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

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ABSTRACT

In this work, a comparative experimental study on performance of domestic refrigerator using R600A and LPG with varying refrigerant charge (w_r) and capillary tube length (L) was carried out. Continuous running and cycling tests were performed on a domestic refrigerator under tropical conditions using refrigerants LPG and R600A with different capillary tube lengths and various charges. The test rig for the experiment was a vapor compression refrigerator designed to work with R12. 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 small refrigerator are achieved earlier using refrigerant charge 60g of LPG with 1.5m capillary tube length. The highest COP (4.8) was obtained using 60g charge of LPG with L = 1.5m. 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. In conclusion, the system performed best with LPG in terms of COP and cooling capacity. In term of power consumption R600a performed best.

Keywords: Cooling capacity, refrigeration system, COP, refrigerant charge, capillary tube length, power consumption

1 INTRODUCTION

Refrigeration plays significant role in domestic, industrial, commercial and health sectors for comfort, necessity of food storage, medical applications etc. There are innumerable applications of such systems and they are the major consumer of electricity around the world [1; 2]. The high energy consumption by refrigeration system is due to the large amounts of units being used and also to their low thermodynamic efficiencies [3]. 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 ozone layer etc. To overcome these aspects there is urgent need of 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 [4].
The energy consumption by household refrigerators depends on its components, the refrigerant charge and ambient conditions. It is well known that domestic refrigerators have highest efficiency when operating with certain combinations of capillary tube and refrigerant charge [5; 6]. Capillary tubes are used as expansion device 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, cheap and

cause the compressor to start at low torque as the pressure across the capillary tube equalize during the off-cycle [3]. In order to enhance the system cooling capacity, the capillary tube and the suction line are usually placed together forming a counter-flow heat exchanger [7]. The heat exchanger may be of lateral or concentric type [8]. 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 refrigerant in a capillary tube and the effects of the capillary tube dimensions and geometry on the performance of vapour compression refrigeration system [3; 8; 9; 10; 11;12;13].

The literature reveals that most of the previous studies have focused on the independent variation of refrigerant charge (w_r) or capillary tube geometries (L or d), while study on 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 performance of a domestic refrigerator using LPG and R600A was also carried out.

The concerns on global warming and ozone layer depletion have mandated replacing chlorofluorocarbon (CFC) and hydro chlorofluorocarbon (HCFC) refrigerants with alternative (Hydrocarbon) refrigerants in domestic refrigerators [14]. The efforts to explore eco-friendly alternative of R12 and R22 have brought out liquefied petroleum gas (LPG) and R600A (Isobutane) as the best promising substitutes of 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 [15]. The ozone depletion potential (ODP) of LPG is 0 and Global warming potential (GWP) is 8 which is significantly negligible as compare to other refrigerant and exhibits properties similar to that of R12 [16]. LPG does not form acids and there by eliminates the problem with blocked capillaries.

From environmental impacts point of view, R600A (Isobutane) has low values (<20) of Global Warming Potential (GWP) and null Ozone Depleting Potential (ODP) [17]. Concerning 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°C to 491°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 domestic refrigeration system. There are only few studies reported in literature for the study of flow of LPG and R600A through capillary tube. Therefore, in order to strengthen the work of LPG and R600A as refrigerants, the present study has been carried out to investigate performance of a domestic refrigerator by simultaneously varying the refrigerants charges and capillary tube length using LPG and R600A as refrigerants.

2. 0 MATERIALS AND METHODS

2.1 Thermodynamic Processes

Figure 3 shows 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. Thermodynamic processes 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 superheated refrigerant. It passes then through the suction line heat exchanger, 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 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 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 saturated liquid at low temperature and pressure, and then the cycle is complete [18].

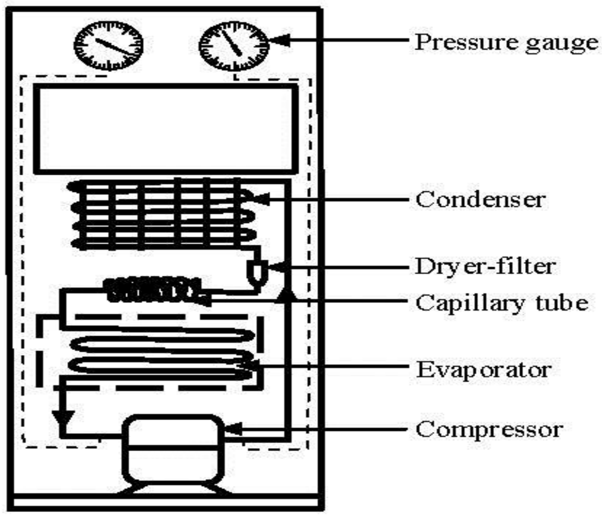

Figure 1. Schematic Diagram of Experimental Refrigerator.

