Effect of Capillary Tube Length and Refrigerant Charge on the Performance of Domestic Refrigerator with R12 and R600a

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Abstract: In this work, the thermodynamic performance of a domestic refrigerator was experimentally studied by simultaneously varying the refrigerant charge (m_r) and the capillary tube length (L). The potential of replacing R12 by R600a was also investigated. The test rig for the experiment was a vapor compression refrigerator designed to work with R12. The enthalpy of the refrigerants R600a and R12 for each data set for the experimental conditions were obtained by using REFPROP software (version 9.0). The results show that the design temperature of -12^{0}C (according to ISO - 8187 standard) and pull – down time of 135 minutes are achieved by using 60g of R600a with L= 1.2m and 1.5m. For R12, the design temperature is achieved at pull – down time of 165 minutes with m_r = 40g and L = 0.9m. The appropriate combination of L and m_r for R600a to be used as a drop-in refrigerant for R12 is found to be 1.5m and 60g on the basis of power consumption per day, pull-down time and COP, whereas by considering the cooling capacity, it is 0.9 m and 60g. The cooling capacity of R600a was about 9.18% higher than that of R12, the power consumed by R600a was about 24 % lower than that of R12 and the COP of R600a was about 6.3% higher than that of R12. In conclusion, the proposed R600a seems to be an appropriate long-term candidate to replace R-12 in the existing refrigerator in terms of power consumption, cooling capacity and COP.

Keywords: Refrigeration, power consumption, COP, refrigerant, capillary tube, refrigerant charge.

1. Introduction

The energy demand for refrigeration systems is escalating due to the increasing demand for comfort, necessity of food storage and medical applications. Domestic refrigerators are categorized as one of the major energy consuming household appliances (Rasti et al., 2012). It is well known that household refrigerators have highest efficiency when operating with certain combinations of capillary tube (expansion restriction) and refrigeration charge (Gonçalves and Melo, 2004; Vjacheslav et al., 2001). 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 0.5mm to

2 mm and length (L)from 2m 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 (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 counter-flow heat exchanger (Dincer and Kanoglu, 2010). The heat exchanger may be of lateral or concentric type (Park et al., 2007).

A number of experimental and numerical studies have been reported on the flow characteristics of refrigerant in a capillary tube, and the capillary tube dimensions and geometry, and their effect on the performance of vapor compression refrigeration system (VCRS). For instance, Wei et al. (2000) compared coiled and straight capillary tubes by using R22, to study the effect of coiling on the performance of VCRS. It was observed that, d and D were the most influencing on the refrigerant mass flow rate, and the helical effect increased with decrease of D. This was consistent with the findings of Kim et al. (2002) who studied the performances of capillary tubes as functions of L and d and with R22 and its alternatives, R407C and R410A. An increase in mass flow rate by 4 % and 23% were observed for R407c and R410a respectively, compared to R22. Also, with D = 40mm, for coiled capillary tube was about 9% less than that of straight capillary tubes. Similar findings were also reported in a variety of subsequent studies (Guobing and Yufeng, 2006; Park et al. 2007; Khan et al. 2008; Boeng and Melo 2012; Salim, 2012; Matani and Agrawal, 2013; Pathak et al., 2014). Joudi and Al-Amir (2014) used a finite difference model to simulate the capillary tube in split air conditioners under high outdoor air temperatures. It was shown that the capillary choking length increased with increasing outdoor air temperature. Also, among the refrigerants tested, R290 needed the maximum L whereas R410A needed the least. Abed et al. (2014) focused on the effect of capillary tube geometry on the performance of VCRS, and showed that the coiled diameter (D) of capillary tube affected the coefficient of performance (COP) strongly, as D increased from 25 to 100 mm and COP increased from 2.8 to 3.7. The increase of D more than 100 mm showed insignificant effect on the cycle COP, while the pitch space showed a minor effect (increased with increase of D).

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 (UNEP, 2012). Although R12 has been phased out in some developed countries, it is still popular in the third world countries such as Nigeria. The relatively high ozone depletion potential (ODP) and global warming potential (GWP) (1 and 2400 respectively) of R12 have prompted researchers to explore eco-friendly refrigerants and their potential to replace R12 in domestic refrigerators. The exemplary works include those reported by Jung and Radermacher (1991), Alsaad and Hammad (1998), Lee and Su (2002), Akash and Said (2003), Halimic et al. (2003), Sekhar et al. (2004), Padilla et al. (2010) and Akintunde (2013).

The literature reveals that most of the previous studies have focused on the independent variation of refrigerant charge (m_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 m_r and L. The potential of replacing R12 with R600a was also explored. The prime objectives of the study are:

- To determine the best combination of L and m_r to give effective cooling.
- To compare the cooling capacities of R12 and R600a under identical conditions
- To compare the COP and power consumption of the refrigerator with R12 and R600a.

2. Experiment and Analysis

The experimental setup (Figure 1) consisted of a domestic VCRS of 1 ton of refrigeration (TR) capacity designed to work with R12, an evaporator of 79 liter 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 R12 and R600a, by varying m_r from 40g to 100g 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 %.

