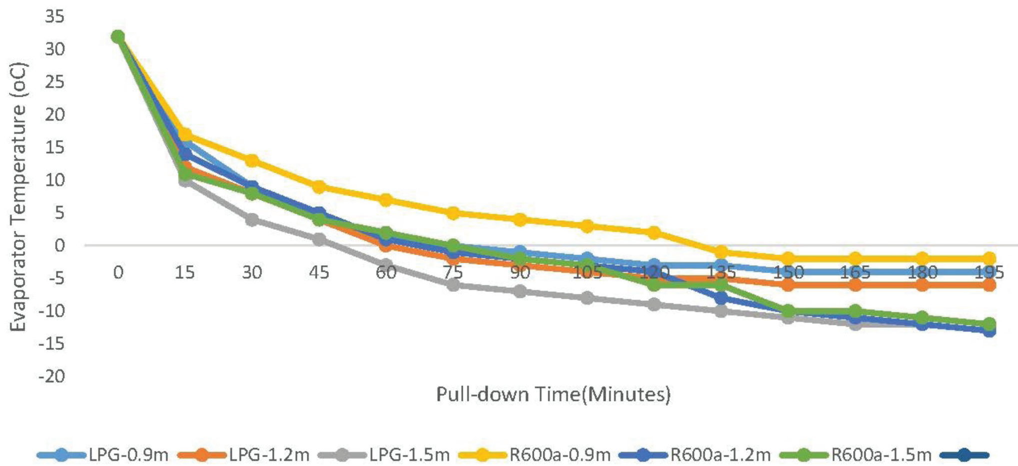

FIG. 15: Plot of evaporator temperature against pull-down time for a 40-g charge of LPG and R600A

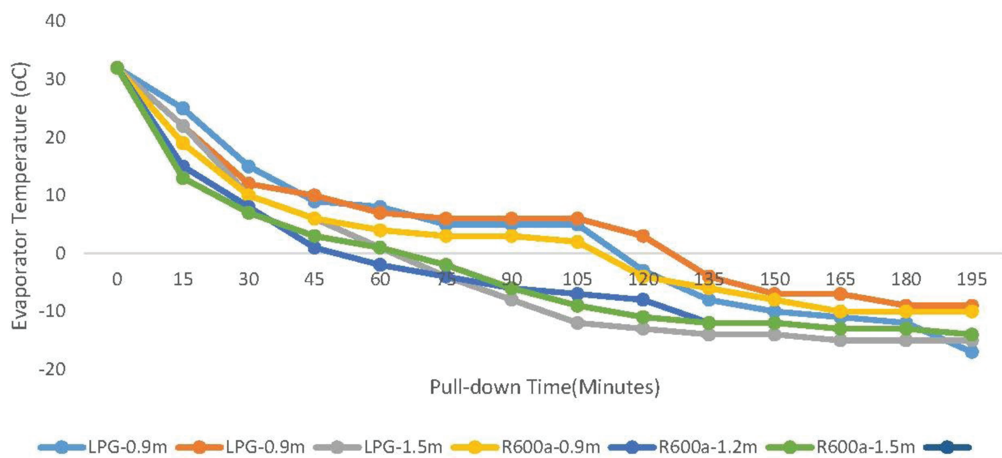

FIG. 16: Plot of evaporator temperature against pull-down time for a 60-g charge of LPG and R600A

cooling capacity, the best combination of LPG refrigerant is L = 0.9 m and w_r = 60 g, while from power consumption per day perspective, the appropriate combination of capillary tube length and refrigerant charge is R600A with L = 1.5 m and w_r = 40 g.

4. CONCLUSIONS

The performance parameters of a domestic refrigerator using LPG and R600A refrigerants have been evaluated experimentally in terms of the cooling capacity, power per ton of refrigerant (PPTR), pressure ratio (PR), pull-down time, and the coefficient of performance (COP) by varying the capillary tube length and refrigerant charge at a temperature of 32^{o}C. After the successful investigation of these refrigerants, the following conclusions were drawn based on the results obtained from the study:

- The average coefficient of performance (COP) of LPG is higher than that of R600A by about 1.14%.
- The average power per ton of refrigerant for R600A is about 20% lower than that of LPG.
- The cooling capacity of LPG is about 1.59% higher than that of R600A under the same environmental condition.
- Mass flow rate of R600A is about 6.63% higher than that of LPG.
- Pressure ratio of R600A is about 10.12% higher than that of LPG. The system using LPG has a relatively low pressure ratio than R600A, which indicates better system performance and reliability of the system using LPG.
- Based on pull-down time and COP, the appropriate combination of capillary tube length and refrigerant charge as a drop in refrigerant for chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants is LPG with L = 1.5 m and w_r = 60 g, on the basis of cooling capacity, the best combination of LPG refrigerant is L = 0.9 m and w_r = 60 g, while from power consumption per day perspective, the appropriate combination of capillary tube length and refrigerant charge is R600A with L = 1.5 m and w_r = 40 g.

ACKNOWLEDGMENT

The authors acknowledge the financial support from Covenant University Management through Covenant University Center for Research, Innovation and Development (CUCRID) for this research work.

REFERENCES

Almeida, I.M.G., Barbosa, C.R.F., and Fontes, F.A.O., (2010) Thermodynamic and Thermophysical Assessment of Hydrocarbons Application in Household Refrigerator, *Thermal Eng.*, **9**(1), pp. 19–27

Alzoubi, M.A. and Zhang, T., (2015) Characterization of Energy Efficient Vapor Compression Cycle Prototype with a Linear Compressor, *Energy Procedia*, **75**, pp. 3253–3258.