2.2 Experiment and Analysis

This study is an extension of work by [19]. Hence, the experimental procedure in the study is similar to the present study. The experimental setup (Figure 2) consisted of a domestic VCRS of 1 ton of refrigeration (TR) capacity designed to work with R12, an evaporator of 79 litre 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 kW h accuracy) for measuring the energy consumption. The test rig was thoroughly checked and commissioned before it was subjected to series of tests at various conditions. The specifications of the domestic refrigerator used in this study are shown in Table 1. Experiments were conducted with LPG and R600a, by varying *refrigerant charge* from 40g to 60g and *L* as 0.9m, 1.2m and 1.5 m, with dry bulb temperature of 32ºC. The temperature (- 30ºC to +90ºC), pressure (100 to 1300kPa) and compressor power (0 to 1100W) were measured with an uncertainty of ± 0.1 %.

Figure 2: The experimental setup.

Table 1: *Specification of the Base Line Test Unit*

ITEM	**SPECIFICATION**
Unit Type	Freezer
Internal Volume	69L
Refrigerant/Lubricant	R12/ Mineral Oil
Compressor	Reciprocating Compressor
Evaporator Diameter	Cross flow fin and heat exchanger 6.4mm
Condenser Diameter	Natural cooling hot plate type heat exchanger 6.4mm
Expansion Device Diameter	Capillary tube 0.8mm

The refrigerants were charged into the system with the 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 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 under the average ambient temperature of 32°C at 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 (P_R), the isentropic compressor work (W_c), power per ton of refrigeration and the COP of the refrigerator, as defined in the following fundamental equations:

The cooling capacity (Q_{ev}) is given by:

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

Isentropic compression work in the compressor can be expressed as:

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

The refrigerant mass flow ($\dot{m}_r$) can be estimated using the following equation [20]:

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

Power per ton of refrigeration (PPTR) is obtained by:

$$PPTR = 3.5 \cdot \frac{W_c}{Q_{ev}} \tag{4}$$

Pressure ratio (PR) is defined by:

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

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:

$$COP = \frac{Q_{ev}}{W_c} \tag{6}$$

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

3.0 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.

Effect of Capillary Tube Length on the COP of the Refrigerator System

Figures 3 and 4 show the effect of *L* on the system COP with 40g and 60g 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 = 60g and L = 1.5m, while for R600a the highest COP of 4.76 was obtained at w_r = 60g and *L* = 1.5m. Based on the result of this study, the average COP of the two refrigerants are very close but the average COP obtained using LPG is about 1.14% higher than that of R600a.

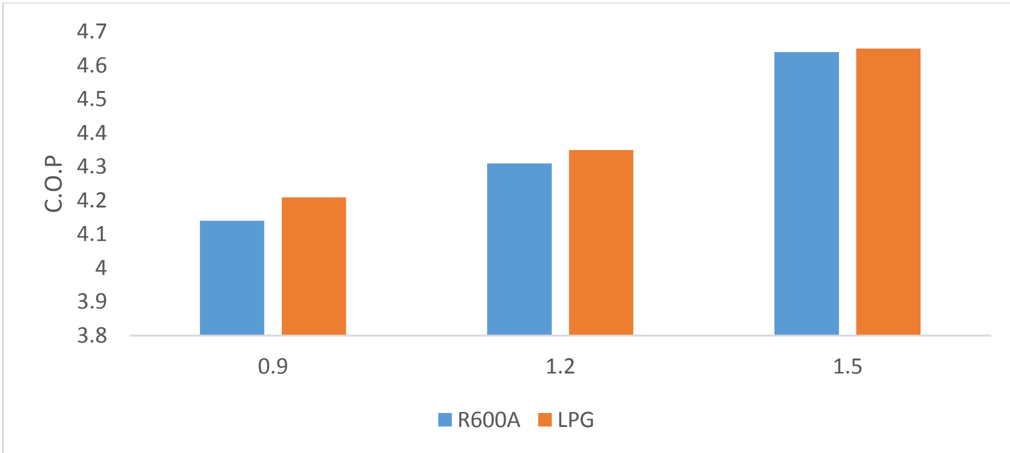

Figure 3: Effect of capillary tube length on the COP of the System at Refrigerant Charge of 40g.

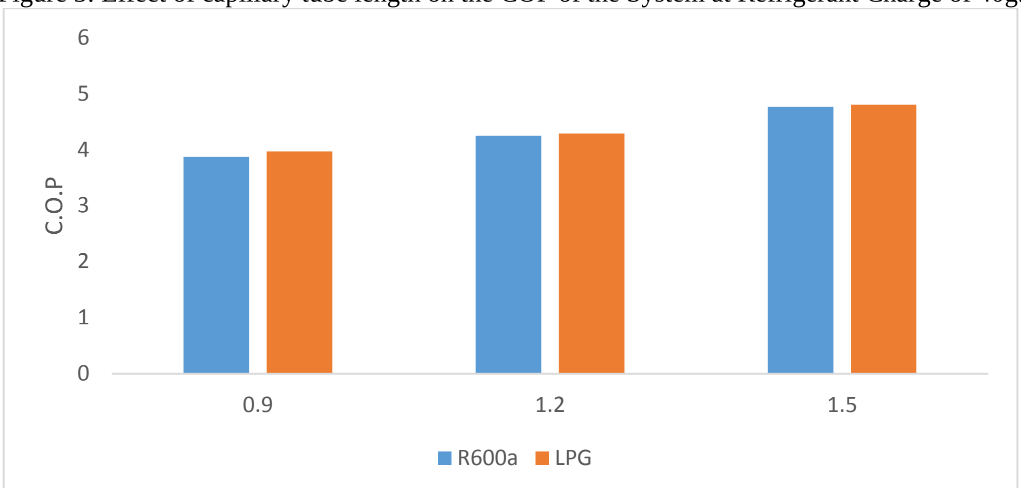

Figure 4: Effect of capillary tube length on the COP of the System at Refrigerant Charge of 60g.