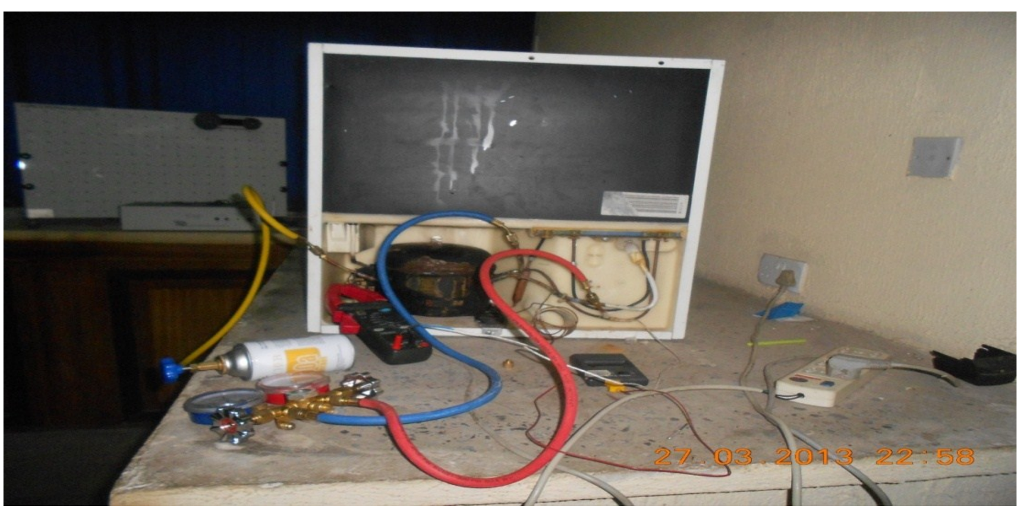

Figure 1: The experimental setup.

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	Cross flow fin and heat exchanger
Condenser	Natural cooling hot plate type heat exchanger
Expansion Device	Capillary tube

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 m_r with an accuracy of ± 0.05. The experiment was carried out under the average ambient temperature of 32^0C 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 refrigerating capacity (Q_E), compressor pressure ratio (PR), the compressor work (Wc) and the COP of the refrigerator, as defined in the following fundamental equations:

$$Q_E = \dot{m}\,(h_2 - h_1) \text{ in kW} \quad (1)$$

$$W_c = \dot{m}(h_3 - h_2) \text{ in kW} \quad (2)$$

$$P_R = \frac{P_{dis}}{P_{suc}} \quad (3)$$

$$\text{COP} = \frac{Q_E}{W} \quad (4)$$

where $\dot{m}$ = 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.

3. Results and Discussion

The effects of L and m_r on the performance parameters were analysed for both R600a and R12, with the objective of obtaining the best values of L and m_r and to study the feasibility of using R600a in a VCRS designed for R12. Figures 5 and 6 show the effect of L on the system COP; the COP increases with increase in L for all values of m_r. The highest COP of 4.64 was obtained with R600a at $m_r = 40$g and $L = 1.5$m, while for R12, the highest COP of 3.81 was obtained at $m_r = 60$g and L=1.5m. The average COP obtained using R600a is 6.3% higher than that of R12. Instantaneous power consumption is the main criterion to choose a right quantity of mass charge.

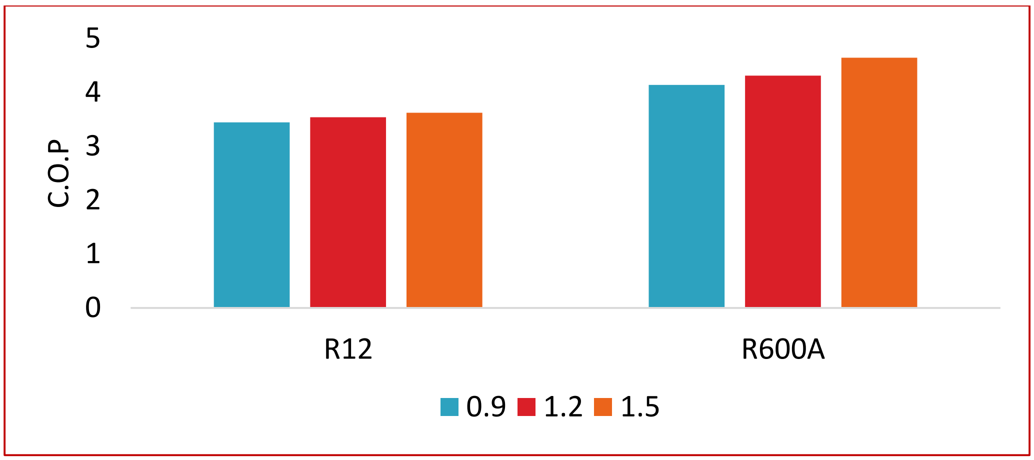

Figure 5: Effect of L on the COP of the System at m_r=40g.