Anand, S. and Tyagi, S.K., (2012) Exergy Analysis and Experimental Study of a Vapor Compression Refrigeration Cycle. A Technical Note, *J. Thermal Anal. Calorim.*, **110**, pp. 961–971.

Bilal, A.A. and Salem, A.S., (2003) Assessment of LPG as a Possible Alternative to R-12 in Domestic Refrigerators, *Energy Convers. Manage.*, **44**, pp. 381–388.

Boeng, J. and Melo, C., (2012) A Capillary Tube-Refrigerant Charge Design Methodology for Household Refrigerators — Part II: Equivalent Diameter and Test Procedure, *Proc. Int. Refrigeration and Air Conditioning Conf.*, Purdue University, West Lafayette, IN, Paper 1180, pp. 1–10.

Bolaji, B.O., (2012) Experimental Study of R152a and R32 to Replace R134a in a Domestic Refrigerator, *Energy*, **35**, pp. 3793–3798.

Copetti, J. and Macagnan, M., (2005) The Use of Hydrocarbons Propanea Isobutane in Refrigeration Systems, *Proc. ABCM 18th Int. Congress of Mechanical Engineering, COBEM 2005*, Rio de Janeiro, Brazil, pp. 1–9.

Dincer, I. and Kanoglu, M., (2010) *Refrigeration Systems and Applications*, 2nd ed., West Sussex, UK: John Wiley & Sons Ltd.

Fatouh, M. and El Kafafy, M., (2006) Experimental Evaluation of a Domestic Refrigerator Working with LPG, *Appl. Thermal Eng.*, **26**, pp. 1593–1603.

Fiorelli, F.A.S. and Silvares, O.M., (2004) Experimental Validation of a Capillary Tube Simulation Model with Refrigerant Mixtures Flow, *Thermal Eng.*, **3**(1), pp. 15–23.

Gonçalves, J.M. and Melo, C., (2004) Experimental and Numerical Steady-State Analysis of a Top-Mount Refrigerator, *Proc 10th Int. Refrigeration and Air Conditioning Conf.*, Purdue, West Lafayette, IN.

Guobing, Z. and Yufeng, Z., (2006) Experimental Investigation on Hysteresis Effect of Refrigerant Flowing through a Coiled Adiabatic Capillary Tube, *Energy Convers. Manage.*, **47**, pp. 3084–3093.

Matani, A.G. and Agrawal, M.K., (2013) Effect of Capillary Diameter on the Power Consumption of VCRS using Different Refrigerants, *Int. J. Appl. Innov. Eng. Manage.*, **2**(3), pp. 116–124.

Mohanraj, M, Jayaraj, S and Muraleedharan, C. (2007). Improved energy efficiency for HFC134a domestic refrigerator retrofitted with hydrocarbon mixture (HC290/HC600a) as drop-in substitute, Energy for Sustain. Develop., **10**(4), pp. 29–33.

Naer, V., Rozhentsev, A., and Wang, C.-C., (2001) Rationally Based Model for Evaluating the Optimal Refrigerant Mass Charge in Refrigerating Machines, *Energy Convers. Manage.*, **42**(18), pp. 2083–2095.

Oyedepo, S.O., Fagbenle, R.O., Babarinde, T.O., Odunfa, K.M., Oyegbile, A.D., Leramo, R.O., Babalola, P.O., Kilanko, O., and Adekeye, T., (2016) Effect of Capillary Tube Length and Refrigerant Charge on the Performance of Domestic Refrigerator with R12 and R600a, *Int. J. Adv. Thermofluid Res.,* **2**(1), pp. 2–14.

Oyelami, S. and Bolaji, B.O., (2015) Experimental Investigation of the Performance of Liquefied Petroleum Gas (LPG) Refrigerant in a Vapor Compression Refrigeration System, *Int. J. Sci. Eng. Res.*, **6**(6), pp. 1158–1167.

Park, C., Lee, S., Kang, H., and Kim, Y., (2007) Experimentation and Modeling of Refrigerant Flow through Coiled Capillary Tubes, *Int. J. Refrig.*, **30**, pp. 1168–1175.

Pathak, S.S., Shukla, P., Chauhan, S., and Srivastava, A.K., (2014) An Experimental Study of the Effect of Capillary Tube Diameter and Configuration on the Performance of a Simple Vapor Compression Refrigeration System, *J. Mech. Civil Eng.*, **11**(3), pp. 101–113.

Punia, S.S. and Singh, J., (2015) An Experimental Study of the Flow of LPG as Refrigerant Inside an Adiabatic Helical Coiled Capillary Tube in Vapor Compression Refrigeration System, *Heat Mass Transfer*, **51**(11), pp. 1571–1577.

Rasti, M., Hatamipour, M.S., Aghamiri, S.F., and Tavakoli, M., (2012) Enhancement of Domestic Refrigerator's Energy Efficiency Index using a Hydrocarbon Mixture Refrigerant, *Measurement*, **45**, pp. 1807–1813.

Salim, T.K., (2012) The Effect of the Capillary Tube Coil Number on the Refrigeration System Performance, *Tikrit J. Eng. Sci.*, **19**(2), pp. 18–29.

UNEP, (2012) Study of the Potential for Hydrocarbon Replacement in Existing Domestic Small Commercial Refrigeration Appliances, UNEP France, accessed May 10, 2012, from http://www.unepie.org