Effect of Capillary Tube Length on Power per Ton of Refrigeration

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 refrigeration (*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 increase in mass flow rate of refrigerant through the compressor. The lowest PPTR of 0.36 kW was recorded at w_r = 40g for R600a with L=1.5m, while the lowest *PPTR* of 0.43 kW was recorded at w_r = 40g with L=1.5m for LPG. The average power per ton of refrigeration for R600a is about 20% lower than that of LPG.

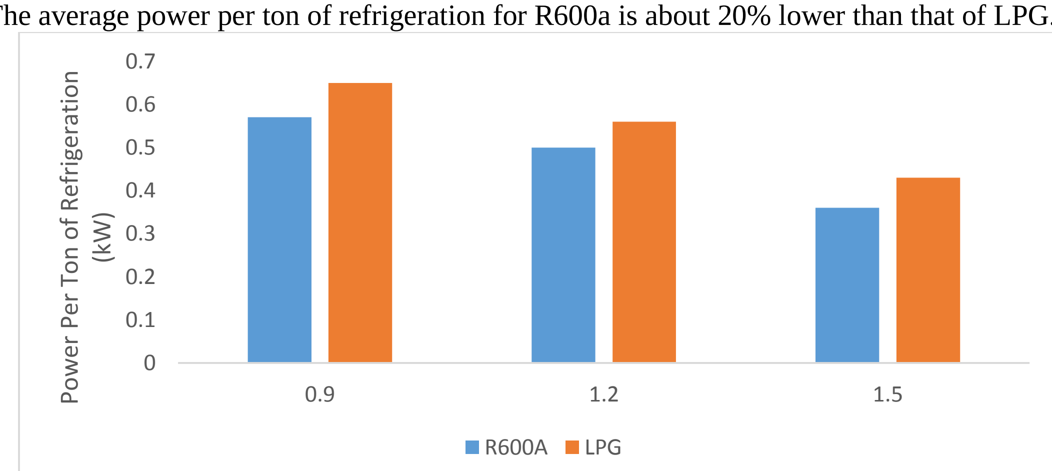

Figure 5: Effect of capillary tube on power per ton of refrigeration at Refrigerant Charge of 40g

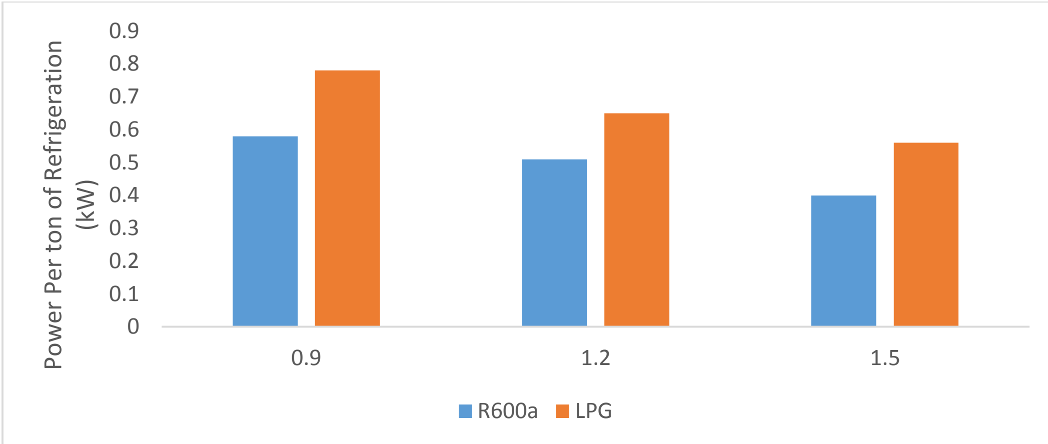

Figure 6: Effect of capillary tube on power per ton of refrigeration at Refrigerant Charge of 60g

Effect of Capillary Tube Length and Refrigerant Charges on Cooling Capacity

Variations of cooling capacity of refrigerants for various time intervals are presented in Figures 7 to 10. From Figures 7 to 10, comparison of cooling capacity for 40g and 60g charges of R600a and LPG, respectively with capillary tube lengths 0.9m, 1.2m and 1.5m are made. At 40g charge, the cooling capacity of R600a varies from 3.095 kJ/s (L=1.5m) to 5.7752 kJ/s (L= 0.9m) while that of LPG varies from 3.105 kJ/s (L= 1.5m) to 5.8752 kJ/s (L= 0.9m). At 60g charge, cooling capacity of R600a varies from 2.005 kJ/s (L=1.5m) to 6.5779 kJ/s (L =0.9m) and that of LPG varies from 2.015 kJ/s (L=1.5m) to 6.6779 kJ/s (L = 0.9m). 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 that of R600A. When the latent heat is high, the energy efficiency and capacity of the compressor would be lower.