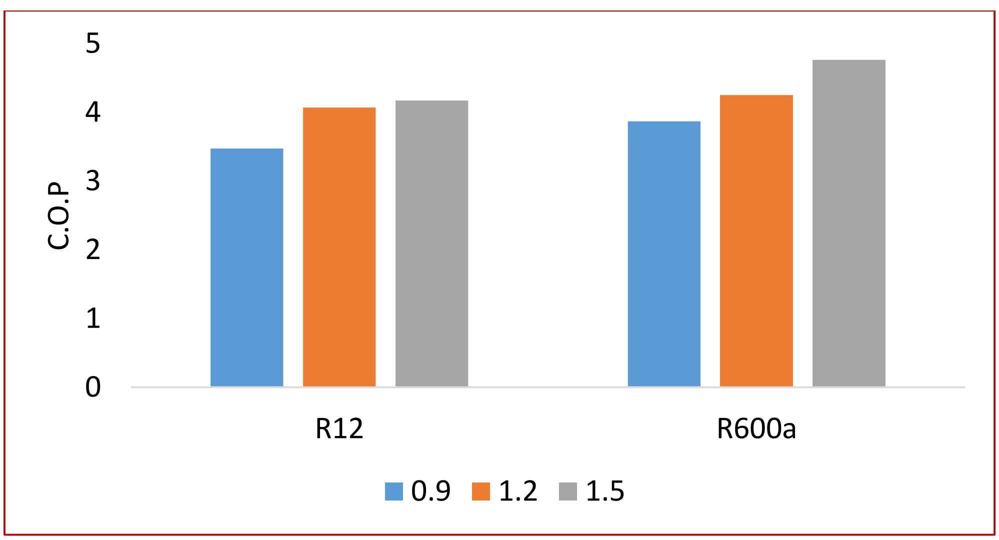

Figure 6: Effect of L on the COP of the System at m_r=60g.

Figures 7 and 8 show variation of the electric power consumption (W_c) with L and m_r. It is observed that W_c decreases with increase in L but increases with increase in m_r. This is mainly due to increase in mass flow rate of refrigerant through the compressor. The lowest W_c of 0.38 kW is recorded at $m_r = 80$g for R600a with L=1.5m, while the lowest W_c of 0.50 kW is recorded at $m_r = 40$g with L=1.5m for R12. The average power consumption for R600a is 24% lower than that of R12.

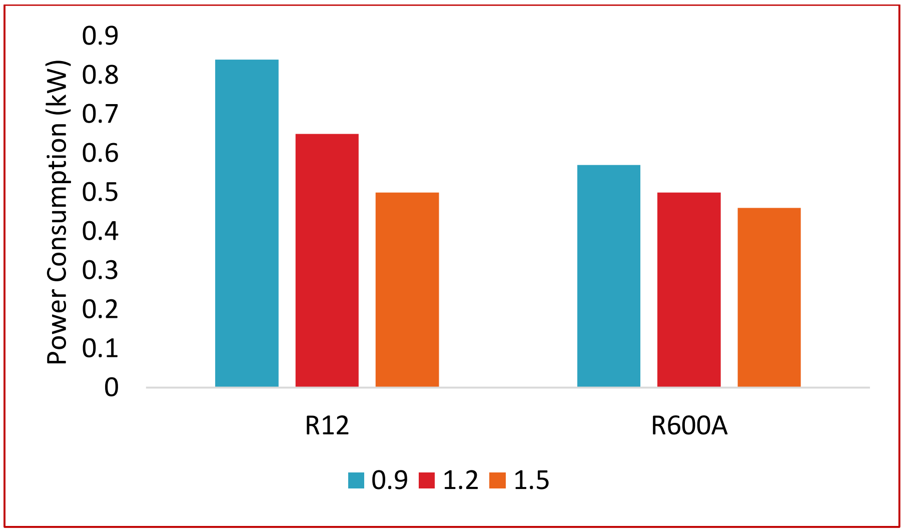

Figure 7: Effect of L on power consumption of the system at m_r=40g.

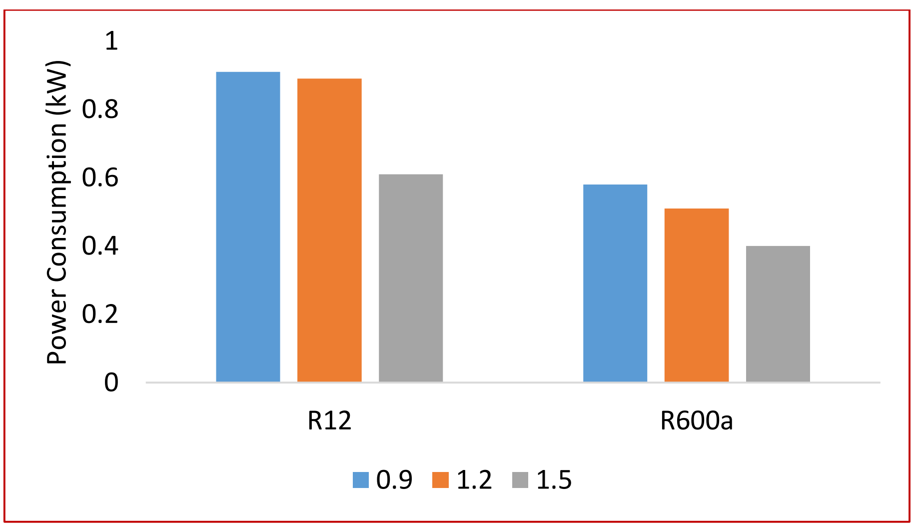

Figure 8: Effect of L on power consumption of the system at m_r=60g.