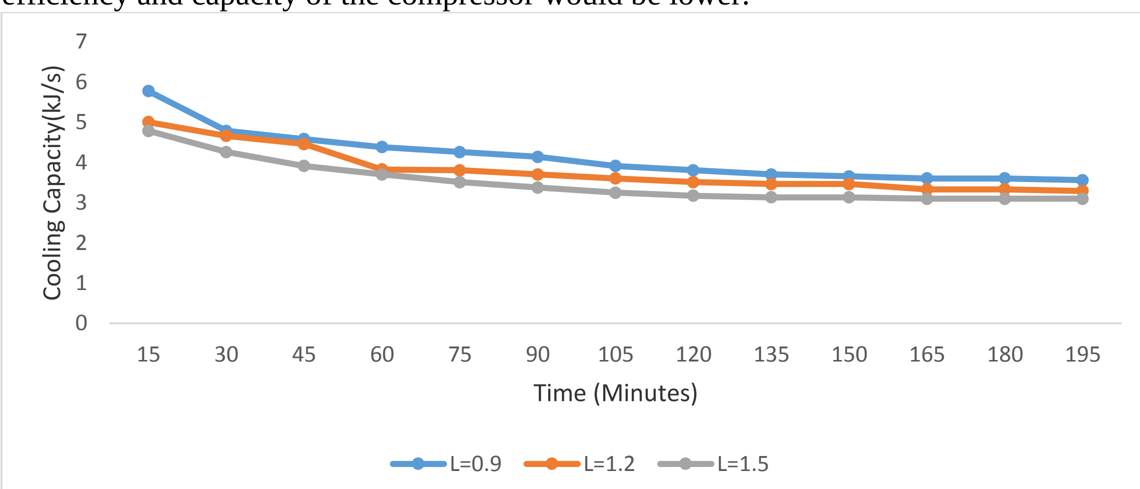

Figure 7: *Cooling Capacity for 40g charge of R600a at capillary tube length 0.9m, 1.2m and 1.5m*

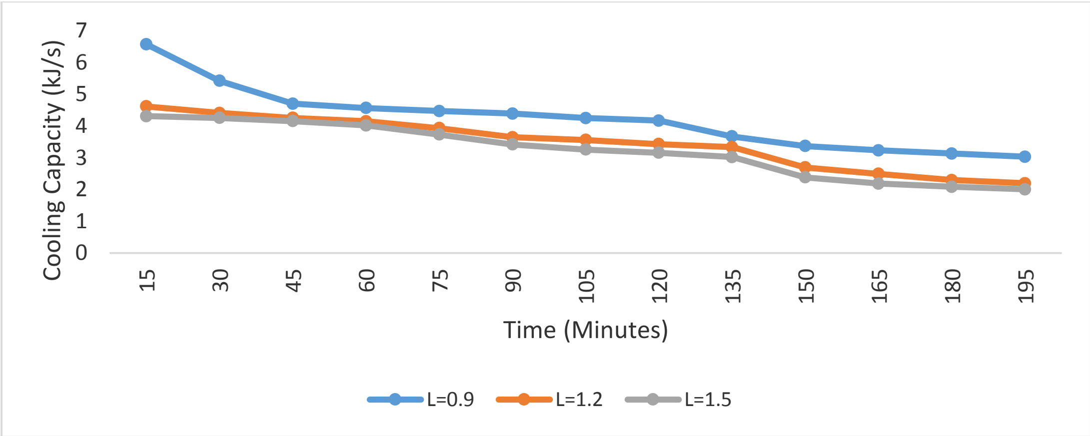

Figure 8: *Cooling Capacity for 60g charge of R600a at capillary tube lengths 0.9m, 1.2m and 1.5m*