Figures 9 and 10 show the variation of P_{dis} with L and m_r. An increase in L decreases P_{dis} and hence m_r. For R12, P_{dis} of 1112.86 kPa is recorded at $m_r = 80$g and L= 0.9m, whereas for R600a, $P_{dis} = 638.02$ kPa is observed at $m_r = 100$g and L= 0.9m. This shows that R600a has a lower discharge pressure than R12 in the system. As high discharge pressure is detrimental to the performance of the system, the low P_{dis} of R600a compared to R12 indicates that using it as drop-in substitute in R12 systems will produce less strain on the compressor and hence a longer compressor life.

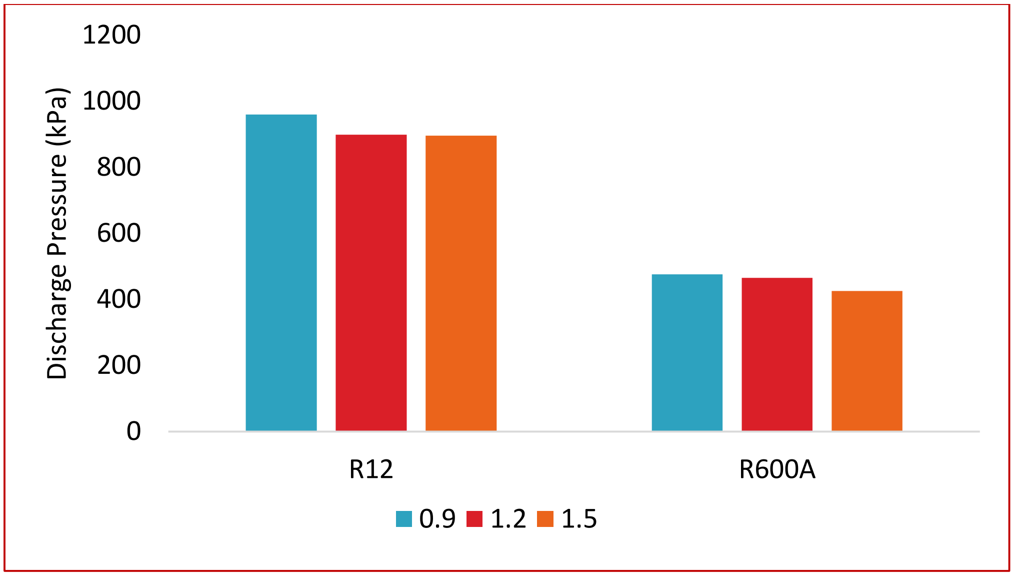

Figure 9: Effect of L on compressor discharge pressure of the system at m_r=40g.

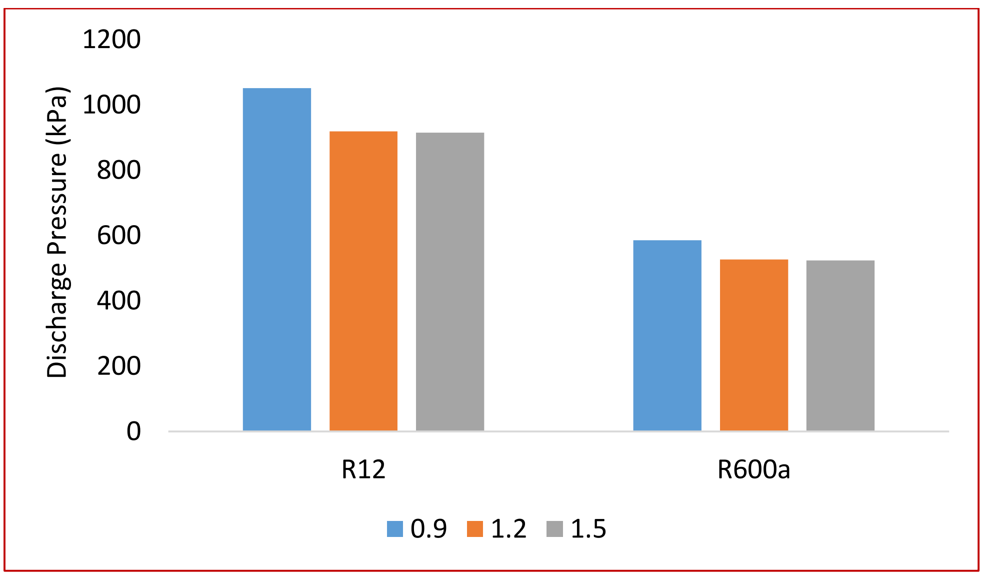

Figure 10: Effect of L on Compressor discharge pressure of the system at m_r=60g.