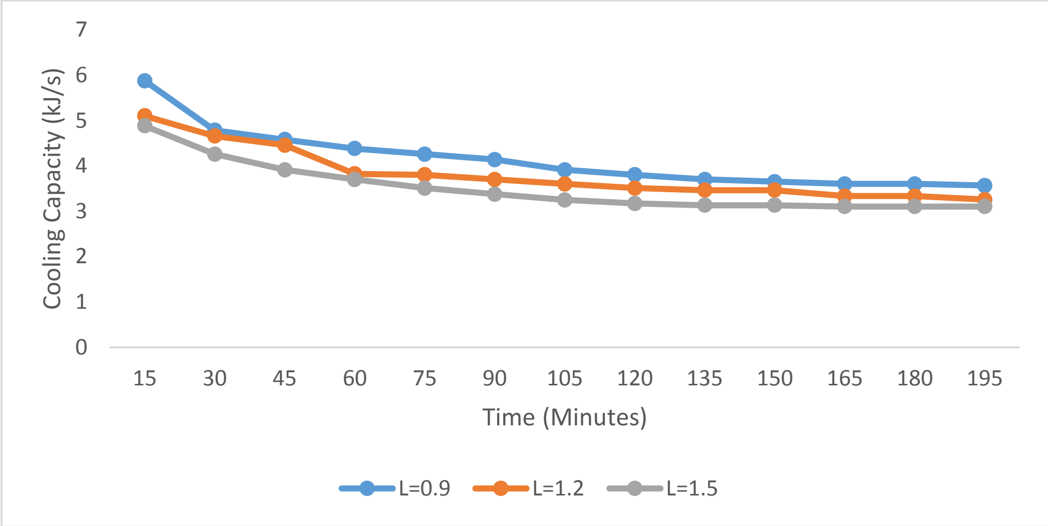

Figure 9: *Cooling Capacity for 40g charge of LPG at capillary tube lengths 0.9m, 1.2m and 1.5m*

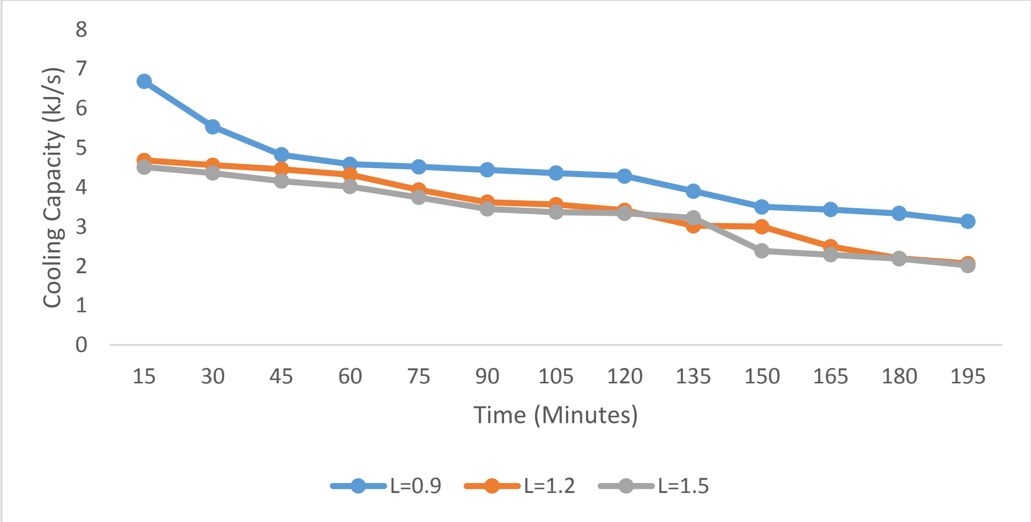

Figure 10: *Cooling Capacity for 60g charge of LPG at capillary tube lengths 0.9m, 1.2m and 1.5m*

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

The effect of capillary tube length on refrigerant mass flow rate was investigated in this study with 0.9m, 1.2m and 1.5m capillary tube length using 40g and 60g refrigerant charges for R600A and LPG. The variations of refrigerant mass flow rate with capillary tube length are presented in Figures 11 and 12. From figures 11 and 12, it can be seen that, with increase in capillary tube length there is decrease in refrigerant mass flow rate because of increase in frictional resistance which is directly proportional to the length of capillary tube. As the capillary tube length increased from 0.9 m to 1.2m and 1.2m to 1.5m, the refrigerant mass flow rate decreased by about 4.89% and 12.8% (R600A at 60g); 5.26% and 13.89% (LPG at 60g); 4.92% and 15.52% (R600A at 40g) and 5.17% and 16.36% (LPG at 40g). The rate of decrease in refrigerant mass flow rate increases with increase in capillary tube length.

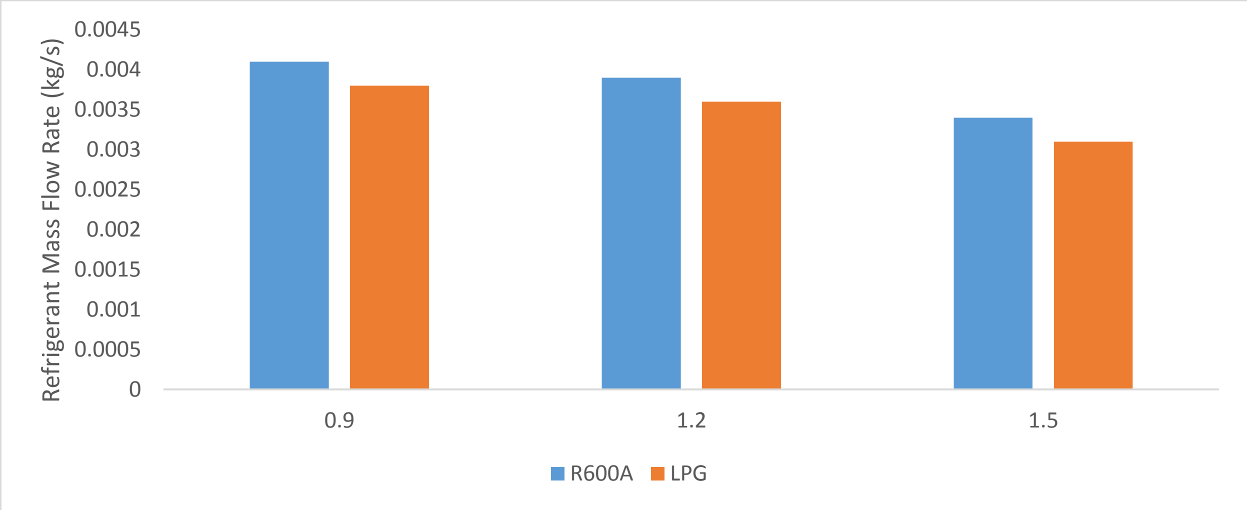

Figure 11: Effect of Capillary Tube Length on Refrigerant Mass Flow Rate at Refrigerant Charge of 60g

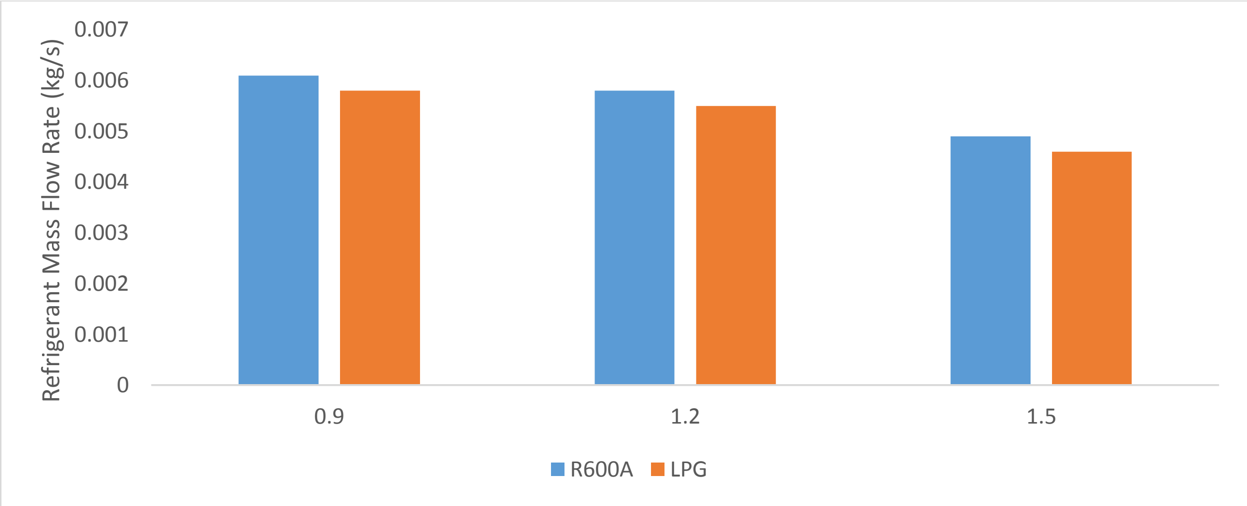

Figure 12: Effect of Capillary Tube Length on Refrigerant Mass Flow Rate at Refrigerant Charge of 40g

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 60g and 40g, respectively. A glance look at Figures 13 and 14, it can be seen that as capillary tube length increases, the pressure ratio also increases. The figures also reveal significant difference between the pressure ratio of LPG and R600A. 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. High pressure ratio will affect the systems performance and reliability of the components negatively, while low pressure ratio will prolong the equipment life and greatly improve its performance.

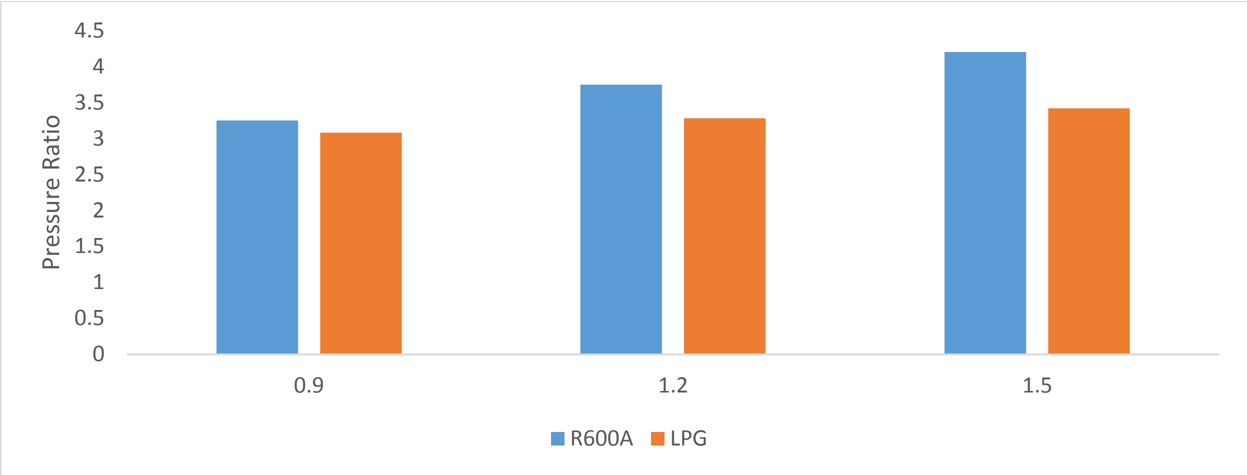

Figure 13: Effect of Capillary Tube Length on Pressure Ratio at Refrigerant Charge of 60g