Figures 11 to 14 show the cooling capacity (Q_E) of R600a and R12 for m_r = 40g and 60g and L = 0.9m, 1.2m and 1.5m for different pull-down times. At m_r = 40g, the cooling capacity of R12 varies from 1.6854 kJ/s (L=0.9m) to 5.9739 kJ/s (L=1.5m) while that of R600a varies from 3.095 kJ/s (L=1.5m) to 5.7752 kJ/s (L= 0.9m). At 60g, cooling capacity of R12 varies from 1.6317 kJ/s (L=0.9) to 5.3735 kJ/s (L= 1.5m) while that of R600a varies from 2.005

kJ/s (L=1.5m) to 6.5779 kJ/s (L =0.9m). For a given m_r, the cooling capacity for R600a (Figures 11 and 12) decreases with increase in L whereas for R12 (Figures 13 and 14), it increases with increase in L. Based on this study, the cooling capacity of R600a is about 9.18% higher than that of R12.

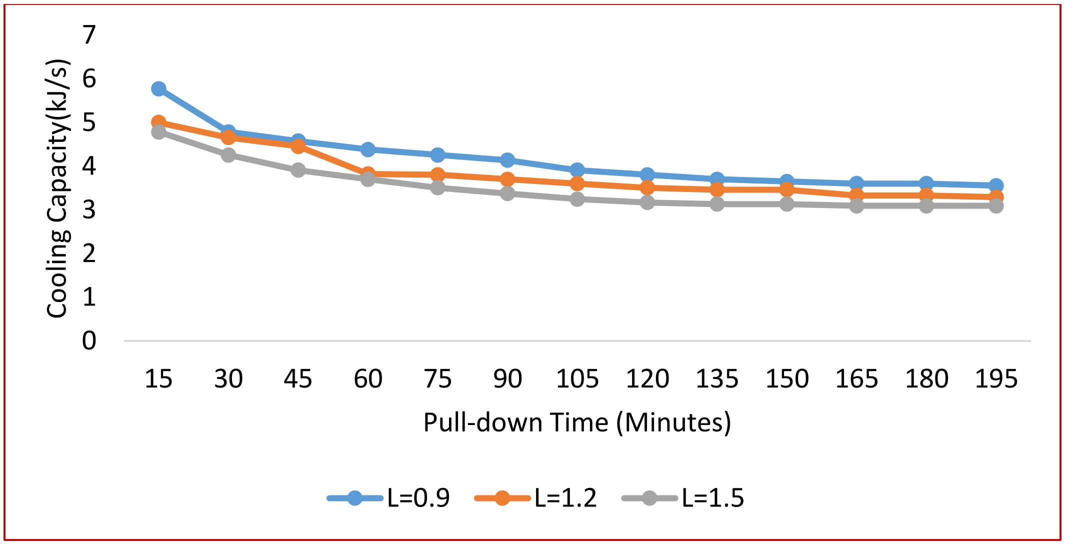

Figure 11: Cooling Capacity for 40g charge of R600a at L= 0.9m, 1.2m and 1.5m.

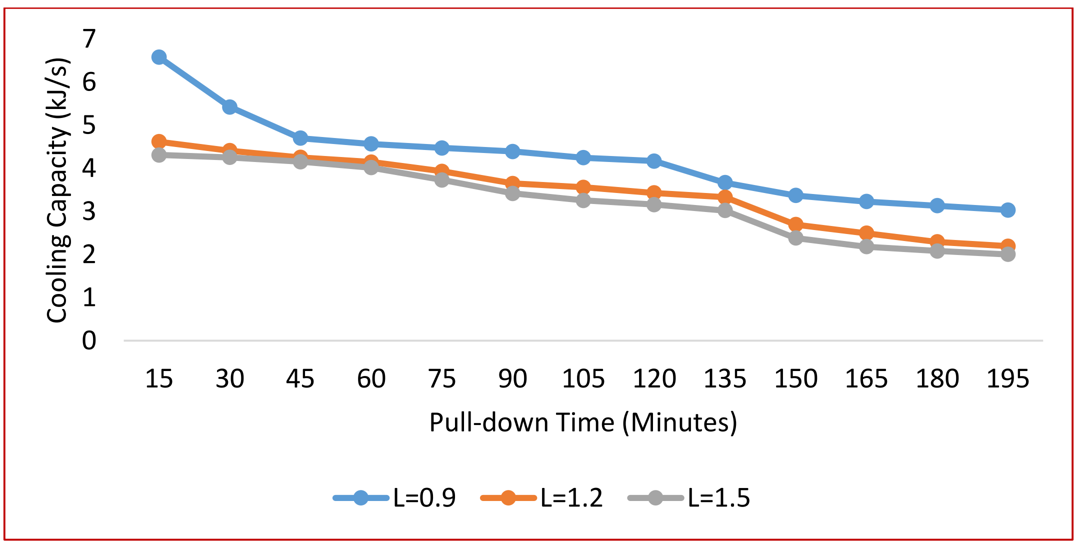

Figure 12: Cooling Capacity for 60g charge of R600a at L = 0.9m, 1.2m and 1.5m.

The pull-down time is the time required for changing the evaporator chamber air temperature from ambient condition (32^{0}C) to the desired final temperature (-12^{0}C) according to ISO-8187 standard for the considered refrigerator class (Bolaji, 2010 ; Fatouh and El Kafafy, 2006; Mohanraj et al., 2007). Increase in pull-down time implies inadequate capacity of the system. In this study, the experimental refrigerator was designed to operate below 0^{0}C. Tables 2 and 3 respectively show the comparison of pull-down times of R12 and R600a in the refrigerator, as functions of L and m_r. It can be observed that, for R12 with

$m_r = 40$g, operating temperatures of -16, -6 and -15^{0}C are obtained at pull-down time of 195 minutes with $L = 0.9$m, 1.2m and 1.5m respectively; for R600a with $m_r = 40$g, the respective operating temperatures obtained at pull-down time of 195 minutes are -8, -11 and – 14^{0}C. For R12, the design temperature according to ISO standards (-12^{0}C) is achieved at pull–down times of 165 minutes and 180 minutes at conditions of m_r= 40g, L = 0.9m and 1.5m (Table 2), and m_r= 60g, L = 0.9m (T) respectively. However, for R600a, the design temperature (-12^{0}C) is achieved at pull–down time of 135 minutes with m_r =60g and L= 1.2m and 1.5m (Table 3). It can be deduced that the results obtained for R600a at m_r =60g and L= 1.2m and 1.5m comply with the ISO standards (operating temperature = -12^{0}C and pull-down time = 150minutes, as reported by Fatouh and El Kafafy, 2006).

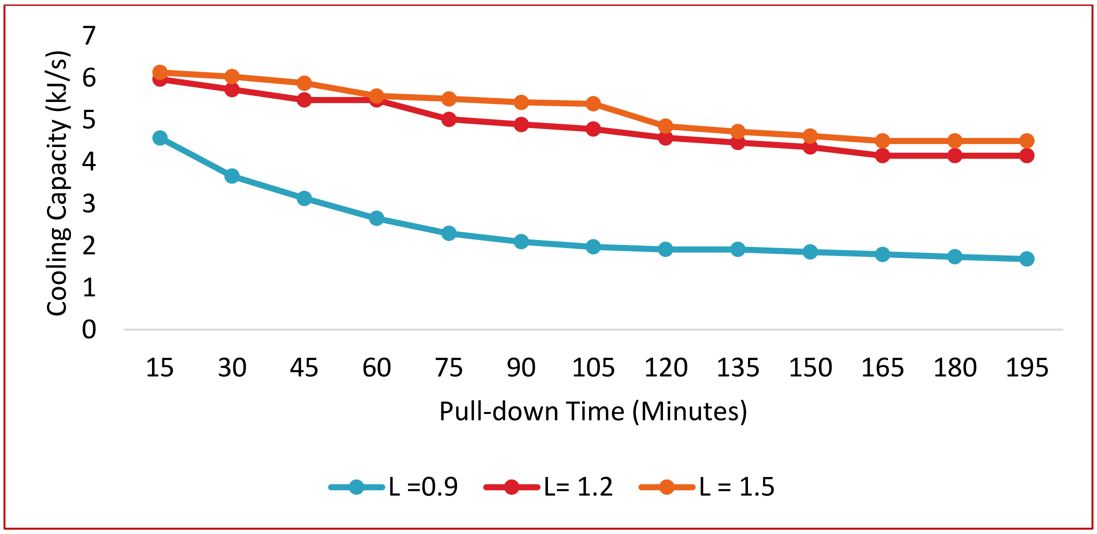

Figure 13: Cooling Capacity for 40g charge of R-12 at L= 0.9m, 1.2m and 1.5m.

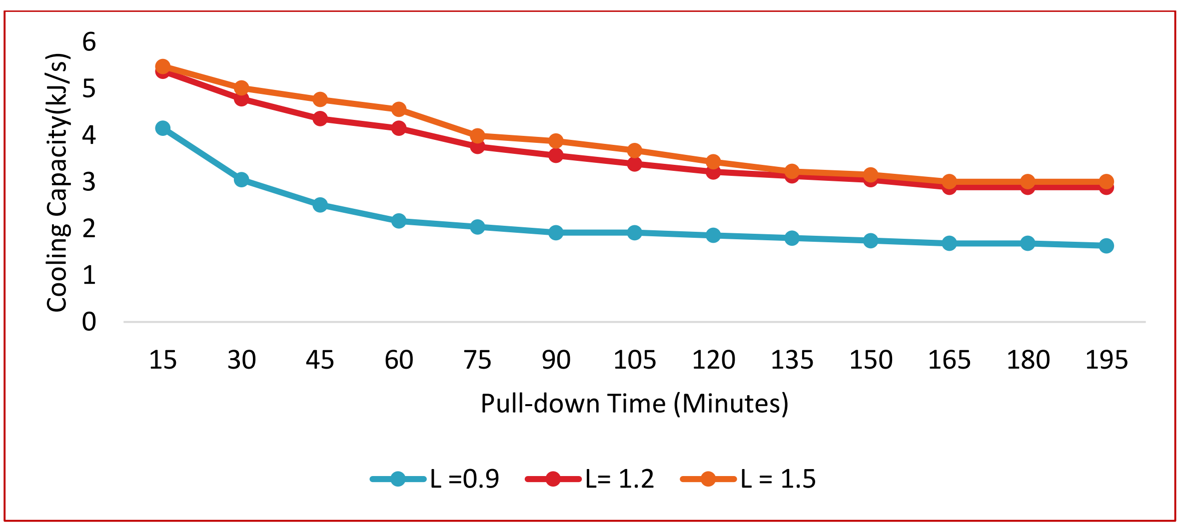

Figure 14: Cooling Capacity for 60g charge of R12 at L = 0.9m, 1.2m and 1.5m.