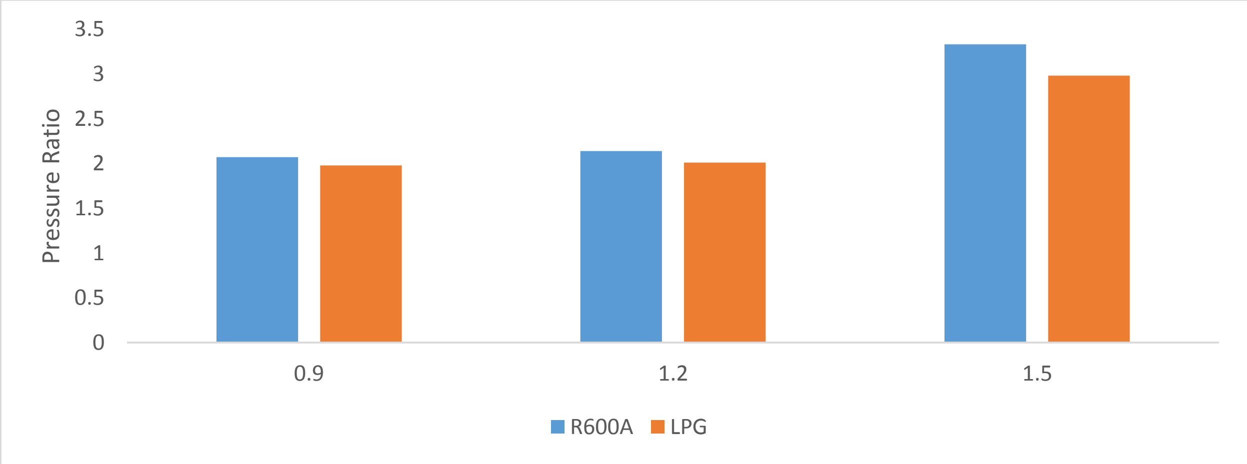

Figure 14: Effect of Capillary Tube Length on Pressure Ratio at Refrigerant Charge of 40g

Variation of Evaporator Temperature with Pull – Down Time

According to [19, 22], the pull-down time is the time required for changing the evaporator chamber air temperature from ambient condition (32°C) to the desired final temperature (-12°C) based on ISO-8187 standard for the considered refrigerator class [14]. Figures 15 and 16 show the comparison of pull-down time of R600a and LPG in the refrigerator for 40g and 60g charges, respectively.

According to ISO standard, the design temperature (-12°C) and pull – down time of 135 minutes were achieved in the refrigerator system using 60g of R600a with capillary tube lengths 1.2m and 1.5m (Figure 16). Using 60g of LPG, the design temperature (-12°C) was achieved at pull down time of 105 minutes with capillary tube length of 1.5 m, These results show that the design standard set by ISO for refrigerator system was achieved with refrigerant charge of 60g and capillary tube length of 1.5 m using LPG at lower time (105 minutes) compare to R600a.

In order to accept a refrigerant as a drop-in replacement, similar or better cooling capacity and power consumption should be achieved [21, 22]. 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 hydro chlorofluorocarbon (HCFC) refrigerants is LPG with L=1.5m and w_r = 60g, on the basis of cooling capacity, the best combination of LPG refrigerant is L=0.9m and w_r = 60g, 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 = 40g.

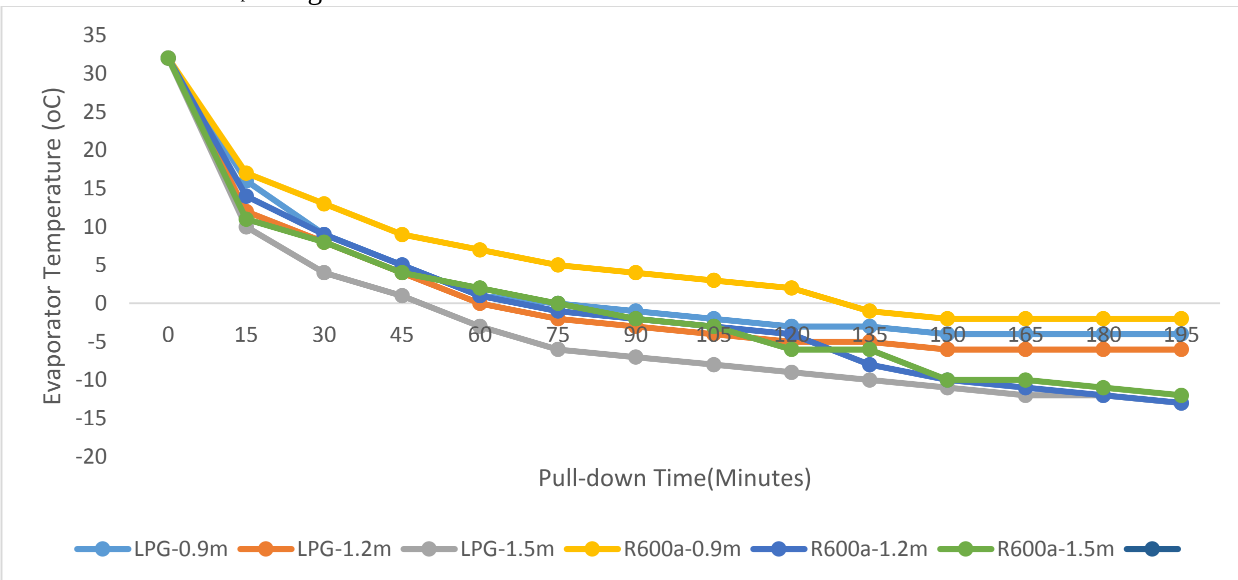

Figure 15: Plot of Evaporator Temperature against Pull- down Time for 40g charge of LPG and R600A

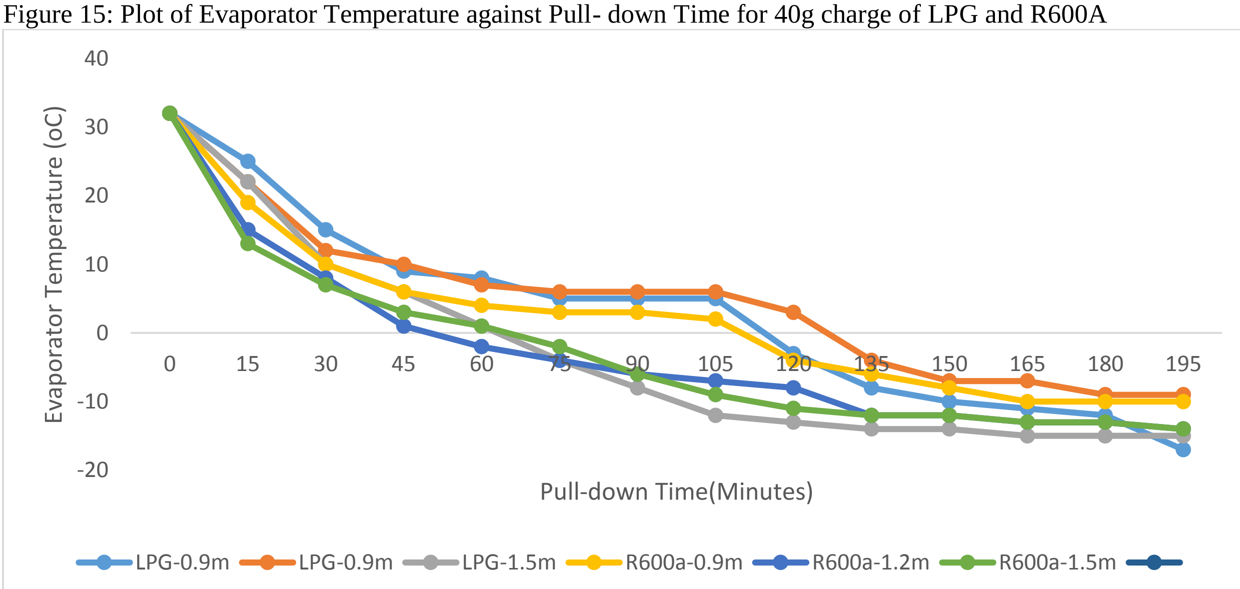

Figure 16: Plot of Evaporator Temperature against Pull - down Time for 60g charge of LPG and R600A

CONCLUSION

The performance parameters of a domestic refrigerator using LPG and R600A refrigerants have been evaluated experimentally in terms of cooling capacity, Power per ton of refrigeration (PPTR), pressure ratio (PR), pull down time and coefficient of performance (COP) by varying capillary tube length and refrigerant charge under 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 refrigeration 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 hydro chlorofluorocarbon (HCFC) refrigerants is LPG with L=1.5m and w_r = 60g, on the basis of cooling capacity, the best combination of LPG refrigerant is L=0.9m and w_r = 60g, 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 = 40g.

NOMENCLATURE

wr - Refrigerant charge

L – Capillary tube length

COP – Coefficient of performance -

LPG – Liquidified Petroleum Gas

1,2 ,3 – Evaporator inlet, evaporator outlet, compressor outlet

PR - Compressor pressure ratio

*W*c - Compressor work (kW)

$\dot{m}$ - Refrigerant mass flow rate (kg/s)

h - Specific enthalpies of refrigerant (Kj/kg)

QE - Cooling capacity (kW)

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