Table 2: Pull-down time for R12 and R600a with m_r= 40g

Pull-down Time (Minutes)	Evaporator Temperature (°C)					
	R12			R600a		
	L=0.9m	*L*=1.2m	*L*=1.5m	*L*=0.9m	*L*=1.2m	*L*=1.5m
0	32	32	32	32	32	32
15	24	30	28	22	24	18
30	15	25	18	12	18	15
45	8	15	9	10	9	8
60	4	12	6	8	9	8
75	-2	10	4	4	7	2
90	-3	6	-3	2	5	1
105	-5	6	-7	2	3	-3
120	-6	4	-10	-5	-3	-6
135	-10	2	-10	-7	-8	-8
150	-10	-4	-11	-8	-10	-12
165	-12	-6	-12	-8	-10	-18
180	-14	-6	-15	-8	-11	-15
195	-16	-6	-15	-8	-11	-14

Table 3: Pull-down time for R12 and R600a with m_r= 60g

Pull-down Time (Minutes)	Evaporator Temperature (°C)					
	R12			R600a		
	L=0.9m	*L*=1.2m	*L*=1.5m	*L*=0.9m	*L*=1.2m	*L*=1.5m
0	32	32	32	32	32	32
15	25	22	28	19	15	13
30	15	12	15	10	8	7
45	9	10	8	6	1	3
60	8	7	6	4	-2	1
75	5	6	3	3	-4	-2
90	5	6	1	3	-6	-6
105	5	6	-3	2	-7	-9
120	-3	3	-8	-4	-8	-11
135	-8	-4	-11	-6	-12	-12
150	-10	-7	-11	-8	-12	-12
165	-11	-7	-12	-10	-13	-13
180	-12	-9	-12	-10	-13	-13
195	-17	-9	-12	-10	-14	-14

In order to accept a refrigerant as a drop-in replacement, similar or better cooling capacity and power consumption should be achieved (Fatouh and El Kafafy, 2006; Björk and Palm, 2006. Based on the results of this study, R600a with m_r =60g and *L*= 1.2m and 1.5m achieves pull down time lower than that of R12 with m_r of either 40g or 60g. Furthermore, for this choice, the COP is higher (4.76) and power consumption is lower (0.40 kW) compared to R12 charge with m_r either 40g or 60g. However, R600a with L= 0.9m and m_r =60g gives higher cooling capacity (6.5779 kJ/s) compared to that of R12 (5.9739 kJ/s); the cooling capacity of R600a is about 9.18% higher than that of R12. The appropriate combination of L and m_r for R600a to be used as a drop-in refrigerant for R12 was determined by considering the the lowest possible power consumption per day with better cooling capacity while other required performances are fulfilled (Björk and Palm, 2006). Based on the power consumption per day, pull-down time and COP, the appropriate combination of L and m_r for R600a is 1.5m and 60g while from cooling capacity perspective, it is 0.9m and 60g.

4. Conclusion

In this study, the performance of R600a which is environmentally friendly refrigerant with zero ozone depletion potential (ODP) and low global warming potential (GWP) was studied experimentally in a domestic refrigerator and compared with the performance of the R12 refrigerant in the same system using different capillary tube lengths and refrigerant charges. After carefully investigated experimentally the performances of R600a and R12 in a domestic refrigerator, the following conclusions can be drawn:

- The pull-down time set by ISO for small refrigerator was achieved earlier using refrigerant R600a than using R12.
- The average COP obtained using R600a is about 6.3% higher than that of R12.
- R600a offers lower power consumption. The compressor consumed 24% less power when R600a was used than when R12 was used in the system.
- The cooling capacity of R600a is 9.18% higher than that of R12 in the system.
- The appropriate combination of capillary tube length and R600a charge to be used as a drop-in refrigerant for R12 is $L = 1.5$m and $m_r = 60$g when considering power consumption per day, pull-down time and COP; however, from cooling capacity perspective, it is 0.9 m and 60g.

Generally, the system performed better with R600a than with R12. This shows that the performance of R600a can be improved in domestic refrigerator originally designed to work with R12 by modifying the capillary length. In conclusion R600a is an appropriate long-term candidate to replace R-12 in the existing domestic refrigerator in terms of power consumption and COP.

References

Abed, AR, Fadhiel, HJ, Mahsun, G and Yassen, TC. (2014). Experimental Study on the Effect of Capillary Tube Geometry on the Performance of Vapour Compression Refrigeration System, Diyala Journal of Engineering Sciences 07 (2):47 – 60.

Akash, BA and Said, SA. (2003). Assessment of LPG as a possible alternative to R-12 in domestic refrigerator, Energy Conversion Management 44:381–388.

Akintude, MA. (2013). Experimental Study of R134a, R406A and R600a Blends as Alternative to Freon 12, IOSR Journal of Mechanical and Civil Engineering 7(1): 40-46.

Alsaad, MA and Hammad, MA. (1998). The application of propane/butane mixture for domestic refrigerators, Applied Thermal Engineering 18:911 – 918.

Boeng, J and Melo, C. (2012). A Capillary Tube - Refrigerant Charge Design Methodology for Household Refrigerators – Part II: Equivalent Diameter and Test Procedure, International Refrigeration and Air Conditioning Conference, Purdue, July 16-19, 2012. Paper 1180. http://docs.lib.purdue.edu/iracc/1180

Bolaji, B.O (2012), Experimental study of R152a and R32 to replace R134a in a domestic refrigerator, Energy 35: 3793 – 3798.

Dincer, I and Kanoglu, M. (2010). Refrigeration Systems and Applications (2nd Ed.), John Wiley & Sons Ltd, United Kingdom.

Björk E and Palm, B. (2006). Performance of a domestic refrigerator under influence of varied expansion device capacity, refrigerant charge and ambient temperature, International Journal of Refrigeration 29(5):789–798.

Fatouh, M and El Kafafy, M. (2006). Experimental evaluation of a domestic refrigerator working with LPG, Applied Thermal Engineering 26 (14-15): 1593–1603.

Gonçalves, JM and Melo, C. (2004). Experimental and numerical steady-state analysis of a top-mount refrigerator. 10th International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, USA.

Guobing Z, Yufeng Z. (2006). Experimental investigation on hysteresis effect of refrigerant flowing through a coiled adiabatic capillary tube, Energy Conversion and Management; 47: 3084-3093.

Halimic, E, Ross, D, Agnew, B, Anderson, A and Potts, I. (2003). A comparison of the operating performance of alternative refrigerants, Applied Thermal Engineering; 23(12):1441-1451.

Joudi, KA and Al-Amir, QR. (2014). A Numerical Simulation of the Effect of Ambient Temperature on Capillary Tube Performance in Domestic Split Air Conditioners with R22 Alternatives', Journal of Information Engineering and Applications 4 (3): 21 – 31.

Jung, DS and Radermacher, R. (1991). Performance simulation of a two evaporator refrigerator-freezer charged with pure and mixed refrigerants', International Journal of Refrigeration 14 (5):254-263.

Khan, MK, Kumar, R and Sahoo PK. (2008). Flow characteristics of refrigerants flowing inside an adiabatic spiral capillary tube, HVAC&R Research ASHRAE 13, pp. 731–748.

Kim, SG, Ro ST and Kim MS. (2002). Experimental investigation of the performance of R-22, R-407C and R-410A in several capillary tubes for air-conditioners, International Journal of Refrigeration 25(5):521-531.

Kuehl, SJ and Goldschmidt, VW. (1990). Steady flows of R-22 through capillary tubes: test data, ASHRAE Trans 96 (1):719–728.

Lee VS and Su CC. (2002). Experimental studies of isobutene (R600a) as refrigerant domestic refrigeration system, Applied Thermal Engineering; 22(5):507 – 19.

Matani, AG and Agrawal, MK. (2013). Effect of capillary diameter on the power consumption of VCRS using different refrigerants, International Journal of Application or Innovation in Engineering & Management 2(3):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 Sustainable Development 10 (4):29 – 33.

Padilla, M, Revellin, R and Bonjour, J. (2010). Exergy analysis of R413A as replacement of R12 in a domestic refrigeration system, Energy Conversion and Management 51(11):2195-2201.

Park, C, Lee, S, Kang, H and Kim, Y. (2007). Experimentation and modelling of refrigerant flow through coiled capillary tubes, International Journal of Refrigeration 30 (7):1168-1175.

Pathak, SS, Shukla, P, Chauhan, S and Srivastava, AK. (2014). An experimental study of the Effect of capillary tube diameter and configuration on the performance of a simple vapour compression refrigeration system, IOSR Journal of Mechanical and Civil Engineering 11(3):101-113.

Rasti M, Hatamipour MS, Aghamiri SF and Tavakoli M. (2012). Enhancement of domestic refrigerator's energy efficiency index using a hydrocarbon mixture refrigerant, Measurement 45(7):1807–1813.

Salim, TK. (2012). The Effect of the Capillary Tube Coil Number on the Refrigeration System Performance", Tikrit Journal of Engineering Sciences 19 (2):18-29.

Sekhar, SJ, Mohanlal, D and Renganarayanan, S. (2004). Improved energy efficiency for CFC domestic refrigerators retrofitted with ozone friendly HFC134a/HC refrigerant mixture, International Journal of Thermal Sciences 43(3): 307-314.

UNEP, 2012. Study of the potential for hydrocarbon replacement in existing domestic small commercial refrigeration appliances. United Nations Environment Programme (UNEP), France http://www.unepie.org Accessed on May 10, 2012.

Vjacheslav, N, Rozhentsev, A and Wang, C. (2001). Rationally based model for evaluating the optimal refrigerant mass charge in refrigerating machines. Energy Conversion and Management 42 (18):2083-2095.

Wei, CZ, Lin, YT, Wang, CC. (2000). A performance comparison between coiled and straight capillary tubes, Heat Transfer Engineering 21(2):62–66.